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MICROBIAL DEGRADATION OF MARINE LUBRICANTS – ITS DETECTION AND CONTROL

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SYNOPSIS

Logical theories can be postulated for the escalation in incidents of marine main diesel engine lubricant malfunction and associated corrosion. If they are correct, the forecast is temporarily gloomy, until preventive measures become generally used and some long term reformulations are carried out. A number of early-warning signs can now be recognised. On-board microbiological tests have proved of great value; biocides are now known which can protect oils in use and many infected systems have been successfully sterilised.

Other 'straight' oil infections are discussed briefly but as yet the financial implications of other known infections are less pressing.

INTRODUCTION

Microbial degradation of cutting oils and coolants, rolling oil emulsions, hydraulic and bearing oils and aviation and other fuel oils has been recognised for many years. Whilst the economic consequences of infections of cutting and rolling oil emulsions is universally appreciated, and considered adequate justification for anti-microbial measures, infections of straight oils (except fuels) have largely been considered an academic curiosity However, in 1969, a ship incident occurred in which malfunction of the main diesel engine lubricant and associated failure and corrosion was for the first time correlated with massive microbial infection. Since then there has been an escalating number of such incidents, many costing hundreds of thousands of dollars. Whilst the better ability to recognise such problems must account partly for an increase in numbers, there is undoubtedly also a real increase for which there must be a logical explanation. To understand this logic we must know something of the characteristics of microorganisms.

MICROORGANISMS

The majority of problems encountered in lubricants have involved bacteria, organisms which are no more than a few microns in size, which can double in size and divide into two every twenty minutes (under ideal growth conditions) and which usually separate from each other after each division. They flourish in warm conditions and most prefer neutral or slightly alkaline conditions. The usual dominant bacterium is *Pseudomonas*. Moulds are often found in small numbers, and in at least one case *Aspergillus fumigatus* has been the dominant

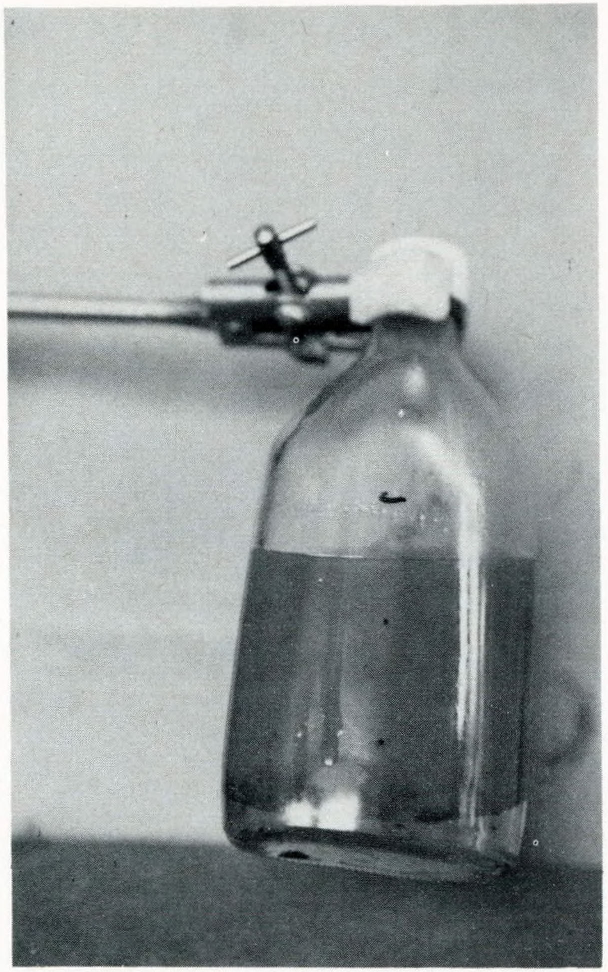

Fig.1 — Laboratory infected slow-speed diesel engine lubricant showing concentration of growth at the oil/water interface and 'milky' appearances of oil due to the formation of a stable emulsion

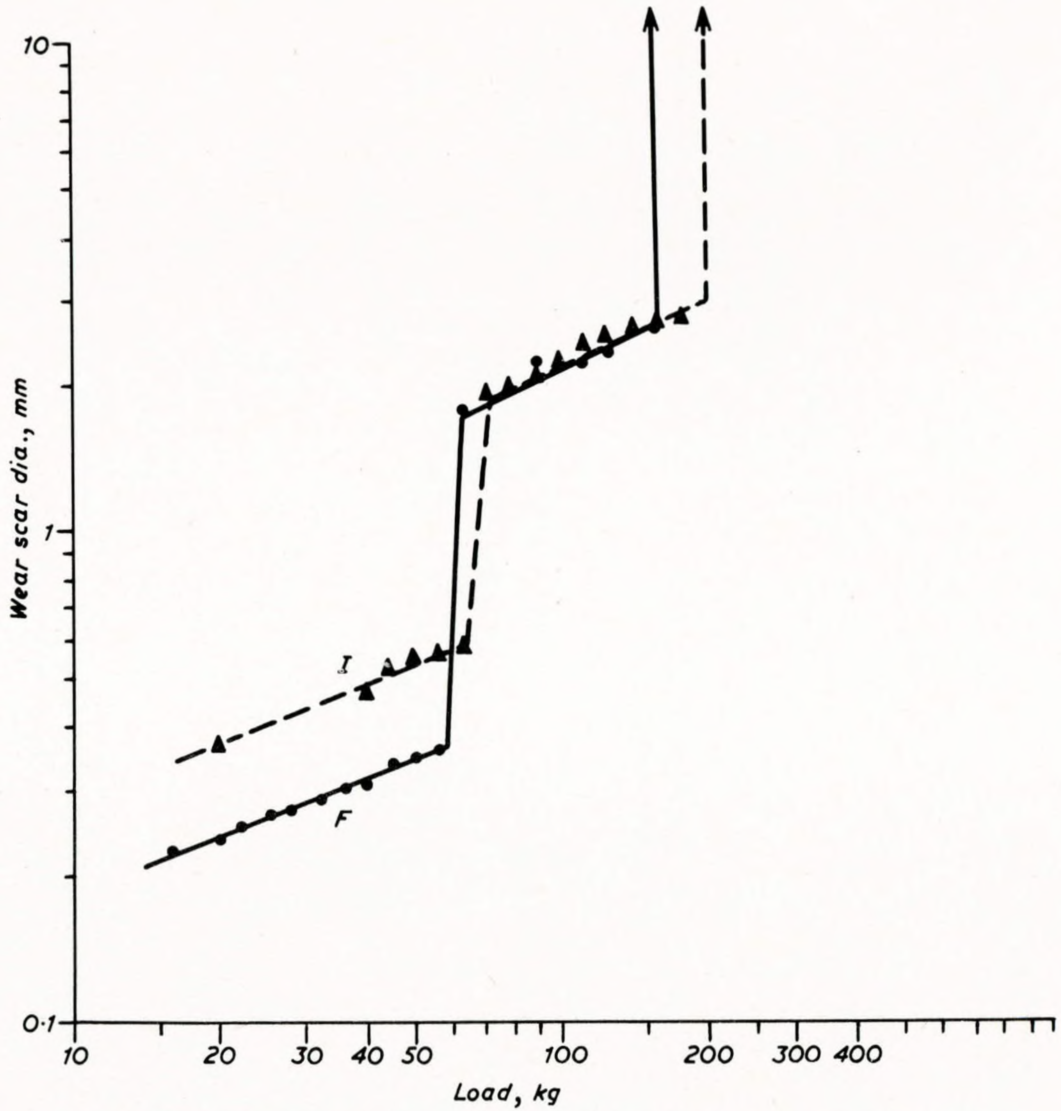

Fig.2—Four-ball wear tests (IP 239/73) on *F*) Fresh lubricant (———)
I) Laboratory infected lubricant (— — —)

Note the increase in wear rates at low loads after infection organism. Aspergillus fumigatus has received considerable attention as a potential hazard to supersonic aircraft fuel systems[1]. It is a filamentous organism and hence is found in discrete 'mats' of growth. Yeasts have often been isolated from hydraulic oils and from steel mill bearing oils but have so far played a minor role in ship's lubricant infections. Both yeasts and moulds prefer slightly acidic conditions.

Figure 1 is a photograph of a typical marine lubricant with an associated mineral salt aqueous phase after three weeks infection in the laboratory.

The milky appearance of the oil is due to a permanent, stable, entrained water phase, and is typical of laboratory generated infections. Infected oils in use also tend towards this condition although the milky appearance is modified by a darkening of the colour of the oil.

THE LUBRICANT PROBLEM

Microbes have an incredible ability to degrade all types of organic (and sometimes inorganic) materials. Different microbes have different preferences for 'diet' and for physical conditions (temperature, pH, light, oxygen tension and electron potential). All must have water to grow. Water is a by-product of their growth, so once started they are to some extent self sustaining. They must have a balanced 'diet' with particular respect to Carbon, Nitrogen and Phosphorus. It is the greater availability of these last two elements, either directly as additives in modern oils, or as coolant additives when these leak into the oil, which is probably a major factor in the escalation of the lubricant problem.

There is nothing magical about the role of microbes in lubricant malfunction and corrosion and this has been described in detail elsewhere[2]. We can summarise their activity in general although individual species of microbes may accomplish only some of the processes listed.

1. Their growth products may be corrosive e.g. organic acids, hydrogen sulphide, ammonia.
2. They reduce inter-facial tension and hence stabilise a water in oil dispersion. Such a dispersion tends to be corrosive and is difficult to resolve by centrifugation.
3. They attack some molecules of the base oil preferentially and hence alter its viscosity and chemical composition.
4. They may attack the oil additives and reduce their effectiveness.
5. Local concentrations of microbes deplete that area of oxygen and hence establish electrochemical anodic pitting due to an oxygen gradient.
6. The physical presence of the mass of microbial cells plugs filters and may restrict flow through pipes and orifices.

The ability of a microbial infection to change the load-bearing properties of an oil is illustrated in Figure 2. Typical "Four-Ball" Wear Data are given for an unused marine lubricant and the same lubricant after a massive Aspergillus fumigatus infection. The increased wear rates at low loads are very marked.

The time-scale for a serious problem to develop can be a matter of weeks but is very dependent on sufficient water being available for a growth 'explosion'. Sulphide generating bacteria may appear after other organisms have flourished and have reduced the oxygen level. They prefer stagnant conditions and hence may be associated with 'lay-up'. Microbial sulphide corrosion is an intensely vicious process.

ESCALATION OF MICROBIAL SHIP INCIDENTS

There must be some logic which explains the increasing number of marine incidents. At least five factors can be recognised which may all have contributed to this escalation.

1. The increased use of non-toxic engine coolant inhibitors to fulfil regulations for coolants also used for fresh-water evaporators. Chromates (where still in use in coolants and at adequate concentration) are anti-microbial, but they cannot be used where the coolant is also used in drinking water evaporators. Other types of coolant additives may actually support microbial growth and hence if they leak into the lubricant may supply nutrients, water and microbes. This appertains particularly to water-cooled pistons.
2. The increased use of sophisticated oil formulations which may constitute a complete 'diet' for microorganisms or may become so when supplemented by nutrients from the coolant.

3. A progressive change in base oils for lubricants. Naphthenic base oils have partially been replaced by paraffinic base oils which may be more prone to microbial attack.

4. Reduced use of renovating tanks. When used routinely these can function as batch sterilisers.

5. Use of undersized purifiers which cannot effectively remove microbes and positioning of purifier suctions to locations which are less able to pick-up and dispose of pockets of microbial infection.

6. Reduced engine temperatures, due to 'slow steaming', may increase susceptibility to microbial attack.

7. Increased lay-up during which growth, particularly by sulphide generating bacteria, may proceed un-noticed.

All of these changes have been made for very sound reasons; it is unfortunate that no-one could have foreseen that when they occurred together a potential microbiological 'time-bomb' would be created. It still, of course, requires an inoculum of those organisms which can flourish at the engine temperatures and in the formulations in use. It is likely that these organisms are acquired in the tropics where the adaptation for organisms growing at high ambient temperature to growth at engine temperatures is more probable; in the North Atlantic the larger temperature difference would discourage this adaptation.

Even when a potential 'time-bomb' has been 'fused' (inoculated with the right organisms), oil changes and corrosion will not develop rapidly unless adequate water is available for growth proliferation. All of the evidence is that early detection is important. In the very early stages no lubricant malfunction and corrosion are seen; at later stages progressive changes can be identified and a major casualty may occur.

Experience has shown that finding large numbers of microbes in the bottom of a system may be no more than an indication that leakages are occurring from infected coolants, possibly during engineering work. Such isolated pockets of microbes have sometimes proved difficult to eradicate, and if the type of organism in the coolant is not able to flourish in the lubricant formulation, these pockets of infection may have no great significance. Where the microbes are able to flourish in the lubricant they tend to spread progressively throughout the system. To draw the right conclusions it is therefore essential that the right samples are examined. It is suggested that the following set should be taken :

1. Lubricant from bottom of sump - by hand pump if fitted or by dropping a sterile tube down the sounding pipe.

2. Lubricant entering purifier, after shutting down heater and allowing temperature to fall.

3. Lubricant from bottom of renovating tank.

4. Lubricant from main circulation e.g. before filter.

5. Lubricant from spare charge.

6. All coolant systems, preferably on forward rather than return flow.

The exact locations of the sampling points, the temperatures of the systems, and the brand of lubricant in use and coolant additive should all be notified to the testing laboratory if one is used.

ON-BOARD INDICATIONS OF MICROBIAL ATTACK ON MARINE LUBRICANTS

All or some of the following phenomena may be observed when lubricant infection occurs. Individually each may be ascribed to some non-biological cause. They should therefore be taken as possible indications of infection which should be confirmed by microbiological tests on-board or in the laboratory.

1. Stable water content in the oil after the purifier.

2. Increased acidity of the oil due to 'weak' acids.

3. Unusual smells.

4. Sliminess of the oil. This may be particularly apparent on the crankcase wall or in a glass sample bottle.

5. Honey-coloured films on the journals.

6. Corrosion pitting of the journals.

7. Rust films on machined parts.

8. Black 'graphitic' pitting in cast iron.

9. Black stains on white metal bearings.

10. Brown or brown/black deposits.

11. Sludge accumulation in the crankcase and sump and sludge separation at the purifier.

12. Corrosion of the purifier bowl.

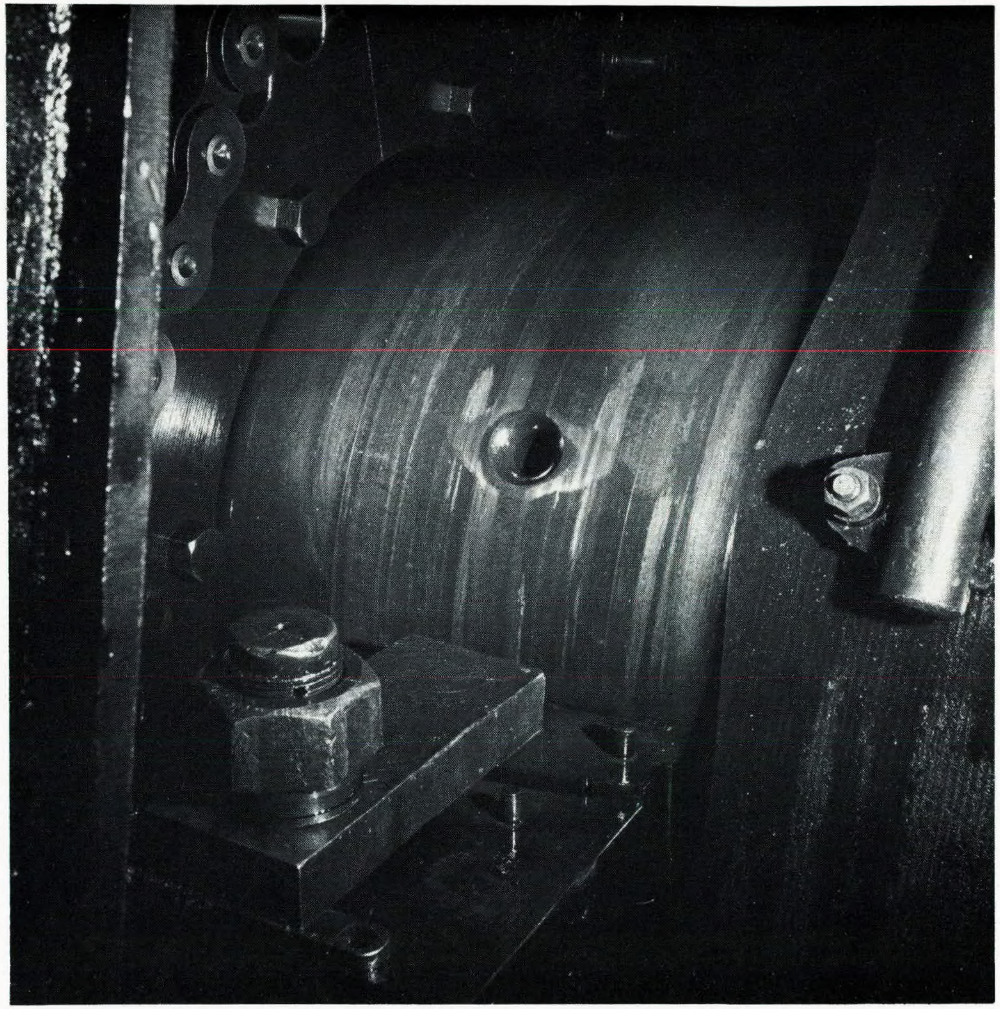

Fig.3—Damage to main journal due to lubricant infection (Case 1)

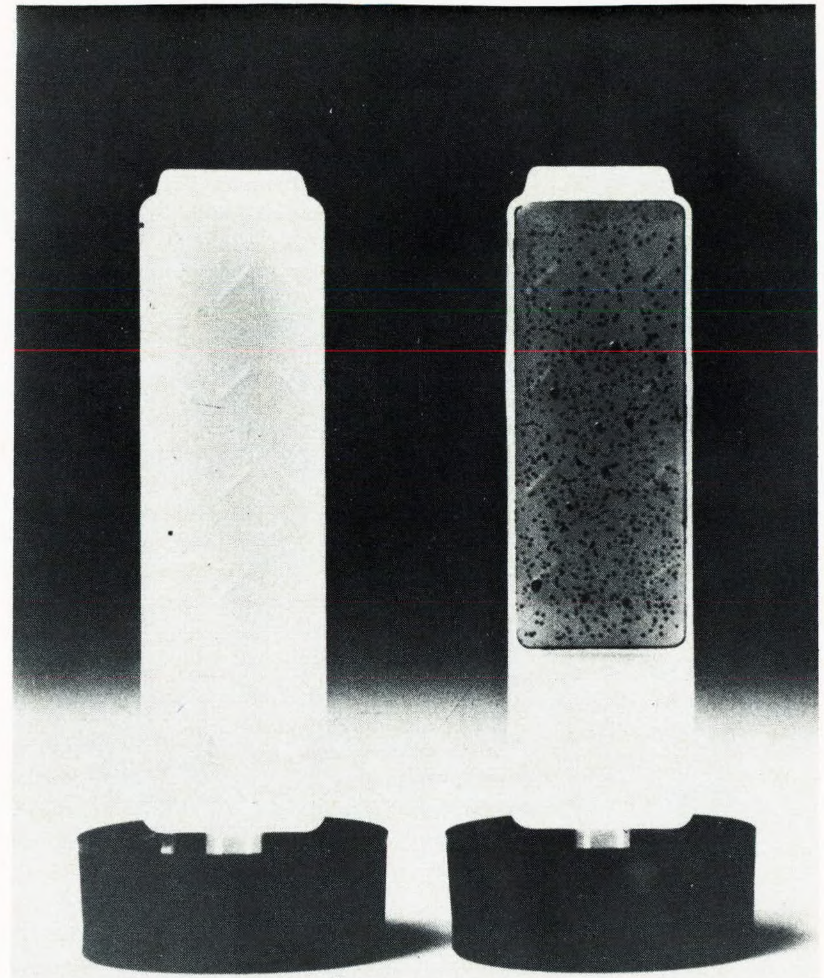

Fig.4 —Easicult TTC dip-slide before (left) and after (right) use on infected lubricant

13. Filter plugging, particularly in rough weather. In total - unlucky thirteen!

These phenomena have also been described by King and McKenzie[3], and a number of Case Histories have been described in detail by Hill[4]. Figure 3 illustrates the typical appearance of a main journal following massive bacterial infection of the lubricant.

MICROBIOLOGICAL TESTS

The regimes for preparing sample bottles, taking and transporting samples and using on-board test methods have adequately been described elsewhere[5]. It is sufficient to say that in on-board tests, slides coated with a nutritive gel can be dipped into the coolant or lubricant, incubated overnight (a yoghurt maker available at retail chemists is a useful cheap incubator) and the resultant intensity of red spots compared to a standard to determine the numbers, if any, of bacteria present. A typical result is shown in Figure 4.

The interpretation of a set of results is best left for the expert although this is outlined elsewhere[5,6].

REMEDIAL MEASURES

A whole spectrum of approaches are available depending on the extent of infection and corrosion. In the early stages it is possible to salvage and sterilise most of the oil by renovating at 180°F or more for 12 - 24 hours, whilst circulating through the purifier and its heater, and then allowing to stand at high temperature for 12-24 hours. At the same time the sump and system can be cleaned out and sterilised with some of the oil containing a biocide. The renovated oil can then be returned to the sump via the purifier and topped up. The renovating tank should then be cleaned out.

Depending on circumstances it may be considered desirable to add a biocide to the oil in use.

Even where this is not considered necessary there may be an appreciable carry-over from the sterilising flushing oil to the new charge of lubricant.

No biocide should be added to oil without expert advice and the approval of the oil supplier. Almost every biocide has some adverse affect and this should be allowed for in any anti-microbial procedures.

For example a proposed biocide may itself interfere with the load-bearing properties of the oil. This is illustrated in Figure 5 where the "Four-Ball" Wear Data are given for an unused marine lubricant and the same lubricant after the additions of biocides B and M.
Obviously biocide B would be unacceptable if the lubricant was operating under conditions of loading likely to approach the initial seizure point.

A biocide may also impair the ability of the lubricant to shed water. This is a particular failing of the oil soluble Quaternary Ammonium Compounds, and as they are also invariably sold as solutions in a low flash point solvent, there are severe practical limitations to their use.

Some biocides are themselves corrosive. Tables 1 and 2 give the results of a simple experiment in which two steel balls were dropped into replicate 50 ml aliquots of two oils plus 1 ml of distilled water and kept at 37°C. A variety of biocides were incorporated in the oils at concentrations appropriate to their biological effectiveness. Inspections were made at intervals of a few days and the onset of corrosion noted and arbitrarily assessed. It will be seen that Oil A (a straight non-additive oil) gave limited protection against corrosion and the various biocides increased or reduced corrosion. Oil B, an additive oil, protected against corrosion, but its effectiveness was impaired by certain of the biocides. Other experiments have indicated that for other brands of oil, different results may be obtained.
Although it is possible in most cases to make a choice of biocide which will result in no known adverse affect, circumstances of availability, transportation and restricted treatment time, may mean that another biocide must be used. Provided that its deficiencies are known, a regime can usually be instituted which allows for them.

If there are already indications that the oil in use has become aggressive, there is obviously little point in trying to reclaim it by renovation. Thus a bearing inspection is desirable as well as a conventional oil test.
On the basis of all the information available it

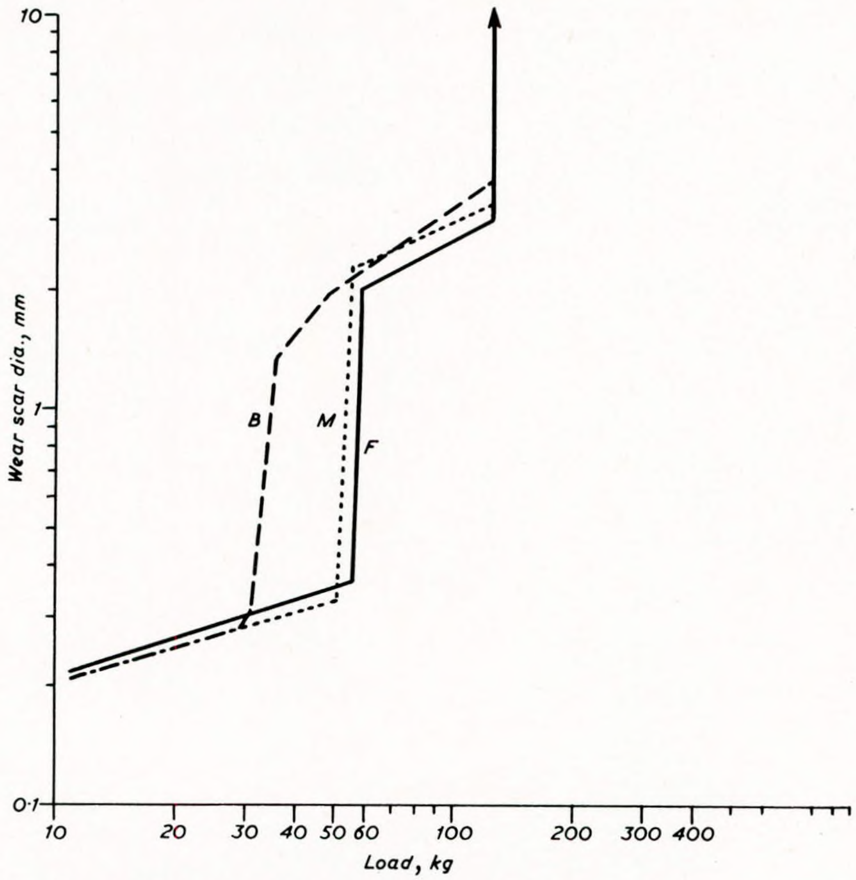

Fig. 5 — Four-ball wear tests on F) Fresh lubricant (———)
B) Lubricant plus biocide benzyl cresol (— — —)
M) Lubricant plus biocide MAR 71 (--------)

Note the marked change in initial seizure load

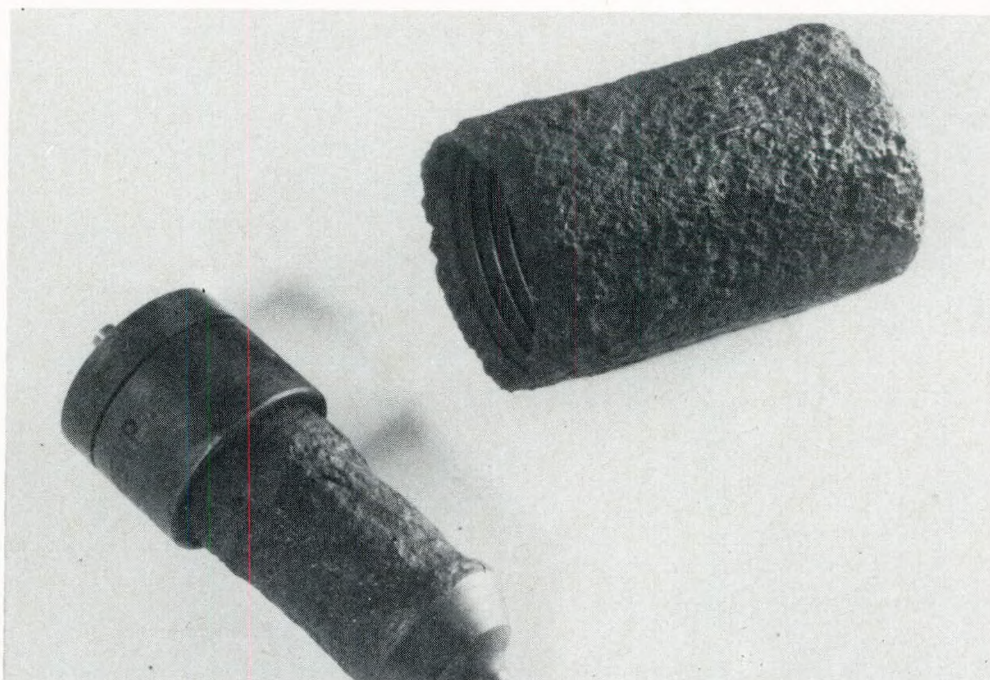

Fig. 6 — Severe corrosion of fuel valve due to growth of sulphide generating bacteria in the coolant. (Case 2)

Tables 1 and 2

Effect of biocides on corrosion of 0.5" SKF Grade 1 steel balls

Steel balls were immersed in 50 ml oil plus biocide and 1 ml distilled water and maintained at 37°C. Corrosion was recorded (+ to +++++) and any precipitation noted (P).

