Current Practice in Tilting Pad Bearing Use


INTRODUCTION

While this paper is primarily concerned with tilting pad bearings, this introduction and the subsequent section on load capacity are applicable to any hydrodynamic type bearing, whether of the tilting pad or non-tilting pad type. The remainder of the paper concentrates on tilting pad bearings though their relationship to non-tilting bearings is also considered.

Hydrodynamic bearings operate on the principle of a lubricant being dragged into a converging film area which is formed between the stationary and rotating element in a bearing system. Normally the bearing element is the stationary part and is lined with a bearing material and the rotating element is the shaft. The nature of the convergent areas in a journal and thrust bearing is shown in Fig 1. When the lubricant is dragged into this converging space a pressure is generated which opposes the load applied through the bearing system. The objective of the bearing designer is to achieve for given operating conditions a film thickness sufficiently large to prevent contact between the stationary and rotating elements of the bearing system. The relationship between the film thickness and pressure generated, is governed by hydrodynamic equations which depend on the lubricant viscosity, lubricant temperature, relative velocity between the surfaces and the design of these surfaces. With modern computer programmes it is possible to calculate with a fair degree of accuracy the film thickness and maximum lubricant temperature for most well known bearing configurations: these parameters form the basis of the load capacity ratings used for hydrodynamic bearings.

LOAD CAPACITY

The load capacity of a hydrodynamic bearing, unlike a rolling element bearing, has no calculable relationship with time in operation. Providing a hydrodynamic bearing is operated within its design parameters with clean oil, it should run for an indefinite time with no wear. In practice, wear does occur with hydrodynamic bearings in the course of time, but lives well in excess of 50 000 hours are often achieved in correct installations.

The limits of load capacity in a hydrodynamic bearing can be divided into two categories:

1) The hydrodynamic limit of adequate film thickness

2) The mechanical and material limits which are a function of mechanical design and bearing surface material.

In Fig 2 is shown a typical operating envelope for a plain (non-tilting) journal bearing and Fig 3 shows the equivalent envelope for a tilting pad bearing. It will be noted that in the case of the tilting pad bearing, the peak of the envelope is cut off; this is a rather arbitrary limit imposed by deflections and stresses in the pivot. This limit does not occur in a plain journal bearing and its importance is probably over-emphasized in tilting pad bearing practice. It should be noted that the fundamental envelopes shown in Figs 2 and 3 depend on
Fig 1. Hydrodynamic Converging Wedge Principle on Journal and Thrust Bearing.

Fig 2. Limits of Safe Operation for Plain Journal Bearing.
the definition of an acceptable minimum film thickness for slow speed operation and an acceptable maximum surface temperature for high speed operation. Suitable values of minimum film thickness for use with tilting pad bearings are given in Ref 1 while acceptable values of maximum temperature will be determined by the choice of bearing surface material which is dealt with in the next section.

MECHANICAL DESIGN

Bearings of the hydrodynamic type are normally provided with a relatively soft surface material for two reasons:

1) To enable hard dirt particles to embed themselves

2) To ensure that any wear which occurs during boundary lubrication is confined to the replaceable bearing element.

Fig 4 shows some typical bearing surface materials which are used in rotating type machinery. An overwhelming majority of applications in this type of machinery use whitemetal (babbit) surface material either of the tin based or lead based type. This material is quite soft and has good embedability while at the same time it has excellent boundary lubrication properties when in contact with steel under oily conditions. Alternative materials should only be used if whitemetal is unsatisfactory for some reason. The principal reasons why whitemetal may be unsatisfactory are temperature or fatigue. In tilting pad bearings fatigue of the whitemetal surface is unusual and the usual problem is bearing surface temperature under design conditions. Generally speaking, a designer will prefer to reduce the temperature of the bearing surface, however, rather than use a bearing with a higher surface temperature capability as at higher temperatures there may be a problem of degradation of the lubricant. It is for this reason that even today most high speed bearings still have whitemetal bearing surfaces though with the increasing availability of synthetic lubricants this is slowly changing.

