Paper read at the Royal Institution of Naval Architects on Tuesday, 3 December, 1974

MEDIUM SPEED DIESEL ENGINES IN BULKCARRIERS

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By mid 1972, twin Ruston and Hornsby 'A.O.' medium speed diesel engines had been installed in each of 11 bulkcarriers in the 22/24 000 dwt class. During the period from May 1973 to August 1974 these engines were replaced by the Stork-Werkspoor TM 410. This paper examines the reasons that made the change unavoidable.
An account of the re-engining procedure is given together with a description of the organization set up to carry it out. The corrosion and cracking of the KaMeWa stainless steel propellers, which occurred at the same time are also described.
The performance of six of the ships fitted with the TM 410 medium speed engines attaining a significant number of running hours by 31st July, 1974 is tabulated. Details of the investigation into the nine exhaust valve failures that occurred early in the life of the m.v. *Cape Grenville* are given, together with the cause and its elimination.
Performance of the re-engined ships has been re-assessed against the advantages envisaged in 1968, when the decision to use medium speed engines for bulkcarrier propulsion was taken. The results of a design study carried out in 1974, using figures taken from slow speed and medium speed engines at sea, operating in similar ships and under the same management company are also mentioned. These results confirm that the most attractive installation for a bulkcarrier of about 26 000 dwt is one using twin medium speed diesels geared to a single propeller.

INTRODUCTION

In 1968 an order was placed for the first twelve 22 000/24 000 dwt diesel engined bulkcarriers. The propulsion plant was to consist of two medium speed engines coupled to a single controllable pitch propeller through a reduction gearbox. All ships were designed for U.M.S.

REASONS FOR THE CHOICE OF A TWIN MEDIUM SPEED INSTALLATION

1) There would be a saving in weight of approximately 300 tonnes and a substantial saving in first cost. Two frame spaces would be saved on engine room length.
2) Time off-hire would be reduced; a bulkcarrier may operate on one engine without significant loss of trading time, particularly on short coastal voyages. When two engines are required port maintenance work on one engine can be continued up to the point of departure. Should there be a last minute snag, there is one serviceable engine available to take the ship to sea. With a clean hull and a controllable pitch propeller the vessel would lose about 3 kn when running on one engine with a full cargo, which in part would be compensated by a substantial saving in fuel costs.
3) Spare parts associated with medium speed diesels are relatively small and light and air transport can be used more frequently. The trading pattern envisaged for these vessels made this factor important.
4) An alternator could readily be driven from the gearbox; this would cut down the maintenance load associated with the auxiliary diesel alternators and save the price differential between distillate and residual fuel whilst the ship was on passage.

THE RUSTON AND HORNSBY A.O. ENGINE (Ref.[1], Fig. 1)

The A.O. engine was first projected in 1959/60 as one of the new generation of medium speed trunk piston diesel engines. At that time the medium speed diesel engine burning residual fuel was offering the first serious competition to the large bore direct drive engine as the most economical means of marine propulsion. The A.O. engine had an initial rated output of 500 bhp (373 kW) cylinder at 450 rev/min. The designers employed the two stroke cycle for its theoretical advantage of high specific output and hence lower unit cost.

The engine frame was a lattice work structure made of cast steel members welded together (Fig. 1). Each cast iron cylinder head had beneath it a replaceable water cooled steel flame plate. The cylinder head and flame plate assembly contained four direct seating exhaust valves.

The piston had a steel crown carrying three compression rings and a cast iron skirt carrying a further compression ring and two scraper rings. The crown was cooled by lubricating oil supplied and returned through telescopic tubes.

The liner was initially made of cast iron but subsequently a change to a sleeve of chromium plated steel was made. The liner had a ring of inlet ports which were exposed as the piston approached bottom dead centre.

REASONS FOR THE CHOICE OF THE RUSTON A.O. ENGINE

A design survey in 1968 had shown that geared medium speed diesel engines were competitive in price and appeared to offer substantial operational advantages. At that time at least one make of four stroke medium speed engine was having unforeseen troubles at sea and the two stroke cycle was being advocated for its simplicity. Moreover, the previous experience with large bore slow speed two stroke engines had been good and although not directly relevant, this undoubtedly influenced the choice.

The A.O. engine was already on order by certain other shipowners including one with considerable experience of bulkcarrier operation. It was also on order for support ships for the Ministry of Defence and it seemed likely that any teething troubles would be overcome at sea well before the new vessels came into service.

The manufacturers were quoting low lubricating oil consumption figures[1], and shore testing appeared to be

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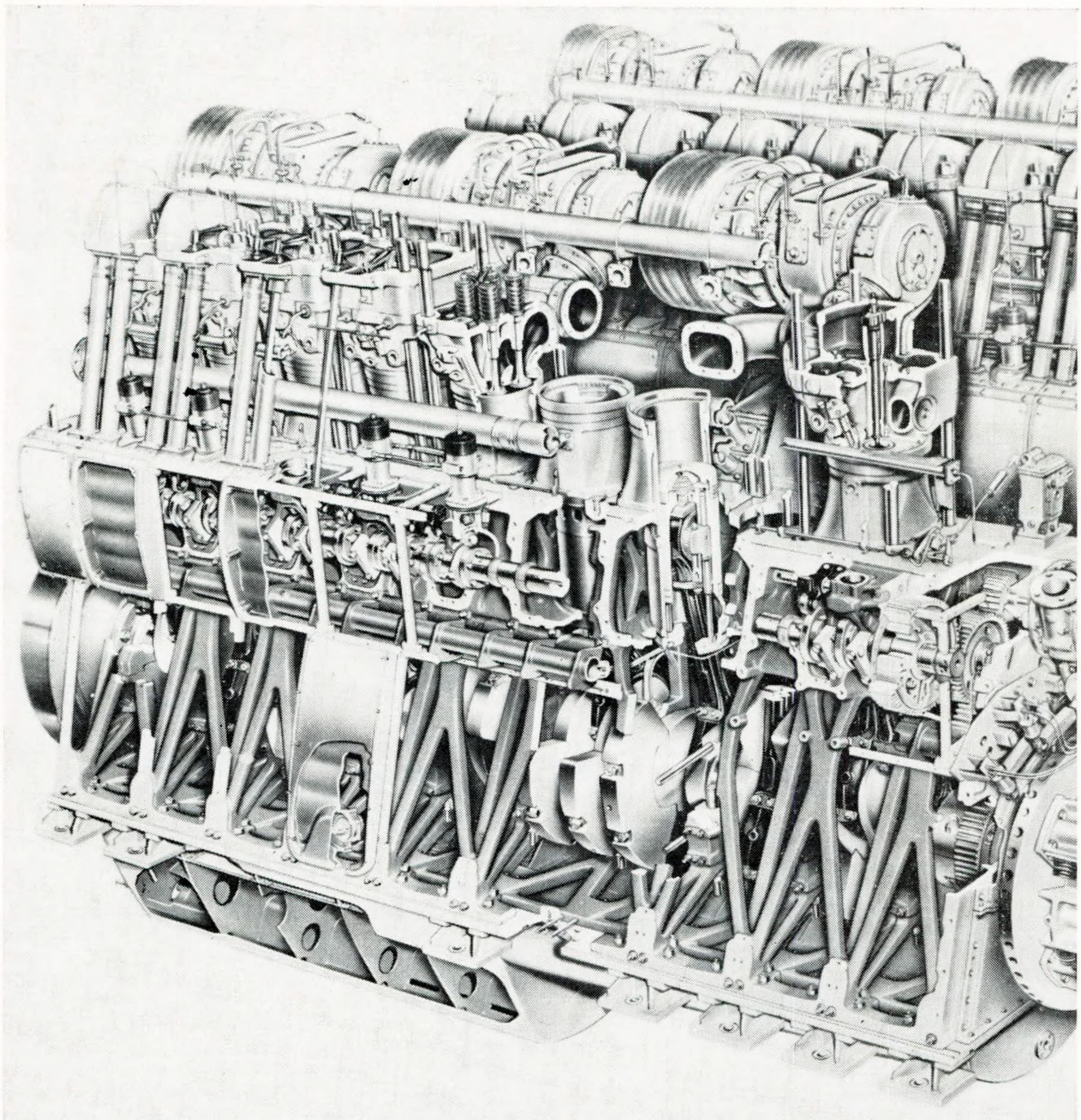

FIG. 1—*Twin 9 cylinder A.O. engine installation showing fabricated steel "space frame" supporting cylinder liners.*

adequate. The engine was stated to have achieved 5000 hours running by 1965 and the manufacturers predicted a total of 20 000 hours test bed running before 1969. Technically and commercially the proposition seemed well based. Standardization of machinery was (and still is) a major consideration in any new building programme. Therefore, early in 1969, A.O. engines were specified for all 12 ships.

PERFORMANCE OF THE A.O. ENGINED SHIPS AT SEA

M.V. *Temple Arch*

This ship was the first A.O. engined vessel of the series. *Temple Arch* differed from the rest of the A.O. engined fleet in having two 9 cylinder in-line A.O. engines (Fig. 1) rather than the twin 12 cylinder vee engines fitted in later ships. The ship commenced her maiden voyage on 4 November, 1969, and on 4 December 1969 the first exhaust valve failure occurred. This was quickly followed by other failures and on that voyage approximately 70 valves burned through, in some cases after a service life of only 250 hours.

To gauge the magnitude of the work involved in coping with A.O. exhaust valve failures at sea it must be appreciated that failures were random and completely unpredictable. The valves were not caged and it was, therefore, necessary to lift an entire cylinder head weighing about 1000 lb (453 kg) in order to replace one valve. This operation also required the draining of about six tonnes of distilled water from the engine system.

On completion of the maiden voyage, all the cylinder heads were removed and new flame plates fitted with recessed exhaust valve seats. This reduced the rate of exhaust valve failures but the number of replacements nevertheless remained unacceptably high. The only solution appeared to lie in a change of material and by October, 1970, all valves previously supplied in stellited E.N. 54 steel were replaced by valves in Nimonic 81.

During the maiden voyage referred to the other principal troubles were:

i) excessive liner wear;
ii) heavy lubricating oil consumption;
iii) high exhaust temperatures.

The problem which ultimately proved to be the most intractable was very rapid wear of piston rings and liners and ensuing heavy consumption of lubricating oil. The first change in piston design resulted in no improvement in liner wear. Consequently, in March, 1971, all liners were renewed and tapered faced rings fitted. Four weeks later it was necessary to delay the vessel at the Panama Canal in order to overhaul the majority of the pistons and liners again and during the subsequent voyage from Panama to Japan a further fifteen liners had to be changed. The lubricating oil consumption during this period varied from 150 gallons/day (682 litres) on one engine to 400 gallons/day (1818 litres) on the other.

On return to the U.K. in November, 1971 all the liners were again renewed and new pistons with three $\frac{5}{16}''$ (8 mm) wide compression rings were fitted in place of the four $\frac{1}{2}''$ (12·7 mm) ring assembly. All the flame plates were replaced due to leakage round the transfer tubes and round the valve pockets (Fig. 2). On arrival in Japan two months later all the liners again required renewal. This time pistons with four $\frac{5}{16}''$ (8 mm) rings were fitted and this ring pack at last produced a significant reduction in the rate of liner wear. Up to this point the vessel had experienced approximately 158 liner changes.

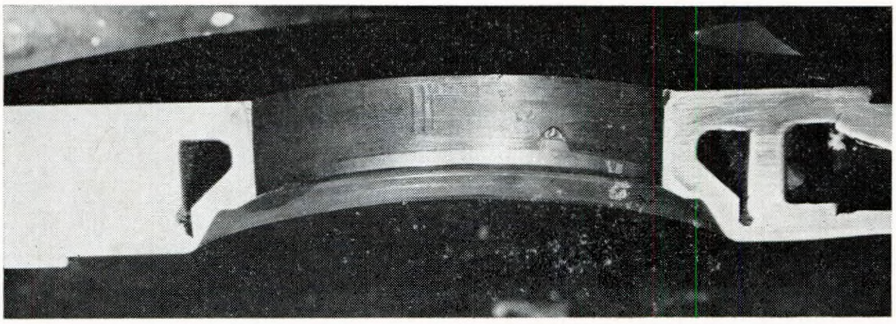

FIG. 2—*Cross section of A.O. flame plate showing corrosion/erosion in the cooling water passage surrounding the valve seat (1070h)*

OTHER A.O. ENGINED SHIPS

Similar troubles occurred in all other ships fitted with A.O. engines to a greater or lesser degree. Modifications gradually reduced the rate of the simple mechanical failures but no solution was found to the fundamental problems of high lubricating oil consumption, which in some ships was between 500/600 gallons (2273/2728 litres) per day, or the very high rate of piston ring and liner wear. Liner wear could range from 0·025 mm to 0·25 mm in less than 1000 hours on the same engine and this wide variation made it impossible to judge the engine condition by examining a single line of moving parts. All liners had to be examined at frequent intervals to prevent ring damage and/or piston seizure.

SEASTAFF

One of the most interesting features of A.O. engined ships was the attitude of mind that developed among seagoing engineers and the superintendents. Despite the immense amount of work that ships' engineers and G.P. ratings were called upon to do in order to keep these engines running, morale on board was high and often quite outstanding.

One voyage of *Temple Bar* took 38 days to go from Kenai in Alaska to Chiba in Japan at an average speed of 4·6 kn. Although the ship left on two engines, virtually the entire voyage was made first on one engine and then on the other, including a period of five days in a force 8–10 gale with no power at all.

The author went to Chiba to meet *Temple Bar* (which arrived under her own power having rejected the offer of a tug) expecting to have to find a new crew to take her to Singapore.

In fact, no one asked to leave the ship and the Second Engineer who had had the most arduous job on board, not only stayed with *Temple Bar* but after taking some leave in the U.K. returned as Chief Engineer to bring her home from Australia for re-engining.

This was by no means an isolated incident. The challenge was there and both the seastaff and the superintendents seemed to enjoy meeting it. By contrast, the TM 410 was a trouble free, smooth running anticlimax.

DEVELOPMENT WORK

The Manufacturer's development effort was stepped up when the magnitude of the problems associated with these A.O. engines was fully appreciated but in spite of carrying out some 400 tests over a period of 18 months and using all available engine test facilities, no marked improvement in liner or ring life was obtained. Tests covered virtually all known methods of supplying lubricating oil to the cylinder liner wall including timed injection. Piston ring materials varied from treated cast iron to a non-metallic polyamide material. A wide range of ring geometries was also investigated.

Unfortunately, most, if not all of these tests were short runs of 50 or 100 hours with each combination of materials and geometry. Extrapolated results from these short term tests tended to be misleading and modifications that looked very encouraging in the laboratory often proved to be of no advantage at sea. Apart from the usual problems associated with the transfer of laboratory results into a ship environment, the random nature of the A.O. liner wear made all short term tests fundamentally suspect.

INTERIM ACTION TO CONTAIN THE A.O. ENGINE SITUATION

By October 1971 seven of the 12 ships were at sea and several of their engines were in different states of modification. No tenable theory explaining the high ring and liner wear rate had yet been put forward. Some ships were now taking in lubricating oil in bunkering quantities and despite the Manufacturer's round-the-clock development work and the enthusiasm and optimism of his engineers, A.O. engines at sea were constantly breaking down, often in the most inconvenient places. All this was making profitable management of these ships extremely difficult. Both technically and commercially the situation appeared to be open ended.

It was clear that an overworked Head Office technical staff could not continue to cope with the tremendous load imposed by the constant operational troubles of the A.O. engine as well as to the routine work associated with managing a fleet of 20 bulkcarriers. An increase in staff was obviously necessary and the author joined the company in June 1972.

By mid 1972, three facts were patently obvious:

1) development work currently in hand by the manufacturer was unlikely to solve the main problem of rapid liner wear in the near future, if at all;
2) in view of the fundamental and intractable nature of this particular problem no more ships should be fitted with the A.O. engine until it had been solved;
3) something had to be done immediately to reduce the maintenance work at sea.

It seemed very probable that a change to a cleaner fuel of lower viscosity would at least reduce some of the work of the ships' engineers. Initially, therefore, the fleet was switched over to distillate fuel. This produced an immediate improvement in reliability and also in the cleanliness of the machinery spaces. However, marine diesel oil is a very expensive fuel for ships which, because of continual delays due to main engine troubles, are struggling to earn their keep. A compromise using 250-300 second Redwood residual fuel was introduced instead.

Two of the original 12 ships designed to take the A.O. engines were still under construction at Haugesund. These were the *Baron Wemyss* and the *Cape Grenville*. In the case of *Baron Wemyss* the engines were already installed and the Yard, not unnaturally, was most unwilling to upset its entire building programme to change the engines at so late a stage. In the case of *Cape Grenville* however, the engines had not yet been delivered and there was time to select an alternative.

ALTERNATIVE ENGINES FOR THE *CAPE GRENVILLE*

Between 1969 and 1972 there was a rapid advance in the design of medium speed diesels for merchant ship propulsion. For the owners of the *Cape Grenville* certain criteria had to be met these were:

a) the engines had to fit into the engine room without major alterations to the ship's structure;
b) the speed of the engine should be such that the existing gearbox could be used;
c) in view of the traumatic experience with the A.O., the engines must have a proven record of reliability at sea and a low lubricating oil consumption;
d) engines capable of burning fuels of up to 1500 seconds Redwood viscosity were a necessity;
e) their structure should be as rigid as possible and have enough internal damping to reduce the effects of vibration to a minimum.

Of the engines on the market in 1972 the Stork-Werkspoor TM 410 appeared to fit these requirements best and it was decided to purchase two and install them in *Cape Grenville*.

OPERATING PROCEDURE FOR THE TWO REMAINING NEW A.O. ENGINES IN THE *Baron Wemyss*

To keep the maintenance of the new A.O. engines in the *Baron Wemyss* to a more manageable level, power output was restricted from the outset to two thirds of the normal continuous rating and a rigid planned maintenance programme was enforced. Each piston was to be removed every 2000 hours and new rings fitted, the exhaust valves being touched in or replaced at the same time; the liner would be carefully inspected and renewed if necessary. This routine was to be observed regardless of the apparent health of the engine, and also, no further experiments on engines at sea could be entertained without the back-up of prolonged shore testing. *Baron Wemyss*, therefore, put to sea with the latest proven modifications only. Engines were run on distillate fuel for about three months and then switched to 250-300 seconds Redwood residual fuel.

THE A.O. ENGINE LINER WEAR PROBLEM

Fig. 3 illustrates the top dead centre region of an A.O. liner after 1897 hours running in *Baron Wemyss*. The circumferential groove is typical of the wear that occurred in a random fashion in every engine. A few liners showed two grooves approximately in line with the top and second compression rings at T.D.C. and the depth of such grooves could reach 1 mm in less than 500 hours.

At first sight the problem appeared to be lack of oil on the upper part of the liner. As the piston passes over the scavenge ports on the compression stroke, oil is scraped off into the scavenge space, leaving insufficient oil to form a lubricating film under the rings. Most of this oil is blown out with the scavenge air and escapes through the exhaust valves This theory might perhaps explain the heavy lubricating oil consumption as well as the lack of sufficient oil to form a film. However, it does not explain why the wear continues to increase rapidly, even when an engine is using 300 gallons (1364 litres) of lubricating oil a day. With this quantity flowing through the cylinders it would be expected that some, at least, would stick on the liner surface above the scavenge ports.

An alternative or perhaps complementary explanation could be that excessive blow-by prevents a film of lubricant forming under the compression rings even when there is an adequate supply of oil in the vicinity. Blow-by has occurred on almost every A.O. engine, sometimes so vigorously that it has caused a very rapid deterioration of the lubricating oil in the drain tank. However, blow-by always appears to be accompanied by seized piston rings, badly scored pistons and heavy liner wear and it is difficult to say which comes first.

It was possible that the trouble might be caused by the use of a chromium plated liner. Depending on the nature of the surface treatment, chromium is usually rather more difficult to lubricate than good quality cast iron but it is unlikely that the use of this material would, by itself, cause such a peculiar pattern of localized wear. Moreover, had the

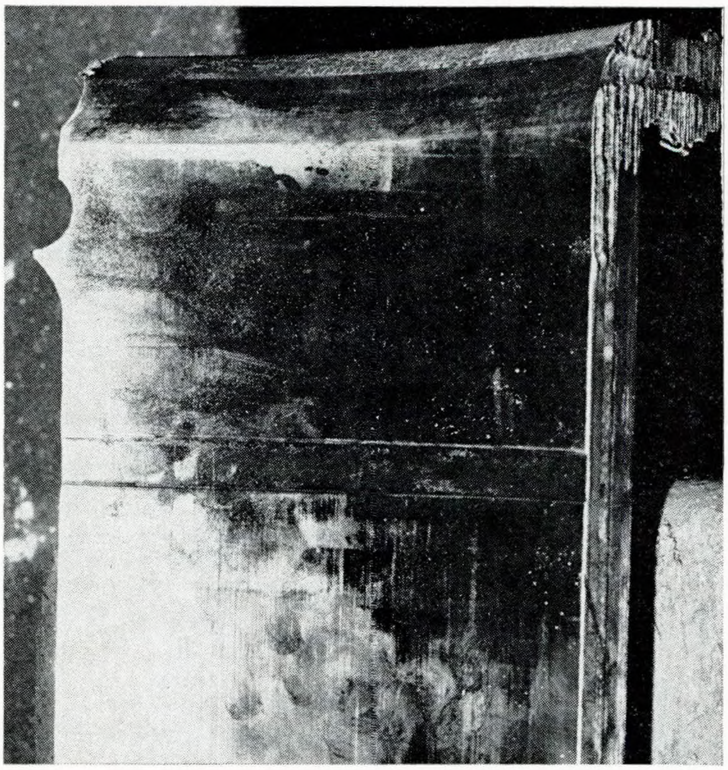

FIG. 3—*Inside sectional view of an A.O. liner showing circumferential groove region of the top ring*

problem been one of blow-by, or optimization of ring and liner geometry and materials, or supplying the right quantity of oil near T.D.C. then it would almost certainly have been solved by the Manufacturer's very extensive research and development work in this field. Moreover literature on the subject is considerable and this had been carefully studied (e.g. [(4)] and [(5)]). It was therefore necessary to look elsewhere.

