HARD-ON-HARD WATER-LUBRICATED BEARINGS FOR MARINE APPLICATIONS

BY

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This article is based on a paper presented by the authors at the Institution of Mechanical Engineers Leeds/Lyons Tribology Symposium at Leeds in September 1986.

ABSTRACT

Current trends in marine engineering are leading to the development of water-lubricated bearings using hard materials for the bearing surfaces. The design and manufacture of such bearings is at the limit of established technology. The most promising designs at present are believed to lie in the area of ceramic/ceramic material combinations, although in the less demanding situations hard coatings on metallic substrates may be satisfactory. The article identifies areas for further research leading to improvements in the technology which will be necessary before such bearings can be consistently designed and manufactured with confidence.

Introduction

Trends in marine technology require equipment that has low first cost, high reliability, low maintenance and is durable. In order to meet these requirements increasing attention is being paid to the design of wearing components, particularly bearings. This article addresses the issues raised in designing water-lubricated bearings and indicates some of the problems that have been encountered in practice.

As far as possible the general design guide that a machine should use its working fluid as a lubricant is followed in marine engineering practice. This has obvious advantages. Water-lubricated bearings are therefore commonly found in circulating water pumps, high pressure feed pumps, extraction pumps, fire and bilge pumps and in the modern glandless pumps. Propulsion shaft bearings in the 'A' frame and stern tube are also water lubricated.

Whilst there are many attractions in using the machine's working fluid as a lubricant, water and in particular sea water bring in their wake certain problems. Fluid film thicknesses are very much smaller than would be obtained using a more viscous fluid. More significantly, sea water carries abrasives, e.g. sand and silt which accelerate wear of the bearing surfaces. In the cases of taper-land designs this can be potentially serious, leading to loss of load-carrying capacity.

The traditional response to these problems has been to use bearing surface materials which are tolerant of abrasives and which are capable of operating under mixed film or boundary lubrication conditions, e.g. ferrobestos or cutless rubber operating against a hard material. Such an approach implicitly accepts a rate of wear which, for reasons that have been described earlier, is unacceptable.

A development presently in its infancy but gaining increasing acceptance is the use of hard bearing surface materials. These are harder than the abrasive particles likely to be drawn into the contact and as such able to withstand the associated wear. However, the engineering of these surfaces is poorly developed and there are many problems to be resolved.

This article considers various aspects of bearing development for sea water lubricated thrust and journal bearings. The article also discusses theoretical aspects of bearing design with low viscosity lubricants and makes reference to experimental and practical evidence. This is followed by a consideration of bearing materials, and indicates an approach which has been found to be helpful in material selection.

Bearing Design Concepts

The ideal bearing system should be self-feeding, maintenance-free and have infinite life. Additionally, when water (or sea water with possible particulate contamination) is to be used as the lubricating medium, the bearing, as well as exhibiting corrosion resistance and material stability, must, due to the low viscosity of the medium, operate with very small lubricant film thickness unless very conservative and often impractical designs are adopted.

These requirements are being met in many cases by using bearing surfaces of extreme hardness, in mutually compatible pairs. Predominately hydrodynamic (i.e. contact-free) operation may be guaranteed by using extremely accurate stable bearing surfaces—both in terms of surface finish and form. Since by the nature of such hard materials no significant running-in capability exists, the alignment of stationary and moving bearing surfaces necessary to promote hydrodynamic operation at the small film thicknesses expected ($< 10\mu m$) must be fundamental to the bearing designs.

For horizontal journal bearings the route chosen is to use bearings of the tilting pad type. The pads are then free to tilt in pitch, roll and yaw on spherical pivots, and can adapt to any equilibrium position for a given lubricant film thickness at the pivot point. A further significant advantage of such bearings is that they are stable to self-excited (whirl) vibrations since each pad can only supply a reaction normal to the pivot. This is in distinct contrast to fixed bearings, which are prone to whirl.

