THE INSTITUTE OF MARINE ENGINEERS

76 Mark Lane, London EC3R 7JN *Telephone: 01-481 8493 Telex: 886841*

TRANSACTIONS (TM)

SOME MEMORABLE BREAKDOWNS AND RESULTING IMPROVEMENTS

T. W. Bunyan

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Some Memorable Breakdowns and Resulting Improvements

T. W. Bunyan BSc, F Eng, FIMarE

SYNOPSIS

The author describes some of the more interesting hull and machinery problems in ships with which he has been personally involved during his career. He covers trouble shooting, redesign and improved construction with particular reference to vibration and bearings.

1. GEARING PROBLEMS

1.1 Metallurgical defects

Due to a metallurgical fault in the heart of the bull gear rim forging on a large tanker, one of the rims burst after only a short spell in service.

A new set of secondary gears was fitted and the ship left her European port to load a cargo of crude in the Persian gulf, whence she went on to Japan for discharge. The ship's Chief Engineer reported from Singapore that he had found heavy pitting, over 100 mm in length, on the aft ends of both helices of the LP secondary pinion.

Within a few days the Author was on a flight to Osaka and, after a hair-raising 40 mile taxi ride, arrived at the discharge jetty in the "neck of the woods". As the evidence suggested serious malalignment of the LP pinion to the bull gear, the tooth contact was first checked by blue marking and found to be excellent. This indicated perfect alignment when the pinion shaft was lying in the bottom of its bearings.

"Leads" were then taken of the top half bearing and it was found that the forward bearing clearance was 0.3 mm greater than the after. As the LP secondary pinion ran in contact with the top half of the pinion bearings, (not in the bottom half) this excessive malalignment error would explain the concentrated pitting on the tooth flanks of both LP pinion helices. It was remarkable that there was no cracking at the roots of the pinion teeth in this dangerous area. Immediate re-alignment was necessary for the running condition. A diminutive fitter with a carpet bag of assorted scrapers was produced. It was explained that the top half bearing had to be scraped to reduce the crown thickness by 0.3 mm, and "washed away" on the sides.

He then set to work in a most alarming fashion with a heavy scraper about half his height, causing the white metal to be dramatically ejected in long wide spiral ribbons which he "miked" from time to time in various places. He was warned that if he took off too much, the ship would be delayed three to four days and big big trouble would arise. The boss man quickly re-assured us that his man was "No. 1 class metal peeler". The strain of watching this performance was too much, so sundry of us interested parties adjourned to the Captain's cabin.

The performance was over in an hour when we were asked to inspect a most excellent blue marking test on the journal and to check that the crown thickness had in fact been reduced by precisely 0.3 mm. The ship sailed that evening.

The gear rims were made of silicon manganese steel, (C. 0.46%, Si 0.9%, Mn 1.0%, S and P 0.4%). However, in this case, due to some confusion, the steel maker had not been aware that the ring forgings were required for anything other than the hoops, normally made by them for shrinking on to the casings of blast furnaces! The trouble, which involved eight ships, with failures occuring after six months to two years service, was caused by choosing the wrong ingots.

For ingots used for forging gear rims it is vital that the non-metallic segregates are concentrated concentrically about the axis so that, when the ingot is pierced before forging, most of them are discarded. What is left must remain concentric so that it lies near the bore of the forged rim where it is totally removed by the rough machining. To centralise these segregates, uniform cooling must be achieved around the periphery of the ingot.

Fig 1 shows nine ingots in the soaking pit: only the central one is suitable for gear rims. After rough machining and smooth turning the cheeks of the rim, the soundness of the forging can be checked by sulphur printing and magnetic crack testing before the expensive finish machining and gear hobbing proceed.

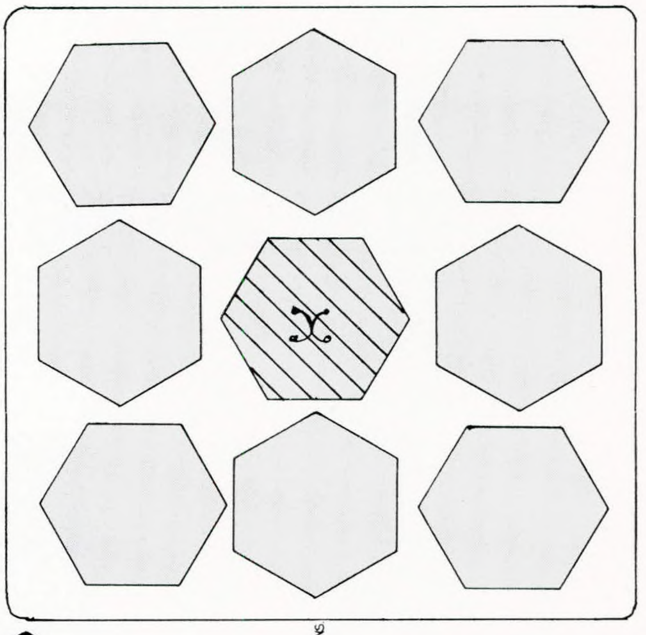


FIG 1 Ingots in soaking pit. Only X is suitable for gear rims as the remainder are unevenly cooled.

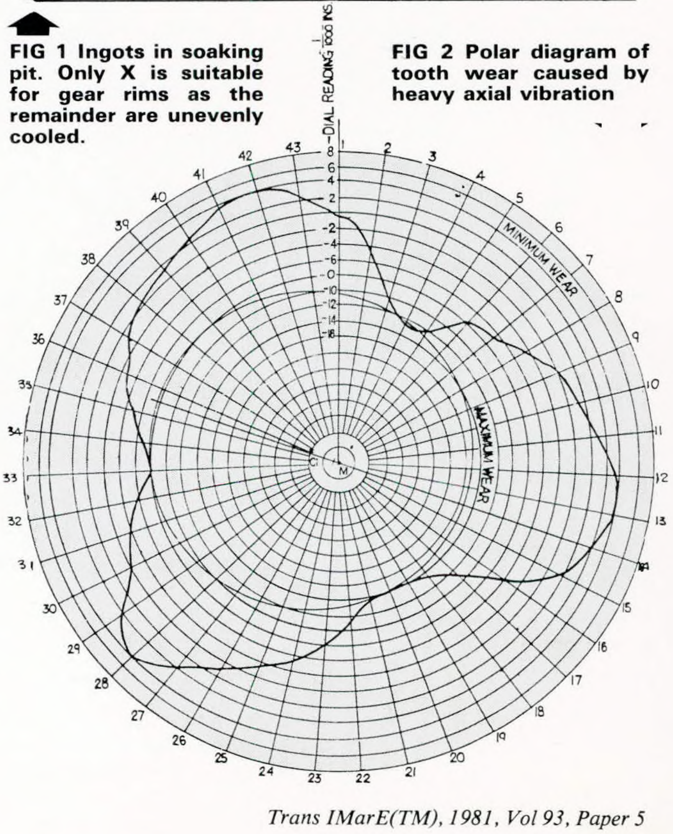


FIG 2 Polar diagram of tooth wear caused by heavy axial vibration

After a student apprenticeship and work in industry, Mr Bunyan spent the next 20 years with Lloyd's Register of Shipping, mainly as Head of their Research Department. In 1957 he joined the P & O Group with responsibility for the engineering of T. S. *Canberra* and the first P & O tanker programme. He is currently Technical Director of Pilgrim Engineering Developments Ltd. He is a past winner of the Institute's Denny Gold Medal.

1.2 Wear in gearing due to vibration

Fig 2 shows a three-lobed pattern which was worn into tooth flanks of an IP secondary pinion towards the end of the useful life of a twin-screw liner. The cyclical errors in the 12-tooth claw coupling excited a multi-noded mode of torsional vibration near the running speed.

The resulting axial vibration levels and tooth wear would reach unacceptable limits in about two years, when the pinion would be changed for the rehobbed spare. This was the cheapest practical solution, causing minimum disruption of voyage schedules, etc.

The readings from one helix, shown plotted in Fig 2, were obtained with the pinion between centres in a lathe; a dial indicator, secured to the cross slide, measured the drop of a roller whose diameter was approximately equal to the original tooth gap at the pitch circle. The readings from the other helix were very similar but approx 180° out of phase.

1.3 Claw couplings

Fine-tooth claw couplings are a vast improvement over the massive 12-toothed couplings originally fitted between turbine rotors and their primary pinions or quill shafts.

A high degree of running alignment is called for if long service life is to be had from these couplings. Indifferent alignment — usually based on the incorrect allowance for "hot alignment" — produces wear which is always cumulative and eccentric. Depending on the extent of initial alignment the rate of wear can therefore increase with unbalanced centrifugal force until a critical situation is reached in which the rotor journal reaction is exceeded by this centrifugal force. At that point in time the journal precesses around the bore of the bearing, reducing effective lubrication and causing edge loading. Complete failure of the bearing is then not far away.

An early warning sign is the low-pitched, first-order rumbling noise, noticeable during the run-up and increasing until it is swamped by the high-frequency noise emitted by the gearing.

Fig 3 shows how the wear can be measured, using a pair of dial gauges. The central sleeve assembly is levered upwards with a wooden batten (making sure there is no jamming) and the movement of the dial gauges is recorded at three angular positions 120° apart. From the maximum throw the centrifugal force at the maximum speed is calculated. If this approaches within 30 per cent of the turbine journal's design load, problems may be developing.

The cold alignment should be checked and corrected for the "hot alignment" or running condition, using the turbine makers "hot" allowances. A simple yet precise method of checking cold alignment from which declivity and parallel errors are obtainable, was developed by Mr Bryan Hildrew, Managing Director of Lloyd's Register of Shipping, when he was a member of their research team. It is known in the trade as the Lloyd's Two Clock method.

In the case of a P & O tanker it was found to be more convenient — because of tight turn-round times, etc — to have the HP fine tooth or "dental" couplings replaced by flexible metallic couplings which were fitted during a normal drydocking.

2. AXIAL VIBRATION

A most interesting case of severe propeller-excited axial vibration was found with a built up propeller which had a blade pitch that increased with propeller blade radius. This is not uncommon. However, the propeller aperture clearances were tight, as shown in Fig 4, and this was long before circular wake patterns at the propeller disc came into vogue which dramatically reduced excitation.