Table 1 Non-additive lubricant

Time elapsed (days)	BIOCIDE ADDED								
	CONTROL	A	B	C	D	E	F	G	H
7	-	+	+	+	+	+	-	+	+
14	-	++ P	++ P	+	+	+	-	+	+
28	-	++++ P	++++ P	+	+	++	-	+	+
42	+	++++ P	+++++P	+ P	+	++++ P	-	+	+

Table 2 Additive containing lubricant

7	-	-	-	+	-	-	-	-	-
28	-	-	-	+ P	-	-	-	-	-
42	-	-	-	++ P	-	-	-	-	-
60	-	+	-	++ P	-	-	-	-	-

may be considered prudent to dump the whole of the oil charge. In some circumstances it might first be 'spiked' with a high concentration of biocide and used as a sterilising flush, preferably with supplementary filtration.

COOLANTS

If infected coolants are detected this may be important in its own right, as corrosion may be occurring in the jacket or piston crown. A very severe microbial corrosion of the fuel valves has occurred on at least one ship (see Figure 6). An infected piston coolant may also act as a reservoir of infection for the lubricant.

Because of the high temperature and alkalinity, infections of coolants are predominantly bacterial.

Bacteria may either oxidise or reduce the nitrite of nitrite-based coolant inhibitors, which thus become inactivated and corrosion can then take place. 'Soluble oil' coolant inhibitors usually become unstable and 'tramp oil' separates at the surface. This is due to the microbial degradation of the emulsifiers and coupling agents which stabilise the oil dispersion. All coolants usually become more acidic during microbial infection.

A variety of water-soluble biocides can be used to flush out infected systems. There are limitations imposed on the addition of biocides to the coolants in use by the toxicityregulations of the Department of Trade. Apart from these limitations any biocide must be compatible with the coolant inhibitor in use, compatible with the lubricant should a leak occur and should not alter the pH of the coolant to an undesirable level.

PRECAUTIONARY MEASURES

Many ship owners have decided to conduct routine microbiological surveys which have proved of some value, as remedial measures can then be instituted before any adverse affects are noted. At the moment, it seems to be prudent to err on the side of caution and eliminate infections when ever they are detected at a substantial level.

Obviously it is sensible to be particularly careful to rectify coolant leakages, to renovate routinely and to organise the purifier lay-out and operation to obtain the maximum benefit.

There is no doubt that the purifier heater and centrifuge both reduce the numbers of living microbes in oil, and a programme of work is now in hand to optimise this part of the oil system so as to reduce the likelihood of a hazardous infection arising.

Concurrently there is much interest in the re-formulation of both lubricant formulations and coolant additive formulations to make them more resistant to microbial attack. Adding a compatible biocide to an existing formulation is only a partial answer, as all biocides are depleted in use and hence will have a finite life.

INFECTIONS IN OTHER STRAIGHT OILS

This paper has largely been concerned with microbial problems in lubricants for slow-speed marine diesel engines. There are good reasons for this. Both the lubricant and coolant temperatures are conducive to microbial growth, and a particularly nutritious environment is created when the chemicals of the coolant inhibitor leak into the lubricant. One does not anticipate severe problems for example in high speed engines, where both lubricant and coolant are hot enough to be self-sterilising.

Gear-oils, 'straight' cutting oils, hydraulic oils and preservative oils can all support microbial growth in the right circumstances. The symptoms of the infection and the remedial measures follow closely those described for slow speed marine diesel engines although some changes in properties may be relatively more important. For example infected hydraulic oils may cause equipment malfunction due to filter and valve plugging, and there is some evidence that cavitation is increased.

Of growing concern are the increasing problems due to fuel oil infection. The American Navy has suffered severe microbial corrosion problems[8], particularly in sea water displaced fuel systems. Light fuel oils support most microbial growth; in severe cases the fuel may become corrosive, but the most common symptoms are coalescer and centrifuge malfunction and filter plugging. Further details of symptoms and treatment are published elsewhere[4].

There may well be other locations in the ship (e.g. stern tube lubrication) where microbial infections will prove to be deleterious.

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CASE HISTORIES

Two case histories have been selected to illustrate the onset, consequences and cure of microbial infections in main engine lubricants and coolants.

In each case the permission of the ship-owners to publish has been sought and given and their cooperation is gratefully acknowledged.

The identity of the ships, the oils and chemicals involved and the make of engines could possibly be deduced or may be obvious from prior knowledge. There is no intention to criticise any of these or any aspect of ship managements.

CASE 1

The ship was 20 years old. The main engine had water cooled lower pistons; the coolant was protected with potassium bichromate (nominally $2\frac{1}{2}$lbs per 1000 galls.) and was common to pistons and jacket. There was no previous history of serious lubricant malfunction, nor unusual engine corrosion. Two months before the first bearing failure a routine survey of the journals and bearings had been conducted without adverse comment.

Three weeks before the first failure a major leakage of coolant into the engine was noted. A "Speedy Moisture Test" showed 7% water, but an oil sample was considered fit for further service. Water became progressively harder to separate cleanly at the purifier; an intractable stable emulsion had formed in the oil, and a watery sludge was being discharged at the purifier. Due to fracture of the main fuel pump crankshaft drive coupling, which was attributed to wear of a drive shaft bearing, the vessel was delayed for a week. A new crankshaft was fitted and the damaged bearing re-metalled. Meanwhile the oil was purified for a week, although the renovating tank was not used. 600 gallons of new oil were added to top-up the system and the ship sailed still trying to purify emulsified oil. As she entered the next port of call, all of the main bearings failed, white metal being extruded. An inspection revealed black scale on the white metal and blackish-brown deposits on the journals overlying corrosion pitting.

Tests on oil samples now revealed the following relevant facts:- oil from top of renovating tank: water content 11.1%, total insolubles 0.32%, initial pH (1P177) 5.9.

Oil from Port Storage Tank (for comparison) : water content - faint trace, total insolubles - nil, initial pH 8.8.

Water in the renovating tank contained 250,000 bacteria per ml and water underlying the sump oil contained 8,000,000 bacteria per ml. No sulphide generating bacteria were detected; no yeasts or fungi were detected. The coolant was not infected.

It was concluded by the oil supplier that the casualty could be ascribed to microbial attack on the lubricant with associated additive depletion, emulsification and corrosion. This opinion was supported by an independent consultant.

Repairs were effected, but before the laboratory results were known a new oil charge was introduced. When the laboratory tests confirmed the presence of bacteria, the oil supplier advised on anti-microbial measures. The renovating tank was cleaned; the oil charge was treated by adding 0.5% of biocide 'G', and the oil was circulated at ambient temperature for more than 24 hrs. Much sludge was removed by temporary gauze filters and it was noted that a black rubbery deposit formed at the purifier bowl. This oil was discharged to bunkers and a fresh oil charge delivered and 0.15% of biocide 'G' added.

The coolant was dumped, the system cleaned and fresh coolant made up containing bichromate and 0.5% of biocide 'G'.

The repairs included re-metalling all bearings, re-grinding the journals and micro-finishing the cross-heads. The approximate direct cost was £250,000 plus loss of hire and wages during the two month delay.

All systems were tested and found to be sterile. The ship sailed and a routine inspection after eight weeks revealed the onset of slight rusting of No. 3 centre crankpin journal. Further inspection revealed a similar condition at No. 5 main bearing and No. 1 side crosshead. On the assumption that a further microbial attack was developing, 0.5% biocide 'G' was added to coolant and lubricant and the ship was diverted for attention. During the voyage a number of water leakages had occurred leading to a water content of the oil of 2.9%. This has been resolved by centrifuging although some temporary emulsification had taken place. Make-up oil (1585 gallons) added during the voyage was treated with 0.1% of biocide 'O'.

A complete engine inspection now took place. The crankcase was clean with no sludge; the white metal appeared normal; the crankcase journals showed bands of rust up to 4" wide near the oil supply hole; the main journal showed small rust patches and the crosshead mainly staining with some etching. Some rust patches had picked up traces of white metal.

Remedial work was confined to removing rust and staining with wire-wool, emery tape and metal polish. Care was taken to preserve the micro-finish on crosshead pins. A modification was made to engine ventilation, and tank top, drain tank and coolers were tested for leakage. A number of 'Easicult' tests were conducted on board and all systems found to be sterile. Further microbiological tests were conducted locally and in the U.K. and it was concluded that there was no significant infection at that time.

There were now two theories to explain the new casualty:-

(1) A renewed microbial attack. If this had occurred it had been effectively concealed by the large addition of biocide 'G' during the latter part of the voyage.

(2) An incompatibility of the biocide with the oil. There was some evidence to support this.

It was decided to continue anti-microbial measures but this time 0.2% biocide 'V' was added to 700 gallons of oil on the basis that it was freely oil soluble, whereas biocide 'G' was very sparingly oil soluble.

This was circulated throughout the system, hosing down in the crankcase at the same time, for about 24 hrs. The oil was pumped to the renovating tank and then disposed of in bunkers.

The coolant was dumped and replenished with bichromate and 0.2% biocide 'V'. The concentration of biocide was maintained in the coolant during subsequent make-up. The new oil charge now added was of a different type with a smaller additive content, but from the same supplier. 0.2% biocide 'V' was added. During engine trials the pungent fumes from the biocide made it impossible to enter the crankcase unless a smoke helmet was worn. After successful engine trials the ship resumed normal trading. No further biocide additions have been made to the lubricant. 'Easicult' tests conducted weekly have given negative results for bacteria.

A light golden film has been observed on the journals but has been attributed to the change of oil type.

Conclusions

A massive bacterial infection caused pronounced oil emulsification and sludging; eventually severe corrosion and failure to lubricate resulted in a complete engine failure. Bacteria were successfully eliminated from the oil and coolant. On the next voyage a further incident occurred which could have been a bacterial attack, as it is known that biocide 'G' is easily leached from oil and hence leaves it unprotected. However such tests for bacteria as were conducted were negative. There could also have been interference by biocide 'G' with oil characteristics. After further repairs and use of biocide 'V' there has been no recurrence of the problem. Biocide 'V' has objectionable pungent fumes which disperse after a period of several days.

CASE 2

The ship was six years old at the time of the casualty. The engine had water cooled upper pistons and oil cooled lower pistons. It had operated normally before the incident with no record of excessive corrosion or lubricant malfunction.

A routine survey was conducted and revealed corrosion. A complete brown film was observed on the journals except for a bright ring in the way of the oil grooves. The ahead guide face showed brown staining, whilst slight pitting was observed on the astern guide bar faces.

An oil sample taken before the survey was reported as fit for further service.

Samples were submitted for laboratory examination. They revealed 92,000 bacteria per ml in an oil sample drawn from the sump and 3,040,000 bacteria per ml in the jacket coolant sample.

As it was not possible to carry out the necessary work on the engine at the port where the ship berthed some temporary steps were taken. A proprietory cleaner was obtained from the lubricant supplier, added to the oil and this was then circulated for 24 hours at 75°C, transferred to the renovating tank for further heating and then discharged to bunkers. The brown encrustation was removed by the cleaning process and revealed that the journal surface below it had a pumice stone appearance and white metal had been picked up. A new charge of the original type of lubricant was delivered and the ship moved port for repairs.

A biocide (0.05% of Biocide 'M') was added to the oil and this was circulated for 48 hours at 50°C. The charge was pumped to the renovating tank, the temperature raised to 85°C and finally discharged to bunkers. There was an unpleasant pungent smell from the biocide in the hot oil but this was not strong enough to make working conditions difficult and it dispersed after a few days.

The jacket and fuel valve coolant were treated with 0.15% Biocide 'P', circulated for 48 hours at 65°C and dumped. All coolant tanks and the renovating tank were manually cleaned. All coolant and oil pipes in a horizontal plane were dismantled, cleaned and pickled in acid.

At the repair yard very severe corrosion of the water cooled fuel injection valves was noted. At this time the coolant system had been treated and was sterile but the valves were sent away for laboratory examination. This confirmed that this was a severe form of sulphide corrosion caused by bacteria.

The engine repairs included extensive remetalling, replacement of bearing shells and machining and polishing of crosshead pins and main journals. Ten of the twelve lubricating oil telescopic pipes were replaced, and all twelve guide bushes were re-metalled and machined.

After re-erection of the main engine the coolant and lubricating oil systems were refilled and treated with biocides as before. After 48 hours these charges were discarded and tanks cleaned.

During biocide treatment, samples were taken from four points in the oil system and all were sterile. Samples taken during the biocide treatment of the cooling systems were also sterile.

After treatment the ship was re-charged with a different type of oil from the same supplier. After engine trials the lubricant was again tested and was sterile. The ship is now operating normally.

Comment

This is an interesting case because of

(a) The type of corrosion on the journals

(b) The successful treatment of the lubricant system with a very low concentration of a new biocide (ten times less than some which have been used previously).

(c) The very severe corrosion caused by microbial growth in the fuel valve coolant.

DISCUSSION

MR R O KEYWORTH said that "Bugs - why now ?" was a question frequently asked and he noted that the author had refrained from stating the obvious in compiling his list of reasons under the heading "Escalation of Microbial Ship Incidents". The obvious was that ten years ago hardly anyone in the marine industry had considered microbes as a possible or contributory cause of crankshaft corrosion. Today, thanks to the efforts of the author and others (King/McKenzie) this was no longer so - in fact some might say the subject had been over-exposed.

There had been a number of recent cases where early determination of bacteria had started a full-scale investigation. This had resulted in the finding of incipient corrosion which was eliminated together with the bacteria before expensive regrinding repairs became necessary. This in itself would seem to more than justify a second airing of the subject before this Institute.

Referring again to the paper under the same sub-heading, item 2 would indicate that sophisticated oils should be avoided because certain of their additives feed microbes. As bacteria could also attack straight mineral oils, and as straight mineral oils offered little or no protection against corrosion during such attacks, his company's experience was contrary to this. Despite the possibility of certain additives being appetising to microbes, the use of the so-called sophisticated oils - by which was inferred alkaline/detergent anti-corrosion oils - had frequently protected against crankcase corrosion even when infested by bugs. He believed that there had been a few medium speed engine infestations recorded but no corrosion had been reported. Medium speed engines invariably used sophisticated oils.

Turning to item 3 of the same subject: It was true that of necessity many diesel engine lubricants had changed from naphthenic to paraffinic base stocks over the last few years. The paper stated that paraffinic might be more prone to microbial attack. Would the author advise if this was based on laboratory test work and state what had been his experience in investigating shipboard problems with oil using these two types of base stocks.

Under the subject "On-board Indications of Microbial Attack on Marine Lubricants"; although Friday 13th had passed us by, might he suggest item number 14: Removal or discoloration of crankcase paint. This had been found to be a good indicator of the presence of microbial attack, particularly if the paint was slimy to the touch and being removed from the top surface inwards rather than peeling off. Mr Keyworth then presented a slide showing the oil film thickness on a crosshead bearing of less than 5 microns at full load. Item 15 could well be the anti-pollution laws which had resulted in bilges overflowing tank tops and causing bilge water ingress to leak into sump tanks. This had been the cause of at least three cases of crankshaft corrosion following bacterial infestation.

As marine engineers, there was much that could be done to counteract not only bacterial infestation but also in the prevention of crankcase corrosion.

Careful thought in the layout of piping in the design of lubricating oil systems to ensure sludges, free water, dirt etc could not accumulate in double bottom sumps. The provision of adequate sized purifiers and renovating tanks equipped with sufficient heating facilities to meet at least the temperatures quoted in the paper, would be significant factors in the war against bacterial attack and corrosion.

Simple precautions such as blanking off the engine sump to the double bottom tanks during overhaul and maintenance work, by preventing water and dirt from entering the double bottom tanks, would also help to prevent the provision of provender for Pseudomonas creatures.

Some older members present may remember the good old days when diesel engines ran happily on up to 25% sea water and oil, so why all the fuss today ?

Figure 7 showed a crosshead bearing load diagram of a modern highly rated engine.

Increased bearing loads meant that often only microscopically thin oil films were available to prevent metal to metal contact. In this case 4 microns or less. Therefore, the slightest deterioration of the microfinished pin surface by rust or corrosion would result in metal to metal contact and ultimate failure.

Similarly in the design of the diesel engine itself, the prevention of water ingress into the engine required even more consideration than given to the problem previously. Future lubricating oil systems would become proportionately smaller in size, therefore the same amounts of water ingress would represent a greater percentage of water contamination of the lubricating oil.

Around 53% of all diesel engine system oil samples analysed by his company showed enough water to merit comment on the lubrication service report and this figure had remained fairly steady over the last few years, any improvements which could reduce water contamination must be beneficial.

From the oil company's point of view the problems arising from corrosion following the ingress of strong acids, fresh or sea-water had been tackled and lubricants were available offering protection against damage from these undesirable sources.

As regards microbial attack, early oil company work had been done in the field of "cutting oils" where a very different technology was applied when compared with marine diesel engine system oils. However, in the last 18 months much work had been undertaken by microbiologists on behalf of oil companies in evaluating base stock oils, and additives relating to diesel engine system lubricants.

This work together with experiences freely exchanged in a number of private and Institute sponsored meetings held in the Institute's Conference Room would certainly go a long way to influencing future formulations for marine system diesel oils. After consideration of all the factors involved the careful selection and blending of the right oils and additives would provide the long term prevention to bacterial infestation and the permanent use of biocides in the lubricating oil was not a long term answer.

Finally to those who rightly insisted on short term protection now, he could only reiterate what he had said in the discussion following the King/McKenzie paper in November 1976 - "the regular testing of system oils for bacteria with test kits together with the use of an alkaline detergent anti-corrosion oil, is the best method of preventing expensive corrosion damage which might otherwise be caused by microbiological infestation".

COMMANDER M J N NEEVES RN wished to report the experience of the Ministry of Defence in operating warships. The Ministry had no low speed diesel engines but did have a large population of high speed engines in the 0.5 to 1.5 MW power range which were used as propulsion engines in small ships, and as generator engines in larger ones. When running on power the coolant and lubricating oil temperatures were of the order of 180^{O}F and should therefore be self sterilising. However a large number of these engines were standby units and

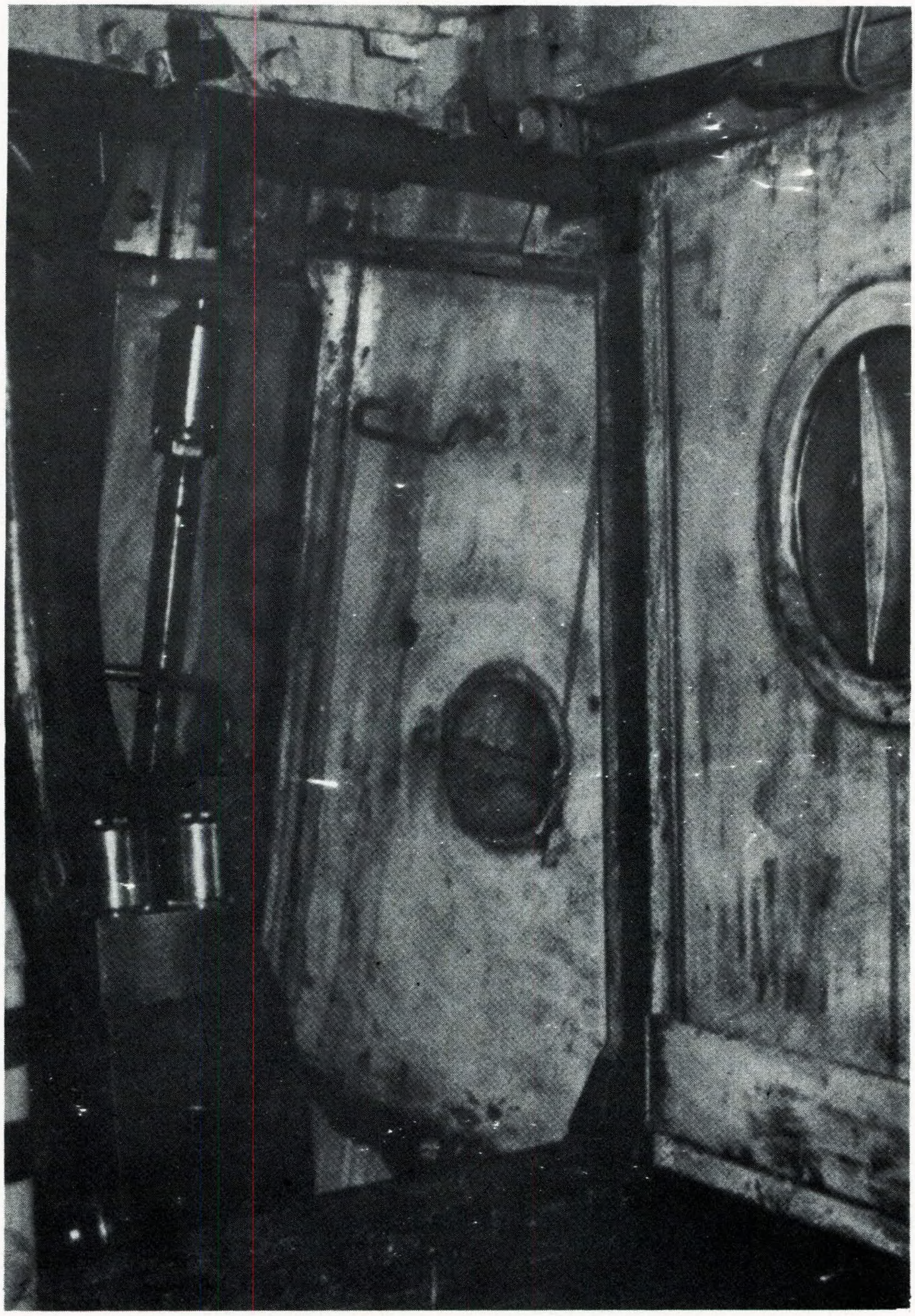

Figure 7 - Paint discolouration and removal in crankcase

were subject to long periods of shut down with the possibility of contamination through the breather arrangement. In spite of this no cases of microbiological contamination of lubricating oil or coolant systems had been reported. This might be due to a normal practice of running standby engines at least once a week for a short period and rigorous weekly tests of lubricating oil water content which the Ministry's regulations limited to less than 1%. Could the author comment on whether they were likely to be at risk in these standby engines ?

Where there had been instances of microbiological contamination this had been in the fuel systems of some of the gas turbine and diesel engined ships, which were run on low sulphur distillate fuel, and particularly in those ships which had water displaced fuel systems. The problem contaminant at sea had been the fungus "cladisporium resinae" although there had been one reported case of contamination by sulphur reducing bacteria in a sullage tank ashore. There were instances of filter plugging but, generally, the problem had been contained, even in contaminated systems, where the fuel had been centrifuged before being presented for fine filtration. The Commander was a little surprised therefore that in his paper the author had doubted the efficacy of the centrifuging process in dealing with these materials.

In Figure 1 the author had shown contaminant at the oil/water interface and this was also, in the Ministry's experience, the location of the growth in fuel tanks. It was not believed that the "bugs" were swimming to keep down there and Commander Neeves reckoned that they, therefore, had a density somewhere between that of water and fuel. If this was so they should be susceptible to centrifuging. The Ministry were starting some trials to test this hypothesis at an MOD establishment at Portsmouth but were encouraged by reports from a ship suffering from contamination which, when cleaning her centrifuge, described that they were getting out material that looked like a "moist cowpat".

Could the author comment on this view of centrifuging ? Would he also comment on whether the problem of growth in fuel was likely to be more amenable to centrifuge treatment than that in lubricating oil ?

MR BRYAN TAYLOR said that although Mr Hill had stressed the seriousness of the damage to bearing surfaces of slow-speed engines which could result from microbial degradation of the lubricating oil, he felt that the extent of this problem must not be exaggerated. So far as he was aware there had been relatively few cases where damage to the engine could be attributed without any doubt to the effects of bacteria in the oil and he, therefore, asked the author whether he could give any statistics on the subject.

Mr Taylor had only one criticism of the paper to make, which was that perhaps more emphasis could have been placed on methods of prevention of infection. Maintenance of a high standard of cleanliness and close attention to the prevention of leakage of water from the bilges into the lubricating oil drain tank were obviously of paramount importance. Another factor not mentioned by the author was that distilled water should be used for the cooling water system. He had mentioned, however, that modern operating conditions with reduced crews, unmanned machinery spaces and frequent changes in engine room staff, made a high standard of housekeeping more difficult.

