Paper read at the Institute on Tuesday, 14 November, 1972

THE DEVELOPMENT OF A DIRECT DRIVE DIESEL ENGINE

Dipl.Ing. P. E. A. Borgeaud, C.Eng., M.I.Mar.E.*

The paper presents some detailed developments in the RND range of Sulzer slow speed Diesel engines and, in his introduction, the author points out the main differences between the RND engine and the RD engine which it has superseded.

He then goes on to discuss various features of the RND engine under the headings of combustion chamber design, cooling water treatment, cooling water system design and top end bearings. Future increase in power output from these engines is discussed from the point of view of increased b.m.e.p., whilst controls and instrumentation are considered by way of a new system, adapted particularly to present day requirements and currently in the final stages of development and testing.

The operation of RND engines on crude oil and dual-fuel engines comes next under consideration, and the author concludes with a section on machinery installation.

Mr. Borgeaud

INTRODUCTION

A few years ago the well known RD-series of slow speed engines, nearly 2000 units of which are in service today, was superseded by the new RND-range. It seems worth-while to point out the main differences of both designs(1) before discussing some more detailed developments within the RND-generation.

The main difference is to be found in the supercharging arrangement chosen: pulse system for RD and constant pressure system for the RND. A comparison of the two possibilities showed advantages for the latter at mean effective pressures above 7 to 8 bar (100–114 lb/in^2), as then the disadvantage of not using the pulse energy as generated in the blow-down process is compensated by the better efficiency of the turbo-charger arrangement. Only on part load does it become necessary to adopt electrically driven auxiliary blowers to get enough air for proper combustion in order to compensate for the loss of pulse energy in the exhaust, which at low loads contains an important part of the total exhaust energy.

With the introduction of the constant pressure supercharging, the rotary exhaust valve between cylinder and turbine as necessary for the RD-type (Fig. 1) could be eliminated for the RND-engines, leading to increased reliability as a number of moving parts could be omitted. The piston skirt had to be made longer so as to cover the ports at top dead centre which also precludes the danger of sparks and dirt particles being blown back and thus reduces the fouling in the under-piston space. Whilst on RD-engines fires in the under-piston space could occasionally be experienced, we do not know of any such incidence on RND-engines.

Based on service experience, other refinements and improvements were introduced, such as a simplified stuffing box for the telescopic pipes with better accessibility to it and a new type of lubricating stud for the liner.

COMBUSTION CHAMBER DESIGN

One of the major problems in the layout of the RND-engines was the design of the parts around the combustion chamber (Fig. 2), where one has mainly to compromise between two requirements: low mechanical and low thermal stressing. A compromise because in the first case one would like to keep the wall as thick as possible, whereas the second requirement is for the thinnest possible walls.(2) In the first part of this paper further developments around the combustion space within the RND-generation will be discussed.

The pistons of the Sulzer slow speed engines are based—since the introduction of water-cooling on the RD-engines—on a design with ribs supporting the area loaded by the gas pressure. Whilst this design ensures an even support of the gas pressure and stiffens the outer part of the piston so that minimum thermal and mechanical deformations are encountered at the piston ring groove, steel casting with incorporated ribs can lead to difficulties during casting and cooling-down of the pieces. On some RD90 cylinder covers we have experienced shrinking defects at the top end of the ribs in the transition to the piston crown top plate. These casting defects were more often in the interior of the piece and were not detectable by magnaflux or X-rays. Only with a refined ultrasonic method could they be located. The defects have proved to be liable to cause difficulties in service and can be the origin of fatigue cracking. Efforts were therefore concentrated on ways and means of producing sounder castings. At the same time extensive use of experimental methods with strain-gauges and computations by the finite-element method resulted in a geometry with very low dynamic stressing in service with a much improved transition of the ribs to the top plate of the piston regarding the change of stiffness at this critical point. This also means reduced localized shrinkage at the mentioned transition, which in turn results in a better cast steel quality.

The stresses are compared in Fig. 3 showing the dynamic stressing in service before and after redesigning the piston head, which is now the same for the RD and RND-engine.

As far back as 1944 the idea was brought forward at Sulzer's for bore-cooling on liners and cylinder covers. Whilst this layout could also be an advantage for the piston-crown, it cannot be realized so easily with flat piston crowns except by cast-in tubing. This effective bore-cooling concept has in the past been used on smaller engines of Sulzer design and was introduced for the slow speed engine on the liner of the RND-series. By this means

* Technical Director, Diesel Engine Department, Sulzer Brothers Ltd., Winterthur, Switzerland.

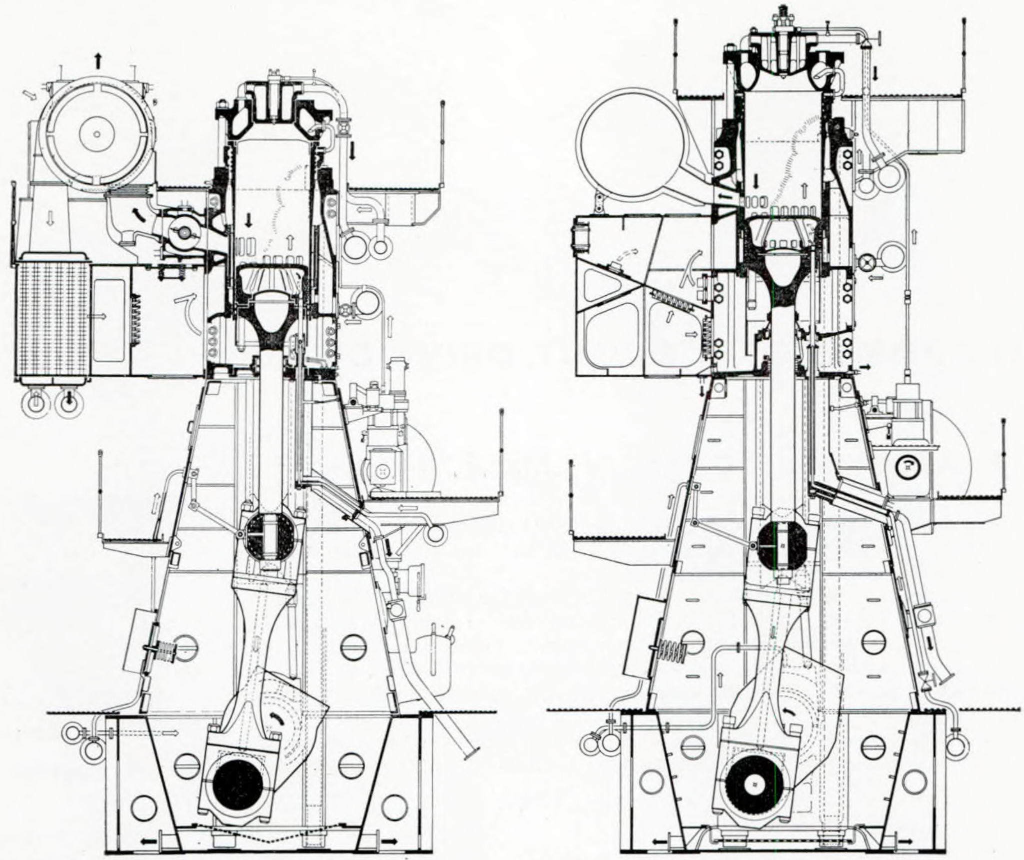


FIG. 1—*Cross-sections of RD90 engine (left) and RND90 engine (right)*

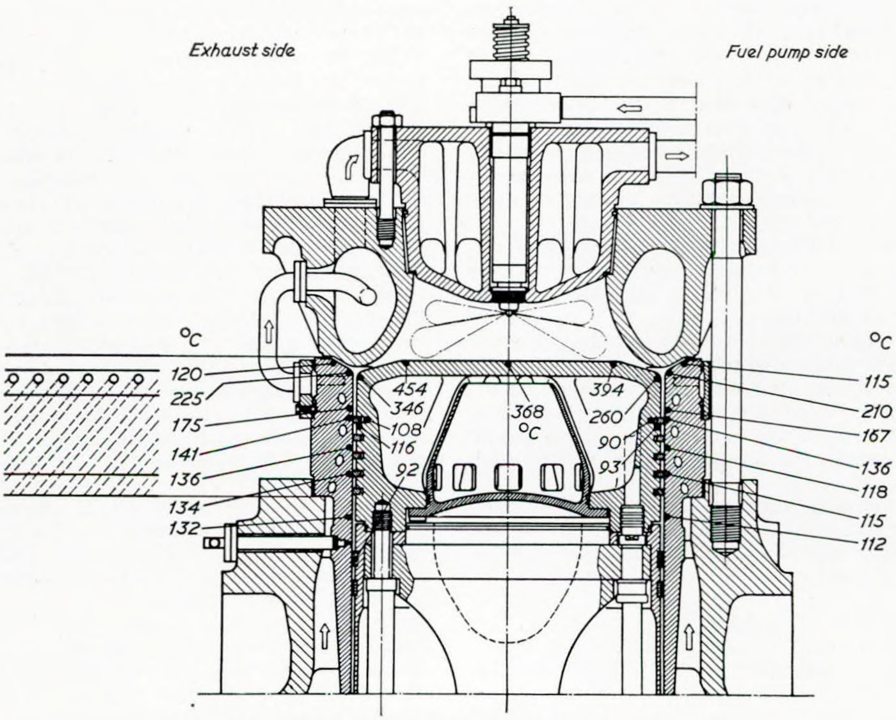


FIG. 2—*Combustion chamber for RND-type engine*

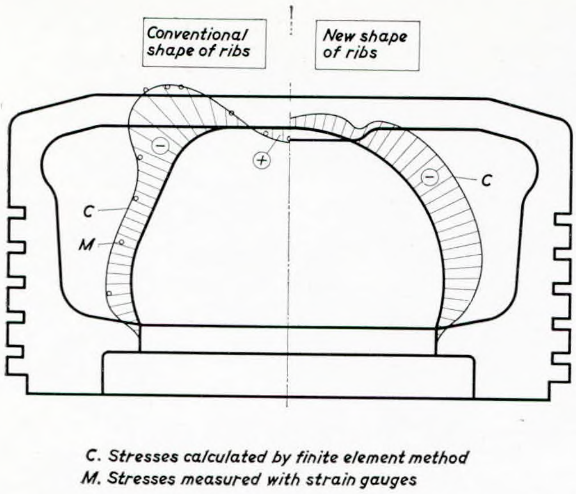


FIG. 3—*RND90, comparison of gas load stresses in different piston crowns*

the cooling medium comes closer to the running surface for instance within some 20 mm for the RND105 liner. The temperatures could thereby be substantially lowered compared with the former head-ring supported design. The collar of a bore-cooled liner can be kept as thick as necessary to withstand the dynamic gas load, and this without any practical influence on the temperature distribution of the inner liner surface.

For the RND-engines, the idea of splitting the cylinder head in two individual concentric parts has been retained up to now. The stressing has been reduced step by step to the lowest possible level inherent in this principle. As an example Fig. 4 illustrates the improvements on the cylinder cover outer part of the RND90 engine since its introduction.

Three points which are felt to be indicative of the behaviour of this component are:

a) the area which is known to be the starting point for possible troubles on RD90 covers, if in service several negative factors add up and counterbalance the safety margin against fatigue failure, one important factor being corrosion on the cooling water side, a problem we will revert to later;
b) the area in which some cracks were experienced for the