Calculations, based on simultaneous vibration recordings at the thrust block and the tail-shaft coupling, showed axial resonance at full speed due to very strong propeller excitation at blade frequency. It was estimated that 85 per cent of the propeller thrust was developed in the first two feet from the tip of the blade. The vibration amplitude of the

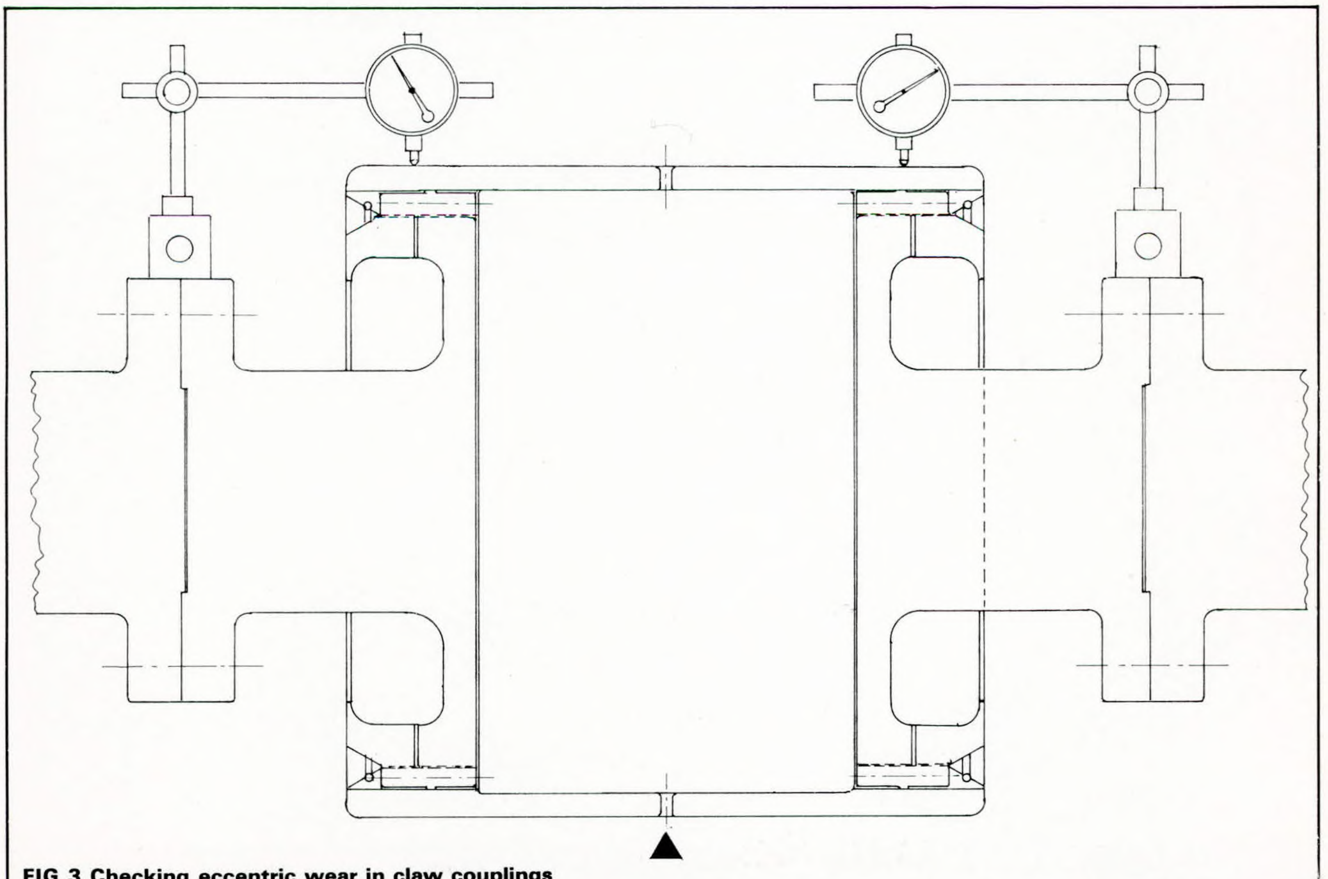


FIG 3 Checking eccentric wear in claw couplings

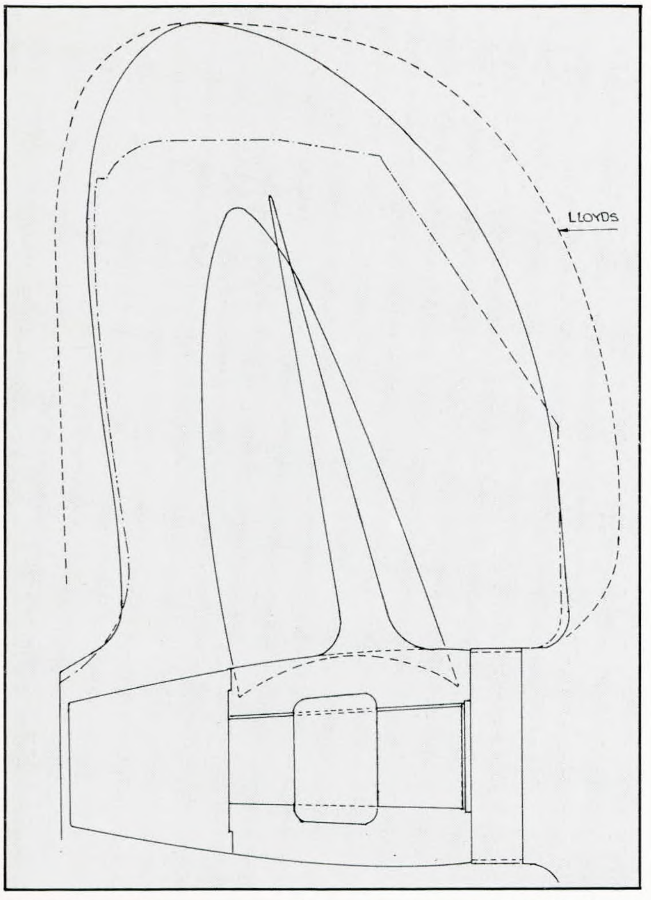


FIG 4 Propeller aperture clearances compared with those required by LRS

bull gear was nearly ± 1.5 mm and the thrust collar was in fact hammering between ahead and astern pads at resonance.

The pitting of the flanks of the secondary gearing was understandable. The cure was a re-design of the propeller. A monobloc design was used with slightly increased skewback and progressive pitch reduction from zero at 0.7R, up to 7.5 per cent at the tip. The trial after the refit with the new propeller produced dramatic results. The axial vibration amplitudes were normal and there was a 0.75 knot improvement in the ship's speed. The owners promptly decided to change all similar propellers. This meant 22 new working and spare propellers of around 25 tons each.

There was a slight embarrassment about the improved speed. The torsional stresses in the rudder stock were above Rule requirements when the rudder went hard over at full speed. A notice on the bridge, requiring a reduction in speed before going hard over, filled the bill.

Fig 4 shows the new propeller in place in the aperture. Lloyd's Rule requirements for aperture clearances are shown dotted.

3. THE STERN BEARING

3.1 White metal

The white metal-lined stern bearing is normally most reliable and will withstand a great deal of hammering in service, even where propeller-excited hull and whirling vibration are involved. The most dangerous time in the life of a stern bearing is when it is new, and particularly where a protracted run must be made at very slow speed, ie, during the prelude to dock trials or during a river voyage in light ballast from the shipyard to the open sea. In these cases the static bearing load is considerably increased because, in light ballast, the centre of thrust is usually well below the shaft centre line.

The abrasive wear, which marks every white metal lined stern bearing at the bottom is due entirely to boundary lubrication conditions which occur at very slow speeds. It is not caused by running under normal conditions. Fig 5, derived from a colour photograph, shows this "6 o'clock" wear pattern in the stern bearing of *Ardtaraig,* a P & O VLCC.

The Author is of the opinion that in single-screw ships with a block co-efficient exceeding about 0.7, the tailshaft runs in the top of the bearing at least until some most ingenious hydrodynamicist can achieve

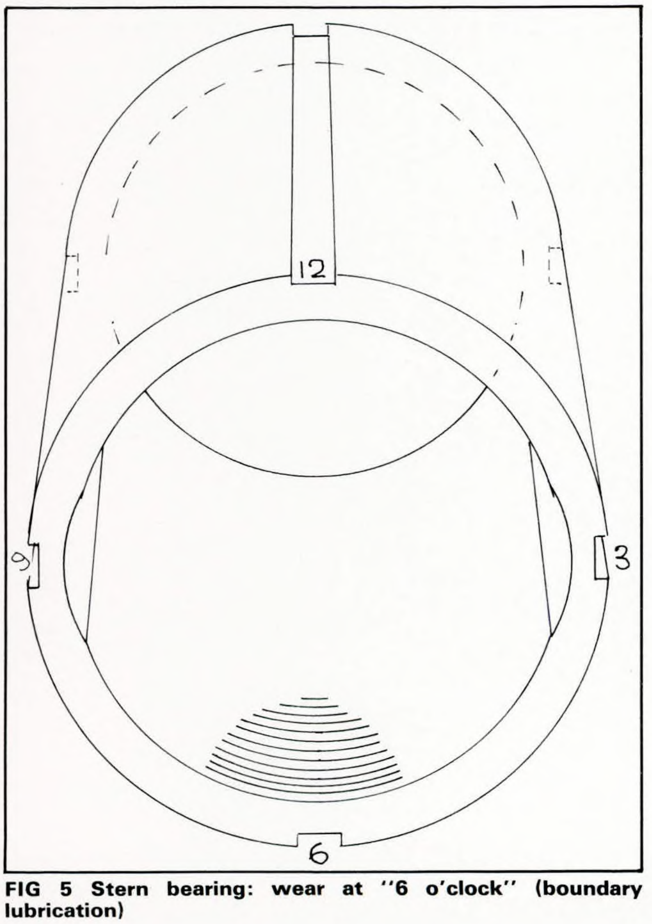


FIG 5 Stern bearing: wear at "6 o'clock" (boundary lubrication)

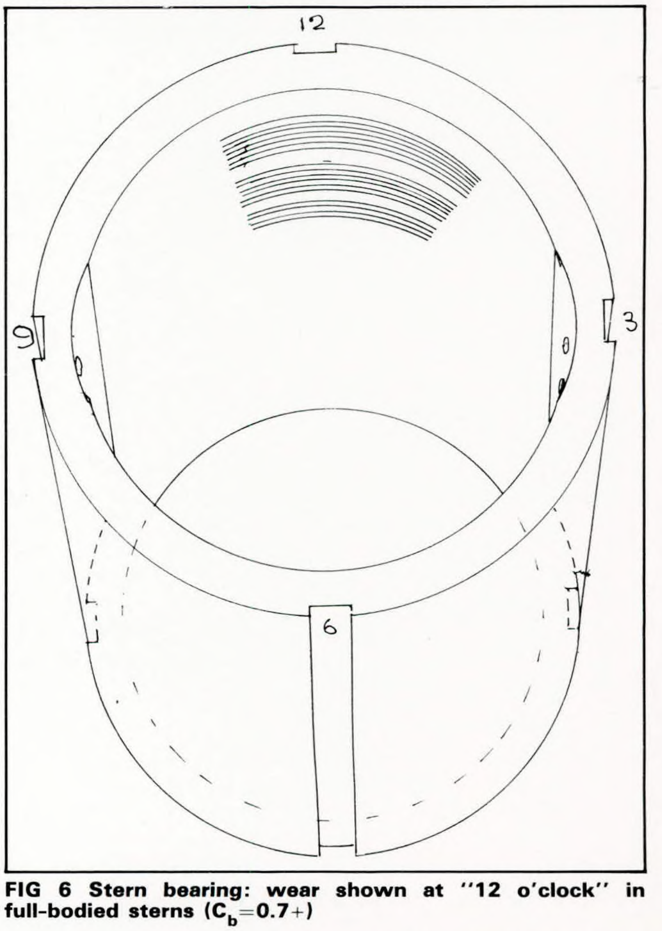


FIG 6 Stern bearing: wear shown at "12 o'clock" in full-bodied sterns (C_b=0.7+)

