Paper No.13

Recent Developments in High Speed Thrust & Journal Bearings

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1. INTRODUCTION

It is the function of a bearing, in common with any other component in a machine, to give trouble free operation over the planned life of that machine. Improving reliability implies a reduction in the chance of failure during that planned life; increasing the overall life can be a quite different task.

Historically, the choice of bearing for a given application was made on the basis of specific load (applied load divided by bearing projected area). With the advent of the digital computer, numerical techniques allowed assessments to be made on a more rational basis. In particular the calculated minimum oil film thickness, and the temperatures of both lubricant and bearing surface have been used successfully for several years in verifying bearing designs.

It is not satisfactory to try and improve reliability simply by overdesigning a bearing, since possible penalties in size, power loss, oil flow requirement etc., will have to be borne throughout the life of the machine, in addition to the increased capital cost. A more systematic approach is needed, starting from a detailed knowledge of the operating conditions that the bearing will be called upon to accept; in many types of machinery these can be difficult to define.

No calculation procedure can be better than the data which is fed into it, and it is not uncommon to find that a basic parameter such as the applied load can be estimated only very approximately. Additionally there are many factors which can drastically affect bearing operation but which normally are not allowed for in calculation procedures; these include dirt contamination of the lubricant, misalignment between the bearing and its mating face, and thermal and mechanical distortions of the bearing and housing.

Thus, whilst tremendous advances have been made in bearing design techniques there are still many factors which affect bearing reliability which are difficult both to quantify and to include in an analysis.

2. JOURNAL BEARINGS

Whilst slow speed bearing design is concerned mainly with keeping shaft and bearing surfaces separated by a film of lubricant \( r \), at high speed numerous other factors become important. These include the need to maintain acceptable temperatures, to keep shear losses as low as possible (for good machine efficiency), and to maintain acceptable vibration levels.

The measures necessary to overcome any of these are not always compatible. For example increasing clearance usually reduces bearing temperature, but it also increases the power loss and can make many types of vibration worse. In such cases it is usually better to move away from a conventional cylindrical bore bearing and to make use of two or more lobes. These can be fixed profile (eg lemon bore, offset halves, 4-lobe) or tilting pad. Carefully designed, they can give both acceptable temperatures and stability.

A general profile bore is shown in Fig.13.1. The major variables open to the designer are number of lobes, lobe clearance, bearing clearance and angle of lobe centre (tilt). When the lobe clearance is increased, the amount of lubricant being dragged into the film is increased, thereby enhancing the cooling effect; this can be done whilst still maintaining a tight bearing clearance and consequently tight positional control of the shaft. Such changes can also make large differences to the stiffness and damping coefficients of the oil film. Eight of these coefficients are required to describe the linearised behaviour of the oil film under imposed displacements and velocities of the shaft.
The oil film stiffness is very important for a rotor operating below its first critical speed, since this can control vibration amplitudes resulting from out-of-balance or other applied dynamic forces. It also affects the speed at which the critical occurs. At the critical speed the damping is the dominant factor in controlling the amplitude. Above the first critical speed the oil film forces may or may not have a significant effect, depending on the distorted shape of the rotor in its vibration mode.

All these vibrations occur at the same frequency as shaft rotation, but many problems are seen with non-synchronous vibrations, which are more directly related to the choice of bearing. Of the eight linearised stiffness and damping coefficients, four are cross-coupling effects relating force to a perpendicular displacement or velocity. These cross-coupling effects cause the sub-synchronous vibrations referred to as oil film whirl or half speed whirl, in which the shaft centre precesses around the bearing centre at some fraction of the shaft rotational speed. This condition can give rise to large vibration amplitudes, and on major damage to the bearing surface. Whilst normally thought of as being due to the existence of the bearings, cross-coupling forces can arise from other components (such as seals) and from the machine itself (aerodynamic cross-coupling). However, the bearings potentially offer the easiest route to solving such problems, provided that the necessary design and manufacturing capabilities exist.