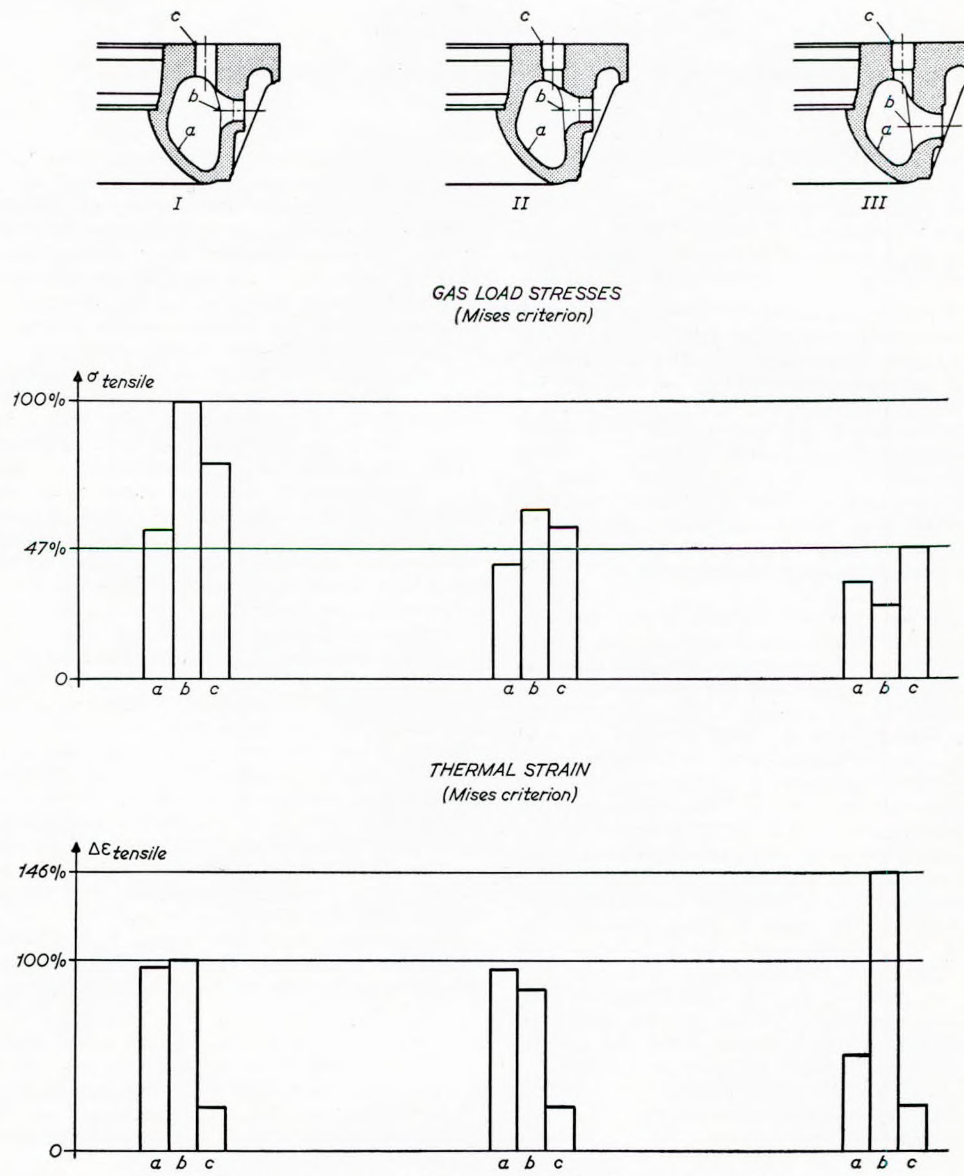


FIG. 4 —*Decrease of stresses in cylinder covers RND90*

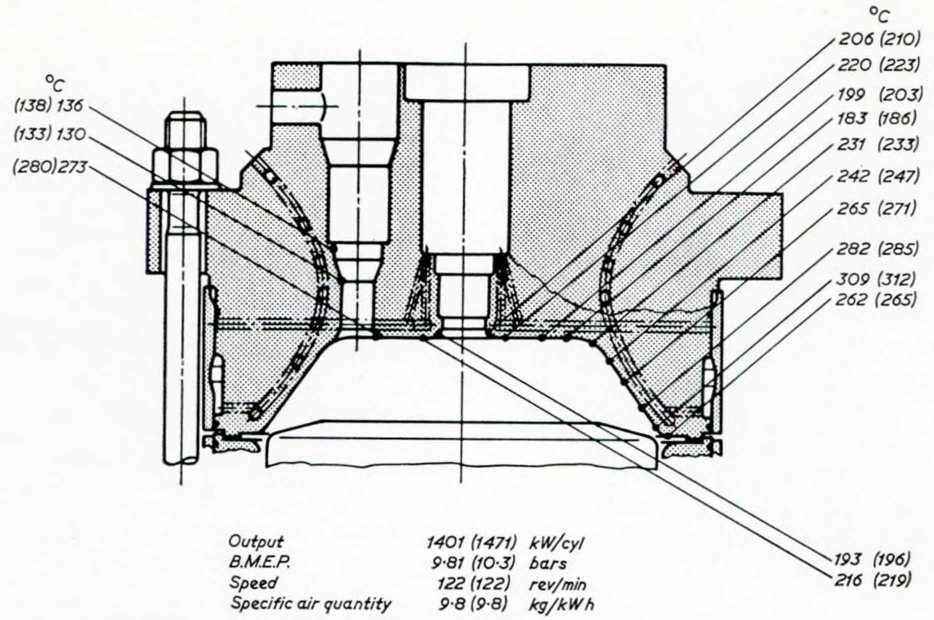


FIG. 5—*Temperatures on the combustion side of the bore-cooled cylinder cover RN*76

alternative I in Fig. 4 since the introduction of the RND90;

c) the area where equivalent service stresses originated by the gas load are experienced for RD and RND engines, the latter being again equipped with covers according to the alternative I in Fig. 4.

Alternative I corresponds to the original version, based on a comparison with RD-experience. After its introduction it was found that the safety margin did not allow for adverse condition in the area (b), and subsequent redesigning resulted in the improvement shown for alternative II in Fig. 4.

Introducing the finite-element method, a wide variation of geometry and its influence on stressing was investigated. This led to the third and final alternative with minimalized stressing. Irrespective of increased output, 26 per cent compared with RD90, the present RND cover outer part shows, in spite of higher maximum firing pressure (+ 13 per cent), lower dynamic stressing and reduced thermal stressing if compared with RD90 cover.

It should be mentioned that stresses and strains given in Fig. 4 were computed by the finite-element method after having cross-checked its validity by strain-gauge measurement on actual components. In these computations the effect of the holes at (b) and (c) was neglected, so that the values had to be corrected by a form-factor which is assumed to be 2 and 3 respectively, and which factors finally led to the stresses and strains indicated in the graph. In assessing the influence of the thermal strain at point (b) it must be taken into account that they are compressive. As mentioned above, the current design is the optimum which can be reached in compromising between mechanical and thermal stressing. The increased safety margin gained could only be kept for further uprating if better materials were considered, a line of thinking which quickly leads to uneconomical solutions. One alternative, is a bore-cooled cover as shown in Fig. 5, where the two parts, i.e. cylinder cover outer part and cylinder cover insert are replaced by one solid forged or cast piece with bore-cooling, which still allows for the fitting of standard parts like injectors, starting valves, safety valves, etc. The results of the temperature measurements indicated were recorded on the old 1-cylinder test engine 1RSA76, which does not allow for running at higher mean effective pressures than 9·8 bar (142 lb/in^2). Therefore, the values valid for today's nominal output of a RND76 of 1471 kW/cyl at 10·3 bar (149 lb/in^2) m.e.p. and 122 rev/min were extrapolated based on readings which were, of course, recorded during different load-levels. These data are given in brackets.

The passages for the cooling of the conical area of the combustion chamber are drilled as for the liner at a certain angle to the vertical centre-line. Their entirety forms a hyperboloid. For the top part of the combustion chamber horizontal bores arranged radially are provided for. Generally speaking and for comparable b.m.e.p. the temperature level is substantially lower than compared with the existing design. A positive surprise are the temperatures recorded around the holes necessary for the safety and starting valves. As these passages disturb the equidistant array of cooling bores, one was anxious to assess the magnitude of possible hot spots. The measurements showed that in this respect the values recorded are very satisfactory.

Simultaneously extensive strain-gauge measurements were done to assess mechanical and thermal stressing. The values measured are related to the RD90 cover, which also stands for a RD76 cover, and where today the author's company has the best statistical recording of cover behaviour. If one compares these two covers, one generally will find that the mechanical stresses of the bore-cooled cover are very low compared to the RD-design and, of course, also in respect of the RND-cover (Fig. 6).

On the side of thermal straining, the improvements are not so considerable. The highest stressing can be found inside the horizontal bores at the bottom of the cover, where at point "d" the maximum dynamic amplitude was measured. The high thermal strains are fortunately in compression. If one combines the effect of mechanical tensile and thermal compressive stress, considering the values recorded, and compares it with the experience of higher mechanical tensile stressing (+ 40 per cent) together with tensile thermal stressing at point "a" of a RD90 cover, then it becomes apparent that the bore-cooled cover means a big step in increasing the strength of this part. Solutions omitting these horizontal bores at the bottom of the cover altogether would still further increase the margin. Therefore, alternative solutions are being investigated regarding this point.

COOLING WATER TREATMENT

All investigations on the behaviour of cylinder covers in service clearly indicated the importance of correct cooling water treatment and its influence on corrosion and on fatigue strength. To define it more clearly, a number of investigations were conducted.

Inhibitors on the basis of nitrites are very effective, but have

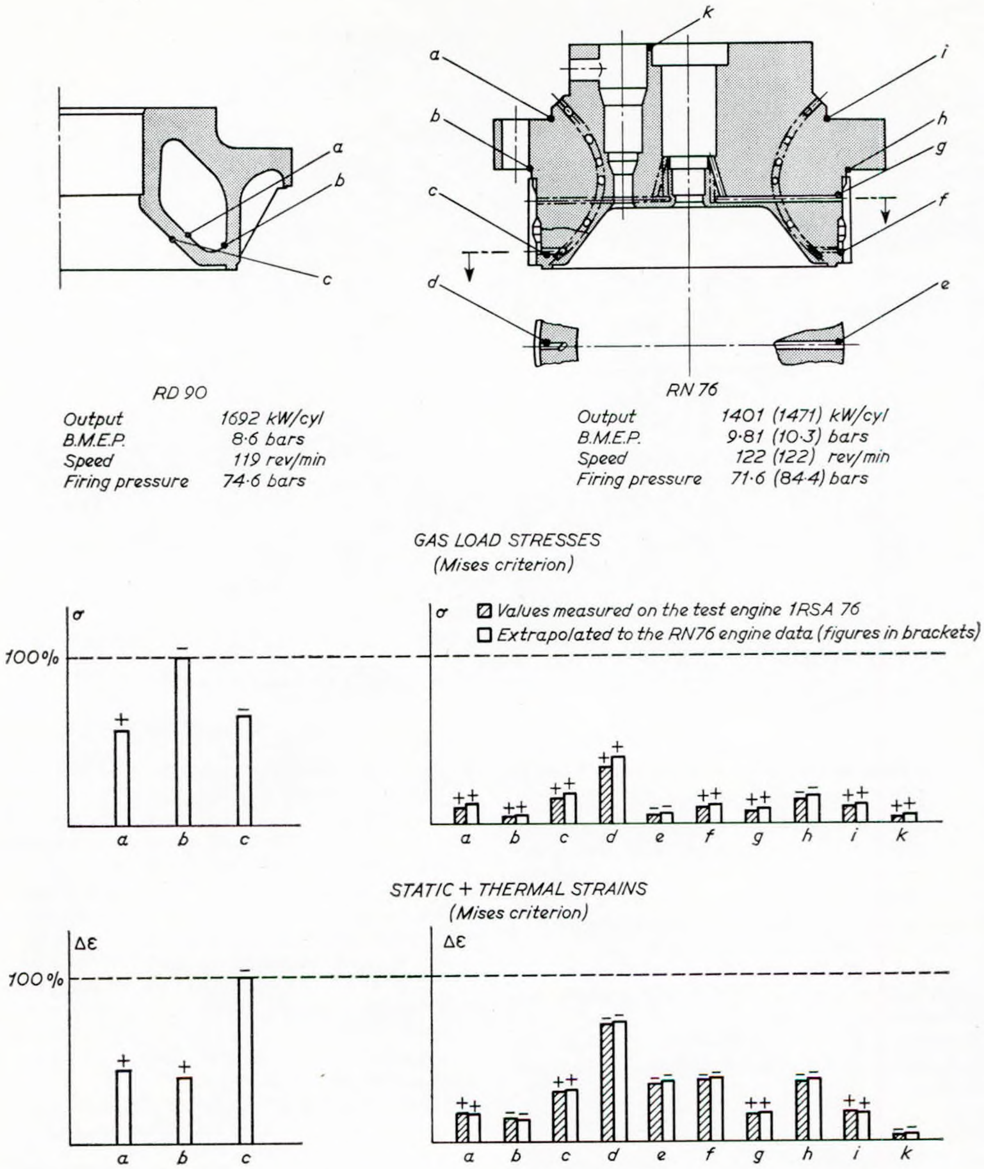


FIG. 6—*Comparison of stresses between the RD90 cylinder cover and the bore-cooled prototype cylinder cover RN76*

the disadvantage of being poisonous and of attacking zinc. Therefore, in a first investigation one tried to find the "ideal" inhibitor, which as such should fulfil mainly the following requirements: to be non-poisonous; not to attack zinc; not to lose its anticorrosive capacity in the presence of chlorides; not to be sensitive to possible cavitation and to be temperature-resistant.

Out of some 80 possible inhibitors tested in our laboratories in a first general search, 15 were retained for more detailed investigations. Finally one organic substance remained which—being nearly "ideal"—still gave good protection with a chloride concentration of 100 ppm. Only the problems regarding cavitation are not yet fully investigated. It was first tested on a RND-engine running on the testbed. After eight hundred hours (200 running hours and 600 hours standing still), practically no corrosion attack could be found. Extending the tests on board ship has up to now given every satisfaction, and we hope that this inhibitor will be marketed by a well known firm in this field in the near future.

In a further step, the influence of corrosion on fatigue strength was investigated on the three materials mainly used in engine design:

a) steel casting GS40 with a rupture strength of minimum 40 kg/mm^2 (25 t/in^2) and in a yield limit of 23 kg/mm^2 (15 t/in^2) or more;
b) nodular cast iron GGG42, having a rupture strength of at least 42 kg/mm^2 (27 t/in^2) with a yield limit of 30 kg/mm^2 (19 t/in^2) or above;
c) grey cast iron GG25 with a rupture strength of 25 kg/mm^2 (16 t/in^2).

The curves shown in Fig. 7 are self-explanatory. The values plotted over the service hours are more or less valid for slow speed engines. As the behaviour was studied on specimens rotating at 1500 rev/min under bending and the results afterwards transcribed to the engine running speed, the qualitative assertion is more important than the quantitative one. As for corrosion, time is a very important factor, the values given are in fact still on the high side. Viewed from the factor time, the decrease of strength would be even more pronounced.

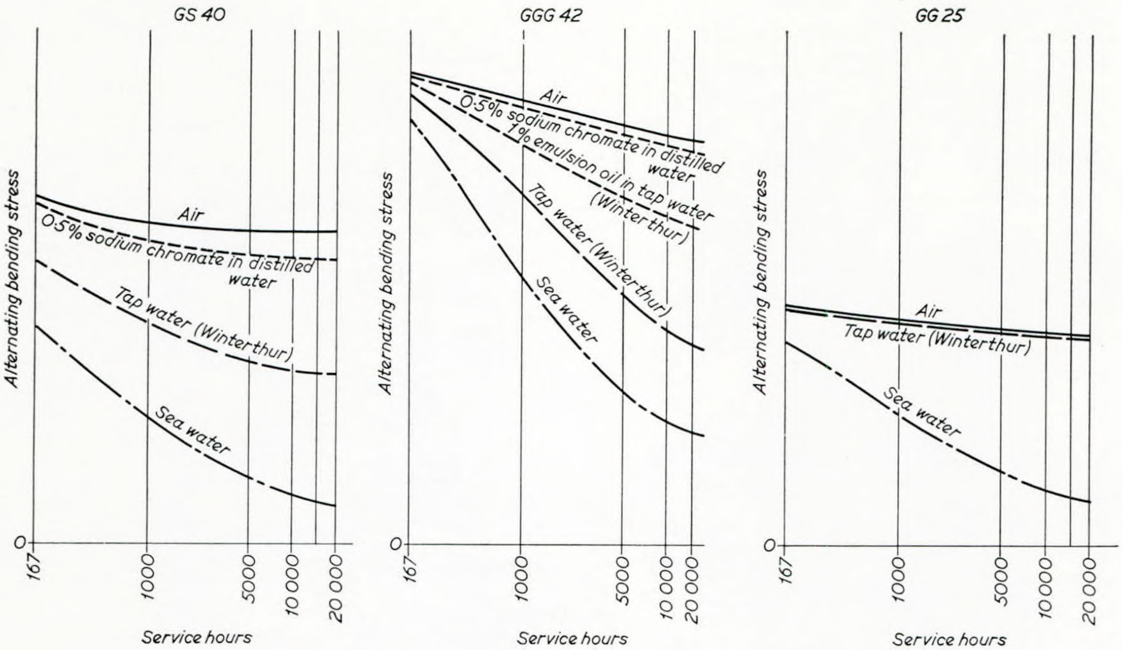


FIG. 7—*Corrosion fatigue of cast steel, nodular iron and grey iron*

Nevertheless, out of these investigations three facts have to be kept in mind:

a) the fatigue strength of steel and nodular iron must be reduced by a factor of approximately 2 if the water treatment is not correctly done in the beginning and not periodically checked. In the presence of sea-water the reduction is still more. Grey cast iron is less sensitive to inadequate water treatment;
b) as could be expected in the presence of corrosion, the fatigue strength relative to the uncorroded condition is increasingly reduced with time;
c) the visual inspection of the corroded surface does not allow a conclusion as to the possible extent of reduction in fatigue strength.