For thrust bearings, particularly in horizontal machines with a single rotation direction, fixed taper-land bearings are most appropriate. The alignment between collar and bearing is assisted by resiliently mounting the latter component.

The dimensions of the bearings are dictated to an extent by the overall dimensions of the machines, e.g. shaft and hence journal diameters are generally determined with little reference to the bearing requirements. Exploratory design studies, have been carried out using the MELBA suite of bearing analysis programs[1, 2, 3] where the various parameters (e.g. bearing length, pad dimensions, pivot position, pad/journal radius difference, etc., in the case of journal bearings) are explored over the expected ranges of load, speed and temperature. If the bearings operate in the laminar regime, with modest power losses, the predictions have been found to be accurate to a

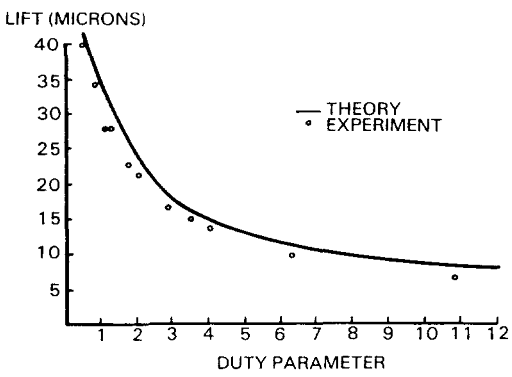

FIG. 1—VARIATION OF LIFT WITH DUTY PARAMETER

high level of confidence—even with isothermal theory. Experimental checks have been made, where journal/pad eccentricity and attitude measurements confirm the accuracy of the mathematical model, particularly at the more heavily loaded conditions (see FIG. 1). The predicted values of minimum film thickness, generally of the order 5–10μm, set the standards to be met for the hard bearing surfaces, both in terms of form (cylindricity or flatness) and surface finish.

In the case of journal bearings, with diameters and lengths of the order of 100mm, cylindricity of 2–3μm is required while surface finish Ra-values of 0.2μm are held. Values of pad/journal radius difference are set at near optimum values for maximizing minimum film thickness at the most heavily loaded conditions. The situation in terms of carrying the loads is therefore satisfactory in that, with near optimum hydrodynamic designs, contact-free operation exists except during starting and stopping transients, where rubbing speeds are low enough to give insignificant wear or distress. This latter feature has been confirmed by long-term testing of various equipments in both rigs and prototype machines. These tests involve multiple start/stop cycling and continuous running both in clean sea water, and in sea water containing measured quantities of silica powder (representing sand contamination).

One feature of journal bearing operation using tilting pads which is not predicted well theoretically and which has caused practical problems particularly in vertical shaft machines, is that at large film thicknesses tilting pads do not behave as predicted. The bearing clearance ratio is determined at the design stage to cope firstly with thermal transients and secondly with pivot setting tolerances of the order of 1/1000 on diameter i.e. in the range where oil-lubricated bearings normally operate. The low viscosity of water dictates that with any significant load on the bearing, pads opposite those taking the load operate at large mean film thicknesses, and therefore with minimal loads generated by viscous entrainment. Under such conditions, pad gravity loading, viscous friction effects and fluid inertia effects may be expected to become at least of equal significance as the viscous wedge effects. Although in principle, and at considerable expense computationally, fluid inertia effects in the film can be calculated, they cannot be expected to give the complete picture, since pressure effects in the pad inlets are undoubtedly of significance—and may even predominate. Measurements of pad inlet pressure in a tilting pad bearing (made via a tapping connected to a manometer) suggest that excess pressures of the order 0.3–0.5 times the surface dynamic head pressure exist. Such pressures are much larger than the specific loading expected on purely viscous grounds so that the normal theory, even if fluid inertia effects are included, would appear insufficiently comprehensive. A separate and more recent investigation using the tilting pad version of MELBA which allows the effect of offsetting moments to be calculated, has indicated that a tilting pad, due to its own gravity offsetting moments (roll and yaw) does not exhibit a stable loaded equilibrium beyond a certain quite small film thickness at the pivot, and as a result will tend to 'sprag'. This has been found to happen in practice.