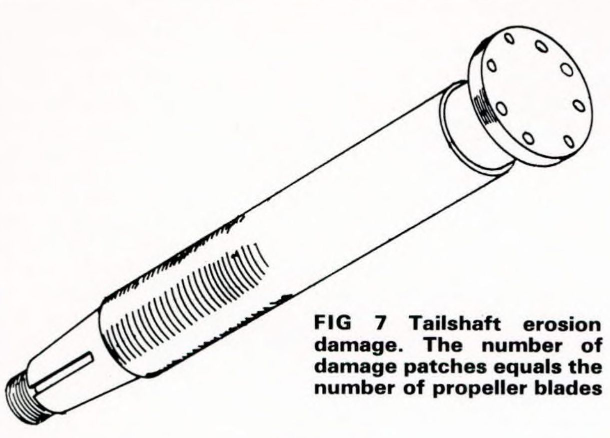


FIG 7 Tailshaft erosion damage. The number of damage patches equals the number of propeller blades

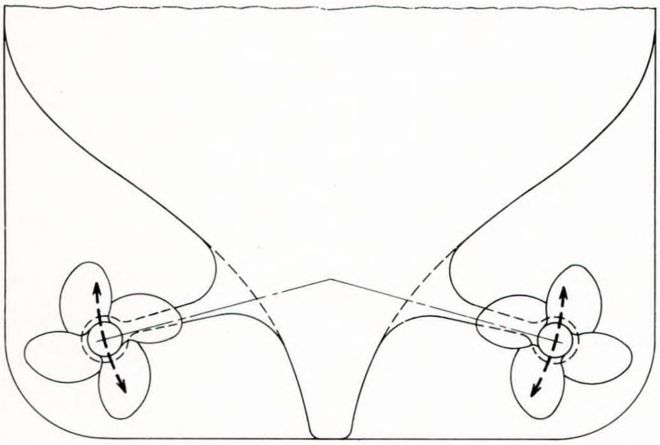


FIG 8 Arrows show mode of vibration of propeller bosses on a passenger liner

a circular wake contour distribution at the propeller disc in these full bodied sterns.

Fig 6 which is also based on a photograph of *Ardtaraig's* stern bearing, shows the marking at 12 o'clock for the running condition. Experience will show that this is typical for full bodied ships with conventional wake contours. Notice the changes of shaft attitude resulting from the differences between ballast and loaded draughts.

What may not be generally known about oil-lubricated white metal-lined stern bearings is that the cooling is almost entirely via the shaft, the bronze propeller and the sternframe boss. The contribution made by filling the after peak tank is around zero.

Table I gives the results of the tests made on a 14,000 shp general-purpose cargo ship which was run, first with the after peak full of water, then empty. Readings of a thermocouple embedded in the white metal stern bearing were taken every watch, together with the seawater temperature during the voyage from Perth to Sydney.

This situation raises an interesting possibility: namely, to seal off the lower after peak tank and retain it as a void space. To ensure its watertightness and to increase the support of the side plate panels, the space should be pumped up with a semi-rigid plastic foam. This technique has been used for many years in ships' rudders.

3.2. Water Lubrication

As we are all well aware, oil pollution of the sea is a very emotive subject, rendering a ship liable to the severest penalties, even expulsion from the area.

From time to time, over recent years, there has been talk of returning to the good old days of water lubrication when *lignum vitae* was king, all tailshafts had beautiful bronze liners, and there was no anxiety about stern gland pollution of the sea. This would be a most retrograde step for hard-driven merchant ships. The problems with water lubrication are legion, and we would throw out any hopes of extending the tailshaft survey interval, and go back to surveys every two or three years.

Fig 7 is familiar to all, except the young generation. It shows the well defined grooving around the periphery of the bronze liner, the number of patches equal to the number of propeller blades. The progressive bell mouthed wear-down of the stern bearing produces a very wide spectrum of whirling tailshaft vibrations at a frequency depending on the number of propeller blades.

So it was not uncommon that, in time, the critical frequency would occur in the service speed range with the result shown in the picture. The damage to the stern bearing would then be rapid and in a matter of days all the stock of gland packing on board used up. This is not serious in the Mediterranean, but quite another matter in the middle of the Indian Ocean. Serious damage to the support structure of the aftermost plummer block often adds to the avoidable trauma.

Table I — Temperatures of stern bearings in GP cargo ship with empty and full after peak

With AP full of water			With AP empty		
Sea (°F)	bearing (°F)	Date	Sea (°F)	bearing (°F)	Date
66.75	67.88		63.85	63.85	
66.88	67.88	3.8.80	63.86	61.85	3.8.80
66.88	66.88		60.84	61.85	
60.84	60.85		63.84	61.84	
60.85	59.84	4.8.80	59.84	60.85	4.8.80
60.84	59.83		61.84	60.84	

3.3 General Guidance on Stern Bearings

The following advice, offered in connection with stern bearings, is based on highly significant experiences.

3.3.1 On no account use white metal-lined bronze bushes with the larger shafts: they have a nasty habit of closing in on the shaft if overheated because hot bronze has twice the expansion coefficient of steel but is prevented from expanding by the cold steel sternframe boss; it must therefore yield in compression. When the bush cools down, it can, and often does, grip the shaft, with disastrous results.

3.3.2 Grey cast iron appears to be the most popular material for stern bushes but it must be de-graphitised. The Kolene process is used to remove the graphite in the surface microstructure before it is lined with white metal. Graphite "poisons" white metal and prevents an intercrystalline bond, which is highly desirable, both for good fatigue strength and optimum thermal conductivity.

3.3.3 A good "6 o'clock" bedding contact with the tailshaft is important as this ensures minimum edge loading and limits the abrasive wear pattern under boundary lubrication.

4. VIBRATION OF TWIN SCREW BOSSINGS

Following complaints from the Captain about the vibration in a large new twin-screw liner, the Author went on board during a short coastal voyage to investigate the trouble. He began the vibration test by taking records in the steering flat while the ship's speed was increased two rev/min at a time, beginning at about 15 per cent below the reported worst speed for vibration.

The test proceeded smoothly and the revolutions on both screws were kept more or less in phase by the engineers, using a simple jury-rigged phase indicator at the forward end of the bull gear journals.

Fig 8 shows a typical butterfly mode of bossing vibration. Its amplitude began to build up at about 80 per cent of full speed, and then disappeared in a few seconds. Naturally this was thought to be due to a speed reduction to cope with some minor hiccough in the engine room, so the bridge was phoned for confirmation. No speed change had in fact occurred. A check of the timing trace on the vibration record confirmed this.

The bridge was then informed that the Author was on his way up for a consultation. He explained that the sudden loss of vibration amplitude at constant rev/min and power and in good weather, could only mean that part of the ship's structure supporting the propellers had been carried away. It was therefore important to examine the after peak tank and this should be opened up forthwith.

Found below was a dramatic situation. On the port side, there was a crack in the bossing plating, seven foot long, towards the peak bulkhead; and one four feet long on the starboard side. Water was gushing into the ship briskly.

The attachment of the boss plating to the floors was a poor job and no doubt the prime cause of the panting in the plating which brought about this early failure.

Sufficient timber and other materials were prepared for the construction of two cement boxes, but they were to be kept out of sight until the local surveyors had made their inspection.

They boarded the ship, took one look below and said "well that's it — a drydocking job". The Author suggested that, as the vibration had disappeared, there was little chance the cracks would extend further, particularly as the support of the shafting and propellers was now concentrated at the cast steel spectacle piece and adjacent frames, well aft of the cracked area. Further, the Author would be making the trip to monitor the vibration. This was agreed.

The cement boxes were installed, shored up and found tight. The ship then completed the 10 day voyage after which she was drydocked and the cracks were welded up externally as a temporary repair, without disturbing the cement boxes.

A check was made that the vibration frequency had not altered significantly due to the welding repair, after which a four months round voyage was completed. Permanent repairs were made on the vessel's return, using furnaced angle bars which were riveted to plating and floors to ensure that there would be the least possible increase in stiffness which would raise the frequency of the butterfly mode of bossing vibration. Final tests at the beginning of the next voyage confirmed that this was the case.

The vessel had a long and most successful career and was only recently scrapped.

5. DIESEL ENGINE FAILURES

5.1 Engine tie rods

Large, direct-coupled diesel engines usually have long tie rods to take the firing loads down to the main bearing girders. To restrain "violin string" vibration of these rods, pinching screws are provided. The engine makers' instructions on the tightening of the tie rods, (usually by hydraulic means) and on the follow-up of the pinching screws must be taken literally. Otherwise, it is highly probable that a neglected engine will quickly and seriously damage itself by cumulative fretting at the bolted connections, which gets worse at an increasing rate.

Fig 9 shows the sensitive areas in which carefully fitted bolts and bolting surfaces are eaten away by frettage requiring expensive and protracted reconditioning and rebuilding.

One dispute in which the Author was involved, concerned the sale of a ship with a large engine, which had been allowed to degenerate in this way. The new owner was faced with a huge bill for repairs and several weeks' delay. In fact the engine had to be taken out of the ship and rebuilt from the tank top up. The Author advised against joining issue with the vendors as the owners had been given every opportunity of examining the engine before purchase.

However, the owner was not satisfied with this advice and called for another opinion. The expert employed was so keen on producing a mathematical explanation for the rocking mode of vibration which was reported during the first voyage after purchase, and which he regarded as a "latent defect" that he completely missed the vital clue in the available reports.

The Author and the expert finished up in Counsel's Chambers. The vibration expert led off, at some length. Counsel, who was obviously impressed by the manner — and possible veracity — of the expert's statements, asked the Author, almost in parenthesis, whether he had anything to say. The Author thanked him for the privilege and said he would only refer to one very short sentence in the Ex-Chief Engineer's report, namely that it was not possible to tighten the first four tie bolts on the port side due to seizure of the nuts but that a good job had been done on the tie bolts on the starboard side. There was no need to spell it out to the expert, who immediately got the message, and gallantly withdrew his report with apologies for wasting everyone's time.

The violence of the rocking vibration in a large, tall engine, when a firing load of some 450 tonnes was eccentrically applied, instantaneously and repeatedly, must have been really spectacular.

These days there is no excuse for this sort of damage as the manual effort has been removed from tensioning tie rods. Hydraulic nuts are now common practice, supplied from a small pneumatically or electrically driven pump which delivers the required pressure. However, in spite of this, broken tie rods are still found at surveys and the resulting expensive damage is the basis for costly charter party disputes.