On the question of cooling water treatment it was recommended that potassium bichromate should be used on engines where its use was not precluded because of the risk of contamination of water produced from a fresh water generator. The use of soluble oils was not recommended, not because of the risk of bacterial infection, as mentioned by the author, but because experience had shown that rubber sealing rings could be affected. In engines with water-cooled pistons it was very difficult to avoid leakage of cooling water into the crankcase but it was interesting to note that in the first case history given in the appendix to the paper, the piston cooling water was treated with potassium bichromate and the water was not infected with bacteria. It seemed reasonable to conclude from this that the use of oils containing anti-rust and oxidation additives was a factor in the escalation of incidents such as described in the paper. It was noteworthy that in both case histories the oil was ultimately changed for a different type, after which there was no further trouble.

Nevertheless it seemed that in engines with water-cooled pistons, attention to the cooling water treatment was very important. When nitrite-based corrosion inhibitors must be used, would it not be possible to add a suitable biocide to inhibit permanently the growth of bacteria in the water ? On the question of biocides, the author mentioned at least eight different types in his paper, most of which appeared to have some disadvantage; was it possible to give some guidance on the best type to use in any given circumstances.

One did not need to be a detective to deduce that both the engines mentioned in the case histories were of Doxford design but it should be emphasized, as the author had pointed out, that the problem was not restricted to any particular type of engine or any particular oil. There were several hundred engines in service of the type referred to in Case 1 and as far as the engine builders were aware it was the only serious incident which had occurred in this type of engine.

Amongst the probable reasons for the escalation of microbial degradation of lubricating oil listed by the author was reduced engine temperatures resulting from running at reduced power. There was no reason for operating with lower cooling water, and lubricating oil temperatures under these conditions and the advantages to be gained from high cooling water temperatures should be stressed. Not only was the growth of bacteria reduced but jacket water outlet temperatures of at least 170°F were favourable from the point of view of liner wear.

It was also pleasing to note that the author stressed the importance of correct operation of the purifier. Many of the oils now used were not suitable for water-washing but it was important that the oil entering the purifier was maintained at a temperature of at least 160°F. It was also important that the throughput of the purifier was kept down to at least half its rated capacity.

MR J E CHURCH FIMarE said that one of his company's ships was among those which Mr Hill had mentioned as having a moderate infection in the crankcase lubricating oil, fortunately without damage and, hopefully, now cleaned and cured, thanks to the advice and help given by Mr Hill.

It was subsequently discovered that the piston cooling water system was even more heavily infected, and this had complicated matters because the engine concerned had telescopic piston cooling pipes, the packings of which require a soluble oil type of inhibitor for lubrication, which Mr Hill had advised should not be used, as it would provide food on which these microbes flourished. It also prevented the use of a chromate based water treatment, which could kill them. On the other hand this ship did not use jacket water for potable fresh water generation, so that chromate treatment had now been put into the jacket coolant, thereby ensuring that any water leakage from that system would act as a deterent.

This did leave the problem of the piston water coolant, and the urgent need remained for a suitable anti-bug additive with lubricating properties for RD type engines with Sulzer approval. In the meantime, untreated distilled water was being used, and changed every month until a solution was discovered.

Having listened with interest to all Mr Hill had said, it seemed that one must now take this matter more seriously and it was now suggested that the following precautions to all ships should be put into effect as standing instructions:

(1) Each time a ship reached port and as soon as engines were shut-down, the crankcase lubricating oil should be transferred to the renovating tank; steam heated and held at 180°F if possible, for the entire time the ship was in port. At the same time circulating through the purifier and purifier heater to ensure uniform heat throughout and the complete extraction of water. In this way the oil would never cool down in port, and so not pass through a temperature favourable to the multiplication of bacteria.

(2) The changing of all ships to chromate based cooling water treatment. Those that had fresh water generators for potable water, which were heated by engine jacket water, to be disconnected from the engine system and be connected to exhaust gas waste heat steam. Scaling-up of steam coils to be prevented by using a proprietary brand of evaporator treatment chemical by drip feed into the sea water feed to the generator. The expense of these extra precautions would be slight compared with the cost of crankshaft damage as described in Mr Hill's paper. The lubricating oil condition could only become better than ever before as a result.

Would Mr Hill comment upon these suggested precautions ?

The question had arisen as to why this problem of microbial degradation had come to the fore recently and Mr Hill had offered some explanations. To these might be added that of the ships sewage, which now had to be retained and treated on board, which meant that all recent ships had some form of sewage tank and treatment plant in the engine room. Mr Hill had told of the tremendous count of bugs in this material and others had stated that dirty bilge water had been established as the cause of some cases of crankcase infection. Would Mr Hill advise if the bugs from sewage were the same as those found in lubricating oil ? If this was the case, steps should be taken to isolate sewage tanks and to prevent any drainage from them reaching the engine room bilges during repairs etc.

<u>CORRESPONDENCE</u>

MR M R QUIRK wrote that the following account covered the experience of his company over the last two years, with fuel suffering from microbial contamination. Whilst acknowledging that the subject matter of this paper was the "Microbial Degradation of Marine <u>Lubricants</u>", the following contribution was submitted, since the problems experienced with fuels, and remedial actions taken, were believed to be similar to those with lubricating oils.

Two years ago, in March 1976, while undergoing full power trials at sea in a modern warship, serious blockages were experienced in the fuel system. The protecting filters of the engines, Olympus Gas Turbines, rapidly blocked up with a slimy glutinous substance, the cause and seriousness of which was not fully understood at the time. In spite of continually changing and cleaning of the filters, little improvement was found and it was realized that a critical situation existed. It was also noticed that large quantities of water appeared to be passing the coalescer filters.

Trials had to be abandoned and the ship returned to Southampton. Samples of fuel were sent for laboratory examination and the results showed very heavy contamination with the fungus "cladosporium resinae". The cleaning up of the fuel system involved the dumping of approximately 200 tonnes of contaminated diesel and the manual cleaning of all the bunker tanks and fuel lines at a very considerable cost.

It was clearly necessary that the company, as the shipbuilder, needed to take positive action to ensure that there would be no re-occurrence of this problem. In consultation with University College, Cardiff, a programme of regular sampling, analysis and dosing of the fuel was set up. The company had recently opened a small laboratory for this purpose, within our fitting-out complex, to enable this work to be carried out on site and within hours of samples being taken. Ships under construction were sampled and analysed monthly in the summer and bi-monthly in the winter, and from the results tanks were dosed with the appropriate biocide. This technique had proved entirely successful and in spite of contamination being detected in certain deliveries of fuel, this had not caused any further disruption.

It was becoming increasingly evident that in the future, the contamination of fuel, as with the contamination of lubricating oil, was a hazard with which all would have to live. The shipbuilder had a responsibility to design into the vessel, means of sampling and dosing, the latter clearly being best carried out as fuel was embarked.

The high flow rates, when a ship was replenishing at sea, could lead to special problems in this respect and automatic metering equipment together with advance knowledge of the fuel state was necessary. Adequate facilities should also be provided on board, or at least at base, to analyse and advise on the strengths of biocides necessary to correct any contamination detected.

AUTHOR'S REPLY

MR HILL agreed with Mr Keyworth that most microbial problems had arisen with moderately additive oils rather than modern highly sophisticated additive oils. Although the latter could become infected, the consequences of infection appear to be diminished. As well as the high alkalinity being able to absorb acidic corrosion products, the detergency appeared to tie the water up into small droplets which were not necessarily available to the micro-organisms.

He regretted that he was unable to give details of the resistance of paraffinic and naphthenic oils without referring to specific products; but he was firmly of the opinion that this was an important factor.

Mr Hill appreciated Mr Keyworth's other comments which had added much from the practical engineers viewpoint.

To Commander Neeve, Mr Hill said that he would believe that the absence of microbial problems in the high speed engines referred to could be attributed to the self-sterilizing running temperatures, the low water content of the lubricant, and the type of oil in use.

The density of bacteria was very slightly greater than water and therefore substantially greater than distillate fuel and hot engine lubricant. It was easy to demonstrate that they were largely separated out at the main engine oil centrifuge, although shipboard tests indicated that effectiveness was less than 100%. He would anticipate that similar success would be achieved with fuel centrifuges, particularly if the density difference and viscosity were more advantageous.

Mr Hill thanked Mr Taylor for his helpful comments and agreed with all of them. He would however like to add that shipboard distilled water could be of a very variable character and was often highly infected itself. One problem of adding biocides routinely to coolants was that they might be considered too toxic for use when the coolant was used in the fresh water evaporator.

He must also confess that having looked at slow-running conditions more closely he did not now think that this was an important factor; indeed it could give rise to higher temperatures if the coolers were operated automatically.

Mr Hill was very impressed by the measures which Mr Church had proposed for his ships and one could anticipate that the likelihood of an infection arising had been very substantially reduced.

However Mr Hill took his point on on-board sewage, and did not know of any experimental work which was relevant. It would certainly be interesting to investigate this.

Mr Hill much appreciated Mr Quirk's comments and were a very clear indication that once recognized, microbial problems could be adequately contained.

Paper read at the Institute on Tuesday 6 December 1977

A REALISTIC ADVANCED STEAM CYCLE FOR SHIPS

K. G. Grossmann, C.Eng., F.I.Mar.E., STG *

This paper describes a marine steam cycle for high steam pressure and low fuel consumption. To achieve a good fuel economy even at partial load, the system is designed to operate from maximum manoeuvring rate to "full away" with a floating boiler pressure p_T. To increase the safety margins of the system and to simplify the feed water control the water level set point is floating proportional to the steam velocity in the superheater steam pipe. The theories of the water level and the boiler pressure behaviour are discussed and evaluated for the control system. Full scale tests at sea trials and dock trials and test on the small steam turbine plant of the Institut für Schiffstechnik were run, to prove the theories and the control philosophy.

INTRODUCTION

Due to the sharp increase in fuel prices, steam turbine plants for ships are in discussion again because of their rather high specific fuel consumption. All over the world new steam systems with higher steam pressure, slightly higher steam temperatures and with reheat are put forward.

This paper deals with a steam plant, which is designed to be self controlling in the "full away" conditions, and is very simple and reliable. To get a low fuel consumption, the plant is run from manoeuvre conditions (CMR) to maximum continuous rating (MCR) with floating boiler pressure p_T, full open turbine control valves, all bleed points in operation and the main feed pump and the main generator attached to the main gear. By this way throttle losses are avoided when steaming at less than full power.

For safe operation, the turbine control system and the boiler control systems are interconnected. During manoeuvres, the latter ones are controlled by the turbine control system.

1. Selection of the reheat system

Two principle modes of operation can be defined:
Mode 1: Manoeuvring and port conditions.
Mode 2: Full away operation.
During mode 1 only 40% of the MCR steam demand is needed to cover the manoeuvring conditions in the ahead speed area.(This is for a ship with rather low velocity, e.g. tanker. For a fast container ship, the manoeuvre load may be even smaller.) The steam demand for full astern rises to 70% of MCR

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on account of the low efficiency of the astern turbine, but this power is usually needed only at crash-stops.
For mode 2 the steam demand rises from 40% up to 100%.
Three conditions can be postulated for the steam cycle.

1. The propulsion system must be as simple as possible, to provide easy service and to demand little maintenance. The elements of the propulsion system must be of proven design and of highest quality.
2. When in mode 1, the plant should be able to perform rapid load changes without any risk to the safety of the system. (This implies an additional requirement for the boiler. The natural circulation must always be guaranteed even with load changes.) Economic considerations - such as low fuel consumption - are not of major importance, because the time, in which the ship will run in mode 1 is comparatively short and the boiler load is usually low to medium.
3. For the whole full away range - mode 2 - the lowest possible fuel consumption should be achieved. With the exception of the emergency stop, all load changes will be rather slow.

If the boiler pressure is raised above 80 bar, some kind of reheat system will be needed to keep the moisture at the exit of the low pressure turbine within acceptable limits. The choice is between flue gas reheat and bled steam reheat.

With the same live steam conditions, flue gas reheat offers a gain in fuel consumption of

$$\Delta \dot{m}_B = 6 - 7 \text{ g/kWh}$$

when the steam is reheated up to the full superheater temperature. With a lower reheat temperature the gain in fuel consumption will rapidly go to zero.

The boiler for a steam cycle with full flue gas reheat has either two parallel secondary passes for the flue gas - one for the reheater and the other one for the primary superheater - or an independently fired reheater. Fig. 1 shows the arrangement of the reheater in the "Esso Norway" [1]. When in mode 1 the reheater is by-passed which means, that the distribution of the flue gases and the heat transfer to the boiler proper is no longer symmetrical. This of course has an influence on the natural circulation.

With the independently fired reheater, which is always connected to the main boiler, there is a change of the flue gas flow downstream of the reheater. This also has an influence on the heat transfer of the whole boiler.

All flue gas reheat systems have a change of the steam temperature at the inlet of the medium pressure turbine. When switching from mode 1 to mode 2, the reheater is activated and the medium pressure-turbine has to be heated gradually to full reheat temperature. When going from "full away" conditions to manoeuvre conditions, the reheater is closed andthe MP-turbine has to be cooled down gradually again. Unfortunately, these load changes may happen very fast, when a crash-stop is initiated or the turbine trip comes. Especially in a crash-stop case the boiler state is changed from the full load condition with reheat rather suddenly to 75% load without reheat.

In comparison bled steam reheat has a higher fuel consumption than flue gas reheat. It utilizes practically only the pressure - and temperature increase. But the boiler for this system is a normal one with a single secondary flue gas pass.

The reheater is heated by bled steam from the first bled point. It is placed in the cross over between the HP-turbine and the LP-turbine. The cross over steam is heated up to below the condensation temperature t_{KA} of the first bled point. This means, that the reheat temperature is always lower than the saturation temperature t_s of the live steam. For this reason the material for the LP-turbine does not need to be of high heat resistance quality. The reheat temperature also decreases with decreasing load, due to the reduction of the pressure at the bled points - and vice versa. When the steam temperature at the reheater entrance is the same as the condensation temperature of the first bled point, the reheater closes down automatically, as the heating steam is no longer condensed.

With a high steam pressure the evaporation heat becomes rather small whereas the economiser heat and the superheater heat increase. This means an equivalent change in the size of the respective heating surfaces. The trends can be seen in table 1. It shows, that the ratio of the heat for the superheater to the evaporation heat increases to nearly twice the value at p_K = 65 bar, the feed water heat to evaporation heat increases even to 3.2 when

Table 1

System No.			1	2	3	4	5
1	p_K	bar	65	65	100	130	150
2	t_{UE}	°C	530	530	530	530	530
3	t_{SP}	°C	138	220	236	236	236
4	r	kJ/kg	1544	1544	1276	1113	985
5	Δh_{UE}	kJ/kg	710	710	745	764	796
6	$\Delta h_{UE}/r$	-	0.46	0.46	0.58	0.69	0.81
7	Δh_{SP}	kJ/kg	630	290	444	520	587
8	$\Delta h_{SP}/r$	-	0.41	0.19	0.35	0.47	0.61
9	Δh_{ZE}	kJ/kg			422	440	441
10	$\Delta h_{ZE}/r$				0.33	0.40	0.45
11	$(\Delta h_{UE}+\Delta h_{SP})/r$	-	0.87	0.65	0.93	1.16	1.42
12	$(\Delta h_{UE}+\Delta h_{SP}+\Delta h_{ZE})/r$				1.26	1.56	1.87

the boiler pressure is increased to p_K = 150 bar.

From a design point of view, mainly space reasons seem to define the pressure limit of a marine boiler with flue gas reheat to p_K = 130 bar. At this pressure, the energy transferred to the superheater and the economiser is already 1.16 times the evaporation energy or 1.33 times the ratio found with the common economiser boiler. To these big heating surfaces we have to add the big reheater volume respectively surface.

The boiler pressure for the system with bled steam reheat can be raised up to p_K = 150 bar. Its volume will still be less than that of the flue gas reheat boiler.

As the simplicity of the plant is considered one of the most important design points, a bled steam reheat cycle with a boiler pressure of p_K = 150 bar was preferred to a flue gas reheat cycle with 130 bar boiler pressure. To achieve a low fuel consumption when in mode 2 the boiler pressure will float proportional to the turbine load from

$$p_K = 65 \text{ bar at } P_T = 0{,}4\, P_{TO} \quad \text{(CMR)}$$

to

$$p_K = 150 \text{ bar at } P_T = P_{TO} \quad \text{(MCR)}$$ [2]

Table 2 gives the figures for the most important values for the natural circulation, the mean water velocity at the entrance into the evaporation tubes and the circulation number $u = \dot{m}_Z/\dot{m}_D$

* space

Table 2 Calculation of the natural circulation

p_T/p_{T0}	p_T	w_0	u	$\dot{m}_Z/\dot{m}_{Z0}$
-	bar	m/s	-	-
1	155	0,734	12,2	1
0,8	125	0,738	16,7	1,09
0,6	95	0,728	23,8	1,17
0,4	65	0,696	36,6	1,20
0,7	65	0,881	23,2	1,32
0,8	155	0,684	14,3	0,94
0,6	155	0,622	17,3	0,86
0,4	155	0,541	22,6	0,74

With a floating boiler pressure the water velocity w_0 remains practically constant, which means that the circulating water volume $\dot{V}_Z$ stays constant too and the circulating water mass flow $\dot{m}_Z$ is even increasing. The cooling of the evaporation tubes is becoming better with decreasing boiler pressure.

(This is a very important fact. It means, that if the boilers on tankships - or any other steam ships - are run with a lower boiler pressure during slow steaming, the circulation will become better and not - as is generally believed - worse. This is caused by the increase of the steam volume V_D in the boiler. The specific gravity of the mixture of steam and water in the evaporation tubes is lowered by the bigger steam volume. This improves the circulation.

The lower boiler pressure would also improve the fuel consumption by reducing throttling losses.)

Another advantage of the floating boiler pressure is the fact, that all rapid load changes happen at the "normal" boiler pressure of p_K = 65 bar with the higher circulation. All big steam driven auxiliary turbines - especially the turbines of the cargo pumps, are supplied with steam of this pressure even under full away conditions. The same happens to the astern turbine.

2. Selection of the steam cycle

2.1 Main feed pumps

The floating boiler pressure as a main design parameter allows the feed pump to be attached to the main gear during the whole "full away" condition (mode 2) Fig. 2. With a reduction in revolutions from 100% to 73%,which represents a load reduction from 100% to 40%, the decrease of the pump's head is still less than the reduction of the boiler pressure.

Due to the higher boiler pressure, the power necessary for the feed pump is P_{SP} = 508 kW $\hat{=}$ 2% of the main drive power. The attached feed pump causes an additional steam flow of $\dot{m}_{DSP}$ = 2985 kg/h into the main turbine.

An independent feed pump, driven by its own turbine with an efficiency of η_{iSP} = 0.7 and a back pressure of p_{NSP} = 3.0 bar would need

$$\dot{m}_{DSP} = 4830 \text{ kg/h}$$

steam. This means 4.8% of the overall steam demand.

With an independent feed pump turbine exhausting into the deaerator,the deaerator pressure has to be as low as possible in mode 1 and in mode 2 to cut down the steam demand of the feed pump. This means, that the make up steam from the bled point has to be throttled at MCR, so that it still can serve the deaerator at manouevre conditions.

The independent feed pump would increase the fuel consumption by

$$\Delta \dot{m}_{BSPEZ} = 2.0 \text{ g/kWh} \hat{=} 0.73\%$$

With a fuel price of DM 175,-- ($ 73) perton and 280 days of full power service per year this means

$$\Delta K = 58\,800 \text{ DM/a } (24\,530 \text{ \$/a}$$

2.2 Generator

The electric load for this steam plant is estimated to be P_E = 750 kW when in mode 2. This includes electric driven boiler fans. The efficiency of the rather small auxiliary turbines with the high boiler pressure would be very poor. There would be also a control problem, if the turbo generator had to work at p_K = 150 bar as well as at 60 bar. For all the auxiliary turbines - including the turbines for the cargo pumps and the astern turbine - a reduced pressure steam line with p_R = 60 bar is provided. This simplifies the auxiliary steam system. Still the turbo generator would run with an efficiency $\eta_{iTG} \approx$ 0.65 - 0.7 only.

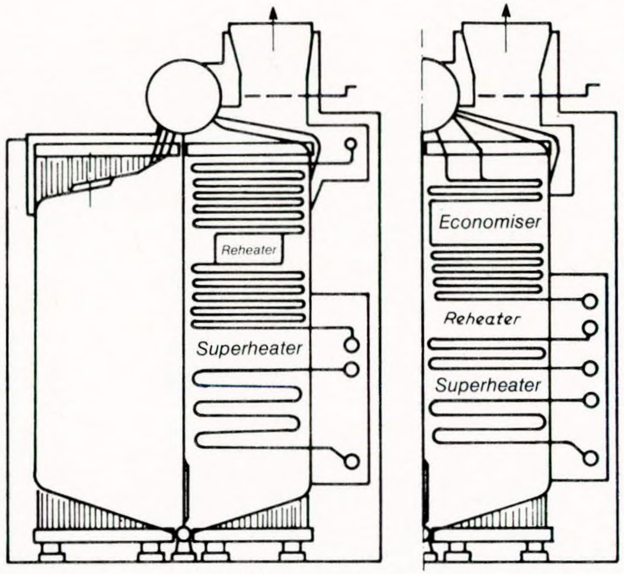

Fig. 1 – Flue gas reheat boiler

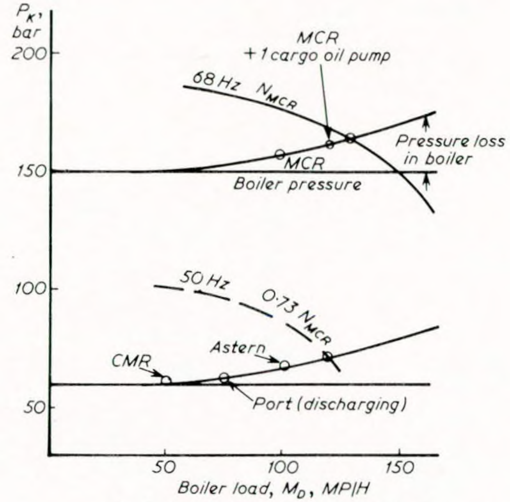

Fig. 2 – Operation of attached feed pump

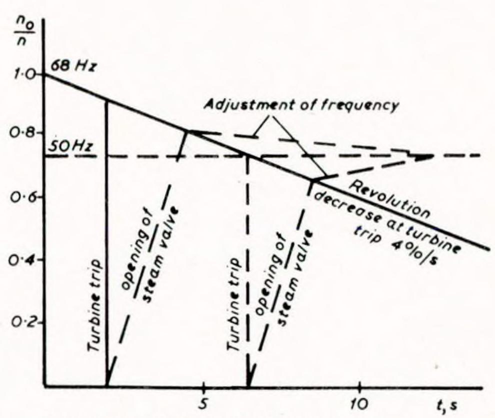

Fig. 3 – Change over operation of main generator

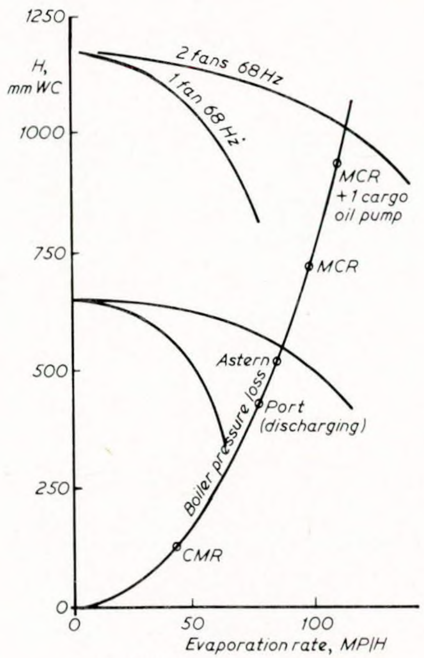

Fig. 4 – Operation of boiler fans with floating frequency

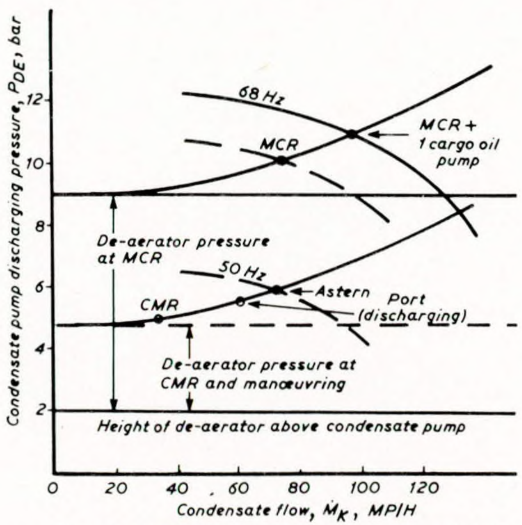

Fig. 5 – Condensate pump operation with floating frequency

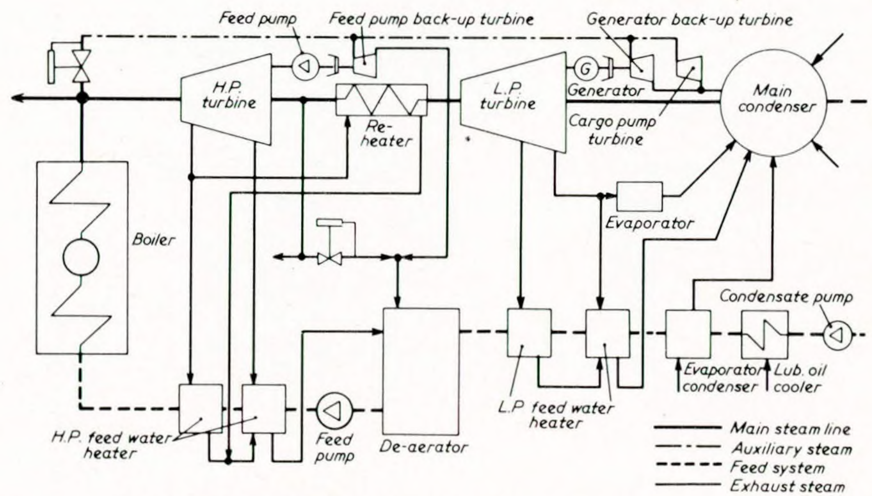

Fig. 6 – Steam cycle with bled steam reheating

When the main generator also is attached to the main gear during mode 2, then the fuel consumption will drop another

$$\Delta \dot{m}_{SPEZ} = 3{,}0 \text{ g/kWh} \mathrel{\hat{=}} 1{,}3\%$$

This would mean a saving of

$$\Delta K = 114\ 600 \text{ DM/a } (47\ 700 \text{ \$/a})$$

There is one big problem with the attached auxiliaries: The switchover from the attached state to the back up turbines under emergency conditions (crash-stop, turbine trip). The critical point is a turbine trip, when the engine is running at CMR (Continuous Manoeuvre Rating), because the frequency is already at the lower limit.