Before leaving the subject of bearing materials, the question of pad backing material needs to be mentioned. The majority of tilting pad bearings nowadays have a steel pad backing material though historically bronzes have been extensively used. A relatively recent development has been the substitution of a high copper content alloy for the steel backing material to help transfer the heat away from the bearing surface. Usually 1 or 2% of some other element alloy is added to the copper to give it mechanical strength with the normal white metal surface layer being retained. In Fig 5 from Ref 2 some results are given for a tilting pad thrust bearing comparing the maximum surface temperature with steel and copper/chrome backing materials. The superiority of the copper/chrome backing material is most evident especially at high speeds.

Another feature of the pad design which affects surface temperature is the nature and position of the pivot. This can either be in the circumferential centre of the pad or it can be offset towards the trailing edge (say 0.58-0.6). It can also either be of a point contact type or be of the line contact type. The effect of using an offset versus a centre pivot pad is shown in Fig 6 where it will be seen that it is only at the higher speeds that the offset pivot gives significant advantages; as it has some practical disadvantages, the use of this pivot position is often confined to high speed machines. Both point
Fig 3. Limits of Safe Operation for Tilting Pad Bearing.

<table>
<thead>
<tr>
<th>MATERIAL TYPE</th>
<th>SAE EQUIVALENT</th>
<th>NOMINAL COMPOSITION (%)</th>
<th>NOMINAL HARDNESS (HV)</th>
<th>MAX DESIGN SURFACE TEMP (°C)</th>
<th>°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>TIN BASE WHITEMETAL</td>
<td>SAE 12</td>
<td>—</td>
<td>3.5</td>
<td>75</td>
<td>89</td>
</tr>
<tr>
<td>LEAD BASE WHITEMETAL</td>
<td>SAE 13</td>
<td>—</td>
<td>0.5</td>
<td>83.5</td>
<td>10</td>
</tr>
<tr>
<td>ALUMINIUM BASE ALLOY</td>
<td>—</td>
<td>60</td>
<td>—</td>
<td>10</td>
<td>6</td>
</tr>
<tr>
<td>ALUMINIUM BASE ALLOY</td>
<td>SAE 770</td>
<td>92</td>
<td>1</td>
<td>40</td>
<td>6</td>
</tr>
<tr>
<td>COPPER BASE ALLOY</td>
<td>SAE 48</td>
<td>70</td>
<td>30</td>
<td>45</td>
<td>170</td>
</tr>
<tr>
<td>COPPER BASE ALLOY</td>
<td>—</td>
<td>20</td>
<td>25</td>
<td>2</td>
<td>44</td>
</tr>
<tr>
<td>ENGINEERING POLYMER</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>20</td>
<td>200</td>
</tr>
</tbody>
</table>

NOTE: THIS IS THE CALCULATED TEMPERATURE MEASURED AT THE HOTTEST POINT ON THE BEARING SURFACE WHICH IS SAFE FOR CONTINUOUS OPERATION. DAMAGE TO THE BEARING SURFACE WILL NOT OCCUR UNTIL TEMPERATURES AT LEAST 30°C (54°F) HIGHER THAN THESE VALUES ARE REACHED. IN THE CASE OF THE COPPER BASE ALLOYS THE LIMITATION IS BREAKDOWN OF THE LUBRICANT OIL WHICH MAY OCCUR AT TEMPERATURES IN EXCESS OF 170°C (338°F) RATHER THAN A MATERIAL LIMITATION.

Fig 4. Bearing Surface Materials for Hydrodynamic Bearings.
Fig 5. Comparison of Thrust Pad Surface Temperatures with Steel and Copper Backed Thrust Pads. 8 Pad Thrust Bearing 124 mm (5 in) Outside Dia. Specific Load 4 MN/m² (571 lb/in²).

ZONE WHERE T_max = 130°C FOR CENTRE PIVOT AND 130°C FOR OFFSET PIVOT PADS.