5.2 Thin-wall bearings

Another interesting problem concerned the renewal of thin-shell main bearings of a medium speed diesel engine which resulted in crankshaft failure two weeks after the refit. The obvious question — not asked of the repairers when they were offered the job — was whether they were experienced in fitting large thin-shell bearings. In fact they were not.

A complete set of spares was on board and a new crankshaft was flown out to the repair yard. However, on removing the damaged shells, it was found that the bearing pockets were heavily fretted — in one case by as much as 1.5 mm. The bending stress in the crankshaft must have been massive. It was arranged to fly out from London a Dallic portable electroplating equipment and a skilled operator. In the meantime a tubular mandrel spanning four of the bearing centres was smooth-turned in way of the bearings. Nickel was progressively electro-deposited to minimise hand fitting. Work went on round the clock to get the ship away six days after arrival of the Dallic man.

The fitting of thin shell bearings in auxiliary and main engines is now common practice and there should not be any problem if they are correctly fitted in unfretted pockets.

6. CONCLUSION

A lifetime's work to do with ships is bound to leave its mark on one. This paper is primarily an occasion for sharing remarkable experiences.

Remembering that experience is one of the few things no one can take from you, — not even my Lords Commissioners of Inland Revenue, — it is a rare anomaly in that the more you share it with others, the more you gain yourself.

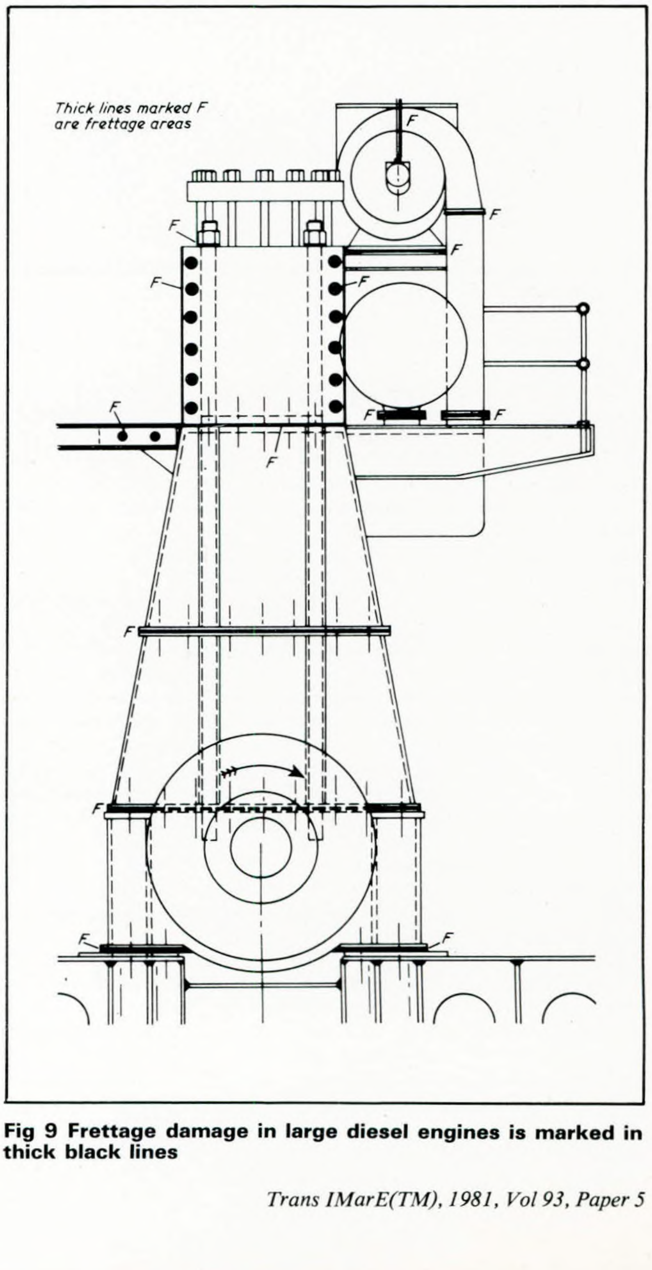


Fig 9 Frettage damage in large diesel engines is marked in thick black lines

Discussion

MR A. HILL CEng, FIMarE: In thanking the author for a most interesting and informative paper, I should like to start with his concluding remarks, and to agree wholeheartedly with him on the benefits of shared experience. Very little is learned from successful machinery or structures. Failures, on the other hand, show that some limit has been reached, there is a problem to be surmounted and a modified approach needed.

It is inevitable that the path of progress should be dotted with failures, each of which was overcome to gain the next advance. The pity is that each new generation tends to ignore the experience of the past, and to buy its own experience, often most expensively. It would be most useful to have more papers to pass on hard-won experience for the benefit of all.

In section 3.1 the author mentions wear in stern tube bearings and cites experience with *Ardtaraig*. Wear in the stern bearings of steam turbine VLCCs is not uncommon and shows how an unforeseen change in operating techniques may result in a new form of damage.

Due to the increase in steam conditions, modern turbines need a longer time for warming up and cooling down. When cargo handling times were reduced to around 24 hours it seemed logical to keep the turbines at full temperature which meant on turning gear for this time. The speed is too low to provide a full oil film and, hence, the shaft is rubbing away on the bearing.

So far as is known, this type of wear does not occur, or at least is much less severe, in diesel-driven vessels. The author's comments on this cause would be appreciated.

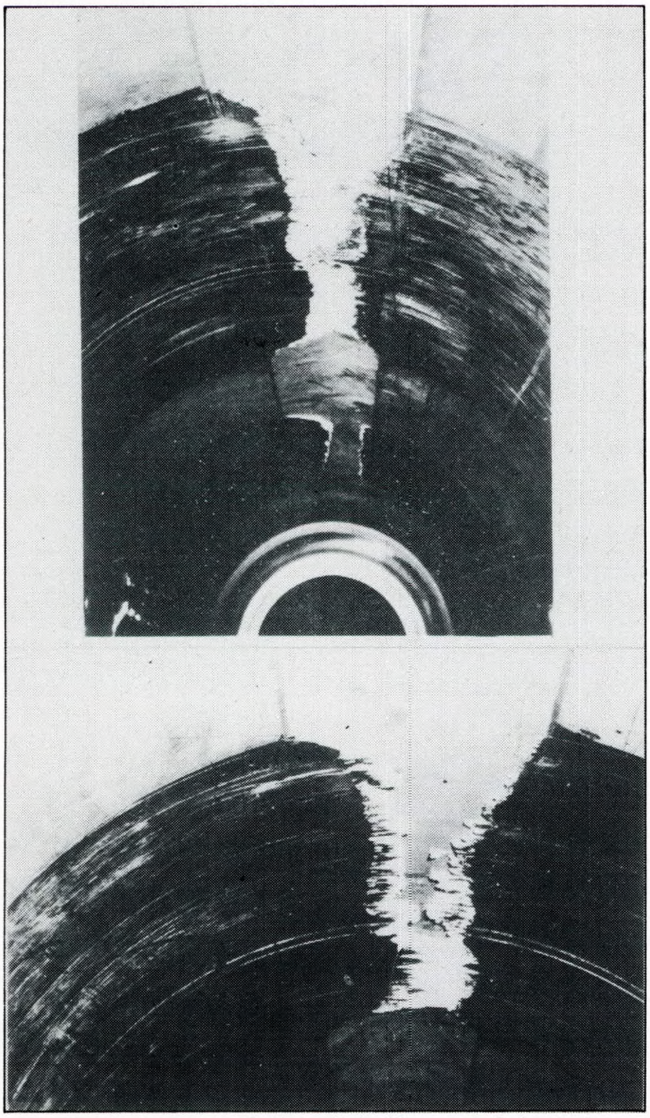


FIG D1 Upper part of the aft (white metal) bush found seized after sea trials

The interesting conclusion is that white metal stern bearings can tolerate this abuse so long as the alignment is good, particularly where it suits the near static condition. Otherwise there would be more instances of damaged bearings.

One must agree with the author that cooling of the stern bearing and seals is assisted by heat flow through the shaft and propeller, but is he not overstating the case in claiming that there is no heat flow through the after peak? It is understood that several cases of overheating in the early days of oil-lubricated stern bearings were found to be due to a dry after peak, and that when the stern tube was submerged by water in the peak the problem was cured. Subsequently, this requirement was written into the Rules.

In Table I, bearing temperatures are listed, with the after peak filled, and empty. In half the cases, the bearing temperature is equal to, or below, sea temperature, which is surprising. As both conditions were recorded on 3rd August and again on the 4th it is possible that readings were taken before steady-state conditions had been established.

With temperatures so close together, one must ask if the instruments had been calibrated together? The figures are given to two decimal places.

Is this the result of averaging and, if so, what was the scatter? Could the author give a little more detail?

In section 3.3.2 the author rightly emphasises the need for cast iron to be 'de-graphitised' if a strong white metal bond is to be achieved. This equally applies to SG irons. The author mentions the proprietary Kolene process, which involves immersion in an electrolytic fused salt bath.

Iron castings are not always free from patches of local porosity. Has the author experienced any trouble due to the caustic soda from the bath being trapped in areas of porosity? It is fair to add that there are alternative processes which provide equal bond strength to those obtained by the salt bath process.

MR G. C. VOLCY CEng, FIMarE (Bureau Veritas): The reading of the author's paper has been a real pleasure, and has reminded me of past experiences that I would like to share.