COOLING WATER SYSTEM DESIGN

Not only correct water treatment but also correct cooling water systems play an important part. Past experience with RD-engines has shown that serious failures like piston seizure, cracked cylinder liners or covers could in many cases be traced to deficiencies in the jacket cooling water system. Possible reasons are:

a) too low a cooling water pressure which favours steam formation, thus leading to local overheating and high thermal stressing. The minimum pressure on RND-engines must be kept between 2·9 bar (43 lb/in^2) to 3·9 bar (57 lb/in^2) at the gauge installed on the engine;
b) too low pressure or even vacuum on the suction side of the water pump due to excessive resistance in the balance pipe between header tank and the pump. The pump may not be working continuously anymore or still worse suck air through the stuffing boxes of the pump. It is therefore important that in service one checks the pressure on the suction side of the pump. It should be just somewhat lower than the equivalent static height of the header tank;
c) insufficient venting caused by unsuitably arranged or dimensioned vent pipes and throttlings. For instance by leading the cooling water from the vent pipes with free jets (open tunnel or other arrangements) into the header tank, air is sucked in which may reach the circuit over the balance pipe. An arrangement as shown, whereby the venting is introduced at the lower part of the tank, eliminates this danger;
d) likewise, in cases where automatic float vents are fitted, the chances of air entering the system are quite real.

To ensure good operational results of the engine, it is therefore important to keep not only the right pressures and temperatures on the cooling water system, but to provide the required vent pipes with the necessary throttlings to reduce the waterflow in the venting system as requested by the engine builder (Fig. 8).

Unfortunately, the cooling water system and its venting is not confined merely to the engine, but is also part of the engine room installation. The people responsible do not always realize the importance of the installation and that the engine is rather affected by air in the cooling system as well, of course, by dirt and inadequate water-treatment as discussed above.

TOP END BEARINGS

Bore-cooled liner, bore-cooled cylinder cover and improved piston crown as described above allow for higher safety margins with regard to the combined mechanical and thermal stressing as compared with the existing RD-range and this notwithstanding a certain increase of power in the future. Higher output to come will mean of course higher mechanical loading, and in this respect one part which also needs further investigation are the bearings in the engine. If we extrapolate the bearing behaviour of Sulzer slow-speed engines for future higher outputs, one can see, based on statistical service experience, that no troubles are to be expected except for the crosshead bearing. The cracking of the white metal lining of the crosshead bearing is to some extent common to all slow speed engines. Hair cracks can become apparent after a short or long service period. In odd cases they develop further till loose white metal pieces form a mosaic-like pattern. Although to our knowledge this has never resulted in a break-down of a RD-engine, it is an unsatisfactory situation for the operator and the engine builder.

The cracking could very simply be eliminated by the use of bronze bearings and hardened pins. But this combination is on the one hand more exacting regarding precision of machining, on the other hand more sensitive to dirt in the oil-system. More important still is the greater risk of the bearing running hot and the possible danger of crankcase explosion. This is the reason why we are very reluctant to do without the use of white metal or

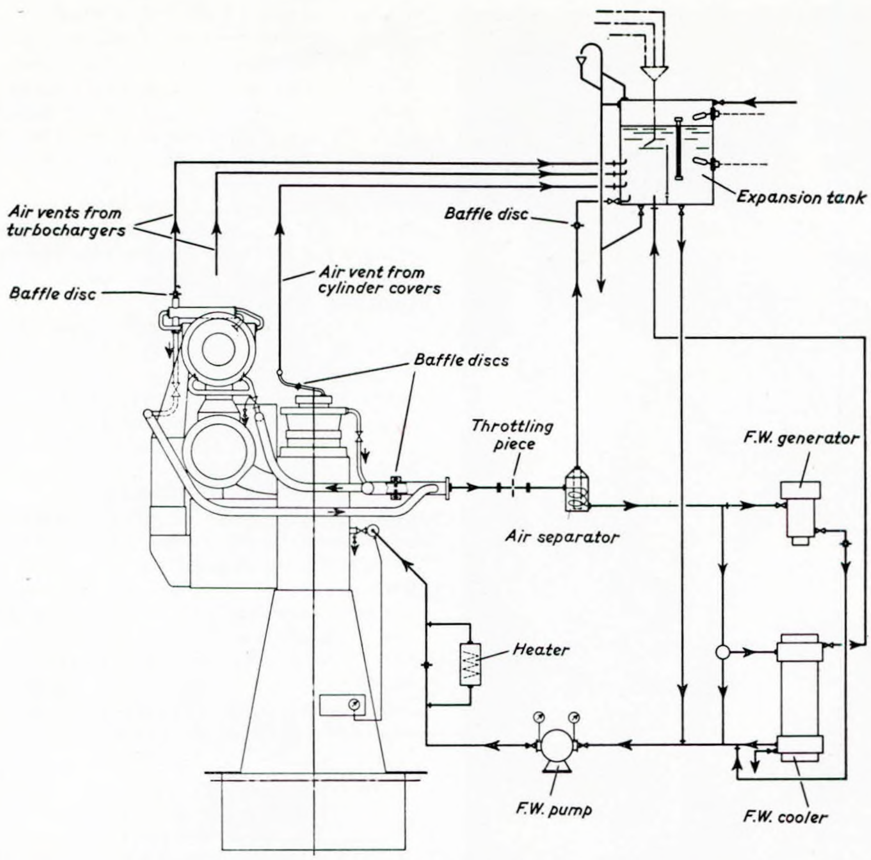


FIG. 8—*RND-engines, air vents of jacket fresh water system*

an equivalent material. To be able to keep the white metal in the RND-design, the bearing pressure has been kept as low as possible by using the biggest possible pin diameters, whereby the bearing body is designed, as for the RD, flexible enough to be self-adjusting to the pin deformations. A few years ago we introduced fine bored bearings which do not require scraping and therefore ensured a more even distribution of the bearing load.

A further step is being made now by the introduction of thin shell-type bearings As shown in Fig. 9, thin shell-bearings have been introduced for the lower half of the bearing, whilst the upper half has been kept as in the past. In the first step, thin white metal lined shells are being introduced, the lining thickness being 1·5–2 mm compared with some 4 mm for the standard bearing used till now. With the introduction of this thin-lined shell-bearing one aims at three goals:

a) An increased average fatigue strength of the white-metal lined bearing. If one compares an ideal thin shell bearing on the one side and an ideal cast white metal bearing, as standard up to now, the fatigue strength will only be slightly increased to the advantage of the former, but as the shell type bearing is manufactured by specialized firms under best conditions, we expect better consistency regarding quality and thus higher average fatigue strength as compared with the components used up to now in Sulzer slow speed engines. Varied service experience with the latter design could to a great extent be related in the past to variations in casting methods and bearing materials used.

b) Fig. 10 shows different types of shell-bearing structures which are considered for possible crosshead application in future, all fully interchangeable without any subsequent remachining in the engine provided the surface finish of the crosshead pin is satisfactory. In the sequence of increasing loading capacity, the following alternatives to the shell-type described above are being tested in engines under service conditions.

Shells with a white-metal thickness still further reduced to approximately 0·5 mm with a bronze-layer introduced between the backing and the running layer. Aluminium-plated shells with a 40 per cent content of tin in the aluminium. At room temperature the fatigue strength of this bearing material is only slightly above that for white metal, but it loses practically no strength with increasing temperature. As a result its load-carrying capacity will be some 25 per cent higher at service temperatures of 60°C (140°F) to 80°C (176°F). According to N. P. E. Desvaux(3) tests performed 40 per cent tin/aluminium alloy showed that it approaches white metal closely in its resistance to seizure, the hardness of both metals being comparable. Experiences with Sulzer engines indicate that for running-in the application of a lead-tin overlay may be beneficial.

The last alternative investigated is of the bronze-bearing type with a thin lead-tin overlay. Additionally an array of equidistant grooves arranged crossways in the bronze-layer, which are also filled with lead-tin, are provided, the idea being that whilst the soft bearing material in the grooves would give, if necessary, the anti-seizure properties, the bronze "isles" in-between, having an overlay of only a few hundredth of a millimeter thickness, will provide for highest possible loading capacity.

c) The introduction of the shell-type bearing in the lower bearing half allows for a quick and easy exchange when necessary. Experience proved that the bearing shell can be renewed within 30 to 60 min. Shims have been eliminated altogether, so that the bearing shells, which are produced to close tolerances, are fully interchangeable. The bolts of the bearing cap are hydraulically

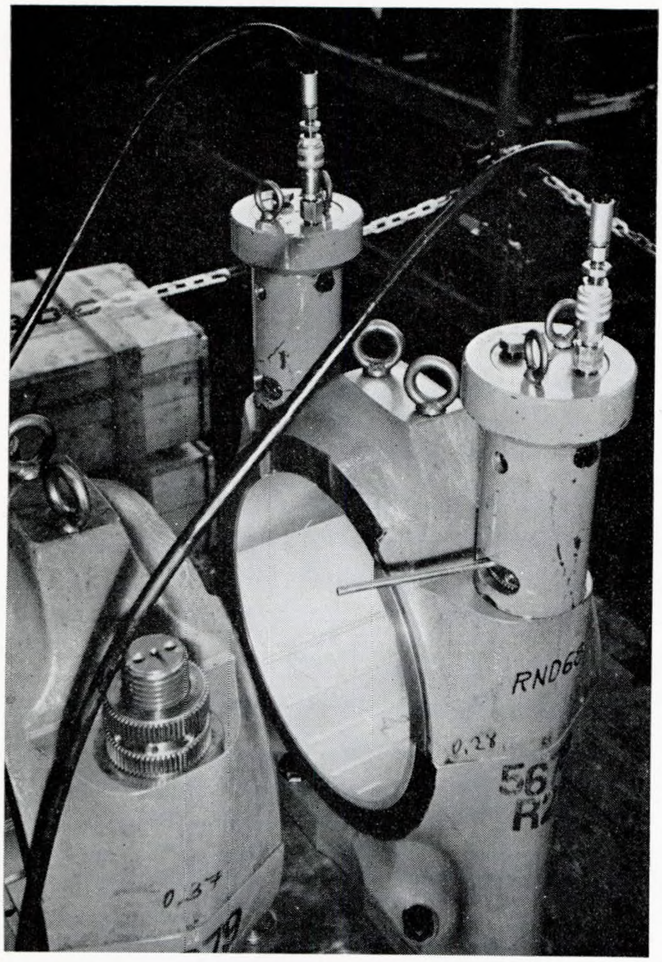


FIG. 9—*Crosshead with thin shell-bearing and hydraulic pretensioning tool*

	Bearing type	Bearing materials Lining materials		Overlay
1	Bimetal WM; 1·5–2mm WM; Steel Ck10	Whitemetal G-SnSb7Cu3Cd1Ni (WM88) 'Hoyt 11R', 'HC2' 'Auto 90R'	——	——
2	Trimetal WM; 0·4–0·6mm WM; 0·5–1·5mm Bz; Steel Ck10	Whitemetal G-SnSb7Cu3Cd1Ni (WM88) 'Hoyt 11R', 'HC2' 'Auto 90R'	Lead-tin-bronze G-CuPb15Sn8Ni	——
3	Bimetal Al Sn; 0·01–0·03mm Overlay; 1–1·5mm Al Sn; Steel Ck10	Tin-aluminium AlSn40 'AS45', 'A40'	——	Pb90 Sn10 0·03mm
4	Trimetal galvanic; 0·05–0·08mm+0·10; 1·5–2·0mm Bz; Steel Ck10	Pb-whitemetal Pb87Sn10Cu3 'Glyco-40'	Lead-tin-bronze G-CuPb20Sn2	——

FIG. 10—*Types of bearing shells for crosshead bearing RD/RND-engines*

pretensioned with minimum dispersion using a tool as shown in Fig. 9, this being an important factor regarding the avoidance of fretting between shell and bearing body. The upper half is still whitemetal-lined as in the past.

INCREASING OUTPUT IN THE FUTURE

The design problems described around the combustion chamber and at the crosshead represent a great number of continuous improvements worked on within the RND-range, which are, wherever possible and necessary, also applied to the RD-family. Major changes with regard to the general outlay of RND-engines are not to be expected in the near future, although work is continuing on an increase in output.

For still greater engine power one could increase the cylinder diameter, the piston speed, the number of cylinders and/or the mean effective pressure.

An increase in engine-bore above today's 1050 mm is technically possible, but does not seem practical for economic reasons, i.e. too high development and investment costs compared to the number of engines one could reasonably expect to be required.

A substantial rise in piston speed is not to be expected either. Keeping the low rev/min required by the propeller efficiency would mean lengthening the stroke, which in turn means a wider bedplate and an increase in engine height and weight or, in other words, a heavier and more expensive engine, whereby the increase of piston speed would only partly compensate for the higher costs. Particularly for two-cycle engines with high mean effective pressure an increase in piston speed will not improve running behaviour. The piston speed usual nowadays for large engines may therefore be raised a little, but not decisively. One visualizes only with difficulty big-bore in-line engines with more than 12 cylinders, as from the shipbuilders' side shortest possible engine rooms are aimed at. Disregarding the possibility of a double-acting engine one is left with the option of a V-configuration or an arrangement of the cylinders in two parallel vertical banks driving a crankshaft placed below and in-between these two cylinder rows. An increase in cylinder numbers does not seem feasible in the near future using these approaches.

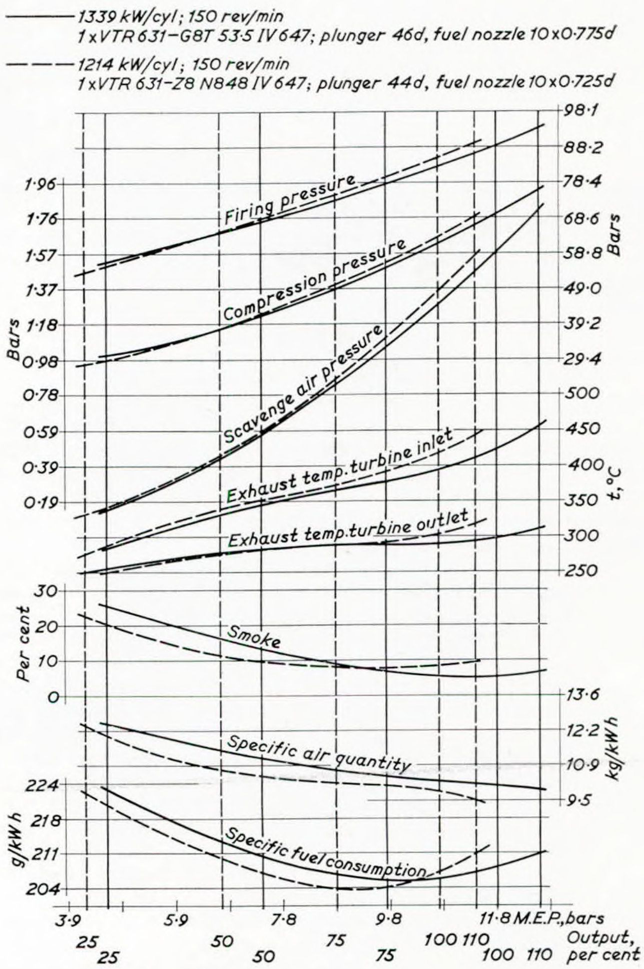


FIG. 11—*Comparison of test results, 6 RND 68*

One is therefore left with the increase in mean effective pressure, which means squeezing still more air into the cylinders. Already a very high purity of air charge is obtained, only 10 per cent of the scavenge air being lost through the exhaust ports. Scavenge efficiency can hardly be further improved, and therefore the only practical way is to increase the pressure level of the charge-air with higher efficiency of the turbo-charger due to better utilization of the exhaust energy or/and by more efficient compression of the air. Preliminary tests on a 6RND68 engine (Fig. 11) showed that with better efficiency of turbo-charger as available in the near future, the b.m.e.p. could be increased by 10 to 15 per cent without increasing the thermal load on the engine components. These results are based on an increase in combined efficiency of approximately 7 per cent.

