Paper read at the Institute on Tuesday, 12th October, 1971

OPERATIONAL EXPERIENCE WITH MEDIUM SPEED DIESEL ENGINES

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The philosophy leading to the adoption of geared medium speed Diesel engines in a cargo liner and a products carrier is described and results obtained in service are presented and discussed against this background. The importance of considering a wide range of factors in addition to purely technical considerations in selecting and judging the performance of equipment is emphasized.

Details of noise trial results, piston, piston ring and liner wear, usage rates of water cooled and rotocap exhaust valves are presented. Service experience with critical items of the installation including reduction gearbox and exhaust gas blowers is also included. Comments are made on the vitally important role of all ship's personnel in exploiting innovations.

INTRODUCTION

The principal potential advantages currently claimed for medium speed Diesel engines, as opposed to slow speed engines for marine propulsion are:

a) Less weight;
b) Less space required;
c) Smaller components which are easier to manipulate and cheaper to freight;
d) Cheaper, even as multi-engine installation;
e) Reduced maintenance costs;
f) Simpler—easier to train operators;
g) Engine assembled under workshop conditions;
h) Multi-engine installation offers the advantage of inherent improved safety, reliability and ability to service an engine underway or alongside whilst remaining mobile on another;
i) Simple adaptation to unmanned operation;
j) Reduction gearbox permits selection of optimum propeller revolutions;
k) Low main component wear rates, i.e. long life.

The principal disadvantages advanced are:

a) High consumption of lubricating oil. This is particularly important when using the expensive alkaline oil necessary for operating on heavy fuel;
b) Short exhaust valve life with heavy fuel;
c) Multiplicity of cylinders, possibly increasing maintenance load;
d) Noise;
e) Need for a vulnerable gearbox;
f) Complicated torsional oscillation model with multiple engines.

MACHINERY COMPARISON

In order to arrive at balanced conclusions from operating experience regarding the suitability, reliability etc. of medium speed Diesel engines for the propulsion of ships, it is essential to consider not only the particular engine for which experience is available, but also installations as a whole, the service for which they have been used and the reasons for their selection. For example, if an installation is designed for one service but used on another for which it is not suitable, then it may be unfair to condemn it. If a particular engine is provided with badly matched or inferior auxiliaries, it could appear to be an unsuitable choice for a particular application, whereas an identical engine in a similar ship but with the right auxiliaries could appear to be an ideal choice. The selection of a medium speed engine could possibly lead to the economical introduction of other equipment the advantages of which can then be considered as consequential to the medium speed engine and so added to the latter's direct advantages in that particular application.

MEDIUM SPEED DIESEL ENGINE APPLICATION TO CARGO LINERS

Service

In 1964 the author's company Thos and Jno Brocklebank Ltd., was considering new tonnage for its liner services, which included Calcutta and Manchester in their itinerary. The ship was to lift a minimum of 10 500 tons of cargo and it had to be suitable for tramping.

Calcutta

Calcutta placed a maximum length of about 500 ft on the ship to facilitate turning in the Kidderpore lock.

Hooghly River

The Hooghly River, between the sea and Calcutta, has a draught limitation and it was specified that the ship should be able to lift a certain deadweight at 21 ft draught. At this draught the vessel would be a deadweight ship where every ton of light-weight saved was theoretically an extra ton of cargo carried. Every ton of ballast necessary to trim the ship was correspondingly a ton of cargo theoretically lost. It followed that the lighter the machinery, the less ballasting was needed and the better the cargo lift at low draught.

Manchester Ship Canal

The requirement to pass through the Manchester Ship Canal

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and round the Weaver Bend, presented a choice of certain beams associated with maximum draughts. The need for maximum deadweight at low draughts dictated that of these combinations, that with the maximum beam was the most suitable. A beam of 63 ft and maximum draught of 28 ft was therefore selected.

Prescribed Dimensions

The main ship dimensions were thus prescribed and it became the designers' aim to pack the maximum cargo deadweight and cubic into these limits without requiring excessive power, and fuel consumption, to attain the specified speed.

Competing Designs

Ship designs were prepared for slow speed Diesel engine and twin medium speed Diesel engine installations. The latter was some 280 tons lighter than the former, and the lower engine room height required to cater for the medium speed engine and its piston withdrawal enabled an additional deck to be fitted in over the engine room, compared with the slow speed installation.

Thus, from the weight and space point of view, the medium speed Diesel engine was attractive and, as will be shown later, it also had advantages in other directions. A geared twin medium speed Diesel engine installation, driving a controllable pitch propeller, was eventually adopted.

Tramping

To be attractive on the charter market for tramping, the main engines had to be suitable for heavy fuel of any composition commercially marketed in the viscosity range selected. To cater for possible financial advantages in the future, it was decided to design the installation to handle 3 500 sec. Redwood No. 1 fuel, but as a general rule 1 500 sec. fuel would be used. To facilitate purchase of lubricating oil in the cheapest market in particular for tramping, when the ship was most likely to be cubically full before reaching maximum draught, excess lubricating oil tank capacity was provided over that dictated by the liner routes for which the ship was principally being built.

Towage

Examination of ship running costs indicated that:

1) Towage costs per ship were significant and were rising annually;
2) On several occasions ships had been delayed awaiting tugs, because the latter were otherwise engaged or because of fog etc. This delay was a demurrage charge;
3) On some occasions ships had been moored alongside for days during tug labour disputes, again at a considerable cost to the company.

These observations were borne out by other owners. Costings for a 9½ ton bow thruster were carried out assuming a 30 per cent reduction in towage charges; resultant possible reductions in demurrage were not credited. The bow thruster was to be powered by an 800 hp electric motor fed from a main engine driven alternator.

Main Engine Driven Alternator

Initially, an hydraulic coupling was considered between the main engine and the gearbox and the main engine driven alternator was placed at the free end of the engine. The adoption of a plate clutch coupling enabled a smaller, faster running generator to be fitted at the main gearbox, permanently coupled to the engine.

The maintenance concept with two engines permitted the ship to enter or leave harbour on one engine, thus an alternator for the bow thruster was required coupled to each main engine. To avoid complicated control gear, a constant speed engine was obviously an advantage, and this led to the adoption of a controllable pitch propeller and the elimination of reversing gear on the main engine.

When not used for the bow thruster, the main engine driven alternator (due to fault current limiting reasons, only one at a time could be generating) was to be used for ship's services, thus reducing running time of auxiliary generators, reducing their maintenance costs and reducing cost of power generation as main engines would be using (cheaper) heavy fuel rather than the (more expensive) Diesel fuel used by the auxiliary generators.

Auxiliary Generators

With main engine alternators available for ship's services, it is possible to reduce the number of auxiliary generators fitted. In this case, this was not carried out and the cheaper first cost but higher maintenance cost high speed Diesel generators were adopted with a cargo weight and space credit to the ship design.

Maintenance

Restrictive practices in the U.K. limit the amount of work that ship's staff may carry out on their main engines in port. Where such work is necessary, owners are generally at the mercy of ship repairers whose costs are rising rapidly and who may not be as familiar with the particular machinery and installation as one would wish, considering the equity involved. Unfortunately, for cargo loading reasons, the time spent in Calcutta was somewhat lengthy. Although ship's staff could work on main engines in that port, frequent requirements to move the ship under her own power made it difficult to plan such work when only one engine was fitted. The twin medium speed engine arrangement adopted gave the facility for working on one engine with the ship mobile on the other. It was, therefore, hoped to confine all major main engine maintenance and survey work to Calcutta, using ship's staff and avoiding unnecessary high U.K. ship repairers' bills.

Again, there was a possibility of maintaining a main engine at sea if staff was available and noise was not excessive.

Unmanned Machinery Space

With the relatively simple main engine control system and the adoption of the controllable pitch propeller, it was a natural step to adopt the U.M.S. concept, so releasing erstwhile watchkeepers for day-work duties and regular maintenance at sea.

Controllable Pitch Propeller

As has been mentioned, the adoption of the controllable pitch propeller enabled the main engine reversing gear to be dispensed with. In addition, it enabled the full power of a single engine to be effectively utilized for propulsion, increased the ship's manoeuvrability, enabled main engines to be more thoroughly checked out before going to sea and put full main engine power at the disposal of the command in the astern direction. With bridge control of the c.p. propeller, there was more rapid and positive response to the command's requirements and the adoption of the U.M.S. concept was easier.

Seaworthiness

From the above it will be seen that the design concept required deliberate single engine operation of the ship. This concept would be completely vetoed if the command refused to operate the ship in this mode on the score that if two engines are fitted, it would be wrong to operate on one, as the ship would be technically unseaworthy. Indeed, there was evidence that some similar geared twin-engined installations had in the past sacrificed natural advantages because this view was taken. From the outset, therefore, it was insisted that the ship was seaworthy on one engine. No claim for increased safety or reliability was made on account of the second engine which was to be considered as a bonus as it was not fitted for these reasons.

MEDIUM SPEED DIESEL ENGINE APPLICATION TO PRODUCTS CARRIERS

Whilst the cargo liners, discussed in the previous paragraphs, were being constructed, Moss Tankers Ltd, which was managed by Thos and Jno Brocklebank, was considering the purchase of two 25 000 dwt products carriers, similar to several others under construction for an oil company. These ships, as offered, were propelled by slow speed engines with steam driven cargo pumps and anchoring and mooring windlasses.

The Moss ships were intended for operation on long-term charter or on the spot market.

Independent Products Carrier Operators' Desirable Trading Pattern

The trading pattern aimed at by independent products carrier operators is to ensure that ships are available during high freight periods and to confine repairs and maintenance, necessitating time off hire, to the low freight periods. For normal planning purposes, high freight periods tend to correspond to European winter months and low freight periods to European summer months. Naturally, time off hire must be kept to a minimum.

Slow Speed Diesel Engines' Maintenance Cycle

The majority of slow speed Diesel engines have a unit maintenance maximum cycle of 6/9000 h. As a tanker is only permitted, in exceptional circumstances, to immobilize itself alongside loading or discharging berths, its main engine maintenance/survey work is generally carried out in lay-up periods off hire. Some operators, it is true, deliberately stop their ships at sea and carry out unit maintenance whilst ostensibly remaining on hire. By this means, very long apparent cycles between unit maintenance are often reported. To do this, the ship must be chartered at a low enough speed to accept such stoppages in the calculation of average speed, without penalty. Again, it is not a procedure which it is prudent for managers to insist upon from sea staff as, should an insurance claim develop during and as a result of this deliberate immobilization, 'negligence by management' could be advanced by underwriters as a valid reason for limiting their liability. Where sea staff are obliged to stop at sea to right faulty equipment, the question of negligence does not, of course, arise, and even in those borderline cases where sea staff may have been over-enthusiastic in deliberately immobilizing ship for maintenance, and they were considered legally negligent, no question of limiting insurance liability arises.

A tanker's main engines log about 7000 h a year underway. Thus, if selection of lay-up periods is to be governed solely by the limit of slow speed Diesel main engines' reliability (6/9000 h), some of such times could fall in high freight periods. The tendency, therefore, has become to lay-up such ships for maintenance in a ship repair yard annually during the summer months, as the main engines would not run satisfactorily for two years. Moss Tankers' experience was that the length of the lay-up period was determined solely by the main engines and that for this size of ship some 17/21 days off hire were needed for the slow speed engines which they were using. For hull maintenance purposes, with modern coatings, 18/24 months between drydockings was acceptable.

Deck Maintenance

For safety reasons, in tankers steam-operated deck machinery has normally been fitted. The piping and valves necessary on deck being hot, suffer from severe corrosion from the salt water which inevitably sloshes over them when the tanker is laden.

Application of Medium Speed Diesel Engines to the Products Carrier

After discussing various alternative proposals, it was decided to:

i) Install two non-reversing constant speed medium speed Diesel engines geared to a single c.p. propeller in the same engine room space, as provided for the slow speed engine;
ii) Fit a permanently connected 1500 kw alternator to each engine;
iii) Drive the four 500 ton/h cargo pumps electrically;
iv) Drive all deck machinery hydraulically;
v) Install instrumentation etc. for U.M.S. operation.

Advantages expected from installing Twin Medium Speed Diesel Engines in Products Carriers

The following advantages were expected from the installing of twin medium speed Diesel engines in products carriers:

a) Ship always mobile alongside whilst an engine is being maintained, i.e. one engine can always be maintained in port;
b) Possibility of maintaining one engine at sea:
   Note: Tank tests showed the ship to be one knot slower with one engine when ballasted than with two engines when loaded. To take advantage of this facility, the ship must be declared at the right speed;
c) Possibility of preparing one engine for survey at sea, approaching a port, having it surveyed in port and reassembly completed at sea after leaving port;
d) Selection of drydocking period and its length, now to be governed by hull maintenance requirements, as opposed to main engines. This could enable time between periods in hand by ship repairers to be extended to 18/24 months and days under repair to reduce to the region of a week or less. Thus, in six years the comparison could be:

	Slow Speed	Medium Speed	Medium Speed
In hand ship repairers	6	3 (at 2 yrs)	4 (at 1½ yrs)
Average days/time in hand	17/21	7/10	7/10
Total days in hand	102/126	21/30	28/40

The periods quoted above are not intended to be indicative of any particular engines or the worst or best time obtainable but are entered purely to indicate a possible range and the tendency to reduce time in hand and thus cost. Time saved would be available for freighting.

In practice, it was expected that operators would aim for 18 to 24 months between drydocking periods, selecting the actual time to suit current freight rate. It was also visualized that under certain high freight circumstances, the drydocking period in hand might possibly be reduced to 3 days;

e) Reduced maintenance on deck, due to the elimination of steam pipes and valves;
f) Electro-hydraulic deck machinery and electric cargo pumps reduce boiler size and maintenance load and eliminate the difficulty of cleaning and maintaining boiler in port;
g) Use of shaft alternator on ship's services reduces operating time of auxiliary alternators, so reducing their maintenance costs. Power generation cost reduced as heavy fuel used in main engine, as opposed to Diesel fuel in auxiliary alternators;
h) More cargo deadweight available, this being particularly important where there is a draught restriction.
   Notes: A) As the engine room space was, in this case by agreement with the shipyard, kept identical with that needed for a slow speed engine in the same hull, no credit for the intrinsic decreased space requirement of the medium speed installation could be obtained;
   B) The major oil companies have excess tonnage on charter at some time or other and it is generally possible to release their own ships for maintenance to suit their overhaul schedules, i.e. they do not have the same interest in extending time between drydocking periods, as they can already do so to as great an amount as the engine will tolerate. Hence, what is right for the oil company ships, is not necessarily right for the independents' ships.

THE PACKAGES

It will be seen from the above that the most important reasons for fitting medium speed Diesel engines were not the same in the cargo liner as in the products carrier.

Cargo Liner

Maximum cargo deadweight at low draught—more freight; maximum cubic—more freight; more self-maintenance—less reliance on repair yards—reduced cost.

Products Carrier

Ability to maintain/survey an engine whilst loading or discharging—reduced cost; time between drydocking periods no longer governed by main engine maintenance and therefore increased. Time with ship repairer decreased—more freight income and reduced costs.

EQUIPMENT SELECTION

The selection of equipment for a ship should be a balanced judgment of a number of factors, including:

1) Technical appreciation
2) Value engineering
   i) price
   ii) life
   iii) maintenance costs
   iv) reliability required
3) Manufacturer
   i) reputation (quality, reliability, delivery promises, back-up service etc)
   ii) financial position
   iii) capacity and order book
   iv) availability of spare parts
4) Operating performance of equipment, as reported by owner and other organizations
5) Shipyard preference
   i) experience with equipment
   ii) experience with equipment manufacturers' design, installation and commissioning personnel
   iii) credit facilities available
   iv) licensing situation
   v) special arrangements with manufacturers
6) What the selecting organization can afford to pay
   i) monetarily for equipment
   ii) riskwise for equipment
   iii) for qualifications of personnel to maintain and operate equipment
   iv) any special shore facilities necessary to support equipment

It is clear that all technical decisions should also be commercial decisions. It is also clear that proper decisions on such matters as the selection of equipment are not simple questions of first cost, what other people are doing and so on, but also demand an appreciation of a wide range of other factors complementary to an intelligent understanding of the technical characteristics involved.

It is important to be careful (and this is not always a straightforward matter) in judging the actual performance of a piece of equipment to review many of the factors considered when choosing it, and also to isolate it from the deficiencies of its ancilliary equipment, any deficiencies arising from its application in a particular installation and any deficiencies arising from unsuitable service for which the installation is used. For example, a badly selected lubricating oil pump could result in high maintenance costs and a medium speed Diesel engine spending more time out of service than operating—this is an ancillary equipment deficiency, not a medium speed engine deficiency. If medium speed engines are installed in an unsuitable power range or next door to and unisolated noise-wise from passengers, these are installation deficiencies, not medium speed engine deficiencies. If a ship with a medium speed engine installation is designed for a relatively short sea passage with fuel within manufacturers' preferred limits and lubricating oil capacity chosen for that specific service and the ship is then scheduled for worldwide trading, it is a deficiency in the choice of the new service which results in difficulties with fuels outside manufacturers' limits and apparent shortage of lubricating oil capacity, rather than a deficiency in the medium speed Diesel engine.

These comments are intended as a warning to analyse all reported deficiencies in depth and not to jump to conclusions.

MEDIUM SPEED DIESEL ENGINES SELECTED

After examining the rival claims of suitable medium speed Diesel engines designed by M.A.N., Sulzer, Ruston, Mirrlees, B and W and Pielstick, it was decided to use the latter in the author's company's liners and, unless otherwise stated in the following paragraphs, references to medium speed Diesel engines are to be construed as applying to Pielstick PC2V engines. The layout adopted in all four ships is shown in Fig. 1.

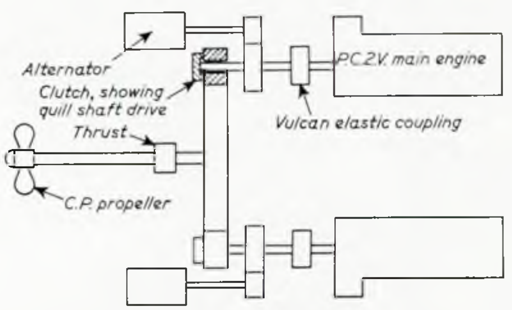


FIG. 1—*Layout adopted in all four ships*

ENGINE AND COMPONENT COMMENTS

Power and rev/min chosen

The following combinations of power and rev/min were adopted:

	Max. Continuous Rating	Max. Service Rating
rev/min	450	450
b.h.p./cylinder	420	420

The rev/min decided upon was less than the maximum for which the engine was advertised, to reduce the strain on the (constant speed) running gear, particularly the valve gear, which at that time had recently been modified to eliminate weaknesses found with earlier engines. From the outset, separate lubrication of the valve gear from the rest of the main engine was adopted, eliminating potential contamination of the latter system by water and fuel.

There has been no adverse experience attributable to choice of power and rev/min and the same criteria is being used in three further products carriers currently under construction. For a ro-ro ship in which PC2V engines are being fitted, the currently advertised higher rev/min of 520 associated with a 500 bhp/cylinder, has been accepted, extensive testing at S.E.M.T. works in France and results obtained in other ships at sea having convinced the company that this is acceptable.

Noise

Noise has always been one of the disadvantages advanced against medium speed Diesel engines. From the moral point of view, the author was influenced by a comprehensive Scandinavian report which showed that the worst sort of noise from the medical point of view was a continuous staccato noise from such equipment as riveting hammers, relief valves lifting etc. Provided exposure was of relatively short duration, followed by a recovery period, it was not expected from information available that the sort of noise in a medium speed Diesel engine room would harm watchkeepers. Nevertheless, for comfort and efficiency, soundproofed machinery control rooms and workshops were fitted in the ships. In addition, ear muffs are available.

Whilst it was hoped to be able to maintain one engine with the other operating, until this was done it was always appreciated that additional portable sound isolating equipment might be necessary, possibly in the form of a quilt with interleafed lead foil hanging down between the two engines. Until this maintenance ability could be proved, no credit for it was advanced in costing arguments. In the event, it has been found entirely feasible, and not particularly uncomfortable to carry out the maintenance as hoped for, and the demand is for improved tools rather than noise reduction.

Representative engine room noise readings recorded in the cargo liner are shown in Fig. 2.

Instrumentation

Generally the instrumentation functioned reasonably well, but increased reliability would be welcomed. Choice of instrumentation was not consciously significantly restricted for financial reasons, and it was felt that the major deficiencies were due to insufficient marinization of equipment designed for and satisfactory on land.

Polycomp recorders are fitted in all four ships to record some 17 important installation parameters. In addition, similar recorders are fitted in the cargo ships to log exhaust gas temperatures. These instruments initially gave trouble, until it was found that they were temperature sensitive. Once adequate ventilation was provided, they have proved very satisfactory and very popular and useful to ship's personnel as they show parameter trends on one sheet and enable abnormalities to be picked out very easily.

In the cargo ship the instrumentation and alarms were operated by a 24 V nickel cadmium battery floating across the mains. When the tankers were designed, there was evidence that perhaps this arrangement was over-cautious in concept. The instrumentation and alarms were, therefore, taken from the emergency switchboard which, in the event of main power

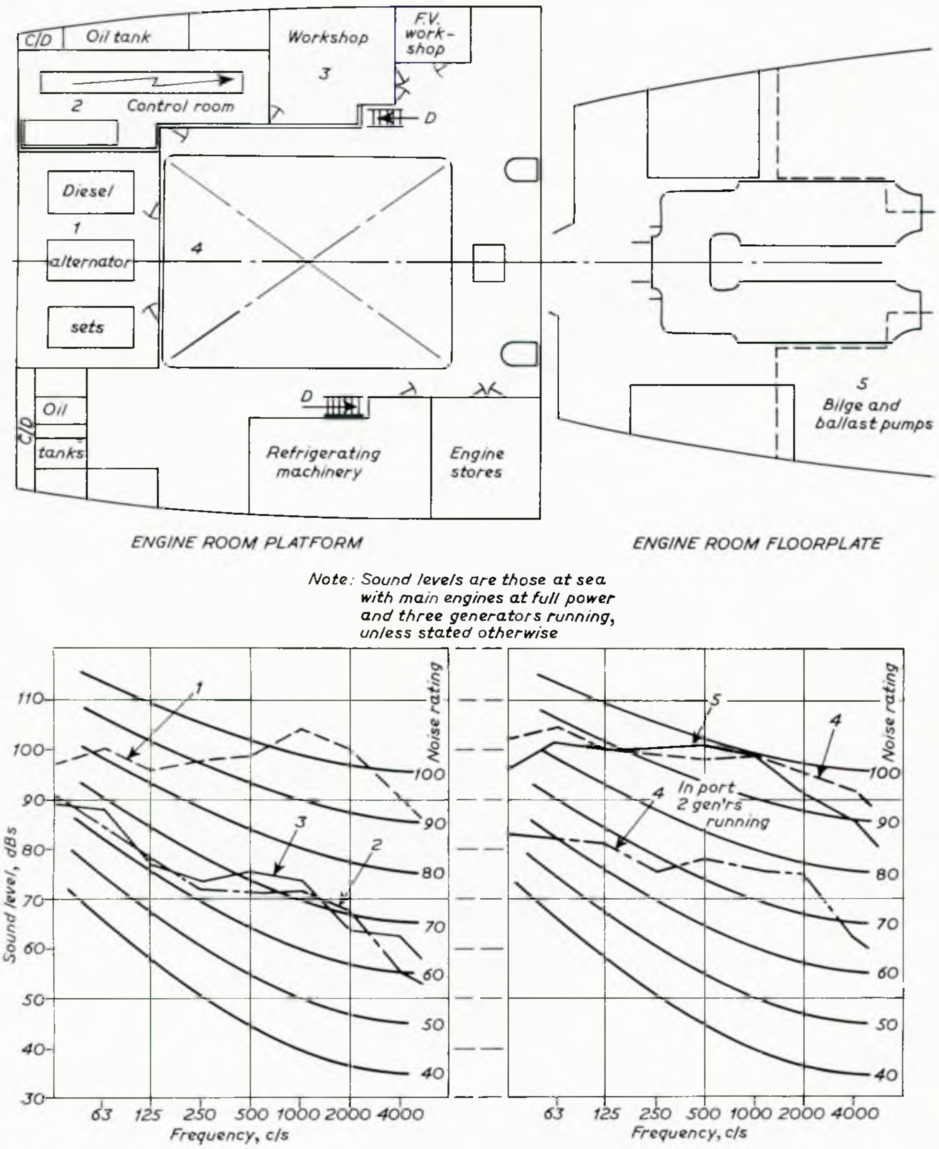


FIG. 2—*Machinery space sound level*

One of the tendencies of the more realistic rules for deduction of engine room spaces for tonnage calculation was to make the most efficient use of the engine room. In the case of the ships being considered, this moved the funnel closer to the bridge than had been customary. As a result, rather high exhaust noise levels were experienced on the wings of the bridge, which officers have reported as disconcerting until they adjust themselves to them. These levels are well below potential damage values, but make verbal communication more difficult. Noise levels in the accommodation were all below the maximum specified.

failure, would be fed from the emergency alternator. This system worked satisfactorily until water leaked from a fracture in a ventilation trunking in the m.c.r. onto switchgear during U.M.S. operation, and caused a main power failure.

Unfortunately the emergency alternator did not start and so there was no power available to alarm the duty engineer. This is being catered for by providing a battery alarm which operates if power is shut off from the instrumentation and alarms.

Control Gear

Experience has shown that the control switches used were insufficiently robust and in several cases their siting has proved unsatisfactory, making them liable to be soaked in water or oil or damaged by vibration. Two cases have been recorded of control gear sticking, leading to engines not shutting down as intended on failure of lubricating oil pressure, and on overspeed. In the first case bearing damage resulted and in the second case engine driven alternator rotor windings were damaged.

Control Air

Considerable use is made of pneumatic controls and, like several other users of such equipment, it has been found that insufficient practical margins for leakage and for equipment not operating at optimum performance was allowed. As a result, the splash lubricated intermittently rated garage type of compressors have had to run longer and more frequently than their design caters for and have required excessive maintenance. In current new constructions an additional compressor and more robust models are being provided.

The system adopted, and which has worked satisfactorily, is shown in Fig. 3.

From the installation point of view, it is obviously simpler and easier to control to a constant cooler outlet temperature, and the engine designers now advise that the jacket water circulation rate is such that with constant inlet temperature, the temperature swing from no load to full load is only a few degrees, and this can be tolerated by the evaporator. The effect on liner wear is considered to be immeasurable, and in any case with high alkaline lubricating oil, acid should be neutralized as it forms.

Lubricating Oil Pumps

The original concept for the cargo liner was developed with the U.K. engine licensees on the basis of engine driven lubricating oil pumps. The order for the ships was eventually placed with a Swedish shipyard and engine builder who had been responsible for a large number of medium speed Diesel installations of this type, particularly in Scandinavian ships. All these installations had incorporated electrically driven lubricating oil pumps and, in fact, the relevant engine castings were not provided with the necessary bossing to fit engine driven pumps. In order to avoid an unnecessary 'extra' and, bowing to experience, the electrically driven system was adopted with the L.O. pumps fed from essential services. The argument was that lack of electrical power or reduction of lubricating oil pressure would shut down an engine. (Some authorities do not like automatic shut-down of machinery on failure of L.O. pressure, as, they suggest, that by not so doing, possible hazard to the ship could be avoided. The author considers this to be an unrealistic attitude, particularly with medium and above speed Diesel engines where the time available between failure of pressure and onset of engine damage is too short to be significant in ship handling, as such machinery operates with high lubricating oil temperature and small clearances.) In practice, there have been a series of incidents in ships of different flags and owners involving loss of lubricating oil pressure from electrically driven pumps and failure of associated engine to shut down leading to bearing and crankshaft damage. These have ranged from shutting down the pump to the wrong engine (with no automatic shut-down) to failure of control gear to shut an engine down. The writer believes that this is enough to demand engine driven primary lubricating oil pumps as a prudent precaution in future installations of this nature, especially where non-reversing engines are used.

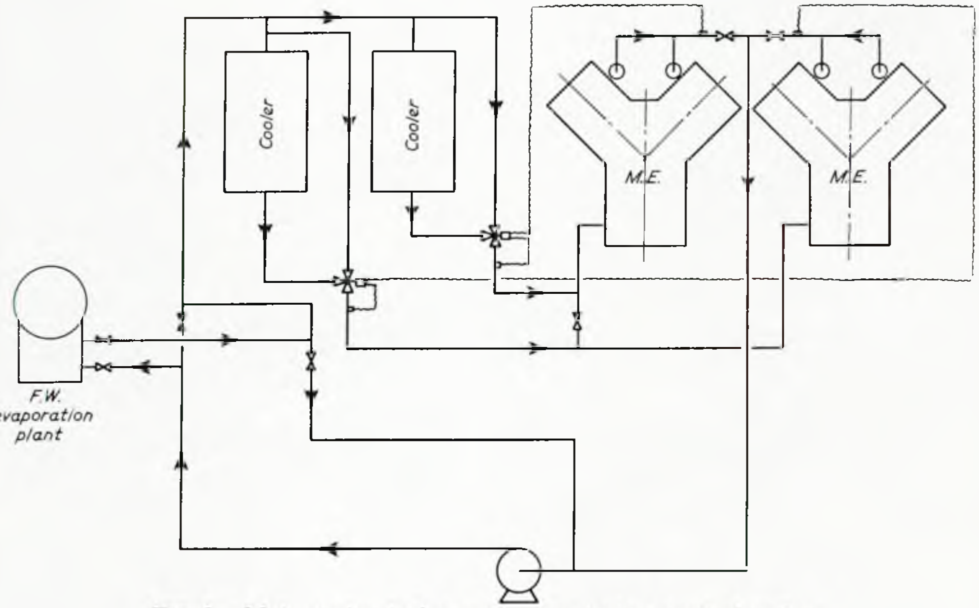


FIG. 3—*Main engine jacket water temperature control system*

Temperature Controllers

For U.M.S. operation, to enable alarm parameters to be set at reasonable values and avoid spurious alarms and shut downs, more accurate control of fluid temperatures, than is strictly necessary for safe operation of the plant, is required. Thus, all temperature controllers were specified to be of the servo-operated type. With two engines and a common cooling water system, the provision of cascade control of jacket water temperature, governed by each engine's outlet temperature, becomes complicated. The author, however, considered it important:

i) To maintain high jacket water temperatures at all loads to reduce liner wear;
ii) To maintain as constant a temperature as possible to jacket water supply to evaporators and so not only provide them with stable running conditions but also permit output to remain constant for all loads.

Bearings

The main and bottom end are steel backed, thin walled bearings, lined with copper lead, overcoated with a lead-tin flash. During examination of bearings after shop trials, there were indications of stray electrical current corrosion on their backs. This was considered due to the electric arc welding of certain securing brackets onto the engine frame after assembly of the bearings. During a recent bearing examination in one ship, pitting of the connecting rod and associated bearing caps was found. This was initially thought to be also due to stray currents, but was finally diagnosed as fretting corrosion when securing bolt tightening measurements were taken, indicating that they had not been properly torqued. Improved instructions and tools are being provided to ensure proper tightening of bolts.

No evidence of significant wear of main bearings has been found. Some of the lead-tin flash is often missing and bearings are only replaced when this reaches over one-third of the area. On one occasion of electrical power, and thus lubricating oil pressure failure, the engine unloaded but continued running for a short period. Damage surprisingly occurred to only one bottom end and its associated crankpin which was scored. After examination of the other bearings, lapping of the damaged pin and fitting of an undersize bearing, the engine again operated satisfactorily.

In one ship, routine examination of bottom ends showed minor wiping in the tin flash area, indicating that the engine had at some time run with insufficient lubricating oil. There had been no indication during the running of the engine, prior to the examination, that anything was amiss.

It is possible to fit the top half of a bottom end shell in the bottom half position. If this occurs, piston cooling oil would be shut off and piston seizure would take place.

The company's experience on bearings, has on the whole been unspectacular and indicates that bearings have possibly more latitude for tolerating lubricating oil supply interruption than one might have expected.

Injector Pipes

Whilst the first cargo liner was being constructed, another PC2V engined ship, recently completed by the same builder, suffered from a rash of injector pipe failures. These failures were traced to a batch of pipes requisitioned from a new supplier by the firm's purchasing department to a simple material specification. This supplier's product had not been tested on an engine.

Injector pipe failures were normally found to occur at the junction with their end connexions. Careful examination revealed decarbonized areas in this vicinity in the pipe bore. It is now the practice to machine inside the pipe to remove this affected area. This, coupled with elimination of distortion by restricting torqueing of fittings to 9 kgm has reduced pipe failure to small proportions.

The author considers that these pipes are a potential danger area, as far as unattended operation of the engines are concerned, and that the enveloping piping should be connected to a collecting tank fitted with an alarm to sound should a pipe fail. A system of this sort should, the author considers, be fitted as a matter of course for U.M.S. operation rather than as an exception.

Exhaust Gas Boiler

To avoid fitting an exhaust gas boiler in the uptake of each engine, and to avoid tube plate differential temperatures with one engine only operating, a Sunrod extended surface/header type of boiler was fitted and has proved very satisfactory in practice.

Exhaust Gas Blowers

Brown-Boveri blowers were fitted in the cargo ships and Napier blowers in the tankers.

The Napier blowers are at the top end of a frame size, whereas the Brown-Boveri blowers are at the bottom. As a result, the Napier blowers are cheaper and were, in fact, of higher efficiency when tested by S.E.M.T.

In order to restrict the blower lubricating oil consumption to a satisfactory rate for U.M.S. operation, and to avoid excessive oil vapour escaping and triggering off engine room smoke detectors, a carbon seal was used in the Napier blower. The tankers use their main engines at very low load when generating for pumping cargo. Whilst all S.E.M.T. and Lindholmen investigations have indicated that engines should run under these conditions without significant production of carbon, experience has shown that the exhaust gas nozzle rings of the Napier blowers gradually become choked. There is some reason to suppose that this gas side fouling is more extensive when operating on Diesel fuel. This is explained, possibly because heavy fuel injector nozzles are fitted and these have larger holes than the Diesel oil version.

The result of this fouling is to reduce engine output and thus maximum available ship speed. Consequently, in order to obtain maximum power on the propeller, the tankers do not use their shaft generators under these conditions. The auxiliary alternators are, therefore, being operated more than was planned and their maintenance has correspondingly increased. The lowered maximum ship speed reduces the opportunity for stopping an engine at sea for maintenance. To date it has been necessary to strip a blower to clean it and, apart from this being laborious and time-consuming carbon seals have generally been broken. At the same time there have been difficulties in obtaining replacement seals.

Water washing is now being fitted to the gas and air sides of these blowers. By this means, it is expected to avoid gas and such air side fouling as takes place, maintain blower performance, obviate the necessity for stripping the blower and so avoid damaging the rare carbon seals, maintain maximum engine power and maximum ship speed, reduce running of auxiliary alternators and carry out more maintenance at sea. The problems with this one component have thus tended to negate several of the potential advantages of the medium speed engine in this application.

At present one spare Napier blower is carried per ship. It follows that a ship seldom has two clean blowers operating. It is intended to carry two replacement blowers in future in case the water washing is not as successful as anticipated and it is still necessary to strip blowers for cleaning. By this means, it will be possible to plan so as to have two clean blowers fitted to one engine at about the same time.

Liners

There have been a number of cases of cylinder liners cracking below the upper flange and dropping down. Consequential damage has occurred to the opposite liner and both pistons.

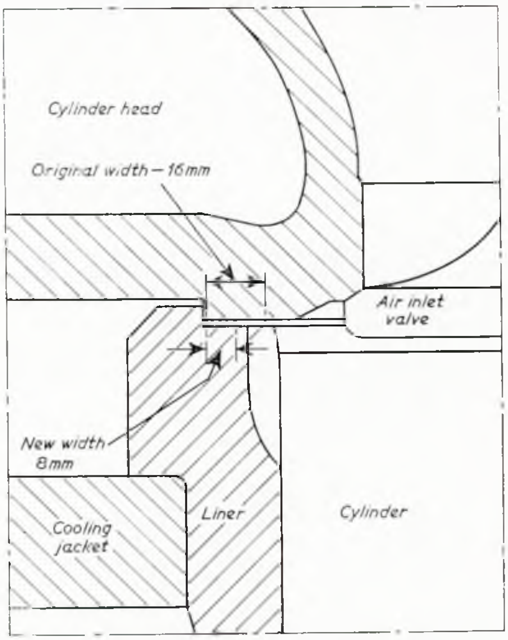


FIG. 4—*Fine concentric groove machined in the cylinder cover opposite the new gasket*

Investigation showed that the cylinder head/liner gasket was of the original 16 mm wide copper design and that over-tightening of the bolts would stress the liner flange, ultimately causing fracture. This was most likely to have occurred during tightening in an effort (seldom, if ever, successful) to stop a leak.

The gaskets are being replaced with the later 8 mm wide brass or soft iron type, and special instructions are being issued, drawing attention to the importance of using the correct tightening procedure.

To assist in making a tight joint, a fine concentric groove may be machined in the cylinder cover opposite the new gasket.

Linear wear measurements have been generally in accordance with manufacturers' claims and some typical readings are given in Table 1.

TABLE I—TYPICAL CYLINDER LINER WEAR MEASUREMENTS IN 'LUSTROUS'

Cylinder No.	Athwart Ship 1/100mm/1000 h	Longitudinal 1/100mm/1000 h
No. 6 cylinder port engine	0	0
	0·8	1·1
	0·7	1·6
	0·4	1·2
	0	0
	0	·01
	0·3	0·7
No. 12 cylinder port engine	0·6	0·4
	1·4	2·5
	0·9	1·0
	0·9	1·0
	1·0	0·7
	1·8	2·3
	0·6	0·5
No. 3 cylinder starboard engine	0·9	0·9
	0·8	0·7
	0·7	0·5
	0·4	0·6
	0·5	0·4
	0·3	0·3
	0·3	0·3
No. 9 cylinder starboard engine	0·9	0·8
	0·8	0·7
	0·7	0·5
	0·4	0·4
	0·3	0·3
	0·3	0·3
	0·3	0·3

Gearbox

One of the disadvantages advanced against a medium speed installation was the need to incorporate a vulnerable gearbox. This criticism, however, caused little concern to the medium speed engine protagonists as, for years, gearboxes had successfully been fitted to steam turbine machinery and gearing design, and manufacturers were considered to be very reliable. Unfortunately, in applying value engineering concepts, i.e. in endeavouring, in the interests of cost, not to over-design the gearbox, unexpected difficulties have been encountered.

Originally it was intended to fit hydraulic couplings between each engine and its pinion, but following successful in service reports of torsional oscillation treatment by elastic couplings and of friction clutches, it was decided to fit this combination and so avoid the inherent hydraulic coupling losses.

A single reduction single helical gearbox of Renk design, with an integral thrust block aft of the gearbox, sintered bronze plate clutches with a permanently connected shaft alternator to each engine through a speed increasing gear within the main gearbox was finally selected for both types of ships.

Whilst the gearboxes were being manufactured for the cargo ships, another owner, with machinery right aft but otherwise with a generally similar installation layout, experienced severe axial vibrations on trials. The immediate solution in that ship was to install a separate thrust block, and seatings for such were designed into the ships of the author's company as a prudent precaution should subsequent investigation show that the same remedy ought to be adopted.

Various authorities, including Lloyd's Register research department, were consulted, and the following action was agreed and implemented in the four ships:

a) The gear case feet and its bolting were modified in way of the integral thrust block;
b) The ship structure was carefully examined and strengthened:
   i) in way of the gearbox seating, taking particular care to compensate adequately for the discontinuity caused by the main gearwheel;
   ii) in way of, and in order if necessary to fit, a separate thrust block;
c) A robust structure was built up on the new potential thrust block seating and connexions were made from it to the centre line of the integral thrust block to assist in resisting the tipping moment on the latter. These connexions became known as the 'clutching hands'.

Instrumented sea trials of the ships were carried out with and without the 'clutching hands' connected with no apparent or measurable difference being detected. It was, therefore, concluded that the measures a) and b) i) above had remedied the deficiency brought to light in the earlier ship.

During the course of discussions on this matter, preferences were declared by some for integral thrust blocks to be positioned forward of the gearbox. The author favours the location aft of the gearbox, as it is possible to transmit the thrust by tension over a longer length of the ship structure in this position—an arrangement less conducive to axial vibration generation than that where thrust is transmitted by compression, or over a shorter length of ship structure, as occurs with thrust block forward due to position of the main wheel.

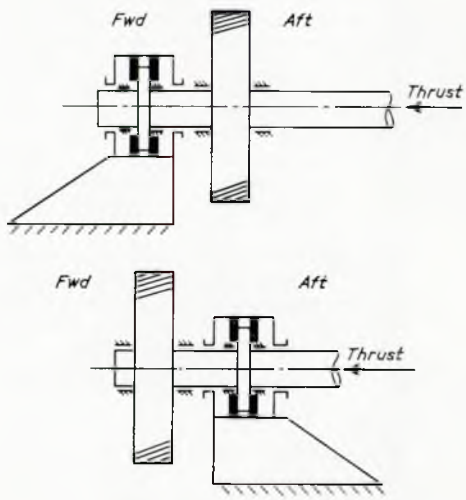


FIG. 5—*Positioning of integral thrust blocks*

The elimination of unnecessary vibration from the engine room is important, not only from the habitability point of view, but also to protect instruments, eliminate piping fatigue fractures, avoid spurious shut-downs of machinery and so on.

Having found on trials that the products carrier engine room was satisfactory as regards vibration, 'value engineering' precluded spending additional money on methods to improve this characteristic for similar ships currently under construction. However, the author was interested in, and seriously considered, the introduction of the Lohmann and Stolterfoht 'five wheel' gearbox for these ships. This gearbox has a small main wheel and avoids the large discontinuity in the ship structure, resulting from a large main wheel. Apart from the increased cost, this gearbox has the disadvantage that intermediate wheel teeth are subjected to reverse bending.

Originally it was specified that the gearboxes for the author's company should be fitted with plain bearings throughout. Although their normal and preferred design used roller bearings for the pinions, Renk agreed to meet these requirements. However, when it was decided to fit the internal integral plate clutch, it was found that this was dependent, as designed, upon using roller bearings. To fit plain bearings would involve additional expense, a complete re-design with the clutch fitted externally and acceptance of a prototype which had not even been rig-tested. On balance, therefore, the roller bearing internal clutch was accepted. Having accepted roller bearings in this position, it was felt illogical to insist on plain bearings elsewhere in the face of the designers' preference for roller bearings. So roller bearings were accepted for the pinions.

Despite every care being taken in the alignment of engines and gearbox, it had been found in other ships that pinion and gearwheel markings indicated misalignment when operating. This was thought to be due to the high temperature of the lubricating oil and proximity and position of the lubricating oil drain tank, causing relative movement of gearbox and engine by differential expansion. Tests were carried out in the shipyard with hot drain tanks, and alignment adjusted as a result. Despite these detailed precautions, the pinion/gearwheel marking was still not entirely satisfactory.

As non-adjustable pinion roller bearings had been accepted, bearing correction to improve the marking was not possible, and the only practical method available was to twist the gearbox fractionally by chocking. This chocking was carried out to designers' instructions and satisfactory markings obtained.

At about this time several cracked main wheel rims were reported in other owners' gearboxes of this type. A variety of explanations were put forward for this damage, ranging from insufficient tooth clearance, pinion teeth too thick and under influence of hot oil on one side of pinion when stopped, differential expansion caused pinion to take up an oval shape and jam into wheel teeth when clutch was engaged. In conjunction with the gearbox designers, each theory put forward was followed up. During sea trials of one of the products carriers, strain gauge readings were taken on the main wheel rim and correlated with its passage past each pinion.

The Renk main wheel rim is shrunk onto a spider and the first evidence of rim cracking had generally been shown by relative movement between the two. Ships were instructed to inspect their witness marks regularly, together with tooth marking and visual tooth inspection.

Late last year information was received that several cracked rims had recently been reported in ships which had operated satisfactorily for over two years. This was now considered to be due to a tumbling action of the pinion when the clutch was engaged, causing uneven stressing of wheel teeth. This uneven stressing, in conjunction with misalignment, would produce overstressing and ultimate fatigue failure of teeth. Furthermore, the tumbling action was now known to be able to cause marking of the teeth ostensibly indicative of good alignment, unless the examiner was aware of and looked for the distinctive marking pattern associated with the tumbling defect. As tumbling was unknown when the gearboxes were originally chocked on the basis of tooth marking, gearbox alignment was suspect.

Magnetic crack detection of the wheel teeth of the four ships was ordered and a revised clutch drill introduced to reduce the tumbling effect.

Unfortunately, one of the cargo ships was found to have started cracks in its main wheel teeth, and a new main wheel had to be fitted. Other modifications are gradually being introduced into the clutch design, which will avoid the necessity for the new clutch drill. The gearboxes of the other three ships were re-chocked according to designers' instructions.

Lohmann and Stolterfoht gearboxes with friction cone clutches are being fitted in the three products carriers currently under construction and a Stal-Laval gearbox in the ro-ro ship.

With variable speed propellers, it is generally possible to eliminate the worst propeller excited hull vibrations by a relatively small change in shaft rev/min. With a constant speed propeller installation and shaft alternator, it would be disastrous to find that the hull critical corresponded with the selected propeller rev/min. It is a wise precaution, therefore, to estimate the hull critical by the best means available prior to selecting the propeller rev/min. In this case, B.S.R.A. methods of carrying out this calculation were employed. During trials of the cargo ships, the propeller speed was varied and the calculated and actual critical rev/min were found to be very close.

The gearbox designers approved the use of engine lubricating oil in their equipment and this has proved satisfactory.

Shaft Generators

In very rough weather some propeller racing inevitably occurs. It was not considered worth the cost or complication of fitting a sophisticated frequency control system to the shaft alternators for this comparatively infrequently expected situation. Under such conditions, the auxiliary alternators are used as a matter of course.

Lubricating Oil Consumption

The lubricating oil consumption of all four ships has been found to be excessive, running in some instances at over twice the figures claimed by the engine manufacturers, which are in themselves high, compared with slow speed engine consumption. As the medium speed engine uses an alkaline oil, which is much more expensive than that used in the slow speed engine, excessive consumption costs more than consideration of quantities alone indicates.

After eliminating excessive losses of lubricating oil through maloperation of lubricating oil separator and filter automatic cleaning, it has been concluded that high consumption is due to:

1) *Excessive Leakage at Gudgeon Pin Cap*
   Examination of gudgeon pin caps has shown that several of them could be turned. The 'O' ring seal which seals the gudgeon pin from the piston has been found to take up a permanent set and it was found that two different size (cross-section) 'O' rings have been provided as spares. New Viton rings of the correct size are being gradually fitted.
2) *Ineffective Scraping of Oil by Scraper Rings*
   Although no abnormal piston ring or groove wear has been reported, S.E.M.T. has now recommended the replacement of all rings with a new set and the blocking of the top row of drain holes in the piston. In the latest design of piston, the bottom scraper ring has been moved to above the gudgeon pin. In this piston tightness of gudgeon pin cap is no longer of the same importance.
   Typical piston ring and groove measurements from an engine in *Mahsud* after 12 700 h are recorded in Fig. 7.
   Before fitting new rings in an engine, it is currently the practice to roughen or deglaze liners using suitable honing equipment. Consideration is currently being given to the use in the near future of an approved running-in compound in the fuel in place of this roughening operation. The effect of using this compound on correctly bedded rings and liners is not expected to be deleterious.
3) *The Constant Speed Engine*
   It is considered that lubricating oil consumption is largely a function of engine rev/min, so that with this constant speed engine the reduction in consumption with power will not be obtained, that would be expected with a variable speed engine where power and rev/min decrease together.

Exhaust Valves

As these cargo ships were specified to be capable of sailing

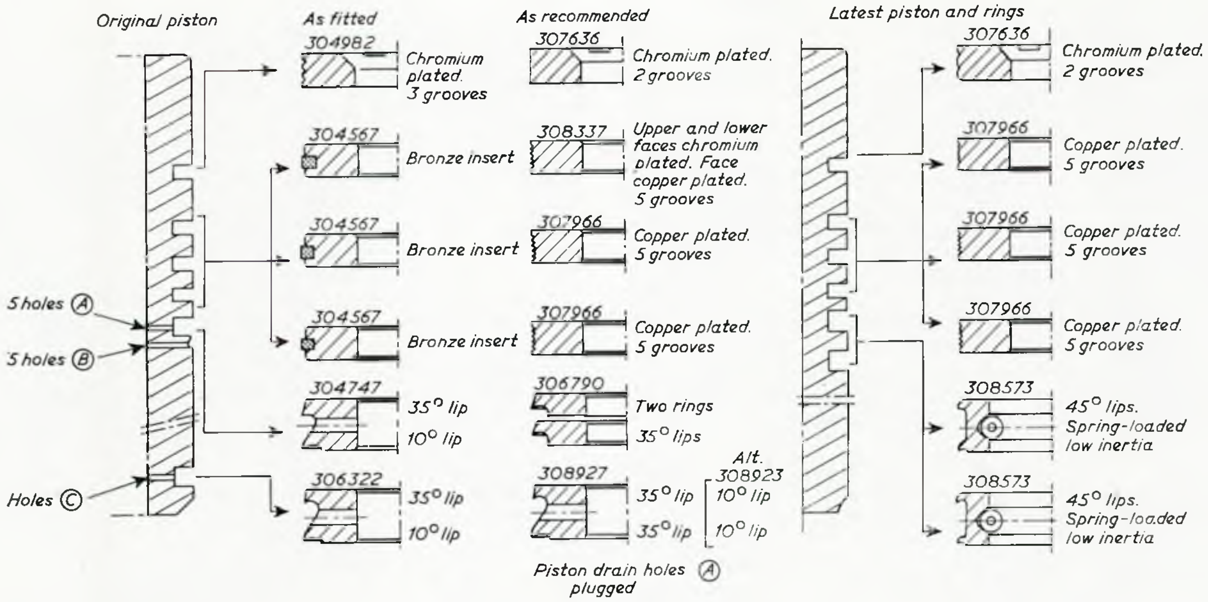
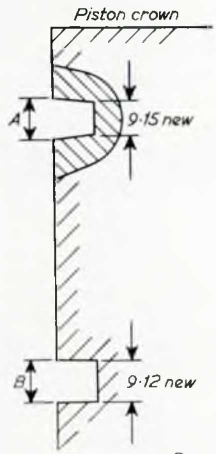


FIG. 6—*The latest design of piston*

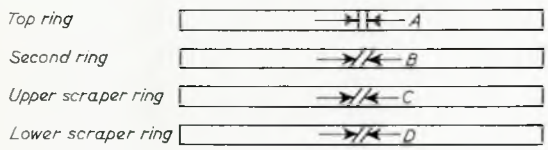


First groove—wear

Cyl. No	A	Average wear, mm	Wear 1/100/1000h
2	9·45	0·3	2·35
9	9·43	0·28	2·20

Second groove—wear

B	Average wear, mm	Wear 1/100/1000h
9·50	0·38	3·0
9·41	0·29	2·28



Cylinder No. 2

	Clearance, mm	Total radial wear, mm	Wear 1/100/1000h
A	4·4	0·38	3·0
B	3·4	0·36	2·8
C	3·2	0·31	2·4
D	2·4	0·135	1·05

Cylinder No. 9

Clearance, mm	Total radial wear, mm	Wear 1/100/1000h
3·0	0·16	1·25
2·7	0·20	1·58
2·7	0·20	1·58
2·3	0·11	0·85

FIG. 7—*Ring and groove wear*—Mahsud—*total running hours 12 700*

worldwide, it could not be assumed that the fuel with which they were supplied would always have a vanadium content of less than 150 ppm, as preferred by the engine builders. It was, therefore, decided to fit water-cooled exhaust valves to both ships.

Before the second ship was completed, the engine builders recommended a change to solid rotocap exhaust valves, even though fuels with vanadium in excess of 150 ppm might be used. In the event, the four ships were equipped as follows:

Cargo liners	*Maihar*	water-cooled (WC)
	Mahsud	rotocap (RC)
Products Carriers	*Lustrous*	water-cooled (WC)
	Luminous	rotocap (RC)

Water-cooled exhaust valves were reputed to have an unlimited valve face life, but the cooled spindles condensed sulphuric acid in way of the bottom valve guide. This acid penetrated the molybdenum plating and it flaked off. If valves were not changed, valve guides were damaged and spindle wear became rapidly worse.

At that time it was not known what spindle life could be expected, and in any case it was confidently thought that it was only a matter of months before a satisfactory spindle reclamation method was discovered and a permanent spindle coating developed.

Early in the life of *Maihar* (1000 h) some 12 WC valves in the starboard engine were found to have fractured faces. In a corresponding period in *Mahsud*, only two RC valves had been renewed. It was therefore concluded that *Maihar* should probably be converted to rotocap. Before this could be done, however, the engine builders advanced the theory that the cracked WC valves came from a rogue batch. In addition, a spate of renewals of RC valves were encountered in *Mahsud*.

By this time results were coming in from *Lustrous* and *Luminous*, indicating about the same life for each type of valve, approximately 1900 h, the WC being replaced because of spindle corrosion and the RC because of face burning.

Japanese sources suggested that WC valve stems could be reclaimed by spraying (£19/£22 per valve), and initially this appeared to be successful.

It was decided to convert *Mahsud* to WC and this was carried out after a year's operation.

A graphical presentation of valve failures plotted against running hours is shown in Fig. 8.

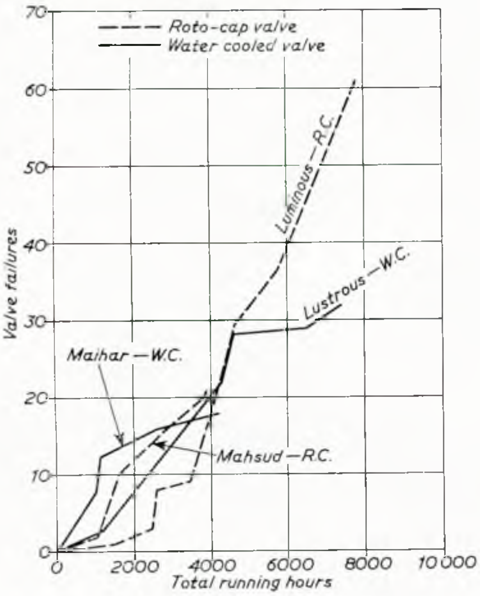


FIG. 8—*Valve failures reported—all causes*

Thinking at this time was coloured by the fact that at no time had a WC engine been obliged to stop because of valve failure, whereas this was frequently reported for a rotocap engine. In addition, it appeared that WC valves could be reclaimed, whereas, unless removed and refaced in time, burnt RC valves had to be renewed; thus, once a basic set of WC valves had been collected, reclamation should be adequate and their replacement rate by new valves should drop off.

Attempts to draw conclusions from information as it came in were confused by:

i) Different types of fuel being bunkered (*Mahsud* (RC)—chief engineers reported increased valve trouble after leaving Caribbean);
ii) On one occasion spare solid valves inadvertently sent to *Lustrous*, which then had to re-install previously rejected WC valves. This resulted in guide damage;
iii) Whilst it appeared that the cargo liner WC spindles could be reclaimed at one of its terminal ports in Japan, reliance on reclamation for the products carriers, which never put in at a port where reclamation could be effected, was not possible;
iv) Failure of replacement RC valves at random and premature intervals: considered, after extensive investigation as being due to excessive guide bush clearances. Instructions issued to ensure clearance within limits when replacing valves;
v) Over a period it became evident that the WC reclamation process was unsatisfactory.

By May 1970, after two years' experience with *Maihar*, the current picture became clearer.

TABLE II—EXPERIENCE WITH THE WC AND RC VALVES

	WC	RC
Affected by fuel quality	no	yes
Valve face deterioration	negligible	controllable
Valve stem corrosion	yes	negligible
Average interval between overhaul	1900 h	1900 h
Availability replacements	4/6 months	immediate worldwide
Cost of replacement	£125	£11/£25
Can be reclaimed	unsatisfactory	sometimes

TABLE III—COSTS WITH THE WC AND RC VALVES

	Maihar	*Lustrous*	*Luminous*
No. of cylinders	2×14	2×12	2×12
Hours	2×8 500	2×14 000	2×10 900
Type of valves	WC	WC	RC
Cost replacements (excludes labour)	£11 502	£13 631	£2 215*
Cost/h/cylinder	2·4 p	2·0 p	0·42 p

*calculated at £21 per valve

The cost of converting a ship from WC to RC ranged between £5000 and £10 000, depending upon the source of the latter. On this basis, and with no sign of an improved WC valve, it was decided that it was no longer economically sound to persevere with WC exhaust valves, and it was decided to standardize on RC for all four ships. One engine in *Maihar* was converted to RC in June 1970 and a fuel additive treatment started. Since then, valve failures have been limited to ten and there are indications that average time between overhauls may be increased to 3000 h if the same fuelling pattern is maintained.

In the meantime, S.E.M.T. has succeeded in developing a WC valve with a spindle flame plated with tungsten carbide by Union Carbide. It is claimed that this valve will be satisfactory for 6000 h between overhauls. Ten of these valves are under trial in *Mahsud*.

The two cargo ships were fitted with Polycomp exhaust gas temperature recorders in the hope that it might be possible to detect a trend in temperature which would indicate incipient valve failure in time to save the valve from being scrapped. In practice, the temperature rises suddenly on valve failure and by then it is too late to save a valve.

There are indications that correct balancing of cylinders, elimination of excess oil in the rocker boxes, correct tappet clearance and correct positioning of rocker arms over valve spindle ends are all important factors in obtaining reasonable RC valve lives.

Unfortunately ships are not provided with a complete fuel analysis on bunkering. Thus, satisfactory exhaust valve results reported may be associated with low vanadium, and general conclusions cannot be drawn from them. Many of the results quoted by other operators and for other engines are in this category.

The author concludes that:

a) For worldwide operation on all fuels, a water-cooled valve is necessary and should be installed as soon as a satisfactory version is available. It is to be hoped that S.E.M.T. have at last accomplished a breakthrough in this respect;
b) Rotocap valves, whilst being considerably cheaper than water-cooled, will inevitably burn at some point, depending on vanadium content. The practical life should lie between 1500/3000 h if guide bush clearances are kept within recommended limits. The need for such frequent maintenance places a heavy load on ship's staff and entails main machinery being out of service more often and longer than one would wish. In addition, these stoppages may be forced upon the operators and not planned and constitute detentions;
c) No particular difficulties or problems have been encountered associated with the water-cooling, provided the water is kept at the correct purity.

MAINTENANCE AND SURVEY

It has proved possible in the products carriers, despite the blower problems limiting ship speed, to carry out all main engine maintenance and survey work by ship's staff at sea and whilst loading and unloading as planned. On arrival for routine drydocking, the ships are expected, to be sensibly up to date with survey and planned maintenance work in the engine room.

Times taken to carry out certain operations:

1)	Change 2 exhaust valves and 1 fuel valve	—$1\frac{1}{2}$ h	(2 men)
2)	Change 12 exhaust valves and 6 fuel valves	— 8 h	(4 men)
3)	Change cylinder head	— 5 h	(3 men)
4)	Removal and replacement of a main bearing	— 4 h	(2 men)
5)	Overhaul of two units and associated main bearing	—16 h	(3 men)

Routine drydocking periods of the tankers are, as hoped, not governed by engine maintenance requirements and the period between them has been increased considerably.

When the exhaust valve problem is finally solved, the main engine maintenance load on ship's staff, of which it constitutes the major proportion, will be decreased significantly and some reduction in engine room personnel may be reasonable.

The cargo liners have operated most of their time on charter between the Pacific and Caribbean, so that facilities built-in especially for Calcutta and Manchester have not been exploited as planned. However, considerable self-maintenance has been carried out, limited only by high charter party speed, and deep draught operation providing little speed latitude for shutting down one engine for maintenance at sea.

Thought is being given to improving tools and equipment carried to ease maintenance load.

TYPE OF PERSONNEL

The author's experience has confirmed that obtained by others, that the correct attitude of mind is required in personnel for the most effective operation and exploitation of this sort of installation. This requirement is not only confined to the engineering department. The navigation department must be fully involved in all plans to maintain equipment at sea and their acceptance of the seaworthiness of the vessel on one engine is essential to the overall concept. Several instances have been recorded by the author's company, and by other operators with multi-engine installations, where the ship has remained mobile and continued operating at half power whilst a fault was corrected on an engine. The fact that these ships operate in the U.M.S. condition has required additional flexibility of outlook from its senior officers.

CONCLUSION

The author does not claim that medium speed engines are the answer to all problems or that they are without their problems, but he does claim that over the relatively short period of 3/4 years during which he has had direct access to operating experience, the performance of the particular machinery fully justified its choice. He is also confident that the WC exhaust valve problem will be solved in the near future and that satisfactory lubricating oil consumptions will soon be achieved. With these two major problems out of the way, multi-engined medium speed Diesel engine installations, particularly of larger bores to reduce the number of moving parts will, it is felt, become increasingly attractive.

ACKNOWLEDGMENTS

The author wishes to acknowledge the help and co-operation of Mr C. J. House, engineer superintendent, Cunard-Brocklebank Ltd, Monsieur J. Gallois of Société d'Etudes de Machines Thermiques, Crossley-Premier Engines Ltd, Mr A. Lindh of Lindholmen Motor AB and of the chief engineers of *Maihar*, *Mahsud*, *Lustrous* and *Luminous* for their help, co-operation and encouragement in the production of this paper and The Cunard Steam-Ship Co. Ltd. for permission to publish it.

Appendix

In reading this paper and listening to the presentation it should be borne in mind that in developing their design concepts the author's company were not reaching out to be first in anything. Their specification required satisfactory sea experience of all equipment before they would consider it. It was their endeavour to harness proven and known equipment. It should also be borne in mind that the four ships referred to in the paper are being worked fairly hard and the author believes that the machinery has responded well and the installation justified the effort which so many put in to achieve it. The ships left the European shores on delivery from their builders and went out to the East. Only one of these has been back to the U.K. and that for but a brief period in the time span being considered.

Luminous was not off hire at all during the first eighteen months. *Lustrous* was five days off hire (three to complete a modification to the switchboard and two to weld up a crack in a boiler). These periods were followed by eight-day guarantee docking periods, during which there was virtually no engine survey work to be done.

The Introduction lists advantages and disadvantages claimed for medium speed engines as opposed to slow speed engines. The company have found that, because these installations have proved satisfactory for U.M.S. operation and because two engines are fitted, the engineers have the time and opportunity, as day workers, to carry out maintenance and survey work which they would not have been able to do if they had had one large engine. As a result, the engineers are worked harder, and they find that the chief engineers are asking for a higher standard of engineer so that they can be fully effective in their day work capacity. These ships have all found it possible, by and large, to keep up to date with continuous machinery survey using own ship's staff resources except when occasionally short sea passages have prevented or delayed it.

Whenever an engine is stopped, logic dictates that exhaust valves be replaced with spare reconditioned valves. If insufficient spare exhaust valves are carried, valuable possible valve replacement time has to be expended on valve reconditioning, a task best relegated to periods when engines are running. For maximum effectiveness of personnel there is an optimum number of spare exhaust valves which should be carried.

It has also been found from experience that younger people settle down better with these installations.

Under the heading "Machinery Comparison" the theme is developed that when assessing operational experience, to be fair one must know what it was designed for, as well as how it was used and be careful to differentiate between the main equipment and its peripherals. This is very important. Perhaps also one should have included that a knowledge of performance of competing equipment is also useful.

Passing on to the section on controllable pitch propellers, the author feels that many more people would fit such propellers if they carried out thorough through-costing, as his company's indications were that they were not as expensive as they appeared at first sight. In fact, one tender which they received quoted less for a c.p. installation than for a fixed propeller arrangement, when credit had been taken for all the various things which can be eliminated or made smaller when a c.p. propeller is fitted.

Their experience with these propellers has been very satisfactory, marred only by a mishap when, due to a chokage in the servo-system, a propeller went to full ahead on starting the pumps, although the control lever was in neutral. The propeller position was not noticed by the bridge from which the engine clutches were then engaged. The ship went ahead, damaging her bow. Since then several additional safety measures have been introduced, namely, improved filtration of oil, modified entry to critical servo-orifice, electric override control of propeller, full pitch alarms fitted and interlocks fitted to prevent clutch engagement except when propeller is in neutral position.

A feature fitted into these ships, which the author thinks is quite useful, particularly for pilots and new officers, is that the propeller control lever may be switched to a one-engine mode so that the full throw of the lever corresponds to maximum permissible propeller pitch.

Incidentally, when carrying out crash stop trials from full ahead, these were done first by bringing the propeller to neutral, dwelling a pause and then moving it to astern. The manufacturers said that this was the quickest way to do it. The author said: "Why not pull the lever right through?" The manufacturers said: "You can, but not us." They then did it, and proved that in practice it was the quickest way of stopping the ship. The deceleration was such that as they went to the wings of the bridge to watch the wake, the pilot remarked that they were looking to see if the propeller was still there.

The author appreciated that he was sticking his neck out by quoting 17-21 days off hire to maintain certain slow speed engines. Before slow speed engine designers get too steamed up, he points out that this period is not only a function of the engine design, but also of repair yard labour available, repair yard expertise in this particular engine, repair yard planning, availability of spare gear and so on. His argument is not dependent upon this number of days and can accept a reduction. He will, however, be interested to obtain operators' as well as designers' views on average periods for different engines.

Under "Equipment Selection" the author has thrown together a number of principal factors which should be considered when selecting equipment. In practice, such a list is of course inexhaustible. Perhaps, however, what he should have mentioned is the fact that there are also factors, other than technical, which govern choice of equipment. Among these are emotions such as hate, anger, jealousy and even love—love of an idea. There is also ignorance and the "not invented by me" attitude. Sometimes, what he terms "Thy masters' preferences" is dominating. You have all met the man who says "So and so at the X Y Z tells me he wouldn't touch A B C with a barge pole—his main engines run 20 000 hours between overhauls". When putting something relatively new to sea, it is well to "know your enemy"; you should know what you are up against, especially if you get your way and run into teething troubles during the introduction of this particular piece of equipment.

Turning to the exhaust gas blowers, the deposit on the Napier blower gas nozzle rings and on the turbine blades has been identified as carbonized lubricating oil. Unfortunately, water washing has yet to be fitted to these, so the author cannot report on how effective it is, but understands that in the contributions we shall be hearing some more about this.

The author's company are puzzled as to why the Brown Boveri blowers are not exhibiting the same choking symptoms as the Napier blowers and he would be interested in suggestions as to why this should be, although the engines associated with the former may be using more lubricating oil than those associated with the latter. Recent investigations have shown that the nozzle choking starts with the top nozzle.

As far as the liner cracking is concerned, it does, on the face of it, look odd that the bending moment has been designed in this way, but the author understands that there is a clever reason for doing so. He was interested to come across the sketch of modifications carried out by FIAT to their B300 engine liners to prevent similar cracking when they raised the cylinder output (Fig. 9).

With respect to tightening of cylinder heads, it has been reported that the initial tightening can be carried out satisfactorily, but once the valve gear etc is connected up, it is not possible to re-tighten certain nuts and leakage can occur.

It is interesting to note in Table I that two of the cylinders have a greater wear longitudinally than athwartships. So far they have not received a satisfactory explanation for this. One theory considers it is due to slower expansion of the engine longitudinally than athwartships when starting up as a result of asymmetrical disposition of material. Another theory is that, particularly where cylinder liner injectors are not fitted, the lubrication is less effective longitudinally than athwartships.

In the section on the gearbox, mention is made of the need to eliminate unnecessary vibration to protect instruments and eliminate pipe fracturing. Some failures of pneumatic

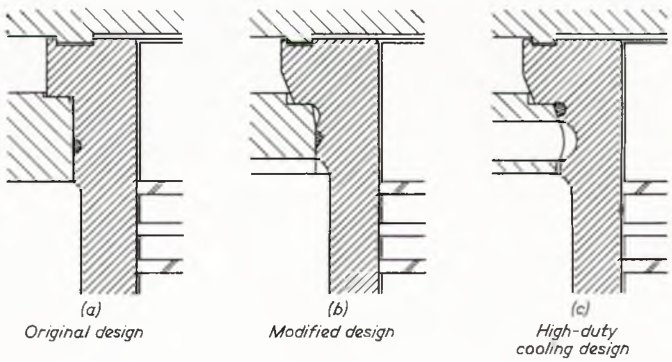


FIG. 9

temperature controllers have been reported, which have been attributed to vibration damaging control reeds and fracturing or holing control pipes. In the "five wheel" gearbox mentioned, the diameter of the main wheel is smaller than for the three gear unit, as it can be hardened and ground.

The author had intended to say a few words about tumbling, but understood that Dr. Pinnekamp would be dealing with this more effectively than he could.

The current drill employed to eliminate the tumbling effect is to declutch after engine has brought main gearwheel up to speed. The pinion then centralizes itself under centrifugal force and clutch is re-engaged.

Current lubricating oil consumption varies from 50 to 100 gallons/day per engine (i.e. *circa* $1\frac{1}{2}/3$ g/hp h). In this connexion, continuous machinery survey is a disadvantage as one never has an engine completely re-ringed/deglazed and back to design consumption.

Trials are currently in hand ashore to extend the running-in compound deglazing technique to engines operating in service. In the event of L.O. consumption starting to rise, it is proposed to introduce running-in compound into the fuel/liner contact conditions. The author understands that there have been some quite good results on trials with this procedure.

It would be of great interest to obtain views on this L.O. consumption problem, particularly from the designers of similar engines with cylinder liner injectors where one would have expected the problem to be more acute.

With regard to the exhaust valves, the attack on exhaust valve corrosion with RC valves in *Maihar* has been four-pronged:

a) maintenance of valve guide clearance within limits;
b) use of additive in the fuel; this has undoubtedly resulted in cleaner double bottom tanks, cleaner fuel system, fuel heaters, pipes and engine exhaust system; on this evidence they would expect cleaner turboblower nozzles and turbine blades; unfortunately they cannot yet report on the use of this additive on the ships with the Napier blowers;
c) elimination of valve cage distortion; there was evidence that incorrect tightening of valve cage holding down bolts resulted in seat distortion which one could visualize as causing rapid valve face deterioration; instructions have been issued to ensure correct tightening;
d) limitation of exhaust temperature; the change air coolers do not appear to have the reserve hoped for; as they become dirty under tropical conditions, the exhaust temperatures would rise above the normal F.P. value of about 430°C if F.P. was maintained; observations had been made that where exhaust temperatures of 440°C to 450°C were regularly experienced, valve failures were to be expected; care was being exercised to restrict exhaust temperature to 430°C by reducing maximum engine output to about 90 per cent of maximum continuous rating.

It was unfortunate that several "changes" had been made at the same time, as the exact contribution of each was not then evident, but the fact remains that since introducing these measures, only nine rotocap exhaust valves had been consumed in 12 months, i.e. 5160 hours on each engine. In addition, it was found that time between necessity for overhaul had been significantly increased with four valves running exceptionally over 3000 hours and two over 3600 hours and still being in order when opened up. It has been estimated that prior to the introduction of these measures, some 30–45 valves would have been consumed in the same period.

During this period whilst actual fuel analyses were not available, it is estimated that 1940 tons of fuel of vanadium over 230 ppm was used out of a total of 7000 tons. Additive used during the same period was 820 gallons; cost is about £900 per year.

Four of the ten tungsten carbide plated WC valves fitted in *Mahsud* have just been examined at 2408 hours. All spindles are reported as perfect and valve faces as good. This is very encouraging news.

Brinnelling of rotocap Belleville washers has been experienced, and care must be taken to maintain the assemblies to ensure correct rotation.

When engineers are working on exhaust valves, they are not doing other things. If the company can increase the valve life significantly they will release effort for other maintenance and can possibly reduce numbers of engineers borne.

Although not raised in the paper, the author would like to mention the question of compatibility of spare gear provided by different licensees. By and large they have found no difficulty with this, but differences do creep in and they were surprised when an alert chief engineer discovered that a Crossley-Premier connecting rod was some two pounds lighter than the original supplied by Lindholmen. On enquiring, they were told that they had nothing to worry about. The author is not a Diesel engine designer and a two pound tolerance on a machined rod seems a lot to him and, whereas he can understand such a difference being acceptable on all rods of an engine set or even possibly on a side by side pair in a vee-engine, to introduce a light rod at random in an engine looks untidy to him. He would be pleased to be reassured on this point by the engine designers.

A few other instances of non-standardization between manufacturers have come to light and could have caused a considerable nuisance if not discovered in time.

The author would make a plea, and somehow feels sure it must have been made many times before from this and kindred rostrums, for licensees not to change the design of engine components for apparent transitory personal advantages. If they have a good enough reason for changing, then they should be able to convince all other licensees—components which are changed should be done so in such a way that they are fully interchangeable with the original. If this is impossible, but change is still required, the new component should be clearly identified as not interchangeable with previous designs.

This leads the author to refloat an old idea for your reactions, which a few people kicked around a few years ago, but which they have not had time or opportunity to develop.

Machinery operators are dependent upon their own stocks and licensees for provision of spare gear. In a dire emergency the "old pals" act may sometimes be invoked to borrow someone else's spare gear until it can be replaced, or it may be possible to appropriate items from similar machinery under construction or awaiting delivery. In the interests of keeping locked-up capital as low as possible, spare gear held by licensees and operators is kept to the essential minimum, depending upon failure probability, usage rates, service, geography and so on.

Feed-back of operator experience, if and where it gets outside the particular company, generally passes to a licensee from where it may eventually get back to the designer. Operators and erstwhile operators are, therefore, dependent upon the goodwill and integrity of designers and licensees to disseminate such information. The organization to do this costs time and money and, to be done intelligently, requires the services of high calibre staff. Human nature being what it is, one can hardly expect a designer/licensee to advertise deficiencies. The designer/licensee is thus limited in what he can afford to provide as an information service, monetarily or politically.

Turning now to the specific piece of machinery which we were considering, because it is easier to illustrate the argument with an example, the Pielstick PC2V engine which is now being built and operated worldwide, the author's proposal is to form what someone jocularly referred to as PUPS, the "Pielstick Users' Protection Society".

The idea is initially for operators of Pielstick engines to pay a modest levy towards and to register all of their spare gear with a small organization (PUPS), which would need access to a telex, and would undertake to catalogue this spare gear and keep a master register.

On requiring spare gear, an operator would use his own or telex PUPS for an alternative supplier, in either case informing PUPS of action taken. The operator appropriating such equipment would be responsible for replacing it or items of equivalent value, as advised by PUPS. One can visualize PUPS becoming a central ordering agency with considerable price bartering power, or in other words, being able to put the squeeze on the manufacturers of the component.

There is currently probably too much spare equipment around the world. PUPS would be responsible for rationalizing this and ensuring a proper spread throughout the world. Gradually it should be possible for the group of owners to stock up on large items, e.g. crankshafts, timing wheels, which individually would not have been an economic proposition for, say a two-ship fleet, instead of overstocking other items. These arrangements would, in due course, not be confined just to the Pielstick engines but would also cover auxiliary equipment essential to their operation.

PUPS would, therefore, organize the immediate availability of more extensive and cheaper spares on a worldwide basis.

The next step would be for PUPS to act as a centre for operating experience and by regular bulletins members could be circulated for information and with information.

It would not be the intention to exclude designers and licensees from some form of membership of PUPS, but the essential point would be that it would be formed for the good of the owners and be an owners' association.

With a properly developed PUPS type of organization and modern data retrieval and updating systems and equipment, we should, the author believes, very quickly be able to provide a world-wide spare gear information and supply collaboration service on all types of equipment and machinery.

It would be interesting to hear views on this embryo proposal. The author is not decrying the service currently provided by licensees, this proposal is complementary to it.

Some years ago he attended a meeting of this Institute at which a paper on operating experience was presented and well remembers a remark made later in a local hostelry "it's all very well him standing up there and saying how well things are working out, but I know some of his engineers and they tell a different story".

The author's role, as he has seen it, has been to examine and distil information provided from sea and to arrive at views compatible with this information. This he has tried to do to preserve the paper's credibility in the operational aspects. But this is not to say that all engineer officers in the ships concerned will necessarily agree 100 per cent with the final views expressed for which he remains legally responsible but does not claim originality.

He personally considers that the choice of this machinery has been fully justified in operation. Such problems as have been encountered with the medium speed engines themselves have been relatively minor and the company are well on the way to reducing their effect.

Conversion factors—S.I. units

1 hp	=	0·7457 kW
1 c/s	=	1 Hz
1 gal	=	4·546 litres
1 ton	=	1·016 tonne

Discussion

Mr. J. F. Butler, M.A., M.I.Mar.E., said that papers describing service experience with engines were always of great interest and this one was of particular importance because of the factual way in which the reasons for the choice of engines and the troubles which had occurred were described.

Heavy expense on lubricating oil seemed bound to occur with trunk piston engines burning residual fuel. Work by Dykes and other later investigators had shown that the net loss of oil past the piston of a trunk engine was the relatively small difference between the large amount of oil which passed the rings during part of the cycle, less the slightly smaller amount which was returned during another part of the cycle. It followed that apart from the actual oil loss, the rest of the crankcase oil was contaminated by combustion products carried back by the oil which had taken this excursion into the combustion area.

In any engine of this type the junction between cylinder cover and liner was difficult to design because of the need for a substantial flange on the top of the liner, the heavy plate required on top of the cooling jacket and the thickness of the lower face of the cylinder cover, which together provided a considerable mass of poorly cooled metal. The necessity to position the cylinder cover joint at a smaller diameter than the abutment on the water jacket resulted in a bending stress intensity at each cycle, as the gas pressure tended to lift the cylinder cover. In addition, the high heat flux in the combustion zone at the top of the liner resulted in steep temperature gradients through the metal and consequently high thermal stresses.

Reducing the width of the joint was an effective palliative because in addition to reducing the amount of the bending moment, it also reduced the bolt load required to maintain gas tightness. There were other solutions to this problem using internally cooled flanges, but Mr. Butler said it would be very wrong of him to advertise his competitors' designs!

Exhaust valves on highly rated engines were also liable to be a source of trouble because of the long heat path between the vulnerable seat and the nearest cooling water. The fact that it was desirable to mount these valves in cages for quick replacement also reduced seat life because the seat itself became virtually uncooled. On this engine one would expect a heat flux to the valve head of five to ten calories/cm^{2}s, if the head temperature were kept below 500°C, which would be desirable to avoid deposits on the seat. However, the distance from the seat to the nearest cooling water in the valve cage was something like ten cm and with this heat flux there would be a temperature gradient between the two of at least 100°C/cm. It followed that the seat temperature had to be well above the melting point of sodium sulphate and possibly above the melting point of the sodium vanadium eutectic. Under these conditions deposits would form on the valve seats which would crack during temperature changes resulting from change of engine load. Hot gas escaping through the cracks at very high speed during the combustion period was then likely to cause local burning of the seat face.

In this connexion, could the author give particulars such as viscosity and sulphur content of the fuels actually used in the engine?

There would appear to be merit in mounting the exhaust valves directly in the cylinder cover to provide seat cooling, and possibly in using sodium cooled valve stems as was common practice with high output aero piston engines. This would necessitate designing cylinder covers for easy removal. The problem of exhaust valve burning was bound to become more acute with increasing cylinder size and output because of the still longer heat flow path from seat to water.

The author and his colleagues were to be congratulated on their courage in choosing an engine type relatively untried for this particular service and in exploiting the advantages of the twin engine arrangement so successfully.

Mr. P. J. G. Mack, M.I.Mar.E., confined his remarks to combustion and load control. There were many factors associated with the recurrent failure of gas joints, in addition to those referred to by the author. The majority of these were usually resolved at the prototype stage and would therefore not normally concern the shipowner, provided that the operational conditions did not depart from the assumptions made in determining the safety margins which were built into the design.

A principal criterion in this respect was the maximum gas load and its magnitude under manoeuvring conditions.

The author had referred to the correct balancing of cylinder power in connexion with exhaust valve performance. The balancing of an engine solely on the basis of exhaust temperature and maximum pressure could be a somewhat tenuous practice. Apart from the questioned reliability of pyrometers and the possibility of exhaust valve leakage, to which the author would no doubt subscribe, there were a number of other factors which could produce high peak pressures in addition to those associated with a fat card.

A uniform distribution of fuel to all cylinders was very desirable throughout the whole range of power and speed but difficult to achieve with single jerk pumps especially where the pumps were calibrated at a specific output without precise matching throughout the rack travel. Balancing at full power might sometimes, therefore, produce an idling condition where less than half the injectors were functioning and where combustion was required to take place at a lower compression pressure. This condition, coupled with the remarks of the author regarding the range of fuel delivered through a fixed orifice fuel nozzle, might contribute to nozzle trumpeting, inefficient combustion, fouling of blowers and their inability to accelerate in unison with a rapid increase of propeller pitch. This condition could, and had in a number of recent cases, been known to produce damaging combustion transients which were undetectable with normal instrumentation.

The hydraulic governors fitted to most engines were basically sensing only speed in relation to fuel delivery. The designed characteristics of the governor were determined from the torque/speed curves supplied by the engine maker. In turn, this data was believed to be developed from the propeller manufacturer's curves. Load control although not always fitted, provided a very necessary governing refinement whereby at the over-fast approach to a maximum load, such as engine racing in a seaway, a modifying pitch reducing signal might be initiated if the fuel control tended to advance ahead of the pitch signal. Excessive operation of a load control/pitch trimming device during normal manoeuvring was, in itself, an admission of a fundamental shortcoming in the control system. Medium speed engine control systems did not generally utilize either boost pressure or shaft torque as control functions.

Fig. 10 illustrated a characteristic set of curves for a controllable pitch propeller and showed the change of slope at the extreme ends which was quite significant in that it produced a slow rate of drop in the power absorbed by the propeller in the region of zero pitch and low idling speed, while the converse was the case at the approach to full power. The derived power control curves should depend on the installation arrangements, the intended modes of operation, the variable number of engines to be coupled at any one time and their torque/speed characteristics, so that the integrated control system ensured that an adequate reserve of engine torque was available to meet the proposed rate at which pitch might be changed without undue perturbation or engine distress. With a low idling speed 3·0 per cent of maximum power could be reached (indicating), and the range of power was, therefore, much greater than with a fixed pitch propeller.

The c.p. propeller manufacturer determined the maximum rate at which the propeller pitch could be changed in relation to the hydrodynamic turning moment exerted by the propeller blades, the design of the hub assembly and the capacity of the servo-pumps. It should, however, be the responsibility of the engine builder to decide the appropriate rate so that excessive transient cylinder pressures were avoided and that an acceptable torque/speed relationship obtained during acceleration from idling speed.

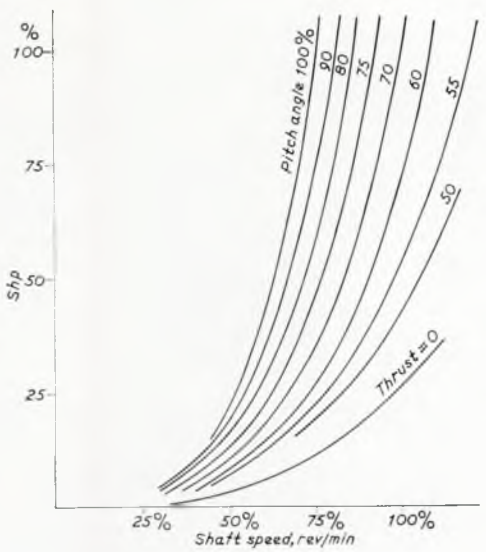


FIG. 10

For instance, if a low idling speed was selected coupled with a rapid increase of pitch and revolutions, due allowance should be made for the accelerating component of the total absorbed torque which, in the case of a standing start, might be initially offset by propeller slip.

The curve (Fig. 11) was derived from the first series and based on the selection of an idling speed coupled with a simultaneous increase of pitch and speed.

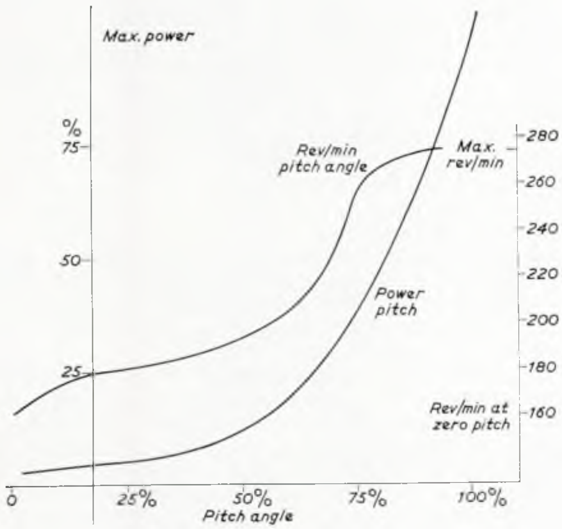


FIG. 11

In the case of the author's ships having a fixed engine speed, the propeller power developed at zero pitch would be in the region of double that of the same machinery programmed for an idling speed. When coupled with a base load alternator, the origin of this curve would be lifted to a level that would improve the distribution of cylinder power and give the turbo blower a reasonable standing start.

The configuration of this curve could vary substantially depending upon the designed pitch/speed relationship, but in all cases it was important to remember that these represented only the steady state condition. If a rapid rate of increase of pitch and speed was intended the engine would be required to develop a greater power or torque to meet the required acceleration rate without inducing excessive transient piston loads—which was Newton's Law. The load control arrangements should modulate this condition.

The curves in the figure were theoretical and should be treated accordingly until demonstrated to be otherwise, in much the same way as the old steam engine hypothetical diagram which had its own correction factor. Provided one knew what was happening in the engine it was all right putting the lever to full astern and saying that the propeller did not fall off.

Against this background it would seem reasonable to expect the builder to carry out a fully instrumented trials procedure in order to confirm that the plant behaviour in the manoeuvring mode was within the predicted limits. Unfortunately commissioning trials left a lot to chance, were often carried out like an eleventh hour Mad Hatter's Tea Party, where the builder wanted his final stage payment without penalty, the shipper wanted the ship and the superintendent had to take pot luck.

Even the standard test procedure for Diesel generating sets did, in fact, provide governing and voltage regulation tests when the full designed load was applied instantaneously. This might be the only instance in the lifetime of the machinery when such severe loading was applied: the generator loading was, furthermore, continuously monitored; overload protection was provided and verified periodically. For some unknown reason, the higher powered medium speed main propulsion machinery did not receive such cosseting. Mr. Mack was not advocating complex control systems—far from it—and like the author he would prefer less old parts, simplicity and greater reliability, and sincerely believed that in all cases, fully instrumented sea trials should be carried out to confirm that the plant behaviour under manoeuvring conditions as well as steady state did not exceed the predicted levels and that sufficient margin had been built into the system to cater for the normal deterioration of components between overhauls.

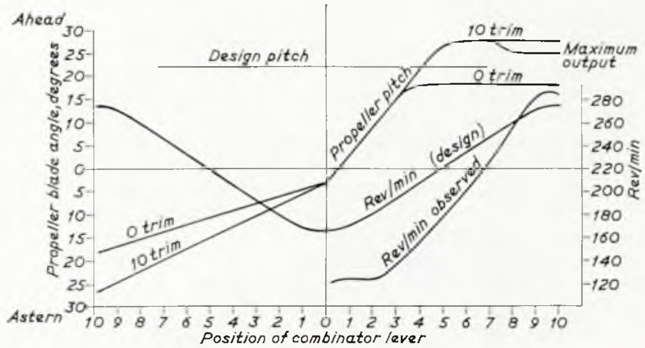


FIG. 12

Fig. 12 was a characteristic combined diagram and related specifically to the original pattern of control built into a twin screw ship propelled by two engines per shaft and also in a number of similar ships operated by the same company. The second of each pair of engines developed half the power of the first and the total power developed was 5800 bhp/shaft in engines of substantially lower rating than those being discussed by the author. The linear pitch changing rate of 12 s (i.e. full ahead pitch—full astern pitch) represented a mean rate of power change of approximately 16 per cent/s, but due to the gradients of the power curve the spread was in the order of 4 per cent/s at the low output end up to in excess of 35 per cent/s at the approach to full power. It was interesting to note that the published accelerating rates of advanced steam and nuclear plant were in the region of 1–4 per cent/s. This instant power would no doubt appeal to the swashbuckling attitudes of R.N. and M.N. bridge officers but it certainly did not endear itself to the average engineer officer. The machinery was commissioned after the normal type of trials where torque and maximum cylinder pressures were measured in the usual way at the steady state in several increments up to full power.

The reduced idling speed was no doubt introduced with the aim of reducing fuel and lubricating oil consumption—particularly relevant since the ship spent about a quarter of its time in the manoeuvring mode. Another relevant point was that the continued lifting of relief valves, where fitted, was not a requisite to excessive cylinder pressures. Small uncooled relief valves, as well as being vulnerable to complete choking, could be very troublesome, and the usual setting of about 18 per cent above the maximum designed combustion pressure might unwittingly be exceeded by an already overworked engineer officer. Following some years of operation associated with piston skirt scuffing and roughening of the cylinder liners and culminating in piston seizures and two violent crankcase explosions, the control system was redesigned on the basis of this modified combined diagram which had been superimposed on the previous diagram (Fig. 13).

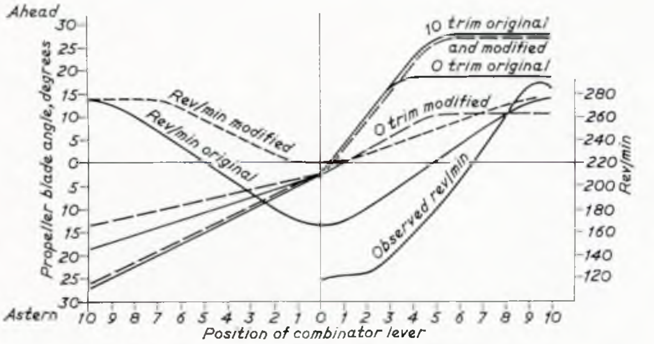


FIG. 13

It would be noted that the idling revs had been lifted from the observed 125 rev/min to 220 rev/min and that the slope of the pitch changing rate had been reduced considerably at the position of zero trim. The effect of these modifications was to increase the blower speed at idling from 2000 rev/min to 4000 rev/min and provide a positive charge air pressure. Manoeuvring was carried out in the usual way with a trim setting of four and a marked absence of blower surge and black exhaust which obtained previously. Acceleration was smoother but the position of the fuel rack was still noted to be held in advance of its full power position during the approach to full power and during crash manoeuvres. Rather than reduce the pitch trimming value in an arbitrary manner, it was decided to attempt to use a piezo-electric transducer to monitor the transient behaviour during manoeuvring.

This was such a new thing intended for use by ships' engineer officers, that the writer had some figures to illustrate his point.

The instrument used was an oscilloscope, with special transducers screwed to the indicator cock or clipped to the fuel pump delivery pipe. The transducer probes, when subjected to strain, generated an electrical signal in the piezo crystals which in turn produced a drawcard image on the cathode ray tubes.

When the engine behaviour was monitored during manoeuvring, it was very successful, and a running picture showed that despite the control system modifications a transient combustion pressure rising to a maximum of 10 per cent in excess of full power pressure was being maintained for periods of up to 80 seconds, depending upon the nature of the ship manoeuvre (see Figs. 14–17).

In view of the availability of this equipment, there was no need for any anxiety when monitoring engine conditions, whether in the steady state or not. The equipment, was available, if needed.

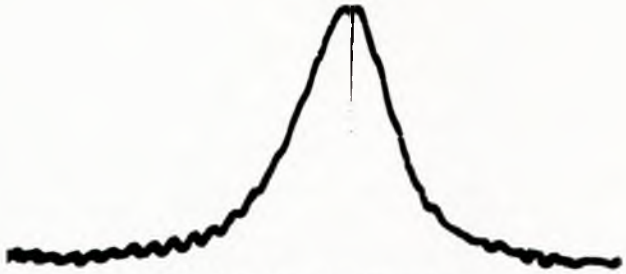


FIG. 14—*Compression pulse*

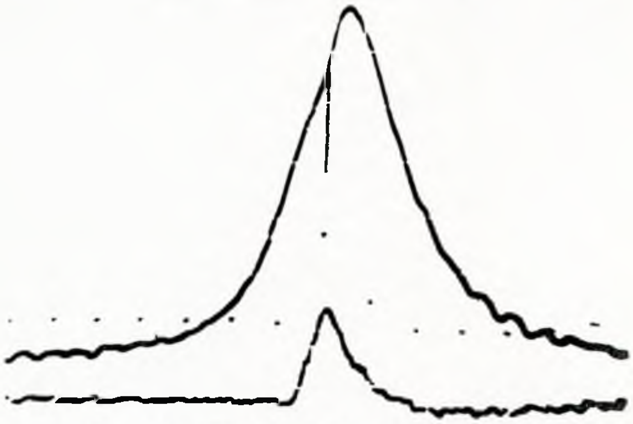


FIG. 15—*Firing pressure and fuel pulse*

FIG. 16—*Fuel line—no unloading*

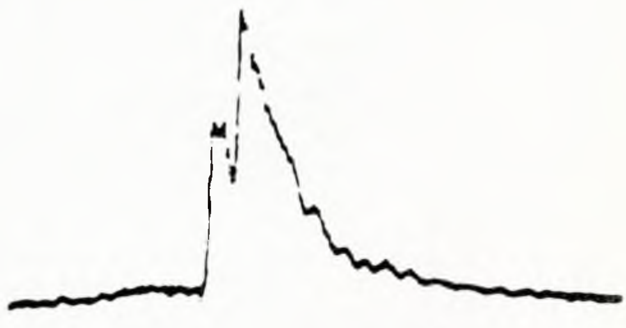


FIG. 17—*Fuel line with unloading*

MR. D. CASTLE said he would resist the temptation to enter into controversy with the previous two speakers, but would simply add to what Commander Short had said and comment on the paper. Among the disadvantages he had listed for medium speed installations was that the multiplicity of cylinders was a possible cause for increased maintenance load.

Maintenance on a large number of small lightweight cylinders in a confined space with low headroom was quite different from maintaining a small number of large, heavy cylinders in a large headroom environment. This was not yet appreciated by ship designers and the same approach as for ship design would not do. The system of lifting and of parts handling needed to be designed at the outset, and not just considered when the engine room was completed. It was vital that those parts needing to be worked on could be easily placed and other parts stowed out of the way.

This was a very common fault, and he was not implying that these particular ships were designed in this way, but from what the author had said one might think so. Features such as a simple overhead travelling crane, to run over the engine and at least off at one end of it should be incorporated, where a travelling lifting overhead block could take over without manhandling. From the flat one should have an open run to a workshop, with jigs designed to hold for instance a piston with rod, a piston alone and possibly a rotating one to remove a gudgeon pin safely from a piston.

If pressure on designers was needed in order to do this, that pressure should be brought to bear.

As medium-speed engines were noisy, it was necessary to plan for this. Remote control and instrumentation were essential and could be made both simple and reliable by using an insulated control room, placed where it could be within reach of the machinery. Windows through which nothing could be seen were of no importance. The workshop location was very important; the control room less so.

With regard to lubricating oil consumption, it was true to say that medium speed engines had a higher usage per brake horsepower than slow speed. In the case of the ships described in the paper, constant speed operation was decided upon in order to utilize main system-driven alternators. This gave the advantage of space and weight reduction in auxiliary equipment and a high overall efficiency in terms of fuel burned. But, partly off-setting these gains, there was the higher lubricating oil consumption which inevitably resulted from running the main engines at full speed all the time. This point was made under the heading "Lubricating Oil Consumption" and was important to keep this factor in perspective.

This subject had been discussed in detail in the paper and it had been suggested that the use of running-in compound in the fuel was to avoid the need to deglaze liners when a new ring pack was fitted.

Unfortunately, this was not so. Deglazing would still be necessary. The benefit derived from the use of a running-in compound was to shorten the period of hazard when the risk of blow-by at high loads was present. Moreover, the high loads were carried with less risk even from start-up. The result was a quickly-achieved, well run-in combination. Full width bed on the top ring was obtained in five hours.

Gearboxes had been classified as "vulnerable" in the paper. It seemed to Mr. Castle that this was a sweeping generalization, based on experience with one make of box.

If tooth marking could be corrected by differential chocking, this was surely an indication of lack of stiffness in the gear casing; was this not the greater part of the problem with these particular boxes? Mr. Castle's company had been concerned with a large number of installations using reduction gears and so far had not experienced the difficulties described. He did not understand the reference to tumbling of the pinion when the clutch was engaged and wondered if the author would be prepared to elaborate on this point.

So far as the position of the main thrust block was concerned, he considered that wherever it was placed, it was merely a matter of taking it into account when designing the structure in the double-bottom. If the thrust was built into the gearbox, it was very important if distortion of the box was to be avoided, that the thrust was taken out of the gear case along the shaft centreline and through the end walls suitably stiffened, rather than attempting to distribute it via the side walls.

Mr. Castle's company had frequently refused approval of shipbuilders drawings on the grounds of inadequate provision for distribution of thrust from a built-in block. Of course, the shortcomings of the ship's structure were sometimes partly compensated for when an exceptionally rigid gear case was provided—and this had been known.

The incidence of axial shaft resonance was very low indeed and usually came about only when the axial location of the shafting was exceptionally soft. He suspected that the thrust block support would be found to have a low axial stiffness, for the system to be able to act as a spring mounted mass. It should not, in his view, be necessary to calculate the axial vibration characteristics of the shafting of any ship.

Reference was made in the paper to the underestimation of compressor capacity. This was a common mistake—losses from air control systems could account for something like three times the functional requirement. Compressors should be sized accordingly.

Under the heading "Temperature Controllers", the author referred to the difficulty of providing accurate temperature control with the two engines on a common jacket water system. This was inevitably true, particularly when one engine was not running and he would be interested to know how the system shown coped with this problem. Mr. Castle was not in favour of common systems—and preferred fully independent engines which need not incur undue complexity or cost if the concept of one engine standing by the other was adopted together with the use of a sea water tank giving a guaranteed level of sea water supply temperature. All the primary engine and gear systems could then use a simple oil or wax filled element thermostatic valve and still be stable.

He supported the author in his insistence on having automatic shut-down in the event of lubricating oil pressure failure. However, his demand for engine-driven lubricating oil pumps, although absolutely sound for constant speed engines was not so ideal for variable speed, where it was not too easy to match the shut-down pressure for any given speed to the safe level for that speed without an over-elaborate monitoring and actuating system. He admitted that the company had done it, apparently satisfactorily, on a number of occasions.

Fractured cylinder liner problems had occurred on very few PC engines and on none built by the writer's company. In his view, these had come about as a result of repeated overtightening in attempts to stop blowing without stopping the engine. A second tightening after initial running was not only unnecessary but undesirable with the steel or brass gaskets now standard on these engines for the past four years.

Finally there was no real reason why the small hole injector nozzles should not be suitable. There was a reference to the need to use a large hole type, but it was only a matter of heating the oil to 90 sec Redwood 1 to use the small hole nozzles.

Dr. Ing. W. Pinnekamp said that the author had mentioned various difficulties with gearboxes in which several breakdowns had occurred. On behalf of the gearbox manufacturer, Renk, Germany, Dr. Pinnekamp had made a detailed investigation of this subject. In his opinion, the results of these efforts had not only revealed a more thorough understanding of some extraordinary problems involved, but had also suggested adequate measures to overcome the difficulties successfully and, moreover, had considerably contributed to the further advance of marine gears to even higher engineering standards.

Therefore, he assumed that it might be of general interest to learn about some of the details of these investigations.

Regarding the measurement of tooth bending stress, conventional approaches to explain the difficulties had eventually failed, and the complexity of the problem became obvious. It could only be solved by the application of special strain gauge measurements to determine the distribution of tooth bending stresses over the length of the gear teeth.

The arrangement of the gear wheels was shown at the upper left of Fig. 18. As shown at the upper right, strain gauges were attached in the tooth root at selected points over the length (1 forward, 2 centre, 3 aft) on several teeth of the main wheel. When in contact with the pinion the tooth force would produce a bending stress. The lower left diagram showed an example of the oscillograph records obtained during one stress cycle. At the three strain gauges, the stress maximum was reached at different times, due to the helix of the tooth cutting. The differences in magnitude indicated an unequal stress distribution. When plotted over the corresponding positions of the strain gauges (lower right diagram) the general pattern of the stress distribution over the length of the tooth was clearly demonstrated.

Again referring to the upper left diagram, it should be noticed that, after one revolution of the main wheel, normally a different

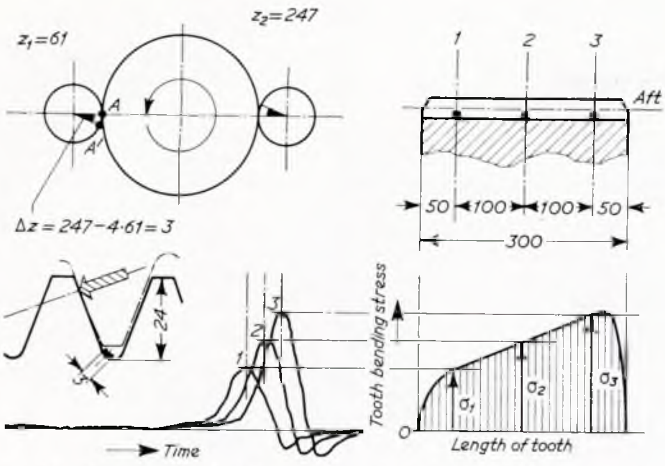


FIG. 18—*Stress distribution by strain gauge measurement*

pinion tooth would be in contact with the measuring tooth. In the course of the time, all pinion teeth would follow in a certain sequence. Therefore, it was possible to detect any irregularities along the circumference of the pinion with the aid of just one measuring tooth on the main wheel.

Application of the unequal stress distribution method revealed that there were irregularities. Condensed from a larger number of revolutions, Fig. 19 showed the variation of tooth

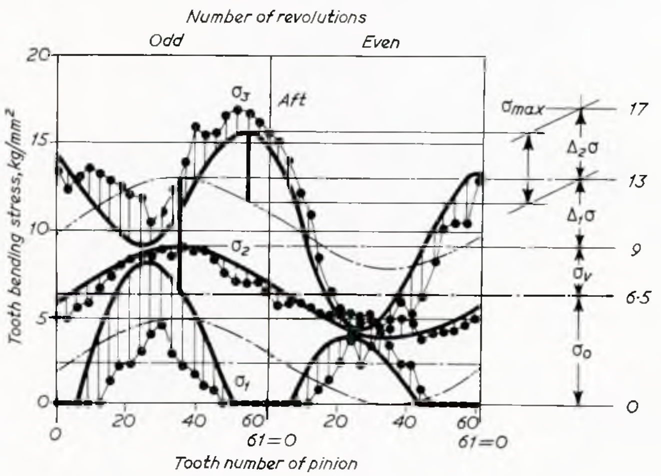


FIG. 19—*Unequal bending stresses during two pinion revolutions*

bending stresses during two pinion revolutions σ_1, at the forward end, σ_2 at the centre, σ_3 at the aft end. The points were the values measured and the solid lines an attempt to analyse their tendency by the super-position of the sine curves, in a simplified manner. The average of σ_2 in the centre was the normal stress $\sigma_0 = 6\cdot5$ kg/mm², produced by the mean torque of the engine. A vibratory stress $\sigma_v = 2\cdot5$ kg/mm² was superimposed, as indicated by the long wave sine curve with a period of two pinion revolutions, due to 0·5 order engine torque vibrations. At equal stress distribution, curves σ_1 and σ_3 would be the same as σ_2. But σ_3 appeared at a higher, σ_1 at a lower level, as outlined by the sine curves parallel to the σ_2 curve. This was an indication of a non-variable component of unequal stress distribution, due to non-parallel shafts or misalignment. Tooth bending stresses were increased at the aft end and decreased at the forward end by $\Delta_1\sigma = 4$ kg/mm².

But, in addition, there was also a variable component as represented by the short wave sine curves with the period of one pinion revolution. This was explained by tumbling of the pinion, i.e. the pinion did not exactly rotate about its geometric axis. Extra stresses due to tumbling were $\Delta_2\sigma = 4$ kg/mm². The summation of normal stress σ_0, vibratory stress σ_v, misalignment stress $\Delta_1\sigma$ and tumbling stress $\Delta_2\sigma$ rendered a maximum stress of $\sigma_{max} = 17$ kg/mm², which was almost three times as much as the normal stress.

Dr. Pinnekamp said it should be mentioned that the diagram did not represent the full load condition. It had been chosen because it was more instructive. At full load, a normal stress of $\sigma_0 = 15$ kg/mm² was measured, and the maximum stress at the aft end amounted to as much as $\sigma_{max} = 43$ kg/mm². Thus, by accumulation of additional stresses caused by excessive misalignment and excessive tumbling, the gear had eventually failed by fatigue.

Fig. 20 showed the stress distribution over the length of the tooth following from the above. The horizontal line at 9 kg/mm² represented an equal distribution encountering normal stress σ_0 and vibratory stress σ_v. By misalignment, $\Delta_1\sigma$, stress distribution became unequal, following the dotted line. Tumbling, $\Delta_2\sigma$, led to a variable distribution which ranged within the shaded area.

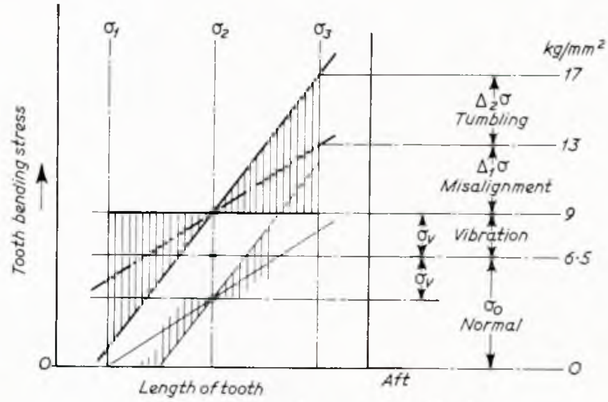


FIG. 20—*Unequal stress distribution*

The diagram also clarified why it was impossible to detect the unequal stress distribution by conventional methods, checking tooth contact over a longer period of time by inspection. The apparent tooth contact would correspond to the upper boundary of the shaded area, which seemed to indicate a more equal distribution than actually existed.

Further investigation had also revealed the reasons for excessive misalignment and tumbling (see Fig. 21). On large vessels, the structure was comparatively flexible. At the support of the forward and the aft pinion bearing, the structure would yield under the action of the vertical bearing forces F_1 and F_2. But the structure was not quite symmetric. Therefore, the spring

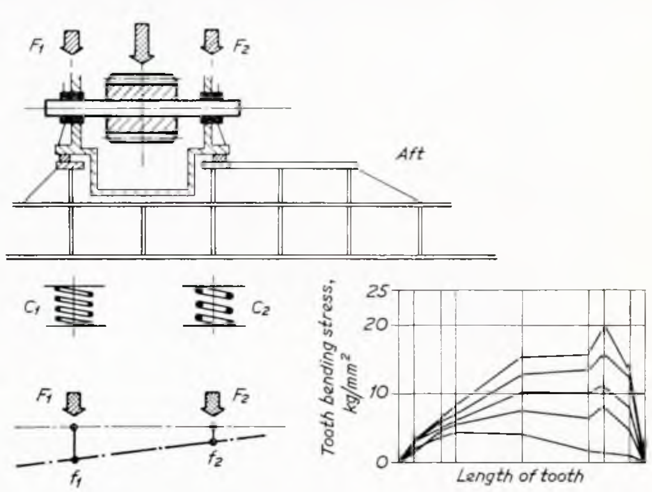


FIG. 21—*Change of alignment under load by flexibility of structure*

constant was greater at the aft end than at the forward end, $c_2 > c_1$. Correspondingly, deflexions f_1 and f_1 were different, and the pinion shaft would suffer a non-parallel displacement or misalignment at increasing load. As shown in the lower right, the distribution of tooth bending stresses changed accordingly, concentrating at the aft end.

Excessive tumbling only occurred with one particular design of pinion shaft bearing arrangement (see Fig. 22). On the input

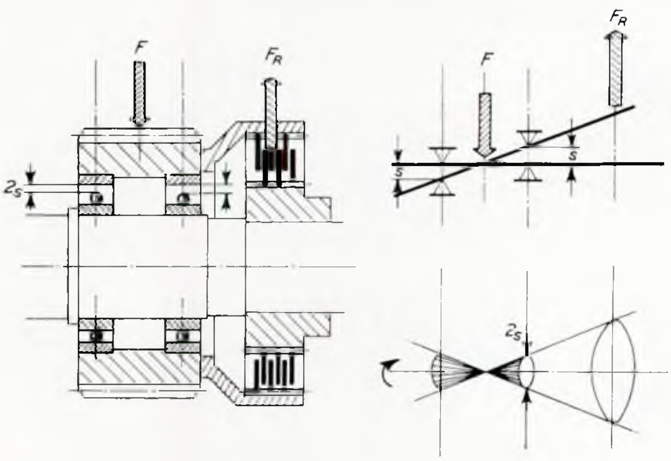


FIG. 22—*Tumbling of pinion generated by radial clutch force*

shaft, the pinion was supported by two bearings, and a multi-disc clutch incorporated. When the clutch had been operated, a radial out-of-balance force F_R had been unexpectedly found to develop within the last fraction of a second. With the particular design in question, the clutch was located outside the bearing support. Therefore, as shown in the upper right diagram, F_R was able to tilt the pinion as much as the clearance 2 s of the bearings allowed. The tilted pinion then started to tumble when rotating (lower right diagram).

From the information collected by the investigations, rapid progress had been achieved by the introduction of effective means to improve the technical standards. In Fig. 23 the left

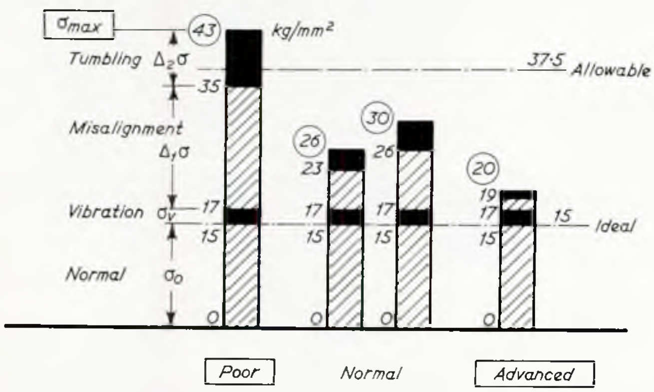


FIG. 23—*Progressive reduction of maximum tooth bending stresses*

column with a maximum stress of $\sigma_{max} = 43$ kg/mm² represented the initial situation, classified as poor standard being beyond the allowable stress limit. Preliminary measures to avoid excessive misalignment and tumbling reduced the maximum stress considerably (26 to 30 kg/mm²), arriving at a normal standard which guaranteed safe operation. Nevertheless, more rigorous application of the knowledge obtained led to further improvements. Maximum stresses went down to as little as 20 kg/mm², which was almost ideal.

Thus, in the end, an advanced standard was achieved. Defective installations had been improved by proper modifications, old designs abolished, and new designs created in accordance with these advanced standards.

COMMANDER N. K. BOWERS, R.N., M.I.Mar.E., confined his remarks on this most interesting paper to a few points concerning the overall installation.

Some of the potential advantages appeared to be border line. For example, recent studies suggested that the first cost of a product carrier fitted with geared twin medium speed Diesels would be little different from the slow speed equivalent. Much appeared to depend upon the method of producing the electrical load. If electric power was produced mainly from an alternator driven from the main engines then the medium speed installation could have a lower first cost. However, it then had a higher fuel cost compared to the slow speed Diesel with a waste heat alternator.

It was difficult to see how the medium speed installation could be regarded as a more simple arrangement than the slow speed direct drive Diesel. Perhaps Commander Short was referring to the elimination of a waste heat boiler and its associated turbo alternator. Even so, a main gearbox, clutches and a c.p. propeller must weigh heavily in the balance on the other side. On the same basis, it was difficult to see why a medium speed rather than a slow speed should be more easily adaptable to U.M.S. and item "k" namely the low main component wear rate i.e. long life, should be no worse with the slow speed engine.

It suggested that there were three real advantages for a medium speed engine in a product carrier (in which height of machinery space was not significant). These were:

a) The ability to carry out maintenance on one engine at a discharge or loading port while still having the capability of moving the ship in emergency;
b) The reduction in cost of main engine repairs and overhauls at drydock;
c) The ability to drive the cargo pumps from the main engines, thus reducing the maintenance associated with auxiliary steam turbines valves and their associated systems.

The bow thruster had been the subject of debate for some years. He would be grateful if the author could state just how much he considered the bow thruster had been worth in retrospect compared, for example with the cost of tugs. He would also be interested to have his views on the relative merits of a bow thruster *vis à vis* an active rudder.

Another speaker had referred to control air. He would support his plea for an ample supply of control air and at the same time suggested that for control equipment only oil-free compressors were used. The garage type compressor all too often discharged oil and water vapour into the control air system and the consequent rust and debris choked the orifices of the pneumatic actuators.

In the table giving the total days in hand for slow speed and medium speed engines the author did not take into account the loss of time due to steaming on one engine. Under this condition the ship probably lost two knots when fully laden and perhaps up to four knots when in ballast. There appeared to have been a substantial amount of unscheduled maintenance with the engines chosen by the author, and if it was assumed that only one engine was in use at sea for, say, 50 per cent of the time the lost mileage was equivalent to 15 days off hire. If this should be the case, then the figures for total effectiveness were inverted, and the slow speed engine showed to advantage compared to the medium speed.

So far as the problems of maintaining steam pipes and steam valves on deck were concerned, there was now sufficient experience to state that this problem appeared to be well on the way to solution. Trials had shown that by applying a suitable anti-rust coating and then painting with a high temperature resistant paint no noticeable corrosion of upper deck steel piping and fittings under high temperature conditions had taken place for five months (the period of the trials).

With regard to item B below this table it must be emphatically stated, that at least one major oil company had a great interest in reducing the time in repair yards to zero. However, one good reason for drydocking was to reduce the resistance of the hull and even given the best anti-fouling compositions there was still a noticeable fall-off in performance at the end of 12 to 18 months at sea.

He would fully endorse the author's view on the desirability of engine driven oil pumps. The hard fact of the matter was that one could never rely on a stand-by engine cutting in automatically. It would be worth going one step further and fitting each medium speed Diesel engine with its own mechanically driven jacket water pump and salt water pump as well. These had proved to be very reliable for Diesel generator engines and the same philosophy could be applied to medium speed engines for main propulsion. It was probable that auxiliary pumps would have to be used for the start up and shut down periods, but under these circumstances there was a full watch below and complete control of what was going on. It was in the full away condition that the maximum degree of reliability was required and this could only be obtained by the use of engine driven pumps. On the same theme, the author apparently used a jacket water cooling sea water system that was common to both engines. From the overall reliability point of view it would, perhaps, be an advantage to have had independent jacket water cooling systems on each engine.

The philosophy of floating a 24 V nickel cadium battery across the mains seemed to be a good way of solving the power source to instrumentation and alarms under failure conditions. Perhaps the author would explain why he did not revert to this system for the product carriers?

Finally, two product carriers had now been at sea with medium speed engines for approximately three years. It would be most interesting to have some published figures for the actual days out of service, the average ships speed both loaded and in ballast and the miles run per year. It would also be valuable to know the fuel consumption both of distillate and heavy oils and the figures for lubricating oil consumption. It would then be possible to make a comparison based on hard experience between medium speed engines and slow speed engines fitted in similar hulls.

Mr. K. R. Macintyres' first point concerned economics. Five years ago a paper was read to the Institute* which considered the application of medium speed engines to seagoing ships and compared medium and slow speed Diesel engines on a cargo carrier of approximately 55 000 dwt. It took into consideration many things, including engine maintenance, L.O. consumption, etc.

It was shown that if the predicted L.O. consumption was achieved, there would be a very small saving in maintenance costs compared with a slow speed engine, but if it exceeded slightly, there would be a penalty. Cdr. Short had raised a very valid point about the possibility of the medium speed engine having less time in dock, and therefore more revenue-earning time. In the small table in the paper he showed that with 18 month dry-docking periods and over a period of six years, the medium speed Diesel ship spent approximately 80 days less in ship repairers' hands than did the slow speed Diesel. A nominal adjustment was made to the Neumann and Carr Line in Fig. 24 to take into account the reduction in ship operating costs due to a saving of 13 days/annum in ship repairers' hands. It was a coincidence that the break-even oil consumption of 1·8 per cent was equivalent to the consumption of 100 gal/day quoted by the author. Did the author agree in general, with this economic comparison?

Regarding L.O. consumption, which was shown to be very important, what level of lubricating oil consumption was expected from the actions described in the paper and what level was expected ultimately from the engine?

What precautions were taken to ensure that an engine, isolated to be worked on was not turned by the running engine, by friction in the clutch or some other malfunction?

On water-cooled valves, had operating experience been satisfactory with the flexible hoses to the valves and the water seals on the valve stems?

* Neumann, J. and Carr, J. 1967. "The Use of Medium Speed Geared Diesel Engines for Ocean-going Merchant Ship Propulsion". *Trans.I.Mar.E.*, Vol. 79, pp 89–129.

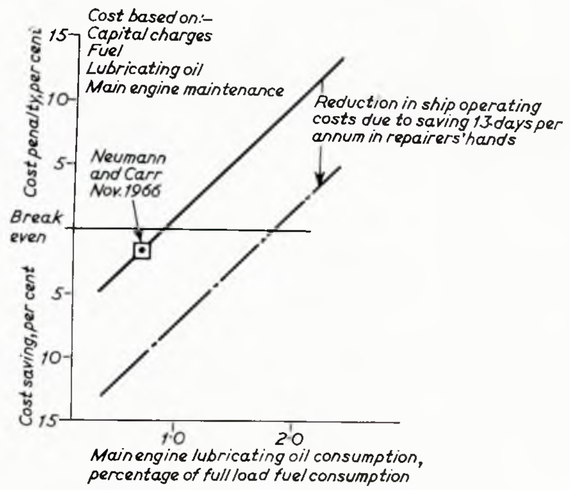


Fig. 24—*Operating cost of medium speed relative to cost of low speed Diesel—Bulk carrier*

With regard to U.M.S. operation, did the author's company have the control equipment serviced by shipboard engineers, or by specialists when the ship was in port? If the former, what training programmes did they have for these engineers, and what was the attitude of the engineers to servicing this control equipment?

Dr. H. M. Hiersig said that in the part of the paper dealing with gearboxes the author had reported difficulties with gear wheels and pinions which had arisen for various reasons. Dr. Pinnekamp had pointed out some of them. Certainly, gearing experts today knew of effective alterations in design, manufacture and installation which would overcome those troubles.

In addition, Dr. Hiersig said it should be pointed out that, according to his company's experience there were some other aspects which should be observed.

First, as they normally used torsionally elastic couplings, the calculation of vibration was of vital interest. In most cases it was possible to avoid resonance between natural frequency and excitation by the Diesel engine, at least by adaptation of the elasticity of the rubber elements. Their experience, however, led to the necessity of calculating not only the vibration situation of the system running under proper conditions but also under the possible condition that one of the cylinders did not contribute to the power generation. In this case the alternating torque could increase to an amount which could easily damage the coupling, gears or even shafts. This was the reason why they recommended (and it was also their own practice) the calculation of torsional vibrations also under abnormal conditions. The choice of coupling elements, their elasticity and their dampening properties had then to comply even with these most unfavourable running conditions (see Fig. 25).

Secondly, as limited torsional vibrations were essential for the performance of a propulsion system the control of torque was very useful indeed. The super-elastic couplings formed a very suitable means of measuring the torsional angle which indicated the torque transmitted by them. One merely had to observe the flashes of a stroboscope to see the angle and its alterations. But it was just as easy to transmit electric impulses from the primary and secondary part of an elastic coupling to electrical inductors and to transform the distances of time in which the impulses were received into values which indicated the torsional angle (see Fig. 26). The system functioned without slip rings and was transistorized; thus it was not submitted to wear and alterations.

The measurement of the torsional angle and the control of the torque was preferably useful in twin engine installations. As, depending on the rigid ratio of the firmly coupled twin gear, both engines had exactly the same speed, there was no indication of the load by a decrease in speed. The only simply available indication

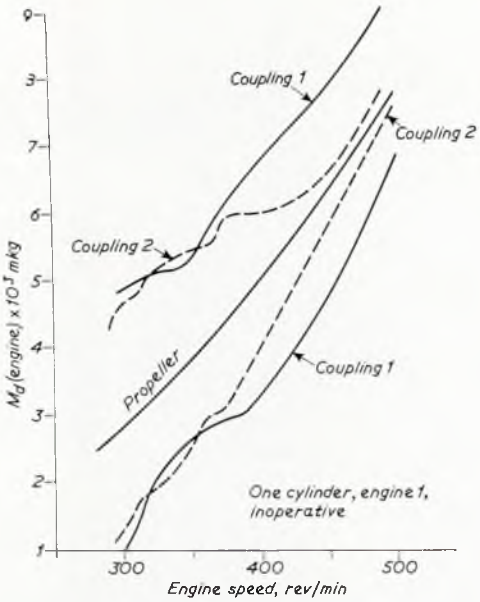


FIG. 25—*Alternating moments over the propeller curve*

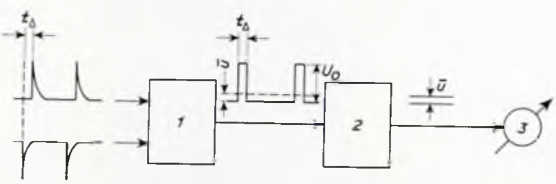


FIG. 26—*Phase angle measuring equipment (Kuhlenkampf and Reuter)*

of torque was, therefore, the torsional angle. In using this parameter both engines could easily be adjusted to equal power. With a small additional device the measurement could be used to give alarm or even to disconnect the clutches automatically if overload was encountered. Many of these torque control devices were in service to avoid overloading of engines and to give protection against heavier damage in case of a seizing engine (see Fig. 27).

Thirdly, the design of gears as described by the author included frictional plate clutches. The company's experience, however, was mainly based on pneumatically operated cone clutches which ran dry and were outside the gearbox. Several hundred of these clutches were now in service and the first had appeared eight years ago. The largest of them transmitted approximately 10 000 hp. The combination of elastic elements and friction parts formed a very compact unit, the most important feature of which was the short way which the generated friction heat had to go to be dissipated (see Fig. 28). Therefore, very high heat ratings had been endured especially on crash-stop manoeuvres.

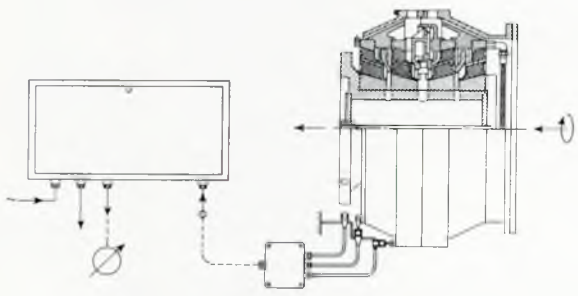


FIG. 27—*Torque measuring device*

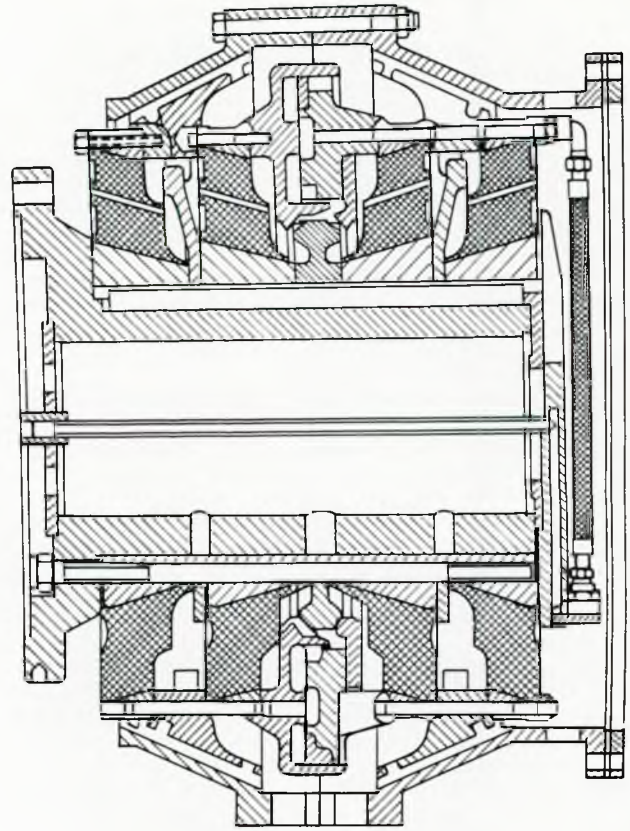


FIG. 28—*Highly elastic clutch coupling*

This led to the growing application of clutches not only with double or multiple engines but also with single engine installations, even in combination with fixed propellers.

In these cases the intermission of a clutch helped to reduce the way until the vessel stopped. Trials carried out in the sea near Tokyo showed a running-out distance of 2400 m without using the clutch and 1400 m with shifting the clutch. The tested vessel had a displacement of 21 500 dwt and performed with 8000 hp, a velocity of 16·5 knots. The engine stopped after 50 seconds; was reversed after 60 seconds; propeller reversal took place after 67 seconds; the vessel stopped after four minutes. The friction energy to take and to dissipate by the clutch was $8{\cdot}77 \times 10^5$ kgm, the clutch performed without notable deterioration (see Fig. 29).

Despite some problems which could never be avoided when introducing new techniques, experience had shown that many

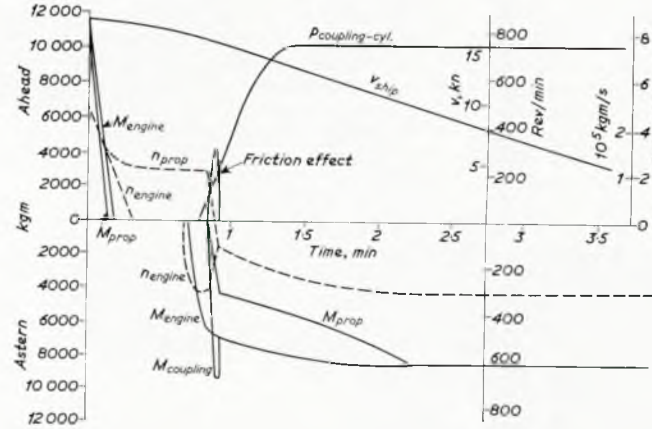


FIG. 29—*Speed, revolutions, torque and friction in the crash stop manoeuvre*

results from a twin engined installation. For this reason, service life in excess of 15 000 hours could be expected.

Concerning excessive lubricating oil comsumption which Cdr. Short stated had been in some instances over twice the figure claimed by engine manufacturers, indications were that, with the currently recommended piston ring pack and piston modifications, lubricating oil consumption was reduced to a more realistic level. Did the author's own experience confirm this? It was also stated that excessive losses of lubricating oil through maloperation of lubricating oil separators and filter automatic cleaning had been eliminated. Probably a greater lubricating oil loss than was generally recognized was lost per annum due to malfunctioning of oil hygiene equipment plus other random leakages, and if would be interesting to know what steps and safeguards had been taken to minimize or eliminate these losses.

It was appreciated by Mr. Davidson that in this type of machinery, liner wear rates were generally low and not regarded as a limiting factor. It was noted as a point of interest that the typical liner wear data given by Cdr. Short indicated that the S,E.M.T. seven position method of calibration was used. This, it was understood, had now been superseded in favour of a three position method. Both methods, however, utilized the lower position, outside piston ring travel, as datum thus compensating for temperature differential. It would appear from the data given that Cdr. Short's company assessed liner wear against nominal cylinder diameter; this being so, did he consider this to be a more realistic method of assessment?

It was interesting to note that the paper mentioned the use of carbon seals in the exhaust gas blowers in order to restrict lubricating oil consumption. This type of installation invariably involved the use of a coupled alternator, with the main engines operating at very low load in port, generating for cargo work. In some cases it had been considered that excessive blower fouling had occurred under these conditions and, in fact, blower lubricating oil sump levels had in some cases risen, indicating contamination by excess engine lubricating oil *via* the exhaust passages. It would be interesting to know if Cdr. Short had observed any differences between tanker and cargo ship operation in respect of blower fouling.

One final point—at the beginning of the paper Cdr. Short had quoted the principal advantages and disadvantages claimed for and against the medium speed Diesel engine, as opposed to the slow speed engine. Stated amongst the former were "reduced maintenance costs" and "low main component wear rates" whilst on the debit side "high consumption of lubricating oil, particularly important when using the expensive alkaline/detergent oils necessary for heavy fuel operation".

Mr. Davidson suggested that the former were complementary to the latter and felt sure that Cdr. Short would agree that these major advantages could not be achieved without the efforts of the petroleum industry in developing special oils for this type of application.

MR. D. ROYLE, B.Sc., A.M.I.Mar.E., said that the author had devoted a considerable proportion of his paper to information about the behaviour of the exhaust valves and the various attempts to achieve longer intervals between valve overhauls. It was fairly evident that a satisfactory solution was essential if medium speed engines were to operate with any degree of reliability on residual fuels available on a world basis.

The author had had difficulty in drawing conclusions for a number of reasons, one of which was the lack of detailed fuel analysis data on the fuels bunkered. Vanadium had a large influence on exhaust valve life and this was confirmed by the remark that chief engineers reported increased valve trouble after bunkering at Caribbean ports.

Technical personnel in the petroleum industry were well aware of the difficulties of supplying detailed fuel analysis on a routine basis at the time of bunkering, but they were always willing to co-operate with an engine builder and/or customer in making special provisions to ensure that first class technical data of the kind given in the author's paper was not jeopardized because one important facet of information was missing.

An illustration of this co-operation which was currently underway concerned the operation of a vessel powered by four medium speed engines working on the two stroke cycle of a well known make. The vessel operated exclusively in the western hemisphere and had been in service since the last quarter of 1970. Fuel bunkers were lifted every other week and each month one fuel bunkering sample was analysed and the average quality of the fuels supplied over a nine month period were summarized in Table III.

TABLE III—AVERAGE QUALITY OF RESIDUAL FUELS SUPPLIED TO 2-STROKE MEDIUM SPEED ENGINES

Viscosity Red. 1 at 100°F sec.	174
Conradson carbon, percentage	6·0
Asphaltenes, percentage	4·1
Sulphur, percentage	1·47
Ash, percentage	0·045
Vanadium, ppm	218
Sodium, ppm	19

The fuel ordered was Intermediate 2—that meant it had a maximum viscosity of 200 Redwood I at 100°F. The various fuel features listed were of particular interest to users of medium speed engines. The complete lack of fuel combustion difficulties and the excellent condition of the piston and liner covering both wear and cleanliness showed that the level of Conradson carbon and asphaltenes of 6·0 and 4·1 per cent was satisfactory. The sulphur level of 1·47 per cent was rather low compared with many fuels of similar viscosity obtainable in Europe. The average vanadium content was 218 ppm.

From Fig. 8 in the author's paper which related exhaust valve failures to engine hours the number of valve failures in either *Luminous* or *Lustrous* was 28 after 5000 hours. There were 48 exhaust valves in each ship. The vessel with four 9-cylinder 2-stroke medium speed engines suffered only seven exhaust valve failures after the same running hours, although there were three times the number of exhaust valves as on *Luminous* or *Lustrous*. This low level of exhaust valve failure rate was achieved using valves which were neither water-cooled nor fitted with rotorcaps and the average interval between overhaul was 3500 hours. No fuel additives were used. The owner now intended to increase the viscosity of the fuel bunkered, and it was expected the average vanadium content would rise to around 250 ppm.

MR. J. BERRING, A.M.I.Mar.E., said that this paper was of great interest to his company and had resulted in a comparison with their own results and experience. He did not quite agree with the introduction, regarding the advantages of medium and slow speed engines. If, for instance, the ability to proceed under own power in case of a breakdown was considered, there was the following range of propulsion machinery available (see Fig. 33)

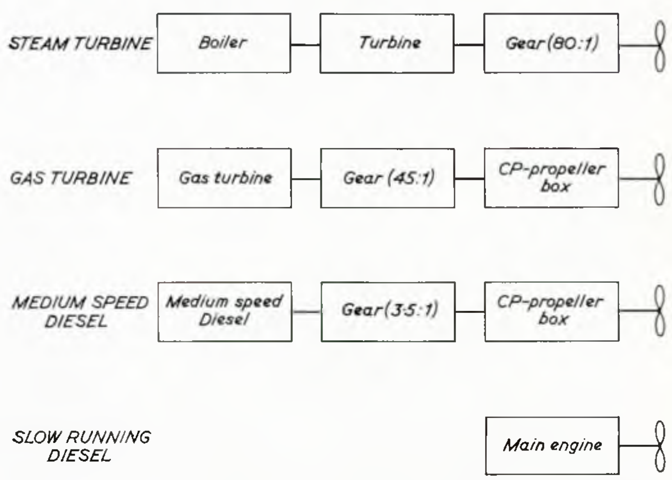


FIG. 33—*Main components for different propulsion machinery*

TABLE IV—OVERHAULS BETWEEN 24 MONTHS DOCKINGS REQUIRING MAIN ENGINE TO BE STOPPED 10-CYLINDER LARGE-BORE ENGINE

Part of engine	Number	Hours between overhauls	Time of overhauls per unit	Provided	Men of work	Number of overhauls between dockings	Total hours
Piston	10	8000	5	spare piston and cover	4	1	50
Fuel-valve/cylinder	3	3000	1	spare valves	2	4 (less)	30
Exhaust-valve/cylinder	1	4000	1	spare valves	2	3 (less)	20
Air cooler	4	4000	1	spare set	3	3	12
Turbocharger	4	15000	8	spare set	3	–	—
							112
						Annually: 56 hours	

Figures based on actual operating experience with large-bore engines. With the new design, engine type K 90GF, it was expected to be able to increase the different periods between overhauls considerably.

in a steam turbine there were three units connected in series, and if a major breakdown occurred, the ship would be unable to proceed under her own power. These were the gears, the turbine itself and, if it was a one-boiler ship, the boiler. In a gas turbine ship there was the c.p. propeller box, the gear and the turbine. In a medium speed Diesel ship there was the c.p. propeller box and the gear, but in the case of a multi-engine installation, a major breakdown on one engine would not prevent the ship from proceeding under her own power. With a slow running direct coupled Diesel engine, only a broken crankshaft aft of the chain drive or a complete breakdown of the chain drive itself would prevent a ship proceeding under her own power.

In other words, considering single screw ships, the risk of not being able to proceed under own power, even at reduced speed, was considerably less for a direct coupled slow running Diesel installation than for any other type of propulsion available today.

When comparing maintenance for medium speed and slow speed engines, the author had been figuring on 17–21 days off hire for the slow speed engine, which was not quite in agreement with the operational experience gained in their own company. Table IV showed the figures for overhauls between 24-month dockings requiring main engine to be stopped. The total hours of 112, giving an annual 56 hours, were based on operating experience with large bore engines.

When discussing the advantages of having a bow thruster, it had to be remembered that the bow thruster caused a reduction in the ship's speed. A Danish shipowner had found a reduction of ¼ to ½ knot at normal speed and loaded condition.

It was interesting to learn about the author's experiences regarding gearboxes and their installation. Mr. Berring's company had, for similar medium speed installations of their own, been faced with the same problems and they agreed with the importance of perfect alignment and proper mounting of the thrust bearing. They now had nine trouble-free ships in service.

The author had shown a preference for engine driven primary lubricating oil pumps and, it was understood, with no automatic shut down. Mr. Berring's company was of the opinion that mechanical failures in the pumps and their drives could very well happen and therefore recommended a shut down also for this arrangement.

Injector pipe failures at end connexions had also been found earlier in several cases in their engine types, but after introducing a high pressure coupling, as shown in Fig. 34, these troubles had stopped. The coupling ensured a good guidance of the pipe in such a way that the threaded pipe end was not exposed to vibrations. The connexion was furthermore made so that in case of leakage of high pressure oil to the protective shield, a special arrangement activated an automatic lift of the corresponding fuel pump which would cause a reduction of the exhaust gas temperature, and an alarm.

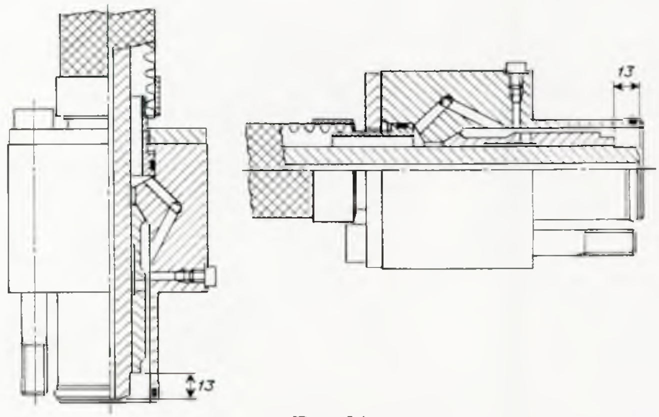


FIG. 34

In the paper excessive lubricating oil consumption was mentioned. It would be interesting to learn of actual figures. Excessive lubricating oil consumption had also been found in the writer's company's medium speed engines. After adjusting of scraper rings and after the vent pipes from the crankcase and bottom tanks had been increased in size, the total consumption was 1·5 g/bhp h. As cylinder lubricators were fitted on these engines, only 30 per cent of the mentioned 1·5 grams of the oil consumption comprised the most expensive alkaline oil.

The figures for piston ring groove wear given in the paper indicated that reconditioning of ring grooves would have to take place after around 8000 hours. Had this been taken into account in the original planning of maintenance?

MR. J. E. GALLOIS, M.I.Mar.E., said that he had discussed the draft of the paper with the author and had been surprised to find the large number of difficulties which Commander Short had met. To gain a better knowledge of them, he had asked if S.E.M.T. could send an engineer on board one of the ships and the author agreed. Mr. Gallois was very grateful, as it had been very useful.

As the author had said, the ships had been away from Europe a very long time, not returning there since the first sailing.

The engineer went on board the product tanker *Luminous* and found that since 15 January 1969 (the ship's first trip) the two engines had had more than 7500 running hours a year—quite significant for such ships. He also learnt that the main engine was used continuously during cargo unloading, running at an average of about 100 hp/cylinder, i.e. about one fifth of full power. Heavy fuel was used, as at sea.

The fuel bunkered on the ship was 1500 s Redwood, with

TABLE V.—ANALYSIS OF SAMPLES OF DEPOSITS ON TURBOBLOWER

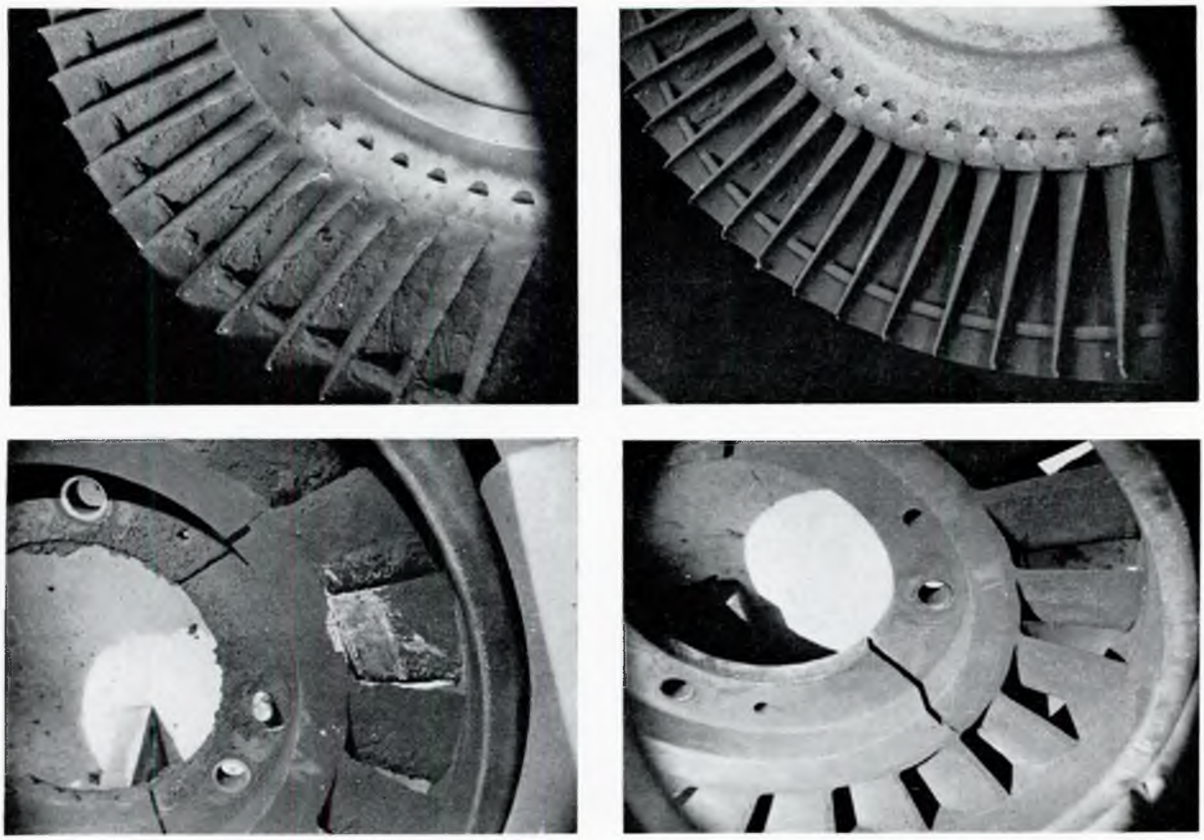
	Deposit 1 taken on the back of the nozzle before cleaning	Deposit 2 taken in the gas chamber under the wheel	Deposit 3 taken on the blades of the turbine wheel before cleaning
Quantitatively:			
Carbon and grease	3·70%	81·40%	81·60%
Vanadium	10·36%	1·63%	1·70%
Sulphuric acid (SO_4)	46·28%	5·60%	4·94%
Iron	1·85%	3·77%	3·91%
Qualitatively:			
Sodium	large amount	small amount	large amount
Calcium	traces	traces	traces
Barium	traces	traces	traces
Nickel	small amount	small amount	small amount
Chromium	none	none	none
Molybdenum	traces	traces	traces
Silicon	small amount	small amount	small amount

probably a relatively low vanadium content, since bunkers in some Far Eastern ports did not have a high vanadium content, but could have a relatively high sulphur content. Analysis of the samples was under way.

The author had said that perhaps the turboblower fouling was special to the Napier turboblowers. Mr. Gallois would say not, S.E.M.T. had met it also on the Brown Boveri type. It depended on service conditions, exhaust gases, temperatures, fuel used etc., rather than on the type of turboblower. It was a relatively old problem and they had solved it about five or six years ago.

They had analysed the deposits (Table V) which clearly came from the ash of fuel with a relatively high vanadium and sulphate salts content. They also found some calcium and some barium, from the additives in the lubricating oil.

The presence of sodium salts was very important because it was in sodium sulphate form and could dissolve in water. During the servicing of the engine they tried water injection. Before and



(a) (b)

FIG. 35—(*a*) *Deposits in turboblower;* (*b*) *deposits in same turboblower after water washing*

after washing the turboblower was dismantled. Fig. 35 (a and b) showed that the washing procedure was very successful. The water was injected for about 20 minutes, the speed of the turboblower being about 0·25 per cent of its service speed. This decreased the speed of the ship, which was sometimes inconvenient, but when the cleaning was carried out regularly there were no turboblower fouling problems. He was sure they would find this with *Luminous* where the necessary devices would be installed.

On this particular ship no cleaning took place—even manual—for 10 000 hours. On the visit which was made, it was found that the nozzle had been overheld, and that more than 50 per cent of one area was completely clogged by deposits. The water washing installation would solve this.

The problem of liners cracking was just a question of original faulty tightening and not a real technical problem.

With regard to lubricating oil consumption—in the beginning they had a particular set of rings (see Fig. 7) which gave a consumption of 1 to 1·2 g/hp h for new engines, but the consumption was not always the same, from one engine to another. This could be explained, for example, by too great a stiffness of the scraper rings when the liner could be more or less deformed. This was modified by using less stiff scraper rings and the lubricating oil consumption was now below 1·0 g/hp h.

Fig. 36 showed the results obtained with the first ship equipped with these rings. She was *Pointe Marin* of the French line and, after 7000 running hours, the port engine gave 1·2 g/hp h and the starboard engine 0·7 g/hp h. The owner gave these figures. In this particular ship they used the engine at constant speed and also in the port whilst unloading, as the author did with ships of the *Luminous* type.

Since then, new tests had been made to find the minimum lubricating oil consumption consistent with a low wear rate. These experiences showed that not only could medium speed engines have a relatively low consumption, but also fairly high speed engines. They were continuing with these tests and had found they had to limit consumption to 0·6 g/hp h minimum and this could be adjusted by the stiffness of the rings. To prevent an increased consumption with service time, they had made tests with chromium-plated scraper rings. The results were very good and this would be adapted in the near future to all PC engines.

To decrease operating costs, the total base number (TBN) had been decreased from about 40 to about 20. So the price of the lubricating oil had decreased by 15 per cent, which was interesting The wear rates remained very low and the oil behaviour was good.

Regarding the exhaust valve figures given in the paper, they had somewhat different figures. After 20 000 hours they found, in *Luminous*, that only 59 valves had been replaced due to burning to some other reason not known, which gave an average valve life time of more than 15 000 hours.

They also found that, by mistake, the coating used for the seat was not to specification and that on some, the seat regrinding for maintenance was so strong that there was no coating left on the seat. This partly explained the number concerned and showed that the normal valve life time, if the crew were well informed, could be more than 15 000 hours.

The author had given some results with water cooled valves and mentioned that this cooling process definitely solved the problem of seat burning. However, he had said that due to the cooling, there was heavy corrosion of the stem that decreased the valve life-time and increased the operating costs. He had also mentioned that ten valves with a new coating had been put into service. Mr. Gallois was very interested to know that these valves were perfect after 2500 hours. This was only a beginning because he knew, from other ships, that the valves would be still in the same state after 8000 hours and they could hope to maintain these valves, without overhaul for more than 12 000 hours.