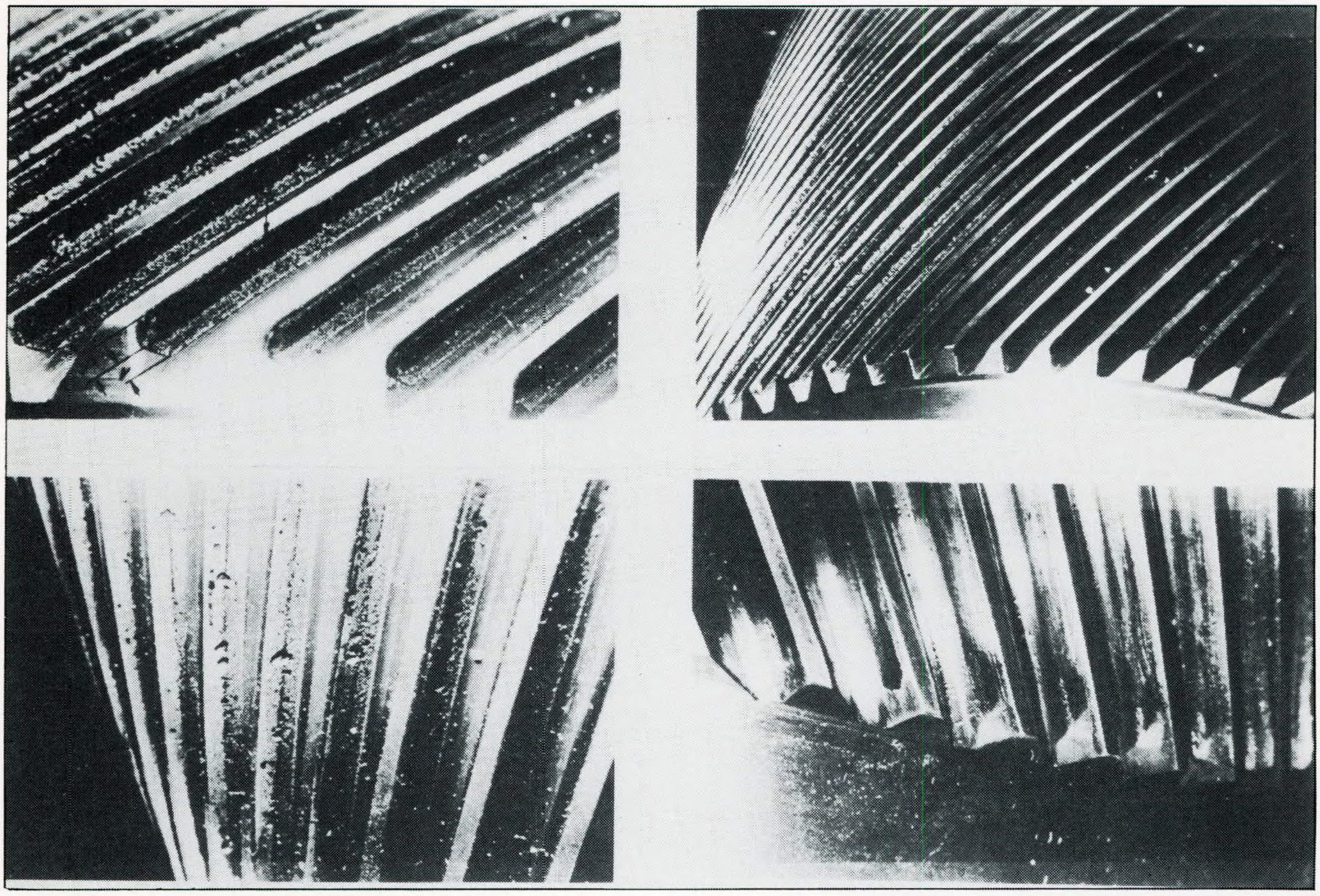
The first occasion concerns main gearing troubles related to pitting. As always, the valuable advice of the author concerning the quality of ingots to be used for gearings is very welcome. But, despite the quality of metallurgy, the pitting often happened in the secondary meshing of main gearing, due not only to internal but also to external effects.

Fig D1 shows the pitting of the aft part of aft helices due to contact between the aft thrust journal and lower shell of thrust bearing. This happened unfortunately often until the aft journal bearing of the thrust shaft was suppressed.

In order to avoid pitting on the forward part of forward helices of the secondary meshing, as shown in Fig D2, it is necessary to proceed with a rational alignment not only of line shafting but of the bull gear wheel shaft journal; the forward bearing always being more loaded during the alignment operation[1,2,3]. Figs 5–7 remind me of the *lignum vitae*-fitted aft bushes of the past. The erosion patches, as indicated on Fig 7, were often accompanied by lateral vibrations of line shafting as the basis of leakage of aft stern and heavy ship vibrations. Fortunately, *lignum vitae* could withstand such whipping of the journal but, for white metal bushes, such cases have shown much more rapid distortion. Fig D3 shows the forward upper part of the forward bush of stern tube which was distorted after only six hours' voyage, without full power, from the outfitting quay to the drydock for cleaning and painting for trials. I experienced such phenomena 18 years ago; these had been predicted by the calculations of the so-called rational alignment which we were trying to introduce into the shipbuilding industry. Neither the shipyard nor the owner wanted to believe the results indicating negative reactions appearing in way of the forward bush; but once they had paid a heavy bill for repairs they were convinced[4]. Since then rational alignment has been universally adopted.

However, new problems have occurred, such as those indicated by the author's Fig 5, due to wear at 6 o'clock (boundary conditions)[5]. Such problems could be solved by introducing double- or multi-slope boring of white metal in the aft part, with the so-called elastic calculations, taking into account not only elasticity of line shafting but also white metal. Surprisingly Fig 5 indicates wear at 12 o'clock. In this respect we agree fully with the author that it is due to the influence of the six components of propeller forces and moments, and it occurs not only on full-body VLCC sterns but has also been encountered on smaller ships and even multi-purpose dry cargo ships.

To solve this problem and stress the importance of the



interdependence of six components of propeller forces and moments on the behaviour of stern gear, it has been necessary to develop a method of calculation of oil film for skewed misalignment[5,6].

But once having at our disposal the tools necessary for predicting the behaviour of the stern gear, as well as the harmful influence of six components of propeller forces and moments (at the origin of which is the non-homogenous wake field) we have tried to establish criteria of judgment and it is possible to discover some coefficients: torque, force and moment. This has been done recently on behalf of the Cooperative Research sponsored by NSMB Towing Tank. We hope this activity will develop so that shipowners and stern gear manufacturers can be protected from harmful influences which act on stern gear behaviour.

The case analysed by the author indicated in section 4 and Fig 8, concerning butterfly bossing vibrations as the origin of ship vibrations, is very familiar to me. It led to the introduction of the philosophy of treatment of the so-called ship vibrations by the detuning of forced vibration resonators. But what was the most amusing to me was that nature itself could detune such forced vibration resonators in the most elegant way, by decreasing the stiffness of bossing. To my knowledge no one had done this before, but the owner of the ship concerned had been lucky to have on board an eminent and far-seeing researcher such as the author who was able to understand the phenomenon and did not allow the stiffness to be brought to the initial value that would have led to the same disastrous ship vibrations.

The author's last point concerning diesel engines is also very interesting. In this case the main engine suffered from lack of machining or maintenance of tie rods. Once again the author's common sense overpowered the academic lucubration of the mathematicians.

I also remember an analogous case of fretting damages to large diesel engines, as shown on Fig 9. It was on a passenger liner equipped with two 12-cylinder main engines where very heavy fretting occurred between cylinder block and transverses of the old-fashioned engine. The primary cause of this fretting was vibrations between cylinder block and transverses, which then decreased the pre-stress of tie rods. All this had been aggravated by deformations coming from double-bottom steel-work.

The fretting was avoided by putting steel shocks in a convenient place between cylinder block and transverses, and connecting newly-machined surfaces by means of 'Lacit otaphom'. This gave good results.

But, in order to avoid harmful influence on the behaviour of main

FIG D2 Pitting, spalling and heavy wear of after helix caused by tilting of thrust block and a non-rational alignment of the shafting supports (17,600 shp, $n = 108$ rev/min)

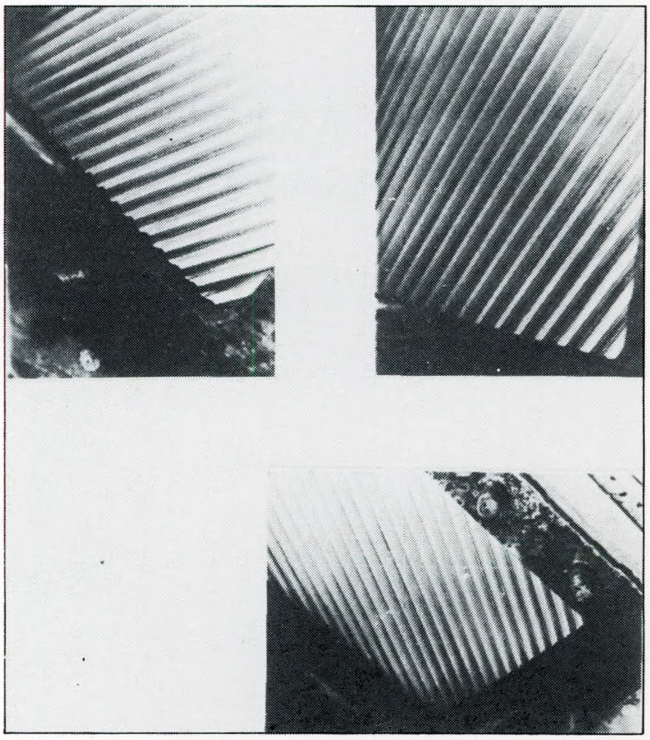


FIG D3 Pitting, spalling and heavy wear of forward helix caused by flexing of double-bottom, tilting of thrust block and non-rational alignment of the shafting supports (17,000 shp, $n = 105$ rev/min)

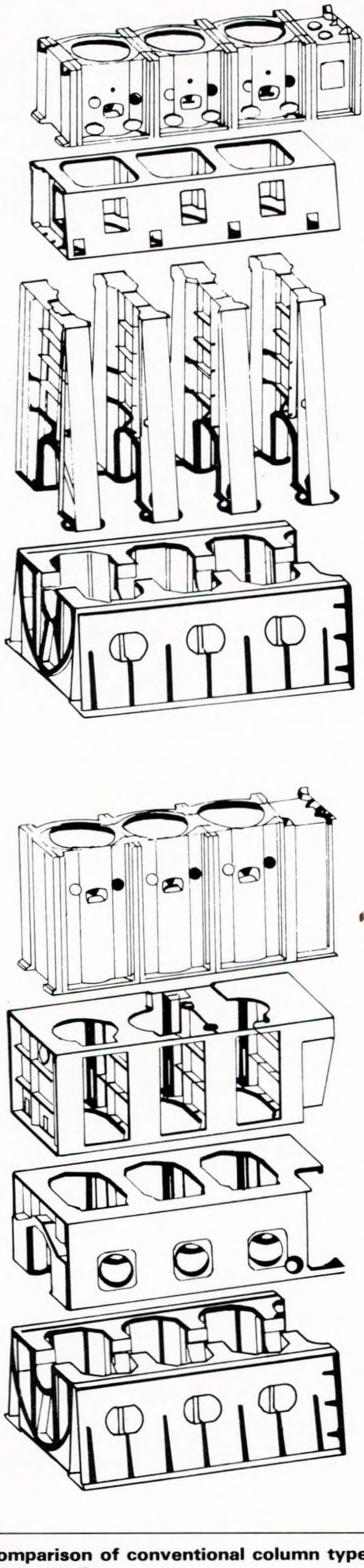


FIG D4 Comparison of conventional column type and new box type engine girder

engine from deformations of steel-work of double-bottom and hull, some diesel engine manufacturers have abandoned the old-fashioned column type engine (which does not present an homogenous girder) in favour of a stiffer solution. Fig D4 compares the old-fashioned column type with the new box-type engine girder[7,8].

In conclusion, I wish once again to agree with the author that there are two types of pleasure a man can feel, one when he is receiving something and the other when he is giving, and the second type of feeling is the highest and the most generous a human being can feel. With the author when he shares his experience with others he gains for himself much more.