Fig. 12 indicates a variation of the heat flows through piston, liner and cover as well as the heat dissipation through charge-air cooling and exhaust, illustrating the margin for increased output without higher heat flows through the components as measured with improved high pressure turbo-chargers. From the operating point of view it must be stressed that higher efficiency can only be of value if it is not penalized by increased susceptibility to fouling as the better efficiency ought to be maintained through thousands of hours of operation.

An increase in output of more than 15 per cent would mean higher thermal load on the component, which would have to be compensated by further design optimization to keep the lower thermal stress levels arrived at today for RND compared with RD. Should this no longer be possible, one would still have the possibility of applying 2-stage supercharging. Using two supercharger stages in series allows a reduction in the individual pressure ratio for each stage, resulting in better stage efficiency. Intercooling of the air after the first stage reduces the power requirement of the second compressor stage, which means that with the exhaust energy available one can compress more air to higher pressure levels.

Based on compressor efficiencies available in the near future for higher pressure ratios, no noticeable drop in the air-fuel ratio is to be expected for b.m.e.p. up to 12·7 bar (185 lb/in^2). With still higher loads this ratio drops rapidly so that viewed from today two-stage supercharging will have to be applied for b.m.e.p. above 13·7 bar (199 lb/in^2). With an air-delivery of approximately 10 kg/kW h (16·5 lbs/bhp h) the following characteristic data would result for b.m.e.p. of 14·7 bar (213 lb/in^2) or 17·6 bar (256 lb/in^2) respectively. Firing pressures of 107·8 bar (1565 lb/in^2) or 127·4 bar (1849 lb/in^2) with exhaust temperatures at turbine inlet of 450°C (842°F) or 480°C (896°F).

In investigating supercharging and its consequences regarding higher stressing of components, a number of factors were analysed on a RND-engine as to its effect of thermal loading and the following findings are worth considering.

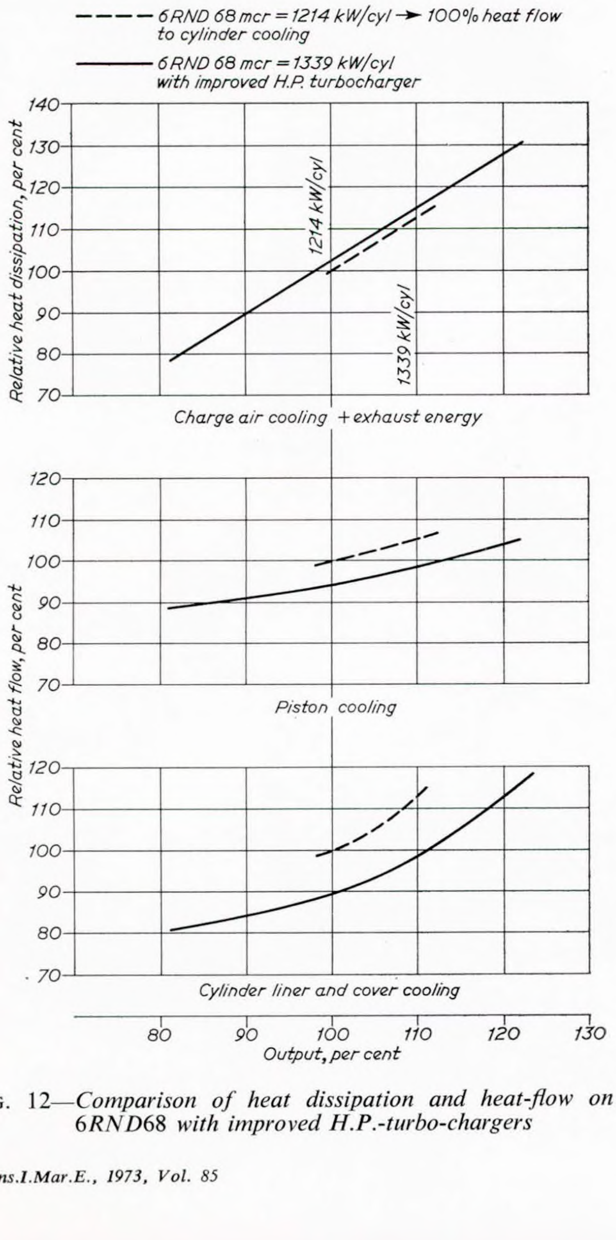


FIG. 12—*Comparison of heat dissipation and heat-flow on a 6RND68 with improved H.P.-turbo-chargers*

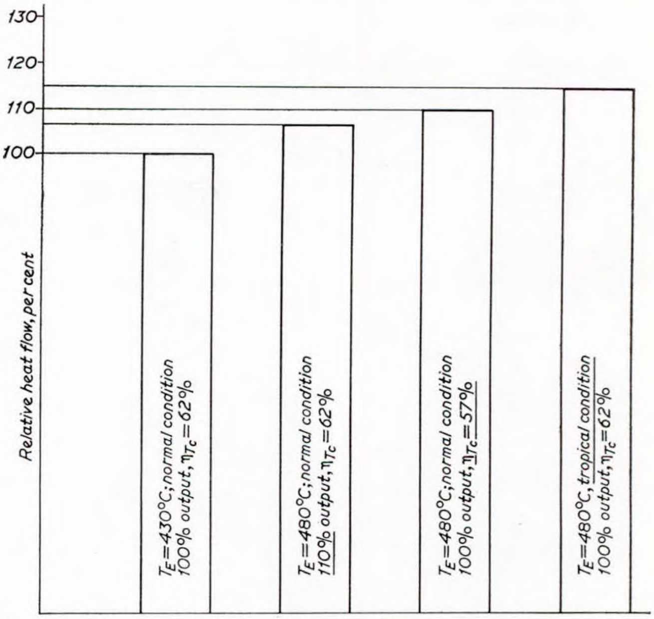


FIG. 13—*Comparison of relative heat-flow in the combustion space*

In practice the exhaust temperature is used to monitor the engine. Fig. 13 shows that a temperature increase in itself does not indicate a corresponding variation of thermal loading as the causes are of similar importance. Compared to the nominal output at normal conditions with an exhaust temperature of 430°C (806°F) before turbine, an increase by 50°C (90°F) to 480°C (896°F), due to either increase in b.m.e.p., reduction in turbo-charger efficiency or tropical conditions, has the following influence:

a) an increase of thermal load by some 6·5 per cent is to be considered when the higher exhaust temperature arrived at is due to an increase of b.m.e.p. by 10 per cent;
b) a reduction of combined turbo-charger efficiency from 62 per cent to 57 per cent would thermally stress the components around the combustion chamber by +10 per cent;
c) running under tropical conditions also gives the same increase of 50°C (90°F), but has even more pronounced effect, the thermal load being 15 per cent higher.

CONTROLS AND INSTRUMENTATION

When remote control was introduced in the beginning of the sixties, it was normal practice to concentrate all controls of the engine at one mechanical stand to enable a single man to manoeuvre the engine. For the Sulzer RD series it is generally situated at the bottom floor, portside, where most space is available and where the various control devices like the reversing valve or the speed-control governor are linked. For the first installations ordered with remote control, alterations had to be made to the manoeuvring stand to adapt the pneumatic servomotors required for the motorization of the various control levers. Additionally a new type of governor incorporating fuel limiting devices was provided.

For the RND-series a mechanical stand located at the same place as for the RD was redesigned in its standard execution in such a way that all pneumatic servomotors for remote control could easily be mounted.

The further development of ship automation meant that the good experience with today's pneumatic equipment for remote control leads to its continued use. As practically every engine supplied today is fitted with remote control this has brought a further step in development of Sulzer engines. Experience has in fact proved that centralized control of the engine is no longer required for the following reasons:

a) in the case of complete failure of the pneumatic control system from the control room, the present extended mechanical stand seems to be a too sophisticated solution for an emergency manoeuvring possibility;
b) experience over the last 10 years has proved the high reliability of the presently used pneumatic equipment;
c) emergency controls of the engine are practically never used in installation with remote control from a control room;
d) in some cases it was hoped that the central manoeuvring stand on the engine could be by-passed and a direct connexion between control room and the different control devices was aimed at.

Based on these considerations, it has been decided to look into a new control system for RND-engines particularly adapted to present day requirements and having the following features:

a) pneumatic controls to take advantage of the flexibility of such a system;
b) the engine to be provided with a standardized pneumatic manoeuvring stand, which can be placed in any desired position in the engine room;
c) this manoeuvring stand to be connected directly to the control devices on the engine;
d) the required pneumatic logic-elements to be of marine equipment available world-wide.

The main alterations of the new control system (Fig. 14B) compared with today's equipment (Fig. 14A) are:

a) the governor UG40 has been replaced by the PGA58 Woodward governor with increased speed range and pneumatic speed setting;
b) the fuel pump control linkage has been improved as it is lighter and balanced. Moreover it is now spring-loaded towards stop position which results in increased safety and reduced control forces in the linkages. This in turn makes a booster as used in the past superfluous.

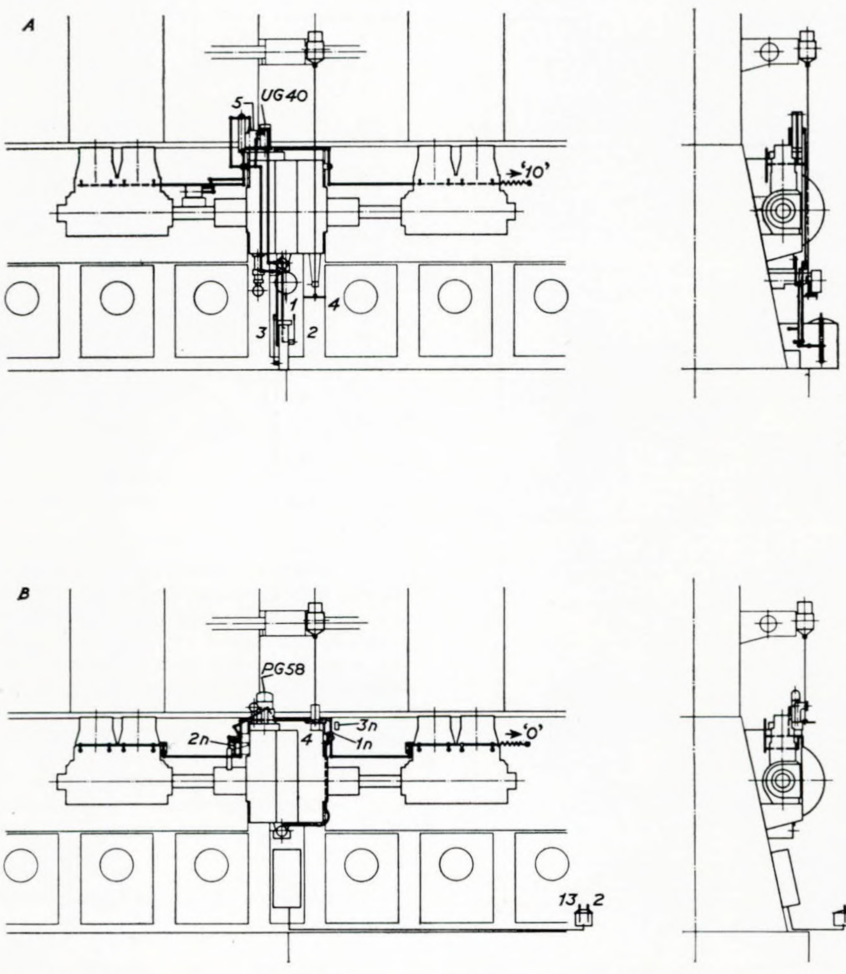


FIG. 14—*Control systems of RND-engines as in the past (A) and in future (B)*

The control of all functions regarding safety devices and interlocks as known on the Sulzer slow speed engine has been retained, but are now realized with pneumatic circuits. The air is delivered over a pressure-reducing station mounted on the engine. Should the air supply fail, then an electrical monitoring system which acts through the interlock devices directly on the fuel pumps automatically takes over the most important tasks of the pneumatic interlocks and safety devices.

The logic elements of this new pneumatic system are built in a box in such a way as to allow for either pneumatic or electrical remote control to be easily connected. It is located at the present position of the mechanical stand.

For emergency running the governor is equipped with manual speed-setting. Should the governor fail, then the possibility would still exist of setting the fuel delivery according to ten possible settings up to full manoeuvring speed. The emergency lever for starting and reversing is situated near the governor and tachometer, and would allow one man to run the engine even under abnormal conditions.

The new control system as described is at the moment of writing in the final stage of development and testing. It is anticipated that it will go into operation on engines ordered in 1973. A last but important point regarding this latest engine control is the fact that the interface between engine and remote control is built into the engine and therefore the system forms an integral part of it. This results in the author's company having to adopt certain restrictions in the indiscriminate application of alternative bridge control system to our engines in future.

CRUDE OIL AND DUAL FUEL OPERATION

Engines of today's RND-range will not only consume any heavy fuel but are also suitable for the burning of crude oils and methane gas provided some alterations are made.

Slow speed engines burning crude oil are running very satisfactorily in stationary plants. Regarding combustion no special problems occur, the only care needed being to keep the temperature of the fuel at about 40°C (104°F) when the latter contains paraffin, as otherwise the centrifuges become chocked. The stationary engines are running without any special safety protection against leaks on the engine, as the machinery hall is sufficiently vented. For ship propulsion the fuel system of the engine would have to be protected. The whole fuel-tubing system is provided with double-piping, and other elements like the fuel pump are boxed-in to avoid any liquid leaks outside this system. An extraction fan provides the necessary vacuum and would in case of leakage suck any evaporation of the crude oil to the atmosphere, thus avoiding any contamination of the engine room.