Records of ship board trials showed, that the propeller revolutions go down with a rate of 2,5 RPM/s when the turbine trips from full power at full speed. For security reasons a deceleration rate of 3.5 RPM/s after a trip is assumed. With a fast opening steam valve, activated by the turbine trip itself, the back up turbine for the generator can take over the load from the main gear 1 s after its activating, so this is also the time, when the generator can be disengaged from the main gear. During this time, the frequency will drop

$$\Delta f = 2.8 \text{ Hz}$$

below the limit. This figure is acceptable for the classification societies. After the disengagement of the clutch, the frequency controller takes over and will adjust the generator on f = 50 Hz (Fig.3).

There is no problem. when the turbine trip is above n_{CMR}, because the back up turbine takes over at frequencies higher than f = 50 Hz, so the controller has to decrease the generator revolutions.

When normal manoeuvres are asked for, the steam valve for the back up turbine will be activated as soon as revolutions lower than n_{CMR} are ordered. [3] [4]

2.3 Boiler fans,deaerator and main condensate pump

With the main generator attached to the main gear in mode 2, the frequency and the voltage will float according to the change of the revolutions. When a frequency f = 68 Hz is selected at MCR, then the frequency will drop to f = 50 Hz at CMR. This means that all electric motors will change their speed accordingly to the changes of the propeller revolutions.

For the electrically driven boiler fans, this means practically a speed and guide vane control. The floating frequency yields a stepless speed control between 100% and 73,5%. Fig. 4 shows these conditions.

The floating frequency is giving problems for the main condensate pump. It has to be designed to serve the deaerator even when running at low frequency. When the deaerator pressure is left floating, too, the condensate pump head is sufficient at low load as well as at high load and the throttle losses between bled point and deaerator are practically zero in mode 2. Fig. 5 shows the characteristic curves for the main condensate pump. The deaerator pressure changes from 7 bar at MCR to 2,8 bar at CMR. Consequently the feedwater temperature will fall from 165°C to 131°C. In mode 1 p_{EA} = 2,8 bar and t_{EA} = 131°C will be kept constant by the exhaust steam from the feed pump turbine and make up steam.

The floating of the deaerator pressure when in mode 2, would mean an extra supply of

$$\Delta \dot{m}_{DEA} = 670 \text{ kg/h}$$

steam when going from CMR to MCR (calculated with a deaerator content of 10 000 kg water). This steam can be deducted from the make up steam when going from MCR down to CMR.

2.4 Evaporator

A single stage condensate cooled evaporator with a heat exchanger, preheating the seawater by the brine, will be provided. Its capacity will be

$$\dot{m}_V = 2000 \text{ kg/h at } p_A = 0.9 \text{ bar (MCR)}$$

At CMR, the destillate flow will drop to

$$\dot{m}_V = 800 \text{ kg/h at } p_A = 0.38 \text{ bar.}$$

For the fuel consumption calculation, $\dot{m}_V$ = 1000 kg/h is assumed. The necessary heating steam is $\dot{m}_{DV}$ = 1250 kg/h.

The condensate cooled evaporator lowers the fuel consumption by $\Delta \dot{m}_{BSPEZ}$ = 0.6 g/kWh.

2.5 Boiler design

This boiler will be designed for full turbine power plus 1 cargo pump running at a boiler pressure p_K = 150 bar (drum pressure p_T = 155 bar) and superheater outlet temperature t_K = 530°C. For 40% turbine power at CMR, the boiler will have p_K = 65 bar (drum pressure p_T = 70 bar). Due to the difference in the heat of evaporation, the evaporation rate at p_T = 70 bar is

$$\dot{m}_{D70} = \dot{m}_{D155} \cdot \frac{r_{155}}{r_{70}} = 0,66\ \dot{m}_{D155}$$

The calculation of the heat balances gives the following results.

MCR + 1 cargo pump	$\dot{m}_D$= 115 500 kg/h	$\dot{m}_K$=83000 kg/h
MCR	$\dot{m}_D$= 98 000 kg/h	$\dot{m}_K$=75300 kg/h
CMR	$\dot{m}_D$= 41 200 kg/h	$\dot{m}_K$=34900 kg/h

Maximum power at manoeuvre conditions with all ble d points closed P_{TCMR} = 12500 kW ≙0.50 P_{TMCR}.

Port, discharging	$\dot{m}_D$ = 76500 kg/h	$\dot{m}_K$=67300 kg/h
Full astern	$\dot{m}_D$ = 77000 kg/h	$\dot{m}_K$=68000 kg/h

From these figures, the boiler system - twin marine boilers- will be designed for:

Maximum load

$\dot{m}_{DMAX}$ = 2 x 60000 kg/h, p_T = 155 bar, p_K = 147,5 bar

Normal load (MCR)

$\dot{m}_{DO}$ = 2 x 50 000 kg/h, p_T = 155 bar, p_n = 150 bar

Manoeuvre load (CMR)

$\dot{m}_{DCMR}$ = 2 x 21 000 kg/h, p_T = 70 bar, p_K = 66 bar

Maximum load at manoeuvre pressure

$\dot{m}_{DAST}$ = 2 x 39 000 kg/h, p_T = 70 bar, p_K = 60 bar

The boiler will be designed for an exhaust gas temperature of t_{AG} = 94°C. [5] Compared with the usual temperature t_{AG} = 110°C this cuts down the fuel consumption by

$$\Delta \dot{m}_{BSPEZ} = 1.6 \text{ g/kWh.}$$

2.6 Steam cycle

The steam cycle with 5 bled points - 2 at the H.P.-turbine, 1 at the cross over, 2 at the L.P.-turbine - and the respective feedwater heaters, with condensate cooled evaporator and lub oil cooler - which lowers the fuel consumption by

$$\Delta \dot{m}_{BSPEZ} = 0.9 \text{ g/kWh.}$$

can be seen in Fig. 6. The specific fuel consumption of this cycle will be

$$\dot{m}_{BSPEZ} = 236.7 \text{ g/kWh (174.0 g/PSh)}$$

(If fitted with independent turbines for the feed pump and the generator, the fuel consumption increases to

$$\dot{m}_{BSPEZ} = 242 \text{ g/kWh} \quad \text{(177.8 g/PSh)}$$

The fuel consumption of the same cycle with flue gas reheat up to t_{RE} = 530°C at p_{RE} = 31,2 bar and with p_T = 135 bar, p_K = 130 bar is

$$\dot{m}_{BSPEZ} = 234{,}7 \text{ g/kWh.}$$

This gain is of the same order as that reached with the attached feed pump. The complications are of a much higher order (also the additional installment costs). This comparatively small gain justifies the selection of the bled steam reheat cycle, as the cost savings of

$$\Delta K \approx 58\ 800 \text{ DM/a } (24\ 500 \text{ \$/a})$$

seem to be too small to cover even the additional first cost, nothing to say for covering the disadvantages of the flue gas reheat system.

2.7 Partial load consumption

The calculations for partial load gave the following fuel consumptions

P_T = 17200 kW, p_T = 110 bar, p_K = 105 bar,
w_S = 14,2 kn, $\dot{m}_{BSPEZ}$ = 248 g/KWh (182,6g/PSh)

P_T = 10000 kW, p_T = 70 bar, p_K = 65 bar,
W_S = 11,8 kn, $\dot{m}_{BSPEZ}$ = 275, 4 g/kWh, (202,4g/PSh)

If the cycle with the floating system pressure is compared to a cycle with a fixed boiler pressure and - naturally - an independent driven feed pump, then the fuel consumptions are as follows

P_T = 25 000 kW, p_T = 155 bar

$\dot{m}_{BSPEZ}$ = 238,7 g/kWh (175,4 g/PSh)

P_T = 17 200 kW , p_T = 155 bar

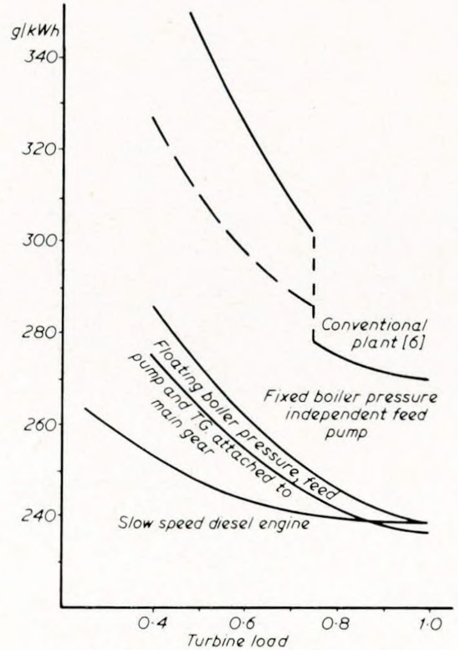

Fig. 7 – Fuel consumption versus turbine load

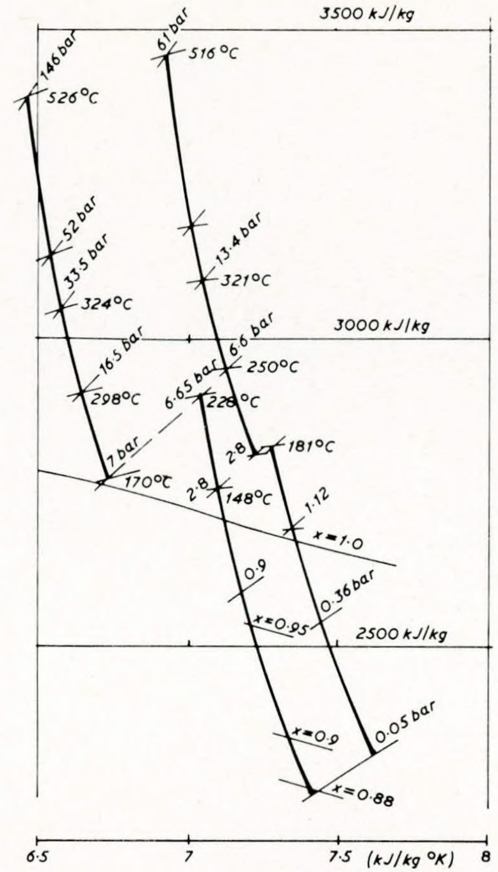

Fig. 8 – Expansion curves of main turbine

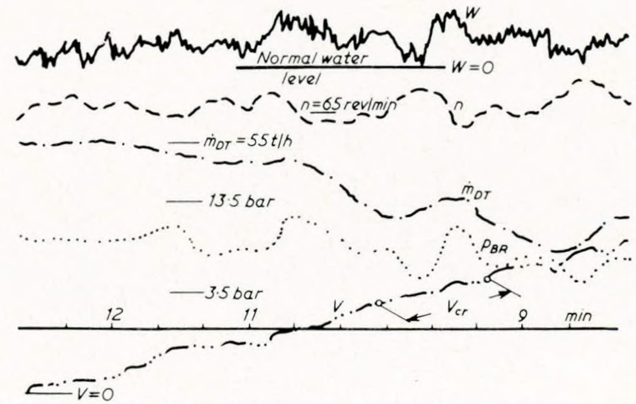

Fig. 9 – Crash stop manœuvre record

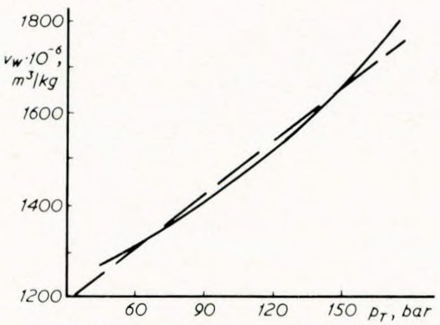

Fig. 10 – Specific steam volume v_W as function of the drum pressure p_T

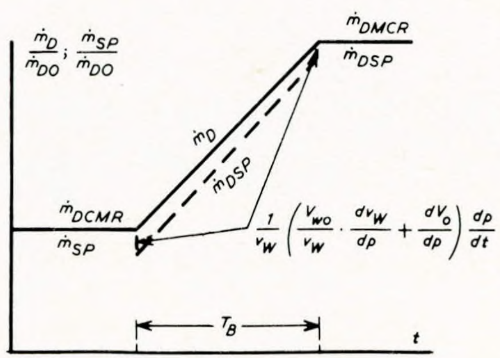

Fig. 11 – Feed water supply and steam demand during load increase from 40% to 100%

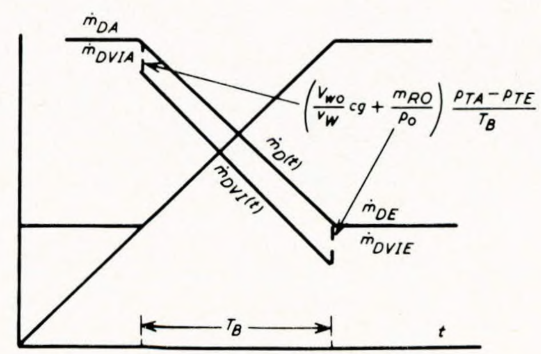

Fig. 12 – Fuel flow m_{DVI} and steam demand $\dot{m}_D$ during a load reduction from 100% to 40%

$\dot{m}_{BSPEZ}$ = 254.8 g/kWh

P_T = 10 000 kW $\quad p_T$ = 155 bar

$\dot{m}_{BSPEZ}$ = 285.5 g/kWh

The influence of the floating pressure can be clearly seen from these figures.

The fuel consumption figures for a slow speed Diesel engine, running on fuel of H_u = 40 600 kJ/kg (9700 kcal/kg) are:

Load	%	100	75	50	25
Consumption	g/kWh	214	215	219	226
Correction for	1 g/kWh lub oil = 5 g/kWh fuel oil				
	g/kWh	219	220	224	231
Correction for higher propeller revolutions (103 RPM → 85 RPM)					
	g/kWh	235.5	236.5	240.8	248.5
Correction for 400 kW generator load.					
$\dot{m}_{BSPEZ}$	g/kWh	239.2	241	248	263.5

The curves behind these figures can be seen in Fig. 7, where for comparison reasons the curve from Rein [6] is also drawn.

In Fig. 8, the expansion curves of the main turbine for MCR and CMR can be seen. At CMR, the reheat is only Δh_{ZE} = 11,0 kJ/kg, whereas it is Δh_{ZE} = 132.6 kJ/kg at full power.

3. Control System

During manoeuvre conditions, the plant should be able to do rapid load changes without any risk, whereas load changes under full away conditions are rather slow. The exception is the emergency stop manoeuvre, but this will be done with a fixed boiler pressure and not a floating one.

The usual control point for the power of the turbine is the revolution of the propeller. This is a very unfavourable value, as the turbine power is proportional to the third power of the propeller revolutions. For the manoeuvre range there are 73% of the revolutions but only 40% of the power. For the full away range with 60% of the power there are only 27% of the revolutions.

It is much easier and safer to go back to the pressure control. When under full away conditions, the control of the steam pressure to the turbine automatically restricts the torque to the main gear, as the turbine control valve does not react to a change in the resistance of the ship due to heavy sea for instance. During the emergency stop manoeuvre, the steam flow to the astern turbine is kept steady, which even a good revolution control system cannot achieve, when the ships speed is between 65% and 45% of the design speed. Fig. 9 shows records of this phase from a trial trip. When rapid load changes around zero load occur, the turbine does nc unnecessary reversing.

At the boiler, the control points are

the drum pressure p_T

the water level h

the steam temperature t_K

3.1 Boiler control at manoeuvring conditions with fixed boiler pressure

Theoretical evaluations show, that the drum pressure p_T is a better control variable than the steam pressure p_K , because the thermal state of the boiler is not changed with every load change, as long as the plant is in manoeuvre state with a fixed boiler pressure. The pressure control system becomes faster and more stable.

The water level in the steam drum depends - when feed water flow and steam demand are in equilibrium - practically on the steam volumes V_D, enclosed within the boiler tubes. This steam volume V_D is practically proportional to the evaporation. When the water level set point is made proportional to the steam flow $\dot{m}_D$, then the water volume in the boiler stays constant - with all the advantages for the fuel supply - and the feed water control valve operates in the same direction as the turbine control valve and not in the opposite direction as it does with a fixed water level set point. By this arrangement the feed water control becomes faster and more stable.

3.2 Feed water control under full away conditions with floating boiler pressure

Generally the movement of the water level in the drum is covered by the equation

$$A_T \frac{dh}{dt} =$$

$$= \frac{dm_W}{dt} v_W + m_W \frac{dv_W}{dt} + \frac{dm_D}{dt} v_D + m_D \frac{dv_D}{dt} \quad 3.1$$

The first term comprises the difference between the feed water supply $\dot{m}_{SP}$ and the steam demand $\dot{m}_D$

$$\frac{dm_W}{dt} = \dot{m}_{SP} - \dot{m}_D \quad 3.2$$

The second term in equ. 3.1 represents the change of the water volume due to the change of the specific volume v_w.

$$m_W \cdot \frac{dv_W}{dt} = \frac{V_{wo}}{v_W} \cdot \frac{dv_W}{dt} = \frac{V_{wo}}{v_W} \cdot \frac{dv_W}{dp} \cdot \frac{dp}{dt} \quad 3.3$$

When for the floating pressure range of $60 \leqq p_T \leqq 160$ bar the specific volume of the water is approximated by the linear equation (Fig. 10)

$$v_W = (1074 + 3{,}85\ p_T) \cdot 10^{-6}\ m^3/kg \quad 3.4$$
$(p_T$ in bar)

then the derivation to the pressure becomes constant.

$$\frac{dv_W}{dp} = 3{,}85 \cdot 10^{-6}\ m^3/kg.bar \quad 3.5$$

$$m_W \frac{dv_W}{dt} = \frac{V_{wo}}{279 + p_T} \cdot \frac{dp}{dt} \quad 3.6$$

Terms 3 and 4 represent the change in the steam volume V_D. The calculation of the natural circulation shows, that the steam volume V_D increases by 20%, when the load is increased from 40% to 100% and the pressure from 65 bar to 155 bar.

Table 3.1 - Steam volume V_D as function of pressure p_T

p_T	bar	65	95	125	155
$\dot{m}_D/\dot{m}_{DO}$	-	40	60	80	100
V_D	m³	0,821	0,93	0,995	1,04

For this pressure range, the steam volume V_D can be set proportional to the steam pressure p_T.

$$V_D = 10^{-3}\ (659 + 2{,}6\ p_T)\ (p_T \text{ in bar}) \quad 3.7$$

Then terms 3 and 4 of equation 3.1 becomes

$$\frac{dm_D}{dt} v_D + m_D \frac{dv_D}{dt} = \frac{dV_D}{dt} = \frac{dV_D}{dp} \cdot \frac{dp}{dt} = 2{,}6 \cdot 10^{-3} \frac{dp}{dt} \quad 3.8$$

When the water level set point h_s is kept proportional to the velocity w_D of the steam in the superheater steam line, then it will be constant during the floating of the pressure range.

$$A_T \frac{dh}{dt} = 0 \quad 3.9$$

With equ. 3.2 - 3.9 equ. 3.1 becomes

$$0 = (\dot{m}_{SP} - \dot{m}_D) v_W + \left(\frac{V_{wo}}{v_W} \cdot \frac{dv_W}{dp} + \frac{dV_D}{dp} \right) \cdot \frac{dp}{dt} \quad 3.10$$

$$\dot{m}_{SP} = \dot{m}_D - \frac{1}{v_W} \left(\frac{V_{wo}}{v_W} \cdot \frac{dv_W}{dp} + \frac{dV_D}{dp} \right) \cdot \frac{dp}{dt} \quad 3.11$$

Equation 3.11 shows, that the feed water $\dot{m}_D$ pursues the steam flow m_D, when the load is increased (dp/dt positive) and leads the steam flow, when the load is decreased (dp/dt negative).

With the numbers from equ. 3.4, 3.5 and 3.8 and $dp/dt = + 0.05$ bar/s the difference $\dot{m}_{SP} - \dot{m}_D$ becomes

$$\begin{aligned} \dot{m}_{SP} - \dot{m}_D &= -2{,}68\ kg/s \triangleq 0{,}103 \dot{m}_{DO} \\ (p_T &= 65\ bar) \\ &= -1{,}69\ kg/s \triangleq 0{,}061 \dot{m}_{DO} \\ (p_T &= 155\ bar) \end{aligned} \quad 3.12$$

Fig. 11 shows these relations.

3.3 Boiler pressure control under full away conditions with floating boiler pressure

The general equation covering the boiler pressure is

$$\dot{m}_{DVI}(t) = \left(\frac{V_{wo}}{v_W(p)} \cdot c \cdot q + \frac{m_{RO}}{p_o} \right) \frac{dp}{dt} + \dot{m}_D(t) \quad 3.13$$

where

$$\dot{m}_{DVI}(t) = \dot{m}_B \frac{HU_{EF}}{\lambda} \qquad 3.14$$

represents the steam flow, which could be generated by the fuel flow $\dot{m}_B$ (virtual steam flow, Profos [8]). The term

$$\frac{V_{wo}}{v_W} c\, q + \frac{m_{RO}}{p_o} \qquad 3.15$$

shows the influence of the self evaporation.

Manoeuvres should be done with the lowest possible steam change rate. The control should be able to achieve a linear increase (or decrease) of the steam flow $\dot{m}_D$.

$$\dot{m}_D(t) = \dot{m}_{DA} - (\dot{m}_{DA} - \dot{m}_{DE}) \frac{t}{T_B} \qquad 3.16$$

T_B is the time passing by to increase the steam flow from m_{DCMR} (Continuous Manoeuvre Rating) to $\dot{m}_{DMCR}$ (Maximum Continuous Rating). As under full away conditions all the steam flows through the main turbine, the steam pressure before turbine is - according to the law of Stodola [9] - proportional to the steam flow.

$$p_{VT}(t) = p_{VTA} - (p_{VTA} - p_{VTE}) \frac{t}{T_B} \qquad 3.17$$

In the full away area, the boiler pressure p_T can be set

$$p_T = c_p \cdot p_{VT} \qquad 3.18$$

as the losses are proportional to the steam flow $\dot{m}_D$, due to the constant steam velocity W_D

$$p_T(t) = p_{TA} - (p_{TA} - p_{TE}) \frac{t}{T_B} \qquad 3.19$$

$$\frac{dp_T}{dt} = - \frac{p_{TA} - p_{TE}}{T_B} \qquad 3.20$$

With equ. 3.20 equ. 3.19 becomes

$$\dot{m}_{DVI}(t) = \dot{m}_{DA} - (\dot{m}_{DA} - \dot{m}_{DE}) \frac{t}{T_B} - \left(\frac{V_{wo}}{v_W} \cdot c \cdot q + \frac{m_{RO}}{p_o}\right) \cdot \frac{p_{TA} - p_{TE}}{T_B} \qquad 3.21$$

Equ. 3.21 shows, that the fuel flow $\dot{m}_B$ has to lead the steam flow $\dot{m}_D$, when the load is increased, whereas it follows $\dot{m}_D$, when the load is decreased. With T_B = 1800 s and p_{TA} = 65 bar and p_{TE} = 155 bar d_p/d_T is o o5 bar/s and the difference between fuel flow $\dot{m}_{DVI}$ and steam demand $\dot{m}_D$ becomes

$$\dot{m}_{DVI} - \dot{m}_D = + 3{,}73 \text{ kg/s} \hat{=} 0{,}134\, \dot{m}_{DO} \qquad p_T = 65 \text{ bar}$$

$$= + 3{,}03 \text{ kg/s} \hat{=} 0{,}109\, \dot{m}_{DO} \qquad p_T = 155 \text{ bar}$$

Fig. 12 shows the behaviour for a load reduction

4. Conclusions

This paper presents a very simple steam cycle with a fuel consumption, which is lower than the corrected fuel consumption of a low speed diesel motor. To achieve this good economy, the boiler pressure controls the turbine power, when the ship is under steady steaming conditions. The boiler pressure floats from p_K = 65 bar at manoeuvre conditions to p_K = 150 bar at MCR. Excessive wetness at the LP turbine exit is avoided by a bled steam heated reheater in the cross over between HP-turbine and LP-turbine. To lower the fuel consumption further, the feed pump and the generator are attached to the main gear, when the plant is in steady steaming conditions. In addition to this the evaporatior and the lub oil cooler are cooled by the main condensate.

An integrated control system for the whole plant is suggested. At manoeuvres turbine, boiler pressure and feed water supply are directly controlled. The control point for the boiler pressure is the drum pressure p_T. The water level set point is proportional to the steam velocity in the superheater line. When the boiler is operated with a fixed pressure, the water level is practically proportional to the steam load. With a floating boiler pressure the water level set point is practically constant.

Numerous tests were run on trial trips, at dock trials and at the steam model plant of the Institut für Schiffstechnik in Berlin, backed up by computer simulation both for the fixed boiler pressure and the floating boiler pressure. All tests proved the feasibility of the control system for the proposed steam cycle.

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NOMENCLATURE

γ = spec. gravity

Δ = difference

λ = $h_{UE} - h_{SP}$

NOMENCLATURE

A	=	Area
K	=	Costs
P	=	Power
T	=	Time constant
V	=	Volume
MCR	=	Maximum Continuous Rating
CMR	=	Continuous Manoeuvre Rating
HU_{EF}	=	Lower effective heating value
c	=	proportional factors
f	=	frequency
g	=	gravity constant
h	=	height, enthalpy
m	=	mass
$\dot{m}$	=	mass flow
p	=	pressure
q	=	evaporation number
r	=	evaporation heat
t	=	temperature, time
u	=	natural circulation number
v	=	specific volume
w	=	velocity

INDICES

A	=	begin of a manoeuvre
B	=	fuel
D	=	steam
E	=	end of a manoeuvre, electric
K	=	condensate, state at boiler outlet
O	=	design point
R	=	pipe, reduced
T	=	state at boiler drum
W	=	water
Z	=	circulation
AG	=	exhaust gas
UE	=	superheater
EA	=	deaerator
ZE	=	reheat
SP	=	feed water
VI	=	virtual
SPEZ	=	specific
i	=	isenthropic, general index
k	=	boiler
m	=	mean value
r	=	real value
s	=	set point, desired
(t)	=	time
v	=	destillat
w	=	water

DISCUSSION

COMMANDER K I SHORT OBE DSC RN CEng FIMarE opened the discussion as follows:

Machinery Selection Parameters

The selection of propulsion machinery for ships was not a simple matter of selecting the installation offering the lowest fuel consumption. A variety of other factors had to be included in the equation such as:

- (i) Type of vessel and its duty;
- (ii) Power range;
- (iii) Fuel available;
- (iv) Fuel costs;
- (v) Capital cost;
- (vi) Weight and space;
- (vii) Lubricating oil and other running costs;
- (viii) Maintenance costs

and so on.