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MR J McNAUGHT CEng, FIMarE (Technicon Consultancy and Services Ltd): Referring to Section 3.2, I would consider it a retrograde step to have water-lubricated bearings in the sterntube and hope this deviation will not materialise. However, I have known the pattern marking on the tailshaft liner, where the sterntube was water lubricated, to be avoided if the clearance between the tailshaft liner and the *lignum vitae* bearing, or alternative material, were kept as close as practicable to the normal 1/16 in or so. Wear on the *lignum vitae* does occur between tailshaft surveys and the tendency is to renew a small area only of the *lignum vitae* of the top half bearing at the after end to reduce clearance. Again, when the *lignum vitae* has to be renewed to reduce or eliminate wear-down, sometimes a part only is renewed. It is in these apparent cost-saving measures that clearances within the sterntube can increase over some years and cause the pattern marking on the liner referred to. It can be particularly bad in twin-screw ships where the forward bush is rarely re-wooded.

MR B. HILDREW CBE, CEng, FIMarE (Lloyd's Register of Shipping): It is interesting to investigate problems on ships. Unfortunately, most problems are inherent in the hull and machinery as built and the solution lies in modifying the installation. However, quite frequently in the developing pattern of engineering practice the problem may be introduced into an apparently reliable piece of equipment, in consequence of modification or inability to recognise a minimum requirement as compared with a maximum allowable. In this latter instance, I recently summarised an interesting case where the intermediate shaft fractured in way of a roller bearing. It was apparent that failure occurred due to a reduction in the material properties of elevated temperature under normal torque and end thrust loading. Investigation revealed that the bearing manufacturer had designed his product from an allowable radial load of 3200 kg, whereas the true loading was of the order of 900 kg. The reduction in friction torque to drive the cage in such a light loaded condition resulted in the roller skidding and subsequently seizing.

A very old case, with which both the author and I were involved, concerned intermittent but excessive hull vibration on a twin-screw vessel. The intermittent nature of the phenomenon was obviously related to the phase relationship of the 4–6 cyl, 4-stroke Blohm & Voss engines. It transpired that the engines had been converted from blast to solid injection and the removal of the compressors from the forward end of the engine had resulted in the compressor cranks rotating in limbo, each exerting an out of balance force which, when all four were in phase, summed to 13 tons. The solution was a simple one: to fit small balance weights to the compressor cranks. Perhaps the most interesting feature on this ship was the single reduction gearing driving directly through to the line shafting without any flexible coupling; my first introduction to Maag gears. The ship shortly afterwards caught fire and sank. The ship was an ex-German war reparation called *Empire Windrush*.

A more recent disaster is the loss of the semi-submersible flotel, *Alexander L. Kielland*. Whilst the results of the enquiry are still awaited it is relevant to note that she was not being used in the mode of operation for which she was designed, ie, as a drilling rig. A flotel which is not subject to vibration arising from the drilling operation would apparently appear to be operating in a less hazardous mode. Further, in order to accommodate access to the fixed platform which she was serving, the symmetrical rig was unsymmetrically moored. Whenever the design criteria by which equipment is created is modified, even for the better, it behoves the engineer to re-examine closely the validity of the design in its new role.

Discussion of the paper with my colleague, Mr Clayton, reminded him of a ship which had a B & W water tube boiler where repeated distortion of the vertical fire row tubes had taken place. There was no indication of flame impingement and the tube material was satisfactory. The cause when found was simple: the boiler safety valve discharge pipe, which rose vertically from the chest, was rigidly secured to an overhead girder. Freed from this restriction the drum was able to move upwards between cold and steaming conditions. Such a solution, whilst simple, is worthy of note as it does identify a need to look further than the immediate problem when making a judgement.

MR R. RUTHERFORD CEng, FRINA (Swan Hunter Shipbuilders Ltd): Being a naval architect, my interest in this paper is focused on the section relating to stern bearings and in particular the author's opinion that, in single-screw ships with a block coefficient exceeding 0.7, the tailshaft runs in top of the bearing. I suggest this also applies in finer block ships where there is poor flow condition in the upper half of the propeller disc. Of course, the ideal solution to this problem is a circular wake distributed over the propeller disc, which is impossible to achieve bearing in mind the shape of the ship form ahead of the propeller.

Swan Hunter Shipbuilders have been actively involved, over the last decade, in the shaping of aft ends to limit vibration from cavitation forces and the goal from the outset has been to get as near as possible to the circular wake. Two samples of Taylor Wake Contours derived from model tests for block coefficients of 0.817 and 0.839 are shown in Figs D5 and D6 respectively, which demonstrate what has been achieved by careful shaping of the hull forward of the propeller. As can be seen the wake flow over a major part of the propeller disc tends to be circular, whilst the flow is very good in the critical area at the top of the disc where cavitation problems normally develop. I am sure the author, with his wealth of experience, will give his nod of approval to these wakes as they will no doubt reduce the tendency for the tailshaft to run in the top of the stern bearing and also minimise the variation in propeller thrust.

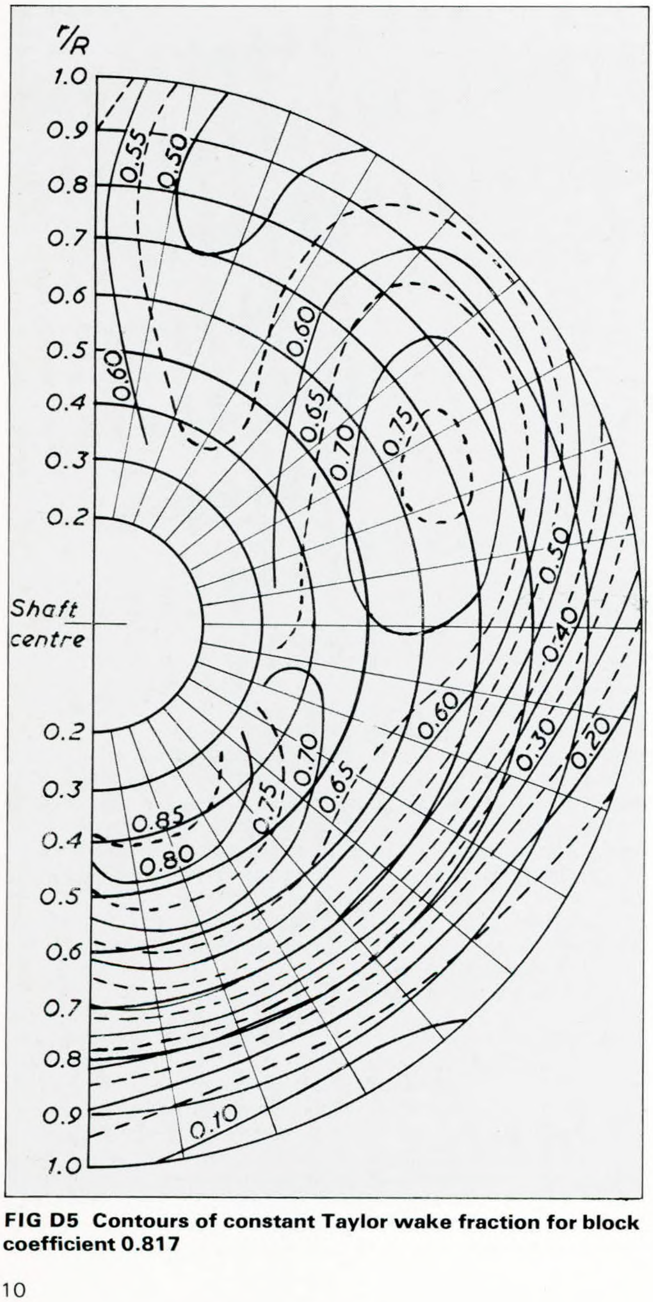

FIG D5 Contours of constant Taylor wake fraction for block coefficient 0.817

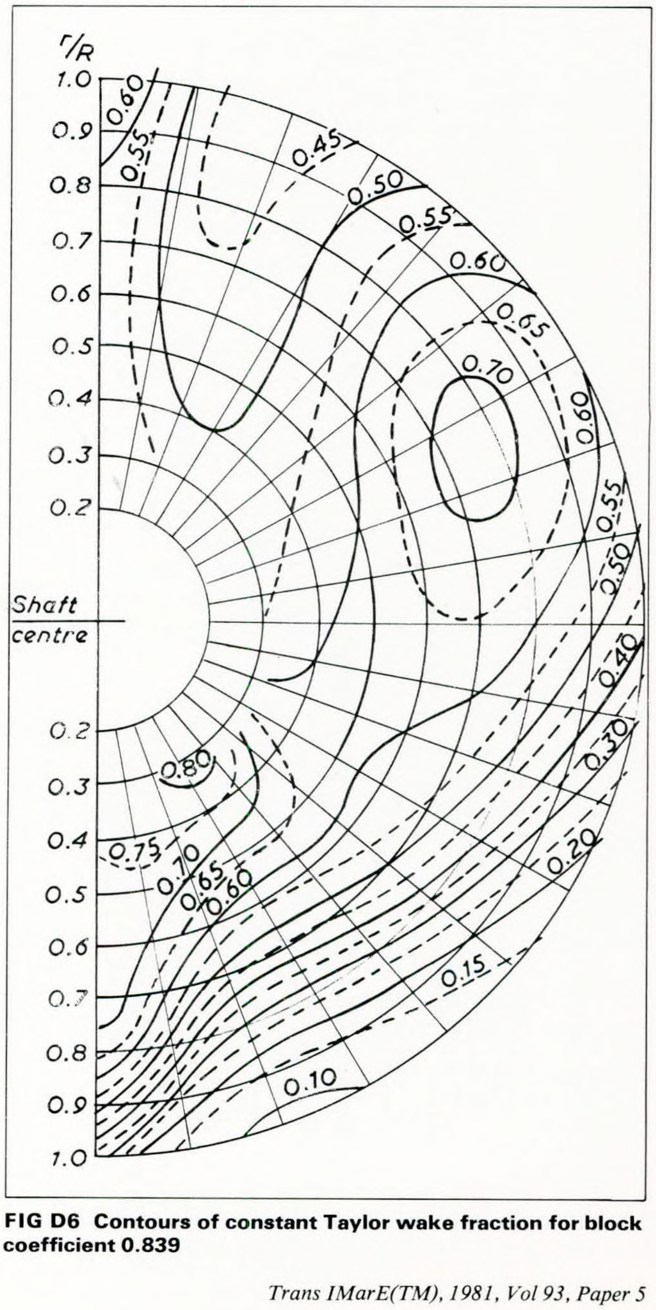

FIG D6 Contours of constant Taylor wake fraction for block coefficient 0.839

There are now a number of Swan Hunter built ships in service with similar type wakes, all of which are operating very successfully with low levels of vibration and noise from propeller sources. I had the opportunity to inspect the propeller of one of these ships after two and a half years in service and found it to be in excellent condition, there being no evidence of erosion anywhere on the blades. All that was required was a polish to bring the propeller back to the as-fitted condition.