Fig 6. Comparison of Offset and Centre Pivoted Thrust Pad Temperatures versus Load and Speed. Pad Size 40 mm (1.59 in). Area 1635 mm² (2.54 in²).
and line pivots are extensively used and it is difficult to say that one has a distinct advantage over the other though the line pivot in tilting pad journal bearings probably reduces the incidence of pivot fretting. The various features mentioned above can of course be combined and Fig 7 shows a comparison between offset and centre pivot thrust pads using steel and copper/chrome backings.

Two other features of mechanical design of the pads used in tilting pad bearings need to be mentioned, firstly, the bearing surface shape and secondly, the plan shape of the pad itself.

The surface profile of a thrust pad must not be concave and, therefore, in practice a small crown is often machined on it to ensure that no concavity exists; this also has the advantage of helping at start-up specially on a centre pivoted pad. From the point of view of bearing operation, the best plan shape of a tilting pad is approximately square and therefore it must have a circumferential width approximately equal to the radial width. Originally thrust pads tended to be segments of the required inside and outside circles but more recent designs have tended to use other shapes as shown in Fig 8. These more recent designs have the advantage that they are not confined to use with one particular diameter and they also allow more space between pads to ensure cool lubricant is available for each succeeding pad.

LUBRICATION ARRANGEMENTS

There are two principal systems of lubricating tilting pad bearings: force lubricated systems with an external pump and cooler and self-contained systems where the oil is circulated and cooled within the bearing assembly. This section will concentrate on force lubricated systems while some description of self-contained systems will be included in the application section below.

Conventionally lubricated tilting pad thrust bearings have been installed in a chamber filled with oil generally as shown in Fig 9. The number and position of oil inlet passages and outlets is not particularly important though that shown in Fig 9 is one of the simplest and most effective. In Fig 10 some of the alternative inlet/outlet systems are illustrated all of which are adequate and will give very comparable performance (Ref 3). For high speed bearings, the fully flooded housing is now often replaced by a system of Directed Lubrication in the interests of power loss and oil flow saving. The installation of a tilting pad thrust bearing using this system is shown in Fig 11 where it will be noted that the bearing housing is no longer flooded and seals are not required where the shaft passes through the housing. An intermediate system between fully flooded as shown in Fig 10 and directed as shown in Fig 11 is sometimes used: this consists of restricting the oil inlet size to meter the oil and having only partially restricted oil outlets. This latter system is generally called inlet orificing. In Fig 12 the savings achieved with Directed Lubrication as compared with flooded lubrication are shown in the right-hand diagram while in the left-hand diagram the power loss savings with directed, flooded and inlet orificing are compared. It should be noted that the power loss savings with inlet orificing can only be achieved at the expense of higher thrust pad temperatures (Ref 4) and, therefore, various forms of Directed Lubrication are now being widely used for high speed machines where bearing safety is important.
Fig 7. Comparison of Operating Envelopes for Different Pad Designs. Same Bearing as Fig 5.

Fig 8. Pad Shapes for Tilting Pad Thrust Bearings.
Fig 9. Typical Installation of Flooded Lubrication Tilting Pad Thrust Bearing.

Fig 10. Alternative Oil Inlet/Outlet Arrangements for Tilting Pad Thrust Bearings.
Fig 11. Typical Installation of Directed Lubrication Tilting Pad Thrust Bearing.

Fig 12. Comparison of Operating Characteristics of Flooded Lubrication, Pressurized Casing Lubrication and Directed Lubrication. Same Bearing as fig 5.
In Fig 13 a typical installation of a tilting pad journal bearing is shown; the oil is introduced to the bearing via the annulus around the bearing housing and normally the quantity is controlled by the clearance of the end seal plates. If necessary some additional control of oil flow can be provided by restrictions at the inlet providing the housing pressure in the journal bearing is not allowed to drop below 0.1 bar (1.4 lbf/in²) above the ambient pressure. Especially when a Directed Lubrication thrust bearing is associated with a tilting pad journal bearing, there can be constructional advantages if Directed Lubrication is also used for the tilting pad journal bearing. This is illustrated in Fig 14 where in the main diagram a tilting pad journal bearing with Directed Lubrication is shown and in the inset the effect of fitting a thrust face with Directed Lubrication is illustrated. The potential power loss savings from using Directed Lubrication in a tilting pad journal bearing are considerably less than in a thrust bearing and the principal benefits are in reducing oil quantity and simplifying construction. In Fig 15 the power loss of a flooded and Direction Lubrication tilting pad journal bearing are shown and it will be noted that the power loss savings are dependent not only on sliding speed but also on the load conditions (eccentricity ratio) of the bearing.