The analysis of an oil film whirl, as with any vibration problem, must take account of all the details of the machine and its bearings. However, assuming that the rotor is rigid and is simply supported in two bearings, very useful design information can be given on the resistance to whirl of any bearing type (2). As an example, Fig. 13.2 shows (in dimensionless form) the predicted region of instability (sub-synchronous) for a conventional cylindrical bore bearing. It is far easier to picture the changes which can be made to alleviate oil film whirl problems by studying a simple cylindrical bore case, with its single value of clearance, than by using a profile bore bearing, with its two clearances. However, the same concepts apply. Note that there are two obvious ways of staying within the stable region: maintaining a low value of the dimensionless critical mass or a high value of eccentricity ratio. A low critical mass implies tight clearance, low speed (normally not a design option), low mass (this is the effective mass of the rotor, at the bearing position, which has to be driven into the whirl), or a high applied load. A high eccentricity ratio can be obtained by de-rating a bearing, normally reducing the length or adding grousing within the loaded area. The eccentricity ratio may also be increased by increasing clearance, which apparently contradicts the stabilizing effect of a tighter clearance within the dimensionless critical mass term.

The dashed line on Fig. 13.2 is a typical operating line resulting from a change in clearance. Near to point A a reduction in clearance is required to move the operating point into the stable zone (but this may cause other problems with operating temperatures). At point B a reduction in clearance merely moves the operating point further into the unstable zone, and to obtain stability the clearance must be increased. Therefore, simple modifications to a bearing showing signs of a sub-synchronous whirl must start from a knowledge of the conditions under which the bearing is operating.

Profile bore bearings show increased resistance to sub-synchronous vibrations, as shown in Figs 13.3a and b. Here a slightly modified critical mass number

$$M_{Cr} = \frac{MCrN^2}{W}$$

is shown plotted against a dimensionless load number

$$W^* = \frac{W}{\frac{Cr}{B} \eta \mu}$$

where

- $b$ is the bearing length (m)
- $Cr/N$ is the bearing diametral clearance (m)
- $d$ is the nominal shaft diameter (m)
- $N$ is the rotational speed (rev/s)
- $\eta$ is the effective dynamic viscosity (N s/m²)

for five different profiles (using typical presets — see Fig. 13.1), with the cylindrical bore bearing (CY) shown for comparison.

The profiles are:
- LB: lemon bore, sometimes referred to as an elliptical bore
- 4L: 4-lobe bearing, with symmetrical lobes
- 3L: 3-lobe bearing, with symmetrical lobes
- T3L: tilted 3-lobe bearing, positive tilt angle, therefore suitable for one direction of rotation only
- OH: offset halves, sometimes referred to as staggered halves; also suitable for one direction of rotation only

Manufacturing tolerances (on the bearing, the shaft and the housing) cause a change in both the lobe and the bearing clearances, and the performance of the bearing is influenced by the ratio of these as well as by their absolute values. This effect can be seen by comparing Fig. 13.3a, minimum clearance conditions, with Fig. 13.3b, maximum clearance conditions. The effect is very significant, so that any design study must take into account the expected manufacturing tolerances. Furthermore, in many cases these tolerances must be kept considerably tighter than would be expected in a conventional cylindrical bore bearing, in order to maintain the benefits of the profile. This is particularly true for smaller sizes of bearings (say less than 100 mm, 4-in bore).

If the load on the bearing is purely due to the weight of the rotor, then the information can be considerably simplified, since the mass (M) and load (W) are then related by the gravitational constant and therefore may be cancelled out. For a given oil grade, a clearance can be

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**Fig. 13.2 Oil Film Whirl Instability for a Cylindrical Bore Journal Bearing**
determined which keeps bearing temperatures acceptable, and the instability speed can then be related directly to diameter, Fig. 13.4. This diagram shows the minimum speed which can be expected before instability occurs, and for many loading conditions this minimum can be exceeded.