The transportation of gas in a liquefied state across the seas is a fast growing venture. In the case of methane tankers a Diesel-gas engine can provide an economical means of propulsion since the gas lost by evaporation from the cargo can in many instances furnish the fuel required for propulsion. Compared to it, a larger amount of liquid fuel is additionally required to the gas supplied by evaporation in the case of steam or gas turbine plant.

Fig. 15 shows a comparison between an internal combustion engine asking for 1670 kcal/bhp h (6630 Btu/bhp h), and a steam or gas turbine plant based on consumption of 2250 kcal/bhp h (8930 Btu/bhp h). Analysing built and projected LNG-carriers, an evaporation rate of 0·2 per cent to 0·25 per cent/day seems realistic, the latter figure being considered for the example given. The propulsion power required is assumed to be 0·59 kW/ton dead weight, values of 0·76 to 0·78 being known for the bigger ships projected for tanker capacities between 88 to 125 thousand m³ (3100 to 4400 thousand ft³). The graph indicates that for a cargo being 100 per cent methane the Diesel-gas engine would require liquid pilot fuel approximating 6 per cent of the total heat input to the engine, whilst the turbine plant necessitates 5 times more, i.e. 30 per cent. The evaporated gas being wasted if not used in the propulsion plant, this would mean a fuel bill nearly 7 times higher for the turbine solution when transporting gas. These considerations would, of course, only be valid for loaded voyages. Under ballast conditions the boil-off rate is considerably less.

Also for stationary applications power plants with outputs below 10 MW are becoming attractive in connexion with waste heat utilization, in which case a large-bore Diesel-gas engine can provide the required energy with even higher thermal efficiency than giant supercritical steam plants.

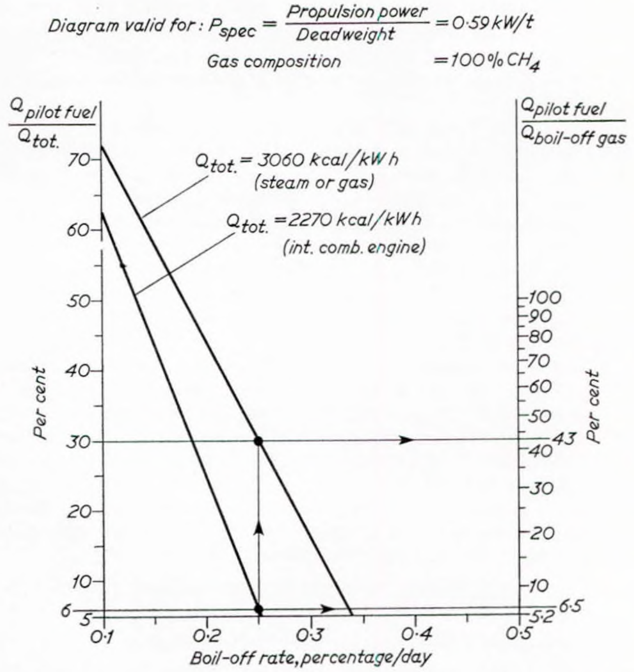


FIG. 15—*Comparison of pilot-fuel requirements of a Diesel gas engine compared with turbine propulsion plant for a typical LNG-carrier*

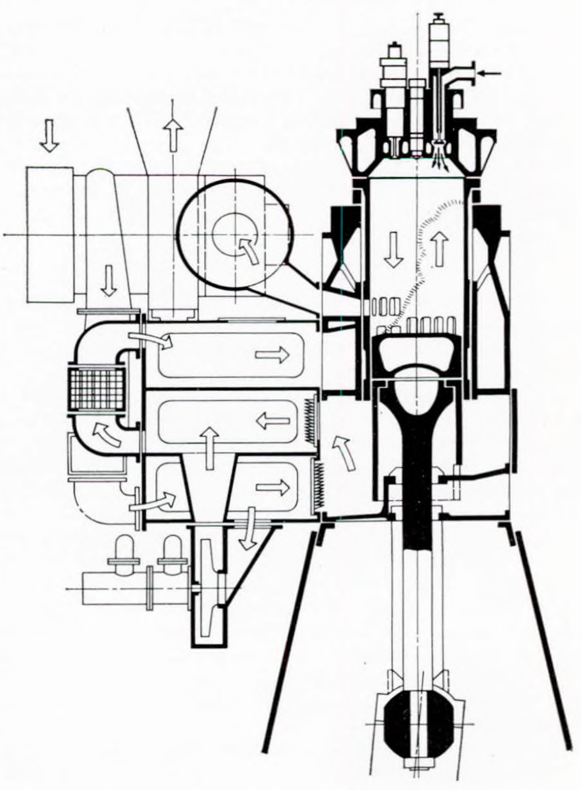


FIG. 16—*Part-cross-section of a RNMD Diesel gas engine*

The gas burning version of the RND90 Diesel engine is designated RNMD90 and shows the following modification compared with the straight forward Diesel version (Fig. 16).

In a modified cylinder cover insert a hydraulically driven gas injection valve is mounted, the fuel-gas being under a pressure of approximately 3 atm. (43 lb/in^2). The gas is introduced in the combustion space during the latter part of scavenging period, and is directed towards the uprising stream of scavenging air to ensure a good mixture between gas and air. The ignition is by a pilot spray, which may be as little as 5 per cent of the total heat input.

One of the main problems in burning gaseous fuel is the knocking effect, which is largely influenced by the temperature level in the combustion space. Therefore additional scavenge air supply, allowing the dual-fuel engines to operate at relatively high b.m.e.p. without knocking, has to be provided. This problem is solved by replacing the auxiliary blower used on the RND-type Diesel engine for part-load operation by several such blowers in parallel and operating over the full load range. They are connected again in parallel with the active piston underside pumps, as can be seen in Fig. 16.

A control equipment which allows switching over from Diesel to gas operation and vice-versa, and which makes it possible to adapt the distribution of the power produced by the gaseous or liquid fuel—depending on the amount of gas momentarily available—is a further important part of the dual-fuel engine. As mentioned above, the temperature level of the dual-fuel engine when running on gas is lower than in straight Diesel operation. This is, of course, due to the higher amount of specific air available. It has further the following advantages:

a) due to clean fuel, combustion and exhaust are clean and therefore the engine parts stay clean;
b) practically no abrasive ash-particles within the combustion space mean low wear of piston rings, liner and piston skirt, as well as lower lubricating costs, since straight forward mineral cylinder lubricating oil can be used;
c) lower temperature levels mean reduced thermal stressing.

As these factors add up to longer periods between overhauls and also reduced maintenance work, and based on experience with smaller trunk piston two-stroke dual-fuel engines, one can expect that the useful life of rings and liners may be extended by some 50 per cent at least compared with straight forward Diesel operation.

The first RNMD-Diesel-gas engine to enter service will be a 7RNMD90 with a nominal output of 14 930 kW to be installed in a LNG/LPG tanker.

MACHINERY INSTALLATION

Finally, it must be emphasized that good service results are not only influenced by the adequate design of engine components alone, but also by the correct layout and maintenance of the propulsion installation as a whole.

The importance of correct pressurizing and venting of cooling water systems has already been pointed out (Fig. 8). Appropriate layout of the air supply to the engine's turbo-chargers becomes more important the higher the specific output of the engine installed, in relation to the engine room space available. It pays in the long run to have sufficient fresh air for combustion to be provided by an arrangement as shown in Fig. 17. Thereby the air filters on the turbo-chargers and the air coolers on the engine will be less prone to fouling and to its resultant adverse effect on thermal loading of the engine. Delivering enough air to the engine ensures good continuous running conditions. Engine rooms where one can feel the under-pressure when entering due to lack of air supply to the running engine, are usually significant for "asthmatic" and thermally overloaded engines.

For big engines special attention had to be given to the engine foundation in the past. The development in engine seating and bolting arrangement can be seen in Fig. 18.

For the RSAD76 and earlier RD76-engine three rows of bolts with originally large single respectively double chocks on each side were introduced. Due to the fact that large chocks and the ones of the inner row were difficult to fit, bad fitting resulted causing additional stressing in the bedplate. When redesigning the cross-members of the bedplate it was also found beneficial to eliminate the inner row of bolts and chocks. The number of double outer row bolts is sufficient to provide the pre-tension required for the functional resistance to overcome the shearing and loading forces acting in the horizontal fixation plane. However, the longitudinal base girder of the ship structure is not symmetrically loaded any more if designed as shown on the lower left-hand sketch of Fig. 18.

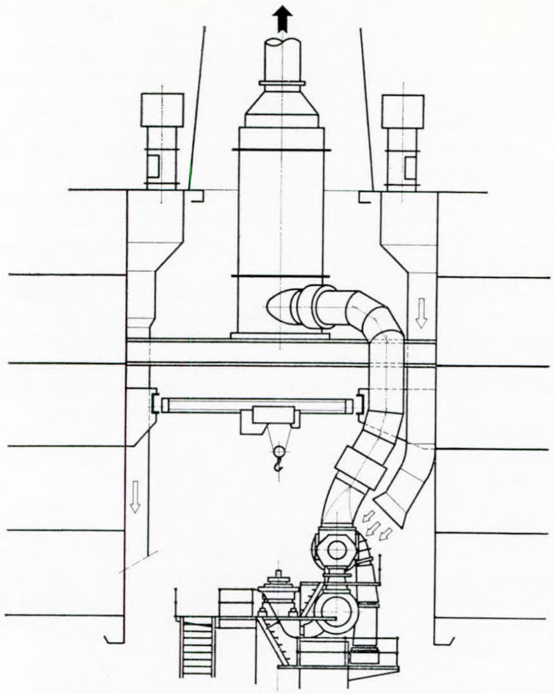


FIG. 17—*Arrangement of air-supply to turbo-charger*

Therefore, a double girder construction has been retained as shown for the RND76 and where the loading and shearing forces are symmetrically transmitted to the double bottom construction.

This layout has been introduced for newly built RD and all RND engines of 760 mm and 900 mm bore, whereas for the RND105 three longitudinal girders are required, and for the RND68 the short middle girder is omitted.

The thickness of the double bottom top plate is slightly more than the one of the lower longitudinal bedplate-girderplate and depends also on frame distance and double bottom height. It is understood that the engine foundation should continue well forward and aft of the actual bedplate to distribute the loads over a larger section of the ship structure.

As can also be seen from Fig. 18, pads have been provided on the older designs to simplify the machining of the actual fixation surfaces. However, modern techniques in today's shipyards allow the machining of the top-plate itself, so that the welding of these pads could be eliminated altogether. If necessary, the engine foundation bolts can also be tightened by hydraulic jacks.

A concern of Sulzers is the relationship between power absorption of the propeller and engine speed, which is a factor influencing the reliability of the ship's prime mover during its whole life in service.

It is recommended that the propeller is designed to absorb at sea trials of the new, fully laden ship 85 to 90 per cent of the maximum continuous output at the nominal speed of the engine (Fig. 19).

This is recommended not because the maximum continuous output indicated is not really the power which can continuously

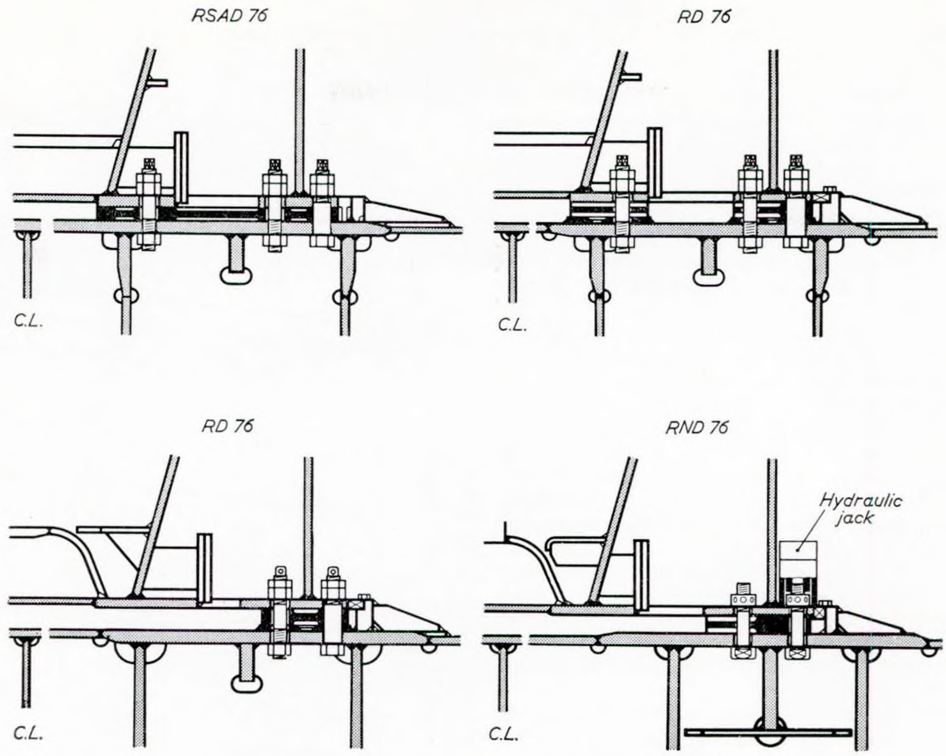


FIG. 18—*Development in engine-seating*

be requested, but because hull deterioration and fouling very quickly imply on average a 10 to 15 per cent increase in power-demand to be able to maintain the engine speed. In tropical waters the influence of fouling is even more marked.

If the above recommendation is not observed for the new ship, it will mean that—as the operator is likely to maintain the engine speed independent of hull conditions for instance with time charter—the engine will be severely and continuously overloaded after one or two years service, this with uneconomic consequences regarding maintenance. These remarks are valid of course for any engine type.