Material Selection

Background

The following materials have been selected and extensively tested for sea water lubricated journal and thrust bearings:

- Phosphor bronze
- Reinforced phenolic resin
- Hot pressed silicon nitride (HPSN)
- Plasma sprayed chromium oxide
- Detonation gun applied tungsten carbide with cobalt and chromium.

The soft materials, phosphor bronze and reinforced phenolic resin, work well as one half of a bearing pair but wear too quickly in the presence of fine sand and corrosion debris.

Of the hard materials, HPSN has performed well against chromium oxide and tungsten carbide. It has been used successfully in journal bearings in the form of 6 mm thick liners lightly clamped into Inconel tilting pad backings and running against a chromium oxide coating on the shaft. The tungsten carbide coating has been found to show signs of deterioration in sea water due to cobalt leaching. In journal bearings chromium oxide has been found to perform well, both against itself and against HPSN. During deliberate failure tests, when bearings were allowed to run dry and seize, some crazing of the coating was observed in the seizure band. However the bearing would run hydrodynamically under full load after the seizure tests albeit with some spalling. This encouraging result has not been repeated in the case of thrust bearings. Several failures have been experienced due to extensive detachment of the coating from the backing. It is not known whether these failures were due to poor coating quality or to a poor performance of chromium oxide coatings during non-hydrodynamic rubbing.

Further consideration has been given to material selection, probably the greatest challenge in hard on hard bearing design. Two semi-quantitative investigations are briefly described below.

Selection Based on Flash Temperature Calculations

The theory was taken from Blok's work of 1937[4], to address the temperature rise due to frictional heating at the interface between two bodies rubbing together. The properties of the materials used are given in TABLE I.

TABLE I—*Properties of materials used in calculations*

Material	K W/m°C	P kg/m^3	C_p J/kg°C
Cr_2O_3 coating	1·89	5000	733
SiC	200/50	3100	800
HPSN	20	3200	750
Inconel (substrate)	9·8	8440	400

TABLE II shows the results of such calculations for a number of material combinations[5]. It can be concluded that:

(*a*) The flash temperatures can be very high.

(*b*) Silicon carbide on itself produces the lowest flash temperatures by a large margin and this, together with its tolerance of high temperatures, probably explains its good performance as a bearing under extreme load conditions as reported by Kamelmacher[6].

TABLE II—*Calculated flash temperature rise*

Material Pair	*Moving Surface*	*Flash Temp Rise* °C	*Heat Partition* watts	
			moving	*static*
SiC/SiC	either	129·3	710	155
HPSN/HPSN	either	474	808	57
Cr_2O_3/Cr_2O_3 (on Inconel)	either	935	832	33
SiC/HPSN	SiC HPSN	154 431	846 736	19 129
SiC/Cr_2O_3	SiC LC4	156·5 727	859·5 647	5·5 218
$HPSN/Cr_2O_3$	HPSN LC4	496 857	847·5 763	17·5 102

(*c*) Tungsten carbide coatings were also considered in addition to chromium oxide. There is little to choose between them in respect of calculated flash temperatures.

(*d*) In all cases the majority of the heat generated flows into the moving body which, if a choice is possible, should have the highest thermal conductivity. Unfortunately either side of a bearing pair could be the moving body in most bearings and it is difficult to derive any advantage from this conclusion. Thus even when using SiC, with its high thermal conductivity, against chromium oxide, the flash temperatures may be almost as high as with chromium oxide against chromium oxide.

(*e*) The SiC/HPSN combination is probably acceptable in respect of flash temperature as the higher temperatures are not likely to damage ceramics.

Thermal Stress in Hard Coatings

The hard coatings mentioned above are applied about 15μm thick on Inconel. If raised in temperature uniformly the Inconel expands more than the coating and away from the edges a perfect coating is put in tension but the bond is not stressed in shear. At about 400°C temperature rise from an unstressed state, a chromium oxide coating can be expected to crack as its tensile strenth is exceeded. At the edges, or local to cracks in the coating, a more complex stress situation is found, with potentially high shear stresses at the bond.