Whilst this was often said - it was often forgotten, and judgments were irresponsibly offered on fuel consumption data only.

When fuel was cheap and abundant there was a limit to the capital expenditure which it made commercial sense to incur to obtain improvements in steam plant efficiency. Nevertheless, in the power range being considered a strong case could be made, economically, for the moderately efficient steam as opposed to the higher efficiency alternative low speed diesel propulsion when all factors were considered.

When fuel prices increased so dramatically a few years ago, existing steam installation and those under construction tended to show up adversely, commercially, compared with diesel engines.

As a result, and doubtless with a few nudges from the diesel manufacturers, the ill informed had sometimes erroneously suggested that "steam was finished".

But the contrary was in fact the true situation for two important reasons:

a) Firstly, just as the increased price of oil had made it commercially viable to exploit the North Sea and other difficult sources of oil, so it had now justified adding to the cost of steam installations to improve their efficiency markedly;

b) Secondly, with the current oil depletion forecasts and the likelihood of more and more degraded fuel being marketed, such fuel was undoubtedly better suited for boiler than diesel machinery.

So investigations of improvements to steam cycles had a serious end in view and were by no means nugatory.

Every reasonable proposal for steam cycles offering improved efficiency must continue to be explored with enthusiasm and zeal.

We were therefore particularly grateful to Professor Grossmann for interesting himself in this subject and for marshalling his facilities to prepare a paper of such topical interest. Commander Short also agreed with him that the current dialogue concerning steam cycles should have taken place some fifteen years ago; in which case steam would have been in a better position today to play its part in the ship propulsion scenario. It should not be allowed to stagnate now or be submerged by over-selling by its competitors.

Coming as it did from a distinguished academic rather than an industrial source the paper was particularly valuable.

Commander Short then commented upon several points in the paper.

Reheat Pressure Threshold The author had suggested a pressure threshold of 80 bar for non reheat systems but it was the Commander's understanding that with a properly designed turbine up to 100 bar was acceptable for 510°C and over 100 bar for the 530°C used in this study without there being cause for concern about erosion of blading at the LP turbine exit.

Floating Boiler Pressure When he first read this paper he was puzzled as to the matter of fact way in which it was suggested that a floating boiler pressure would improve efficiency at lower powers by reducing throttling losses.

He had been brought up to believe in nozzling down on turbines to maintain a high inlet pressure to maintain turbine efficiency and he sought an explanation in the paper for the bald claim that a fixed pressure installation involved additional fuel consumption of circa 2.3% at 69% maximum power and 3.3% at 40% power. But to no avail.

In a SNAME paper read earlier this year by Siegel of General Electric (USA) "Improvements in Part Load Performance of Marine Steam Turbines" he had produced the graph shown in figure 13 and commented "It is seen that the plant fuel rate suffers with variable pressure operation". As the vessel mentioned regarding the re-heat boiler was the "ESSO NORWAY" fitted with GE machinery, Commander Short would have expected the Siegel conclusion to apply to the author's proposal.

Then the Commander wss introduced to Donald's recently published slim volume "Marine Steam Turbines" (which, incidentally, he recommended to all those currently interested in steam propulsion), and from it he deduced that the author's comments were probably based upon the use of a Blohm & Voss propulsion unit with its unconventional 50% reaction type HP turbine employing throttle valve control with full 360 degrees inlet steam admission area.

According to Donald, whilst this turbine could be marginally more efficient at full power than a nozzle controlled impulse turbine it was less efficient at part load.

If his own supposition was correct the turbine would of course not mind how the reduced steam pressure was obtained whether by throttling or by floating the boiler pressure but the latter would save throttling losses as claimed.

Had a nozzle controlled impulse HP turbine been used then perhaps part load efficiency would have been higher with a fixed boiler pressure.

Perhaps the author would be good enough to comment upon this point which had been so fundamental to his proposals.

Flue Gas Versus Steam Reheat The Commander left any detailed discussion of the boiler and turbine material problems with flue gas reheat to their designers and manufacturers. He would however comment that General Electric had more than nine large flue gas reheat installations 100 bar 510°C already at sea.

The MP turbines were designed to accept the temperature changes associated with sudden load changes (reheat to non reheat and vice versa) without placing any restraints on the operator. No turbine problems had been experienced in service, boiler control had given few problems and fuel consumption rates as low as 239 gms/kWh had been obtained at full power with some of them.

It was thus not correct to suggest that all MP turbines had to be cooled down gradually and that the freedom of action of the operators was therefore inevitably restricted with flue gas reheat.

Limiting Pressure for Flue Gas Reheat Boiler
The author had suggested that for marine boilers there was a space reason for limiting the boiler pressure with flue gas reheat to 130 bar.

A current study by the General Electric Company of America for the US Maritime Administration was seriously proposing a reheat plant in excess of this, namely:

165 bar
560°C/560°C

with an all purpose SFC of:

219/232 gms/kWh

for a plant of 30,000 to 45,000 kW. So he believed that the space difficulties were not insuperable.

Boiler Circulation at Reduced Pressure The author had presented the results of some calculations which suggested that circulation improved with lowered pressure due to the reduction in the specific gravity of the mixture of steam and water in the evaporation tubes resulting from the bigger volume of steam in the boiler.

Did not the appearance of more steam in the mixture tend to reduce circulation by "choking" - and if pressure reduction was carried to the extreme with tubes practically filled with steam, surely mass flow must be reduced ?

Could the author please expand upon the method of calculation involved and any verification carried out.

Astern Turbine Pressure In this study the astern turbine used steam at the lower pressure. This must inevitably increase the size of the turbine above that of one designed forthe higher pressure and thus increase the astern turbine losses.

Independent Feed Pump - Increase in Fuel Consumption
Could the author please explain why the increased fuel consumption was only shown as 0.73% whereas the figures given earlier indicated that:

(i) 2% of Main Steam was needed when attached;

(ii) 4.8% of Main Steam was needed when separate.

As steam produced must be roughly proportional to fuel, he would have expected a higher gain on these figures even allowing for the additional steam needed for the de-aerator when pump was attached.

The combined attached feed pump and generator full power saving was quoted in the paper as 2%. In their MST-14 conservative reheat fixed pressure propulsion plant which was in service, GE(USA) had calculated that 2.0/4.0% of the fuel rate improvement was attributable to this arrangement, the actual figures varying with power. Fuel rate figures at full power circa 239 gms/kWh were being obtained.

Not only was the GE estimate higher than claimed by the author but it was based upon a main engine driven feed pump contrary to his suggestion that fixed pressure systems would 'naturally' have an independent feed pump.

Slow Speed Diesel Engine Consumption Figures
The author had been conservative in calculating these figures. Using SNAME Technical and Research Bulletin 3-27 "Marine Diesel Power Plant Performance Practice", Commander Short arrived at the following full power consumption figures:

		Cf Author
SFC	214 gms/kWh	214
Poor Combustion 2%	4	-
Lubricating Oil 4%	8	5
Correction for rev/min (7% for 18)	14	17
Centrifuge Loss (1%)	2	-
Generator Load (2.7%)	5½	4
Guarantee Margin (4%)	9	-
	256 gms/kWh	240

This revision very much improved the relative position of the steam plant. He had re-drawn Professor Grossmann's figure 7 (see figure 14) and he had included some figures for other steam plant designs. From the GE MST-14 full power point he had deduced the part load consumption curve from Siegel's paper for the two row/one row design as shown dotted. This was flatter than the author's curve.

He thought that in presenting studies of this nature such estimates should be as realistic as possible from the beginning even though perhaps every margin was not necessarily needed to make one's case. If not claimed originally it could be more difficult to establish their credibility later.

Control System The full implications of the control systems calculations were not clear to him and he would welcome a practical explanation. Did they for example suggest a simplification or the elimination of certain controls as a result of the floating boiler pressure ? During the presentation the author had shown slides of some additional graphs to those in the paper and these might have clarified some of this had there been time to study them and correlate them with the paper.

Comments

Commander Short believed that techniques already existed - or under the impetus of need were not far away - for the design and construction of boilers, turbines and ancilliary equipment which if properly tied together would offer fuel rates, over the service range and maintenance costs in particular, which were extremely competitive with other forms of propulsion.

The tying together was a matter of selecting the right cycle, the correct matching of components and the installation of effective control gear.

He was sure that with modern technology the material problems could be solved if properly posed.

We were therefore very dependent for advance upon finding the right cycle.

He concluded by urging Professor Grossmann and others to continue to use the ingenuity, as had been done in this paper, to suggest and evaluate cycle improvements however radical they might appear initially.

MR. N.A. BRICKNELL said that the paper had referred to improved fuel consumption at manoeuvring speeds by reducing the boiler pressure to reduce throttle losses. It was current practice to attain similar results by nozzle control and he had been led to believe this was more practical than altering the boiler pressure. This alternative was mentioned in the paper; would the author please comment ?

The author had referred to quick opening valves to start back up turbines. Whilst this was reasonable for small items such as feed pumps, it was normal practice to warm turbo alternators slowly. Would the author explain what procedure was adopted to avoid damage due to suddenly applied high temperatures, or were the turbo alternators continuously warmed If so, how did this affect overall running costs. Also despite all preventative precautions the supply lines were sure to be filled with water at some time when the quick opening valve was actuated. He wondered how long it would be before severe damage occurred to pipe and turbines etc., due to resultant water hammer.

As the boiler load reduced the author proposed altering the output of feed pumps, fans, etc., by varying voltage and frequency. This would presuppose that these items had the same performance characteristics so that the same frequency/voltage reduction would give the required output reduction for all items. Also as the varying frequency was supplied by the main alternator it meant that only a frequency variable supply was available for all equipment. This would not be suitable for hotel services. electronic, radios and other equipment. Presumably some form of voltage and frequency stabilised supply was made available for these items. He would be interested to know how this affected the overall cost in comparison with a conventional steam cycle.

Whilst cycle efficiency was increased by using condensate to cool the lubricating oil, this idea was discarded some time ago due to the danger of oil leakage into the system during shut down times. Such leakage would be particularly dangerous at the high operating pressures mentioned. What safeguards were considered necessary to prevent such an occurrence and at what additional cost ?

MR. CHRISTOPHER MURTHI said that he had had some operating experience with a reheat turbine installation and flue gas reheat boiler, W.P. 86 bar, superheat and reheat steam temperature 515°C, engine driven feed pump and alternator and most of his comments would be based on the practical aspect of the cycle presented.

However, he would first like to know more about the construction of the H.P. turbine. It would appear that for a pressure drop from 150 bar, the turbine would have to be much longer than the average marine turbine. Alternatively, the turbine would need to have initial velocity stages which, he suggested, would drop the efficiency and negate any increase obtained due to the initial very high pressure.

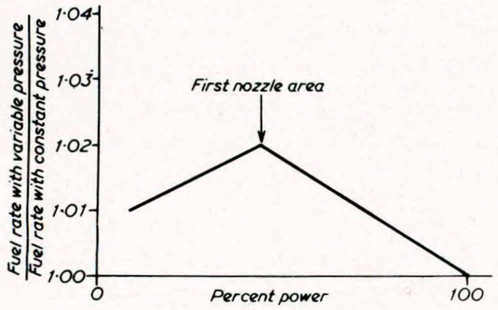


Fig.13 – Relative fuel rate

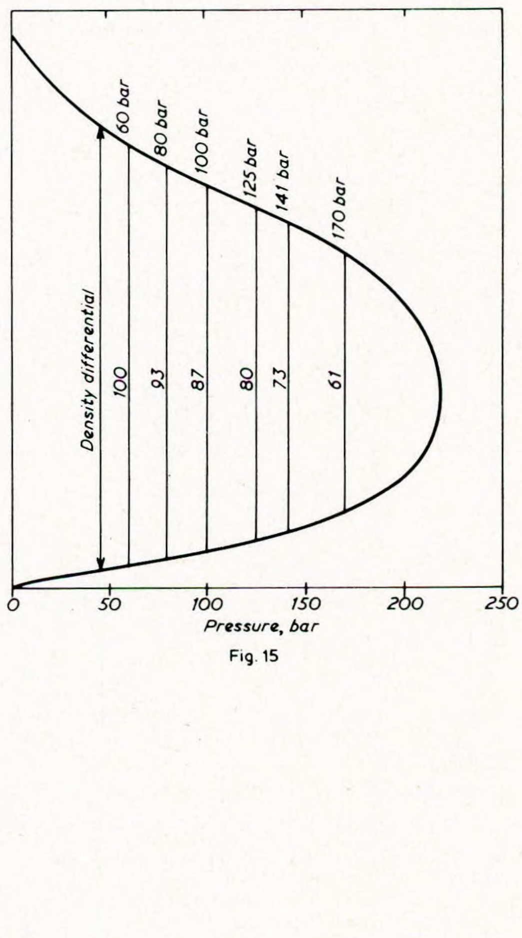


Fig. 15

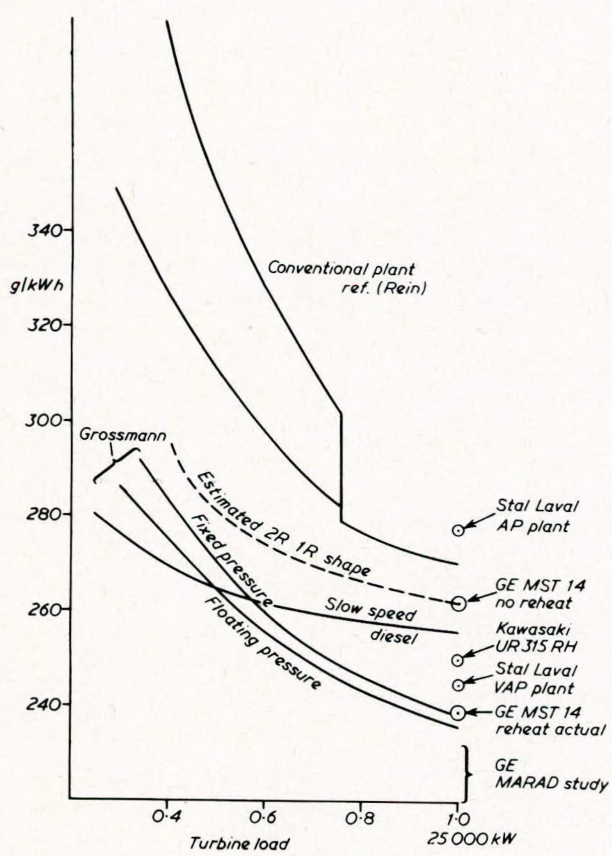


Fig. 14

During the operation of boilers at 86 bar, enormous troubles were experienced with chemical hideout. Could the author indicate the feed water treatment that would be used at pressures of 150 bar and how effective it would be in overcoming problems similar to chemical hideout, etc.

Another area of operating trouble was the direct reading gauge glass which was of the bulls eye type. Could the author indicate the type of direct water level indicators to be fitted or would he rely on differential pressure cells only for indication of the boiler water level. If so, it would be interesting to know the thoughts of the D.T.I. regarding this.

Finally, two comments which he thought bore special reference to the advanced steam cycle as described. In one instance while operating with a "clutched-in" alternator, when the revolutions were decreased, the clutch disengaged but the electric steam valve for the back up turbine failed to open and the ship blacked out. Worse still, on another occasion, during a speed reduction, the steam valve for the back up turbine opened, but the clutch failed to completely disengage, with catastrophic results.

MR. A.F. HODGKIN (in a contribution read for him by MR. CULLEN) said that the effort made by Professor Grossmann to produce a simple yet economically attractive steam cycle was applauded by all who had an interest in seeing steam propulsion stay as a viable proposition. The boiler designer would have little quarrel with him over steam/steam reheat or floating pressure operation. It was a little disturbing however to find statements which could tend to deter those who might be currently interested in what we call conventional reheat using boilers similar to those in figure 1. Whilst referring to figure 1, he was sure that Professor Grossmann realised his error in that the reheater was not as shown but was the top two banks of tubes in the left hand view of the illustration.

There were a number of ships at sea with reheat machinery using boilers with parallel gas paths, similar in style to figure 1 and it was believed that these had given reasonably acceptable service. Certainly, in the case of "ESSO NORWAY" mentioned in the paper, experience was such as to encourage reference to the ship, on more than one occasion, as the "best ship in the fleet". Possibly, this was partly due to the care and attention devoted to the ship by Professor Grossmann himself. Nevertheless, there was felt to be sufficient evidence to support the contention that although reheat machinery might be a trifle more complicated than straight cycle plant, this did not cause operational difficulties of any kind.

It was true that as steam conditions advanced the reheat boiler became more difficult to design. Changes took place in the proportion of the various types of surface required - as mentioned in the paper. However, it was far from impossible as proved by the work done by G.E. (USA) and reported by MARAD in report number MA-RD-920-77056. The boiler design prepared for this had steam conditions of 170 bar, 568°C at the superheater outlet and 28 bar, 565°C at the reheater outlet.

Circulation, that is natural circulation, of boilers was mentioned many times in the paper and it was rightly an important aspect of any boiler design. Marine boilers had always had to face the requirement of a rapid manoeuvring situation especially in warships. There was no substantial history of any shortcoming in this respect. Modern watertube boilers of radiant type now show improved circulation characteristics compared to earlier bi-drum designs. See Table II.

TABLE II

- Absorption per Tube per Unit Flow -

Circuit	M.R. Boiler	Early Integral Furnace Boiler
Furnace side wall	42.8	53.9
Screen	38.1	47.0
Furnace front & rear	41.5	52.0

As the pressure at which a boiler generates steam was raised, there was a continual reduction in the difference in density between steam and water until at the critical pressure this difference disappeared (figure 15). Since natural circulation was dependent upon this density difference, natural circulation could only be obtained at pressures up to a fraction of the critical value. It followed that for a design made at a pressure equal to this limiting fraction, circulation would tend to improve as the pressure was reduced until at a very low pressure the rapidly increasing steam specific volume caused a large increase in the resistance of riser circuits. However, between the pressures of 65 bar and 155 bar the effect described in the paper could be obtained. This was not to say that natural circulation at the highest pressure was in any way unsatisfactory. The design of the circulating system took account of the operating pressure and by a judicious arrangement of heated circuits, downcomer/feeder and riser arrangements, a satisfactory circulation ratio could be obtained up to 170 bar and the boiler offered for this pressure would be as able to safely execute the demands upon it as a boiler designed for a lower pressure. The degree of security originally built into the design rendered any improvement due to manoeuvring at reduced pressure of little consequence.

MR. M.S. BRADLEY, M.I.Mar.E., said that he would like to make a few comments on the remarks of previous contributors regarding nozelling down turbines, particularly with regard to reduced power steaming and Professor Grossmann's proposed cycle.

Some time ago he had been involved, separately, with two Shipping Companies, regarding economical reduced speed operation of steam VLCCs. Over several measured trials it was invariably found that the lowest full consumption occurred for the conditions applicable to nozzle groups fully open or fully shut. These points were those marked 'N' on figure 16. The continuous line normally drawn through these points to produce a speed/power (fuel consumption) curve was in fact a fiction. Operation at intermediate powers due to throttling at the manoeuvring valve introduced losses, principally due to the higher LP exhaust temperature. This is indicated in figures 17 and 19.

Attempts to operate the plant at intermediate powers bv alternating between nozzle positions as proposed by Mr. Larsen also introduced losses, since the average steaming rate lay somewhere on the chord line connecting the nozzle positions, the position being determined by the relative percentage of the time steamed at each setting (figure 18).

Professor Grossmann's proposals on the other hand would appear to allow the turbines to be steamed under nozzle fully open conditions, without the need for throttling for intermediate powers. This was a laudable achievement and was attained solely by the, to Mr. Bradley's mind, fascinating and so far unexplored principle of a floating boiler pressure.

Perhaps the author would care to comment on the possibility of existing steam plant being operated at reduced boiler pressure for service at reduced speeds. Obviously the lower limit for such operation would be defined by the capabilities of the turbo alternator and feed pump governors. Nevertheless, worthwhile savings are apparent in principle.

Regarding the authors comments on a floating deaerator pressure, with which he agreed in principle, he wished to make the following observations: during the above mentioned investigations into economical reduced power operation, various attempts were made to ensure that the main turbine bleeds remained open as long as possible. In order to continue extracting bleed steam from the turbines at low rev/min the live steam make up reducing valve set points were adjusted downwards, the net result being that all system pressures, including that of the deaerator, floated downwards. The plant operated satisfactorily until the deaerator pressure became so depressed that the feed pump governor was unable to cope with the reduced exhaust pressure and instability set in. Could the author state whether this problem, and others similar to it, had been examined and satisfactory control methods been achieved without adding any further complexity to the system. It had been Mr. Bradley's experience that plant and equipment might only be termed "Self Regulating" or "Self Compensating" over a comparatively narrow bandwidth of operation.

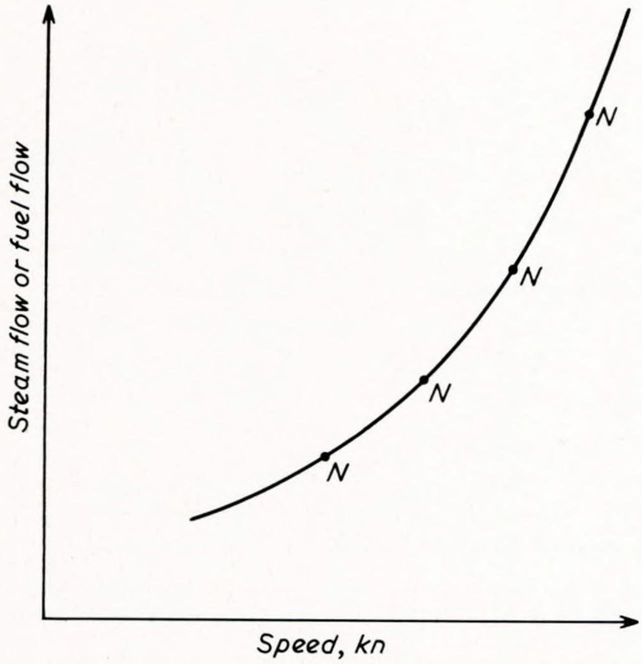


Fig. 16

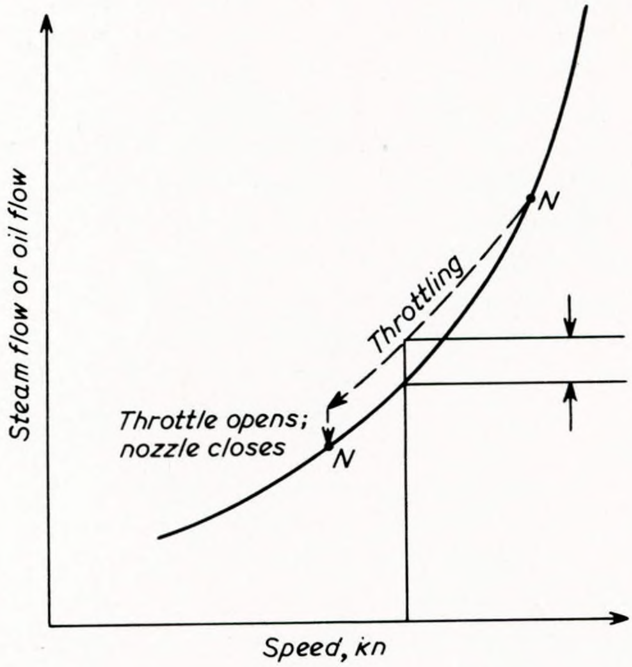


Fig. 17

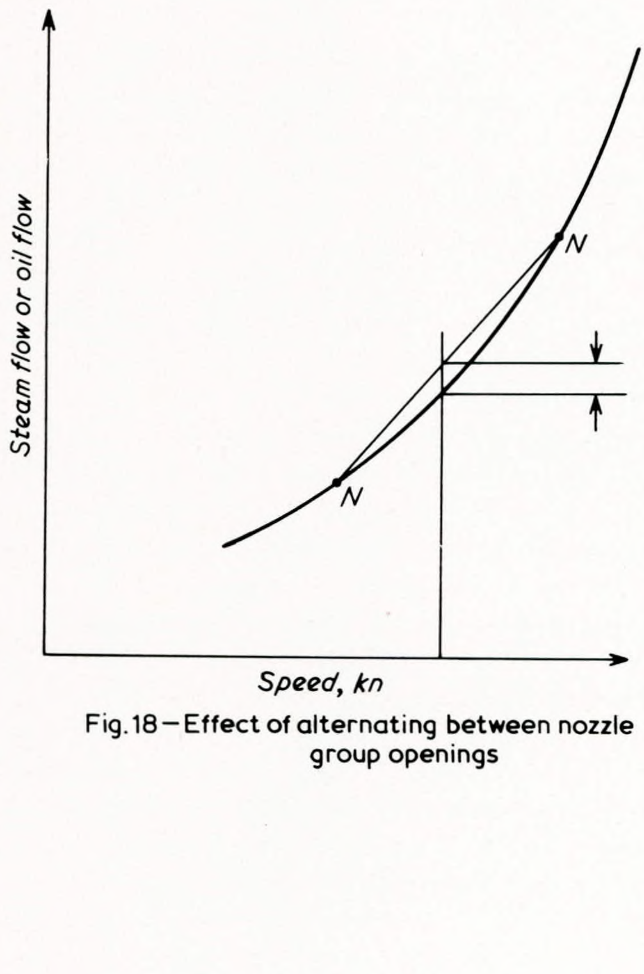


Fig. 18 — Effect of alternating between nozzle group openings

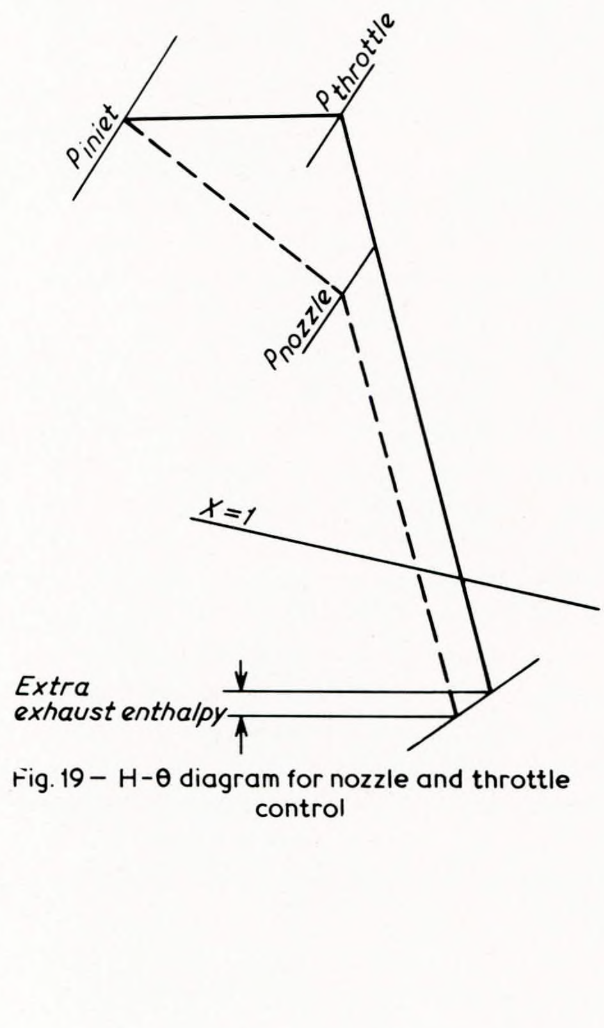


Fig. 19 — H-θ diagram for nozzle and throttle control

AUTHOR'S REPLY

PROFESSOR GROSSMANN thanked all contributors to the discussion, especially Commander Short, who had very thoroughly worked through the paper. He would try to answer all questions as well as possible, condensing them to some major points of issue.