The most unusual feature of the A.O. engine, apart from the liner wear pattern is the structure of the space frame on which the cylinders rest and underneath which the crankshaft is slung. An examination of this structure suggested that the liner wear problem might be associated with the "space frame" design criteria. These were high torsional rigidity[(1)], safe working stress in each member, and safe fatigue stress in all welds. All these factors had been carefully calculated and the calculations checked by model tests. However, no firm design limit appeared to have been laid down for the maximum relative deflexion of the cylinder head abutment and main crankshaft bearing support. Under the firing load this deflexion could be significantly higher than that occurring in an engine with a conventional solid cast iron bedplate.

A rough calculation suggests that the deflexion under a "static" firing load would be in the region of 0·5 mm but in practice the actual relative deflexion under dynamic conditions could obviously vary widely from this figure. The space frame is resilient and has little internal damping, and interaction between cylinders would almost certainly occur.

At this point it is worth comparing Figs 1 and 4. The TM 410 engine illustrated in Fig. 4 has a massive cast iron bedplate and an entablature that constrains the relative positions of the cylinder head, cylinder liner and main bearings to within very close limits. By virtue of its large surface area the cast iron structure suffers only a small change of stress when the engine fires. As long as the tie bolts are tightened so that their combined pull exceeds the firing load on the cylinder head by a generous margin, relative movement of any of the parts should be very small. By contrast, significant relative movements in the A.O. engine will occur very rapidly, probably at some harmonic of firing speed.

Maximum relative displacement between the A.O. piston and liner due to distortion of the space frame probably occurs at the moment of firing when the pressure forcing the top compression ring on to the liner wall is at its greatest. With a flexible support and a lack of internal damping and perhaps a lack of oil it is not difficult to visualize how a fretting type of action could occur that would damage the liner and form a circumferential groove of the type shown in Fig. 3. The hard $Cr.O_2$ debris from the fretting action would probably embed itself into the rings and so cause general wear over the length of the liner.

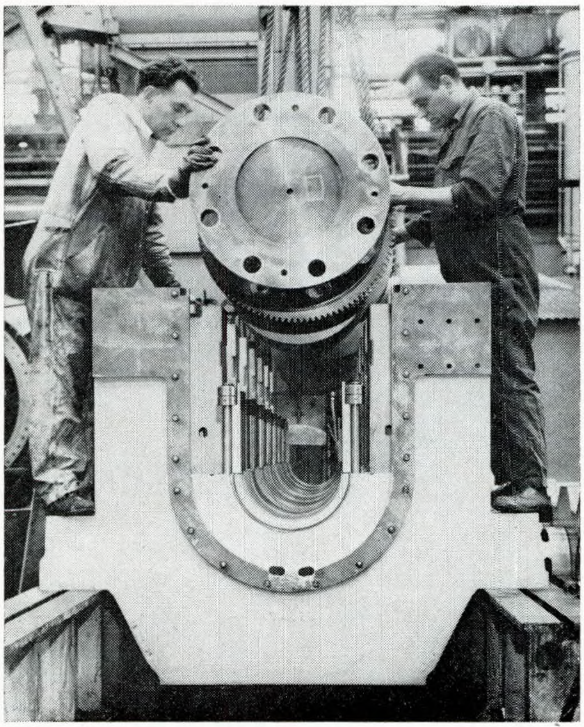

FIG. 4—*Cast iron bedplate of an In-line Stork-Werkspoor TM 410 engine*

THE FUTURE OF THE A.O. ENGINE IN SHIPS ALREADY AT SEA

The immediate problem was to decide what to do about the ships already at sea with the A.O. engine—whether to redesign the engine to the point at which it would be possible to live with or, alternatively, to replace it with another make of medium speed diesel. The former was not necessarily impossible, given adequate finance, time, enthusiasm and good intuitive engineering as a medium speed two stroke engine of similar power had been running more or less satisfactorily for approximately five years. However, if the fretting hypothesis proved to be even partly correct (and this by itself would take time to establish) then a major change of design thinking would be required. The light space frame concept would have to be modified or even discarded completely and lateral rigidity built into the engine in the plane of each piston, liner and the associated main bearings. This would mean a complete rebuild of the engine, obviously.

Furthermore, it was likely that a re-designed A.O. engine with an acceptable rate of liner wear would still be inferior to the new generation of four stroke medium speed engines coming on to the market and this would undoubtedly be reflected in the re-sale value of the ships.

After a careful study of the alternative courses of action it became evident that the only commercially viable solution was to re-engine the remaining 11 A.O. engined ships. This decision was taken with extreme reluctance because, apart from the magnitude of the task involved, an operation of this sort has a serious effect on the cash flow situation.

Another and more detailed survey of the medium speed diesel engine market was made and again the technical assessment suggested that the TM 410 would be the most suitable replacement. This was re-assuring because it endorsed the original change decided upon for *Cape Grenville.*

THE TM 410 ENGINE

The design philosophy underlying the TM 410 is very different from that of the A.O. engine. In the case of the A.O. considerable ingenuity and skill had been exercised in making the whole assembly as light as possible (Fig. 1 and Ref.[(1)]). The designer of the TM 410 had not regarded weight-saving as an important feature but had placed the emphasis on rigidity, both torsionally, and laterally between the cylinder liners and the main bearings (Fig. 4).

The main elements of the TM 410 consist of a U shaped bed plate and a massive cylinder block connected together by tie rods. The cylinder liner is made from a thick cast iron quill and has cooling holes drilled at an angle in the top section. Although the engine operates on the four stroke cycle, lubricating oil is injected from two holes in the cylinder liner about halfway up its length. The cylinder head is of cast iron and each exhaust valve is secured in a water cooled cage that can be removed without lifting the entire cylinder head. Apart from the tie-bolts the main structure consists almost entirely of cast iron and makes a very rigid assembly with good internal damping. The engine is consequently much heavier. The 12 cylinder TM 410 at the 1972 rating is over 40 tonnes heavier than the A.O. engine of equivalent rating. Stresses and temperatures in the 1972 TM 410 are conservative and the history of the engine has been remarkably free from mechanical failures.

THE RE-ENGINING PROGRAMME

Office Organization

Two small but necessary changes were made to the Technical Office structure to cope with the A.O. situation just before and during the re-engining period. A Superintendent Engineer was recruited and put in charge of the spare gear and stores section; at this stage the supply situation was critical and engineering judgement was required before deciding which A.O. spares should be allocated to any particular ship. A comprehensive visual display was set up on the wall of the Superintendents' office showing the current state of the machinery of all ships, whether large bore or medium speed; this "Information Board" was up-dated weekly from information sent in from each vessel and a plot was made of the important parameters affecting ship performance. This enabled trends to be observed and action taken in time to avoid any major engineering disasters during re-engining when the chartering situation was going to be difficult due to vessels being withdrawn from service.

Tenders

Tenders were invited from Yards interested in re-engining the 11 A.O. ships, with a request that time and cost should both be quoted thus indicating the value of time to the Owners. Quotations were obtained from other manufacturers of competitive medium speed diesels as well as from the preferred engine manufacturers. A rough programme was drawn up to establish the cost of the whole re-engining operation, and the effect that time out of service would have on the forward chartering commitments of the Company. It was then possible to put realistic figures into the cash flow equation. The freight market survey carried out by the chartering department predicted a probable upswing in 1973–1974 and, consequently, the sooner the re-engining was completed the less expensive it would be.

Contracts and Letter of Co-operation

In November 1972 separate contracts were negotiated between Amsterdam Dry Dock, Stork-Werkspoor Diesel and the Owners. Much thought was given to the co-ordination of the work and whether the whole contract should be placed with the Yard as the main contractor. However, for several reasons it was thought better to deal separately with each party and a joint Letter of Co-operation was drafted, setting out quite clearly the responsibilities of each. It is an indication of the excellent relationship that existed throughout the re-engining period that the Letter of Co-operation was referred to only once and that on a minor issue. Suitable break clauses were established in these Contracts should the performance of the TM 410 in *Cape Grenville* fail to meet the guaranteed performance figures.

Control of Re-engining

Both the Engine Manufacturer and the Yard were situated in Amsterdam. The Owners therefore put a senior Superintendent Engineer in charge of the whole Amsterdam operation, and he lived in Amsterdam while the work was in progress. It was made quite clear to everyone concerned that he had full authority to act on the Owners' behalf and that there was no appeal over his head. Without this type of delegation the re-engining of 11 ships in less than 16 months would not have been possible. Many decisions had to be taken while the work was in progress that could not await the approval of someone in Head Office.

In addition to the re-engining work there was inevitably a large number of owners' extras. Many of these, e.g. the overhaul of the controllable pitch propellers, the re-conditioning of the propeller hubs and the examination of the stern tube bearings were intimately linked with the re-engining procedure. The majority of the engine room auxiliaries had suffered due to the concentration of all the maintenance effort on the main engines and most of them required a major overhaul. Other items, such as the cleaning and re-painting of the holds, the cleaning of the accommodation and the overhaul of the cranes and upper deck machinery were not closely connected with re-engining and were handled from Head Office.

Staff

The staff at Amsterdam included the Commodore Chief Engineer, who already had extensive experience of the TM 410 engine in *Cape Grenville* and who accompanied each ship for the first leg of her first voyage after re-engining. There was also an assistant Superintendent, a Master to cover deck items, a Store Keeper, and one man whose task it was to clean the accommodation of each ship from top to bottom. Continual maintenance work on the main engines had left its mark on the accommodation and the restoring of each ship to its original high standard of cleanliness was a task that carried with it a great deal of job satisfaction.

Training at the Engine Manufacturer's Works

Twenty-seven Chief and Second Engineers took part in a series of training courses arranged by the Manufacturer. These courses were held at the Manufacturer's works and it was therefore possible for each class to see the engines in all stages of manufacture. As a result, those taking part acquired an excellent three dimensional understanding of the TM 410 as well as being thoroughly indoctrinated into the routine maintenance requirements of the engine.

Chartering

Perhaps the trickiest part of the whole exercise was programming the 11 ships to arrive in the repair yard as near as possible to schedule bearing in mind the chronic unreliability of the A.O. machinery. At the same time it was essential to incur the minimum time out of service. The most practical approach was to bring the ships to Western Europe well before time and then trade them on short voyages until they were required. Even so, there were some nasty moments. Only *Temple Inn* whose owners had run her on distillate fuel throughout and *Baron Wemyss* working at reduced power and under a rigid planned maintenance routine actually arrived on time and without problems.

Re-engining Programme

Work was scheduled to start early in April 1973 with the re-engining of the *Baron Renfrew* and to finish in August 1974 with *Baron Wemyss*. Two ships at one time were the most the Yard considered could be handled efficiently with the work force available, but it was also realized very quickly that it would pay to concentrate as much work as possible between September and March and thus have the minimum amount of ship overlap during the holiday period. Between September 1973 and March 1974 six ships were re-engined with an average time from arrival to departure of 47 days plus one day at the trials berth.

It would have been possible to have brought *Baron Wemyss* into the Yard earlier and thus to have achieved the original completion date of the end of August. However, the ship was operating without major problems and with freight rates still at a high level there seemed to be no point in risking a delay due to the shortage of labour. Consequently *Baron Wemyss* was brought in for re-engining after the previous ship had left.

The Engine Manufacturer managed to alter his production schedule to fit in with the revized requirements of the Yard and the final result was an excellent example of what can be achieved by co-operative organization and above all good communications. The latter requirement is sometimes under-rated but if there was one single factor that made for the success of the whole operation it was the free flow of information on planning, progress and problems between the Yard, the Engine Manufacturers and the ship management team under the senior Superintendent at Amsterdam. Feedback from sea came either directly from the Commodore Chief Engineer returning from each first voyage or from Chief Engineers' letters and the weekly performance reports to Head Office.

Cleaning

With three exceptions the engine rooms were so heavily contaminated with soot, fuel oil and lubricating oil that a tank cleaning vessel had to be used to get rid of the worst deposits. Hand cleaning was then used to reach the condition in which a gas free certificate could be obtained and burning and welding commence.

De-storing and Security

Frozen stores were removed and the accommodation sealed by welding straps over all external doors except one to prevent pilfering. The hatch to the engineers' store was similarly secured and a watchman placed on board.

Access to the Engine Rooms

As the forward end of the engine room was also the after end of the aftermost hold, a section was cut out of this bulkhead to allow the A.O. engines to be pulled out into the hold and thence taken away by crane (Fig. 5). A smaller hole cut through the next bulkhead provided direct access to another hold and this was used as a store room for the items of equipment that were to be replaced. A door was cut in the ship's side at dockside level to allow easy access for the work force.

FIG. 5—*Forward engine room bulkhead of* Cape Horn *showing a section removed prior to withdrawal of engines, and separate access door*

Removal of the A.O. Engines

The engines were disconnected from their couplings, control gear and pipe work, and pulled along tracks built up from heavy steel girders into the aftermost hold and lifted out on to the dockside (Figs 6 and 7).

Auxiliary Machinery

It was necessary to clean and overhaul all the auxiliary machinery. In particular, the electric generators and motors that were not totally enclosed were badly fouled with soot and grease and had to be very carefully cleaned and checked for insulation damage.

Change of Gear Ratio

Not all of the 11 ships had the same hull form or the same gearboxes. In the case of four ships it was decided to fit in-line engines to improve the general layout and obtain more room for routine maintenance. With in-line engines the number of cylinders was restricted by the length of the engine room and, therefore, the engines had to run at their maximum design speed in order to obtain the required power. In these ships the gearboxes were removed, new bull wheels, pinions and power take-off drives fitted and the gearboxes realigned to the tailshaft.

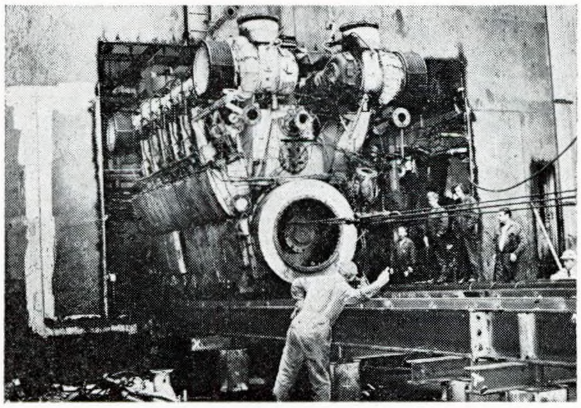

FIG. 6—*12 cylinder A.O. engine being withdrawn through engine room forward bulkhead into aftermost hold*

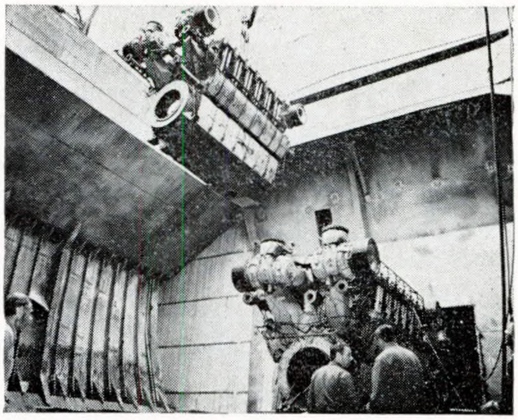

FIG. 7—*Removal of A.O. engine indicating relative size of engine and hold access*

Drydocking

The most expensive and time consuming operation in drydock was the refitting of the KaMeWa controllable pitch propellers. Cracks and corrosion were found in the stainless steel propeller hubs and in the blade roots (Fig. 8). The propeller hubs had to be sent back to the manufacturers for re-conditioning and the cracks in the blade roots removed by grinding. Sharp edges around the bolt holes were also faired off. A second drydocking was required to replace the hub assembly. Later in the re-engining programme it was possible to obtain a spare hub and blades before the vessel docked and thus avoid a second costly period in drydock with the inevitable interruption of the work on board.

All the tailshafts were found to be corroded in the vicinity of the gaps between the stainless steel covers protecting the main bolts holding the propeller to the tailshaft (Fig. 8a). It was necessary to remove the tailshaft, grind out the corroded area, crack detect the adjacent region and then inhibit the shaft against further corrosion. The opportunity was taken to survey the tailshaft bearing and gland, which in every case was found to be in good condition.

Renewing the Main Engine Bedplate Seatings

In several ships this proved to be more difficult than had been expected due to the distortion that occurred as the old bedplate was removed and the tank top cut. Correction of this distortion added four days and a substantial sum to the cost of re-engining. The main bedplate was prefabricated as a single unit, lowered into the aftermost hold and pulled into position on the tank top (Fig. 9).

Alignment

The gearbox was replaced, or, when no change of gear ratio was necessary, the alignment of the bull wheel was checked with the tailshaft flange. The TM 410 engines were pulled into position on fabricated steel girders and were then aligned to the pinion flanges. The engine was bedded down on to epoxy resin chocks. During this operation the engine room had to be maintained at a temperature of 12–15°C in order to assist curing. The bedplate was warmed locally. The work of alignment and chocking was carried out over a week-end to ensure that the structure suffered a minimum of disturbance. Even so, some differences of opinion arose between the Classification Society Surveyor, the Engine Manufacturer, the Yard and the Owners' representative. This was because not every one appreciated at the time that the distortions caused by local heating were not permanent. In fact it took four days before the readings returned to normal.

Spare Gear and Documentation

The platform deck in the engine room was re-designed to improve the movement of stores and the stowage of spare gear and all A.O. spares were removed and sent to other ships still fitted with A.O. engines. TM 410 spares were then substituted and new stowage arrangements made. The docu-

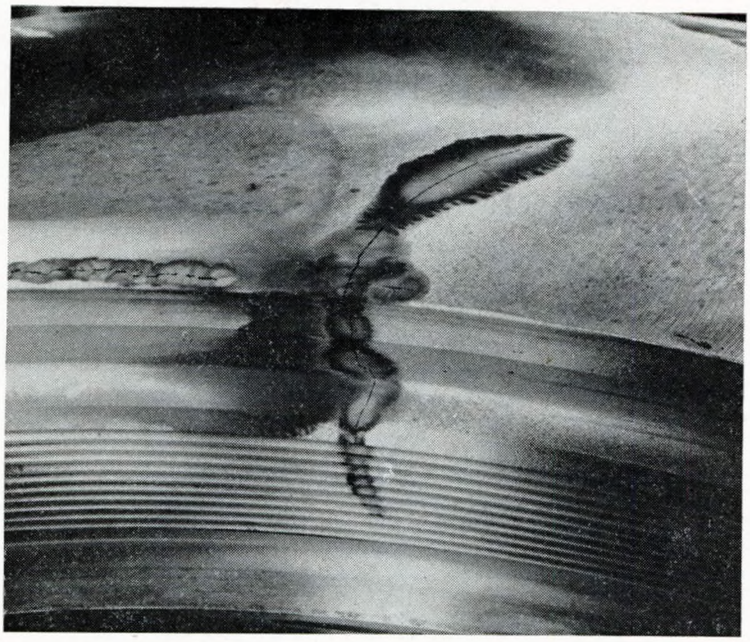

FIG. 8—*Typical crack in stainless steel hub of a KaMeWa propeller*

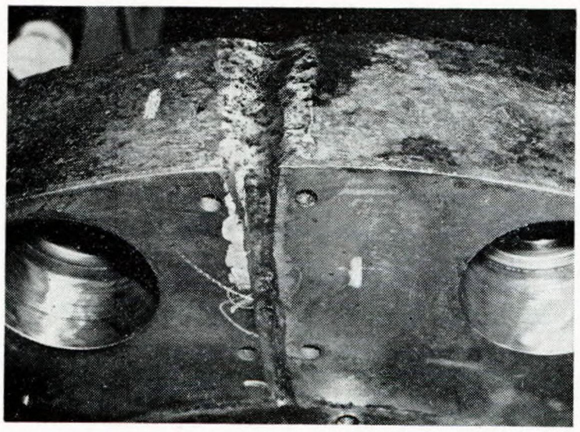

FIG. 8a—*Corrosion of a tail shaft, flange adjacent to the Kamewa stainless steel cover plates*

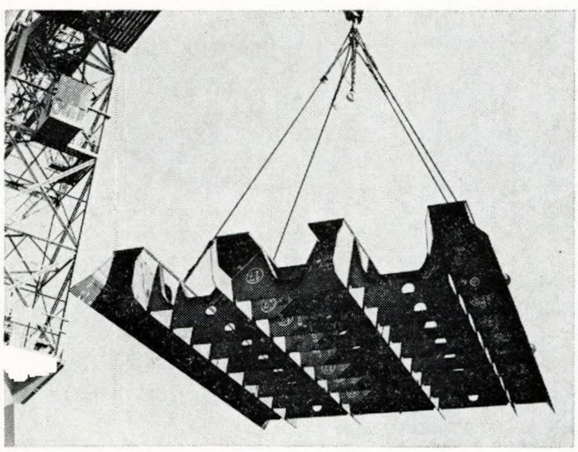

FIG. 9—*Seatings for TM 410 engines being lowered into the hold*

mentation was amended and relevant instruction books were supplied, together with a new planned maintenance routine. This was no small task, particularly in the case of the first two ships, as this work coincided with the setting up of computer programmes to handle both the spare gear and planned maintenance routine.

Re-assembly of Systems

Pumps were re-positioned, pipe work replaced or renewed and new instrumentation fitted. Exhaust uptakes were lagged and the forward bulkhead re-welded. The ballast control panel, which was removed together with the forward engine room bulkhead, was re-positioned and reconnected to all the hydraulic actuators. The electric cables crossing the forward bulkhead were reconnected through junction boxes or in some cases renewed completely. During this period the repainting of the engine room was completed.