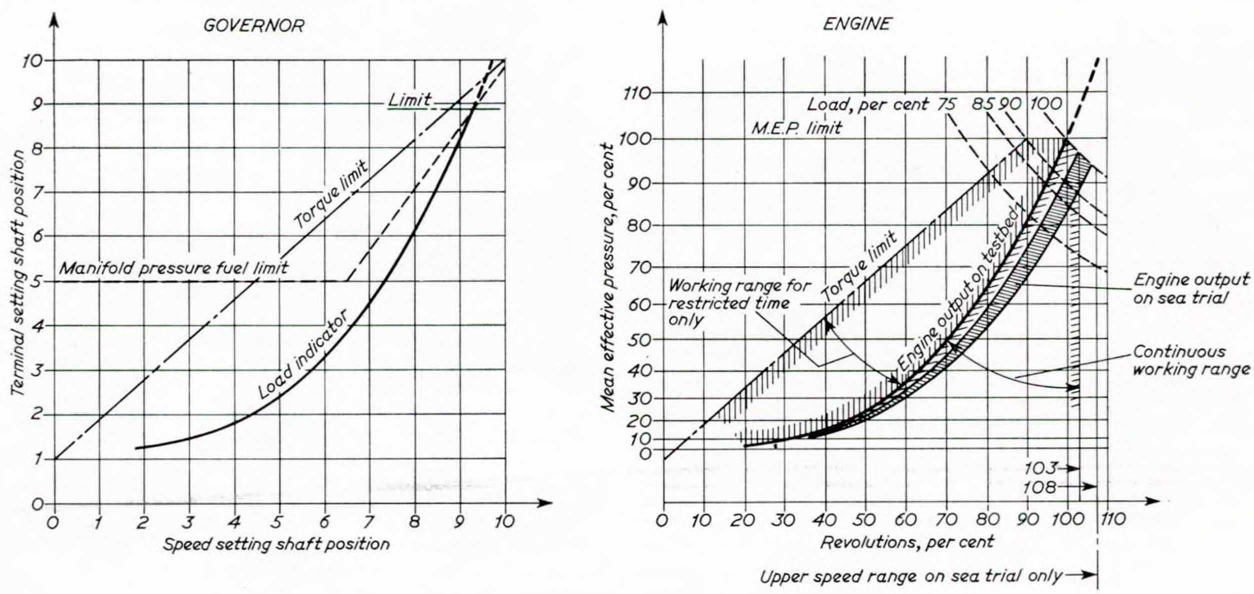


FIG. 19—*Chart of recommended propeller layout*

REFERENCES

1) "Sulzer Marine Diesel Engines". *The Motor Ship*, Special Survey, October 1969.
2) WOLF, G., 1971 "Large Diesel Engines Development Problems of Cylinder Covers, Pistons and Liners". CIMAC, Stockholm
3) DESVAUX, N. P. E., 1972 "Development of a High-Tin-Aluminium Plain Bearing Material". *Tribology*, April.

Discussion

MR. G. C. VOLCY, M.Sc., M.I.Mar.E., said that for some time, he had been following with interest the development of the design of the main propulsion engines built by the author's company, especially types SD, RD and the latest one, RND. The main structural changes between these types were evident to everyone, but he pointed out that the company had always paid particular attention to the design of the construction of the engine base, not only by way of the engine girder and bed plate, but also for the main engine seating incorporated in the double-bottom steelwork.

Bureau Veritas had often been asked, by shipbuilders, to express its opinion on the requirement, by Sulzers, for main engine seatings such as were shown in the upper part of Fig. 18, from which it would be seen that they attached great importance to resistance to bending stresses and required the shipbuilders to apply an important measure of inertia I/v by increasing the upper flanges of these seatings, not only by horizontal inner bottom plating, but also by heavy section longitudinal plates. Nevertheless, although pleased by the care taken to protect these engines from deformation of the double-bottom steelwork, Mr. Volcy had not been happy to see that account had not been taken of deformation due to shear forces. However, the lower part of Fig. 18 showed that the engine builder had now taken account of the resistance of engine seatings to shear force deformation by installing heavy section longitudinal girders beneath the engine bed plate.

Unfortunately, he was still not completely happy, because the author's company, in seeking to simplify bolting the bed plate to the seating, recommended installing the double longitudinal girders in the double bottom in such a way that only the outside bed plate girders could contribute in resisting the effects on the bed plate of double-bottom deformations which were, it was now universally agreed, not only increasingly important, but also increasingly detrimental to the performance of main engines. This recommendation did not fully employ the advantages of the sound concept of the Sulzer engine bed plates, wherein the crankcase walls were formed by strong longitudinal girders which could and should, resist all external influences from the bed plate. He had not quite understood the author's explanation of the reasons for this recommendation; since the engines were condemned to having to withstand harmful outside effects arising from even greater double-bottom and hull steelwork deformations, every effort should be made by main-engine builders to counteract these effects and not to imperil the prospects of installing large Diesel engines in current and future large tonnage vessels incorporating increasingly flexible steelwork.

Turning to the positive aspects of the development of the RND engine, Fig. 20 showed the sound design of the bed plate. Mr. Volcy drew attention to the strong longitudinal plates forming the webs of the engine girder and allowing the cylinder block better to withstand the deformations imposed on it through the columns. This was a sound improvement over the earlier design of these engines. It seemed to him that this indicated that the author's company were reverting to the general concept of the main engine as represented by the earlier SD type. In his opinion, this was the best solution for resisting the effects of hull steelwork deformation, by creating a main-engine girder, where the upper and lower flanges, formed by the cylinder jackets and bed plate respectively, were joined by the double vertical webs. Such a girder was able to withstand currently existing and future anticipated deformation levels and protect the performance of the main bearings and crankshaft. His own idea of such an engine girder had been put forward in 1966* and was shown in Fig. 21.

Fig. 21 was self-explanatory and a similar solution had recently been adopted by a British engine builder. Even though the solution shown in the illustration might not eliminate all

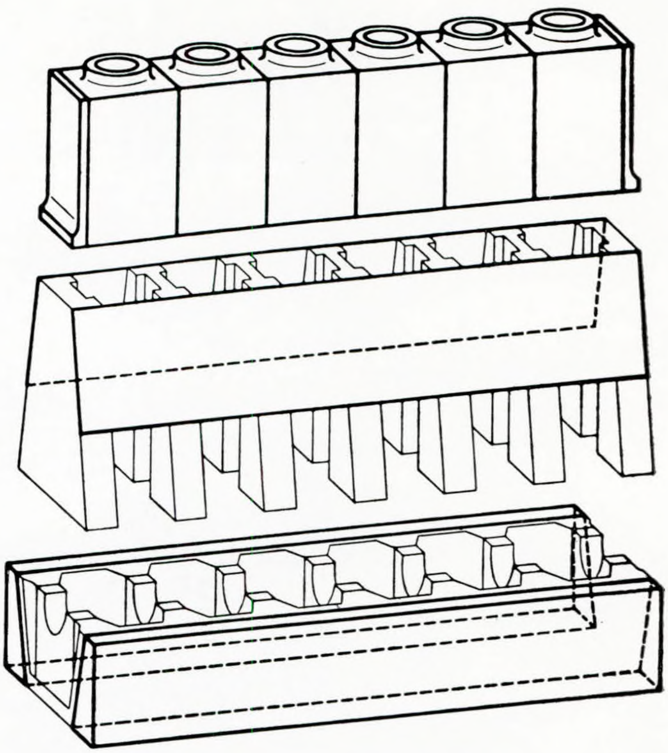


FIG. 20—*Schematic layout of the bedplate, column and cylinder jacket assemblies of the RND engines*

Courtesy of 'The Motor Ship'.

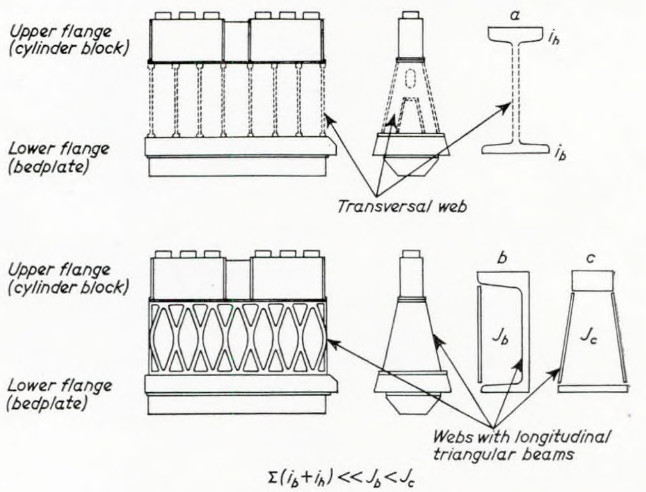


FIG. 21—*Development of the concept of the engine as a beam*

influences detrimental to the performance of main bearings and crankshaft, it could be suggested as a supplementary remedy and arrangement to assist engine builders in combating the harmful effects from hull steelwork deformation. These effects were discussed in a paper in 1964†.

Bureau Veritas, being well aware of the importance, and

* Bourceau, G., and Volcy, G. C. 1966. "Some Aspects of the Behaviour in Service of Crankshafts and Their Bearings." *ATMA*.

† Bourceau, G., and Volcy, G. C. 1964. "The Influence of Hull Flexibility on Main Propulsion Units in Motorships." *Nouveautés Techniques Maritimes*.

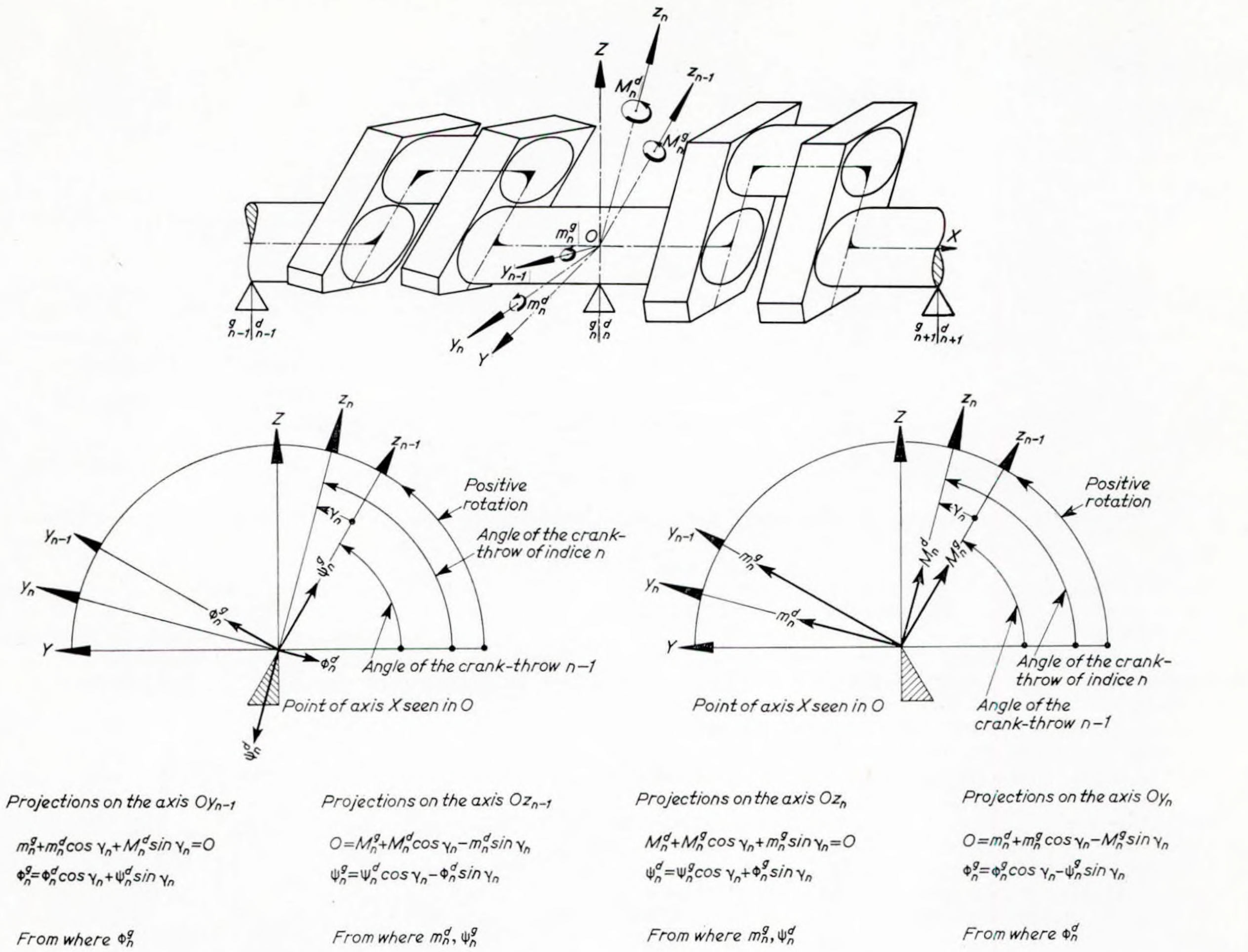


FIG. 22—*Summary of angular deformations and moments in way of a journal bearing of two adjacent crank-throws*

effects on the performance of propulsion plant, of double-bottom deformation, had continued to study the problem and some results had recently been published‡.

Fig. 22 showed the angular deformations and moments, in way of the journal bearing of two adjacent crank throws, which had served as a basis for determining the actual position of the main journal bearings, on the basis of deflexion values.

Mr. Volcy presented one example of the calculation when, on a seven-cylinder engine, the deflexion value, as indicated in Fig. 23, permitted the determination of the position of the journal bearings, shown in Fig. 24. The crankshaft had already been pre-deformed in hogging, with a 0·5 mm maximum deflexion, during installation. It was worth noting that the calculation carried out had shown a maximum deflexion of 0·46 mm.

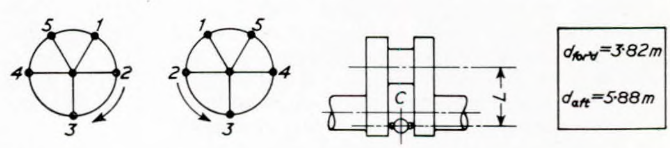
The degree of correlation between calculation and measurement was quite satisfactory and the quality of the calculation of crankshaft deformability quite promising; it would be of assistance to engine builders in overcoming double-bottom deformation effects so as to ensure that main propulsion engines were built to run and not just to be repaired.

MR. J. H. MILTON, M.I.Mar.E., said that the author had given plenty of food for thought on Sulzer's latest large bore engines and had described in some detail the steps which had been taken to overcome troubles experienced with pistons, covers, cooling water systems and crosshead bearings.

‡ Volcy, G. C., and Trivouss, A. 1972. "The Crankshaft and Its Curved Alignment." STG/ATeNA Meeting, May.

Fig. 3 showed stresses in a piston crown which had apparently been measured with strain gauges. This in itself, when one considered the difficulties in getting the gauge leads out from a moving piston, was little short of miraculous.