Floating boiler pressure This point was raised by practically all contributors. The floating boiler pressure was first of all a method to reduce the stresses in the boiler, the high pressure steam pipes and the HP turbine in the manoeuvring state, when rapid load changes occurred. By reducing the pressure due to the steam pressure by the factor 2.5, the overall stresses were reduced considerably and the overall safety was definitely increased, especially at flanges, valves, packings and gland seals. The floating pressure was based on a turbine with a 360° inlet steam admission area. This type of turbine was built by AEG/KWU for Howaldtswerke ships. The full inlet area gave an undisturbed temperature field at the steam inlet, which should be reached with the proposed high pressure, otherwise additional thermal stresses would be found at the HP turbine inlet. (The above consideration might be a point which would influence a prospective owner to decide for a high pressure steam plant).

The second main point for the floating pressure was that it allowed the main feed pump to be attached to the main gear during the whole full away range from 40% power at 73.6% revolution to 100% power and 100% revolution. With a fixed boiler pressure, the discharge pressure of the attached feedpump would drop at a load reduction underneath the boiler pressure, so either the back up turbine or the stand by feed pump would have to go on duty. The attached feed pump made the floating deaerator pressure possible, so that the bled points could be used best under full-away conditions. This would also take care of the stability problems of the feed pump governor as mentioned by Mr. Bradley. The feed pump turbine was not connected to the deaerator during full-away mode, so the change of the back pressure could not effect the feed pump governor.

The fuel consumption decreased because, due to the floating boiler pressure, the steam consumption of the feed pump decreased from

d_{sp} = 0.048 kg steam/kg feedwater

to

d_{sp} = 0.0304 kg steam/kg feedwater

But as the heating steam for the deaerator had to be made up by bled steam, the gain was not more than the 0.73% shown in the paper. This difference was naturally achieved with a throttle controlled turbine.

To sum up this discussion point, the main benefits of the floating boiler pressure were

(i) the higher safety;

(ii) the better operational possibilities for power down to slow steaming.

The floating boiler pressure was not restricted to the bled steam reheat system. It could be used with all flue gas reheat cycles and he would strongly recommend that this should be done.

Floating pressure versus nozzle control The nozzle controlled turbine certainly had a better steam consumption than the throttle controlled turbine at low loads because the enthalpy drop was greater. This gain was partly offset by the higher steam consumption of the usually throttle controlled feed pump. The difference in fuel consumption was shown by the flatter curve in Commander Short's figure. However, for the reasons mentioned before, we would like to stick to the throttle control and the floating boiler pressure.

Mr. Larsen had shown that the relative change in fuel consumption for the VAP-System was the same as for the bled steam-floating boiler pressure system. His paper in MER October 1977 showed that the VAP main turbine was throttle controlled by the valve between main boiler and fluidized bed superheater. The throttle losses at the feed pump turbine were offset by having this turbine nozzle controlled. The throttle losses at the deaerator were avoided by leaving it independent from the main turbine and heated by the exhaust steam of the feed pump turbine only. As the differences in generator load were below 1% at 40%

power of the main turbine, the similarity of the two curves was no longer surprising.

Flue gas versus steam reheat For the flue gas reheat boiler two temperatures must be controlled: the superheater outlet temperature and the reheater outlet temperature. The vane or damper control at the cold end of the boiler was regarded as being insufficient to control the superheater outlet temperature and the reheater outlet temperature simultaneously, as the ratio between normal steam flow and reheat steam flow might change due to secondary load changes. With the VAP system there was the additional control system for the fluidized bed heaters, which included an additional fan.

If GE did not recall any problems in changing the temperature at the inlet of the IP-turbine by 200°C at the change over from non reheat to reheat, this was only because the boiler control system was taking care of the time factor already; which meant that the boiler needed more time to attain the new state rather than the turbine.

On the "ESSO NORWAY" the switch over time for the reheater had to be changed from 20 sec as per design to 300 sec - the maximum time for the installed controller. This was necessary, because the primary superheater got out of control. As the temperature rise at the IP-turbine was 180° (323 - 503°C) the 300 sec for activating the reheater were surely no problem for the turbine.

With the TT "GOLAR PATRICIA" it took even 30 min to go from reheat to non reheat and vice versa with a temperature difference of 177°C (343 - 520°C) (see Japan Shipbuilding and Marine Engineering Vol. 6, No. 1, 1971). On this ship, there were two dampers, in series, installed to protect the reheater pass of the boiler.

The difference in steam pressure indicates only that the ratio of the space for the superheater and the economiser to the volume of the boiler proper (the space between the water walls) was the same for a normal boiler with 155 bar drum pressure as for the flue gas reheat boiler with 135 bar drum pressure; taking into account the ratio of the space for superheater, economiser and reheater with this boiler. Up to very near to the critical point, there would always be a higher boiler possible for the "normal" boiler than for the reheat boiler, if designing the superheater and reheater into the water walls was considered as one criterion.

For the steam conditions of the bled steam system and a flue gas reheat system with equal steam temperatures but 135 bar drum pressure, the gain of 3.6% mentioned by Mr. Larsen was reduced to 1.8 - 2.0%.

Circulation The floating boiler pressure brought no danger of "choking" to the steam generating pipes. As the steam mass flow was reduced proportional to the pressure, this meant that the steam volume flow stayed practically constant and the ratio of the steam volume inside the water walls to the water volume stayed more or less constant. (The steam volume at 40% load was practically the same as that at 100% boiler load).

The difference between the specific gravity of the boiler water and that of the steam decreased with increasing boiler pressure and so did the "driving force" of the circulation, as shown in the following table, compared with the 60 bar boiler.

p	bar	60	100	130	150	165	180	190
F_A/F_{A60}	-	1	0.96	0.92	0.88	0.84	0.79	0.74

Mr. Hodgkins showed this quite clearly in figure 15. Both this figure and the above table show that for the 100 bar plants like "ESSO NORWAY", there should be no problems with the circulation. But with higher boiler pressure above 140 - 150 bar and the lower evaporation rate, the boiler designer achieved circulation numbers of 10 - 12 instead of the usual 18 - 24 for the 60 bar boiler, so special care had to be taken to ensure a good and easy natural flow.

Pressure for auxiliary turbines and astern turbine There would be enough problems to get the flanges and the valves tight in the main steam line. To reduce these trouble spots we would suggest having only the main steam line from the boiler (boilers) to the ahead turbine designed for the high pressure and the high temperature. Just before the main turbine, a branch line for the whole auxiliary steam system was provided. This steam was reduced to 60 - 65 bar and supplied to the turbogenerator and the feed pump - if these engines were not attached to the main gear - and to the astern turbine, the cargo pump turbines etc. This certainly would mean a higher fuel consumption at full away conditions, when feed pump and generator were steam driven, but the whole steam system became safer. There was no problem with the astern power, as all existing turbines were designed for just this pressure. He did not believe that the size of the astern turbine could be reduced considerably by going to a higher steam pressure, so we had to live with the ventilation losses.

Independent feed pump - Fuel consumption

Commander Short was correct in quoting the steam consumption of the feed pump to be

$$m_{DSP} = 4830 \text{ kg/h}$$

with an independent feed pump designed for an overall efficiency of 44% and a volume flow of 0.0395 m^3/s

and

$$m_{DSP} = 2985 \text{ kg/h}$$

as an attached pump. Unfortunately with the attached pump roughly 2900 kg/h steam from a bled point was needed to heat the deaerator. This meant an increase of the steam flow to the turbine by 1160 kg/h, so the real saving was only 740 kg steam/h, which meant 0.73%.

Even with the above mentioned good efficiency the independent feed pump system delivered more exhaust steam than the deaerator could take, so the cycle had to be redesigned to get rid of the surplus steam. Mr. Larsen had shown one way in his article in MER October 1977, by cascading the steam to the evaporator. To avoid further surplus steam the feed pump system must have a well improved efficiency. If the efficiency was just what nowadays was built into ships, then there had to be more cascading of the exhaust steam. (Professor Grossmann still had to see the feed pump with an overall efficiency of 49% in service. Up to now ship owners and ship yards only found the efficiency to be below the design point).

Floating frequency - attached generator

Mr. Bricknell and Mr. Larsen had mentioned that part of the pumps as well as the radio equipment needed a constant frequency. The following distribution of the electric power had been assumed:

Equipment, which could stand a floating frequency	500 kW
Air-conditioning 150 kW (also floating) for half the voyage only	75 kW
Equipment with constant frequency 148 kW. Frequency and voltage converting efficiency factor 0.85	175 kW
Electric load on main generator	750 kW
Real electric load	723 kW

Even if the load with constant frequency was changed to 348 kW which would bring down the frequency independent load to 375 kW, the main generator load would increase by 30 kW and the fuel consumption by 0.3 g/kWh.

The floating voltage and frequency gave an automatic power reduction for all motors directly connected to the generator. These motors would run with constant speed otherwise, so that a lot of the power would be destroyed in the throttle valves. The reduction of the electric load at low loads lowers the fuel consumption accordingly. Professor Grossmann would draw attention to the point that the comparable electric load of the VAP plant should be 783 kW due to the additional requirements of the F.B. heater booster fan.

General With respect to Mr. Murthi's experience with clutches at main gear driven auxiliaries, we would revise the system: the attached auxiliaries would have no back up turbines; two steam driven feed pumps, one steam driven generator and one diesel generator would be provided as stand by engines.

Also we would no longer suggest the condensate cooled lubricating oil cooler as, in agreement with Mr. Bricknell, this was thought to be a rather dangerous arrangement.

Feed water treatment was mentioned by Mr. Murthi. As Professor Grossmann could not give any direct comments on the feed water treatment he thought that with this high pressure the best possible feed water treatment which included on-line filtering and ion exchanging should be taken. The nuclear merchant ships had shown that with a good design, very exact and clean fabrication and on-line ion exchange, boilers could be operated without any trouble for more than eight years.

Slow speed diesel versus steam turbine system
The figures of Commander Short for the fuel consumption of the slow speed diesel engine were impressively higher than their own. In their opinion, the poor combustion and the guarantee margin should not be included in the comparison, because the combustion could be very bad in a boiler too and there existed experience with turbine and boiler manufacturers, where the steam or fuel consumption was above the design figure within the guarantee margin.

If we were to include the centrifugal losses into our figure and raised the influence of the lubricating oil to 6.5 g/kWh, we would have a total of 244 g/kWh, which seemed to be a realistic figure for comparisons.

If the lubricating oil cooler was cooled by sea water, if the exhaust gas temperature was raised to 110°C again and even if the turbo-generator was driven from the 60 bar line, then the bled steam reheat cycle had a fuel consumption of

$$m_{BSPEZ} = 242.2 \text{ g/kWh}$$

which meant that the "fuel consumption" per mile was lower than that of the equivalent slow speed diesel driven ship. This was achieved with a very simple and most reliable cycle, comprising a completely normal boiler system.

Control system The control system for the proposed steam cycle became much more simple than a control system for a normal non-reheat steam cycle.

Due to the floating water level, the feed water control system became a simple two point - water level and steam velocity - control system, which would work in line with the other control systems and not against them.

When the drum pressure was taken as control point instead of the superheater outlet pressure, the range of the fuel control system became the same as the one of the steam system, because the latent heat of the boiler water stayed constant during manoeuvres - due to the constant drum pressure. Usually there was a turn down ratio of 1 : 15 for the boiler, when the real steam change rate was 1 : 10. With higher steam change rates a burner had to be closed. With drum pressure control, steam turn down of 1 : 15 could be accepted without closing of a burner. At the same time, the thermal state of the boiler stayed constant - with our proposed system during manoeuvring conditions - which definitely increased the safety.

Both the above features were independent of the kind of the reheat system, so he would strongly recommend them for adoption.

The steam temperature control was the same as for any normal boiler, which meant that it was simpler than the temperature control of any flue gas reheat boiler.

It should always be considered that part of the automatic control could go out of action by failure, so that a part of the plant - for instance the boiler - would have to be operated manually. With two temperatures to control this would be rather difficult.

There was also no need for steam or condensate cascading for the power range from 40% to 100%, so the bled points control system also became more simple. The only additional thing was the variable deaerator pressure set point during full away mode. (Mr. Bradley's question as to the bandwidth of the "Self Regulating").

Existing plants Mr. Bradley questioned, whether there would be a substantial saving, when existing plant changed to floating boiler pressure. The answer was quite plainly "No". The turbo-generator load was more or less constant regardless of the main turbine load. If the boiler pressure went down, its steam consumption would go up so the savings at the feed pump were negated by the turbo-generator.

Professor Grossmann could see only one possibility to better existing steam plants; that was to replace the whole high pressure steam and feed water system by a new one with about 130 bar and with a bled steam reheater. With the new pressure the new HP-turbine would deliver too much power, but this surplus power could be taken care of by a directly attached generator. The new high pressure feed pump would be electrically driven. With this refitting, the fuel consumption of a normal economiser plant could be lowered from 285 g/kWh to about 256 g/kWh. The yearly savings at todays oil prices would be about DM 750,000. As the refitting at todays man hour and material prices would be around 8 - 10 Mio DM, he was afraid that the gain would be not good enough to justify the refitting cost. Only appreciable government investment grants could help there.

Conclusions

Professor Grossmann again expressed special thanks to Mr. Bricknell, Mr. Bradley, Mr. Cullen, Mr. Hodgkin, Mr. Murthi and Commander Short. Due to their contributions, it was possible to clarify some ideas and also to weed out some weak points. It was hoped that this paper had provoked some ideas or points of view, which would help to put the steam turbine plant into ships again.

Paper read at the Institute on 13 December 1977

C.P. PROPELLER DESIGN CONSIDERATIONS IN RESPECT OF VIBRATORY LOADS

Dr.-Ing. W. Wührer,
Escher Wyss, G.M.B.H.

INTRODUCTION

The controllable pitch propeller has acquired importance in the field of marine propulsion where high manoeuvrability and adaptability to changing conditions in the ship or the ship's power plant (eg. multiple engine plant operation Fig. 1) are called for. Certain classes of ship are, therefore, equipped with CPP such as ferries, Ro/Ros, reefers, tugs, trawlers and many naval vessels. The latter use combined gas turbine and diesel propulsion machinery and, without CPP, could not reverse or operate on a single engine. In some cases, power plant configurations with shaft generators, which call for constant speed operation, necessitate the adaption of a controllable pitch propeller.

It is also worth noting that there is some economic advantage in the application of CPP to vessels operating over long distances under changing load or weather conditions. In such cases, it is advantageous to adjust the power absorption of the propeller in order to operate the engine(s) continuously in the optimum condition, ie. at rated speed and output. This can be effected automatically by means of a load control system. Such systems are successfully applied by all the major makers and render CPP attractive from both the engine makers' and operators' points of view.

For the design engineer, the CPP poses a considerable challenge as it includes also a complex mechanical engineering task. It is characterised by the fact that in an adverse environment a mechanism must be provided within the confined diameter of a hub capable of changing the configuration of the blades which transform large amounts of energy in the process. Further it should be appreciated that the inflow to the propeller is inherently non-uniform, subjecting the CPP to major vibratory loads, and these wake field induced fluctuating forces require special attention and must be kept in mind when evaluating design concepts. Their magnitude must be known to enable a dynamic stress analysis to be carried out to check the safety factor of components subject to vibratory blade loads, for fatigue failure. Naturally these components are mainly the blade attachment sections of the design. Table 1 gives some examples of the kind of bending moment fluctuations (at the radius of the blade palm face) the design has to cope with. While dependent on the wake field pattern the naval architect succeeds in achieving, there is also a strong correlation between the type of ship and the magnitude of the vibratory loads. The numbers given are related total fluctuations, ie. the difference between maximum and minimum bending moment values referred to its mean value. The amplitude of the first order harmonic is somewhat less than half the values indicated. The quality of the wake-field with regard to uniformity of inflow to the propeller disc is clearly reflected in the table with naval vessels showing the smallest fluctuation in bending moment. The maximum values, which are unexpectedly high, should act as an incentive to the naval architect to improve wake field uniformity wherever possible. (The other main areas of concern are cavitation and propeller induced vibrations in the ship).

1. CPP DESIGN CONCEPTS

The effect of dynamic loads on a propeller hub depends on the design philosophy applied. Two essentially different concepts are available, the main difference being the blade bearing system chosen which has a marked effect on the type of actuating mechanism. Figure 2 shows the two basic principles; one the collar type (left) and the other a trunnion or journal type bearing. Escher Wyss use the latter design throughout its range of CPP as well as in water turbine wheels which deliver up to 110 MW. The advantages of the design are :-

i) Support of blade force in hub body

The characteristic long internal part in the trunnion design relieves the outer hub body from supporting one force of the couple that is in equilibrium with the blade force (blade upsetting moment). The blade bearing forces are characteristically supported axially in this design, with the largest force supported at the forward end where the hub body additionally stiffened by the shaft flange; the support of the largest blade-carrying force may be termed optimal. The cyclic "vibratory" loading does not effect the outer hub structure. In contrast the collar bearing concept introduces the largest radial force at the open aft end of the hub body where it is subject to constant and cyclic

Table I: Blade bending moment fluctuations at blade root (hub radius)

Vessel	Power, h.p.	rev/min	Dia.	No. of blades	Bending moment fluctuations $\Delta SM_x/\overline{SM}_x$	$\Delta SM_\eta/\overline{SM}_\eta$	resultant $\Delta SM/\overline{SM}$
Cargo	1 x 17,500	120	6.1	4	0.49	0.65	0.63
Reefer	1 x 23,200	120	6.1	4	0.42	0.59	0.56
Lash	2 x 21,600	134	5.5	4	0.79	1.06	1.01
Ro-Ro	1 x 10,400	187	4.2	4	0.66	0.90	0.87
Frigate	2 x 26,000	260	3.9	5	0.26	0.37	0.34

elastic deformation. This in turn may have an adverse effect on the linearity of the bearing pressure distribution. There is also the danger of fretting if cyclic deformation subjects the bearing surface to a small amplitude/high frequency movements.

The trunnion bearing concept furthermore allows the openings in the hub body to be kept relatively small compared to the collar type bearing where the "width" is needed to support the force couple; the smaller opening improving the hub rigidity.

ii) Dynamic loads cause bending stress in the trunnion

While the outer hub structure remains little affected by the large dynamic forces described, the dynamic loading is absorbed in an obvious and well defined way in the trunnion. Such a feature is welcomed as the necessary margin of safety is easy to demonstrate. It has been noted that this portion of the design duplicates the situation of the hub - propeller flange connection, which is also subjected to alternating loading due to propeller weight and blade thrust variations induced by the wake-field.

The trunnion which is subjected to tension (centrifugal-force) and bending can, with advantage, be designed or shaped to yield low stress concentration factors, and compound radii are most favourable. Figure 3 shows a less than optimal fillet contour producing a measured SCF of 1.6, while values of approximately 1.35 can be achieved within the confines of the structure of a hub. This standard case was taken from PETERSON-Tables.

With the stress concentration factor and the loading (static and dynamic) known, the safety factor against yielding and fatigue failure at usually 10^8 cycles can be calculated either from the Smith diagram of the material used, or by comparing an equivalent stress

$$\sigma_{EQU} = \sigma_{ALT} \cdot \frac{\sigma_U}{\sigma_U - \sigma_M}$$

with the fatigue limit stress where σ_{ALT} is the alternating and σ_M the mean stress in the fillet area. Both values are computed as nominal stress times the respective (static and dynamic) stress concentration factors. Safety factor in excess of 3 is advisable, and relatively easy to achieve.

In contrast, the collar type bearing system has much higher SC-factors usually reflecting the limited space in the fillet area of the blade carrying disc, where an enlarged radius immediately decreases the bearing area and increases bending in the collar. Favourable contours yield SC-factors of equal to or smaller than = 2.5.

The same approach to a dynamic stress analysis must govern the calculations on the blade bolts and an elastic bolt diagram should be derived to determine actual stress variation at a given prestressing level. Again a good bolt design in respect of dynamic loading will include :

(a) A reduced shank diameter (0.8 of thread core diameter) of as much length as possible, allowing a relatively large compound radius at the bolt head for SC-factors not exceeding the value of 2; furthermore the bolt head should be kept rigid to avoid increasing the head fillet stress through tilting. (These points are less readily achieved with the otherwise elegant hollow More Grip bolt);

(b) Rolling of threads to reduce the SC-factor by 2 in the threaded portion;

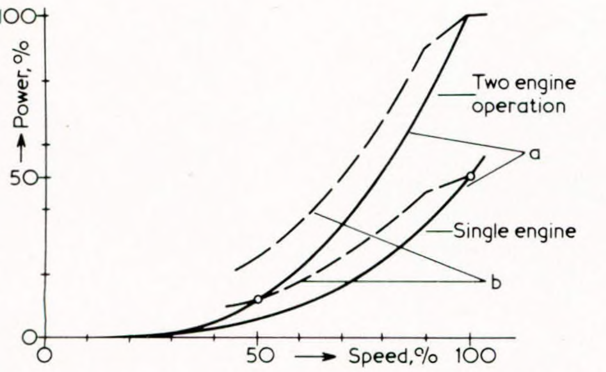


Fig. 1 — Two engine operation (medium speed) with fixed pitch and CP-propellers

Single engine operation

	Max. power Shaft	Engine	Speed	
Fixed pitch	12·5	50	50	%
CPP	50	100	100	%

a. Propeller curves b. Engine load limit curves

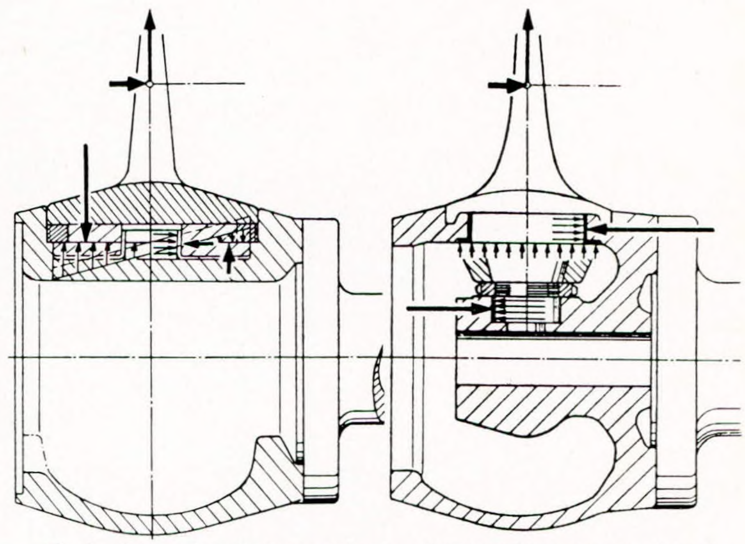


Fig. 2 — Comparison between collar type (left) and trunnion type (right) blade support designs for CPP

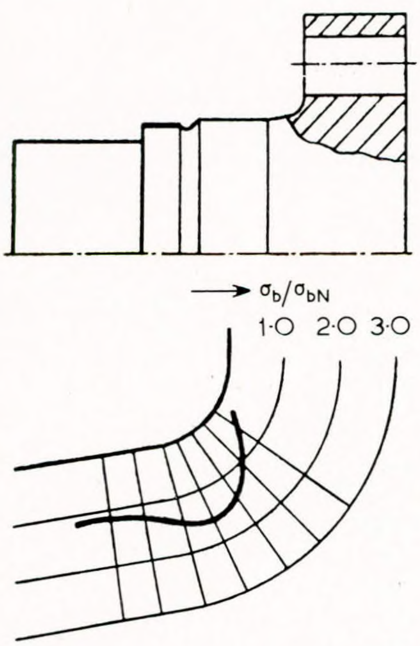


Fig. 3 — Stress in fillet area of blade trunnion according to strain gauge measurement