Trials

The main engines were run unclutched for one hour to check the instrumentation and the safety devices. The vessel was then trimmed until the draught was 8 ft. forward and 20 ft. aft and moored at the trials berth with the bow hard against a soft mud bank.

Both engines were clutched in and run for approximately six hours at 60 per cent. This was the maximum power that could be absorbed by the propeller with the ship stationary. However, each engine could be run at 100 per cent power separately, and full power trials were carried out on each engine for approximately one hour.

The sea trial commenced by setting up the KaMeWa control gear and adjusting the power sharing between the engine governors. With both engines running the propeller speed was increased and the main engine shaft driven generator put on load. One engine was then quickly declutched. This was to prove that the load control gear was capable of keeping the propeller revolutions within $\pm$ 10 per cent and that the shaft driven-alternator remained on load and did not cause a blackout.

The power was increased to 85 per cent of the maximum continuous rating and this power maintained for approximately one hour. Towards the end of this time a complete set of readings was taken. A further two hours trial at 85 per cent power was carried out and the instrument readings checked for any variation. The ship then proceeded on passage. No ship has returned to the Yard for modifications after sea trials.

MEDIUM SPEED ENGINE INSTALLATION IN NEW BUILDINGS

One conclusion that emerged from the re-engining of 11 medium speed diesel ships is that it would be a simple matter to design an engine room for a medium speed diesel bulk carrier so that there was a clear route for the main engines from the forward bulkhead of the engine room. All pipe work or electric cables could run above or below the main engine profile on the bulkhead and the pumps could be positioned elsewhere. This leads to the tempting thought that medium speed main engines could be the last units to be fitted when the ship is built thus saving time for the ship-builder, improving his cash flow situation, and, one hopes, saving the owner a little money.

TABLE I—OPERATING EXPERIENCE WITH SIX BULKCARRIERS WITH TM 410 MEDIUM SPEED DIESEL ENGINES DURING THE PERIOD JANUARY 1973 TO 31 JULY 1974

Vessel	Operational time		Total period without shaft power	Period underway on one engine	Effective days lost due to period on one engine	Effective days lost to hull fouling	Exhaust Valve Failures		Valve spring failures
	Days in service	Days underway					Burnt	Bent	
Cape Grenville	540	305	2 day clutch rubber change. 2 days dry dock	3·5 days pipe joints and leakages. Investigating exhaust valve failures.	0·7	4 days	9	6 +24 due overspeed stbd. eng.	4
Baron Renfrew	354	230	0	6·0 days pipe joints and leakages Investigating exhaust valve failures.	1·2	4 days	13	18 due overspeed stbd. eng.	14
Cape Horn	313	187	0	2·0 days pipe joints and leakages. Modifying valve gear lubricating system.	0·4	1.5	0	0	5
Cape Grafton	273	165	0	1·5 days replacing cylinder head inlet valve inserts	0·3	0	0	0	5
Baron Ardrossan	225	110	0	12 days (turboblower) failure.	2·4	0	2	0	7
Baron Inchcape	138	86	0	0	0	0	1	0	0

SUMMARY OF OPERATING EXPERIENCE WITH THE TM 410 ENGINE FROM JANUARY 1973 TO JULY 1974. (TABLE I)

By July 1974 sufficient experience had been gained with the TM 410 engine to make a tentative assessment of how closely medium speed diesel engined ships now met the original objectives set out at the beginning of the paper:

1) *Saving in weight*—Using the heavier TM 410 engines running below design speed (to avoid the cost and time of changing the gearbox) the saving in dwt has been reduced from 300 tonnes to 200 tonnes.
2) a) *Time off-hire*—In 18 months service *Cape Grenville* was never without shaft power at sea. During the same period, apart from the routine two day Guarantee Dry Docking, this ship was only immobilized in harbour for 46 hours—the time required to change two damaged rubber couplings in the clutch assembly. In fairness to the clutch manufacturers it should be stated that the failure of the rubber did not immobilize the ship, but it was thought prudent to renew the flexible elements before the commencement of the next voyage. Of the re-engined ships, none has been without shaft power to date, either at sea or in harbour.
   b) *Time on one engine*—The *Baron Ardrossan* had a series of unexplained bearing failures on two turbo blowers and as a result operated for 12 days on one engine, thus incurring an equivalent time off-hire of 2·4 days. The *Baron Renfrew* had to operate on one engine during the replacement of a cracked cylinder head and while replacing exhaust valves and changing a piston. This amounted to an equivalent time off-hire of 1·2 days.
   The other ships listed in TABLE I had each incurred less than one day of equivalent time off-hire (up to July 31st 1974) in spite of the fact that some modifications were found to be desirable, particularly to the valve gear lubricating oil system. These modifications were carried out in harbour on one engine at a time and the ship could move using the other engine when required to shift berth.
3) *Spare Gear*—There has been no problem in obtaining spare gear. The Manufacturer's stocks in Australia and Canada were not complete until after *Cape Grenville* had been in service for over fifteen months. During that period, spare gear was flown out as required and at no time was any ship in difficulty. The heaviest item sent out by air was the spare cylinder head for *Baron Renfrew*.
4) *Shaft Alternator*—The shaft alternator was in use for approximately 90 per cent of the time underway and has proved to be a most reliable and maintenance-free source of power. In view of the saving in distillate fuel these vessels have cost less to run than the direct drive slow speed diesel equivalent.

THE TM 410 ENGINES IN M.V. *Cape Grenville*

By January 1973, there were over 60 TM 410 engines at sea and many were operating on residual fuel. No exhaust valve trouble had apparently been reported to the Manufacturers and an independent enquiry among other users had confirmed that there had been few, if any, failures. It, therefore, came somewhat as a surprise both to the Manufacturers and to the Owners when the port engine in the *Cape Grenville* showed a steady increase in the temperature from No. 2 port cylinder exhaust, and on the 14th March, 1973 after only running 897 hours, a hole about ½ cm across was found in the after valve of that cylinder (Fig. 10).

This was followed by a similar exhaust valve failure on 13th April 1973 after 1380 hours running and another failure on 19th May after 2070 hours. Two more valves were found to be slightly burnt when a routine inspection was carried out at Vancouver on 28th May. During this inspection six valve stems were found to be slightly bent.

The effect of these failures of caged exhaust valves on the ship's performance was very small. In fact, *Cape Grenville's* accumulated lost time (i.e. time on Charter but below Charter

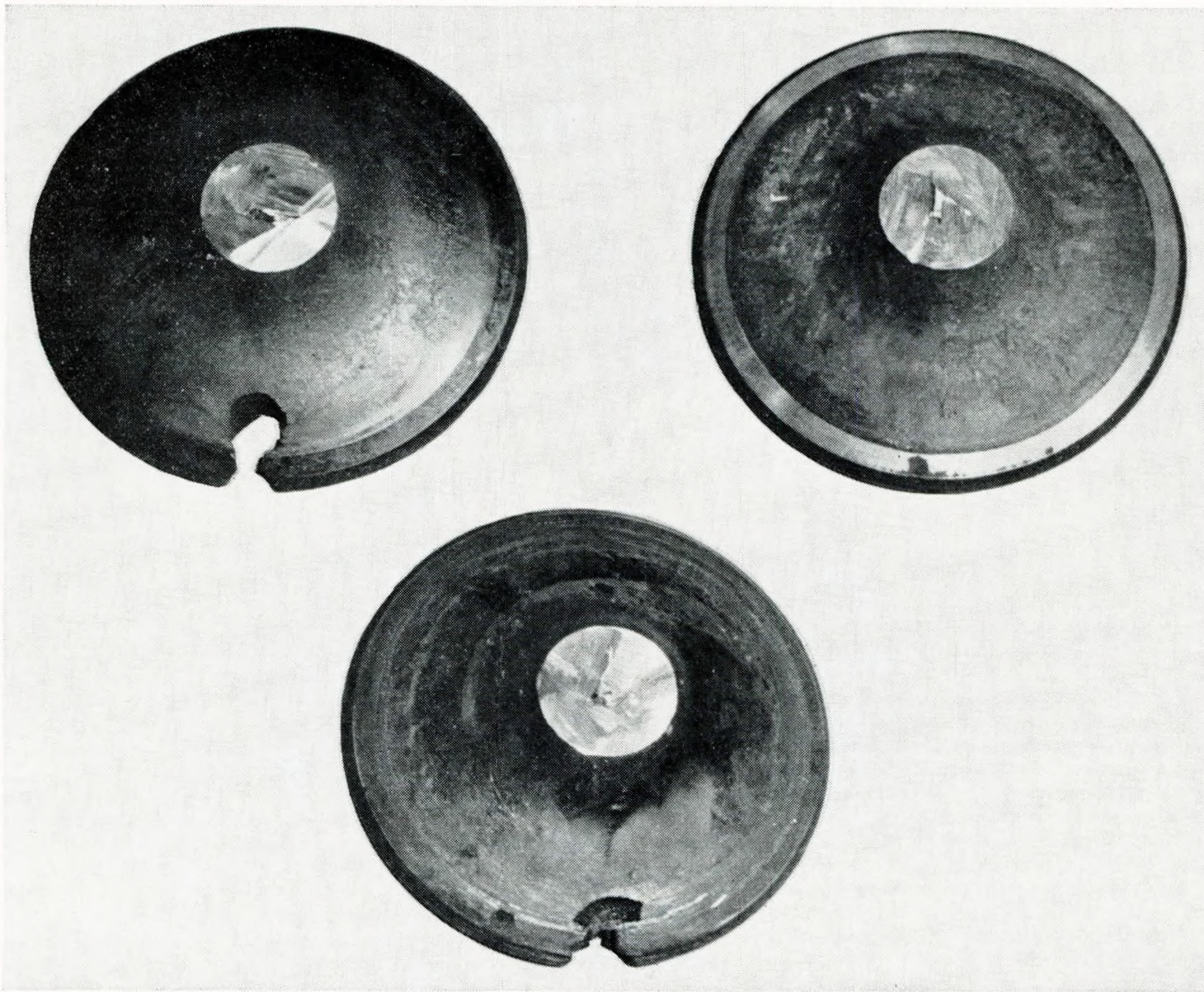

FIG. 10—*Failures of three TM 410 exhaust valves in* Cape Grenville *port engine*

speed) due to the main engines during the first 18 months service was less than one day. This was much less than that due to hull fouling and the routine 48 hours in drydock. Nevertheless, both the Engine Manufacturer and the Owners considered that a further investigation should be carried out to establish the cause(s) of these failures. Records showed that this was the first time that a TM 410 engine had propelled a bulkcarrier engaged in routine voyages of three weeks or more, operating U.M.S. and usually at constant power. It was thought there might be some unknown problem and that these failures should be investigated in depth. Various theories were put forward to explain the exhaust valve failures. These included:

High Vanadium

Cape Grenville might have taken in fuel with a very high vanadium content while bunkering before passing through the Panama Canal. To dispose of this possibility all the exhaust valves were removed from both engines and spare valves substituted. Thus no vanadium contaminated valves were left on board. However, occasional exhaust valve failures continued.

Sticking Valves

It was suggested that the trouble might have started by exhaust valves sticking in their guides due to inadequate lubrication. There was some evidence to support this theory. To begin with it also offered an explanation of the six bent valves. Secondly, the Chief Engineer of *Cape Grenville* had reported knocking noises from the engines which suggested that valves were occasionally sticking. Thirdly, the valve lubrication arrangements on the TM 410 engines consisted of a network of small pipes connected to distribution manifolds inside the cylinder heads and through flexible hoses to similar manifolds on the rocker arms. The distribution manifolds contained little plungers that delivered a small quantity of oil to each of the various lubrication points on the valve gear. The whole system was triggered by pulses of oil from a mechanical interval pump. This became known as the M.I.P. system. This system is very susceptible to air leaks. Any air in the system acts as a spring and prevents the high pressure pulse from activating the distribution plungers.

After re-engining with the TM 410, the *Cape Horn* suffered noticeable valve gear wear due to the failure of the pulse system to operate properly. At the Owner's request the engine manufacturers agreed to change the valve lubrication system on the TM 410 engines for all the bulkcarriers, including those in *Cape Grenville* and *Cape Horn*, to a continuous flow system for the rocker gear whilst retaining the pulse system to supply oil to the bottom of the valve guides. At the same time an annulus would be cut into the valve guide about 12 cm from the top with holes drilled into the annulus so that the exhaust blow-by would not interfere with the lubrication of the top 12 cm of the stem (Fig. 11). Apart from fitting a separate 60 litre oil tank and a small electrically driven pump in order to avoid risk of contaminating the entire lubricating oil system should a fuel pipe leak occur, the only changes were modifications of the piping system to and from the cylinder heads.

After 2000 hours of operation with this modified system *Cape Grenville* showed no measurable wear on five exhaust valve guides selected at random and there was certainly no

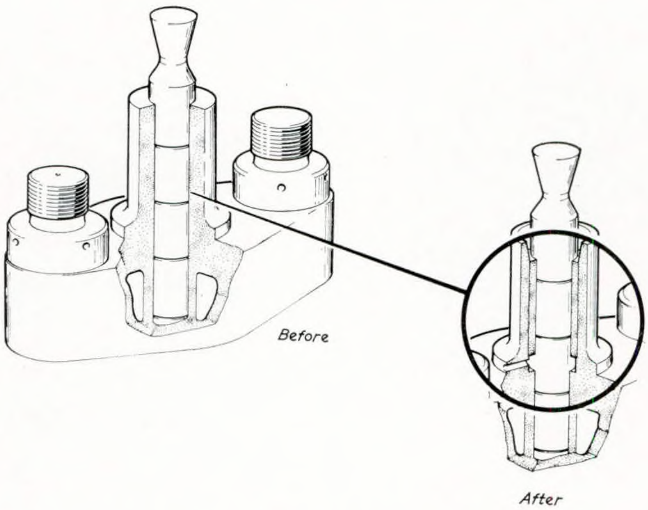

FIG. 11—*Modification made to a TM 410 exhaust valve guide in* Cape Grenville

sticking of valves. Similarly there was no further trouble with the valve gear of the engines in *Cape Horn.*

Fuel Viscosity

The majority of the TM 410 engines were operating on fuel of between 300 and 600 seconds Redwood and it was suggested that operation on 1500 seconds Redwood might be the cause of the exhaust valve failures. This was investigated in the first ship to be re-engined, namely, *Baron Renfrew*.

The A.O. 12 cylinder vee engines in that vessel were replaced by two 9 cylinder in-line TM 410 engines. At the same time the fuel systems were re-arranged so that each engine could be supplied with fuel from a separate tank. This enabled trials to be carried out on different fuels on two engines running under identical conditions. One engine was to run on 300 seconds Redwood fuel and the other on 1500 seconds Redwood fuel and ultimately on 3500 seconds fuel. Just after these trials commenced the oil crisis blew up and there was clearly no chance of choosing one's fuel for a long time to come. However, one significant fact did emerge. On the voyage to Australia via Japan burnt holes appeared in five exhaust valves of which three were in the engine using 300 seconds Redwood fuel and only two in the engine using the heavier fuel (600 to 1000 seconds Redwood). Although it is still the intention to complete these trials, including a prolonged period of running on 3500 seconds Redwood fuel, the initial results suggested that viscosity was not likely to be the fundamental cause of valve failures.

Steady Steaming at High Power

The theory that long periods of steady steaming with the exhaust valves at a more or less constant temperature in a salt laden environment, thereby allowing thick deposits to build up on the valve face, was, and perhaps still is, a possible part cause. However, a new light was thrown on the whole problem when, in February, 1974, the starboard engine in *Cape Grenville* was inadvertently allowed to run up to 630 rev/min (40 per cent overspeed) during a routine check of the trip gear. The only damage was to bend the exhaust valves slightly. All except two of the valves were replaced before the vessel was due to leave Vancouver for Sydney. It so happened that the author was in Sydney and able to meet *Cape Grenville* on arrival.

When the two remaining valves from the starboard engine were removed for inspection at Sydney they were both found to be just sufficiently bent to prevent the valve from seating properly all the way round. Part of the valve face was bright and the rest covered with a thin layer of deposit that could easily be removed with emery paper (Figs 12 and 13). Neither of these two valves was burnt, although the cooling effect of the water cooled seat must have been reduced. By contrast, another valve which had run for the same number of hours on the port engine had a hole burnt through the seat

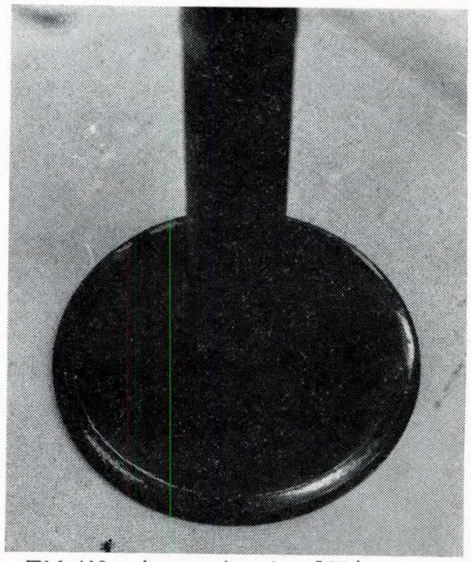

FIG. 12—*TM 410 exhaust valve after 500 hours operation with the seat* $\pm 1{\cdot}5$*mm out of true*

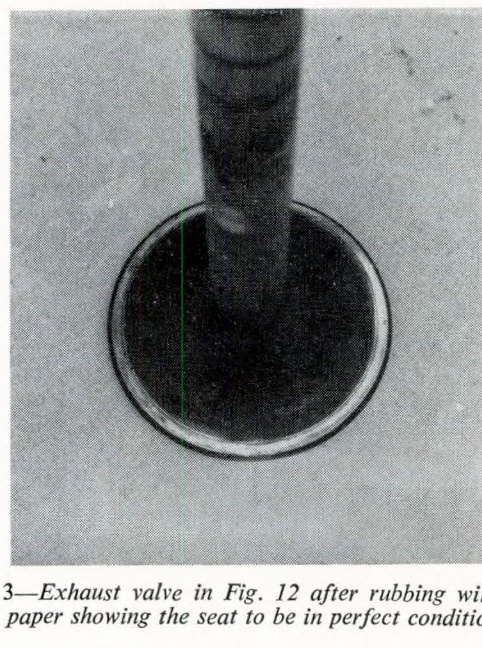

FIG. 13—*Exhaust valve in Fig. 12 after rubbing with emery paper showing the seat to be in perfect condition*

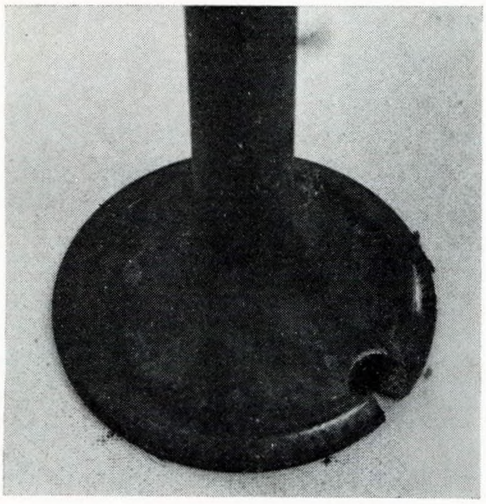

FIG. 14—*Burnt TM 410 exhaust valve with the same number of running hours as valve illustrated in Figs 12 and 13 but with a true seat*

(Fig. 14). This burnt valve was set up in a lathe and the face found to be true but it had a thick hard deposit round the periphery which could not be removed without scraping. The voyage from Vancouver to Sydney had taken approximately 500 hours. The exhaust temperatures had been about 380°C from both engines. This evidence suggested that:

a) in the absence of deposits, the stellited faces of exhaust valves did not break down, even when the valves were slightly bent (and therefore not so well cooled by the valve seat);
b) because these valves had been supplied with the same fuel processed by the same centrifuge and had run for the same number of hours some factor in addition to the fuel had had a major effect on the formation of the deposits on the valve;
c) a comparison with the bent valves removed from the starboard engine showed that the six bent valves discovered earlier were almost certainly the result of a previous overspeed.

By this time other TM 410 engines were in service and were providing significant evidence since, apart from *Baron Renfrew*, no other exhaust valve failures had been reported. The engines in *Cape Horn* had run for 2880 hours on similar fuel to that used in *Cape Grenville* without a single valve failure and at that time no failures had been reported from *Cape Grafton*, *Baron Inchcape* or *Baron Ardrossan*.

Tappet Clearances

It was suggested that incorrect tappet clearances might be an important cause of these failures. However, as all these engines had been built to the same standards and with the same tappet clearances it appeared unlikely that this was the case although that possibility could not be ruled out entirely. To make quite sure all ships were requested to check tappet clearances on their engines.

THE EFFECT OF LUBRICATING OIL

One of the factors that influenced the choice of engine to replace the Ruston A.O. was the low lubricating oil consumption of the TM 410. Reports from other owners suggested that the probable lubricating oil consumption at maximum continuous rating would be in the region of 0·85g/bhph (1·14 g/kWh). In fact the highest figure reported by other TM 410 owners was 1·6 g/bhph (2·14 g/kWh) and the lowest was 0·61 g/bhph (0·82 g/kWh). These figures covered engines ranging from 6 to 18 cylinders and included a number of 12 cylinder engines. The figure given in the manufacturers brochure was 0·8 g/bhph (1·07 g/kWh) at maximum continuous rating. For the 12 cylinder engine this is equivalent to 37·5 gallons (170 litres) per day.