Fig. 5, illustrating a bore cooled cylinder head, was ex-



Cylinder	Forward					Astern				
	1	2	3	4	5	1	2	3	4	5
1	0	−2	−1	−2	0	0	−2	+1	−1	0
2	0	+1	+7	+3	+1	0	+2	+7	+3	0
3	0	+3	+8	+1	0	0	+1	+8	+4	0
4	0	+6	+11	+1	0	0	+1	+11	+9	+2
5	0	+6	+10	0	−1	0	+1	+11	+7	+1
6	0	0	+2	0	1	0	−1	+2	−2	0
7	0	0	+3	+4	+2	0	+2	+3	0	0

FIG. 23—*Deflexion values of a crankshaft realized during curved alignment*

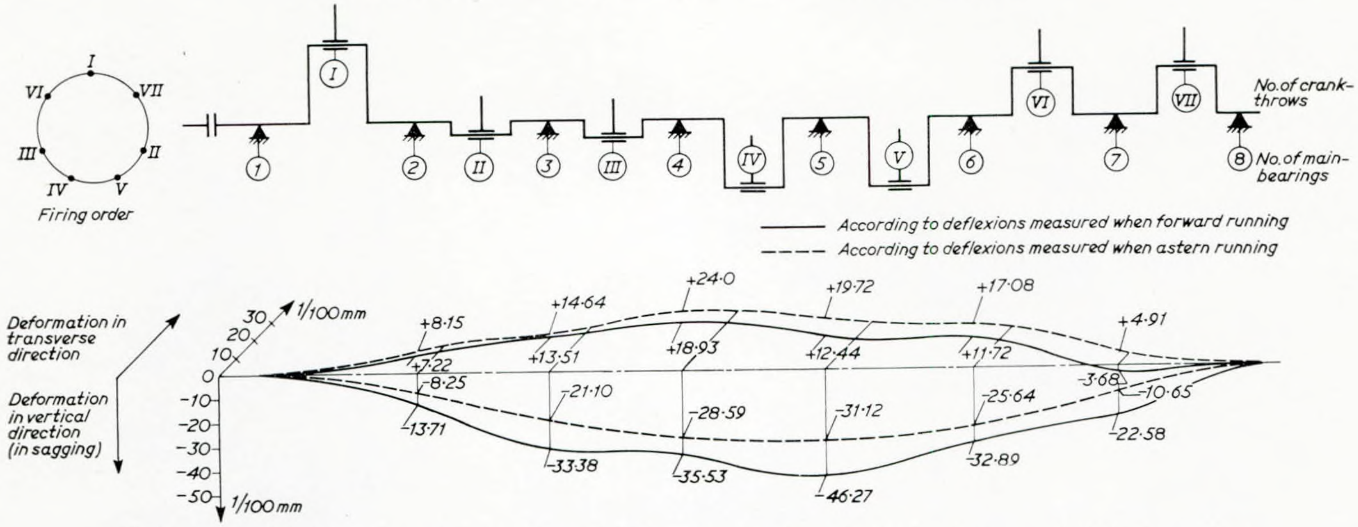


FIG. 24—*Real deformation of a curved crankshaft determined on the basis of measured deflexion values*

tremely interesting, but apart from the initial cost, made one consider what would be the effect of even the slightest coating of scale in the bores.

Fig. 8 showed the venting of a fresh water cooling system. The importance of this in new installations was not, he imagined, always realized. There had been recent cases where, in new installations, cooling water had become aerated through the use of open ended vent pipes, the resulting corrosion after only one year's service being serious enough to require renewal of cylinder liners.

The author had discussed his attitude towards bearings and had stated that, for higher outputs, no troubles were to be expected, except for the crosshead bearings.

During Mr. Milton's career as a surveyor, top end bearings of most types of engines had been examined, in varying states of deterioration, from minor crazy cracking of white metal to fusion with the steel bearing shell, resulting in crankcase explosions and occasionally even loss of life.

He imagined that most people present were more concerned with operation than with design and, with this in mind, he postulated the following to the author. Undoubtedly, improvements had been made by increasing the bearing area, stiffening the pin assembly (one designer had pins nearly 80 per cent of the cylinder bore in diameter), super-finishing the pins and using very thin shell bearings in flexibly mounted bodies. However, was not the missing link for trouble-free operation to be found in efficient lubrication?

From experience, there had often been indications, both from the burnished surfaces of top end pins and thermal fatigue cracking of the white metal bearing surfaces, that the lubrication had been of a boundary nature, the white metal only just being maintained below melting point by the cooling effect of the flow of lubricating oil through the bearing channels.

It was felt that designers of crosshead two-cycle, slow speed oil engines realized that, owing to the microscopically thin oil smear which existed in their top end bearings, they must keep the loading down and use very stiff superfinished pins. These factors might see an engine through its early days, but such bearings were extremely sensitive and would not tolerate any lubrication deficiency. Did the author not agree that, in addition to experimenting with very thin potentially dangerous white metal linings in bearing shells, every effort should be made to obtain all the advantages of improving the oil film between the bearing surfaces?

Why did the white metal of top end bearings apparently need a greater fatigue strength than that used for bottom end bearings, bearing in mind that, area for area, the loading on the former was less than the latter? Also, with regard to the so-called fatigue strength, what mode of fatigue was inferred? In his opinion, the fatigue from which crosshead bearings suffered was a combination of thermal and surface shear, resulting from poor lubrication.

As a means of improving lubrication and flexibility of these crossheads it had been suggested some years ago that a floating pin should be used (see Fig. 25). The contention (yet to be

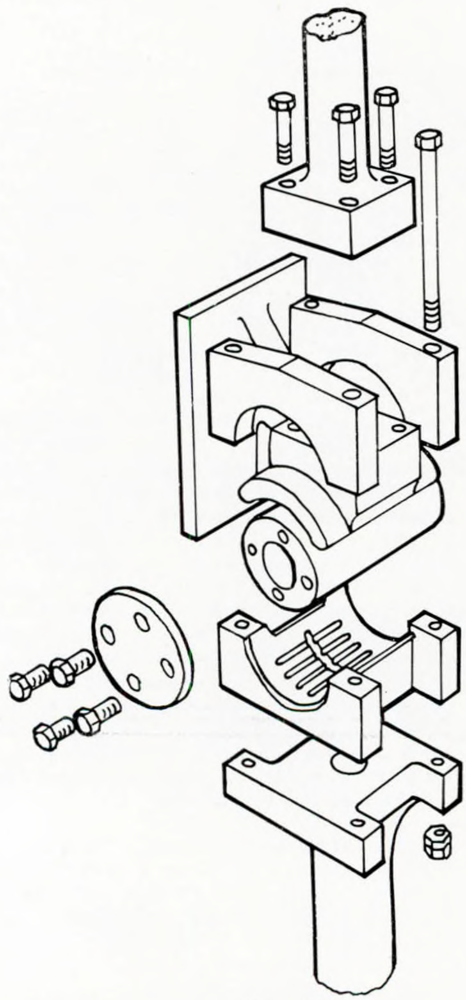


FIG. 25

proved) was that the floating pin would rotate in service due to variation in load on compression and firing strokes, and, carrying lubricant with it, would, in addition to overcoming the lubrication problem, give the crosshead assembly the additional flexibility of oil films above and below the pin.

In the author's opinion, might the combination of shell bearings and a floating pin be a possible solution justifying further research? All other factors being equal, a slow speed main propulsion oil engine free from top end troubles would be more reliable and thus have a greater market potential.

MR. J. A. DUNCAN, M.I.Mar.E., said that the information in the paper impressed one with the extent of the thought and care which had been given to the development of the RND engine, both with regard to the various component parts and the engine as a whole. Whether or not this careful designing would be successful in producing a faultless large marine Diesel engine would take some time to establish. However, Mr. Borgeaud could be sure of one thing: no one would be more anxious for its success than its operators.

It was obvious that in a relatively short paper on such a major subject the author was obliged to be selective in his presentation. So it was in this case and, consequently, in many ways, the information was incomplete.

Referring to combustion chamber design, the author had described the arrangements for bore cooling, which seemed to be an improvement on current practice. Unfortunately, he made no further reference to cylinder liner cooling, although he dealt fairly extensively with cylinder cover cooling. There was one problem associated with cylinder liner cooling which the structural changes described would not eliminate, however, and that was contamination of the cooling water by cylinder lubricating oil and products of combustion arising from leakage at the cylinder oil injection quill joints—what had been done in the RND design to prevent this leakage of oil into the water space? In the RD engine, these joints had always been a hazard, making it almost impossible to keep the jacket water free of oil. In his own experience, a number of liner and cylinder cover failures by fracture had been considered to be due to overheating as a result of oil deposits on the cooling surfaces. The amount of oil that could enter the jacket water, in even a few days of leakage, had to be seen to be believed and, unfortunately, it was not readily detectable until it was there in quantity, by which time it was too late.

With regard to cylinder cover cooling, while the design changes described were not questioned, they did not represent all that might be necessary to remove risks. There could be external hazards and one was the arrangement of cooling water inlet internal pipes in the RD engine covers. In certain instances these could be fitted the wrong way round, resulting in an area of reduced circulation which ultimately allowed the cooling surface of the cover to overheat and fracture. He would be interested to know if the arrangements for correctly locating these pipes in the RND covers had been improved. In the RD engine the pipe flanges had been rectangular, with four stud holes, and had been provided with a locating dowel in the flange face. If this dowel were lost, or removed because of damage and not replaced, through carelessness or inexperience, the flange could be turned 180° and fitted without difficulty in that attitude. This hazard could be eliminated by using a triangular three stud or pentagonal five stud flange, giving positive one-way location.

He was rather surprised at Mr. Borgeaud's reference to nitrites being poisonous, in his remarks on corrosion inhibitors. It was not so very long ago that engine cooling systems used for circulating potable water distillers had been required by the DTI to be corrosion inhibited by non-toxic chemicals, as an alternative to those such as potassium bichromate which were regarded as poisonous. Sodium nitrite was approved as non-toxic at that time. However, he supposed that "poisonous to humans" was not the only criterion. An inhibitor poisonous to certain metals could be equally objectionable. It was commendable that a lot of trouble seemed to have been taken to find a really good inhibitor, which he noted was to be offered in the near future. He hoped that though its quality was high, its price would not be.

He had never quite been able to understand why the Sulzer engine jacket cooling water system should apparently be regarded with some lack of confidence by its designers. As a result, its arrangement had gone through innumerable permutations without there being really any basic change. It had always seemed to him to work reasonably well, although vent pipe bores in some of the earlier arrangements had perhaps been on the small side. He strongly suspected that the dangers ascribed to it in the paper, mainly risk of air ingestion and inadequate venting, together with formation of vapour pockets, were not in fact nearly so prevalent as supposed, but that in many cases where liners and covers had overheated and fractures had resulted, these were more likely to have been due to oil contamination than to faulty cooling water circulation.

With regard to crosshead bearing fatigue failure, he was pleased to note that shell bearings were now being tried. This seemed to be a positive attempt at radical improvement, but why had their introduction taken so long? The nature of the problem had been clear for at least eight years and no one dealing with large marine Diesels could have been in doubt about its prevalence. Shell bearing production techniques were fully established and manufacturing capability had been available at any time during that period. Despite this, years had been spent trying one variation after another on the old standard bearing in efforts to get it to accept loadings which were at the limit of or beyond the physical capacity of its materials when arranged in the conventional design. If it was considered too dangerous to depart from the use of the traditional material, then surely the method of application of that material must be radically changed in a way which would enhance its load bearing capacity, if it was to continue to be used successfully. In that context, he noted that, in the paper, the thickness of the white metal in the new shell bearings was stated to be between 1·5 and 2·0 mm. Some years ago, when investigating the possibility of using shell bearings in crossheads, he had been informed by manufacturing experts that the maximum tolerable thickness for adequate support of the white metal surface and establishment of acceptable fatigue strength was of the order of 0·25 to 0·5 mm. As these dimensions were considerably less than those quoted by Mr. Borgeaud, Mr. Duncan would be interested in his comments on them.

Turning to the question of what relevance some of the improvements described in the paper had to the existing nearly 2000 RD engines in service, could they also be fitted with the improved covers, liners, shell bearings and so on? Surely it would be advantageous to their operators if these improved parts were also available to them, when replacements or repairs were required.

Finally, on a general rather than a particular point, nowhere in the paper was there any reference to the need for field testing, as distinct from test bed running and detail design testing. By field testing, he meant intensive instrumented study of the thermal and physical performance of the engine in its working environment with intent to explore and define its capabilities fully. It seemed to him that the marine engine, and particularly the large marine Diesel engine, must be the only instance of a large, complex, costly engineering product used intensively in vital and what could be, on occasions, dangerous service, which did not have its behaviour and characteristics thoroughly investigated in its operating environment. How was it possible to discover the problems and determine their solutions otherwise?

MR. P. S. POWELL asked if the author could pursue the question of cooling water treatment a little further. He would probably know from the researches that his company had carried out that this had been a subject of considerable interest in the United Kingdom during recent years, and that work had been done by such authorities as the British Ship Research Association, independent research laboratories commissioned by Diesel engine manufacturers, the Admiralty and, of course, chemical manufacturers themselves.

The author had quite rightly drawn attention to the comparison of chromates, soluble oils and nitrites, and dismissed chromates as being poisonous and ineffective with salt water contamination, and soluble oils because of their tendency to form gummy deposits which could very much impede heat transfer. So one came back to the standard treatment at the moment which, of course, was nitrites in most cases. However, he wondered if Mr. Borgeaud had considered all the implications of the nitrite based treatment and the improvements which had been brought about in this form of treatment. Going back through the years, the use of nitrite had been recommended to be carried out in an alkaline solution so that it had been necessary to add a borate buffer to it, and he wondered whether Mr. Borgeaud's reference to nitrite being corrosive to zinc was, in fact, a factor of the borate buffer and not the nitrite itself.

The author had also referred to the effect of chlorides on nitrite. It was quite true that nitrite would not protect indefinitely if chloride contamination was experienced. He would be very interested to know if Mr. Borgeaud had any figure in mind for the new inhibitor which his company had investigated. Mr. Powell assumed that there was a chloride limit and that one could not run the engine, for example, on neat sea water. In this connexion, in view of the importance of adequate cleanliness in the cooling water systems on the proposed new engine, Mr. Borgeaud's company would presumably recommend distilled water, although the paper did refer to tests carried out with fresh water.

Finally, he said that a previous speaker had already stolen his thunder on the toxicity of nitrites. As had been pointed out, these had been approved by the British Department of Trade and Industry for use where low temperature evaporations were employed for the production of distilled water for drinking. He assumed that, at the concentration at which these products were normally used (i.e. about one per cent), one must consider them as having a low toxicity.

Captain J. A. Smith, D.S.C., V.R.D., B.Sc., A.C.G.I., M.I.Mar.E., thought that, on the whole, those who had operated the RD engine had had an interesting and fairly happy time.

Concerning crosshead bearings, the original RD90 white metal in cast steel bearings had, in his experience, proved possible to live with. They had shown signs of cracking which, in fact, had been more of a threat than a hindrance to operation. The builders had very rightly corrected this, and in his company three prototype shell bearings of the intermediate thickness, i.e. 16 mm, had been in service for three years in a tanker operating with an mip of about 8·8 bars. They had been a noticeable improvement on the original and, although the prototypes had been superseded by the thin wall bearings, they had remained in satisfactory service in his company.