I am sure Tom Bunyan, who still retains that wonderful sense of humour, will appreciate that we naval architects are not insensitive to the problems facing our engineering colleagues and are doing our utmost, as we must, to design hull forms which will result in greater efficiency of machinery, less maintenance and reduced levels of vibration and noise for improved crew comfort.

MR T. D. FISHER (IEC Ltd): First I would like to congratulate Mr Bunyan on producing such an interesting and informative paper. As he so correctly concludes there is no substitute for experience and we must all be indebted to him for sharing his own remarkable experiences. However, there is one area where I strongly disagree with him and this is in connection with water lubrication.

To make the statement that returning to water lubrication 'would be a most retrograde step' is, I feel, short-sighted to put it mildly. Such a statement is tantamount to concluding that research and development is a waste of time. This I just cannot accept. Furthermore, it is surely indicative of the view that many other people take: that the majority of development work in this field is in association with water-lubricated, non-metallic bearings. Nobody is going to say at this stage that water-lubricated bearings give, across the board, a better system than oil lubricated but the potential advantages are there for all to see:

(1) The requirement for a constant and complex oil supply is eliminated.
(2) Other than at the inboard, and less troublesome end, no sealing is required. It is a fundamental problem of oil-lubricated shafting that the total integrity of the system relies entirely on the seal.
(3) Bearings are easily finish-machined prior to fitting and future replacement is easier and less expensive.
(4) Bearing installation is simpler—an interference fit is all that is required although back-up mechanical securing is easily accommodated.
(5) Start-up and running at low peripheral speed (ie, under boundary lubrication conditions) is readily accommodated without the fear of 'wiping'.
(6) Life is predictable and safe (water for lubrication is the most abundant element available to mankind).
(7) Overall design is less complex.
(8) There is no risk or expense of oil spillage.

Therefore some, or all, of the following benefits can be derived: simplicity of design and installation; lower operating costs; low maintenance requirement; no pollution; reliability and long life.

It is fairly obvious that Mr Bunyan, even if he accepts most of the above points, will totally disagree that water-lubricated bearings offer 'reliability and long life'. Perhaps then I can be permitted to expand on this point. There are many vessels operating with water-lubricated bearings and I have not heard of any more complaints relating to these than I have of oil-lubricated bearings.

Other than *lignum vitae*, against which Mr Bunyan seems to be relating his comments, there are other more recently developed materials most of which can now boast excellent experience data. What about phenolic laminates such as Railko and the synthetic elastomer/plastic material, Thordon? Both of these materials are now approved to operate for the same tailshaft survey interval as white metal bearings; indeed, indications are that the survey interval could be increased.

Whilst it may be concluded that I have made a meal of a relatively small point, in no way do I wish to diminish the usefulness and the overall validity of this excellent paper. However, I will again repeat that to say that water lubrication is a retrograde step is unacceptable in terms of today's advancements in non-metallic materials. Perhaps I should finish by admitting that I have, as you have probably guessed by now, a personal interest since my company is the European Technical Manager for Thordon.

DR S. ARCHER FEng, FIMarE: As a former colleague of the author and a member of the small team under Dr G. H. Forsyth, who founded Lloyd's Register's engineering research department towards the end of the 1930s, I found the paper both stimulating and evocative, since some of the examples described in the paper occurred during my service with Lloyd's Register.

As regards example 1.2 and Fig 2, there seems no obvious explanation as to how the torsional vibration mode near the running speed resulted in heavy axial vibration of the double helical IP secondary pinion. Torsio/axial coupling seems unlikely, whether from propeller or turbine. However, if tooth separation occurred (an improbable event near the running speed, owing to the extremely heavy excitation required), one could perhaps visualise the possibility of impactive excitation of axial vibration due to very small changes of axial alignment between pinion and wheel during the unloaded part of the cycle. Could the author elucidate?

In the case of example 1.3 and Fig 3, it is stated that when measuring the coupling eccentric wear, the central sleeve is levered upwards with a wooden batten (making sure there is no jamming). Could the author state with what degree of confidence such a procedure could be expected to achieve adequate simulation of the running condition, bearing in mind the propensity of such couplings to attempt to centre themselves under torque? It was interesting to note that in the case of the P & O tanker the HP fine tooth couplings were replaced by flexible metallic couplings, a modern practice adopted as preferred standard by one well-known British gearmaker. Were the couplings in question of the diaphragm (alternatively, link) type and could the author state his own experience with such couplings?

Figures 5 and 6 of example 3.1 were instructive, although I find it difficult to believe that Fig 6 is typical of 'full-bodied sterns ($C_b = 0.7+$)', in that it would require an extreme eccentricity of thrust to overcome the propeller deadweight bending moment to the extent of lifting the shaft through the complete bearing clearance of, say, something like 0.040–0.060 in. In Fig 4 of the IMAS '73 paper by Wilkins and Strassheim*, for the loaded ship condition the calculated lift of the tailshaft was given as less than 0.005 in at the after end and zero at the forward end of the stern bush bearing. These results were stated as typical of large, high-powered vessels and probably included high block tankers. In the circumstances, could there possibly have been some other factor, such as an alignment condition in example 3.1 causing, or helping to cause, such an unexpected wear pattern?

I fully agree with the author's general guidance advice on stern bearings. On 3.3.1 concerning white metal-lined bronze bushes, the expansion of hot bronze is indeed much greater than that of steel by about 60 per cent and the danger described by the author is a very real one. It calls to mind the case of a large vessel which broke down on its maiden voyage. In drydock the stern bush bearing was found to have seized on the shaft (having sheared the dowel pins) and to have worn a considerable depth into the sterntube. At first, inadequate bearing clearance was blamed (correctly so for bronze bushes), although the true mechanism of the failure was not immediately appreciated. Lloyd's subsequently issued recommendations for minimum bearing clearances.

Example 5.1 caused a great deal of expensive concern to owners, engine designers and classification surveyors until its major cause was established. Various palliatives were tried, including horizontal staying between engine tops and engine room casings, none of which was really effective.

The author's well-known success as a trouble-shooter and inventor may well owe a good deal to his undoubted ability to separate the vital factors in a problem from the totality of the evidence; also, perhaps, to his unwillingness to reject the obvious in favour of the more esoteric solution, at least until the obvious is disproved. Example 5.1 is typical of this characteristic.

MR A. ROSE MSc, FIMarE (B & E Boilers Ltd): Over his long and distinguished career the author has benefited this Institute by often allowing us to share his wide experience through his papers, always instructive and often provocative, resulting in wide-ranging constructive argument. This is no exception.

He has described the modern white metal lined, oil-lubricated stern bearing as being most reliable; as indeed it ought to be when its relatively modest loads are considered. Any lack of reliability in the past has been mainly due to stern seal failure and the paper would be considerably enhanced if the author would include comments regarding sterns seals in his reply.

It is well known that the author has been involved in stern seal development, and seals to his design are at present at sea. Perhaps he can give us statistics on leakage rates, wear rates and failure (or lack of failure) rates. In his seal design the author has taken the industrial principle, used in difficult sealing situations, of introducing a buffer fluid between the two fluids, oil and sea water, requiring separation. In his design, air at atmospheric pressure is the buffer fluid and, as the buffer

* Wilkins and Strassheim, 'Some Theoretical and Practical Aspects of Shaft Alignment', *IMAS '73 group 9*, p. 1.

space is open to atmospheric drains in the engine room, leakage of either oil or water can be rapidly detected. Would it be feasible to increase the air (or void space) pressure on this seal so that it approaches that of the sea water and lubricating oil? In this way the pressure difference across the sealing elements would be reduced, thereby increasing reliability.

It may also be queried as to whether air is the best buffer fluid to use. Using similar industrial seals as an example would it not be preferable to use fresh water as the buffer fluid? Obviously it would have to be maintained at a pressure below that of the lubricating oil and possibly also below that of the sea water. However, by employing water in the buffer space, considerable cooling of the sealing elements would result, with a corresponding increase in life. Sea water and oil contamination of the circulating fresh water can easily be detected.

It seems unfortunate that in opposing water-lubricated stern bearings the author has advanced, as argument, the problems that beset staved stern gear of two or three decades ago. Nobody who has proposed the use of water-lubricated stern gear has advocated that they should be of the *lignum vitae* staved design and it is unfair that the author has failed to consider the benefit of modern materials and design methods. Numerous synthetic materials are available with extraordinary wear resistant and resilient properties. A group of such materials is polyurethane based and one proprietary brand, Thordon, has found widespread industrial and marine use. In addition to its wear-resistant properties it has the advantage that various properties can be introduced during manufacture to meet various bearing problems. For instance, the elasticity and hardness can be readily varied. Since the material is elastic it conforms to varying tailshaft slope, which white metal can only achieve by wiping.

With the enhanced wear resistance of polymeric materials it is unlikely that the younger generation of engineers would see the tailshaft damage shown in Fig 7. As the author has pointed out, this erosion has been caused by the effect of bearing wear upon the point of support of the tailshaft within the bearing and the corresponding change in critical speed of the shaft system. Critical speed is now relatively simple to evaluate in the as-new condition and the wear resistant properties of the new generation of plastic materials will ensure that the critical speed does not approach running speed as time passes. Surely the author will agree that, with new materials already available and in the course of development, it would be folly for engineers to neglect their application to stern bearings with all the technical and cost advantages that would ensue.

Returning to Fig 7, it is noted that the tailshaft in question has a keyway. The author's views on keyed and keyless fastening will be appreciated.

Author's Reply

Mr Hill raises a most interesting point about abrasive wear being more pronounced in modern steam turbine tankers because of keeping the engines turning during discharge or loading. This confirms my suspicions which prompted a remedy based on standard practice with large turbo alternators, ie, oil jacking the journals with a small high pressure pump, which cuts in automatically during stand-by and start-up. As to the cooling of stern bearings by flooding the after-peak, I will come back on this matter when the results of tests on two sister ships are known.

I am aware of alternatives, but only mention the Kolene process of de-graphitizing cast-iron stern bearings as this is the one that appears to be in common usage. Because porosity in a stern bearing reduces performance of the white metal lining, every effort must be made to have sound castings. If for any reason a porous area has to be accepted, then its position should be marked, and disposition arranged to be in the unloaded position of the bearing, ie, the forward end. Porosity in sand-cast bearing bushes is usually to be found at the header end of the casting and can be controlled by allowing a generous head of metal.

I thank Mr Volcy for Fig D3 where the forward bush was destroyed during a 6-hour light ship voyage—it can only be described as holding the 'malalignment record'. It must have been difficult to get the tailshaft coupling bolts fitted. Another point of interest is that there appears to be no adhesion between the white metal lining and the wall of the sleeve.

I should be glad to receive further details—preferably with sketches—of the preparatory work that was necessary on the heavily fretted surfaces (as shown in Fig 9) before the epoxy build-up was commenced.