Rotor flexibility complicates this simplified approach, but in general it lowers the speed at which instability occurs. However, any damping within the system which is external to the bearings (e.g., from a gear mesh) can raise the speed. Experience suggests that the rigid rotor assumption is acceptable for all but very flexible rotors.

If a profile bore bearing cannot be designed to cater for the speed/load conditions, then a tilting pad journal bearing will invariably reduce a sub-synchronous vibration, provided that the problem has its origins within the bearings; if seals or aerodynamic effects are the cause then a tilting pad journal bearing may not always be able to suppress the whirl. A tilting pad bearing has a high stabilising effect because its cross-coupled stiffness and damping are extremely low, perhaps two orders of magnitude less than the direct-coupled terms. Many design methods conclude that the cross-coupled effects are zero, but in practice inertia, thermal and loading effects can combine to give significant cross-coupling [3]. Whilst a tilting pad journal bearing is an extremely valuable asset in many high-speed machines, it should not be thought of as a panacea for all vibration problems. It cannot necessarily control externally forced vibrations more effectively, and indeed may be more prone to damage under such conditions by, for example, fretting on its mechanical pivots.

3. THRUST BEARINGS

As plant is refined and updated, there are conflicting requirements for machine designers wishing to increase the loads and speeds acting upon the thrust bearing, whilst at the same time, demanding increased life and less downtime, without incurring significant penalties in terms of increases in running or capital costs.

The increase in severity of operating conditions can be divided into three areas: firstly, a demand for increased load capacity at high operating speeds; secondly, a demand for a bearing to operate at high temperatures, and finally, a demand for a bearing to accept increased levels of misalignment.

The demand for higher load capacities has often arisen either from upgrading equipment, or from design methods for calculating the thrust load not being sufficiently...
Fig. 13.6 Minimum Measured Film Thickness and Maximum Measured Pad Temperature for Different Combinations of Pivot Position and Pad Material
refined to give accurate values. Subsequent measurements on test then indicate the need to withstand higher thrust loads than originally expected. In such situations the structural envelope is already defined and resorting to a significantly larger bearing would result in an expensive re-design of both the casing and thrust collar, and in some cases the shaft as well.

Clearly the fitting of a bearing with a higher load capacity within the existing envelope results in the most convenient solution to this problem. There are several ways in which the load capacity can be increased over that of a standard centre-pivoted tilting pad thrust bearing.

Firstly, in high speed applications the temperature may be reduced with improved film thickness using directed lubrication. This is achieved by directing oil to the inlet and outlet edges of the pad adjacent to the collar, from jets fitted between the pads.

Three benefits to the hydrodynamic operation arise from this arrangement:

a) The feed oil to the pad is at supply temperature instead of the mean housing oil temperature (which approximately to the housing outlet temperature),

b) Higher local oil velocities exist around the pad, thus increasing the local transfer of heat and reducing the pad temperature.

c) A greater proportion (but by no means all) of the hot oil from the previous pad can be displaced from the collar.

The overall effect is to reduce the operating temperature, which in turn results in a higher effective oil viscosity and a thicker hydrodynamic film.

This method is particularly effective at high speeds. It allows the function of the collar to rotate in air, which has the added benefit of reducing the power consumed by, in some cases, 60% to 70%. On high speed machinery the power saving alone may well justify the increased complexity of directed lubrication because of the saving in energy costs resulting from reduced power loss.

For example on a large generating set the saving may be as high as 400kW (540hp). This energy may now be turned into a saleable commodity and there will be a reduction in the capital cost of the plant resulting from the reduced oil cooling requirement of the bearing. The power loss on such a bearing with both flooded and directed lubrication is shown in Fig.13.5.

Offset pivoted pads may also be used to improve the load capacity of a bearing. This method is achieved by moving the pivot from the centre of the pad towards the trailing edge. In this case the pad operates with a greater tilt thus taking in more cool oil. Improvements in both the pad operating temperature and the film thickness are achieved. However, this is at the expense of a loss of reversibility of the bearing and, in a double thrust unit, it has the added complications of the necessity to add or modify components if it is required that the bearing cannot be incorrectly assembled in the field.