The 12 cylinder TM 410 engines replacing the A.O. engines were set to run at a maximum speed of 450 rev/min in order to avoid the additional cost of modifying the gearbox ratio. At 15 kn these engines usually have a mean effective

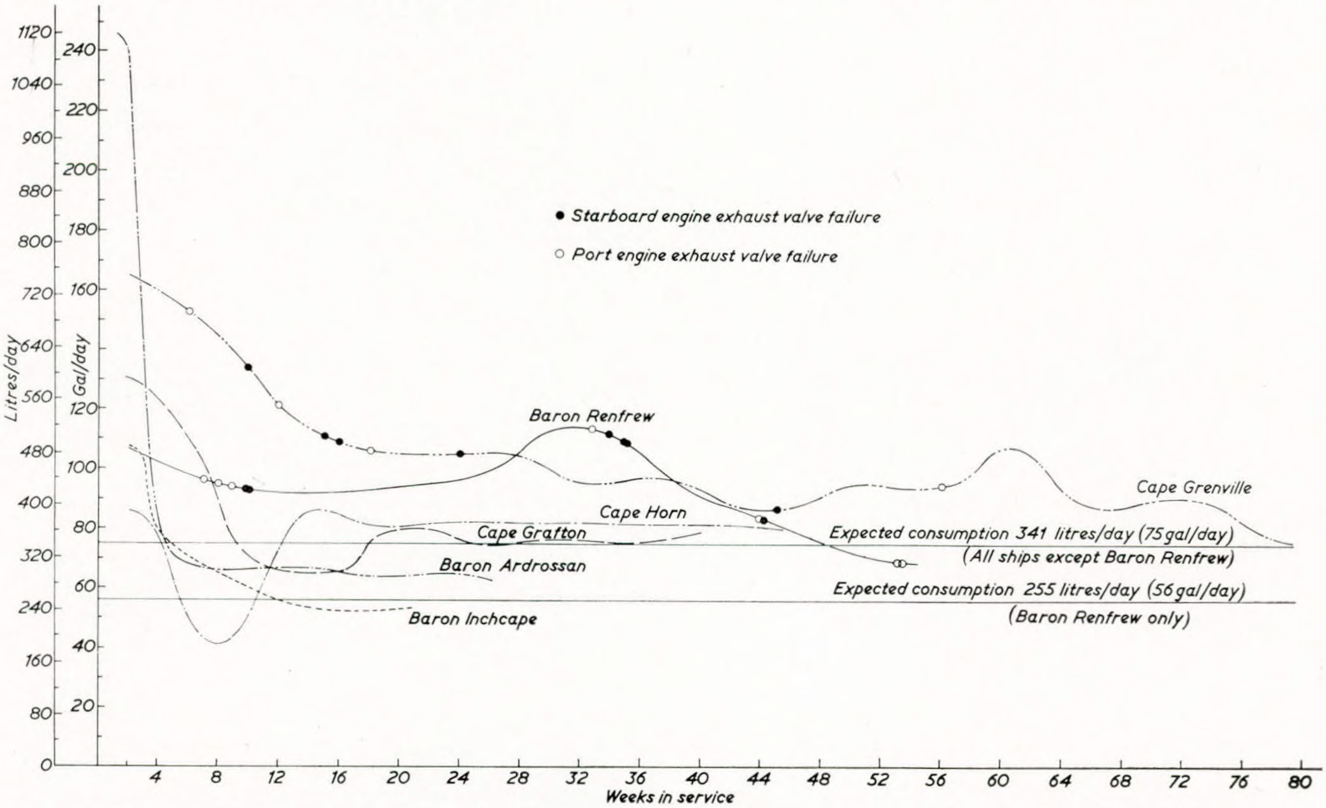

FIG. 15—*Curves of lubricating oil consumption showing points at which exhaust valve failures occurred in* Cape Grenville *and in* Baron Renfrew

pressure of about 80 per cent of the continuous service rating figure, depending on the length of time out of dock. At sea the shaft generator is normally in use and this requires engine speed to remain constant at 450 rev/min.

Curves of Lubricating Oil Consumption (Fig. 15)

Lubricating oil consumption is difficult to measure accurately on a day to day basis in these ships because the tanks under the engines are long and flat. However, over a period of one month it is possible to arrive at a reasonably good figure provided all the oil rejected by the automatic back flushing filters is properly accounted for and leaks are not ignored. The curves shown in Fig. 15 have been plotted after taking into account all known losses. It can be seen from these curves that the lubricating oil consumption of the twin TM 410 engine installation in three of the ships dropped to approximately 75 gallons (340 litres) per day per ship after about four months service and this has now been taken as a good average figure for the 12 000 hp (8950 kW) twin TM 410 installation. For the twin 9 cylinder installation in *Baron Renfrew* the target has been set at 56 gallons (255 litres) per day i.e. 28 gallons (127·5 litres) per engine.

It will be noticed that the total lubricating oil consumption of the two engines in *Cape Grenville* has been well above 75 gallons (340 litres) per day for the first thirteen months after the ship entered service and has only recently started to drop towards that figure. The *Baron Renfrew* has also had a high lubricating oil consumption for the first twelve months after entering service. These are the only two ships which have had consistently high level of lubricating oil consumption for more than five months; also, these are the only two ships with a significant number of exhaust valve failures (Fig. 15).

It seems possible therefore that one important contributory cause of these exhaust valve failures is the high consumption of alkaline lubricating oil. The calcium residues from high T.B.N. oils could assist in the formation of the hard deposits found on valve seats. These deposits would eventually permit channelling to occur and the stellite would subsequently break down in the manner described by Zinner[2] and Ayabe, Yano and Motooka[3].

High temperature accelerates the process, obviously. Exhaust valve failures occurred earlier in the 9 cylinder engines of *Baron Renfrew* probably because the average exhaust temperature from the 9 cylinder engine (410°C with a maximum of 440°C) is significantly higher than the average exhaust temperature (380°C) from the 12 cylinder engine of *Cape Grenville*. However, it seems possible that there would have been no exhaust valve failures in either ship if the lubricating oil consumption had not been so high.

Lubricating Oil Supply to the Inlet Air

Wear on the inlet valve seats has been avoided completely in the TM 410 by the injection of lubricating oil into the inlet air in accordance with the manufacturers instructions. The amount required is approximately 7 gallons (32 litres) of oil taken from the sump of each engine, i.e., about 20 per cent of the lubricating oil consumption for the entire installation. A deposit of hard carbon has been found on some of the inlet valve spindles. Injection of oil into the inlet air is a simple and effective method of preventing heavy wear of the inlet valve seats but it would clearly be worth using a cheaper oil and, in view of the probable connexion between an excess of alkaline oil and exhaust valve corrosion[3], an oil that does not contain calcium.

INITIAL LUBRICATING OIL SUPPLY

The burnt exhaust valve problem in *Cape Grenville* would probably have been avoided completely if, in addition to using a straight mineral oil to prevent fretting of the inlet valve seats, the initial charge of oil into the sump had been a good detergent oil with a low T.B.N. The normal 25 T.B.N. oil would be supplied to the cylinder liners and the sump T.B.N. would have risen slowly to the normal equilibrium figure of 18 to 20 T.B.N. during the first few weeks running. During this period, excess calcium would have been avoided during combustion.

REASONS FOR THE HIGH LUBRICATING OIL CONSUMPTION IN *CAPE GRENVILLE* AND *BARON RENFREW*

The TM 410 engines in the *Cape Grenville* were supplied while the ship was being built at Haugesund. It so happened that at that particular berth there was no convenient crane capable of lifting the 12 cylinder TM 410 engine into the ship. As a result, the TM 410 which is over 40 tonnes heavier than the equivalent A.O. engine had to be partly dismantled by the Engine Manufacturer and then re-assembled after installation in the ship.

The pistons fitted to the engines in *Cape Grenville* were originally to an old design which did not have cut-out areas in way of the exhaust valves. With the engine dismantled, the Manufacturer decided it would be worth changing the pistons to the new design. The old rings, however, were put back as these were (and are) standard. The engine had already run for more than 30 hours on the test bed before the change of pistons and removal and replacement of rings took place. It therefore seems likely that slight distortion during the change prevented the rings, and particularly the scraper ring, from sealing properly.

There is no obvious reason why the two 9 cylinder engines in *Baron Renfrew* should use any more oil than any other 9 cylinder TM 410 engine. The lubricating oil consumption is now decreasing but further investigation is necessary.

Additives to Counteract Exhaust Valve Corrosion

No additives have been used for this purpose in any of these engines, because, even if they are as effective as their protagonists claim, the cost is much more than the cost of replacing a few burnt valves in a twin engined ship. To treat a single ship with one well known brand of fuel additive for 12 months costs as much as replacing 40 burned exhaust valves a year. With an anticipated failure rate of perhaps two or three exhaust valves a year, there is clearly no point in doping the fuel.

LUBRICATING OIL SUPPLY AND FILTRATION

One of the most attractive design features of the TM 410 is the arrangement whereby 80 per cent of the fresh lubricating oil supplied to the engine is fed directly to the liners. This means that the surfaces carrying the highest load, i.e., the compression rings and the cylinder liner, always receive a constant supply of new clean oil at the right T.B.N.

The automatic back-flushing filters used in conjunction with the TM 410 are satisfactory but unless care is taken to note the amount of oil rejected and then subsequently to purify and return this oil, erroneous figures for engine lubricating oil consumption can, and do, easily arise. It might be preferable to use a large, single cartridge filter with ample capacity and throw away the cartridge every year, even if the pressuıe drop across the filter is still low.

SHAFT DRIVEN ALTERNATOR

When both main engines are in use the shaft driven alternator has proved to be an extremely reliable source of power to the main engine auxiliaries. It takes power from engines burning residual fuel and therefore should produce cheaper electricity than that supplied from the two diesel alternators.

The only problem is to implant the necessary degree of cost consciousness in the mind of every Master or Chief Engineer. It is sometimes overlooked that if a shaft alternator is used 3 kn below the design speed of the ship, any savings from using residual fuel to produce electricity will be swamped by the loss of efficiency of the propeller. For prolonged steaming at low power the direct diesel alternators are used and the main engine speed reduced. This applies whether running on one or two engines, obviously.

THE CONTROLLABLE PITCH PROPELLER

Experience with this piece of equipment during the last three years has been somewhat mixed. The main problems have been the corrosion and cracking of the stainless steel propeller hubs. This phenomenon was first noticed in the *Cape Race* and the *Baron Belhaven*. These ships are fitted with large bore direct drive diesels and the fluctuation in propeller torque is greater than with the medium speed installations.

Hub failures are consequently more dramatic. However, the root cause of the trouble has been the same both in the large bore and in the medium speed ships. This was the use of a stainless steel with too low a chromium content for the hub body.

The propeller Manufacturer realized this and to avoid corrosion behind the seal rings, an austenitic insert was welded into each blade cavity. The welding was not followed by the necessary heat treatment to allow the chromium to soak back into the heat affected zone round the weld and consequently corrosion and pitting occurred in that area. A line of corrosion pits developed in a highly stressed region of the propeller hub and four serious cases of cracking occurred (Fig. 8). In each of these, a substantial quantity of oil was able to leak out of the system and sometimes water entered into the hydraulic oil tank. Small cracks were detected in 9 other hubs during drydocking and all the hubs had to be changed.

This trouble now appears to have been cured and there is no doubt that the controllable pitch propeller gives excellent manoeuvreability and is very well liked by every Master who has handled a ship fitted with one. The fact that these ships were fitted with the controllable pitch propeller is now undoubtedly an advantage. Whether, in view of the high initial cost, one would be justified in fitting a controllable pitch propeller to a new construction medium speed diesel bulkcarrier installation appears to be open to argument.

DESIGN STUDY FOR FUTURE NEW CONSTRUCTION

In February 1974 a machinery design study contract was awarded to a well known firm of marine engineering consultants. The aim of the study was to optimize the type of machinery to be fitted to a 26 000 dwt bulkcarrier to give the best return on capital.

The machinery alternatives considered were a range of TM 410 engines and an equivalent large bore engine of proven design. A feature of the study was that manufacturers' data, often obtained under idealized conditions, would be given some credence but not to the same extent as information obtained from actual service experience. Medium speed engine experience and data were drawn largely from the performance of the first TM 410 engined bulkcarrier, *Cape Grenville*, which by then had been in service for over 12 months. Additional experience from the other TM 410 re-engined ships was made available as the study progressed. Experience of the large bore engines was based mainly on results obtained over some years from ships fitted with and without controllable pitch propellers.

Altogether, six machinery alternatives were examined and the through cost calculated for a period of 30 000 hours. The main conclusions drawn from the study were as follows:

i) If the reliability of all the systems was assumed to be the same, the installation with the lowest combined capital and operating costs was the single geared medium speed engine. However, in practice its reliability would certainly be no better than that of a single large bore diesel and in fact, with the extra number of cylinders, its reliability might be slightly worse.

ii) The single large bore diesel had a higher combined capital and operating cost than either of the two twin medium speed engine installations using standard gearboxes.

A ship with a twin engined medium speed installation would be much less likely to lose all power at sea and would have a greater overall availability than a similar ship with a single engine. For example, if a large bore and two medium speed engines each had four stoppages per year and each was of six hours duration, then the chance of the ship coming to a complete halt would be 4:1 on for a single engined ship. For a twin engined ship the odds are one chance in every 32 years. This figure would have to be modified slightly to take into account the possibility of a gearbox failure.

The optimum machinery configuration for a 26 000 dwt bulkcarrier was therefore either a twin engined medium speed installation with a fixed pitch propeller or a similar installation with a controllable pitch propeller. The latter installation (controllable pitch propeller and shaft alternator) was, on the data available, marginally more economic than the fixed pitch propeller installation which had conventional diesel generators. This was largely due to the saving of the extra cost of the distillate fuel.

The study did not include the use of a waste heat recovery system for power generation. However, calculations carried out since the study was completed suggest that if such a system was incorporated into the twin medium speed installation with a fixed pitch propeller, the overall costs differentials between the competing twin engined installations would be negligible.

CONCLUSION

The experience to date with six ships re-engined with the TM 410 has shown conclusively that the decision in 1968 to use twin medium speed diesel installations for bulkcarrier propulsion was correct but that the original engine chosen was not suitable for bulkcarrier operation.

A design study covering six alternatives and based on 1974 material and installation costs supports this view.

ACKNOWLEDGEMENTS

The Author wishes to acknowledge the assistance in the preparation of this paper by his colleagues at Scottish Ship Management and in particular by the Chief Project Engineer, Mr. F. M. Lo.

This paper is published with the permission of the Scottish Ship Management Board but the responsibility for any statements of fact or opinion rests entirely with the Author.

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Discussion

Mr. D. G. Edgar, F.I.Mar.E., said that much of the first part of the paper was in some ways history, but there were a couple of points which, with hindsight, could be looked over. The problem of the valve failure in the A.O. had been largely overcome, not by fitting the very expensive Nimonic valves, but by recognizing that, apart from the need for recessed flame plates and increased tappet clearance, the valve seats when finally ground at the maker's works were finished by a machine which faithfully followed the existing machined geometry of the seat. Experience had shown that the seats finished in this manner could be anything from 0·005″ to 0·0625″ or 1·5 mm (and on occasion even larger amounts) out of concentricity with the valve stem guide centre and hence the valve head. The purchase and use of a more sophisticated valve seat grinding machine which precisely centred the valve seat to the guide centre and virtually guaranteed a full face gas-tight seating of the valves reduced the incidence of valve failure from approximately 10 or 12 per month to five or six per year, given that all regrinds and even new replaced flame plates were done. The machine, whilst fairly expensive compared to the former types mentioned, would not cost as much as five or six Nimonic valves, and gave reasonably trouble free service when using the much cheaper EN 54 steel valves. Investigation of this aspect might interest the author with regard to his burning and bending problem in the valves of the TM 410 engine.

Mr. Edgar agreed that the rigidity of the "space frame" would seem to have been less than its designer would have hoped. The major cause of the circumferential grooving of the liners at the upper travel limit of the top ring probably stemmed from the design of the strut type load reverser fitted in the Mk 10/8 piston. Over the short arc at top dead centre where the load reverser was active, the piston became unstable, being supported solely on a $1\frac{1}{4}''$ (6·35 mm) diameter strut, and a high frequency "shudder" (for want of a better descriptive word) was experienced for a very short period, thus producing a groove not much wider than the top ring itself. The validity of this hypothesis was borne out by the fact that pistons of the Mk 5 type with hydraulic load reversers did not groove the liners, but did have a wear pattern which might be attributed to the distortion of the frame in conjunction with other factors, since the wear pattern extended further down the liner, thus perhaps confirming the author's opinion about chrome oxide debris.

Although these experiences were largely history now, since the A.O. had virtually ended its career as a marine propulsion engine, they could be of value as guidelines should similar problems emerge in other engines.

Ir. J. H. Wesselo, F.I.Mar.E., said in the name of Stork-Werkspoor and of the TM 410 he would make a few introductory remarks, and comment on a few points mentioned by the author.

In technical management and design there were two basic attitudes. The first, the natural attitude of the technical man, was due to the fact that if he was a good man he was a man with ideas; particularly if these ideas were his own he would defend them, and thus give the impression of being rigid and conceited. If he did not escape this first stage of wisdom, he was really an egg-head. If he did, he might adopt the second attitude and turn into a flexible man, ready to change his ideas at every indication that something was was wrong. After some time he would find the other ideas also had some disadvantages. Thus, this second degree of wisdom would lead to a process of cycling of designs going on all the time, improvements being introduced continually, but teething troubles also going on continuously. Having learned this, the third degree of wisdom might appear: this implied being flexible in ideas, but trying to be consistent in principles, and not dropping a good idea on secondary observations.

Ir. Wesselo said that as compliments to the author were not encouraged he would just make a cool remark that the decision to stick to the idea of medium speed propulsion was a demonstration of this third degree of wisdom. Another example followed.

Valve Gear Lubricating System

This system as applied and described in the paper was considered an intermediate solution giving better lubrication but entailing a separate system of oil circulation. Ir. Wesselo's Company now had a different system which still maintained the advantages of the original system. The improvements included the following:

a) the high pressure part of the system was limited to fixed components, seldom needing to be detached;
b) the distribution part of the system was equipped with flexible tubes with automatic couplings which also prevented air entering;
c) small pipes were thicker and stronger;
d) the total quantity of oil was increased, but was still so little that it could be drained.

These improvements, it was thought, would be nearly ideal, but results must prove them, alternatively the Company would be flexible enough to change the system again.

Exhaust Valve Behaviour

Ir. Wesselo said that his Company would have to become accustomed to the idea that exhaust valves could burn. Up to a year and a half ago this had not been known with the TM 410 engine. The general situation over the fleet today was still favourable but, also, in some other ships burning of valves had occurred now and then: more than 200 from all TM 410 engines in service; about the same number of burned valves had been reported as from the ships mentioned in the paper alone. This meant that at about one per cent of all exhaust valve positions, a burned valve had occurred once.

For 99 per cent of the cases Fig. 16 illustrated the normal picture:

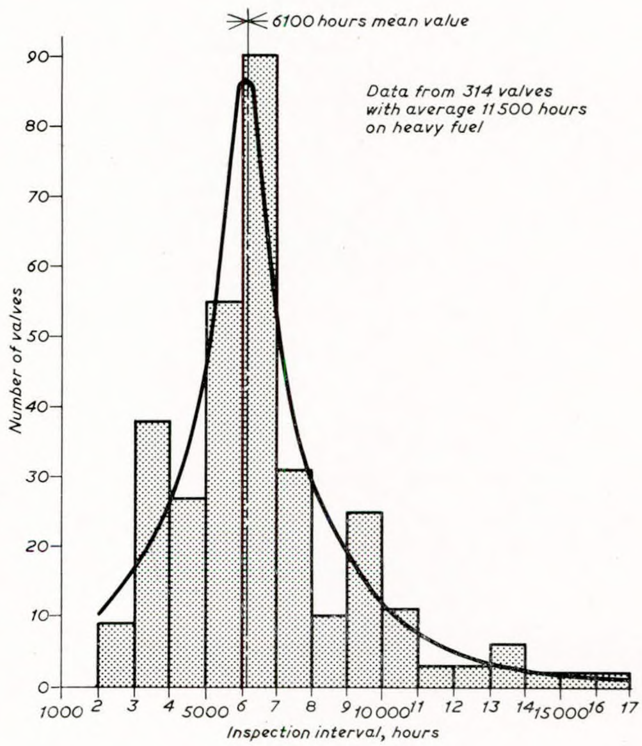

Fig. 16

The data were inspection times as applied by users and a result of statistics from 19 ships, all ships using heavy fuel and with more than 5000 running hours it could be seen that most people accepted 6000 h which was in the instruction manual data, but some people did not believe they would make 6000 and checked after 2000 and others ran much longer; the latest recorded had been running 16 to 17 000 h.

As could be seen from Fig. 17 all these valves showed some black patches which disappeared by light lapping, and all of them were used again.

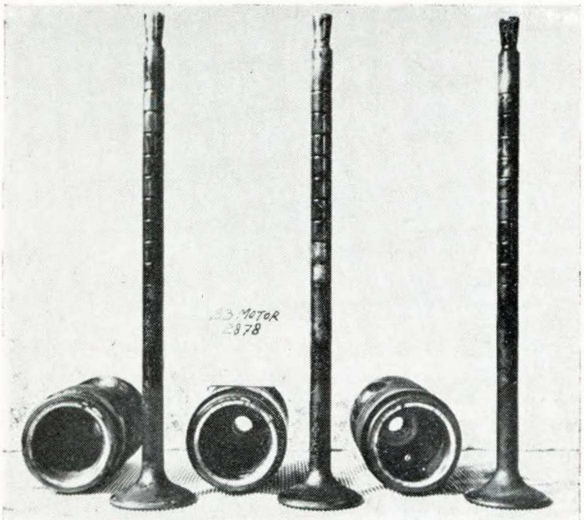

FIG. 17

The cause of burning valves was not quite clear. Everyone knew the process described in literature; vanadium and sodium being the critical materials, but what could be done to prevent burning, with certainty, was another question.

From the other random spread incidental cases no relation with high lubricating oil consumption could be found, or it should be higher lubricating oil consumption in individual cylinders.

Although the author had observed bent valves without and true valves with damage, Ir. Wesselo said he did not believe the eccentric seating was favourable, and therefore had not completely ruled out engine overspeeding as a possible cause of valve burning. Also defects in the valve gear lubricating system were still suspect.

Therefore in the first place the overspeed cut out system and the valve gear lubricating system had been improved, rendering the engine less sensitive to human error or to less than ideal maintenance. In the second place it was very important to prevent high lubrication oil consumption, and he thought the most effective way was to handle piston rings with great care. Deformation must be avoided, and piston rings removed at inspection must be fitted in their original groove and position, or preferably renewed.

Lubricating Oil Consumption

In TM 410 engines this was generally a very reliable matter. The piston ring pack, applied for the first engines supplied, had never been changed, and was still applied today.

Fig. 18 showed a piston inspected after 17 000 service hours in one of the first engines in the ocean-going tug *Rode Zee*, which used 0·85 g/bhph as an average over three years and this was still the normal figure for TM 410 engines. The ring pack consisted of a chrome plated spherical top ring; three cast iron compression rings with bronze insert; one conformable slotted scraper ring and one simple block-ring at the bottom. This normal figure was so reliable that a cause outside the engine could nearly always be found for a higher consumption. If piston rings had been spoilt running in would take longer, but the consumption would come down, otherwise the Company had a strong suspicion of other sources.

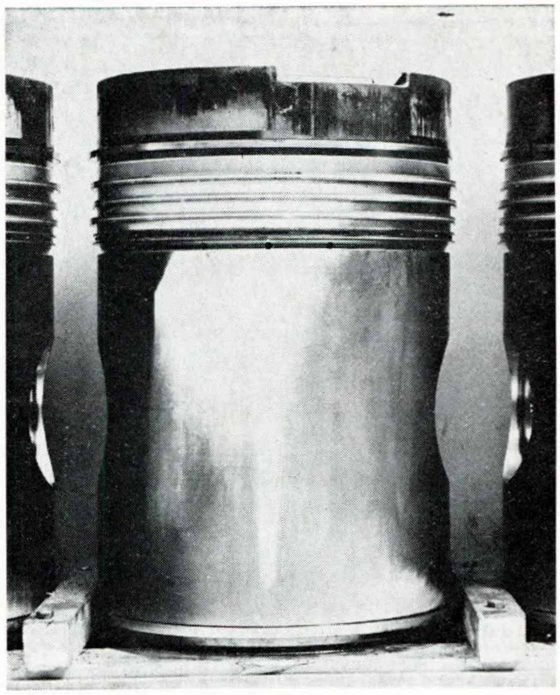

FIG. 18

Off-hire

Ir. Wesselo said that this was the last point he wished to make. He agreed with the author that a twin engine installation was fundamentally better in terms of off-hire, so his own Company would promote this idea.

The resulting off-hire figures of main engines of 20 ships, 32 engines with more than 5000 service hours, twin-engine installations were 0·48 per cent. Unfortunately the theory was spoilt because TM 410 single engine installations appeared to be even better at 0·375. Obviously the statistical material was too small to prove the statement.

This was perhaps the time to say that he tended to be less dramatic than the author about the dangerous diesel engine conception with high thermal and mechanical stresses. Now, all important processes were controlled, all main components ran perfectly. Of course, little things could be annoying enough, and that was the reason for trying so hard to rule out the small problems also. This process would never be completely finished, but looking to the diesel engine history, Ir. Wesselo felt that achievements had been made too.

MR. T. K. L. RENKEMA referred to the last part of the author's statement about various studies of the appraisal of medium speed engines; the Company had studied bulk-carriers of 26 000 dwt and 140 000 tdw and a tanker of 70 000 tdw, alternatively equipped with crosshead engines and medium speed engines.

From these one could see there was in each case considerable difference in the total annual operating costs of the ships to the advantage of the medium speed engine alternative.

In fact such studies made by themselves and by third parties as well gave a lower costing rate per tonne/mile of 5 to 8 per cent for the "medium speed" ship.

MR. P. A. D. FENTON, M.I.MAR.E., said that in Holland, England and the United States Van der Horst, Europe had chromeplated the great majority of all the A.O. liners produced and re-standardized; comparatively few had been plated in Australia and Japan, and the information at the time was that the performance was certainly no better and the price very much higher.

The Company had also worked very closely with the builders in their particular part of the R & D programme on the liner wear problem, which was the chromeplated bore surface.

The author had advanced, as a possible cause of random liner wear, lack of lubrication at the top of the liner, first suggesting this might be because the engine was a 2-stroke, but it would appear to be a little ingenuous to suggest that the porting in a 2 stroke engine left insufficient oil on the surface above the ports to prevent excessive liner wear, even in a trunk piston engine normally without separate cylinder lubrication. In the very many A.O. liners the Company had looked at, they had never found any evidence of lack of a lubricant film. He wondered whether the author had found any. It was significant that even with accurately timed injection of oil into the critical wear area, no improvement had been obtained.

Secondly, the author had suggested that the lubricant film might be destroyed by blow-by. This was of course always a possibility, and sometimes occurred above the exhaust ports on loop-scavenged engines but not, in his experience, at top ring reversal on uniflow engines, especially with the oil consumption of the A.O. In his experience there was always a little blow-by until the rings bedded in, and it was possible that in some units the ring and liner wear were so rapid that real bedding in was never achieved. However, once rings were bedded in, blow-by usually followed, and did not cause stuck rings, and these also caused heavy ring and liner wear. Again, the normal cause of stuck rings was too much lubricating oil, or the use of an oil with too high a T.B.N. for the sulphur content of the fuel. Blow-by could also be caused by eccentric liner wear or, indeed, by the circumferential deformation of the liner.

Thirdly, the author had stated that a possible cause of the trouble might be the use of a chromium plated liner. This was a wrong hypothesis drawn from wrong premises, said Mr. Fenton, and it was not supported by facts. Porous chromium plate, as supplied in the great majority of A.O. liners, was not more difficult to lubricate than cast iron; it was easier, and in trunk piston engines, 2 stroke or 4 stroke, lubricating oil consumption had been reduced in comparison with that on cast iron, with simultaneous improvements in liner and ring wear rates, by changing to a porous chrome-plated bore. Further, in the A.O. engine, as the author certainly knew, plain cast iron and various specially finished cast iron liners had been tried without finding a solution to the problem.

Mr. Fenton said he did not understand why the author ventured upon this particular possible cause, which he subsequently appeared to withdraw.

He had been delighted to see that the author had shown as an example of a stiff engine frame the Smit Bolnes engine because all production liners for this engine were chrome-plated.

The author's suggestion that hard chromium oxide debris from his so-called fretting action was embedded in the rings, so causing general wear over the length of the liner, was also not in accordance with the facts. Particles of chromium oxide embedded in the rings must cause scores the full length of the liner; this was unusual in the A.O. liners except where rings were stuck or broken, and the running surface of the liner was normally smooth and the wear limited to the top 4 to 6 inches (100–150 mm) of the swept column. The consequent wear to which the author had referred was not general down the length of the bore.

These remarks of his were a little misleading, and not in keeping with the high standard of the rest of the paper. Perhaps for that reason the author had glossed over this part of the paper in his presentation.

Clearly, said Mr. Fenton, if they had been able to assist the builder in solving the three problems of liner and ring wear, and excessive oil consumption, this paper could more happily and more productively have been devoted to describing the solution to the problem. However, as far as they knew the problem remained unsolved, and they must agree with the author that the probable cause lay in the unique space frame design of the entablature of the A.O. engine. However, they did not agree that a possible cause of the liner wear was a fretting action; fretting resulted from a continuous high frequency, relative movement between two parts restrained in close mechanical contact over a finite length of time. These conditions did not obtain between the ring and the liner at top ring reversal, or anywhere else in the stroke.

It was significant that the same circumferential groove (shown in Fig. 3 of the paper) near top ring reversal was often visible, though to no measurable depth, at the bottom of the stroke, and he believed that these indicated a resonant vibration between the liner, gripped insufficiently rigidly in its resilient supporting frame, and the piston and its rings. It was in fact a high frequency mechanical hammering action which wore away the liner at the top.

It was suggested, therefore, that it was a combination of the resilient low mass space frame with rather a flexible liner carried at the top which caused the random wear which had never been satisfactorily explained.

It was also suggested that the wear rate was a function of the magnitude of this resonant vibration of the liner, and that the magnitude of the vibration depended on the variation between limits in the tolerance between the external locating diameters of the liner and the bores in which these were held in the entablature. The impact would be a maximum at the "free" end of the liner, and the hammering action could be increased by an induced ring flutter.

In this way the random wear pattern could depend on the individual fit of each liner in the engine, and a "wear/no-wear" situation could exist between liner and liner, irrespective of all other parameters in the engine.

DR. R. A. NORRBY said that Mr. Bowers had dealt not only with medium speed diesel engines but also with KaMeWa propellers for the Owners' 14 ships, and comments were needed on some of the paragraphs.

As at now there were approximately 2500 KaMeWa controllable pitch propellers in service. Approximately 80 per cent of these were in stainless steel, and the rest in bronze. In the early seventies there was a slight change in design and production of the stainless steel propellers, which had resulted in problems with cracks in a number of cases. However, these problems had now been overcome, and where cracks had appeared they had been repaired, as Mr. Bowers had mentioned. In some cases replacements had been made.

It was said in the paper that a line or corrosion pits had developed in a highly stressed region of the propeller hub. This was not so, as the region in which the pits had developed was to be considered as having low or moderate stress, in the order of 19–39 MN/m². The yield stress of that material, based on standard tests, was at least 440 MN/m².

Mr. Bowers had mentioned that for the case when the ship speed was reduced by approximately 3 knots, which condition could be reached with, for instance, one engine driving, it was better to reduce the shaft speed and change over to direct diesel alternators. (See Fig. 19.)

In the diagram, which was valid for *Baron Renfrew*, the Stork-Werkspoor 9TM 410 load curves had been drawn.

The point made in the paper was that one should, with one engine drive, use reduced shaft speed, whereas his Company recommended full shaft speed, in this case 118 rev/min. With one engine driving there was no possibility

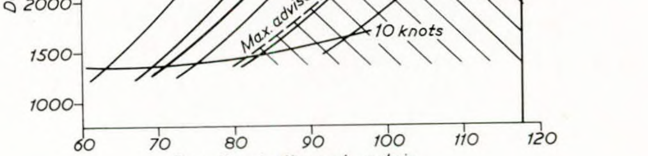

FIG. 19

of using a fixed pitch propeller instead of the controllable pitch propeller (CPP), as this would overload the engine excessively. Dr. Norrby said he had indicated in the diagram that the fixed pitch propeller would have a P/D ratio 0·8; the engine would never reach that, not even with this load curve.

A cost comparison was presented: the input data, in addition to the diagram mentioned, were fuel consumption curves for the actual diesel and fuel prices in Rotterdam in the fourth quarter of 1974. The power for the alternator had been assumed to be 90 per cent of the maximum value. Two cases, concerning one engine drive at approximately 12 kn, had been studied, viz:

1) CPP with shaft driven alternator at full rev/min (CSR);
2) CPP with reduced shaft speed and auxiliary diesel alternator (power as per maximum advisable load curve).

Case 1), which was the standard with CPP and shaft driven alternator, was just as favourable as case 2) advocated by Mr. Bowers in the paper. There was practically no change compared to using 1972 fuel prices.

The author had stated that it was favourable to use the shaft driven alternator because it had proved to be an extremely reliable source of power. Further, the obvious saving in lubrication oil and spares as well as maintenance cost by using the shaft driven alternator instead of the direct diesel alternators must be taken into account. Thus it was questioned on what the author based his recommendation regarding reduction in shaft speed with a CPP in combination with shaft driven alternator at one engine drive.

Dr. Norrby said, on controllable pitch propellers, it was not difficult to design such a propeller for quite an appreciable time on one engine and at full shaft speed, but one had to know this from the beginning, or else one could make minor changes to the propeller after a drydocking.

Mr. D. B. M. MATHEWS said, as a member of a Company which, although perhaps on the sidelines of the marine industry as presented today, was engaged as designers and suppliers, he would like to say that he and his colleagues were most interested in all aspects of progress in this field. They were indeed very fond of contemplating the challenges and attractions of life at sea, but essentially from the comfort of firmly shore-bound armchairs. He wished to record their deep interest in this paper, admiration for the author, his staff, the ship repair yard and the engine builder, and the tremendous achievement described in the paper.

He had been particularly impressed by the comprehensive staff work of all departments. Without their efforts this challenging project could not have succeeded. It was quite clear that the Company had a first class management team who could formulate and execute big decisions quickly and effectively, an ability not notably prevalent amongst management teams in these days.

Mr. Mathews said he found himself full of admiration for the sponsors of the undoubtedly complex contractual arrangements. Judging by the relaxed disposition of the author, there were clear indications that the outcome of this operation was likely to be profitable, which must mean that the contractual organization must have been sound. Indeed, with the many variable circumstances which could not be spelt out with any certainty, these arrangements must have been particularly challenging to all the parties concerned.

Mr. Mathews said he would be grateful for an outline of the philosophy incorporated in the financial framework of those arrangements, as, he was sure, would all who were aware of this fascinating story. Why was it decided to deal separately with each party rather than have the yard as a main contractor?

It was of course not possible for the author to enlarge on all aspects of this fascinating case history in his paper, but Mr. Mathews said he would much appreciate any further comments on the following points:

Although Mr. Bowers and his staff made a deep appreciation of all suitable propulsion engines for the re-equipment programme, what long-stop arrangements had been made to cover the unlikely eventuality of further trouble after the re-engining programme?

Would the author care to comment on the performance he had really anticipated at the time of committal, bearing in mind that guarantees were of little value in the event of problems arising after a change of this magnitude?

In the paper reference was made to "Change of gear ratio". He was sure this evening's audience were amazed, as he was himself, at the way this aspect of the conversion had been executed, particularly bearing in mind the total absence of subsequent problems. Perhaps the author would care to comment on the work involved in this area, and how it had been so successfully executed; it was a major item.

Reference had been made to the significant changes made at short notice to the engine valve lubricating systems. Would the author please enlarge on this event, and outline the major problems involved both in the material execution of this change and in extracting the obviously full co-operation of the engine builder to adopt this major change of design philosophy?

With regard to improvements to the original engine lubricating system they had heard about, Mr. Mathews said he was intrigued in so far as it appeared that a simple lubricating system met the requirements. Why then was the engine builder progressing with the original philosophy, which was more complex, more expensive? He was sure there were good reasons for this and would like to hear what they were.

Another interesting area was the lubricating arrangements made for inlet valves. The majority of engines currently available required no such special arrangements: why were they necessary in this engine? Investigation developments must be going on towards eliminating this aspect, which complicated things, as the engine used more oil and it cost more money. If this consumption could be avoided, the overall performance of this engine must be extremely attractive.

Mr. Mathews said he was also very interested in the assessments of propulsion arrangements mentioned by the author.

Obviously there were many more broad issues concerned than those the author had been able to mention. Why had he specifically chosen a life of 30 000 hours? Obviously broader parameters had been considered: could he expand on that?

In conclusion, Mr. Mathews thanked the author and all those who had assisted him in the preparation of this most interesting paper; personally and privately he was interested in the final disposal of the superfluous ironmongery arising from this operation, and would not be surprised to hear that a profit had been made from it.

CDR. E. R. MAY, F.I.MAR.E., said that his interests were limited to controllable pitch propellers. Towards the end of this paper a very careful reading detected a slight wavering on Mr. Bowers' part of his faith in them. Commander May could not have that, for his life was bound up with them in his work.

With regard to corrosion and hub cracking, no material was entirely immune to either trouble, but the record so far was that bronzes were better than stainless steels. To the best of his own belief, and he was open to correction, there was no known case of a large high tensile bronze CPP hub suffering cracking except in connexion with a weld repair, and these bronzes did not in general suffer from crevice corrosion.

His main point was Mr. Bowers' slight doubt in relation to overall costing. There were ships with two medium speed engines per shaft which were employed solely on long sea routes between major ports with good availability of tugs and no need for tricky manoeuvring. The case for fitting CPP's to such ships rested mainly on the ability of a CPP to allow the full power of one engine to be developed when it was either not necessary, or not possible, to run both engines. To meet this requirement a CPP only need be able

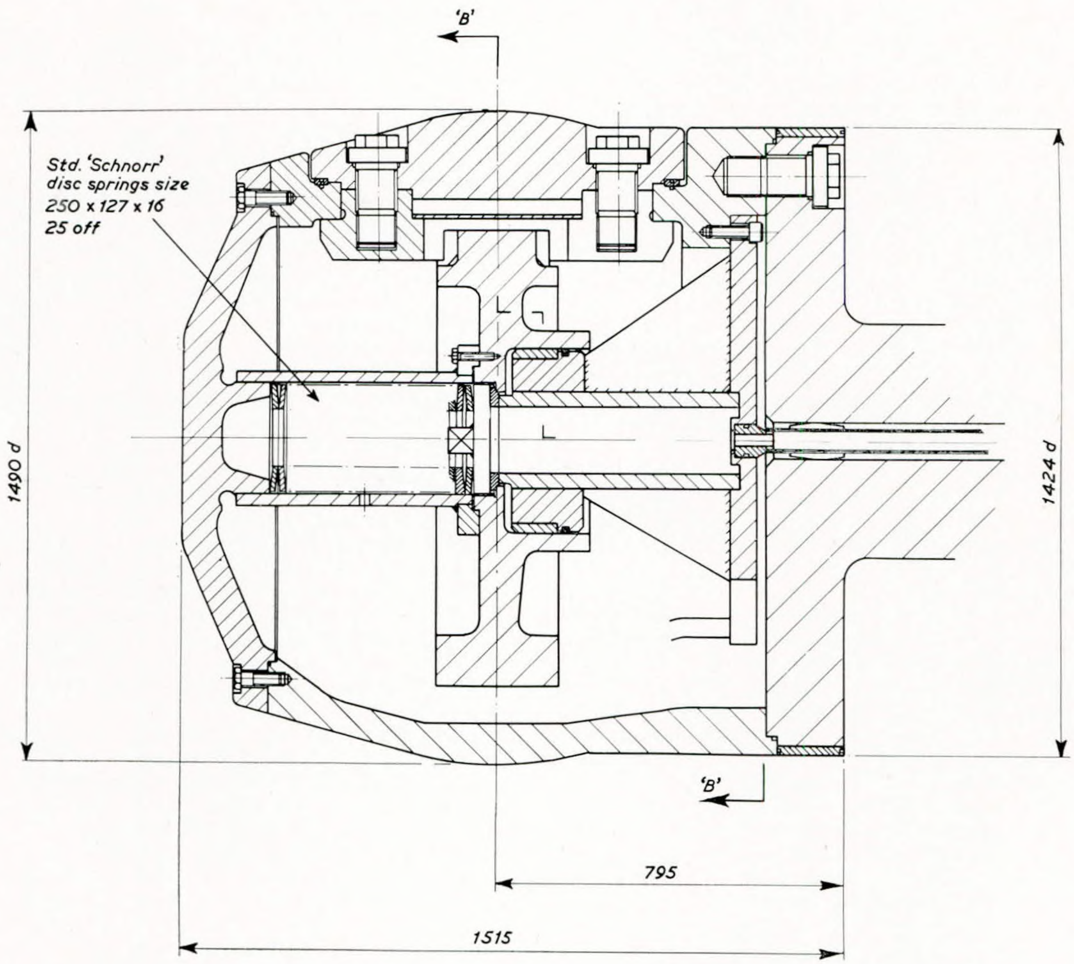

FIG. 20

to adopt one of two pitches—full ahead, and about 80 per cent ahead.

Fig. 20 showed a much simplified CPP which was automated to stay in low pitch at all times except when both main engines were running at over 75 per cent full rev/mm. This propeller improved the ship's acceleration, compared to that available from a FPP; and enabled full use to be made of available engine power, but was substantially less expensive than a normal CPP.

MR. O. SYASSEN commended the author for the unique candour of his paper. To his knowledge, existing difficulties and problems had never been presented and put forward so unreservedly: he believed this paper was extremely interesting, instructive and valuable for the medium speed diesel engine makers and users.

He hoped, therefore, that this contribution to the discussion would not be taken as cheap criticism of past mistakes committed by others, but as a positive effort to clarify and assess the causes of damage, thereby contributing to an elimination of such fatal mistakes which greatly injured the cause of the diesel engine, and were a disadvantage to all those involved.

Medium speed engine plant—the owners' decision to install a medium speed diesel engine plant embodying two medium speed engines onto a CPP and saving two frames in length and 300 tonnes in weight, was certainly well founded. Today, after bunker oil prices had risen three or four-fold, there could hardly be any doubt that this decision was right, at least in so far as steam versus diesel was concerned.

In this context, Mr. Syassen said he would like to mention an example which his Company had recently examined. The plant involved, i.e. the container vessel TS *Remuera* with 2 × 24 000 h.p. (17 500 kW) was, however, considerably more powerful. Instead of a steam plant they had entered on the drawing two large 4 stroke engines of the 14V 65/65 type (650 mm bore and 650 mm stroke) from the joint M.A.N.-Sulzer development programmes. (Figs 21 and 22.)

What was remarkable was the effect on the assessment due to the oil crisis and how much better the medium speed plant was now. The decision of the Owners to opt for a medium speed engine layout was therefore probably right in principle.

On the Ruston A.O.: Mr. Syassen said that in his opinion, the design of this engine type had embodied some mistakes:

a) The engine was designed with a view to permissible stress and not to permissible deformations, additionally, since the space-frame was a steel casting, and as the loadability of steel when compared with cast iron rose more than the E-modulus which determined deformation, excessively large deformations and relative movements resulted when adjoining components were subjected to varying loads.
b) 2 stroke cycle and trunk piston were, in principle, always a complicated combination. Owing to the oil escaping via the ports, the piston could only be "starvation-lubricated" and with high specific loads this was not enough.

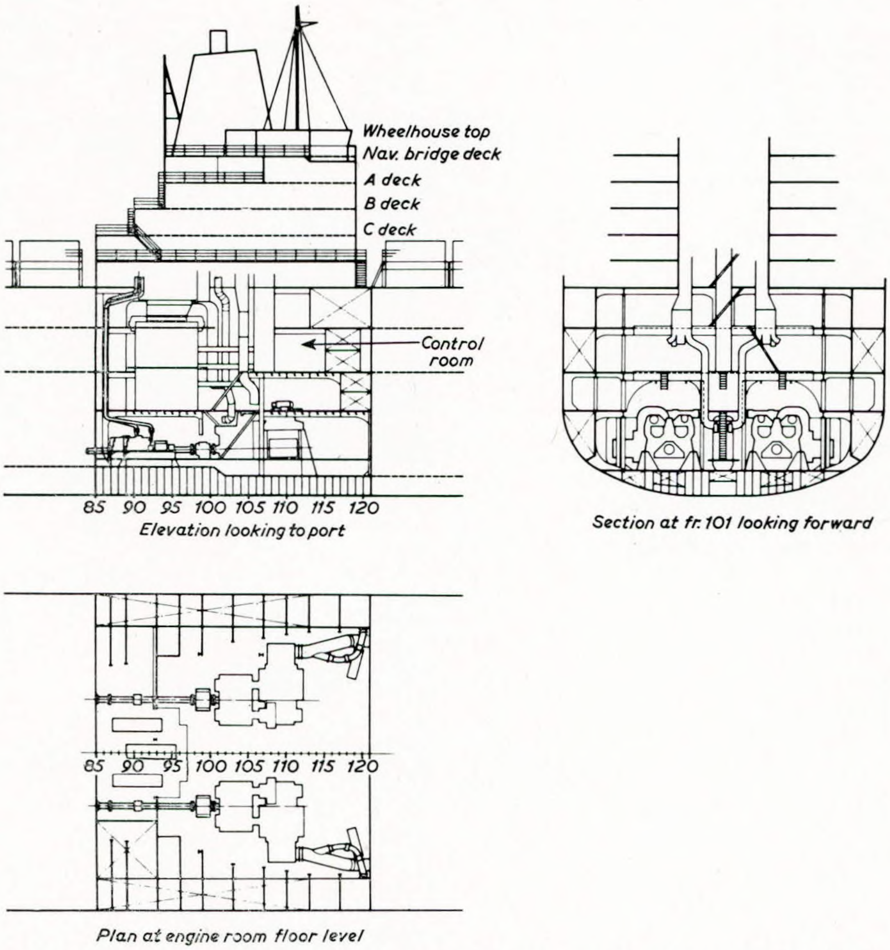

FIG. 21—*Engine room layout for a container vessel with a steam plant* $N_e = 2 \times 24\,000$ *shp*

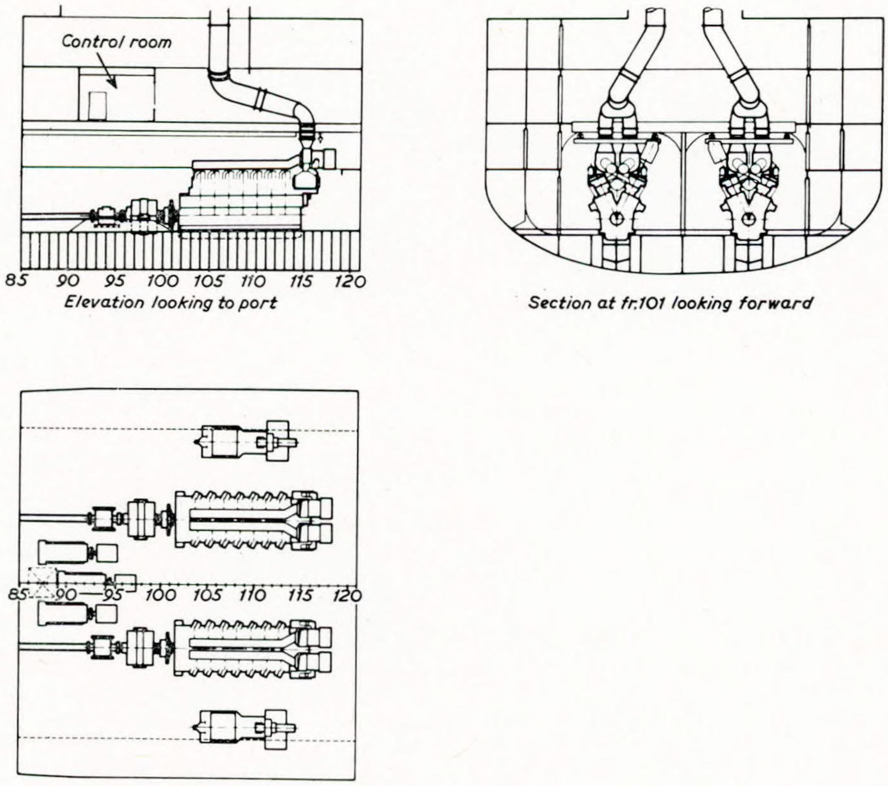

FIG. 22—*Engine room layout for a container ship (same engine space as in* FIG. 21) *with two medium-speed 4 stroke trunk-piston engines of type* 14 *V*65/65 $N_e = 2 \times 23\,000$ *shp*

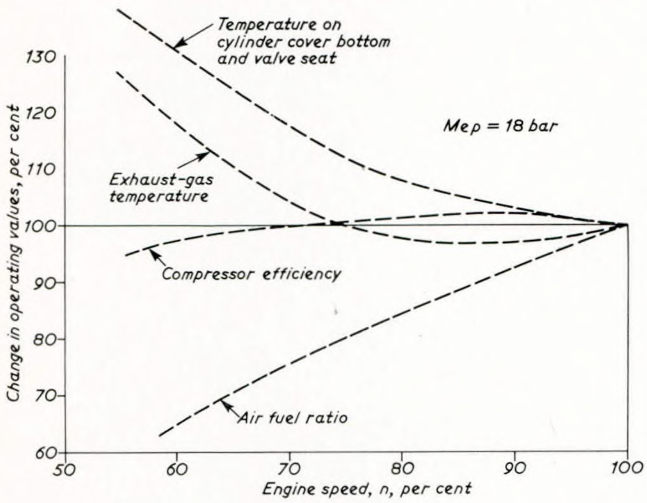

FIG. 23—*Rise of outlet-valve and cylinder-cover temperature of a 4 stroke engine, speed being reduced at constant b.m.e.p.*

c) If 2 stroke trunk pistons were used at all, then the unfavourable lubricating conditions as mentioned under b) must be taken into account by using special pistons, which were particularly resistant to seizures: e.g., rotating pistons (GM) or step-by-step rotating pistons (Sulzer).

On lubricating oil consumption: To be able to assess this problem correctly, further data was required. Judging by the overall situation, Mr. Syassen assumed that liner and water jacket had too little rigidity, so that the author was right in his assumption. M.A.N., for instance, had the experience that with long-time operation the ovalities due to wear which occurred on the liners, increased the specific oil consumption rate considerably; in this context, Mr. Syassen said he would like to ask the author to state the wall thickness of the liners.

On liner and piston ring wear: Mr. Syassen said he was convinced that the author was right in his assurance that here again deformations were the chief culprit. The fact that a clearly defined circular groove of up to 1 mm in depth worked itself into the liner was an astonishing phenomenon. How sharp was the boundary of the upper and the lower wear edges? Mr. Bowers had remarked "A few liners showed two grooves approximately in line with the top and second compression rings at T.D.C. and the depth of such grooves could reach 1 mm in less than 500 hours". Did that mean that the upper edge of the first and the lower edge of the second compression ring coincided exactly with the upper and lower boundary of the groove worn in the liner?

If the liner was comparatively thin-walled—as it appeared to be from Fig. 3—apart from deformations, vibrations must also be considered as a contributing cause to such extreme wear rates. It was known from vibration measurements carried out on liners for the purposes of the preventive maintenance programme that thin liners might be subjected to considerable longitudinal and transverse vibrations.

Apart from this, the piston of this engine seemed to be comparatively short for a 2 stroke, so that running and guiding characteristics were not so good as they could be.

Moreover, it seemed that the piston did not have a dished combustion space, but tapered towards the outside. This might cause a disproportionately large quantity of fuel to be sprayed against the outer wall. Due to this, the lubricating oil film might be washed off under certain conditions so that running and wearing behaviour—especially of the first piston rings—would deteriorate significantly.

As to cylinder lubrication, the time-injection examined was no doubt the right technique to ensure oil supply to the decisive point at the proper moment: it was, however, insufficient in the case of major deformations. Mr. Syassen would like to know the number of oil admission holes around the circumference and their distribution and position in relation to the first and second compression ring and TDC.

He could not agree with the statement claiming that cylinder lubrication had positive effects. The favourable characteristics mentioned there might almost have come from a ten year old M.A.N. pamphlet. In those days they had advanced the same arguments for cylinder lubrication. In the meantime, however, they had learnt that the admission of fresh oil through lubricating oil feeds had only a very limited effect in terms of area and by no means reduced the wear rate to a marked extent.

The oil sprayed from underneath against the cylinder liner walls was about 100 times the amount of that admitted by cylinder lubrication. This was on the assumption that the cylinder lubricator had been set at a lubricating rate which corresponded approximately to the consumption. It was even questionable whether the negative influence of the calcium additives on the valves made it really desirable to feed a maximum quantity of fresh oil of high TBN to the upper liner zones for combustion. It seemed that this question was implied in the author's suggestion to use a low TBN oil charge in the engine sump.

Mr. Syassen was somewhat surprised to learn that this engine faced such serious inlet valve problems that it should become necessary to feed 20 per cent of the lubricating oil consumption into the inlet air in order to lubricate the valve seats. They were, as a matter of fact, also confronted with these problems in earlier years, as could be seen from Professor Zinner's article,[2]. In his Company's opinion a number of techniques had meanwhile been introduced which made it possible to control such wear on the inlet valve seats without any particular difficulty.

Mr. Syassen considered it an impossible situation to use three different types of lubricating oil, i.e. as an additive for the combustion air, for cylinder lubrication and for the engine sump. He was quite convinced that it was possible to use a single lubricating oil type without any disadvantage whatsoever.

He concluded his comments on the aforementioned difficulties and problems by a glimpse of the future: the problems described had, today, to a great extent been mastered by the large diesel engine manufacturers with certainty. It was now first and foremost a matter of improving maintainability, general reliability and, by reducing fuel and lubricating oil consumption, thus further enhancing the existing superiority of the diesel engine over other forms of propulsion.

MR. J. A. ROOS said that the audience had been shown three slides of the Smit-Bolnes engine frame. This all welded frame had been developed almost 25 years ago and all their engines were successfully operating with such frames. With the first four engines nothing had been done on stress annealing of the engine frame: all subsequent engines had been annealed, from which it was ascertained that a frame of this type could not be surpassed.

There were engines which had an exhaust valve fitted into a special housing which could be extracted from the cylinder-cover. However, it was much easier to remove the cover of a Smit-Bolnes engine completely, than to remove a valve housing such as other engines had.

He would remind the audience that there were engines in the world which had a very simplified cylinder cover with only one exhaust valve, and the weight of that cover was only 300 kg. Simplicity was needed to-day, certainly in the top of the engine.

With chromium plated liners they had been much more successful and these were a necessity today. He thought it fantastic that Mr. Bowers and his Company had continued with medium-speed engines after so much trouble.

It was a pity they had not kept to 2 stroke engines although there was a 2 stroke crosshead medium-speed engine available. It was a question of size and revolutions; and they also wanted to keep to the gearbox. All right, but there was a medium speed 2 stroke crosshead engine, and in this case the crosshead was circular, worked as a piston works, and was used as a scavenging air pump.

Mr. Bowers' paper had proved that there were still companies in the world who were willing to try a new type of engine, and willing to help to develop something new, although they ran the risk of losing money. In connexion with the torque of a 2 stroke engine when using a controllable pitch propeller, it was very important to know that the 2 stroke engine was stronger.

With a 2 stroke engine at 75 per cent revolutions one would still find 106 per cent torque, compared with a 4 stroke engine at 95 per cent revolutions giving only 100 per cent torque, and one could imagine what happened when it dropped to 75 per cent revolutions.

Mr. Bowers had said something in the paper about the crew on board. As an engine manufacturer, his Company said the engine was as good as the engineers on board.

MR. J. W. ALLEN, F.I.MAR.E., said that the first part of the author's paper might be described as the life and death of the A.O. engine. It needed saying from this platform that its demise was a blow to the pre-eminence of British engineering in general and the British marine industry in particular. The author through the medium of his paper had given them a very factual insight into the build-up of circumstances and factors which had led to that final decision to re-engine 11 ships with 22 new engines. One noted that the decision had been taken "with extreme reluctance" and on reading the paper Mr. Allen had been left with the thought that sleep might have eluded Mr. Bowers on many occasions around that time.

To come to the question of the exhaust valve problems: this was the component which always appeared to be the "Achilles heel" of the medium speed engine and more often than not provided an "interface" of owners, builders and oil suppliers. Looking at the fuel aspect first, Mr. Allen would certainly have an area of agreement with the author when he said that fuel viscosity was not likely to be a fundamental cause, though Mr. Allen would add that fuel viscosity at the injector tip could be; but experience suggested that the medium speed engine was more susceptible to fuel quality than its slow speed counterpart. Controlled and consistent fuel quality were beyond the bounds of economic viability, even before the disruption of oil supplies in December 1973, and when the author decided to change all the exhaust valves of the *Cape Grenville* because of a shipment of high vanadium fuel, this was a situation that could arise at any time.

Whilst vanadium contents were very often below the 120 ppm which many engine manufacturers would wish to see as a maximum norm, contents of 400 ppm were not unknown, and this could not be controlled either at the refinery source or at shipboard level. The vanadium itself was no problem, providing the exhaust valve seat temperatures were below about 500°C.

Werkspoor were to be congratulated that by diligent design their exhaust valve seat temperatures were contained at substantially lower figures than this. However, the inclusion of sodium in the fuel could give rise to complex sodium/vanadium compounds which lowered the melting point of vanadium. The presence of sodium was more usually manifest by a falling turbocharger efficiency caused by fouling of the gas blades and nozzles. One of the most prolific sources of sodium was salt water contamination, which could arise from shipboard handling, and from levels of about 100 ppm onwards deposition could be expected.

The importance of delivering the fuel to the engine in as clean a condition as possible was the only way of minimizing this problem; therefore could the author say if at the time of the exhaust failures, and in view of the subsequent fact that the exhaust valves were now running 6000 hours, the sodium levels were looked at, if there were any problems occurring with the shipboard handling of fuel, and if any turbocharger blocking had occurred. Water in the fuel could also give rise to a trumpeting at the fuel valve tips. What was the condition of the injectors at that time?

On the part possibly played by the lubricating oil in the failures, with controlled injection of the cylinder oil the incidents would appear to coincide with high crankcase oil consumption associated with the apparent failure to run-in. Therefore could the author say what kind of running-in procedure had been adopted? Had a low T.B.N. crankcase oil been in use it was possible the problem would have met with some alleviation. Accepting that the lubrication of the pistons was a two-way process, the use of a lower T.B.N. oil in the bearing system was a worthwhile possibility, as suggested by the author. There were medium speed engines running today without separate cylinder lubrication and the T.B.N. of the crankcase oil would stabilize at 25 per cent below that of new oil. There was a 4 stroke medium speed engine in use today which had suffered from valve problems, and one of the methods of attacking this problem had been to reduce the ash throughput through the valve system.

Looking at the consumption curves the abcissa, it was noted, was in weeks, and these were assumed to be calendar weeks and therefore not a measure of actual running time. If it was assumed that running in was complete when minimum consumption was reached, then with the exception of *Baron Renfrew* and *Cape Grenville*, running in appeared to be taking 6 to 12 weeks. The curves themselves also suggested some piston disturbance, and it was possible that the inflexion in the curves coincided with piston removals: no mention had been made of the frequency with which pistons had been pulled or were expected to be pulled on the basis of the accumulated running hours to date: however, Mr. Allen believed these should be left undisturbed for as long as possible, and 20 000 hours should be reached without problems. Could the author say if the inflexions coincided with piston removal, and describe his procedure in running-in pistons when pulled?

Mr. Allen had one final remark on the deposit build up on the air inlet valves spindles: if this was a persistent problem, the use of an oil with some alkalinity was suggested, but incorporating ashless additives, and therefore of very low ash content. However, this required alteration of pipework, and the provision of a separate tank, so it had not been taken up. The oil for this lubrication of inlet valves was taken from the sump, therefore this again gave rise to the thought that crankcase oil of lower T.B.N. might be of some value.

MR. A. J. WICKENS said that with such a wealth of information it seemed ungracious to ask for more.

In his paper Mr. Bowers had described the original installations as being designed for unmanned operation. Presumably this had been abandoned during the A.O. era; was it reinstated with the introduction of the TM 410?

Several speakers had commented on the unusual stepped groove formed in the liner. Coming from the opposed piston stable, he would suggest that the upper pistons in such engines would, at firing, be subject to similar or even greater stretch phenomena. He had never observed any effect of this nature, so while the explanation was ingenious it was not entirely satisfying.

Could the author say if both the 9 and 12 cylinder engines were run at similar outputs per cylinder when the re-engining took place, and had there been any noticeable difference in performance?

The saga of the exhaust valves of both engine designs was enlightening but, for those in the opposed piston business, interest in such failures was purely macabre. However, whilst subscribing to the theory that the deposit on the seats was at the root of the problem he would like to know if there had been other instances of exhaust valve failure? Had there possibly been any failure due to stress corrosion which usually manifested itself as a circumferential crack in the valve head?

The lubricating oil consumption curve was interesting. He would have expected a classic "bath tub" consumption/time profile for the trunk piston engine with the stable low level phase ultimately followed by a rise. Had there been any evidence of such a terminal increase in the consumption of any of the earlier re-enginings since the compilation of the paper?

Was there any means of separately metering the flow of oil to the cylinder injectors and loss from the crankcase?

Mr. Roos had made a point in favour of the 2 stroke engine which he would like to develop. The 2 stroke engine was a horse power machine and could tolerate to a greater degree than the 4 stroke engine the adverse torque situation obtaining when operating with one engine out of action. Mr. Wickens considered that a twin 2 stroke installation reduced to one engine would enable a vessel to maintain 80 per cent normal rev/min, whereas in the same circumstances a 4 stroke engine, where one was limited to a linear rev/min/mip relationship, the speed would drop to little more than 70 per cent rev/min.

Mr. Wickens said that he hoped the point made by the author about the reliability of twin engines had not been lost on the insurance companies. He would take issue with the author's estimate of the chances of engine stoppages causing a complete halt. He would have thought with a single engine the probability of a stoppage causing a complete halt was unity. He then added that a statistically minded colleague was prepared to offer better odds on two engines and considered that the chances of a halt under the circumstances described were once in 146 years.

Finally Mr. Wickens said, whilst gearboxes had achieved a very high standard of reliability the author made an oblique reference to the possibility of a failure. It would be interesting to learn if he had had any trouble from this source in his fleet. It was not often that the industry was favoured with information about important equipment drawn from such a large population of similar items.

Captain M. W. Anketell-Jones, R.C.N., F.I.Mar.E., said that it seemed that every possible question had been asked. He pointed out that the question of procurement economics for the TM 410 had not been discussed in the paper. He realized that it might not be an area into which prying would be welcome but wondered if the author's company had considered machinery alternatives other than an extensive modification on the A.O. engine. Could the author comment on this point please?

With regard to the fuel situation, Captain Anketell-Jones noted that heavy fuels were in use in the TM 410 engines. He commented on the rather broad translation which was possible when using terms like heavy fuel and high viscosity fuel. The importance was in the engines' tolerance to fuel impurities. Could the author define what levels of impurities he considered to be reasonable for continuous use with the TM 410?

With regard to the possible effects of higher levels of impurities, had Mr. Bowers had any experience indicating difficulties in this area other than those already mentioned? And what would be the expected effect on the overall engine economics of using higher impurity type fuel?

On the question of engine availability—from Table I it could be seen that approximately 60 per cent of service time was at sea underway, viz. out of a total of 1843 ship days only 1083 were spent underway. It was surprising therefore that notwithstanding the 760 spare days, a further 25 days had been spent steaming on one engine. Could the author indicate whether there was a prohibiting factor which precluded engine overhaul in port, thus leading to the need for further work at sea, and whether a shorter in-port time would lead to even higher ineffective time when ships were underway.

On the question of specific lubricating oil consumption, the makers claimed a specific consumption of 0·8 g/bhph., which was a good figure. Based on the information in the paper, it would appear the figure achieved was 1·08 g/bhph. Did the author consider that satisfactory, and did the manufacturer think that an adjustment to the figure he was claiming was justifiable?

Mr. J. Cliffe asked if the author would enlarge on the A.O. engine exhaust valve failures? Were seat distortion effects the mode of failure, or high temperature, high vanadium, or some other cause?

During the design stage the makers had deliberately rejected a caged valve seat construction, believing that some of the distortion problems experienced at times with caged seats could be avoided by a very stiff flame plate construction designed for highly effective cooling.

In retrospect could the author comment on this decision?

Did the achieved engine exhaust temperatures prove to be higher than the design temperatures?

The A.O. engine appeared at the time of its inception to be a very promising development, but perhaps with hindsight the makers had tried to incorporate too many new features all at once. Possibly the space frame concept would have been more successful in conjunction with a 4 stroke cycle engine, whilst still enabling important savings in weight to be made.

The contrast with the TM 410 engine weight was very apparent. Nevertheless, equally successful but lighter 4 stroke engines than the TM 410 were available. How important was weight saving in a bulkcarrier?

Regarding the TM exhaust valve troubles, and the steady cruising salt build up theory, was there any evidence of salt build up in other parts of the engine? In such a case one would expect salt deposits on the compressor, and fouling of the turbocharger.