This result would be obtained with every kind of fuel, even with 300 to 400 ppm of vanadium, because the temperature of the valve seat was 380°C (716°F) maximum. That was the only solution to the problem of valve seat behaviour with any kind of fuel.

Table VI showed the wear results reported from *Luminous*. There was a radial wear of only 0·02 mm/1000 h on the first ring, and could use these for more than 15 000 hours in spite of the difficulties with exhaust valves and turboblower fouling. This meant that it was possible to extend the time between overhauls to more than 15 000 hours.

In his introduction, Commander Short had said that he had had connecting rods which were two pounds too light, but as they weighed 500 lb, one could accept this, as it was only 0·4 per cent. This difference in weight arose from a drawing alteration which made the connecting rod easier to forge and machine.

With regard to licensees and co-operation, S.E.M.T. had to progress gradually. Mr. Gallois thought that a big step forward had already been made in the standardization of parts used on

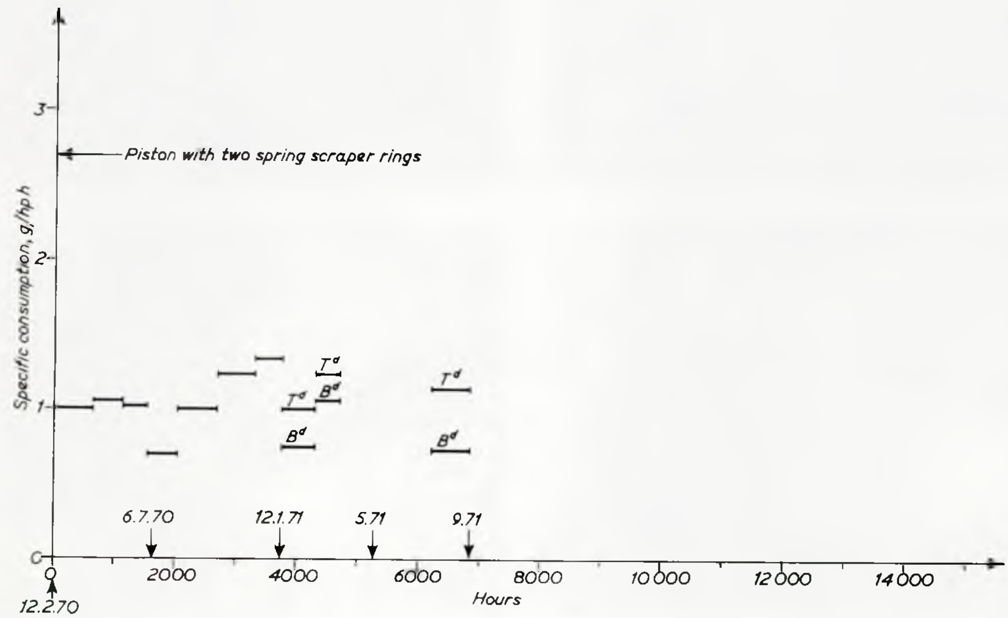


FIG. 36—*Specific lubricating oil consumption against running hours of two 18PC2V engines equipped with pistons using two spring-loaded scraper rings*

TABLE VI—AVERAGE WEAR RATE MEASURED ON "LUMINOUS"—TWO ENGINES AFTER 20 000 HOURS

Engine wear 100/1000 h — Tanker *Luminous* 2 × 12 PC2V — 20 August 1971

	Liner wear	Pistons: Groove wear: First Groove	Pistons: Groove wear: Second Groove (with copper plated ring)	Ring equipment (radial wear): Top ring: Rubbed (304 982 without notch)	Ring equipment (radial wear): 1st compressor: Copper plated (307 966)	Ring equipment (radial wear): Scraper rings: Upper (304 747)	Ring equipment (radial wear): Scraper rings: Lower (304 747)	Operating conditions
								total hours port engine 20 291 on 20.8.71; starboard engine 20 012
Port	0·80 (11)	0·33 (6)	0·77 (6)	1·83 (12)	3·63 (12)	1·50 (10)	1·30 (11)	Recording effected between 5000 and 20 000 hours
Starboard	0·80 (10)	0·28 (8)	0·72 (8)	1·91 (10)	3·88 (9)	1·50 (10)	1·50 (10)	FOL. 1300/1500 Rw; lubricating-oil Esso Tromar SR 30; Manoeuvres with heavy fuel oil
Average	0·80 (21)	0·30 (14)	0·74 (14)	1·86 (22)	3·74 (21)	1·50 (20)	1·40 (21)	

engines in Japan, the U.S.A. and Europe, as could be seen by shipowners using European engines in Japan, or by European shipowners using both European and Japanese built engines.

All data on engines in use were welcomed by S.E.M.T. who would go anywhere to get them, with the shipowner's approval.

MR. J. H. BARKER said that the author had made a comparison between the experiences with his two cargo liners and two product carriers. It should be remembered that the cargo liners had 14-cylinder engines and the turbo blowers would have four turbine entries, whereas the product carriers had 12-cylinder engines with 2-entry turbines. The total nozzle area for the 14-cylinder engines would be 50/60 per cent larger than the total nozzle area for the 12-cylinder engines. The flow of gas carrying the contaminant would be in proportion to the number of cylinders, ie only 17 per cent greater. Therefore the effect of nozzle fouling due to high lubricating oil consumption would tend to change the turbine efficiency on the 12-cylinder engines more rapidly.

All four ships had been experiencing high oil consumption, and the product carriers, because of cargo handling, did considerably more low load running than the cargo liners. This again would tend to increase the nozzle fouling rate with the 12-cylinder engines.

Difficulties and delays from stripping and cleaning blowers arose from the compactness of the installation, which produced the necessity for the complete removal of the turbo blower to clean the turbine end. This difficulty of servicing must be common to both classes of vessel which had blowers of generally similar construction, although those on the cargo liners were somewhat larger.

It was perhaps, of interest to owners contemplating this type of machinery arrangement in future to consider the advantages of the most modern blower designs developed since the ships with which the author was concerned, were planned (see Figs. 37 and 38).

The method of removing the centre section containing the complete rotating assembly and the bearings should be noted (Fig. 38). On the right were the turbine outlet and inlet casings which remained on the engine, leaving all exhaust and water connexions undisturbed. A spare centre section could be held on board as a service replacement. The complete change operation should not take more than one hour.

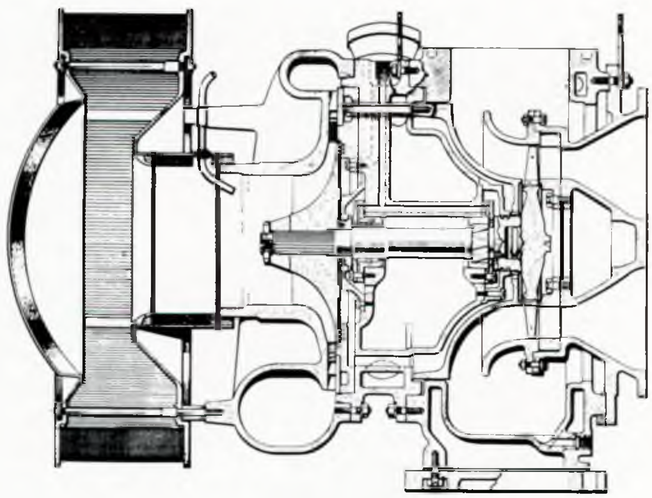


FIG. 37—*Napier SA type turboblower*

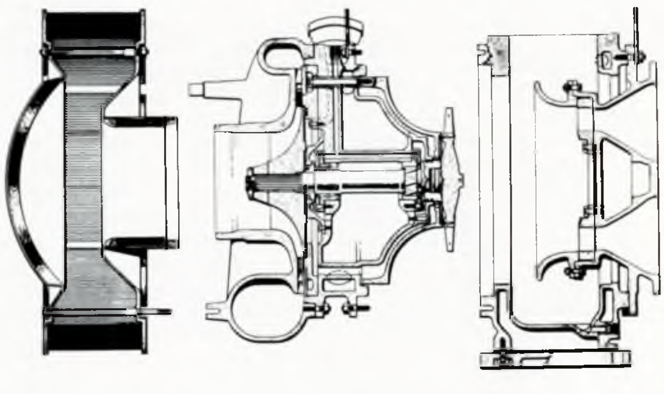


FIG. 38—*Removal as one assembly of internal components of SA type turboblower*

The author mentioned the use of non-standard carbon seals. This was brought about by a very restrictive stipulation by the engine builder concerning the period between oil examinations during U.M.S. operation. As a result, the carbon seals were adopted to give the best possible margin of cover in this respect, but, by their nature, they were more susceptible to damage when the blower was stripped and, for the reasons given above, such stripping had been necessary at more frequent intervals than was originally foreseen. At the same time, there had been no problem concerning blower lubricating oil consumption, and it was therefore proposed to adopt alternative cast iron components to overcome the problem of the fragility of the carbon. This would also prevent the recurrence of manufacturing problems which were experienced by the manufacturers of the carbon seals.

It was noted that water washing was now being fitted to the gas and air sides. Wide experience elsewhere showed that turbine water washing was generally a very effective means of maintaining turbine efficiency between normal overhaul periods, even when severe deposition was experienced of products arising from unfortunate combinations of fuel and alkaline oils. Nevertheless, it was obviously desirable to attempt to reduce engine lubricating oil consumption to a satisfactory level.

Correspondence

Mr. P. R. Belcher wrote that since his company did not lubricate any of the ships which were the subject of Commander Short's excellent paper, he had no first hand knowledge of their lubrication costs. Nevertheless it appeared that a misunderstanding had crept into the first paragraph under the heading "Lubricating Oil Consumption" which might be worthy of reconsideration in the final version of the paper.

He agreed that the alkaline oil used in medium speed engines was about 20 per cent more expensive than the system oil used in slow speed engines; however, the latter was only a minor part of the cost of lubrication of slow speed engines, the major part being the consumption of highly alkaline cylinder oil, which was about 15 per cent more expensive than that used in medium speed engines.

Typical oil consumption figures for slow speed engines were 0·5 g/hp h for cylinder oil, plus 0·2 g/hp h for system oil, compared with about 2 g/hp h implied in the paper for medium speed engines; thus Commander Short's broad conclusion was still valid, but not from the same reasoning.

Mr. J. Neumann, B.Sc., A.M.I.Mar.E., stated in a written contribution that he had rarely read a more fascinating paper, and wished particularly to thank the author for discussing so fully his own and his company's requirements which led to the selection of the equipment and overall ship designs.

Regarding the factors which influenced the equipments selection, would the author agree that another feature to be considered was the likelihood of future use of the particular equipments in other ships yet to be built? All the factors listed were important, and it was to be expected that one or another might dominate in any particular case. It would be of great benefit if the author could give any hint of a procedure or any strategy that he might have found helpful in making the deciding judgement as quantifiable and as objective as possible.

The question of lubricating oil pump drive, engine driven or motor driven which was discussed by the author, had an uneasy parallel in the design if lubricating oil arrangements for the main gearbox in naval ships. For various reasons an overhead gravity lubricating oil tank was not attractive for such an application, and an engine-driven pump not only presented problems as regards the physical arrangement of the drive, but also required to be greatly "oversize" to ensure an adequate oil supply down to a reasonably low shaft speed. Past experience suggested that would be foolhardy aboard ship to rely absolutely on the assurance of a continuous electrical supply. He believed this would remain true whatever the integrity and standby sophistication of the electrical system; so long as there were human operators, who, in the heat of some immediate crisis, when they were trying to react extremely rapidly to some emergency, in so doing, might, in fact perpetrate an operating mistake. Recent discussion in the technical press in respect of mistaken in-flight shutdowns of aero engines indicated that this problem was not exclusively a marine one. The solution in the naval problem to which he referred was tending towards the use of a safety air-turbine driven pump, which was triggered automatically and which relied on a modest air reservoir to provide a period of a few minutes of operation.

Mr. C. W. Herbert, M. I.Mar.E., wrote that what had been done most ably in the first part of the paper was to formalize the thought processes which essentially preceeded logical decision.

There were inevitably some risks involved in any process of advancement, as Commander Short had revealed in the second part of his paper, and the statistics given confirmed that he and his colleagues had applied commendable diligence in seeking the ultimate truth.

A summary of the logic followed in developing the actual machinery arrangement would have been most useful. The writer had the privilege of looking over one of the tanker installations and in that ship this important aspect of design was certainly not neglected.

The need to create an environment in which the machinery could be supervised and serviced to best advantage called for a great deal of thought and careful attention to detail, and many of the medium speed installations made to date were quite deplorable from this point of view.

With regard to the table depicting off hire time under the heading "Advantages Expected from Installing Twin Medium Speed Diesel Engines in Products Carriers", could a summary of the work load responsible for the number of days in hand for the slow speed and medium speed alternatives be given? Why also, was this analysis worked over six years rather than eight, to include two propeller survey periods?

It was inferred from Fig. 3 that the engine services were common with one working and one stand-by pump serving both engines.

Would it not be more in keeping with the reasoning applied under the heading of seaworthiness to separate the auxiliary loops to each engine and dispense with stand-by pumps? This reduced capital cost of pumps and pipework and considerably simplified automation.

Again, was it not advantageous to add engine driven water and fuel pumps to the preferred engine driven L.O. pumps? There would of course be a need for some smaller electric pumps for starting and cooling down, but the reduction in electric load and electrical installation cost would be significant.

With regard to the gearbox history referred to, the writer had to face the same basic situation in ships built somewhat later, and was most grateful to Commander Short for explaining his assessment of the problem at that time.

The remarks on tumbling were of particular interest. It would be of inestimable value if photographs showing the distinctive marking associated with this could be published.

A word about the modified clutch drill and the eventual clutch modification would be appreciated.

In several ships with which the writer had been concerned there had been evidence of gas side fouling of the turbo blowers, as the author reported. Water washing seemed beneficial when it was commenced on a clean blower and regularly applied. This problem had been found to be worse in ships whose schedules demanded long periods of low load operation as in a Panama transit, due it was felt, to pumping of lubricating oil into the exhaust system at quite low temperatures.

In consequence the writer had been concerned at the prospect of carbon deposits preventing the proper operation of corrugated expansion bellows in the manifold sections. Had any problem of this nature been encountered?

Regarding rotocap valves, the writer's experience had not been so depressing as the paper reported, and the work of reconditioning had been considerably lessened since Swedish grinding equipment, which ensured narrow band contact towards the middle of mating faces and eliminated hand grinding, was made available

In one ship, with two engines of the type referred to in the paper, the original WC valves were changed by the builders to RC at the end of guarantee.

In another engine of the same type it was found that the seats could be distorted when the cage was fitted.

In two otherwise identical ships each fitted with two Fiat C. 420 engines with RC valves, only one valve required to be changed during 4500 hours operation before guarantee inspection, and the majority inspected at that time did not warrant attention.

When comparing the relative performances of water cooled and rotocap valves in the PC2 engine, it was important to note that the valve seats in this engine were uncooled.

In the Fiat engined ships referred to, the rotocap valves were fitted in fully cooled cages. Each cage was made of two steel components welded together after precise internal machining of the cooling annulus. The arrangement provided high velocity coolant circulation in close proximity to the seat and combined the benefit of heavy cooling of the interface with the advantage of valve rotation.

The three ships had operated very similar schedules and used 1500 fuel taking some bunkers in Panama and U.S. West Coast.

The service statistics given in the paper could only be the result of disciplined reportage and it was hoped this would be properly appreciated by the manufacturers concerned. All too often one received elaborate diagnosis without the plain statement of fact necessary to support it. There could be little doubt that one of the main difficulties facing marine equipment manufacturers was that of obtaining accurate data on service defects. The establishment of service contracts with engine builders was perhaps one way of narrowing this communication gap to the benefit of all concerned.

Captain W. A. Stewart, C.B.E., R.N., M.I.Mar.E., said, in a written contribution, that he was interested to hear of the author's troubles with exhaust valves and wondered if they could not be solved by the adoption of some other solution rather than his Hobson's choice between rotocap pre-cooled or fixed water cooled valves.

He had not mentioned the combination of a water cooled cage and valve seat with rotocap valves of high conductivity Cr, Ni, Si material which was used both by Mirrlees in their K major series and M.A.N. in their 40/54 or 53/55 medium speed engines.

This would seem to be the right compromise in order to prevent sulphur corrosion at low outputs, and avoiding the destructive aspects of sodium vanadates at the highest outputs and temperatures whilst using low quality residual fuels.

It might also be possible to add a special thermostatically controlled bypass for the seat (or valve) cooling to help the low output condition and it would be interesting to know if this had been tried.

It was also surprising to learn of the need to place such operational importance on accepting almost any grade fuel at any port. Slightly larger bunker capacity could possibly avoid this necessity, even when on charter, so that certain well known areas of high sodium and vanadium fuels could be avoided.

Mr. J. G. Poole, A.M.I.Mar.E., referring to Commander Short's remarks regarding the vulnerable gearbox, said it seemed to him that gearboxes fell into this category mainly because the owner, shipbuilder and engine builder at times chose to make it the "necessary evil" part of the installation. Engineering judgement was clouded by the requirement of choosing the lowest priced box in order to show the overall concept to its best advantage. With demurrage rates of up to £3000 per day, this must be bad economics.

In other cases, installation and associated component defects, all outside the gearbox manufacturers control, were attributed to inadequate gearing.

Gearing with excellent records of freedom from defects was available. One could point to the numerous trouble free turbine geared installations produced in the U.K. before the smaller steam turbine vessels became uneconomical as well as typical of charter times of two hours in two years in recent bulk carrier geared Diesel installations to prove the point.

The problems associated with gears at sea were well known to the traditional marine gear builder. This experience had been utilized by his company in developing a series of gearboxes which to date had been virtually trouble free. The basis of the design was an extremely rigid base capable of maintaining the gear element alignment irrespective of the ship structural condition. Telescopic sightings were made to ensure that all planes of gearing in the manufacturing shop were duplicated in the vessel. To date, using this technique excellent tooth contact conditions had been maintained in ten installations, the earliest of which had been at sea some five years.

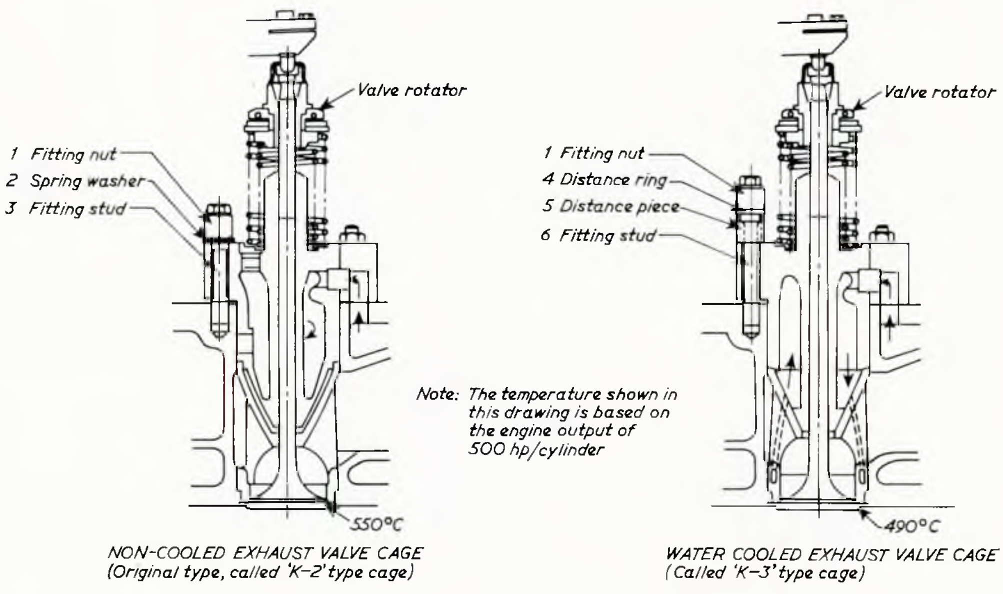


Fig. 39—*Comparison of exhaust valve types*

Regarding the position of the integral main thrust, contrary to Commander Short, the forward position was favoured. Besides giving a neat engineering arrangement, it also allowed the engine and thrust seating to be combined, so ensuring that the thrust load was transmitted to a large area of the hull of the vessel. The major problem when locating the thrust block aft was the effect of the discontinuity in the ship's tank top structure caused by the gap for the main wheel. Unless specially reinforced thrust seatings were employed, an extremely flexible arrangement could occur and in one such controllable pitch propeller installation with an aft integral a movement of 2 mm was measured when moving from ahead to astern.

Commander Short referred to the tumbling action of pinion when the clutch was engaged, could he elaborate on this point?

With respect to the general defects associated with Diesel gearboxes Mr. Poole would like to suggest that many problems were associated with misalignment due to thermal, dynamic and hull stability problems. This aspect required further detailed investigation from two view points, firstly the gathering of practical results to corroborate calculations, and secondly, the development of flexible couplings capable of absorbing those misalignments which were calculable and also capable of providing a contingency for those which could not be calculated. It was also necessary for the shipbuilder to work closely with the engine and gearbox manufacturers in the design stage to ensure that they were fully acquainted with the most adverse misalignment and vibration characteristics which were likely to occur in practice so that this could be taken into account in the gearbox design, and also in the selection of the clutches and couplings.

Mr. K. Yamada wrote that his company had manufactured 180 sets of PC2 engines. During this time they had met with difficulties of piston seizures, excess consumption of lubricating oil as well as premature failures of exhaust valves.

As far as exhaust valve failures were concerned, they considered the most important countermeasures to be intensive cooling of the valve seats and minimizing valve cage deformation.

Water cooled valves developed by S.E.M.T. had been installed for three engines manufactured by his company and they had experienced stem corrosion by sulphur attack and breakages of the rubber hose for cooling water, although the seat condition was very good. The reason why they did not apply this type of valve on their serial production engines was not due to these troubles but to the very high price.

After various experiments extending over three years and in contact with S.E.M.T., his company had succeeded in developing a water cooled exhaust valve cage. As could be seen from Fig. 39, the conventional cage consisted of two parts, the upper water cooled holder and lower non-cooled seat, whilst the new cage was fabricated into one block by welding and the seat was water-cooled.

With this cage, the temperature of the valve seat was about 60 deg. C lower compared with the normal cage. Another advantage of this cage was that there was no thermal expansion and no deformation.

Exhaust valve records (see Fig. 40) for one ship fitted initially with K2 rotocap valves (non-cooled) and later, at 4565 hours, with the new K3 valves (cage cooled valves) showed that performance recorded was:

	Hours	Failures
K2 valve	4565	18
K3 valve	9215	6 stems burnt
		4 water leakage at cage

Regarding water leakage, it was found that this trouble was due to the thickness of the rib containing the water passage. This problem had been eliminated by the design modification.