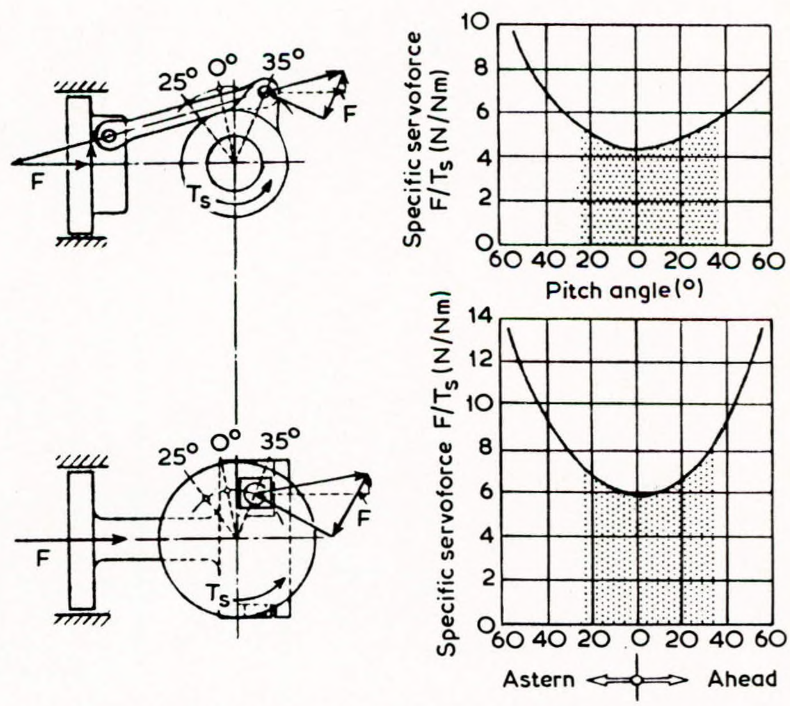


Fig. 4 — Servoforce F needed at the same blade spindle torque T_s for a turning mechanism with positioning lever (top) and sliding block (below)

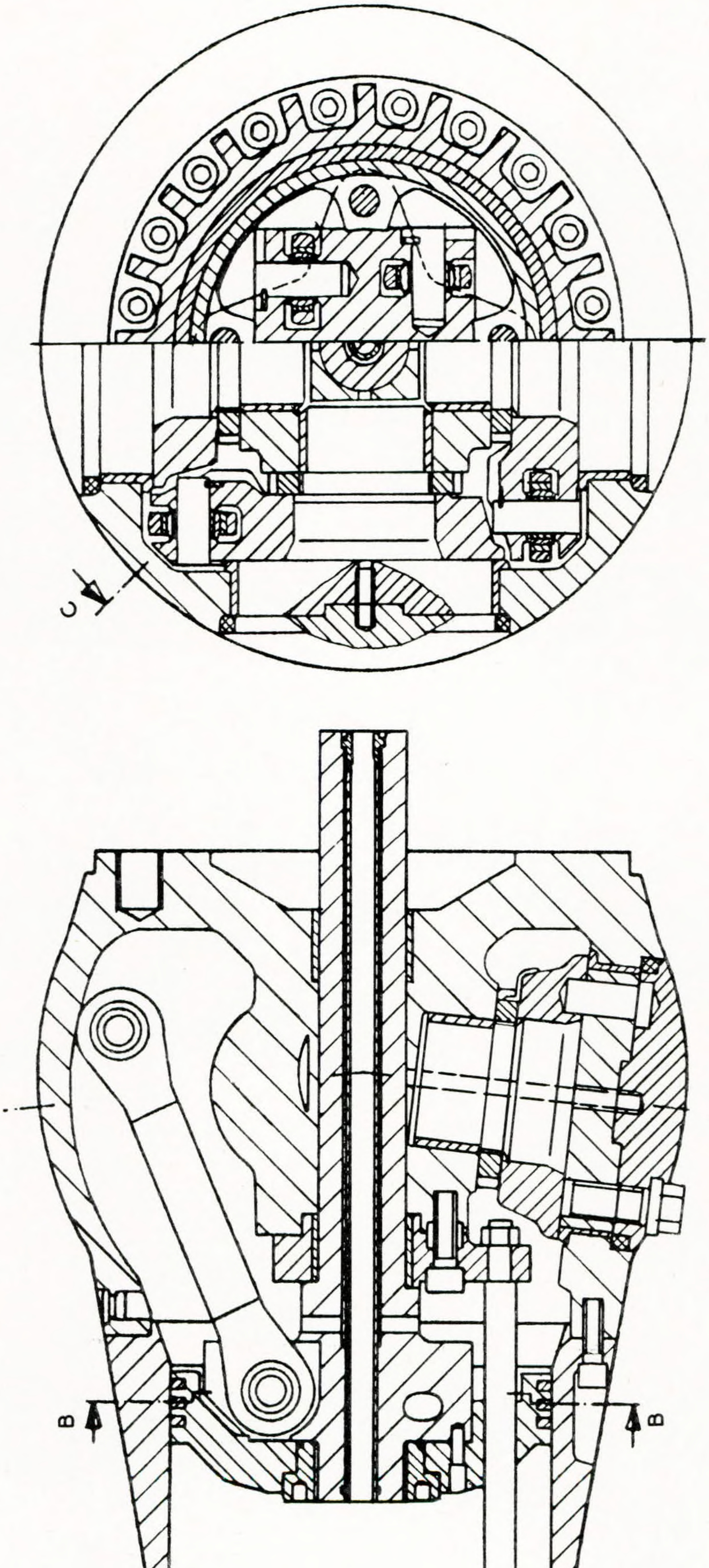


Fig.5—CPP with trunnion type hub

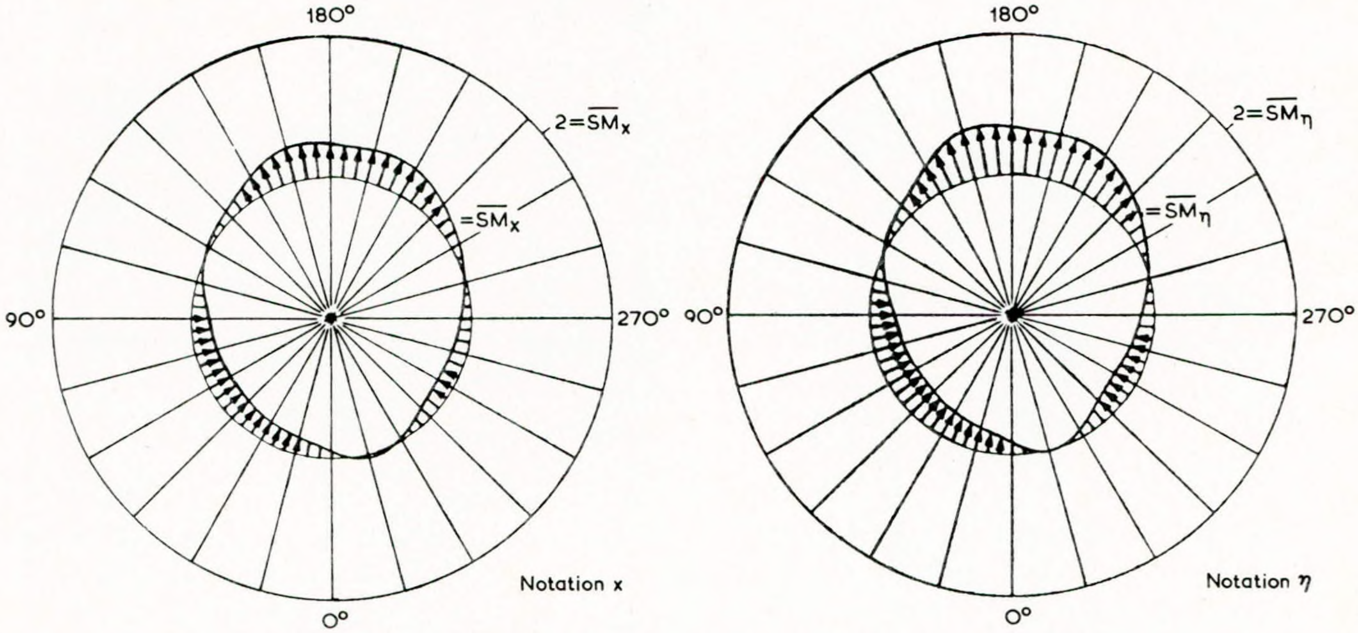


Fig. 6 — Wake field induced blade bending moments at hub radius (blade root section); single-screw reefer vessel, 23 kn, 122 rev/min, 23,200 hp, propeller diameter 6·1 m

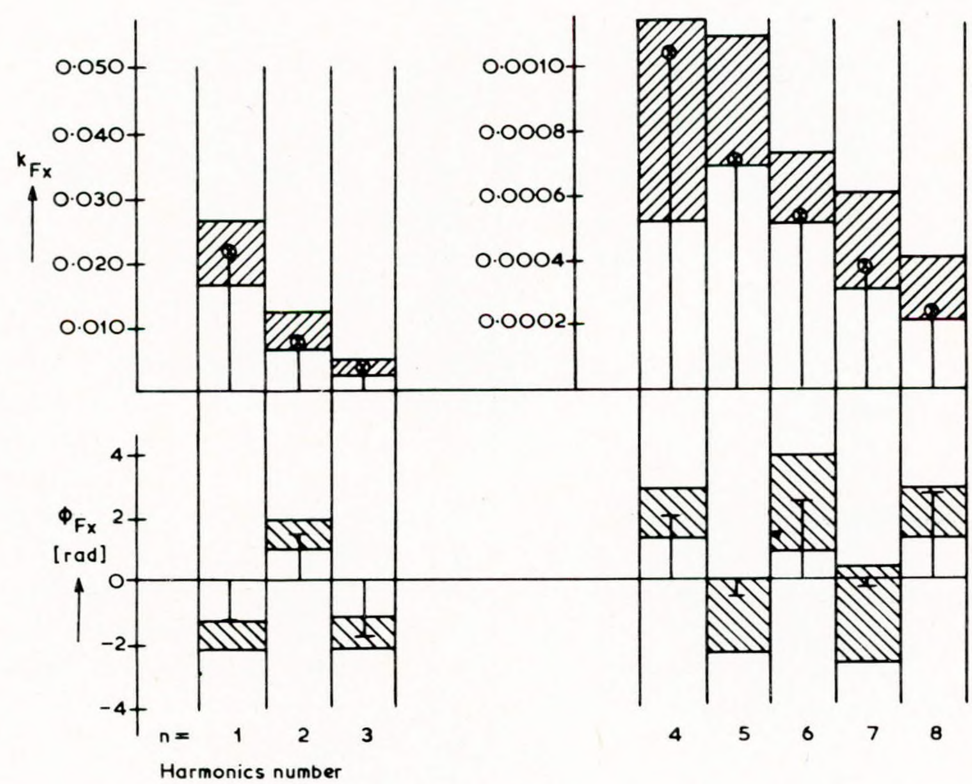


Fig. 7 — Amplitudes and phase angles of first eight axial blade moment harmonics of the ITTC specified model propeller; results of Escher Wyss programme (points) in relation to range of 3-dimensional computation results.

k_{Fx} — axial blade force / $(\rho n^2 D^4)$

(c) close tolerances on the parallelity of the bolt head and the concentricity of the thread bore to avoid distorting the bolt.

The collar bearing system with its normally larger disks (and hub openings) allows a larger bolt circle diameter than in the trunnion design, thus easing the task of the bolt designer. On the other hand, the trunnion system offers the option of using blades with cast trunnions for highly loaded applications. In such a case the blade is secured with a large nut leaving the palm undisturbed structurally and hydrodynamically.

iii) The influence of blade bearing concept on the type of actuating mechanism

For geometric reasons, the trunnion design with its smaller hub opening allows a larger radius (about 1.4) for the actuating mechanism, compared to the standard collar bearing design with the added advantage of a reduced actuating force. Also in the collar bearing design, sliding blocks and a single shear bolt is employed contributing to this problem as the single shear pin has to be made considerably bigger. The single shear pin needs careful design in the fillet area to keep the stress concentration within tolerance in view of the dynamic load resistance. Also there may be a complex reaction of stresses from the bolt, the blade disc collar and even the blade bolt thread. In the trunnion concept, the actuating mechanism stresses are not passed on to the blade trunnion.

Figure 4 shows the actuating forces in both designs.[1] Figure 5 shows a cross section of a trunnion type hub.

2. COMPUTATION OF WAKE-FIELD INDUCED BLADE FORCES

The effective dynamic loading of the system is needed for the hub stress analysis. The author's company has developed its own computer program[2] using the following data:

i) the wake field data, axially and peripherally;

ii) any shaft inclination or turning effect;

iii) the blade geometry.

The stationary and fluctuating loads on the blades can then be calculated as well as on the propeller in all axes (3 forces and 3 moments). The program has sufficient capacity to allow it to be used for routine applications (computing time on an IBM 370-125 approximately 3 minutes).

Figure 6 shows the computer graphs of the bending moment at the hub radius in the form of a polar plot, ie. the momentary values of the bending moment as a function of the turning angle through one revolution of the shaft. Notation x refers to the shaft axis, notation η to the axis perpendicular to the shaft (revolving with it or the blade respectively).

The resultant moments are derived by returning to the absolute values in both axes, vectorially adding and relating again to the mean resultant force. A Fourier spectrum of the output is also available.

2.1 Some Remarks on the Program

The non-uniform wake data, axial and peripheral, is first harmonically analysed. Plotter options are arranged to produce the inflow to the propeller in various forms as desired.

The computation of the transient efforts is then based on the Karman-Sears type two-dimensional instationary flow treatment at cylindrical blade sections, whereby not only the transverse, but also the longitudinal disturbance flow according to Horlock's stationary two-dimensional flow theory is considered at each blade profile. The latter is not usually included in this type of approximate program. The correction, ie. reduction of lift effectiveness as the major effect in the real three-dimensional flow versus the two-dimensional theory is effected through factors given by Breslin (1970).

2.2 Comparison Between Design and Test Figures

A welcome proof of the reliability of any computation is a satisfactory correlation with test results.

The propeller committee of the ITTC has reported on the results of detailed measurements on a model propeller of published geometry in a known wake field. Fig. 7 shows the amplitudes and phase angles of the first to eight axial blade force harmonics obtained from three dimensional and consequently more detailed calculations which define the limits of the hatched area.

A set of full scale test results were carried out in Japan on a CPP ship with strain gauges located on the blade root section with the cables led into the ship via a terminal box at the aft fairwater cap and the rudder stem.[3] The wake field of this propeller was known and a comparison of results is as follows :

i) Theoretical results using a computer

Bending moment fluctuations at the hub radius was $\Delta SM/\overline{SM} = 1.27$

which was also the designed stress level as the bending moment and stress were proportional.

ii) Measurements taken in a ship

The result was : 1.32

The theoretical and practical results are, for all practical purposes, the same and quite satisfactory for the fatigue strength analysis of the hub blades and hub components, bearing in mind that a high safety factor is used in the dynamic analysis to allow for other uncertainties.

REFERENCES

1) SCHANZ F and WUHRER W "Controllable pitch propellers for Naval Vessels
2) C.P. THOMSON, ESCHER WYSS - Computerprogram PROCAL
3) KAWASAKI - H I Internal Report, May 1972

DISCUSSION

MR P H GEE BEng FIMarE congratulated Dr Wührer on his paper and in particular on the presentation.

Manufacturers invariably considered that their designs were superior yet this could only be proved by their long term reliability in operation.

He would like to outline briefly one or two failures, which had recently been investigated by Lloyd's Register Technical Investigation Department.

The first involved a controllable pitch propeller unit incorporating the trunnion or journal type bearing. Failure was found to have occurred in all four trunnions and on each it was found that a bending fatigue crack had propagated from deep corrosion pits, present in the recessed fillet, between flange and boss on the pressure side. Metallurgical examination of the trunnions showed that both chemical composition and mechanical properties conformed to the classification rules for carbon steel forgings. The cause of the damage was attributed to salt water ingress into the propeller hub.

Another investigation involved the alternative collar type blade bearing. It was found that a series of crank pin rings had failed again due to corrosion fatigue, originating in the fillet between the flange and the boss and attributed to salt water ingress.

It was appreciated that since these particular failures, manufacturers had made some improvements in altering the material of the crankpins and modifying the sealing rings.

As the result of a computer analysis of failures to c.p.p. units which have been in service for five years it was concluded that the incidence of failure for both systems was approximately the same, that is, in the region of 86% fault free operation. To be fair to the trunnion type propeller manufacturer it must be stated that out of some 637 units classed with Lloyd's Register only 72 were of this type, therefore only a very small increase in numbers with low fault conditions could have given a very different picture.

One other type of failure, occasionally encountered, was that of crevice corrosion in the blade flange pockets above the sealing ring. Would the author please comment if his company had any knowledge of this, and if so, what were the recommended repair procedures ?

It would appear from both Mr Gee's survey experience and the statistical evidence that the main failures lie in the following areas:

1) poor material castings;
2) damage due to impact;
3) failure of blade securing bolts and of the blade flange in way of the bolt head recess;
4) failures due to salt water ingress into the hub.

He understood that 1) had now virtually disappeared due to improved casting and inspection techniques.

With regard to impact damage, this could not always be avoided but by harmonization in design between the various components it should be possible to contain its effect.

For a c.p.p. the replacement of a single blade was a relatively simple operation in that the number of components involved was minimized; consequently the strengthening of other components to withstand the maximum load of a collapsing blade would appear reasonable.

Would it be correct to assume that the blade securing bolts were intentionally designed as ultimately the weakest component and could this have an effect upon their relative fatigue life, so contributing to the quite high incidence of broken bolts which had sometimes led to more serious failures.

It was understood that for ice conditions the author's company fitted blades with an integrally cast trunnion. What then was the likely effect of ice on a blade and was the operating mechanism designed to withstand such a condition.

Mr Gee suggested that based upon the fact that neither type of blade bearing would appear to have a clear advantage, the manufacturers should make efforts to improve blade sealing and to possibly modify the blade securing arrangements, thereby eliminating the possibility of sea water ingress into the hub, which could enhance the conditions for fatigue failure regardless of how carefully the components were designed.

MR C A LINDAHL MSc said that in his paper Dr Wuhrer had mentioned the two main design concepts for c.p. propeller hubs used today: the "collar" type design and the "trunnion" type design. Of these two designs, the "collar" type was used with success by the majority of the leading c.p. propeller manufacturers. It was believed that the company Dr Wuhrer represented was the only one using the "trunnion" type design. Lips had used both designs but had now settled on the "collar" design according to a paper by Mr Wind of Lips.*

Although the "trunnion" concept was used by a number of water turbine manufacturers, this design was not generally considered optimal for c.p. propellers.

In Dr Wührer's paper, suspicions were cast on the "collar" type design with regard to lack of strength and rigidity of the hub structure. The blade ports and the open aft end of the hub were mentioned especially. Hubs with opened as well as closed aft ends had been used by the writer's company. Experience showed that neither of these hub types were exposed to any adverse deformation, which possibly could risk the performance of the hub structure or the bearings.

However, with the dimensioning used today for the "collar" type designs, the stresses in the outer hub structure were of such a low magnitude that nothing useful could be achieved by further reducing the stresses at the blade ports or at the aft end of the hub.

There was therefore nothing to gain by reducing the diameter of the blade ports in the hub. Instead the larger the blade flanges could be made the better the blade root configuration would be. A large blade flange also provided a better bolt connection. The design with bolted on propeller blades always had an advantage with regard to easy replacement of blades.

Figure 8 showed an example of the influence of the blade figure diameter on the cavitation margin at inner blade sections. From this comparison it was clear that the safety against cavitation was appreciably lower for the blade with the small flange, and consequently thick root sections.

The standard design in the "collar" concept for the actuating mechanism was "sliding block - crank pin". Thorough studies had indicated that this design utilized the confined space in the hub in an optimal way. The crank pin radius was smaller. This resulted in the actuating force being larger for a certain spindle torque. However, the resulting stresses and bearing pressures were lower than in the design with levers and links as used in the "trunnion" concept.

In Figure 9 the completed diagram of Dr Wuhrer's Figure 4 showed the specific servo force in the two different hub concepts with two diagrams showing the corresponding stresses and bearing pressures.

The comparison was based upon the diagram for specific servo force given, and the actual dimensions of the actuating mechanism in hubs of the two different concepts both with the same outside diameters.

It was noted that the author's company had their own computer programme for estimating dynamic forces on the propeller blade. Comparison with good conformity had also been made with measurements on one full scale propeller.

It would be valuable to obtain more details of the ships mentioned in Table I and explanations for the different designations could be given. He assumed $\Delta SMx/SMx$ referred to the propeller thrust and that, therefore, the resultant $\Delta SM/SM$ should be closer to $\Delta SMx/SMx$ than to $\Delta SM\eta/SM\eta$.

At KMW the dynamic forces on a propeller were as a routine calculated by means of a computer programme. However, for several projects these calculations were followed up with model test in the company's cavitation laboratory. Special 5-component dynamometers were used for measuring the dynamic forces on propeller blades. The five dynamic components measured by the dynamometer were:

a) blade thrust;

b) bending moment of blade thrust;

c) transversal force in torque direction;

d) bending moment of transversal force;

e) spindle torque

The tests were made in behind conditions with simulated full scale wake. The close contact between testing and calculation was essential for safe designing. Figure 10 showed a comparison between measured and calculated thrust on a propeller blade.

It was, however, not only the wake field which exposed a c.p. propeller to fatigue and vibratory loads, but ice conditions could be much more severe with regard to fatigue.

It was well-known that, for the US Coast Guard, Escher Wyss had built the biggest c.p. ice breaker propellers in the world so far. It would be interesting to hear something about the methods used for determining the dynamic forces for these big propellers.

* "International Shipbuilding Progress" February 1971 number 198

MR M S BRADLEY MIMarE wished to refer to three points in Dr Wührer's paper:
The author had quoted an expression for equivalent stress which he had compared to the material yield stress to give factors of safety in the order of three.

This expression appeared to derive from the Goodman criterion, but with the use of yield stress instead of endurance stress. Mr Bradley wondered if the author would not consider it more appropriate in his comparisons if the endurance limit of the material were to be applied since the results would be more applicable to the use of alternative materials in construction. The apparent factor of safety presently derived incorporates an allowance for the ratio of yield to endurance stress, and this was not necessarily the same between differing materials.

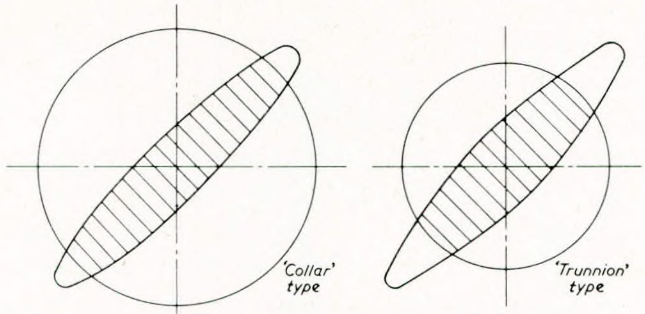
He would be grateful if the author would provide further details regarding the pre-stressing of the blade bolts and their elastic diagram. This was mentioned in passing in the paper and it was a pity that such an important topic had not been more fully dealt with in view of the high vibratory loads described. Might he suggest that the author should incorporate his verbal presentation on bolting into his reply to the discussion, thus enhancing an already interesting paper.

Finally, a point of interest: the ratio of fluctuating bending moment to mean bending moment at the blade root in the X direction given, example, for the LASH vessel in Table I as 0.79 would seem to imply a shift in the centre of thrust of approximately 13% of the propeller radius, and Mr Bradley wondered if this was a correct interpretation.

CORRESPONDENCE

MR J S CARLTON MIMarE wrote that the author had produced an interesting paper dealing with the philosophy behind his company's particular design of controllable pitch propeller, which, no doubt, would provide some lively discussion throughout the propeller industry.

The design of a controllable pitch propeller brought together the two disciplines of mechanical engineering and ship hydrodynamics often with a certain degree of conflict arising in the area of the blade palm. As the author had correctly pointed out, from a mechanical engineering viewpoint, the hub design with the smallest opening for the blade palms was the most desirable in order to achieve the greatest strength in the outer regions of the hub body. However, on the other hand, the hydrodynamicist always looked for the greatest palm diameter in which to site his blade, this having already been constrained in its form due to the blade to blade interference considerations. Some advantage could be gained by allowing the root sections to "overhang" in terms of blade strength; this effect being shown in Figure 11 by means of an increase in the section modulus coefficient of a particular N.A.C.A. series section for various amounts of



C.P. propeller with maximum section length

		'Collar type'		'Trunnion type'	
r/R	Section length	Section thickness	Cavitation margin %	Section thickness	Cavitation margin %
0·35	982	126	23	152	7·5
0·40	1104	111	21	130	9·5

$$\frac{\text{'Collar' type flange dia.}}{\text{'Trunnion' type flange dia.}} = 1{\cdot}33$$

Fig. 8 – Influence of blade flange diameter on cavitation margin (Example)

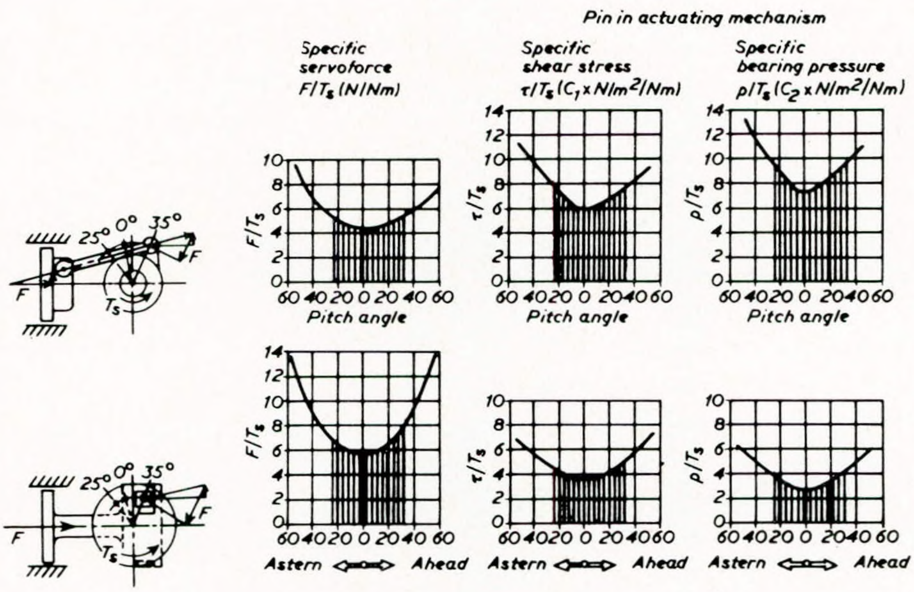


Fig. 9 – Servoforce F needed at the same blade spindle torque T_s for a turning mechanism with positioning lever (top) and sliding block (below), as well as corresponding shear stress and bearing pressures

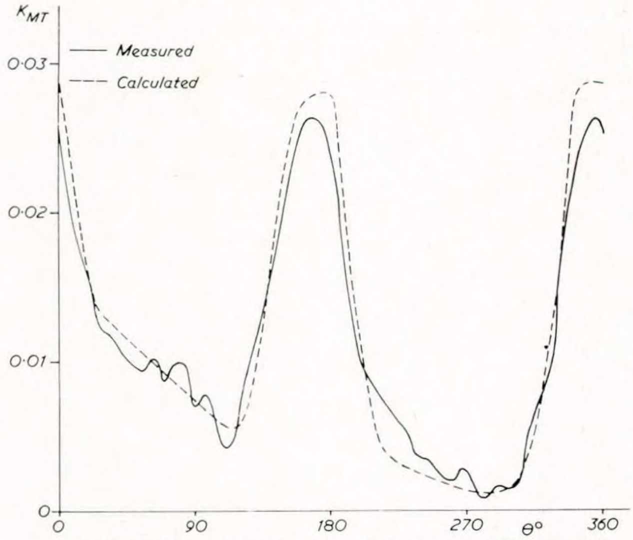


Fig. 10 – Comparison between calculated and measured dynamic bending moment of thrust

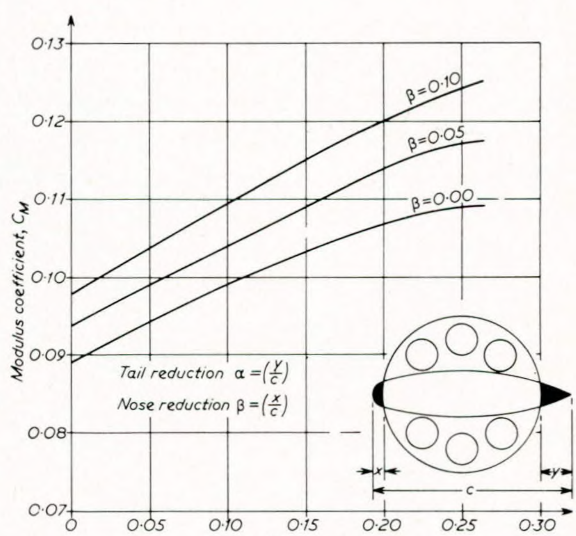


Fig. 11 – C.P.P. root section modulus coefficients

overhang of the leading and trailing edges. This process must not be carried too far for obvious reasons. Furthermore, another problem associated with the design of blades which were to be sited on relatively small palms were the stress concentrations arising from the penetration of the actual root aerofoil section profile by the blade bolt holes. This latter problem could, as mentioned in the paper, be very satisfactorily overcome by the use of cast trunnions, however, this did not appear to be the normal practice for the higher power c.p.p. applications in which the blade bolt system seemed to predominate. Would the author care to comment further on these particular aspects of the design of c.p. propellers in relation to the constraints used in his particular design ?