This was, said Mr. Cliffe, a problem with compressors of marine gas turbines, but so far as he was aware it had not previously been reported for diesel engines, the diesel being much less sensitive in this respect, exhaust valve temperatures being well below the salt corrosion point.

Despite recent engine development, it seemed that the Achilles tendon of diesel engines was still the exhaust valves.

Mr. R. H. Morton, F.I.Mar.E., said the author was to be commended for a very forthright paper. It was not always that such frankness was, or in the past had been, encountered. Not really surprising was the fact that the builders had so conjoined because, for years past and many must have found so, their openness in regard to a trouble, rare in their case, had been readily conceded.

The trend towards the smaller sized medium speed diesels, batteried in multiple units for relatively high powers, might be induced not merely by the consideration of space and weight saving, possibly of greater significance for a container ship than a bulk (and commensurate deadweight) carrier, but also by the supply of the heavy slow speed unit being insufficient to meet an increased demand for the higher powers, coupled with the cost of them, both initially and for replacements which, must be required in the passage of time. Those costs had increased to alarming proportions.

Every design from the earliest, even those which were short lived, had provided a lesson or lessons to be learned, but one could question whether those learnings had spread widely enough. Mr. Morton firmly believed that records of problems and what was done to cure them, not merely an alteration of design with no indication of the reason for it, could help to prevent what might be termed repetitions, and they did occur involving both work and effort. Thus, for example, excessive lightness of steel structure with its inherent flexibility might be avoided. Exhaust valve trouble, meaning the work of it, might be minimized by design, especially if whatever amount of it there might be is to be multiplied ninety six times: whereas the idea of even one valve being seated direct in the cylinder head of a propelling motor especially would seem to have been unheard of since earliest times, while a caged valve was so obviously called for. But, in the haste of competition it would seem that such a backward step was deemed acceptable. The time needed to change a valve on this motor, small as the valve was, must have exceeded that required for the most awkward that he could recall: the bottom valve of a B & W double acting four stroke which would take about two hours. As for the simple single acting, including the Vickers so long ago on which a change at sea was infrequent and for some reason other than a burned valve, a rarity because valve as well as cage was water cooled, the stoppage would be about twenty minutes at most. As to work there were some operations which still existed despite all improvements but because of increasing powers and loadings had remained on the work list, though for different reasons. For example, in the 1920's it could have been reasonably supposed that the need to withdraw pistons out of season for repair or change, that was to say not merely for cleaning or to see the liner but to up-end them with rods on the platform, had been satisfied and was at an end; however for different reasons and with the increase of powers that had not been so and it could be believed that the latest decade had seen more of it than the 1920's.

It was interesting to see that in changing, the Owners had headed in an opposite direction, notably choosing a rigid and relatively heavy design, from builders who were among the earliest producers of propelling motors.

The future would seem to hold extending prospects for the multi-unit power pack, but it might be found necessary to give way somewhat, not being too ambitious, in the matter of space saving which the system promised, for nothing was worse for maintenance than being "unable to get at the job properly".

COMMANDER E. TYRRELL, R.N., F.I.MAR.E., said that when an approach was made by the engine manufacturer in 1966 the Government had decided to support the development of the A.O. engine. The proposals were made by a competent and well known engine builder and represented an attempt to leapfrog overseas competition and to produce an engine of low specific weight and small size at a low cost. The design allowed for a moderate increase in piston speed and in brake mean effective pressure over those obtaining in engines operating satisfactorily at that time. It was considered that these moderate increases were well within the scope of short term technical development.

At that time it was not fully appreciated that satisfactory performance of a component in a few tests would be insufficient to guarantee acceptable operation in the long term and 10 or perhaps 20 tests under different operating conditions were required before the designer could be fully satisfied with the engine's operation.

He said he had been interested to hear the proposed solutions put forward from the floor to solve many of the problems encountered. It always distressed him to hear a manufacturer's representative plugging his company's products to a learned society. Although his own countrymen were not alone in this respect he would say to Mr. Fenton that he wished that Mr. Fenton had come forward with his solutions when the engine was on the test bed and was in trouble.

The engine was put into service at sea when it seemed that its many problems were resolved. In fact its service performance was bad and the engine soon, and deservedly, acquired a bad reputation for reliability. He had no doubt that given time and money the problems could have been solved, but the engine's already bad reputation precluded sales in any quantity and escalating development costs coupled perhaps with expense of the components necessary to overcome the troubles would have priced the engine out of the market. It had therefore been decided that there was no point in proceeding with the development.

Correspondence

MR. E. T. KENNAUGH, wrote that during the discussion on this frank and excellent paper one of the speakers had stated that the A.O. engine would have benefited from a more extended period of development, and with this he would agree.

The modern Sulzer medium speed engines had been designed on the basis of more than 20 years of development experience, starting in 1955, during which time over 12 000 hours of testing had been accomplished, both for the 2 stroke and 4 stroke versions.

It could be said that the thermal load of a 4 stroke engine was less than that of its 2 stroke counterpart with the same specific output. With the "Z" engine, all the highly thermally loaded components of the combustion space, i.e. cylinder cover, cylinder liner, piston and exhaust valves, had been developed for the more exacting 2 stroke version and tested under 2 stroke conditions. All these parts had been retained and were practically identical for the 4 stroke version. This additional safety factor had had a positive effect on the operating behaviour.

There was a good deal of discussion about the exhaust valves, but comparatively little mention as to how the valve seats were cooled. Sulzer had adopted press-fitted valve seats with very effective cooling and the maximum temperature on the exhaust valve during the 4 stroke operation was only 555°C, with the face itself reaching only 430°C. In addition, specially patented coatings had been developed for high performance trials for 2 stroke operation, and very successful results had recently been obtained during a 600 hours endurance test at an output of 800 bhp (595 kW) per cylinder, using heavy oil of 1620 seconds Redwood 1, with a sulphur content of 2·5 per cent vanadium 190 ppm and sodium 58 ppm. The results from a large number of these valves already in service were very promising, and some had achieved approximately 15 000 hours in service.

The design of the cylinder liner and the cooling methods employed also played a very important part, and the principle of bore cooling enabled the cooling water to be brought close to the combustion space surface, and kept the maximum temperature at the running surface at a low level.

The piston head was a two-part design with a rotating piston which also contributed to the successful operation of the engine. The crown itself only reached a maximum temperature of 310°C and the turning mechanism practically eliminated the danger of piston, or piston ring, seizure, which was one of the most exacting problems with a modern highly loaded engine.

Thanks to the rotating piston, the temperature distribution was symmetric and local over-heating was prevented, leading to less, and more even, wear. The turning mechanism itself had proved to be very reliable in service.

Long term service results with the Z40/48 mostly concerned the 2 stroke version. However, since the same engine components were concerned under a higher thermal load as in the 4 stroke version, the same, or even better results might be expected with the 4 stroke version.

To date, there were approximately 48 2 stroke and 23 4 stroke in service. The first ship with a 2 stroke engine went into service in May 1967 (M.V. *Finlandia*) and to date these engines had each accumulated 38 000 running hours, and had been running on fuel of up to 400 seconds Redwood 1 since 1968.

The cylinder liner wear on average had been between 0·02 and 0·03 mm/1000h, giving an expected service life of 60 000 hours. Piston overhauls now took place at approximately 10 000 hours, which was considered to be satisfactory for a medium speed 2 stroke engine.

The cylinder lubricating oil consumption was approximately 1·0 g/bhph at full load, and the system oil consumption was recorded at 0·4 to 0·5 g/bhph, including all losses, giving a total lubricating oil consumption of 1·4 to 1·5 g/bhph over long periods.

MR. P. MANSON wrote that Mr. Bowers had given a very good report on the problems experienced with the A.O. Ruston and Hornsby medium speed engines. It was rather puzzling to a mere observer that, in spite of all the research and testing done on the prototype engine, not even a slight indication of the troubles of the kind reported had been experienced during the prolonged shop test. It was agreed that conditions under a marine environment were quite different to those obtained in an engine shop, but nevertheless one would have thought some indication of the problems reported would have shown up from time to time.

From all accounts, all of the A.O. engines had suffered similar troubles, and he was not sure that all had now been replaced. Mr. Bowers stated that the previous experience with the large bore slow speed 2 stroke engines, had influenced him in the selection of the medium speed 2 stroke Ruston A.O. engine. Some engine designers nowadays referred to the power rate factor, which was derived from the normal formulae used for expressing the b.h.p. of a diesel engine, and by re-arranging the terms of constants, used the values PME (brake mean effective pressure kg/cm^2) CM (mean piston speed m/s) in relation to the engine output per unit area of piston, and obtain the formula:

$$PME \times CM \propto \frac{\text{BHP}}{D^2} \text{ (Piston Dia.)}$$

These values were also used as representing the thermal load of the combusion chamber. In other words the higher the mep and the mean piston speed were, the higher the thermal load of the combustion chamber would become, and this was where the importance of specific air throughput required special consideration.

The writer had done an exercise to calculate the power rate of the different designs of large slow speed diesels on offer at this time, and found, that the power rate ($PME \times CM$) worked out over a range of between 70 to 88 with satisfactory air throughput at their existing ratings, and in the case of the 4 stroke medium speed diesels a similar power rating was obtained but starting from a lower figure of 49 up to 80, it could also be said that satisfactory air throughput was achieved with this cycle, based on figures published in the technical journals. On the other hand the power rating of the medium speed two stroke engine works out between 70 to 80. It was, however, found that a considerable difference existed between the airflow kg/bhph in three cases of the medium speed two stroke engines investigated, and in his opinion could well explain the difference between success or failure, in so far as thermal conditions were concerned, with its associated problems. The maximum air flow in one case was found to be 8·0 kg/bhph and that, of one of the other two engines 6·35 kg/bhph. The power rating in each case being 70·56 and 74·6 respectively, which would indicate that the engine with the higher power rating had in fact a much lower specific air throughput, and therefore could be subject to problems associated with high thermal loading.

When considering the two stroke cycle as against the 4 stroke, in the latter case we obtained a far more efficient scavenge cycle, as against the 2 stroke. Would the author agree, based on his experience with the A.O. Engines, that for such a highly rated engine the air flow was perhaps on the low side, especially when one had to consider the drop in efficiency of the turbo-charger after a certain period of time in service. In fact one wondered what proportion of the 300 gallons/day (1364 litres) of lubricating oil was in fact burnt up. The piston ring and liner and exhaust valve problems were all indicative of high temperatures, which could lead to carbonization of the lubricating oil. Perhaps this was the basic difference between his experience with the large bore 2 stroke slow speed diesels and the medium speed 2 stroke engine, namely efficiency of the scavenge system.

As a matter of interest in working out the power rate of the 4 stroke Werkspoor T.M. 410 engine, this worked out at between 68·57 and 76·39 at maximum continuous rating with an air charging pressure of 2·15 bar and air flow 6·3 kg/bhph, which might be considered as one reason for the success of this 4 stroke engine with good scavenging. It was however, doubtful whether the air flow would be satisfactory for a 2 stroke engine with a similar maximum rating of 76·39.

It would be appreciated if the author could indicate the scavenge air pressure obtained under service conditions and also the designed air flow for the Ruston engine.

Commander M. F. Griffey, R.N., F.I.Mar.E., wrote that the saga of the A.O. engine was a sad one; it had had tremendous potential, but in the event was apparently found wanting. During their detailed and comprehensive investigations, however, Rustons must have pioneered an immense amount of research into fundamental lubrication, materials, liner and piston matters. He would like to put in a plea for a paper on their recent work to be published.

Turning to the paper and to the references to alignment. Why were resin chocks chosen? Could Mr. Bowers elaborate on the requirement to apply local heating and the temporary distortion which took four days to disappear? What had been the in-service experience with these chocks to date?

As a result of the re-engineering experience had any thought been given to the cost effectiveness of carrying out major overhauls of the TM 410's by engine replacement?

R. M. Dunshea, F.I.Mar.E., wrote that the author's description of troubles with the A.O. engine was a sad account of the British attempt to enter the advanced medium speed diesel engine field, now dominated by continental designs. The fortitude displayed by engine room staffs was the only bright aspect of this engine's short and unhappy history.

As one who had investigated the A.O. for installation in ferries it was difficult to understand how the engine progressed beyond the test bed.

In the writer's opinion in any two cycle engine, but more particularly in a medium speed engine, the pressure differential between the cylinder and scavenge trunk on opening to scavenge should be always such that air immediately flowed into the cylinder. In the A.O. the reverse situation was apparent. Thus scavenge parts "coked up" and, in a scavenge trunk awash with lubricating oil, fires were a constant risk. Scavenge efficiency would rapidly deteriorate giving rise to high exhaust temperatures and unsatisfactory combustion.

The flame plate arrangement did not look satisfactory and the valves in head promised difficult maintenance.

On the test bed lubricating oil was literally being pumped through the engine, a good deal being possibly swept up from the scavenge trunk. From the test bed performance the writer had decided upon the Pielstick PC2 engine, which for seven years had given satisfactory results on residual fuel of about 1000 seconds Redwood viscosity.

At this point one wondered why with all its very apparent shortcomings the decision to change the A.O. to a 4 stroke cycle was not made. Was a specific weight per bhp the fetish? Had such a change been made Britain might have been more fully represented in the advanced medium speed diesel engine field.

Injection of lubricating oil into induction air was a practice which was not known to the writer. Very little oil would actually be deposited on the valve faces. Could the author give the considerations which prompted this procedure.

The author's experience with TM410 engines was similar to that of the writer in the case of PC2 engines and demonstrated that well designed and developed medium speed diesel engines operating on residual oil with moderate vanadium and sodium content, at not above 85 per cent M.C.R. would perform satisfactorily. They must, however, be given regular intelligent maintenance.

Here a criticism of medium speed engines must be made. Overhauling procedures were still too time consuming and did not match the optimistic figures given by engine builders. This was at last being realized and better equipment was being made available to shorten operations.

The writer had not experienced as many exhaust valve failures as the author had reported. Could Mr. Bowers give particulars of the vanadium and sodium content of the fuels used and the performance of the valve rotating mechanism.

Mr. J. Morrison and Mr. S. R. Frederick wrote a joint contribution stating that it was a tremendous pity that the author's Company had had to go to the trouble and expense of changing engines in so many relatively new ships, but if the decision to do so was almost inevitable, it was nevertheless courageous and seemed to have been carried out with considerable efficiency. The paper had described the whole exercise very adequately and gave many pointers to the reasons for and possible solutions to a number of problems which would surely be of assistance to many operators of medium speed diesels. There were one or two points, however, in the paper on which the writers would like to comment.

The author had mentioned the possibility of redesigning the A.O. engine as a way of solving the problems. It was difficult to conceive of modifications to the extent he had postulated (virtually a new engine) being made and proven in less than seven to eight years, when the ships would be that much older, when they were in fact re-engined. In the meantime the troubles and consequential loss of service of the ships would continue. It would seem that the decision to go to new and proven engines as soon as possible was correct.

With regard to flameplate failures, the author had reproduced a photograph showing corrosion/erosion in the cooling water passages surrounding the valve seats. Approximately 18 months ago, he had asked BSRA to undertake a metallurgical examination of a section cut from a welded flameplate and the location of the corrosion was as shown in Fig. 24 and details of the welded joints as in Fig. 25, from which it could be seen that weld B contained a root gap. If welding did not ensure full penetration the joint was liable to be associated with a crevice.

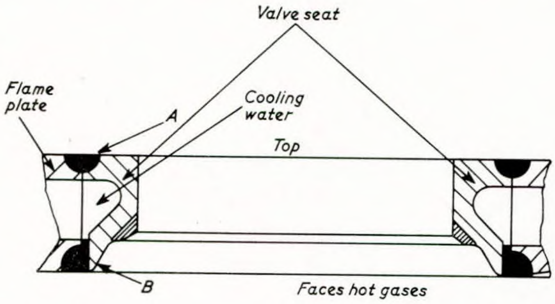

FIG. 24—*Illustration of welded joint between exhaust valve seats and flame plate*

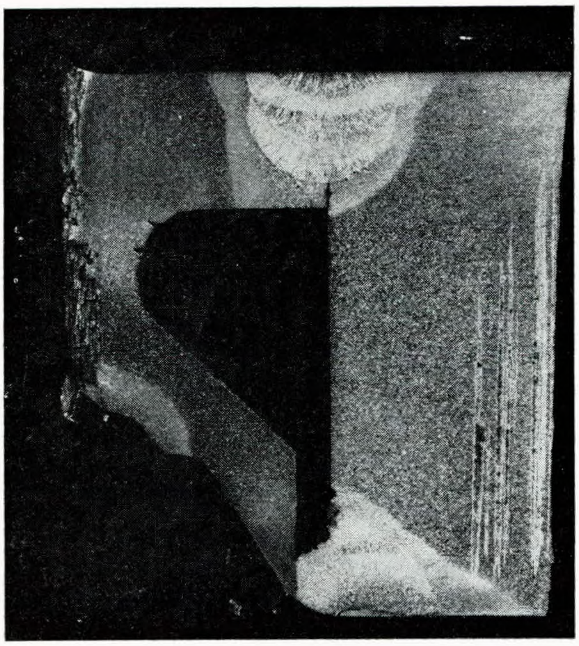

FIG. 25—*Macrosection showing two types of weld securing flame plate to exhaust valve seat*

It was also reported that a nitrite/borate anodic inhibitor was used in the cooling water, and such an inhibitor could be considered "dangerous" if the concentration became too low, as would occur at the root of a crevice. In this instance, leakage of cooling water was considered to be due to the initiation of a differential aeration cell at the root of the crevice, and to failure of the nitrite/borate inhibitor to prevent localized corrosion.

Jumping a little, but taking things more or less as they came in the paper, the writers would refer to the cracks in the propeller blades and hubs. BSRA was recently asked by a shipowner to examine the stainless steel blades of a variable pitch propeller following the discovery of cracks between the retaining bolt holes on the pressure face of the blades. No information was given regarding the composition of the steel, but a micro-examination revealed a structure typical of a straight 13 per cent chromium steel in the heat treated condition. It was found that all four blades had fractured due to fatigue, and the fact that sister ships in the fleet had also suffered cracks at the same position confirmed this diagnosis. These blades were to be repaired by welding and when such a repair was carried out, a prolonged annealing treatment at 690°C was essential, otherwise stress corrosion cracking would occur at an early date.

The author made an interesting point about calcium deposits on valves, due probably to the excessive lubricating oil consumption. It seemed from the evidence carefully collected that there was a possible contributory cause of valve failure. Calcium deposits certainly did build up on engines using alkaline lubricating oil, and one quite often saw heavy build up on the pistons of slow speed diesels at circumferential positions corresponding to those of the cylinder lubricating oil quills. The liner usually carried axial streaks where the deposits had polished it. These deposits, presumably, could also cause ring sticking and possibly give rise to adverse temperature distribution in the piston.

The last point the writers wished to make concerned the recent study made for the author in which various alternative types of machinery were compared for use in 26 000 dwt bulk carriers. It was noted that the data used in that study came mainly from experience. In their opinion this led to greater confidence in results of such studies. However, machinery comparisons were of considerable interest and the writers wondered whether it would be possible for the author to publish, in more detail, the basis of the calculations, all the factors included, the assumptions made, and the actual margin of difference between the various options. Could the author also say what the results of his calculations would have been if he had included reliability as a factor in terms of ship down-time and differences in maintenance costs.

MR. A. D. RUSCOE, F.I.MAR.E., wrote, observing that one of the features of the A.O. engine which was apparent already in the early days, before the onset of service troubles, was the very short piston crown land; he could recall no one remarking on this. The proximity of the top piston ring to the top of the piston would make things easier for the piston during its working stroke (but would not show the benefit during blow-down which might be expected with a cross-scavenge or opposed piston design). The top ring, however, would certainly run much hotter for a given working crown land-to-cylinder liner clearance than if it were lower down. He believed the builder's figures for top ring groove temperature were, nevertheless, quite low due to the internal cooling arrangements. Perhaps this was another case of service conditions deviating, as they all too often did, from test conditions. He wondered if any of the piston variations tried incorporated a ring pack lower down the piston or if the geometry of the design precluded this?

With regard to the groove in Fig. 3, of the paper, some years ago he had seen a 4-stroke marine engine of some 8″ to 10″ (203 to 254 mm) bore whose liners had very sharply defined deep grooves at the T.D.C. position of the top ring. These had all the appearance of having been machined, they were so bright and smooth, although slightly eccentric in the liner. These liners had been chromed but by a different process from that used in the A.O. No explanation had been found at the time and he believed this had occurred in only one engine of a series.

Very small but sharply defined wear steps were seen by the writer in an old opposed piston engine extending over only about a quarter of the circumference. These were outboard of the ports. The liner was not chromed.

Author's Reply

Replying to the discussion Mr. Bowers agreed with Mr. Edgar that valve mal-alignment might well explain the very high rate of exhaust valve failures in the early A.O. engines. In the author's case the need to honour tight chartering commitments meant that a quick and certain solution had to be found. Nimonic valves provided the best answer at the time.

Circumferential grooving of the A.O. liners was an unique phenomenon which had drawn considerable attention and speculation. Mr. Edgar had put forward yet another possible explanation based on service experience which to the author seemed quite plausible. Unless the A.O. engines were somehow resurrected as a viable prime mover, a situation difficult to foresee, then uncertainty as to the cause(s) of the liner grooving would always remain. However, from the available evidence it seemed possible that the flexibility of the "space frame" together with the instability of the strut load reverser as described by Mr. Edgar both contributed to the liner grooving phenomenon. The fact that pistons with hydraulic load reversers did not groove the liners, supported his theory that the strut load reverser was the principal cause of the trouble. However, generally speaking, experience seemed to show that the more rigid the design of a diesel engine structure the less the rate of liner wear and also the lower the rate of lubricating oil consumption.