TABLE III—*Coating tensile and bond stresses*

Coating Thickness t μm	*Max Coating Tensile Stress* $f(o)$ MN/m^2	*Peak Bond Shear Stress* τ *max* MN/m^2
50	217·1	99·8
150	217·1	178·4
450	217·1	315·0

It can be demonstrated that the coating tensile stress and bond shear stress distributions have forms shown in the examples of FIGS. 2 and 3. TABLE III indicates the variation of coating tensile stress and peak bond shear stress with coating thickness[7, 8].

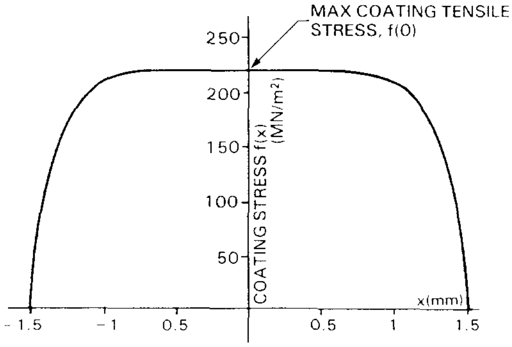

FIG. 2—COATING TENSILE STRESS DISTRIBUTION

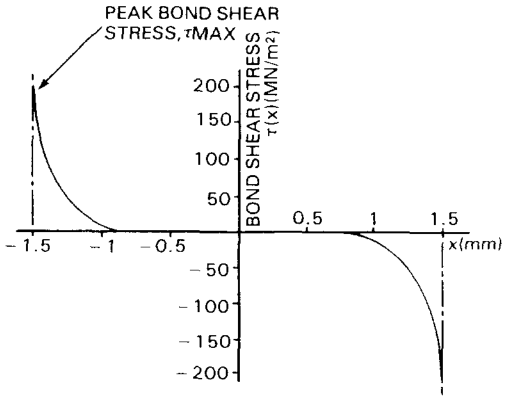

FIG. 3—BOND SHEAR STRESS DISTRIBUTION

It will be seen that the maximum coating tensile stress occurs away from the edges of the specimen and the peak bond shear stress rises to very high values at the edges or at a discontinuity, e.g. a local imperfection. Equally, coating thickness significantly influences the peak bond shear stress. There is a clear advantage in minimizing coating thickness.

Discussion

The requirement for 'hard' materials in water-lubricated bearings is established and experience to date indicates that the concept is practicable. However, the technology is imperfectly understood, and is very much a matter of development for particular applications.

Tilting pad designs have been found satisfactory in horizontal journal bearing applications. However, in applications where pads operate vertically in the unloaded condition serious instability has occurred. It is believed that the pads can tilt to present a divergent film shape. This generates hydrodynamic forces tending to pull the pad into contact with the moving bearing component and leads quickly to catastrophic failure. Variations to the pad shape, by way of 'ramps' to the pad leading and trailing edges, have been attempted with some success to try and modify the film shape and so prevent the divergent film from becoming established. This is clearly an area where further research is required. In particular, conditions at the pad inlet and the influence of fluid inertia effects merit investigation. Practically, the problems experienced with unloaded pad instability have led to the more conservative taper-land designs being adopted for thrust bearing applications.

It will be apparent that water-lubricated bearings using hard materials are indeed precision engineering components. Fluid film thicknesses of the order 5–10μm, surface finishes of 0.2μm Ra value, flatness of runners (26 cm diameter) of the order 10 helium light bands and swash of 13μm present design problems which are difficult to resolve in practice. Quality assurance during the manufacturing process is of paramount importance. Particular care must be taken in designing adequate stiffness into the bearing yet enabling sufficient compliance to accommodate misalignments which are inevitable given an accumulation of manufacturing tolerances.