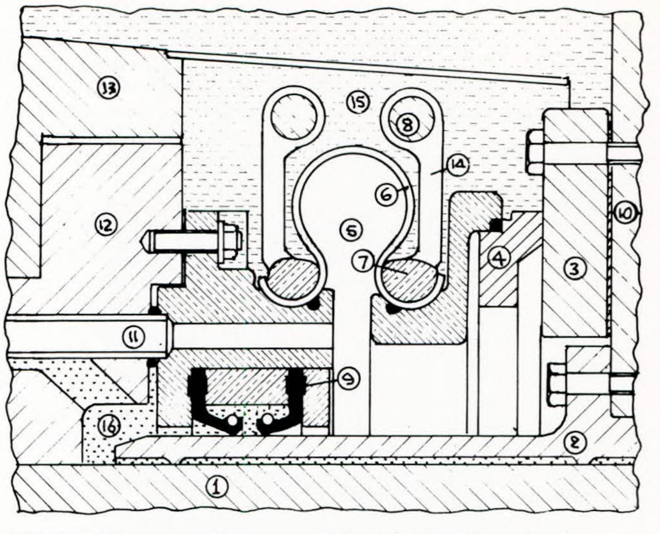

FIG D7 'Coastguard' stern seal (section at 12 o'clock)
(Key: 1 tailshaft; 2 liner; 3 rubbing plate of face seal; 4 wearing ring of face seal; 5 void space (bled into ship at 6 o'clock); 6 bellows annulus; 7 clamping rings; 8 clamping screws; 9 'Simplex' labyrinth oil seals; 10 propeller boss; 11 air vent tube; 12 stern bearing bush; 13 sternframe boss; 14 clamp lugs; 15 sea water; 16 oil)

It is clear that Mr McNaught's agreement regarding a return to water lubrication as a means of dealing with oil leakage from stern bearings is based on many years of similar experience with the special problems associated with water lubrication. Replies to Messrs Rose and Fischer deal with other aspects of this dilemma.

Mr Hildrew's description of the failure of the roller bearing plummer blocks, which resulted in the fatigue failure of the shaft itself, is most dramatic and is probably the main reason why there has never been a rush to fit these bearings despite their claims to energy saving. I remember well *Empire Windrush*, and the outstanding improvement in habitability produced by a simple lump of cast-iron in the right place. The hardened and ground Maag gears connected to the engine without a flexible coupling won our admiration for the courage which first prompted the system. Incidentally, the arrangement doubled up as a sensitive and very audible indicator of irregularities in fuel injector performances.

Bob Rutherford won my respect when, provided only with tank tests and lines plans, he was able to grade 11 ships in terms of their performance as to propeller-excited hull vibration. His decisions proved correct in every case when compared with hull vibration tests carried out in service.† Such assurance is of the greatest value to builders and owners alike.

It is good to know that experience makes the wake factor $\Delta W/(1-W)$ at the 'characteristic propeller radius' so reliable a parameter that consistent full-scale predictions are ensured for the normal run of ships.

The two wake contour plots (Figs D5 and D6) show that, with a bit of ingenuity and a lot of service experience and juggling of afterbody lines, we have a smooth running ship.

Mr Fisher takes strong objection to my comment, which nevertheless is repeated by Mr NcNaught, that returning to water lubrication would be a most retrograde step. My experience is coloured by being deeply and responsibly involved with some 200 cases of trouble, mostly big trouble, every year for almost a working life-time. Because of this it is possible that I have developed a jaundiced view of some marine engineering practices.

The stern bearing gland and tailshaft assembly had achieved the singular distinction of being the most dangerous and unsatisfactory feature in marine engineering, becoming progressively worse as the size and power of ships increased. These factors have left a deep impression of insecurity in the minds of those involved—including shipowners. Latterly a new dimension of frightfulness has been added, namely oil pollution of the sea.

† Rutherford, R: 'Aft end shaping to limit vibration', *North East Coast Inst. of Engineers and Shipbuilders*, Vol 95, 1979.

I would remind Mr Fisher of a most important contribution towards solution of this problem made by Shell International Marine a few years ago. There had been two disastrous stern bearing failures in Shell tankers in 1968, as well as similar failures in WM lined stern bearings in some foreign-owned VLCCs. These failures triggered the Shell research programme using a massive test rig in Harland and Wolff's shipyard in Belfast. The results of this very costly research were published (W.N. McIlvean; 'Stern tube seals and plastic bearings', *Proc. I Mar E*, Vol 88, 1976). The nett result was an improved lip-seal and a cryselic-resin bonded reinforced wound-asbestos yarn stern bearing bush.

Like Shell, I am open minded as regards new developments which, of course, include bearing materials for stern bearings, especially when demonstrated by full-scale testing. Again, like Shell, I prefer to keep to oil lubrication because of the ' creepy-crawlies' that are taken on with water lubrication; notable amongst these is the tailshaft liner. The main problem is not with the bearing, which, with adequate care, has every chance of lasting the life of the ship. The problem has largely been the oil seal. But we can now see the light at the end of the tunnel. The Glacier Coastguard seal (Fig D7) was first fitted in a P & O 100,000 dwt bulker three and a half years ago (720 mm diameter shaft), and quickly followed by three P & O gas carriers (650 mm) and two high-powered container ships (820 mm). In terms of proving a new product this a short time, but it must be remembered that the 'Coastguard' seal combines two well-tried seals, the 'Crane' face seal and the 'Simplex' oil seal, both of which have been around for more than the life of a ship. The unique feature of the 'Coastguard' seal is that, without smashing the seal completely, it is not possible for oil to leak into the sea. If it does leak, it leaks into the ship together with the slight drip of sea water (less than a cupful in an hour) which weeps through the face seal. The other important feature is that the wearing parts can be replaced in two or three hours in drydock or afloat, without any disturbance to the shafting.

In reply to Dr Archer, it must be remembered that this ship was under the sentence of the long voyage of no return to Taiwan, having served her masters faithfully and well (if noisily) for around 30 years.

The fall in value of the pound brought a fresh rush of blood to cruising world-wide, and a temporary reprieve to the ship. So given normal conditions, the emergency repairs, which were the cheapest and least disrupting, would have been different. The 12-tooth claw couplings were fretted and worn, ie, the faces of the coupling teeth were no longer in the same plane as the turbine/pinion axis, but took on a progressively increasing inclination. This, however, had an increasingly strong axial component which achieved dramatic significance as the revolutions approached a sharply resonant torsional condition near the full speed.

As regards the accuracy of measurement of eccentric wear in fine tooth claw couplings using the two clocks of Fig 3 in the paper, unless you can repeat each reading (three at 120 deg rotation) to ± 0.001 it suggests that the tooth spaces have entrapped frettage products which must be removed and the coupling re-connected for another try.

A popular design of fine tooth couplings had a continuous spherical landing solid with the male member, which, without wear, perfectly centralized the female member. Later designs dispensed with this feature and utilised short, spherically surfaced, extensions of the tooth tips of the male member which supported the female member in the roots of the teeth.

The contact markings around the periphery of the continuous spherical landing are a clear indication of the running alignment. Uniform marking, usually very slight, is an indication of the good alignment necessary for smooth operation over many years. However, harder marking tapered over a 180 deg arc shows indifferent alignment and the probability of progressive wear; the female member is running thrown out at the same radial position. In one serious case, where frettage developed early in the life of the ship, the marking was found to vary from 1.5 mm to 5.5 mm diametrically opposite. These couplings were replaced by flexible metallic couplings of the link type.

In regard to the 12 o'clock stern bearing marking (Fig 6) it is, of course, not possible to be dogmatic as to the shafting alignment, as these measurements were not taken at the time. The same marking was also found in all four sister ships. Mr Rutherford confirms the '12 o'clock factor', but goes further to say that the phenomenon is to be found in ships with less then $C_b = 0.7$ but characterised by concentrations of high wake fractions at around 12 o'clock of the propeller disc.

I have noticed this feature, partially developed, in six high-powered, high-speed container ships which were rough-running at full power. In these cases there was some anxiety as the shaft climbed only half-way up the bearing and appeared to run in way of the 9 o'clock oil groove, producing some wiping at the groove shoulders. Incidentally, this 9 o'clock performance produced serious wiping of the bearing in two other large, high-powered, diesel engine driven container ships and called for a bearing re-design with a 12 o'clock oil groove replacing 9 and 3 o'clock grooves. The 12 o'clock marking is, however, no cause for concern. Experience with a 100,000 dwt tanker (notorious for hammering away merrily at full power), apart from the bright 12 o'clock markings of the stern bearing, showed no other indication that the ship was a rattletrap.

Mr Rose refers to the 'Coastguard' seal and suggests pressurising the void space (Fig D7). The air bleed pipe (11) is intended to be used for this purpose in the event of the wear rate of the face seal being excessive. So far the performance of all six seals has indicated that this additional complication is not necessary. I consider that, if the wear rate of the face seals makes renewals necessary only at the 5 year tailshaft survey, the design is quite satisfactory. Regarding the Perbunan labyrinth oil seals with the low header tank, ie, 1 m above the tailshaft periphery, experience has shown that no replacements should be necessary before 10 years; nor is significant grooving wear of the liner to be expected, due to the low contact pressures and corresponding low levels of frictional heat, which is the cause of hardening of the labyrinth seals and grooving the liner.

Mr Rose also takes up the cudgels on behalf of the return to water lubrication of tailshafts. He is referred to the reply to Mr Fisher. However, Mr Rose may be already persuaded that there is a good chance that the solution lies with us. He is aware of the performance of the first group of six 'Coastguard' oil seals (Fig D7). Twelve further seals are on order for shafts up to 890 mm in diameter, and this in spite of the increased cost of the 'Coastguard' over the usual 'Simplex' seal, a factor which has retarded its adoption. The number of enquiries now being received does, however, indicate a quickening interest.

With regard to tailshaft whirling critical speeds, calculation and experimental on-site methods have been known for many years, and computer programs are now in every-day usage providing optimized data for shafting alignment, which could be used to advantage in ensuring that the whirling critical is well below the running speed. With the drive for greater economy, the large-bore, slow-speed diesel engine is king for ships trading deep sea. The high levels of torsional vibration critical stresses have meant excursions into medium alloy steels to permit lowering the I-node critical speed by reducing shaft diameter. How shafts with bronze liners will stand up to these high dynamic torsional strains is another problem, which goes hand-in-hand with those already associated with water-lubricated stern bearings.

Fig 7 is authentic in that the bronze liner and water lubrication mostly 'bowed-out' with the 'key'—as fellow conspirators in disaster—in the 1960s. The keyless propeller was almost exclusively used with larger, higher-powered ships, though the smaller ships are still fitted with the key. Lloyd's statistics show that of the 343 keyless tail-shafts at risk over 1968/69, there were zero failures; but 39 failures in keyed shafts, with 1427 at risk. In other words, the risk of a tail-shaft failure in keyed propellers is ten times greater than in keyless.

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