Mr. Wesselo's exposition of attitudes and degrees of wisdom desirable in technical management and design was, in the author's opinion, unassailable. Perhaps for completeness Mr. Wesselo should have added another desirable attitude of mind namely an acute awareness of the need for design simplicity. From the Owners' point of view the TM 410 high pressure lubrication system for the valve gear was both complicated and expensive. By contrast several current designs of medium speed diesel engine employed a continuous flow system supplied from a small separate oil tank. This arrangement was relatively simple, operated at a low pressure, and had only one moving part, namely the pump impeller. It provided a continuous flow of oil that virtually eliminated all wear from the valve gear including wear at the top of the valve guides. In addition the steady re-circulating flow of oil ensured that the drains in the cylinder heads were always kept clear. With all these advantages, which included lower maintenance and lower first cost, it was difficult to understand why Mr. Wesselo's Company did not use the continuous flow system.

On the subject of exhaust valve failures, experience so far had shown that in the absence of deposits, valves did not burn. He agreed with Mr. Wesselo that a higher than average lubricating oil consumption in certain cylinders could account for an excessive build-up of valve deposits. If so one might reasonably expect exhaust valves to fail more frequently in these cylinders and not in others. However, statistical evidence to date did not support such a hypothesis although the true effects could have been masked by random fluctuations of individual consumption rates.

Whatever the process that caused valve deposits to build up, the author suggested that a rubbing movement between the valve and the valve seat might be beneficial. Ideally the movement should be in a circumferential direction although the brief, unexpected experience with bent valves in the *Cape Grenville* suggested that any action which removed the valve deposits would tend to prolong the valve life.

The logic behind Mr. Fenton's remarks was rather difficult to follow. The hard fact of the matter was that the rate of wear of the porous chrome liners in the A.O. engine had been unacceptably high, both at the top where heavy grooving had occurred and also along the length of the liner. Heavy liner wear was almost always associated with the lack of an adequate lubricating oil film. Mr. Fenton's statement that his Company had never found any evidence of a lack of lubricant film was therefore difficult to understand.

In reply to Dr. Norrby, Mr. Bowers agreed that the controllable pitch propeller offered very substantial advantages in twin engined single shaft installations. The acceptability of stainless steel as a propeller material appeared to be well founded, certainly according to the figures given by Dr. Norrby. These figures showed an overwhelming bias towards stainless steel. Apart from the initial hub cracking problems mentioned in the paper, all the stainless steel propellers fitted to the re-engined ships and including the *Cape Grenville* had so far operated satisfactorily. It therefore supported the statements made by Dr. Norrby regarding the problems encountered in the early 1970's.

The author agreed that for ship-board electric power generation a gearbox power take-off and controllable pitch propeller combination provided worthwhile benefits at normal ship speeds (15 kn). However at about 12 knots, i.e. at a finer pitch, the fall-off in propeller efficiency, although small was significant. In fact it was enough to shift the economic advantage from a constant speed propeller trimmed to a finer pitch and using a shaft driven alternator to the same propeller running slower at a coarser pitch and using auxiliary diesel alternators.

To Dr. Norrby's question, the author said that his views and overall conclusions on shipboard power generation were based on detailed calculations which included propeller efficiency losses, kW loadings, fuel rates and end-1974 fuel prices as major study parameters. Factors such as maintenance and lubricating oil costs mentioned by Dr. Norrby were also taken into account.

In reply to Mr. Mathews the author said that the decision to deal separately with each party was made to sharpen accountability, to simplify communication and above all to ensure that the decision-making paths were as short as possible. Moreover the speed at which the whole re-engining project had to be executed, together with the need to keep the cost below budget, had made it essential to pare all management overheads down to the irreducible minimum. The administrative systems that evolved perhaps exhibited stark simplicity but nevertheless provided adequate communication and the necessary feedback to Head Office.

The possibility of further trouble after the re-engining programme was reduced by the fact that the *Cape Grenville* had had TM 410 engines installed from the outset. This ship was at sea for over six months prior to the commencement of the re-engining programme and a clause in the contract stipulated that the re-engining programme could be postponed or cancelled if the performance of the *Cape Grenville* was not satisfactory.

With regard to the performance anticipated from the new engines, three points were worthy of comment:

1) the engine performance with heavy fuels up to 1500 seconds Redwood was better than had been expected;
2) it had been hoped that with the 12 TM 410 engines running at a lower than designed speed, the lubricating oil consumption rate would also be lower by a corresponding margin. In fact, the actual current consumption rate at 75 gallons (340 litres) per day was higher than originally anticipated but nevertheless was still within the guaranteed figure;
3) the emergence of the valve gear lubrication problem came as a complete surprise. The first TM 410 engine had been in continuous service elsewhere since 1968 and in view of this long gestation period it had been assumed that the valve gear lubrication system was fully developed and trouble free. This was not the case and fundamental modifications had to be made as described in the paper.

The gear ratio change referred to by Mr. Mathews was applicable to only four ships all of which had in-line TM 410 engines. Apart from changing the pinions and the wheel there was no other major modification to the gearbox. With regard to alignment, the only problem encountered was that of obtaining good tooth contact together with accurate alignment of the pinions and the main shaft. There was a great deal to be said for having a very stiff and rigid gearbox casing, otherwise the task of obtaining correct alignment and good tooth contact could be both elusive and decidedly tedious.

With regard to the change of valve lubrication from an impulse system to a continuous flow system, the engineering work involved was comparatively simple. Convincing the designers of the TM 410 engine that such a change was necessary proved to be a more difficult task. However once they were convinced that there was no alternative, at least in the short term, if heavy random wear of the valve gear was to be avoided, the change to a continuous lubricating system for the valve gear was made very quickly.

With regard to the lubrication of inlet valve seats, Mr. Mathews had correctly echoed the strong views held by the author. The need to inject expensive lubricating oil into the inlet system had been avoided in other comparable medium speed diesel engines and therefore the author felt that it was only a matter of time before this requirement was removed from the TM 410. The specific lubricating oil consumption would then reduce considerably.

With regard to the comparison of the various propulsion arrangements, the choice of a figure of 30 000 hours was made simply to ensure that whatever plant was chosen, and regardless of the trade on which the ships operated, the period would be in excess of the interval between the Lloyd's main engine surveys. Experience had shown that continuous survey was not the best way to operate medium speed diesel engines. Instead it was the author's policy to run the engines for as long as possible with no more than the necessary routine attention. Obviously, when assessing a variety of propulsion arrangements, the preferred type of survey cycle had been taken into account.

In reply to Commander May, Mr. Bowers stated that his only objection to the controllable pitch propeller (CPP) was that of high first cost. This objection was tempered by Commander May's proposal to introduce a cheaper and simpler controllable pitch propeller which had only two pitch or position settings. From the Figure shown, the propeller design seemed ingenious and simple to operate. Compared to the conventional CPP, the two position CPP operating in conjunction with twin medium speed engines would provide a less expensive installation but then it would not be possible to use shaft driven alternators. However, if waste heat recovery were used to generate electric power and the maintenance costs of the steam plant were not too high the overall installation could be very attractive.

In reply to Mr. Syassen, the author said that the A.O. liner wall thickness was between 19 and 28 mm. As to the geometry and disposition of oil holes the author stated that the manufacturers had tried many combinations and that it would be impracticable for him to venture into details in a paper of this type. He agreed with Mr. Syassen that the clearly defined circular grooves were an astonishing phenomenon. In most cases the upper edge of the top groove was sharp whereas the lower edge was bevelled over. Where two grooves appeared in the liner the upper edge of the top compression ring coincided with the upper edge of the top groove and the lower edge of the second compression ring coincided with the lower edge of the second groove. However he would like to emphasize that there were two separate and distinct grooves and therefore that the top compression ring must, presumably, enter and leave the lower groove as the piston travelled down the cylinder. That this should happen without severe damage to the rings was almost incredible but two grooves had been observed in several cylinders although in some cases the lower groove only went part way round the circumference. This supported Mr. Syassen's view that vibrations might be a contributory cause to such extreme wear rates.

He also agreed that the use of three different types of lubricating oil was impracticable. The problems that came immediately to mind were those of storage, administration and identification. In time an error must occur which would result in the incorrect oil being used and one such error could easily swamp all the likely benefits.

In reply to Mr. Roos, Mr. Bowers said that the two stroke engine admittedly had a better torque/speed characteristic but for bulkcarrier propulsion this was not necessary. For bulkcarriers the torque capacity available from the four stroke engine was sufficient. Moreover the four stroke engine had the advantage of lower cylinder liner wear and a lower rate of ring wear. Both these characteristics were, in the author's opinion, more important than high torque.

In reply to Mr. Allen the author said that for reasons already given he was indeed very sorry to have had to take the decision to replace the A.O. engine.

On the subject of fuel quality, it was a remarkable and little known fact that medium speed four stroke diesel engines appeared to have virtually no combustion problems. This had been verified by the TM 410 engines which over several thousand hours service on 1500 seconds Redwood fuel had, if anything fewer combustion problems than those experienced by large bore diesel engines operating over the same period.

With regard to the sodium content of the fuel the author could only answer Mr. Allen's question in part by saying that at the time of the exhaust valve failures in the *Cape Grenville* a careful check of the fuel oil centrifugal separators had been carried out. These were apparently in good working order but no check was made of the sodium dissolved in the fuel itself. Nevertheless significant correlation did appear to exist between exhaust valve failures and lubricating oil consumption rates and therefore it was probable that lubricating oil consumption had a greater effect on valve failures than fuel quality.

The author agreed with Mr. Allen that the running-in procedure could be improved, for example, by using low TBN oil. Recently this method had been tried out on the *Baron Wemyss* but it was too early to make any firm statement about the value of such a procedure. It would be interesting to see firstly, the rate at which the lubricating oil consumption decreased in those engines and secondly, whether the incidence of valve failures was significantly reduced. There was no relation between the inflexions in the lubricating oil consumption curves of the *Cape Grenville* with piston removals. Only one piston had been pulled and that was simply to establish the state of the working parts at the agreed six months guarante inspection period. The author supported Mr. Allen's remarks that pistons should remain undisturbed for as long as possible. Present thoughts were to run the pistons for the longest interval permitted by Lloyd's Register of Shipping and then to overhaul all the pistons together, deglazing the liners if necessary. New rings would be fitted at the same time and the engine would then run undisturbed until the next survey.

In reply to Mr. Wickens, Mr. Bowers stated that the TM 410 installation was in fact designed for unmanned operation and that since re-engining, all ships had received full UMS certification.

Mr. Wickens had stated that no stepped grooves had been found in the upper pistons of opposed piston engines. This would suggest that the explanation given by Mr. Edgar, suggesting that the load reverser was the principal cause of this phenomenon could well be correct. On the other hand he was not sure whether it was valid to compare the behaviour of a crosshead engine with a trunk piston engine. The crosshead engine by design had greater lateral rigidity than the trunk piston engine and thus provided a very stable and vibration-free structure. Nevertheless the author would be the first to agree that the explanation of the circumferential grooves given in his paper was at best only one of several possible hypotheses.

He confirmed that both the 9 and 12 cylinder TM 410 engines had approximately the same bmep but the horsepower output per cylinder of the 9 cylinder engine was greater because it ran at a higher speed. From the limited information available to date there was no noticeable difference in performance.

Mr. Bowers agreed that for those in the opposed piston business the best medium speed engines had no exhaust valves at all. However it really was necessary to get an indication of the order of the problem. The cost of replacing exhaust valves at the current rate of failure was less than 0·15 per cent of the cost of the fuel. This was less than the difference in specific consumption between the slow speed engine and the medium speed engine. It seemed possible that once the problem of eliminating valve head deposits had been solved, and there were a number of possible solutions under development at this moment, then the cost of exhaust valve failures could be reduced to negligible proportions.

With regard to lubricating oil consumption, it was perhaps too early to form any conclusions about the trend following the initial running-in period. He would agree with Mr. Wickens that a "bath tub" consumption/time profile should emerge but to date there was no evidence of this. Mr. Bowers confirmed that the flow of oil to the cylinder lubricators was metered separately from the oil supplied to the crankcase. He would also agree with Mr. Wickens that the torque/speed curve for the two stroke engine permitted a greater output on one engine, always assuming that a controllable pitch propeller was not fitted. However, against a background of rising fuel and maintenance costs the shaft alternator and controllable pitch propeller combination was commercially very attractive and, given a controllable pitch propeller, the torque/speed curve of the 4 stroke engine was quite adequate.

Mr. Wickens' assertion based on the figures given in the paper that, for a single engined ship, the probability of losing propulsion in a year's operation was unity was of course correct. To be precise it is 98·2 per cent or in other words there is a 1·8 per cent chance that there will be no failure in a year's operation. For a twin engined ship with assumed engine failure and repair rates, the reliability calculation was slightly more complex. The results of a Markov analysis showed that for a twin engined ship the likelihood of losing all propulsive power was one chance in thirty-two years as stated in the paper. To date, since re-engining, none of the ships had stopped at sea although on several occasions they had been reduced to one engine due to minor faults on the other. With reference to gearbox failures, there had been, to date, one failure due to incorrect initial assembly and three cases of tooth scuffing. The single gearbox failure which occurred on the second A.O. engined ship was caused by the failure of a locking device inside the gearbox. This allowed a nut from the main wheel to run amok and cause severe damage to the teeth on both pinions. The cause of scuffing was by no means clear but it appeared to be linked to poor oil quality and abnormally high temperatures in the gearbox. In each case, honing of the gear teeth and the use of an E.P. oil had provided a complete cure.

In reply to Mr. Cliffe, Mr. Bowers said the A.O. engine exhaust valve failures appeared to have been due to the combination of a number of causes. Mr. Edgar had mentioned mal-alignment whereas his own impression had been that the majority of the failures were caused by high temperatures and/or corrosion. He was not sure whether the corrosion was primarily due to vanadium or due to the excessive quantities of alkaline lubricating oil which were burnt in the engine. The A.O. engine exhaust temperatures had occasionally been very high particularly when large quantities of lubricating oil were leaking past the piston crown and causing after burning. Referring to the decision to use a flame plate construction in the A.O. engine, Mr. Bowers said this design had a great deal to recommend it. For engines of this size it was debatable whether there was much advantage in fitting caged exhaust valves. By careful design of the cylinder head fastenings and attachments it would probably be just as easy to remove a cylinder head as it would be to remove separate exhaust valve cages. If the life originally anticipated for the exhaust valves had been achieved then the flame plate and the cylinder head design of the A.O. would have been quite acceptable to the operator.

It was very easy to be critical of the A.O. engine design in hindsight but from the point of view of a bulkcarrier operator it was doubtful whether the weight saving achieved by the space frame was worth the development effort put into it. However, every tonne saved would mean an extra tonne of payload especially if a ship were trading in depth restricted areas.

With reference to salt build up, there were no signs of any appreciable build up in other parts of the engine. The compressor was washed as a matter of routine, but much of the dirt that collected was of a greasy nature.

The rise and fall of the A.O. engine, referred to by Commander Tyrrell, had provided a valuable lesson to us all in the marine industry. In particular it emphasized the need for extensive development under realistic service conditions before putting an engine on to the open market.

Mr. Kennaugh had provided a useful account of the design and progress of the Sulzer "Z" engine. In the context of the A.O. problems and Commander Tyrrell's remarks the Sulzer policy of extended development, as emphasized by Mr. Kennaugh, had obviously played an important part in the development of the "Z" engine.

Mr. Manson had referred to power rate factors and suggested that low specific air flow could possibly account for some of the piston ring, exhaust valve, and liner problems on the A.O. engine. Independent full load tests carried out on one A.O. engined ship gave figures of 0·88 bar for the boost pressure and 5·8 kg/bhph for the air flow. The manufacturer's projected air flow figure was about 6·25 kg/bhph. It would appear that both the actual and projected A.O. specific air flows were below those quoted by Mr. Manson and therefore his association of low air flow and engine thermal problems might well have substance. Significantly however some A.O. engines, notably those fitted with pulse converters and operated by other Owners, had had much higher air flows (7·2 kg/bhph) but problems, including the rapid wear of the liners nevertheless still persisted.

To the questions raised by Captain Anketell-Jones, the author said that several machinery alternatives had been examined before the TM 410 engine was finally selected. With regard to the fuel specification the TM 410 engine was designed to operate on fuels up to 3500 seconds Redwood. None of the re-engined ships had yet run on fuels higher than 1500 seconds but it was hoped that tests with 3500 seconds fuel would soon commence on one ship. From the excellent combustion characteristics exhibited to date on the TM 410 no troubles were expected when burning the impurities currently present in high viscosity marine fuel oil.

With regard to engine availability, the author said that the 25 days attributed to single engine steaming might seem excessive as it probably did to Captain Anketell-Jones. However a closer study of Table I in the paper would show that almost half of the 25 days was due to a single turbo blower failure at sea (12 days) whilst the remainder was due to items such as exhaust valve failures and leaky joints in the valve gear high pressure lubrication system. If only the direct engine items were examined the 25 days would become 13 days and the resulting effective time lost by the six ships would fall to 2·6 days total. As a percentage of the steaming days (1083) the effective time lost was therefore only 0·24 per cent. Most of the problems mentioned occurred whilst the ship was under way and therefore the question of overhaul in port had little relevance. It was usually a simple matter to work on one engine while the other was running and there was, obviously, a strong commercial advantage in correcting a minor fault at sea rather than to await arrival in port.

With regard to specific lubricating oil consumption, the manufacturer's figure of 0·8 g/bhph was based on the designed MCR output whereas the 1·08 g/bhph referred to by Captain Anketell-Jones was based on a much lower output. Compared to other medium speed diesel engines, and even some large bore engines, a specific consumption of 1·08 g/bhph was an acceptable figure although like most ungrateful ship managers the author would like to see this reduced much further. Once the lubricating oil supply to the inlet valves on the TM 410 had been eliminated then the figure quoted by Captain Anketell-Jones would reduce to just over 0·85 g/bhph.

The author concurred with Commander Griffey's remarks that the saga of the A.O. engine was indeed a sad one. He would also support Commander Griffey's plea that Rustons should publish a paper, if only to show the immense amount of effort they had expended on the A.O. project and the many problems that they had successfully overcome.

For the re-engining programme, resin chocks were used primarily because of the need for speed of installation. The material chosen had a good record of reliable service and also had the approval of the major classification societies. Local pre-heating of the chocking area was recommended to assist the uniform bedding down and the final curing of the resin material. The temporary distortions referred to by Commander Griffey were probably due to several causes. The most likely was the uneven heating of the very large engine room space, only a very small portion of which was filled by the medium speed diesels themselves.

With regard to major overhauls by engine replacement, the author believed that this concept was eminently practicable for prime movers such as aircraft type gas turbines, but not for medium speed diesels fitted to bulkcarriers. However, there was an excellent case for the overhaul of medium speed engines by the replacement of combined piston, liner and cylinder head assemblies particularly if the piston rings and liners had previously been "run-in" on a reciprocating rig using a fine abrasive. He hoped that medium speed diesel manufacturers would provide units of this type as this would greatly reduce time required for both survey and overhaul. The "running-in" of such units would also reduce the high initial lubricating oil consumption that invariably occurred when rings and liners first came together without pre-treatment in an actual engine.

Many of Mr. Dunshea's remarks were to the point and added to the value of the paper. He had quite rightly expressed surprise at the need for injecting lubricating oil into the TM 410 air inlets to prevent valve seat wear. The author, along with many others was also surprised but nevertheless oil injection was a standard feature of this particular engine. It was hoped that the engine builder would very soon come up with an alternative solution which if successful will save over 20 per cent of the total lubricating oil costs.

On the subject of exhaust valves, the TM 410 engined ships all burn heavy fuels up to 1500 seconds Redwood obtained in the normal way from commercial sources worldwide. Insufficient experience had yet been obtained with valve rotocaps to assess their value on this engine. One engine in the m.v. *Baron Wemyss* was fitted with rotocaps while the other engine was not. It was intended to run both engines without exhaust valve maintenance to try and establish the relative values of rotocaps in this size and type of exhaust valve configuration.

The author was indebted to Mr. Morrison and to Mr. Frederick for amplifying many of the points raised in the paper. With regard to details of the study, the principal factors examined were:

i) capital and installation costs;
ii) fuel, lubricating oil, maintenance and spare gear costs;
iii) engine weights, gear box weights and overall dimensions of the total installation;
iv) reliability and availability.

The study concluded that the twin medium speed diesel installation had the lowest combined cost in all but one of the six alternatives examined. The cheapest installation was the single geared medium speed diesel but it was considered that the advantage of increased availability justified the additional cost of the twin engined installation. The overall cost differential in favour of the medium speed installation at 1974 prices ranged from £55 000 for the twin medium speed/CPP installation to over £130 000 in the case of a single medium speed engine with a fixed pitch propeller. In view of these figures the 4 stroke medium speed diesel would, in the author's opinion, have to become very unreliable indeed before it would lose its economic edge. Therefore, even if engine reliability could be quantified in cost terms, itself a daunting task, the overall conclusions of the study would probably remain unchanged.

The study itself was being updated at monthly intervals by comparing information supplied from the medium speed engined ships and slow speed engined ships in the author's fleet. So far the results agreed with the predictions made by the consultants although one could not be too positive until the overall running costs had been checked over a full five year cycle, thus including the survey costs.

The author thanked Mr. Morton for his historical survey of diesel engine development, which must have revived many memories among those present. He agreed that the uprating of diesel engines had not always been accompanied by the reduction in maintenance that the steady improvement in engine technology might have led one to expect. It was undeniable that design problems had a habit of repeating themselves and also that it was risky to take short cuts in the development of any new prime mover. Attention to detail was, of course, vital, but perhaps the most difficult judgment that any designer had to make, working as always on a limited budget and against a tight time schedule was how much to rely on short term rig tests, how much to evaluate by prolonged test-bed running and how much to leave to experience and intuition. As Mr. Morton had inferred, with marine machinery, there was no real substitute for an extensive proving period at sea.