Perhaps the most significant problems in designing 'hard-on-hard' water lubricated bearings lie in the area of material selection and design for the use of such materials, particularly ceramics. The design problem can be seen in two parts: firstly to provide adequate structural strength and rigidity to withstand high transient loads, e.g. shock and stiffness to maintain the bearing form; secondly to accommodate situations where the bearing operates non-hydrodynamically and touchdown occurs.

Without taking a particular design for detailed study and presentation it is only possible to make general observations for the first case. Where monolithic ceramic components are used, given that the component sizes are within the size limit that is possible (presently about 26 cm diameter disc for SiC components) then the method of attachment of a thrust runner to its shaft is often the area where the greatest problems arise. It is important to avoid stress concentrations. Securing the runner by bolts through holes in the ceramic is not likely to succeed. More complex designs using friction devices may be necessary.

Particular care also has to be paid to the design of the bearing flexible support system. Under load the bearing will deflect. Given the small film thicknesses that exist only minor deflections can be tolerated. Resolution of such problems in practice is as likely to be a matter for test rig development as of accurate calculation and will therefore be costly.

It is possible to give more quantitative, general guidelines for the second case, i.e. non-hydrodynamic operation. Essentially we are trying to design to resist wear and potential failure as a result of the high flash temperatures that can be generated. There are three material approaches that are available:

(*a*) Hard coatings (plasma sprayed or applied by detonation gun).

(*b*) Ceramics (either monolithic or fabricated structures).

(*c*) Hard thin films (ion deposition process).

A useful material screening method has been developed by applying Blok's flash temperature theory and by theoretical analysis of the thermal stresses between a coating and its substrate. It has been shown that coatings in contact (chrome oxide and tungsten carbide) generate substantially higher flash temperatures than a coating against a ceramic or ceramic against a ceramic. Indeed temperatures sufficiently high to melt the coating can occur. Theoretical work also indicates that the temperatures generated are certainly sufficient to cause coating cracking and bond failure. This would explain catastrophic failures that have been observed in practice, and which resulted in almost total destruction of the bearing surface. Where such damaging contact conditions can be avoided a coating *v.* coating combination has been found to operate effectively with minimal wear.

It is believed that a more promising way forward lies in the use of ceramics in both thrust and journal bearing designs. For tilting-pad journal bearings, ceramic inserts in a metal backing have been used succesfully, albeit not without some difficulty in manufacture. Ceramic sleeves on shafts have not been so successful and it still remains for a wholly reliable attachment method to be devised.

It is not thought that ceramic facings can at present be successfully engineered into large thrust bearings, given the difficulties of manufacture and the close tolerances that are required. The most promising way forward is seen in monolithic designs paying particular regard to the problems of runner attachment to the shaft.

Work has only recently started to investigate the likely advantages of hard thin films formed by the ion deposition process. This route offers attractive prospects. The coatings promise to be less prone to catastrophic failure and might even be tolerant to penetrating abrasives in much the same way as traditional white metal bearings. Components would be relatively cheap to manufacture and in the structural sense would certainly be more attractive than ceramics.

Finally, an area which would particularly benefit from further research is that of non-destructive examination of coatings. At present there is no proven method which can measure the quality of attachment of a coating to its substrate. Equally, the ability to identify coating delamination or inclusions in monolithic ceramic components would be extremely valuable. The most promising way forward probably lies in the direction of ultrasonic scanning techniques.

Conclusions

1. The design and manufacture of water-lubricated bearings with hard bearing surface materials is possible but is at the limit of established technology.
2. Combinations of ceramic materials are believed to have better prospects of success than ceramic/coating or coating/coating combinations.
3. Hard coatings require a highly corrosion-resistant substrate. Nothing less than Inconel has been found to be satisfactory in sea water.
4. Further research is required in the areas of:
   - lightly loaded tilting pad bearings;
   - non-destructive methods to assess the adhesive quality between a coating and substrate;
   - the use of ion deposition methods to produce hard coatings for such applications.

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