There was no mention in the paper of any measures to maintain the surface finish of the crosshead pins. With the older type of crosshead bearings—those in the RD—the surface finish of the pin had been critical. After three or four years' service the surface finish would have deteriorated from the original four microns to 12 or more, and it would be necessary to polish the pin to restore it to something like the original four micron finish. This was an expensive process, and in another make of engine with a similar but more catastrophic problem, the trouble had been reduced by plating the pins. Had this or any other similar process been considered for the Sulzer engine? From the operator's point of view, to do the super-finishing, or restoration of surface finish properly in service, entailed removal of the crosshead to a workshop, because nobody he had yet come across had been able to guarantee a satisfactory finish *in situ*, and the process could cost £200 per pin once the pin was in the workshop.

He then referred to the burning of crude oil, a development which would give everybody something more to think about. It took him back to the existing type of engine, the RD, burning heavy fuel which required heating. His company was having the utmost difficulty in maintaining steam tracer pipes in a tight condition. In fact, they seemed to be faced with a Forth Bridge problem because, as soon as one pipe was repaired, another would begin to leak. He wondered whether they were behind the times and whether developments in materials of tracer pipes might have solved their problems; he would welcome the author's advice.

Correspondence

Mr. R. F. Munro, M.I.Mar.E., wrote that the importance of the arrangement for bore cooling the cylinder cover together with cooling water treatment and system design was emphasized by the amount of space devoted to these matters in the paper. Practical difficulties were experienced with the fettling of conventional cylinder cover castings and they were liable to sustain indentations from pneumatic tools, resulting in stress raisers at the position "a" in the graphs of Fig. 4. This particular hazard was to be removed by the adoption of bore cooling for these components and it would be interesting to learn whether means had been devised for detecting the formation of deposits on cooled surfaces in service. Local deposits in horizontal bores could result in hot spots which, together with corroded surfaces due to a breakdown in the water treatment, might well combine to affect the fatigue strength and fatigue crack development characteristics of the material. A similar effect could be expected from surface roughness within the bore hole and it would be interesting to know whether these various factors had been studied in detail. Were any figures available regarding the decrease in strength with the time factor for the materials tested and, in connexion with the corrosion fatigue curves in Fig. 7, what might be the error, expressed as a percentage, in transcribing the test results to apply to the slow running engine? Many of the troubles associated with the cylinder covers would be avoided when steel castings were no longer employed, but there would remain the effect of thermal stress increments due to one part interfering with the expansion of another and he wondered to what extent this was being considered in the design of future cylinder covers. It was clear that the importance of the proper operation of the cooling water and venting systems could not be over-emphasized and it was concluded that active measures were taken to bring this to the especial attention of the operating engineers on board.

In conclusion, what measures did the author's company take to ensure that all items of spare gear supplied to owners, from whatever source they might be obtained, complied with the designers' specifications for materials, dimensional tolerances and standard of workmanship?

Author's Reply

Mr. Borgeaud, in reply, thanked all who had contributed to the discussion.

He thought that Mr. Volcy and himself were not really in disagreement. His company had always tried, in the design of their slow speed engines, not to analyse the structure for stressing only, but to give the aspects of stiffness the necessary considerations, as they were of the opinion that, if the worst came to the worst, it was preferable that the engine stiffened the ship structure than the other way round.

At the interface between bed plate and tank top, the interconnexion must take up the shear and normal forces which resulted from the required compatibility of forces and deformation of the two counterparts at this point.

For the RD engines, the bed plate seating was changed from the two-row chocking per longitudinal girder to the one-row double chock, after having experienced some difficulties with cracks around the bearing saddles. Thorough investigations and model testing* had shown, at that time, a better stress flow between engine and tank top seating when using the one double-row chocking support. At the same time fitting of the chocks was proven more reliable, as with two rows one could never really ascertain which of the chocks were load-carrying or whether undue deformations were imposed on the bed plate. His company had made sure that, in spite of only one row of chocks, the necessary shear load could still be transmitted at the said interface and it was felt that the subsequent experience had fully proven this layout.

Mr. Borgeaud agreed with Mr. Milton that it was astonishing what one could achieve with today's measuring techniques, but by using radio-telemetry it was relatively easy to record the strains of moving parts.

Referring to crosshead bearings, the author said that although he agreed with Captain Smith that "people lived with it"—the more so as Sulzer had no records of RD or RND crosshead bearing failure wherein the mosaic pattern of the fatigued white metal had developed into a break-down of the gear train itself—this was a component which had definitely to be improved on.

Insufficient lubrication would result primarily in wiping, a problem which was not known with Sulzer crosshead bearings, except for some isolated cases where rusty pins led to a more or less localized wiping of the white metal, but which could be neglected in this context. The problem was one of pure fatigue, whereby the relatively sharp blow resulting from the maximum firing pressure had to be taken up by a bearing where the relative movement between the pin and the bearing was not of rotating but only of oscillating character, and this within a relatively small angle. This again resulted in a poorer capacity for building up an oil film and a lower specific loading capacity compared with the bottom end bearing. Sufficient lubrication oil was also necessary as a coolant to counteract the heat generated by friction. Whilst white metal gradually lost its fatigue strength with increasing temperature, the aluminium-tin bearing would keep its fatigue strength within the temperature range to be expected under normal service conditions.

As to the crosshead bearing with a floating pin, proposed by Mr. Milton, Mr. Borgeaud agreed that with some effort it could be designed to allow for fitting shell type bearings, but it still needed to be proven that this relatively long pin would allow for an even load distribution.

Referring to Mr. Duncan's comment on cylinder liner cooling contamination, the author pointed out two distinct changes from the RD to the RND series. The first was that on RND engines the lubricating quills—which had given some trouble with the RD design—had been designed with a neutral space between the combustion chamber and the cooling water system. This meant that if the quill leaked, the gases from the combustion chamber could not reach the cooling water system but would be led to atmosphere. The second important change was the fact that emulsion oil was not recommended any more, thus the possibility of a break-down of the emulsion was avoided altogether.

Regarding the toxicity of nitrites, it was accepted that sodium-nitrite water treatment was regarded as non-toxic in the concentration used. As the lethal dosage of nitrites was only about five times higher than that of chromates which, as Mr. Duncan had said, were regarded as poisonous, the author might be excused in classing nitrites as poisonous to a certain extent.

The jacket cooling system and the importance of its venting had not been brought up as a general excuse for troubles, and it was accepted that in the past many troubles could be imputed to break-down of the oil emulsion of the cooling water and subsequent overheating of the component. This had, of course, nothing to do with the system venting. For a long time he had been of the opinion that as long as the system was statically vented, e.g. before starting the engine, and that water was present, everything was alright. But this was not sufficient and one had to provide for continuous and effective venting of the system whilst the engine was running. If one had trouble with either liner or cylinder covers and the case could not be explained by break-down of the emulsion oil, the first step Mr. Borgeaud would suggest was to check whether the pressure on the suction side of the water pump of the jacket cooling system was not below atmospheric condition. Alternatively it would also be worthwhile checking, at the head tank, whether too much water or air were circulating, the latter being usually re-aspirated into the system and, he was sorry to say, detrimental to good operational results.

The point regarding the wrong fit of the internal pipes of the RD cylinder cover water inlet was noted and the counter proposal welcomed. Reverting to Fig. 10 of the paper, where the thickness of the white metal in the new shell bearing was indicated to be between 1·5 and 2·0 mm, Mr. Borgeaud said that a layer thickness of less than 0·5 mm was usually used in conjunction with a tri-metal bearing (type 2 of Fig. 10). It was known that, with white metal, reducing the thickness below a few tenths of a millimeter would give an increase in fatigue strength as the thickness was reduced. Increasing the thickness above the figure quoted of 0·25 to 0·55 mm, would only marginally increase the fatigue strength, but a thicker white metal layer would not be of disadvantage and certainly be less sensitive to dirt and surface damage due to mishandling.

Regarding the relevance of some of the improvements to the RD engines in service, where possible and necessary they had been adapted to RD engines also. This was true for instance for piston crown, cylinder cover and to some extent for bearings and liners.

The author's company did carry out tests on board and believed that these must be done. However, there was often the difficulty of sorting out the different factors leading to a phenomenon. This could be very tedious and the test bed was, therefore, more appropriate when not the "how much" but the "why" was more important. Usually, measuring on board led to a more statistical way of analysis which, as such, valuably complemented test bed information.

Referring to Mr. Powell's points about cooling water treatment, Mr. Borgeaud confirmed that his reference to nitrite being corrosive to zinc was attributed to the nitrite itself.

Regarding the chloride limit, up to 100 to 150 ppm chlorides, the new inhibitor would give adequate protection.

Referring to Captain Smith's question, up to now chromium plating the pins had only been considered for repairs, but as such had given good results provided the plating had been properly done. But for aluminium-tin bearings this surface treatment of the pin could be considered, the more so if, as mentioned, maintenance costs could be reduced effectively. For white metal, shell type bearings the same surface quality as for the older type of crosshead bearings was required.

The bore cooled cylinder cover gave rise to some comments which might finally be answered, together with Mr. Munro's written contribution.

Careless fettling of the cooling water space of the conventional cylinder cover castings had in the past caused harmful indentations in the surface. Immediately after realizing this danger, Sulzer made an exact specification, which defined the minimum required quality of the casting surface. Since additional work was involved for the attainment of the given surface standard, the introduction of the bore cooled cover would eliminate this problem automatically. The surface of the drilled cooling holes was in any case smoother than the casting surface, giving an additional advantage to the new solution.

No special means had so far been developed for detecting the formation of deposits in the cooling bores. As the velocity of the cooling water in the bores was relatively high, loose deposits in this area were very unlikely. The experience gained with bore cooling the top part of the cylinder liner—for the RND and the medium speed Z engine applied since their introduction—as well as with the area of the exhaust ports, also bore cooled,

* Borgeaud, P. E. A., 1967. "Stress Investigations on Large Diesel Engines." *Trans.IMar.E.*, Vol. 79, p.1.

was very good. So far no failure was known due to a blocked cooling hole. The recommendations given to all customers regarding water quality, water treatment and periodical supervision of the cooling water, fulfilled the purpose of preventing operating difficulties caused directly (corrosion) or indirectly (contamination and resultant inadequate heat transfer) by the cooling water. If they were acted on by operators, break-downs of the water system should be eliminated; otherwise the danger, for all designs, was real; but still the smaller the deposits the lower were the stresses in operation.

Measurements on the RSA 76 test engine with a deliberately blocked cooling hole in the cover showed, for the affected area, a temperature increase of around 25°C.

Because of the time involved in the investigations into the influence of corrosion on fatigue strength, it was impossible to run the tests at the service speed, of some 120 rev/min, of a slow speed Diesel engine. In order to assess the error due to the time factor, another GGG 42 specimen was tested at half the speed (750 rev/min instead of 1500 rev/min). The additional reduction in fatigue strength for a corresponding service time of 15 000 hours was some seven per cent, which meant that slightly reduced strength must still be considered at service speed.

The mutual influence of the different deformations in service of two engine parts (e.g. cylinder cover and cylinder liner) to the stressing of the respective parts was automatically studied by strain gauge measurement. They were also studied with modern calculating methods, though considerable additional effort was necessary.

The author's company was very anxious to keep up a combined high quality and full exchangeability for all parts of the engines. The basis of all manufacturing done by them, or by their licensees, were the drawings, which give all the informations required for the manufacturing process in a complete form. In the case of highly stressed parts, additional specifications were published, regarding material quality, necessary testing methods, quality of rough and machined surfaces, repair of casting defects etc. These specifications were approved by the classification societies; licensees were bound to comply with them and were checked by the classification societies.

Spare parts manufactured by any other manufacturer but Sulzer licensees were not covered by the above mentioned instructions. Therefore, it was left to the customer to assess the risk involved and to decide for himself, for or against such purchasing.

Related Abstracts

Reliability analysis of piston rings in low speed Diesel engines

Because a piston overhaul of a large low speed, two-stroke marine Diesel engine can be laborious and time consuming, the Netherlands Ship Research Centre recently conducted an investigation to establish the optimum piston overhaul interval without jeopardizing safe operation of the engine. The decision as to exactly when a piston should be drawn depends on several factors, such as the extent to which the piston rings have worn, ring breakage, the extent of fouling, the overall condition of the piston, and the time available. A questionnaire was designed and ship's engineers were requested to fill in the form every time a piston was drawn from each respective engine. The questionnaire was distributed to 103 ships equipped with main engines of six different makes amounting to 19 types. Calculations were made, based on the average wear of the radial width of a piston ring because this was identified as the most important factor in ring renewal. The maximum allowable wear was taken as 20 per cent or the original width. The calculation of the number of running hours at which the wear criterion is exceeded was based on the assumption that the wear rate of each piston ring runs parallel to a regression line that fits a set of wear data. The wear and extent of fouling of combustion chamber components depend on a number of factors, including power rating and construction of the engine, number of starting manoeuvres, and fuels and oils used. The possible influence of each of these factors on the wear has been neglected because such investigations are difficult on board ship. It was attempted, however, to derive for the engine types and ships involved a correlation between wear and running hours, in terms of an average that could be used to calculate the reliability when making certain assumptions. It was found that the wear of the second ring was notably less than that of the upper ring. Perhaps the second ring, after the upper ring has reached the maximum allowable wear, takes over part of the work previously done by the upper ring, and that this occurs without any harmful influence on engine performance. In that case, prolonged periods between piston overhauls can be used. Evidently, prolonged overhaul periods are obtained when one would raise the wear criterion to a higher value.—*Brandenburg, P. J.: Motor Ship, March 1973, Vol 53, No 632, p 543.*

Studies on low compression ratio Diesel engines (2nd Report: Effects of combustion chamber configuration in a direct injection type Diesel engine)

In a Diesel engine, the form of combustion chamber is considered to be important in designing a low compression ratio engine. In this paper, the optimum form of combustion chamber for performance was ascertained and, by using this chamber, the reduction of the compression ratio to approximately 12 was attained without deterioration of performance, for example maximum output and specific fuel consumption. This seems to depend mainly on the increase of the degree of constant volume owing to the shortened duration of combustion as the compression ratio decreases. To determine the form of combustion chamber, the air velocity, with varied diameters of chamber entrance, was controlled and the relationship between combustion and performance obtained. As the air velocity increased, so did the degree of constant volume, because of increased maximum rate of heat release and shortened duration of combustion; this worked effectively on the thermal efficiency, but in contrast, the cooling loss increased. Thus, this resulted in the optimum diameter of combustion chamber entrance, especially in the heavier load range. When the diameter of the combustion chamber entrance decreased, the throttling loss was considered to be negligible in the direct injection type of engine.—*Miyamoto, N. et al: Bulletin Japan Society of Mechanical Engineers, November 1972, Vol 15, No 89, pp 1423–1433.*