A third method is to change the material of the pads from steel to one of very high thermal conductivity. The most commonly used is copper 1% chromium (Cu-Cr). This alloy, in its hardened condition, is as hard and strong as the steel from which pads are normally made, but has a thermal conductivity approaching that of pure copper (nine times greater than steel).

The effect of the high conductivity under high shear rate oil film conditions, i.e. thin films and/or high speeds, when compared with steel pads, is twofold. It reduces the pad operating temperature by allowing heat to circulate within the pad with reduced temperature gradients, which in turn reduces the amount of thermal crowning which occurs. The result is a large reduction in maximum operating temperatures which, in turn, increases the effective viscosity and leads to thicker operating oil films.

The various methods of improving the load capacity of a thrust bearing outlined above may be combined. The bearing having the highest load capacity will be an offset-pivot copper chrome backed bearing with directed lubrication.

The results of an experimental investigation into the effect of these various changes in pivot position and material are illustrated in Fig.13.6, where direct comparisons in terms of film thickness and pad temperature have been made for different loading conditions on an 8112 Glacier standard thrust bearing (125mm, 5 in collar). Fig.13.7 compares an operational envelope defined by a minimum film thickness and maximum pad temperature. The limits used are not necessarily recommended operational limits, but are values used to enable the relative performances to be compared. Fig.13.8 compares the effect of using an offset pivoted Cu-Cr pad instead of a standard centre-pivoted steel pad on a large thrust bearing (650mm, 25.5in collar).

The advantages of using the increased load capacity, obtained by material and pivot changes, have been demonstrated in users' plants on several occasions. One case concerned a gas turbine where approximately double the calculated design load was found to be acting in
but under favourable conditions these limits can be exceeded without damage to the bearing. Fig. 13.11 shows a bearing from a pump where, due to internal wear, the thrust load was five times the design limit. The only observed damage was slight witness marks on the pivots, but a high level of cleanliness within the lubrication system had been maintained.

For the majority of applications tin based white metal historically has been the preferred material. This is not without justification since it still reigns supreme as the most seizure resistant and dirt accommodating bearing alloy available. However, there are applications where these qualities do not outweigh other disadvantages. These conditions are high temperature, corrosive, environments and applications where thermal cycling, resulting from regular changes in operating conditions, is prevalent.

To meet the demand for bearings to operate in high temperature environments, there is a choice of materials. For moderately high temperatures, aluminium 40% tin is used and this has found favour particularly in high speed gas turbine applications; however, this material itself has an operating limit below 155°C (310°F).

For applications which demand operation at even higher temperatures, polymer bearings may be used. Geothermal pumps are one application where such bearings are supplied. These pumps may be lowered into deep wells to pump hot water or other hot natural fluids to the surface. The pump casing temperature can be as high as 300°C (570°F), which is well beyond the temperature at which the common low melting point bearing alloys operate. Bronzes have been used, but the film thicknesses at such high temperatures are extremely small and seizure is therefore common. A bearing supplied for this application, coated with Glumat 58™, is illustrated in Fig. 13.12; this has a potential working temperature up to 350°C (662°F).

Corrosive environments arise either from contamination of the oil by process fluids, or from the lubricant itself degrading. The appropriate bearing material will depend upon the chemicals involved. Generally acidic environments are best met by aluminium based materials.

![Fig. 13.9 Excessive Load at High Speed causing Plastic Flow of the Bearing Surface.](image)

![Fig. 13.10 Operating Point of a Nominally Overloaded, but in Practice Satisfactory, Thrust Bearing - Compared with Normal Design Limits.](image)

![Fig. 13.11 Bearing after Running at the Operating Point Shown in Fig. 13.10.](image)
It is expected that some polymer linings will operate well in corrosive environments, but no field experience is available.

An effect known as faceting or thermal ratcheting occurs when tin based whitemetal is thermally cycled. It is caused by the grains of metal expanding at different rates along the different crystal axes. If the thermal stresses built up within the grains exceed their yield point, then permanent deformation will occur and this may be seen as a "crazy paving" effect on the surface, with measurable differences in height between adjacent grains, see Fig.13.13. Excessive faceting can give rise to intergranular cracks. The simplest solution, assuming the plant operating cycle cannot be changed, is to change from tin based to lead based whitemetal, where this anisotropy does not exist.

![Fig. 13.13 Faceting Damage to a Thrust Pad Surface Due to Thermal Cycling.](image)

![Fig. 13.14 Improvement in Operating Envelope Due to the Use of Di-Ester Based Synthetic Lubricant.](image)

One variable so far not considered has been the choice of lubricant. With mineral oils a balance has to be drawn between the increased shear rate generating more heat with a thick lubricant (resulting in a bearing running hotter with increased power loss for a small improvement in film thickness) and using a thinner lubricant, which will run cooler with possibly a thinner oil film. In general, thicker lubricants are used at low speed and thinner ones at high speeds.

However, with the general availability of di-ester based synthetic lubricants there is the opportunity of obtaining thicker oil films, with their inherent improvements in reliability, without the penalties of increased power loss or higher operating temperatures. The reason for this is not understood, but measurements have shown that film thicknesses are increased significantly, as is seizure resistance. Synthetic oils have been marketed industrially on the basis that their increased initial cost may be justified by the longer maintenance intervals; these lubricants are much more stable and do not degrade in the same way that mineral oils do. The increased reliability has been noted by operators, and it is thought that this is due to the (measured) increase in hydrodynamic film thickness. Results from tests comparing the effects of different oil types is shown in Fig.13.14.

Misalignment will always occur to some extent in any mechanical equipment. Excessive misalignment will result in significant differences in load sharing around the bearing. The most highly loaded pads may well suffer premature damage and then shed their load onto adjacent pads so that, in the worst cases, a "domino effect" could arise. Mechanical systems for sharing the load between pads are built into some ranges of bearings, such as the Glacier CO shown in Fig.13.15. The commercial demand for such a range of bearings stems from specifications demanding such systems, generally originating from North America. Although such bearings meet this requirement, they do so by significantly increasing the number of components and consequently the overall reliability of the system may not be any better. The incidence of bearing damage resulting from misalignment is very small and probably less than the incidence of fatigue damage. Fatigue normally results from a swinging collar and such a form of misalignment is likely to create fretting on the levers of a self-aligned bearing, thus making it less reliable than a standard unit. The choice, therefore, has to be carefully made, based upon the type of misalignment expected.

The various ways of improving the reliability of a bearing outlined above, generally result in the bearing
having increased load capacity. The increased load capacity could be used by the designer to enable a smaller unit to be fitted, with a significant reduction in the power absorbed.

However, before advantage is taken of this extra capacity, a higher level of confidence is needed in the ability of some industries accurately to calculate the bearing loads. It is not unusual in mechanically balanced systems, such as multi-stage pumps and turbines, for calculated thrust loads to be incorrect by a factor as high as two or three and, doubtless, there are many machines operating in the field under conditions that would be considered completely unacceptable had the actual bearing loads been known at the design stage. It is for this reason that bearings still tend to be conservatively rated and the high intrinsic overload capacity explains the very high level of reliability achieved in the past.

As improvements in computer-aided-design techniques give a higher confidence in load predictions for machines it will become safer to use smaller, higher loaded bearings with lower power consumption. With the selection of some of the alternatives outlined above, and a high level of lubricant filtration, there is every reason to believe that bearing reliability will continue to improve.

ACKNOWLEDGMENTS

The tests carried out on synthetic oils were sponsored by Exxon Chemical Co.

Glacier CQ is the subject of British Patent Application No. 42 945/78.

Glamat 59™ is the subject of British Patents 1 513 727, 1 513 378, 1 523 282, 1 521 531, 1 556 047, 1 590 254, and PCT applications 81.00125-2 and 81.00129.

REFERENCES