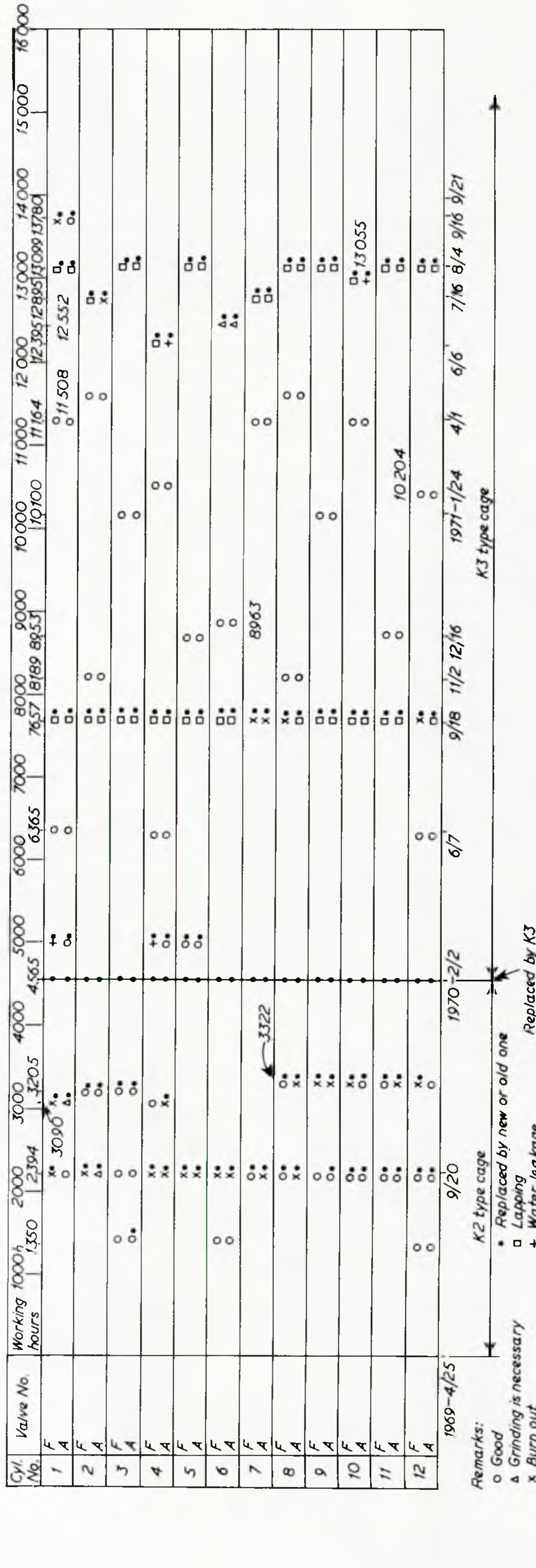


Fig. 40—*Running record of exhaust valves for 12PC2V—5130hp; 500 rev/min*

Author's Reply

Commander Short in reply to the discussion, said that he would like to make and stress one important point about a paper of this nature—that one was dealing principally with the things which went wrong and not with things which went right. Managment consultants had a high-faluting "gobbledegook" expression of "management by exception" which covered this procedure. In spite of a variety of things going wrong, the engines reported upon had worked and worked well. It was all right, on an occasion such as this when talking to mature engineers, to admit to problems of the nature described, as they could interpret and put things into perspective. Unfortunately, however, when those with insufficient experience or from other disciplines, e.g. accountants, lawyers etc got hold of what appeared to be one glorious defect list, they were liable to misinterpret things. As a general remark, he was always delighted to hear how well the PC2 engine competitors had solved its problems, particularly exhaust valves and lubricating oil consumption. One then knew that as one persevered with one's own equipment, one was not, as it were, trying to push water uphill.

He was indebted to Mr. Butler for his explanation about the joint problem and its solutions. As stated in the paper, ships were not provided with a complete fuel analysis on bunkering, but it was known from random sampling that they had been covering practically the whole spectrum, using 1500s Redwood 1 fuel, between the extremes of Caribbean high vanadium low sulphur to Middle East low vanadium high sulphur. As far as was known, the characteristic of a high ash content was the only significant one likely to be encountered which has not been included in the fuels for these ships at some time or another. With reference to Mr. Butler's final remark, Commander Short noted that he was not referring to the engine type as being relatively untried, but only to their application of it. The engines and the general arrangement had, of course, been successfully used in ferry ship applications, in particular, for some time.

Mr Mack had made a most valuable contribution in respect of control systems in general. The author said he had played into his hands somewhat by talking about throwing the propeller from full ahead to full astern. Of course, proper safeguards had been designed into the system. There was a limiting speed of movement of the control lever and automatic reduction of pitch if the engine was overloaded. Once one put controls into the hands of the officers on the bridge, it was no good saying to them "Do it slowly' and expect them to do so in an emergency. If they saw something coming towards them, they would pull the pitch lever right through, whatever their instructions. It was not a matter of swashbuckling at that moment, but *sauve qui peut*. The author was a firm believer in simplifying such matters for the command, especially if he was on board at night or in any fog approaching the Straits of Dover. The command at such moments had enough to do and think about and it was the duty of engineers to design out unnecessary problems or worries for them. His reason for wishing to demonstrate this manoeuvre was in line with Diesel generating set governing and voltage regulation tests, overspeed trip tests etc. Perhaps it did only happen so severely once in the lifetime of the machinery, but it had at least been demonstrated, under controlled conditions, that the safety arrangements were adequate as designed. The majority of Mr. Mack's remarks were more directly applicable to variable speed engines. Their constant speed engines, although suffering from marginally decreased propeller efficiency, except at the design point and from higher lubricating oil consumption at low powers, did have more easily solved control problems. He was interested in Mr Mack's recommendations regarding engine monitoring equipment. As he had previously and continuously made it quite clear, he was not in the engine design business, but his experience told him to be wary of misinterpreting the effect of transients. He recalled the United States Navy elaborately instrumenting the structures of a Royal Naval aircraft carrier fitted with a steam catapult. According to these instruments there was no bow left after lifting the catapult, if one misinterpreted the transient stresses recorded. Was an infrequent transient combustion pressure of ten per cent in excess of full power outwith the design of the engine to tolerate? If it was not, what should be done? Should the engine design be altered—should its rating be lowered—should increased and more complicated control gear be fitted and maintained? His plea for simple controls spilled over to simple and easily interpreted instrumentation for general use at sea. He fully supported Mr. Mack's plea for properly designed, conducted and reported machinery trials at sea.

Mr. Castle had mentioned the advantages to be extracted from a multiplicity of small lightweight cylinders if proper maintenance facilities were designed into the engine room and workshop arrangement. The author could confirm that every effort was made to do just that and they were improving by experience all the time, although probably not as rapidly as some of the ships' engineer officers would like.

He agreed about the elimination of machinery control room (M.C.R.) windows. Where this was done, there was additional space for mounting mimics, instruments etc. Some people advocated the next step as moving the M.C.R. to the bridge. He did not support this, as when the M.C.R. was manned as it had to be from time to time, he, like Mr. Castle, favoured a position fairly near to the machinery so that operators had easy access between the two. He thanked Mr. Castle for his correction upon the benefits and effect of using the running-in compound.

Commander Short did not regard gearboxes as being any more vulnerable than other parts of an installation which were not duplicated. The summary in the introduction represented an unedited listing of claims put forward by opponents and protagonists of medium speed Diesel engine installations. With regard to the pinion tumbling problem, he understood that this was to be dealt with later in a contribution by Dr. Pinnekamp. He was generally in agreement with Mr. Castle's remarks about correct design of double bottoms. The discontinuity introduced by the main gearwheel must be adequately compensated for. He maintained his preference for the thrust block aft of the gearbox for the reasons advanced in the paper and agreed with Mr. Castle that, with a properly designed and located thrust block on an adequately strengthened double bottom, axial vibration problems should not be encountered. Suffice it to comment that they had been encountered in several ships as the sharpening of economic pencils had resulted in reduced scantlings etc. To be forewarned was to be forearmed.

Ideally, he would have preferred two independent jacket water systems, but this would have been more expensive, requiring an additional pump, cross-connexions and associated valves and would have created problems with the fresh water evaporator, possibly requiring the fitting of two of half size. The system shown in Fig. 3 maintained a constant outlet temperature of the shutdown engine by bypassing its cooler and raising the inlet temperature to the engine. It should be noted that in Fig. 3 various isolating valves had been omitted to simplify the presentation of the control arrangement.

It was interesting to note that Mr. Castle had successfully used engine driven lubricating oil pumps with variable speed engines. That he had not objected to doing so strengthened the author's view that such pumps ought normally to be fitted in preference to electric driven pumps.

Dr. Pinnekamp was to be congratulated upon several counts. First of all for his discovery of and clear exposition on the phenomenon of pinion tumbling, secondly for the thoroughness and speed of his investigation, and thirdly upon his excellent condensation and presentation of his results. Dr. Pinnekamp and his company had, by the frank publication of the failures they experienced, their investigations and remedial measures advocated not only restored any waning confidence in the particular manufacturers, but had made a contribution to gearing design generally. Commander Short thanked Dr. Pinnekamp, his managing director, and his other colleagues for attending this meeting and making such a valuable contribution to the Institute's records.

Mr. Bowers had distilled the potential advantages claimed in the Introduction to three of significance for product carriers. By and large the author was in agreement, as described under the relevant heading of Packages.

With regard to bow thrusters, he doubted if these would be worth fitting on a products carrier of the size they had been considering, as tugs were provided at loading ports by the oil companies and it was generally a requirement that they be used. This was understandable. Where expensive jetty etc facilities were provided, the oil companies did not wish to risk damage to them by relying upon the tanker's own bow thruster (the maintenance and reliability of which was unknown), as opposed to using the tugs provided. Again, unloading was only likely to be carried out at a limited number of ports, unlike the general cargo vessel, which normally, had several loading and unloading ports. As far as his company's general cargo vessels were concerned, the bow thruster had more than lived up to expectations and reduced the towage charges by over 30 per cent. Towage charges were increasing yearly and were particularly high in North American ports where these ships had operated extensively. With regard to time in hand for maintenance of slow speed engines, the figures of 17–21 days were picked by the author from their experience, as representative of time required for proper maintenance. As he had said in his presentation, his argument was not dependent upon this figure and could accept a reduction. Where lesser periods were claimed, he would ask how much main engine maintenance was being carried out at sea or illegally alongside and how much additional time off hire was necessary during the year for main engine repair or maintenance. Sometimes he had heard working days quoted, as opposed to time off hire. He did not believe that slow speed main engines were normally adequately overhauled in 7–10 days. The lost time resulting from "steaming" on one engine should have been allowed for before selecting the charter party speed. Ignorant/irresponsible selection of ship speeds was one of the factors which we, as engineers, had constantly to be watching, and unfortunately too often slipped by and presented a constant cross for ships' personnel and superintendents. It called for constant vigilance and constant education of those responsible for declaring charter speeds.

He was most interested in Mr. Bowers' remarks regarding the solution to the corrosion of steam pipes on deck. He wondered whether the method developed was as effective with old (partially corroded) pipes as with new ones.

He did not recall inferring that major oil companies were any different from independent operators in wanting to reduce the time in repair yards. The only difference he had suggested was that because of excess chartered tonnage, major oil companies already had a flexibility to select optimum times between overhaul periods and that they did not have the same interest in the particular inherent advantage of the medium speed engine installation. With regard to engine driven sea and fresh water pumps, he did not believe these to be in as essential a category as engine driven lubricating oil pumps and in fact he questioned the desirability of fitting them as to do so would complicate pump maintenance (more necessary with water than lubricating oil) and, for variable speed engines, temperature control. Again, circulating water failures were not normally as disastrous as lubricating oil failures and did not, therefore, merit the same philosophical approach.

The battery was not fitted to the instrumentation and alarms of the products carriers because to do so would have involved a large "extra" and by this time they were mistakenly confident that the emergency generator would cut in satisfactorily on loss of primary power and provide an emergency supply. With more care to detailed weaknesses, which had been found by experience, he was satisfied that the emergency generator solution was adequate. It might be of interest to note that the emergency generator in these product carriers was located in the forecastle, as it was in the oil company ships which were the basis of the offer made by the builders. In their latest three medium speed products carriers, the emergency generator had been moved to a less vulnerable position in the accommodation block.

The performance information requested by Mr. Bowers was not yet readily available. At one time one of his colleagues agreed to produce a chart for them to fill in and compare with his company's results. Unfortunately this chart did not materialize. Perhaps Mr. Bowers would like to reinstate the project, which the author certainly believed would be an interesting and worthwhile exercise.

Mr. MacIntyne had pointed out an amazing coincidence of the balancing of the saving in time and the lubricating oil consumption. The author could assure him that it was a coincidence. The engine designers confidently predicted a reduction in lubricating oil consumption to less than 1 g/hp h or about 33 gallons/day when the engine had been overhauled and all recommended measures taken. These figures were generally in line with those claimed by other medium speed Diesel engine designers, but the stricture brought out in the presentation of the paper that "continuous machinery survey is a disadvantage as one never has an engine completely re-ringed/deglazed and back to design consumption" must be remembered.

Interlocks were fitted to ensure that an engine was effectively isolated when being worked upon and could not be inadvertently turned by air or by the running engine. Friction within the clutch when it was disengaged was insufficient to turn an engine. Surprisingly enough flexible hoses and their connexions to the valves had given little or no trouble and no one had expressed any concern about them.

The engineer and electrical officers on board did the servicing of the control gear. Their training had been on the job and they knew more about it than any "cowboys" brought in from shore. This was relatively unsophisticated control gear. With an unmanned engine room, there was a great incentive to make sure that one's equipment worked properly to avoid being called out at night.

Dr. Hiersig had mentioned the desirability of checking the alternating torque when a cylinder was cut out. The correlation between calculations and practice could be established during trials. He had also described an interesting adaptation of the elastic couplings as glorified torsionmeters for balancing the load as between engines, and for operating alarms and disconnecting clutches in the event of overloading of an engine. The reduction in time to stop a ship with slow speed reversing engines, when using a dry pneumatically operated cone clutch, was shown by trials to be significant.

Mr. Crowdy had commented that there was no need for gearboxes to be vulnerable. The author agreed, of course, as he had reitereated in his reply to Mr. Castle. Mr. Crowdy had commented upon their selection of an engine "with high inherent maintenance costs"—of course they should, whilst rejecting this statement as unfounded, excuse him his little gibe as an interested competitor. When he stated, however, that engines were "only rated at 420 bhp/cylinder", the author had to point out that he was asked to talk about these ships two years ago, but refused until they had had practical operating (as opposed to paper) experience. The design of these ships pre-dated the Neumann/Carr paper of 1967 and was some six years old. He agreed that they were not exactly what one would do today in the light of experience. For example, they would probably up the horsepower per cylinder, in which case their installation would show to better advantage. However, he agreed with Mr. Crowdy that the principal practical advantages stemmed from the fitting of more than one engine—a possibility not practical or economically attractive with slow speed engines.

Dipl. Ing. Luther had suggested that the number of auxiliary generators could have been reduced. The author agreed and said so in the paper. However, the installation was designed six years ago and it was not certain what could be achieved with sea maintenance of the main engines. Today they would fit fewer auxiliary engines, but whether he would reduce to a single auxiliary engine, he did not know, but probably would.

As far as noise was concerned, the vicinity of the turbochargers was omitted when taking measurements, as this particular noise was localized and in any case was similar to that prevalent with low speed engines. Silencing measures were taken at the turbochargers and the adjacent air trunking, but he could not comment as to whether such could be considered as standard or in excess of standard.

The interesting remarks on the separate lubricating oil system in the new M.A.N. engines emphasized the importance of keeping the lubricating oil separate from the fuel and water. This was so obvious once one thought about it that one wondered why anyone ever did it in a different way. He agreed that additives were helpful but did not think it would be appropriate to select one by name as several were used.

Regarding tungsten carbide and extra wear, their limited experience to date had not indicated that this was a problem.

Ir. Wesselo had asked whether the author would recommend fitting PC2 engines if there were no draught limitations and no hull design advantage to be obtained. This would need to be looked at carefully for any particular application. As had been mentioned in the paper and by Mr. Crowdy, one of the principal advantages of medium speed engines was that they lent themselves to multiple installation with all the advantages that this produced. Despite the exhaust valve problems and high lubricating oil consumption, he believed their selection of these engines for these particular applications had been justified. Cure these two problems and the engines would rocket ahead in interest. He was always interested to learn of successful exhaust valve experience in other engines on the score that if one could do it, so ought another to be able to do so. He remarked similarly on the lubricating oil consumption figure quoted.

He did not dispute Mr. Davidson's claim regarding the possible service life with the present generation of lubricating oils and looked forward to retaining oil long enough to confirm that claim.

Regarding malfunctioning of the equipment, it was always easy to blame something like that when lubricating oil consumption went up, so they had tried not to put too much blame there, although it was known that some was due to it. On wear in cylinders and whether they are satisfied with the method of measurement, he could only remark that wear was so low that he saw no reason to be unduly complicated and sophisticated about how the results were presented. With regard to differences between tanker and cargo ship experience with blower fouling, he could only point out that not only did they have different modes of operation of the machinery in the two ships, but also different blowers.

Mr. Royle had said that oil companies were always glad to provide an analysis of oil bunkered in the interests of completing information for a project such as this. Unfortunately, it was easier to say this than to get the information. The author's experience had been that you did not get Mr. Royle at the end of your telephone when you wanted information, but rather someone who probably did not know what a fuel analysis was. You had to plan well ahead to get this sort of information.

Regarding the two-stroke Diesel engines with such low exhaust valve failures, he was very surprised indeed and wondered at what powers and temperatures they were running. These ships appeared to be on a "milk run", so fuel analysis could be arranged well ahead.

Mr. Berring must, the author felt, have had his tongue in his cheek when he suggested that one large slow speed engine was more reliable than a multi-engine medium speed engine and he commented no further on that.

With regard to the claims for time off hire per year to recondition a large bore engine, he was amazed. He was reminded of a Boeing 707 full of exhaust valves which he heard was ready to go any place on the globe at a moment's notice for a certain slow speed engine.

The effect of a bow thruster on speed increased with speed and decreased with the efficiency of the design of the cut away behind the orifice. This latter was illustrated in Fig. 41. There was also another loss associated with bow thrusters termed "inertia drag". Due to pressure differences, there was a flow through a bow thruster tunnel. Thus water at rest was picked up, accelerated to the speed of the ship, i.e. given energy, and then spilled out. This energy addition represented a drag or resistance on the ship.

In this case the ship speed, fineness of form and design of cut away was such that model testing was unable to detect any resistance effect from the bow thruster.

During trials the speed was established at which the effect of the bow thruster, as planned, became zero and then reversed as

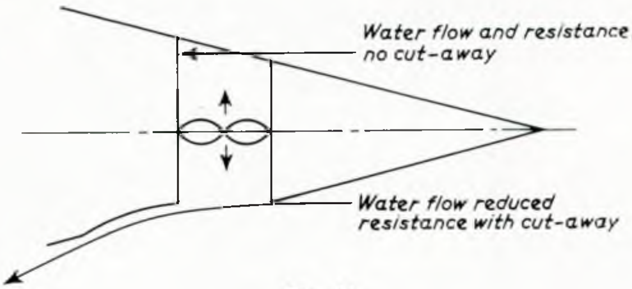


FIG. 41

speed increased due to interference of the ejected water with the ship's boundary layer.

From an owner's point of view, it was interesting to note the current B and W lubricating oil consumption claim which, from the experience of the author's company, appeared more likely than the less than 1 g/bhp h claims which some engine builders said they could achieve. The author felt Mr. Berring had misunderstood him regarding perturbations in the lubricating oil system. He advocated immediate and automatic shut-down on loss of lubricating oil pressure.

Whilst the rate of ring groove wear shown in Fig. 7 was higher than measured in *Luminous* after 20 291 hours, it did, nevertheless, if it continued at the same rate, give a piston life of over 40 000 hours before groove reconditioning, so the statement regarding 8000 hours was not understood.

Mr. Gallois had given some interesting possible lubricating oil consumptions of 0·7 g/bhp h. The author looked forward to worrying about the limiting consumption of 0·6 g/bhp h. He could confirm that Mr. Gallois was always ready to despatch someone at the drop of a hat and whomever he sent was extremely well versed in the engine problems and possible solutions. One would invariably get a straight answer from Mr. Gallois, even though sometimes he had to qualify it as an opinion pending further testing etc. But at this stage the author had difficulty in accepting his arguments regarding the Napier blower, although he accepted that water washing if started early enough, would prevent the vanadium salt/sulphate build-up. He was relieved to receive the assurance regarding the difference in weight of the connecting rods and congratulated Mr. Gallois on the degree of standardization achieved worldwide to date, but the author wagered it was an unstandardized piece that went wrong.

Mr. Barker had provided a most convincing explanation of the difference between the Napier and Brown Boveri blowers used in these particular ships, and the author accepted that the former were more critical with regard to change in turbine efficiency, as the area for gas flow was less. One must just hope that regular water washing would maintain the efficiency in the future.

Commander Short was grateful to Mr. Belcher for cleaning up a rather woolly paragraph. His calculations indicated something on the lines shown in Table VII.

TABLE—VII

	TBN	Basic cost	g/bhp h	Cost	Total cost
Slow speed	—	68	0·2	14	64
	70	100	0·5	50	
Medium speed	20	85	2·0	170	170

Thus, if medium speed consumption approached 0·7–0·8, then the two engines should break even, but for constant speed, medium speed engine consumption would remain essentially at calculated full power value throughout the power range, eg.:

Slow speed	100 per cent f.p. L.O. cost 64
,, ,,	30 per cent f.p. L.O. cost 19
Medium speed	100 per cent f.p. L.O. cost 64 say
,, ,,	30 per cent f.p. L.O. cost 64 say

Mr. Neumann had suggested that the likelihood of future use of the particular equipment in other ships yet to be built was a factor influencing choice of equipment. This might be so for types of ships such as ferries where replacement requirements etc. could be seen well ahead. In this case, that was not a factor which they were able to consider. The task of applying operational research procedures to the selection problem and allocating points, as it were to the various factors was, the author feared, "too difficult" until there was more experience available. At this stage one was very much dependent upon heuristic methods to reduce the problem to a manageable size and ultimately to experienced judgement.

The naval approach to the standby lubricating oil pump using an air turbine compared with the air bottle fed, air operated overspeed shut-down fitted to each pump on the Werkspoor TM 410 engines installed in the latest two Cunard cruise ships, *Cunard Adventurer* and *Cunard Ambassador.*

Mr. Herbert had asked for a breakdown of the work load for the off hire drydocking periods given in the paper. As explained previously, it was not necessary to insist rigorously upon the 17–21 days for the slow speed engine for the author's argument, nevertheless he contended that it was not an unreasonable figure. In the case of the slow speed engine, work load would include its complete overhaul, piston rings etc. and associated continuous survey, together with other ship work. The medium speed engine would have no main engine work load, purely other ship work. In reply to Mr. Herbert's remarks regarding the water systems, the author would invite his attention to his reply to Mr. Castle's comments. He trusted that Dr. Pinnekamp's detailed description of the tumbling would satisfy Mr. Herbert. As for the distinctive marking, this consisted of heavier marking at the tooth ends crossing over at the centre (see Fig. 42). The modified clutch drill consisted of declutching after the main wheel had been brought up to speed. The pinion then centralized itself under centrifugal force and clutch was re-engaged.

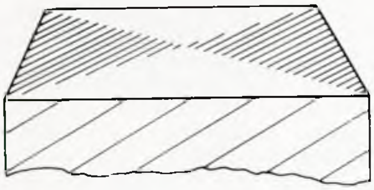


Fig. 42

Mr. Herbert's experience with water washing of blowers was encouraging. The author's company had had no trouble reported with carbon deposits in corrugated expansion bellows.

He had no comments to make upon Mr. Herbert's remaining remarks, the value of which should not be underestimated, coming, as they did, from a person well steeped in Diesel lore and with considerable medium speed operating experience at first hand.

Captain Stewart had suggested that a different combination of valve cage design/valve material/rotocap, as used by Mirrlees and M.A.N. might be tried in the PC2 engine. The normal PC2 injector and valve cooling systems were in series, at low power, the circulating water being heated electrically.

It was not only operationally but commercially important to be able to accept almost any grade of fuel to give complete flexibility in developing the most economic schedules.

The author entirely agreed with Mr. Poole's remarks about the importance of not choosing gearboxes by price. Skating over other such important factors as delivery, he could assure him that at the time when the gearboxes with which there had been difficulties were chosen, they were satisfied with the engineering quality as they saw it and would have been hard pressed to justify more expensive equipment. Of course, it was easy to say now that the gearboxes were not rigid enough and the clutches required additional support, but at that time the problems which they now knew about and had reported had not been encountered.

The author was interested to read Mr. Poole's comments on the thrust block position. Perhaps neither of them should lose any sleep over it, provided the problem of the compensation for the discontinuity of the tank top was understood. Otherwise, he agreed with him regarding the need for a better understanding of "thermal, dynamic and hull stability" problems to reduce Diesel gearbox problems.

Mr. Yamada's contribution was particularly valuable, coming as it did from the representative of such a large manufacturer of PC2V engines. Taking the various points which he had mentioned, piston seizures, the author understood, were traced to an unsuitable liner material and surface preparation coupled with increased b.m.e.p. and had now been eliminated. The author's company, as mentioned in the paper, had had no problems with rubber hoses. Unfortunately, no fuel analysis was provided to support Mr. Yamada's evidence of success with the new water cooled cages being applicable to high vanadium fuels, but nevertheless the reduced seat temperature must be an advantage and Commander Short personally looked forward to hearing more about the performances achieved. He was not clear from Mr. Yamada's comments whether the stem burning was, in fact, corrosion. If so, this should easily be dealt with and might have been engendered from fitting the K3 valves in worn guides. When we could confidently obtain 10 000–14 000 hours between valve inspections with any fuel, he would feel that success was within grasp. Imagine the surplus effort which this would make available, or alternatively one could then really visualize meaningful personnel reductions.

In conclusion, Commander Short thanked the various contributors, a quarter of whom came from abroad to attend the meeting, for presenting such an interesting exercise in replying. He hoped that he had adequately responded.