TILTING PAD APPLICATIONS

In broad terms, high speed, long life applications are likely to use hydrodynamic type bearings while low speed, short life applications are likely to use rolling element type bearings but there is a very large range of applications where both types of bearings could be used and the choice between them is not clearly defined. In the left-hand column of Fig 16 are listed four principal considerations which should be borne in mind when a choice between hydrodynamic and rolling element bearings is made. For many applications the most important factor will be a combination of bearing life and cost; the rolling element bearing has a calculable life while the hydrodynamic bearing, if it is correctly used, should have a considerably longer life than the rolling element bearing. Thus for capital equipment where the cost of downtime is large, hydrodynamic bearings are favoured while for auxiliary equipment where stops for maintenance are easier to arrange or life requirement is short, rolling element bearings tend to be favoured.

Assuming the choice is a hydrodynamic bearing, the question then arises as to type: a major division here is whether it should be a plain (non-tilting) type or a bearing of the tilting pad type. In the right-hand column of Fig 16 some of the considerations governing this choice are indicated. While life can be important with this choice, normally it ranks equally with the other three factors though in any one particular application, one factor may predominate. For instance, on high speed, lightly loaded journal bearings, the dynamic characteristics of the journal bearing will virtually dictate the type of bearing to be used: this particular aspect is discussed in further detail below. When there is the choice between thrust bearing types, the question of load capacity is likely to be the predominant one: for very light loads (say less than 3.5 bar or 50 lbf/in²) a plain grooved thrust washer might be adequate while for heavy loads, say 20 bar or 255 lbf/in² and up, tilting pads are normally used. For medium loads between these two extremes, taperland thrust washers may often be the best choice.

In Fig 17 the stability characteristics of various journal bearing
Fig 13. Typical Lubrication Arrangements for Tilting Pad Journal Bearing.

Fig 14. Arrangement of Directed Lubrication Tilting Pad Journal Bearing.

Fig 15. Power Loss Saving with Directed Lubrication Tilting Pad Journal Bearing. 120 mm (4.72 in). b/d = 1.
### Fig 16. Factors Affecting Choice of Rolling Element and Hydrodynamic Plain Bearings.

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Suitable Direction of Rotation</th>
<th>Resistance to “Half-Speed” Whirl</th>
<th>Stiffness and Damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylindrical Bore</td>
<td><img src="#" alt="Diagram" /></td>
<td>WORST</td>
<td>MODERATE</td>
</tr>
<tr>
<td>Cylindrical Bore with Shaped Groove</td>
<td><img src="#" alt="Diagram" /></td>
<td>MODERATE</td>
<td></td>
</tr>
<tr>
<td>Lemon Bore</td>
<td><img src="#" alt="Diagram" /></td>
<td>MODERATE</td>
<td></td>
</tr>
<tr>
<td>Three Lobe</td>
<td><img src="#" alt="Diagram" /></td>
<td>INCREASING</td>
<td>GOOD</td>
</tr>
<tr>
<td>Offset Halves</td>
<td><img src="#" alt="Diagram" /></td>
<td></td>
<td>EXCELLENT</td>
</tr>
<tr>
<td>Tilting Pad</td>
<td><img src="#" alt="Diagram" /></td>
<td></td>
<td>BEST</td>
</tr>
</tbody>
</table>