Blade bending moment fluctuations were frequently less for twin screw ships than for their single screw counterparts due to the greater uniformity of the induced wake field. This general, although by no means infallible, rule was adequately borne out by the Frigate results in Table I, since these types of propellers normally operated in what could effectively be considered to be open water. It was interesting to note therefore that in the case of the other twin screw vessel mentioned in the Table we observed the greatest bending moment fluctuation of all of the examples cited. Consequently, it would be interesting to learn of any particular anomalies in the wake field of this ship which might have been responsible for this apparently large increase in bending moment fluctuation.

Some controllable pitch propeller designs and proposals emanating from the author's company had over the years been notable for the composition of the total propeller rake by means of both rake of the blade relative to the palm and also of the palm relative to the plane of the shaft centreline, indeed, Figure 5 of the paper demonstrated this latter component of rake. Now, in the case of unraked propellers it had been shown that the cylindrical blade sections underwent considerable deformations of profile when the pitch was changed, consequently, having a similar effect on the hydrodynamic properties of zero lift angle, lift slope and pitching moment. It, therefore, would be of interest to learn what additional effects the two above mentioned components of rake have upon these hydrodynamic properties and how such changes are accounted for in the author's program, since this, as indicated in the paper, was based upon a cylindrical blade section analysis rather than a Hess type approach. The author also gave examples of the excellent correlation obtained between his computed results and the results of his full scale trial analysis, both of which are presumably at the design condition. Could the author perhaps give some examples of the extension of this correlation into some of the important off-design conditions normally met with in controllable pitch propeller operation.

In the paper much comment had been made of the advantages in terms of the steady state and dynamic stress distributions achieved by use of the trunnion c.p. propeller mechanism over the collar type. Consequently, it would be of interest to know more of the development stages that had taken place during the avolution of this design and also to learn something of the author's experience with the wear rates of the various members in the trunnion linkage as opposed to those of the collar type.

AUTHOR's REPLY

Dr Ing Wührer replied that Mr Gee had understandably suggested that the merits of any design should be reflected in a lower failure rate as compared to others. The author could only agree that it was of supreme importance to the operational success of the c.p.p. that the makers should conscientiously observe the rules of reliable detail design and quality control in such items as seals, because these elements - if they should fail - could obscure the fundamental soundness and/or the principal advantages of a given concept.

In the author's company's case and in relation to the cases of sea water intrusion mentioned by Mr Gee, the seal design did not need to be changed but a stringent quality control was introduced. This included the measurement of the compression set module of the endless, molded rubber ring, which for good, lasting elasticity should not exceed 10 to 15%.

Furthermore, the natural aspiration (uncharged) closed loop hydraulic system which allowed pressures below the head tank level to occur under certain operating conditions had been changed or (for other reasons) abandoned.

Some experience had been acquired with crevice corrosion in the hub above and outside the seal rings in older designs, particularly in stainless steel. The obvious repair was to insert a separate ring to "replace" the corroded surface. Crevice corrosion was not found in carefully electrolytically protected propellers and when the clearance between hub and blade was neither too small nor too long. Also according to experience a second "outside" O-ring could protect the large diameter seal ring from this kind of corrosion or other adverse environmental influences. The author's company's propellers showed these features.

Subsequently, Mr Gee had asked whether the blade bolts were, or should be, made to fail next after the blade was damaged in bending upon impact. This was in fact attempted according to what had become known as the pyramid of strength. Certainly there then was the danger of having bolts which were not strong enough in respect to the necessary fatigue life, hence the emphasis on the knowledge of the dynamic forces and the careful detail design of the bolts in respect of the stress concentration effects mentioned in the paper. As complying with the pyramid of strength was prudent, a detailed dynamic analysis of bolt strength was required. Also the threads must be of superior strength by a sufficiently large factor in comparison with shank and bolt head. There was no doubt that both the pyramid of strength requirement and sufficient safety with respect to fatigue life could be achieved.

On the subject of the hub for an icebreaker propeller, for which the author's company would recommend fitting of integral trunnions, could it be said that the structure was designed to withstand forces that bend the blade in the weakest direction. As the section modulus of the blade - which applied when it was subjected to spindle torques - was excessively large, the above reasoning could not be fully followed up in respect of the mechanism. The mechanism must instead be designed to withstand the operational iceforces with an as large a safety margin as attainable (see some further comments in the answer to Mr Lindahl's comment).

MR LINDAHL suggested that except for the author's company all major c.p.p. makers applied the collar type design, implying that this situation had developed through a basic process of evaluation and long term development. Whereas the author was of the opinion that the background of this situation was, in the author's opinion, mainly historical. The author's company, a leading water turbine maker, had designed and produced the first hydraulically operated c.p.p. back in 1934, along the lines of its Kaplan turbine design which was then, and is today, based on the trunnion bearing concept also and, after having carefully checked the merits of both systems periodically, they were still convinced of its special suitability in c.p.p. design.

KMW, on the other hand, had (to the author's knowledge) been the only water turbine maker who applied the collar type bearing. This being adopted for KMW's marine propeller, which - due to the leading position of KMW in the first post war years - had influenced the then newcomers in the c.p.p. market, generally fixed pitch propeller foundries, to opt for the collar type design. The wide spread use of the collar bearing therefore did not hold technical proof of "optimal" features for a c.p.p. application. More specifically Mr Lindahl referred to the larger blade palms as one advantage of the collar concept stressing the cavitation aspect.

The author commented that this drawback did not apply in the majority of propellers, but was met, where necessary, by providing larger blade palms (see Figure 12) with only marginal infringement of the favourable width between ports which the trunnion concept allowed.

Mr Lindahl's denial of any adverse effect of the very much larger hub ports on the rigidity of the hub structure was contrary to both the reasoning in this paper and some structural hub failures (cracks) between large ports which had become known. Some publications[1] described

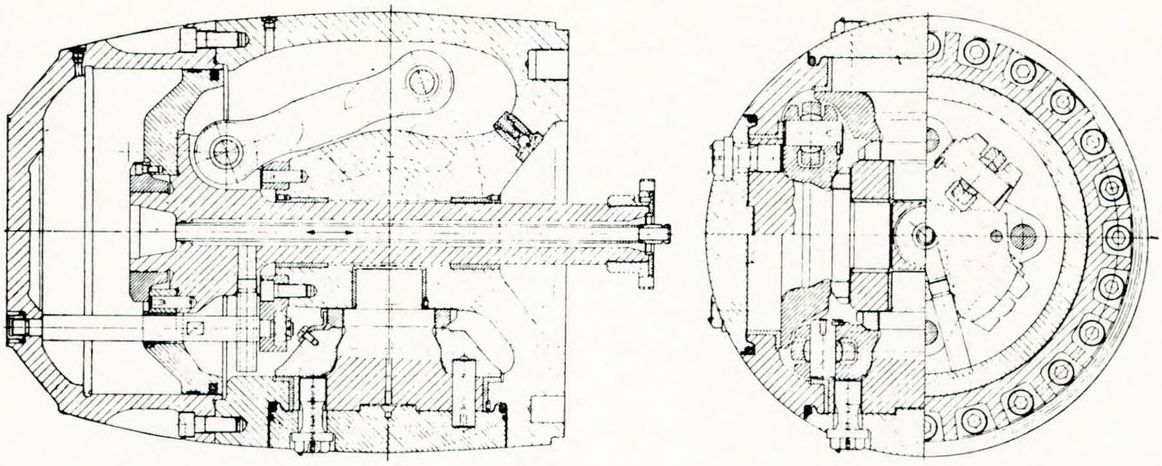


Fig.12 — Hub for fast naval craft. Power 2 x 8500 kW (11 500 PS at 640 min^{-1}. Hub diameter 665 mm

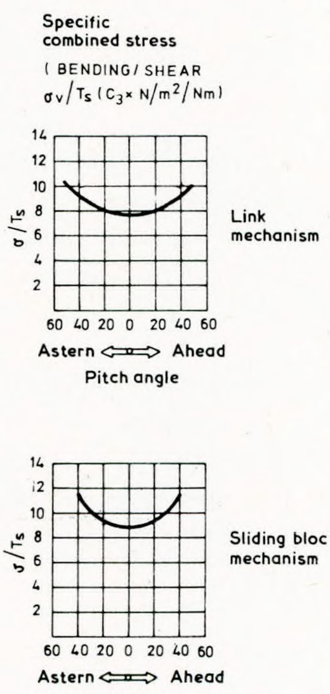


Fig.13 — Comparison of stress in pin of actuating mechanisms (influence of frictional forces in the blade bearings not included)

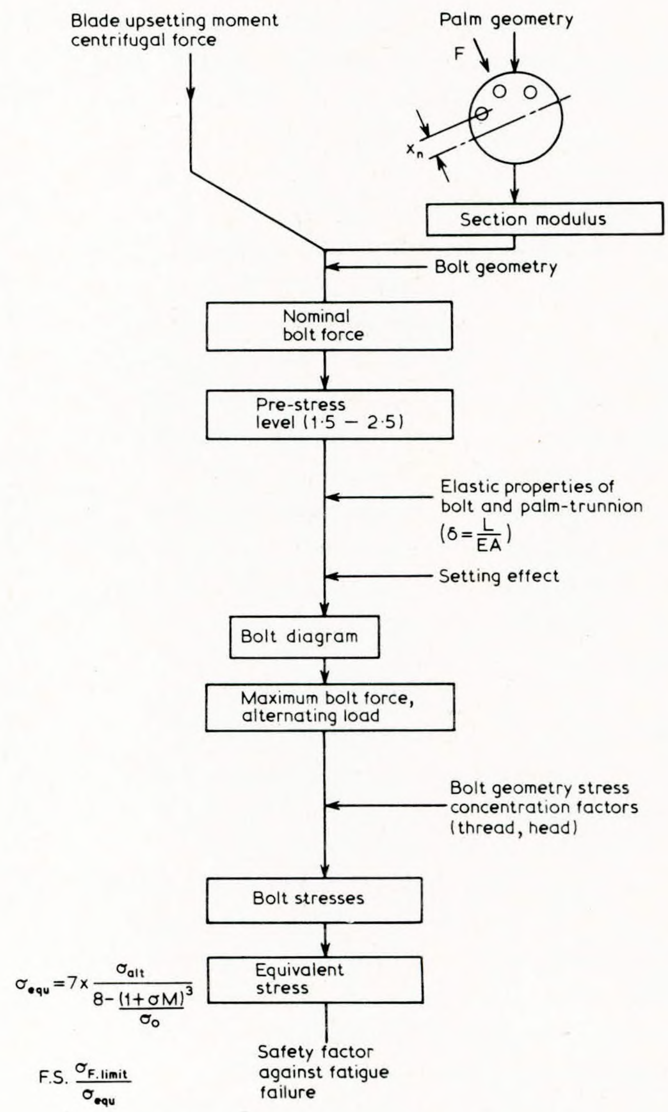


Fig.14 — Dynamic bolt stress analysis

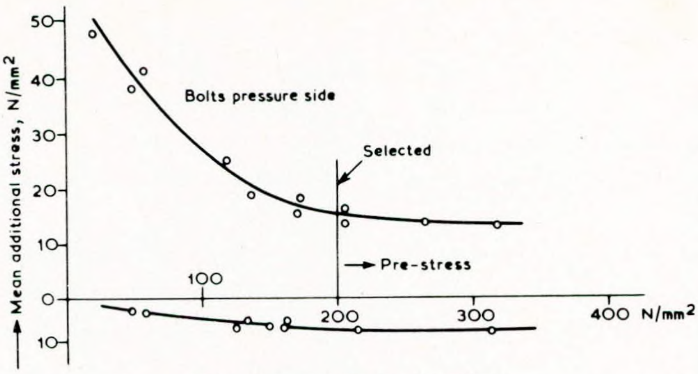


Fig.15 — Selecting the pre-stress level

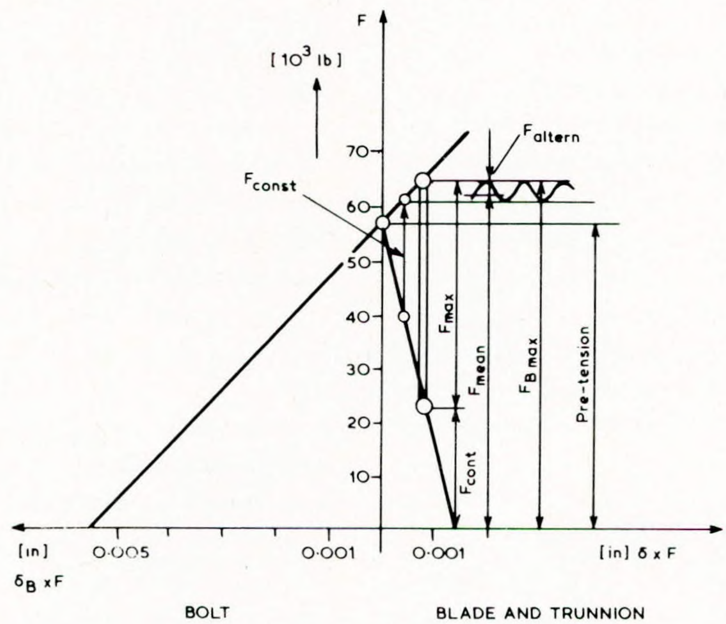


Fig.16 — Blade bolt diagram (Example: M42)

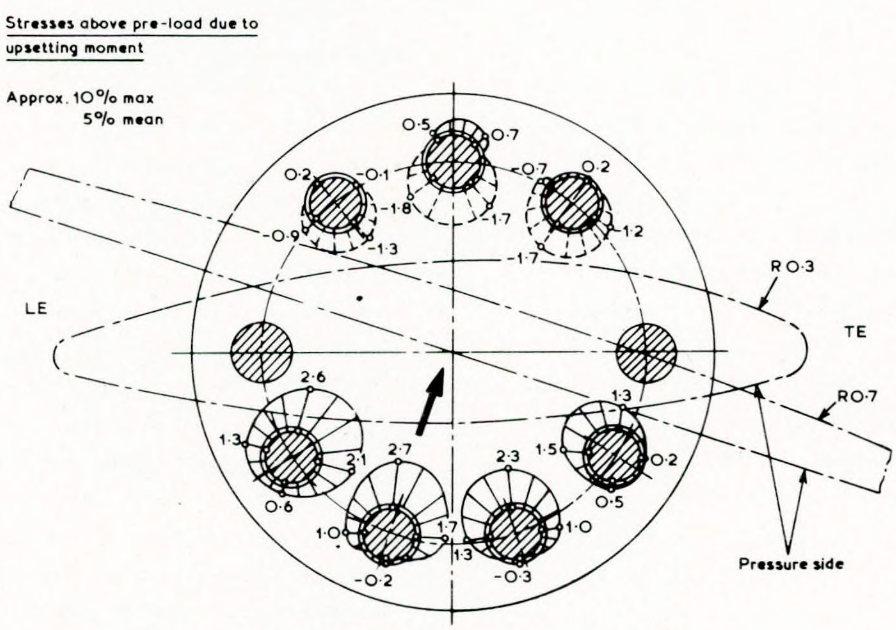


Fig.17 — Blade bolt (shank) stresses (strain gauge measurements)

The question of rigidity might be less important in a stainless steel hub of a three or four blade configuration at normal specific loading, but it was definitely accentuated in five bladed bronze hubs and also in hubs of large loadings, generally, such as in special icebreaker applications.

Subsequently Mr Lindahl had commented on the mechanism forces in both designs, confirming the forces in the trunnion concept to be lower. In an accompanying diagram, however, the pin shear stresses were shown to be higher in spite of the lower forces. This comparison however omitted the bending stress of the single shear pin in the sliding block arrangement which also sees a stress concentration factor of considerable magnitude (probably about 2.0 depending on the detail design). This reversed the comparison of Mr Lindahl's Figure 9 again in favour of the link configuration. The author's Figure 13 exhibited the comparison of the effective (combined) stress in both designs on the basis of the forces as shown in Figure 4 of the paper. In reality the forces in the sliding block mechanism were even relatively higher than indicated due to the larger friction radius. Thus the comparison of forces and stresses was rendered even more favourable.

On the bearing pressure the actual (special) bearing surface was larger than the pin area so the active bearing pressures were somewhat lower than in Mr Lindahl's diagram. Furthermore, the calculation of the bearing pressure in the sliding block arrangement neglected a possible, unfavourable deformation, while the links mechanism's spherical bearing was self-adjusting with matched hardened surfaces. Moreover, the bearing pressures were conservative in relation to the ratings in the manufacturer's catalogues.

Table I in the paper was only meant to illustrate the correlation which existed between the type of vessel and the vibratory loads to be expected, and to emphasize the importance of the designers task to cope with those forces. The only additional information which could be given was obviously the wake field - which would again be typical of the type of vessel concerned. Therefore it had not been separately given.

The related bending moment $\Delta SM\eta \,/\, \overline{SM\eta}$ was caused by forces in shaft line direction, i.e. it corresponded to the thrust component so that the Table's designation was correct, and still in line with Mr Lindahl's noting that the thrust related moment should be the higher one.

Finally Mr Lindahl had referred to the world's first large polar icebreakers with c.p.p. (3 x 14000 kW Allis Chalmers - Escher Wyss propellers). The method of calculating the blade forces including the spindle torque generally followed Jagodkin's paper(2), which distinguished crushing and cleaving forces when milling ice, and reflected the obvious fact that the forces in increase with an increasing ratio of the speed of advance/speed of the shaft - as in the tool forces on a lathe. Also a computer program was developed defining the blade in finite elements which saw varying forces due to the local "angles of attack". The forces were integrated to give the whole spectrum of blade forces as a function of the slip speed/shaft speed ratio, pitch and the rotational blade position.

In view of the general importance of the experience with these propellers under extreme operational conditions some additional comments might be of interest:

The "Polar Star" propellers had originally been designed to mill ice at a speed of advance of 3 kn in the gas turbine mode (175 rev/min). The operational advantages of the c.p.p. system became fully apparent on the first tests but due to operating the ship "beyond" the design conditions, namely at 6-7 knots and on diesel power with 110 rev/min, which dropped to about 60 rev/min when hitting ice blocks, the original links were damaged. Under a US Coast Guard contract to strengthen the mechanism in order to withstand the full scope of possible operating modes, the links were subsequently redesigned and stood up in an Antarctic mission to break channels into 2.4 m thick ice in record time. The operational results were reported to have been very favourable with no blade damage encountered due to the continuously high shaft speed. A check up of the hub mechanism had now revealed only minor remaining items such as the turning of some bearing bushings due to insufficient securing.

MR CARLTON had comprehensively summarized the conflict between the mechanical and hydrodynamic design of the c.p.p. which should not be seen as a simple alternative of either a small palm in one design and a large one in the other.

Instead a large palm could be made, if necessary, with the trunnion design as well, as was suggested to Mr Lindahl (see Figure 12). This could be achieved without notably infringing on the original hub rigidity as the overhanging palm disc area only supported the seal and was kept comparatively thin.

As to the bolt circle there was also nothing in the trunnion concept itself to prevent the realization of a given or selected (i.e. equally) large bolt circle. The inner part of the trunnion hub would still unload the outer aft hub section as described in the paper and therefore render such a hub (with the same parts) favourable in the sense described in the paper. Furthermore, a given bolt circle required somewhat larger ports in the collar design due to the required collar radii and the unfavourable superposition of bolt thread and collar stresses. Additionally, Mr Lindahl's company used a threaded-in-bearing ring requiring a still larger port, relative to a given bolt circle.

In summary it could be stated that it was not an inherent drawback of the trunnion design to have smaller palms and/or bolt circles.

Mr Carlton had inquired on the implications of the cast-on trunnion in c.p.p. applications. Obviously the cast-on trunnion avoided the blade attachment problem associated with bolting altogether but introduced some inconvenience in changing blades, as the hub cylinder had to be removed to release the trunnion nut. This procedure would tend to become more inconvenient as the units became large. With large propellers the author's company applied and preferred the bolted-on blade. In general, however, experience with the integrated trunnion type propeller indicated that a blade change had been regarded by owners as less troublesome as the major effort was in preparing the staging, rigging etc. In larger naval applications also, complete hubs could be and were removed and/or exchanged if necessary as, in practical cases of impact failures (grounding), more than one blade was usually damaged, and the hub might need to be inspected. With the large Polar icebreaker hub previously mentioned the same procedure was adopted.

The anomaly in the Lash ship shown in the Table was a wake field with a large peripheral component and might not be quite typical.

Concerning the rake in some propeller hubs produced by the author's company it might be said that this had been the introduction of a general, proven fixed propeller practice to the c.p.p. Only the author's company was in position to do this as the link mechanism alone allowed such a rake in the palm/shaft. The advantages of rake (increased tip clearances) could thus be realized.

A disadvantage was a moderate added blade and bolt loading, and a (marginal) inconvenience in manufacture and blade mounting. Of course the off-design performance was also effected, i.e. there were notable deviations from the calculated values. However, the author's company had not noted any detrimental consequences.

The computer program mentioned did not include a rake effect and might not correctly represent the off-design conditions, although the blade geometry, i.e. also the resultant geometry of the off-design position, was an element of the input.

Concerning any developmental stages leading to the present design of the linkage mechanism it should be mentioned that the linkage mechanism was the only natural actuating system available for the trunnion type bearing arrangement. The linkage could be arranged in different ways however:

i) either coaxially with the shaft, as in the older Escher Wyss designs, requiring heavier guides to absorb the resulting torque in off-design positions;

ii) or with links pointing towards the propeller axis as in the newer design, thus minimizing the said torque and simplifying the guide design.

In the newer form the author's company had not experienced any wear in the links after eliminating Teflon coated spherical bearings as used in the very first small scale application.

iii) Mr Bradley had remarked that the equivalent stress as defined in the paper should be compared to the endurance limit of the material. This was indeed the author's intention and a correction to the preprinted text had been made accordingly.

Further to Mr Bradley's comments on bolt design and prestressing a dynamic bolt strength analysis followed in principle the pattern given in Figure 14 starting out from the blade force configuration (both stationary and dynamic) and the palm and bolt geometry. After having determined the nominal bolt force (by dividing the section modulus of the attachment by the distance of the bolt farthest away from the bending axis and multiplying by the shank area) the prestress level must be selected. This should be only as high as needed to minimize the alternating force, as unnecessarily high prestress levels caused unfavourable high local stresses in the radii of the blade palm recesses.

Figure 15 showed the measured additional mean stress seen by the bolt due to the applied blade force as a function of the prestress level. It could be seen that, in the example shown, the mean additional stress could not be substantially reduced further by increasing the prestress level beyond 200 N/mm^2. A prestress factor of 1.5 to 2.0 was recommended as a general guide line.

With the prestress level selected the bolt diagram could be derived. Figure 16 showed an example and defined the various forces. It could be seen that the alternating force was found after introducing the maximum bolt force and the constant bolt force into the diagram. With the bolt forces and the various stress concentration factors known, the stress, and finally an equivalent stress, could be determined.

Figure 14 included a formula for the equivalent stress which was recommended for high strength bolts in lieu of the formula given in the paper (which had been mentioned in connection with trunnion design considerations).

After these remarks on the analysis of a given design some general design considerations relative to bolted blade attachments might be of interest:

In the above analysis, in determining a section modulus of the attachment, a uniform application of the load by the palm was normally assumed. This was an acceptable simplification only in "stiff" palms supported by a comparatively stiff trunnion flange. Figure 17 showed measurements of bolt stresses (prestress level and additional stress) in such a palm. While here, also, palm deflection caused bending in the bolt (and emphasized the importance of parallelity in the countersink) largely uneven palm thickness distributions could cause a most unfavourable concentration of almost the whole blade load in one bolt[1].

Concluding, the author wished to thank the Institute for the opportunity to present this paper and for the attention it had been given in the discussion. He was thankful for the profound and demanding questions and the opportunity to answer in detail.

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