### Fig 17. Comparison of Journal Bearing Types.
profiles are compared; it must be emphasized that a diagram of this sort can only be used as a rough initial guide and that the final choice of profile to be used is best made by comparing the dynamic characteristics of several types in conjunction with a rotor stability programme. There is no one particular journal bearing which will solve all problems in rotating machinery though probably the most versatile type is the tilting pad type with centre pivoted pads. For this type of bearing it is possible with modern computer techniques to predict the dynamic coefficients with some degree of accuracy taking full account of the geometry variables which are possible in tilting pad journal bearings. The effect of many variables on dynamic characteristics are given in Figs 18 and 19 which are taken from Ref 5. It will be seen from these diagrams that the effect of design variables on stiffness and damping coefficients is very dependent on the load and speed of the application and, therefore, it is necessary to consider each application on its own merits before deciding the exact design of the bearing. The accuracy of theoretical predictions of stiffness and damping coefficients may be queried; therefore some comparisons with experimentally measured coefficients are given in Fig 20 and some comparisons between theoretical methods of calculation are given in Fig 21. It will be seen that generally speaking, the methods of theoretical calculation are in agreement with one another and the calculations themselves are in reasonable agreement with experimental results.

In conclusion, some applications of tilting pad bearings in complete assemblies will be studied. Applications can be divided into two major categories: vertical shaft and horizontal shaft. For each of these categories the method of lubrication can be another useful division of types i.e. force lubricated and self-contained.

In Fig 22 a diagrammatic illustration of a horizontal force lubricated double thrust and journal assembly is shown. This type of assembly is commonly used on machines such as steam turbines and compressors and it will be noted that the double thrust is installed on either side of a single thrust collar. For some applications a more compact assembly can be achieved by having a double thrust collar in between which the double thrust and journal assembly is placed. An assembly of this type is shown in Fig 23: the right-hand thrust face is of the equalised type, while the left-hand thrust face is of the non-equalised type as it has to take surge loads only. Both thrust faces are lubricated by a Directed Lubrication system and the journal bearing is of the tilting pad type. Another feature which will be noted is the provision of instrumentation for monitoring bearing and machine performance and this is a feature of many current bearing applications. In this assembly both main thrust face and journal pads have temperature sensors, while both axial and radial shaft position probes are fitted as well as a shaft speed sensor.

In Fig 24 is shown a diagrammatic illustration of a vertical self-contained bearing assembly. This type of assembly is commonly used on vertical motors and pumps and, in particular, in nuclear applications where the large oil reservoir gives a degree of safety for running during emergency conditions. In Fig 25 an actual application of a double thrust and journal assembly to a vertical nuclear coolant pump motor is shown. During normal operation the oil is circulated through the assembly by the centrifugal pumping action on the thrust collar with cooling being provided by a water cooling coil in the outer reservoir. Under emergency conditions, however, the bearing must be
Fig 18. Effect of Various Design Parameters on Stiffness of a Tilting Pad Journal Bearing.

Fig 19. Effect of Various Design Parameters on Damping of a Tilting Pad Journal Bearing.
Fig 20. Comparison of Theoretical Predictions and Experimental Measurements of Dynamic Characteristics for a 20 mm Diameter, 4 Pad, Tilting Pad Journal Bearing.

Fig 22. Outline Arrangement of Single Collar, Horizontal Flooded Lubrication Bearing Assembly.

Fig 24. Outline Arrangement of Vertical Self-contained Bearing Assembly with Air Cooling.
Fig 23. Arrangement of Double Collar, Horizontal Bearing Assembly with Directed Lubrication Thrust Bearings.
Fig 25. Arrangement of Vertical Self-Contained Bearing Assembly with Water Coil Cooling for Nuclear Coolant Pump Motor.
capable of running for significant periods with the cooling water turned off during which time the heat generated by the bearing is absorbed into the mass of the cooling oil and, to some extent, dissipated through the housing to the surrounding air. It will be noted that both upper and lower thrust bearings are fitted with hydrostatic jacking as thrust loads in both directions can occur at starting. The hydrostatic oil is supplied at high pressure from an external electrically driven pump which normally is only in operation during starting and stopping cycles. Again, however, in an emergency, the bearings must be capable of taking the full load during shutdown without the provision of high pressure hydrostatic oil.

ACKNOWLEDGEMENTS:

The author would like to thank The Glacier Metal Co Ltd for permission to publish this paper and express his gratitude to his colleagues for help in its production.

REFERENCES:


