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Journal Bearings in Turbine and Generator Systems

Principles, Design, and Operation

Technical Brief — *Maintenance Management and Technology Research Area Generation Sector, Steam Turbines and Generators Program*

Abstract

Fluid film journal bearings are the most common type of bearing used in power plant rotating equipment because they are capable of carrying high loads at high shaft speeds. Although they can sometimes appear to be as simple as a cylindrical sleeve supporting the shaft, their principle of operation is not trivial, and a substantial amount of deliberate consideration and engineering experience is required to select the appropriate bearing for the corresponding application and to operate it safely. This report provides an introduction to the principles of hydrodynamic lubrication, the operating characteristics of various bearing configurations, and an overview of bearing damage mechanisms.

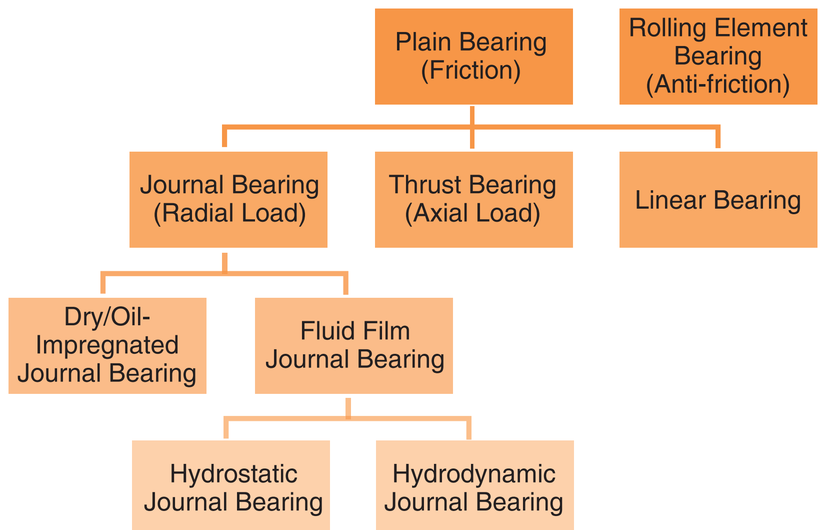

Figure 1. Classification of bearings (focusing on journal bearings).

Introduction

A journal bearing consists of a cylindrical sleeve, fitted with a tight clearance around a rotating shaft (or journal), and is designed to support radial loads. It is a type of plain bearing, since it operates by sliding contact, as opposed to the anti-friction bearings that use rolling elements (such as ball bearings). As Figure 1 illustrates, journal bearings can be further subdivided into those that operate without lubrication or with partial lubrication (such as nylon bushings and oil-impregnated sleeve bearings), and those that operate with journal and bearing surfaces completely separated by a film of lubricant. In the latter case, the fluid film is typically maintained by hydrodynamic pressure generated when the viscous lubricant is swept into a tight clearance by the relative motion of the journal and bearing. However, if the application is such that the shaft speed is insufficient to generate adequate film pressure hydrodynamically, the lubricant can be introduced at high enough pressure to separate bearing and journal surfaces even in the absence of shaft rotation (hydrostatic lubrication).

Each of these bearing types has a specific range of operability, as well as an associated cost. At the high shaft speeds and loads typical of power-generating equipment, the bearing of choice is of the hydrodynamic type. This report therefore focuses on hydrodynamic journal bearings.

The intent here is not to instruct the reader on how to design a bearing from scratch. Rather, it is to provide a basic understanding of journal bearing operation and the factors that should be considered in the choice and use of hydrodynamic journal bearings.

Principles of Hydrodynamic Lubrication

Figure 2 illustrates the basic principle of hydrodynamic lubrication. There is a radial clearance between the journal and bearing (which is usually on the order of 1/1000 of the bearing diameter), which is flooded with a viscous lubricant. The external or gravity load (W) forces the journal to be positioned with an eccentricity (e) relative to the bearing center, forming a converging wedge in the clearance space. As the shaft rotates, the lubricant is dragged into the converging wedge by virtue of the no-slip condition on the journal surface. A resulting pressure field (p) is generated, as illustrated in Figure 2, providing the desired supporting force and maintaining a minimum film thickness (h_{min}) sufficient to prevent contact between the two surfaces.

The generation of positive hydrodynamic pressure in a converging fluid film can be explained by the following arguments. We first make the simplifying assumption of an infinitely long journal, reducing the problem to two dimensions. Noting that the radial dimension of the fluid film is much smaller than the circumferential dimension justifies also neglecting the curvature in the problem, and designating the circumferential direction as a Cartesian “x” coordinate and the radial direction as the “y” coordinate. The corresponding velocity components are “u” and “v,”

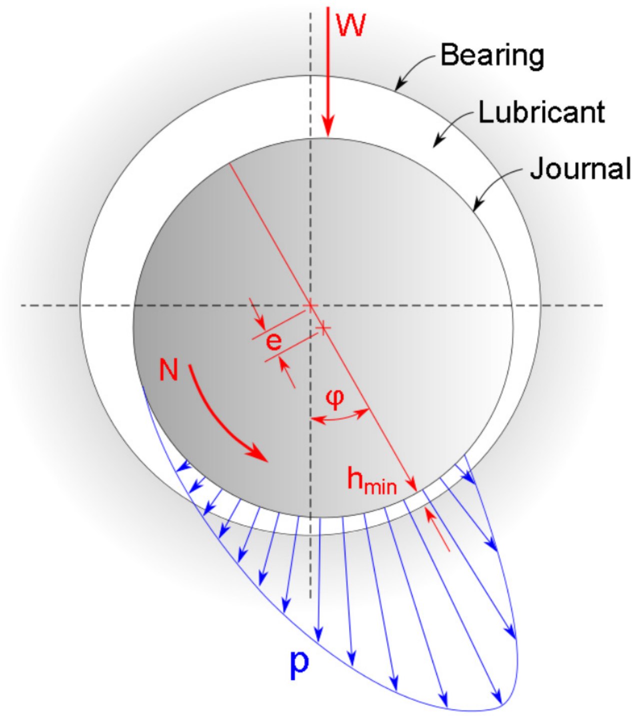

Figure 2. The principle of hydrodynamic lubrication in journal bearings.

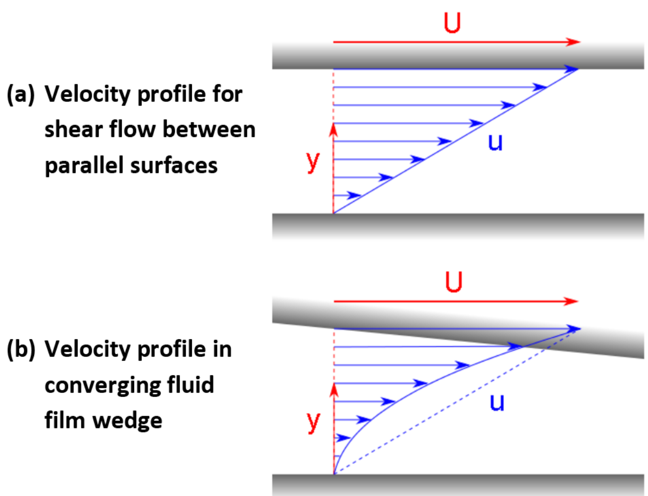

Figure 3. Shear flow between parallel surfaces and in a converging wedge.

respectively. It can be demonstrated by dimensional analysis (beyond the scope of this text) that fluid inertia terms can be neglected, leading to a reduced momentum (Equation 1):

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \qquad \textit{(Eq. 1)}$$

Consider the case of a fluid film between parallel surfaces separated by a distance y, with the top surface sliding at a velocity U, as shown in Figure 3a. This simple shear flow will have a linear velocity profile, with u=0 at y=0, to u=U at y=h. Such a profile has a zero second derivative in the y-direction, hence no variation in pressure in the x-direction. A journal with no eccentricity will not generate hydrodynamic pressure.

Now, consider the case of a converging wedge, as shown in Figure 3b. Once again, the boundary conditions are set by the bounding surfaces. If we postulate a linear velocity profile at some x location, we find that the shape of the velocity profile at other x locations must deviate from the linear in order to accommodate the same mass flow, despite the changing film thickness, h(x), and the fixed boundary conditions (u=0 at y=0, to u=U at y=h). Although not necessarily intuitive, the change will tend to be concave for converging films (see Figure 3b). This creates a velocity profile with a positive second derivative with respect to y, and by Equation 1, an attendant increase in pressure. Conversely, the pressure will decrease in diverging fluid films. In fact, the pressure downstream of the minimum film thickness (h_{min}) can fall so low as to induce cavitation (due to both the release of dissolved gases and lubricant vaporization). The onset and extent of cavitation not only affect the bearing load capacity and stability, but also can cause surface damage [1].

Note that journal eccentricity will vary in accordance with the load. At a given shaft speed (N) and lubricant viscosity (μ), a higher load will lead to greater eccentricity (that is, a more severe converging wedge) so that the hydrodynamic pressure balances the applied load. We also note in Figure 2 that, although the applied load is shown as vertical, the journal center is positioned at some attitude angle, φ, relative to that. This characteristic is related to the cross-coupled stiffness of hydrodynamic bearings, which will be discussed in the later section on bearing dynamics.

Operating Regimes

The preceding discussion gives a qualitative sense of the principle of hydrodynamic lubrication. One should note the essential nature of lubricant viscosity, journal speed, and journal eccentricity (a result of the radial clearance and applied load). When some of these essential parameters are outside of the range required for maintaining a stable fluid film, the bearing may operate in a state of mixed or boundary lubrication, as surface contact cannot be prevented.

A useful plot demonstrating the various operating regimes of hydrodynamic bearings is that of friction coefficient against the non-dimensional group μN/P, where μ is absolute viscosity, N is the journal speed, and P is the unit load, defined as the load divided by the journal projected area. This plot, shown in Figure 4, demonstrates that, with sufficient viscosity and journal speed, full film lubrication can be maintained. However, if the load is increased, the lubricant viscosity drops (such as the result of high temperatures) or the shaft speed is low (such as during startup conditions), and at a certain point the coefficient of friction will rise steeply. This occurs as the minimum film thickness is insufficient to prevent asperity contact between journal and bearing surfaces. In the boundary lubrication and mixed (or thin film) regimes, friction and wear are affected by the properties of the contacting surfaces, as well as the lubricant's chemical properties (rather than the lubricant's viscosity).

Another aspect of the transition between the boundary and full film lubrication is that to the right of the curve minimum, the fluid film is stable with temperature, whereas to the left of the minimum, it is unstable. An increase in lubricant operating temperature would reduce its viscosity, leading to a smaller μN/P value. In the full film regime, this will reduce the heat generated by viscous shear, allowing the lubricant to cool down and the viscosity to increase. Conversely, in the mixed lubrication regime, the opposite is true, and the viscosity of the lubricant will continue to drop as the coefficient of friction and lubricant temperature rise.

The requirements for stable full film lubrication dictate the range of operation for hydrodynamic journal bearings, as illustrated in Figure 5. The bold lines indicate the preferred type of bearing at a particular load, speed, and diameter. Hydrodynamic bearings come out as winners at larger diameters and higher shaft speeds, where a stable fluid film can be maintained.

Bearing Dynamics

Another important aspect of hydrodynamic journal bearing operation is the journal bearing's dynamic performance. This is a very significant consideration for avoiding vibration problems and instability in rotating machinery.

A bearing has both stiffness and damping associated with it. In other words, perturbations in journal position (Δx, Δy) and the rate of such perturbations ($\Delta\dot{x}$, $\Delta\dot{y}$) will elicit reaction forces. These forces, in general, are very nonlinear [3], meaning that they are not directly proportional to journal position variation; however, a linearized approach is sufficient for predicting and avoiding dynamic instability. Therefore, for most design purposes, the linearized expression in Equation 2 is used:

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{Bmatrix} \Delta x \\ \Delta y \end{Bmatrix} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{Bmatrix} \Delta\dot{x} \\ \Delta\dot{y} \end{Bmatrix} \qquad \textit{(Eq. 2)}$$

This expression can be depicted schematically by three springs and three dampers between the bearing and journal, as shown in Figure 6. Note that the spring and damper representing the Kxx, Cxx, stiffness and damping terms, are drawn so that they react to perturbations in the x direction with forces in the x direction. A similar statement can be made for the Kyy, Cyy stiffness and damping. However, the spring and damper representing the off diagonal stiffness and damping terms (Kxy, Kyx, Cxy,

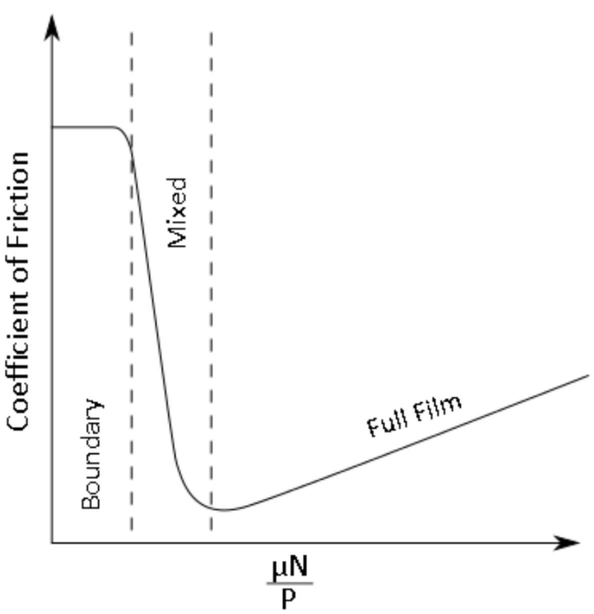

Figure 4. Regimes of full film, mixed, and boudary lubrication.

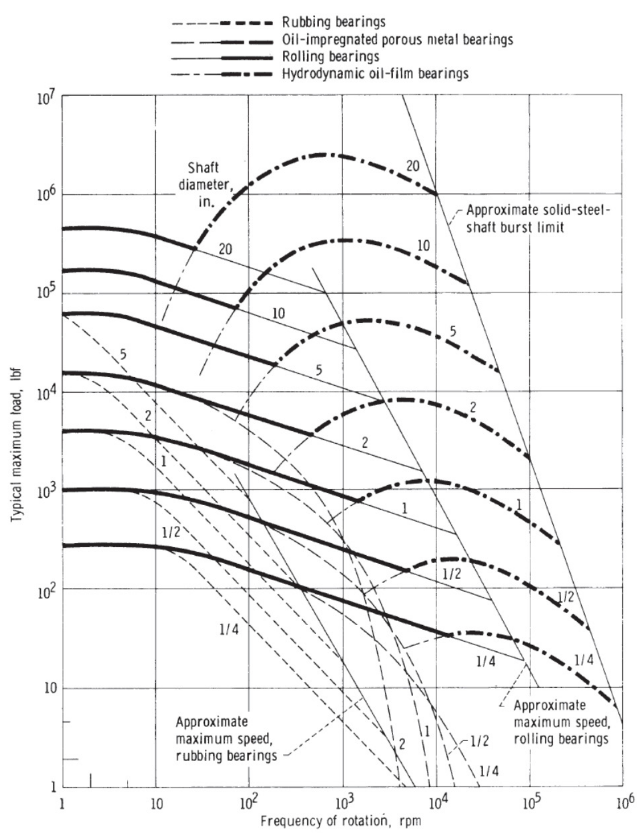

Figure 5. Operating range of bearing types. The journal bearing length is assumed to be equal to the journal diameter, and a medium viscosity mineral oil lubricant is assumed for hydrodynamic bearings [2].

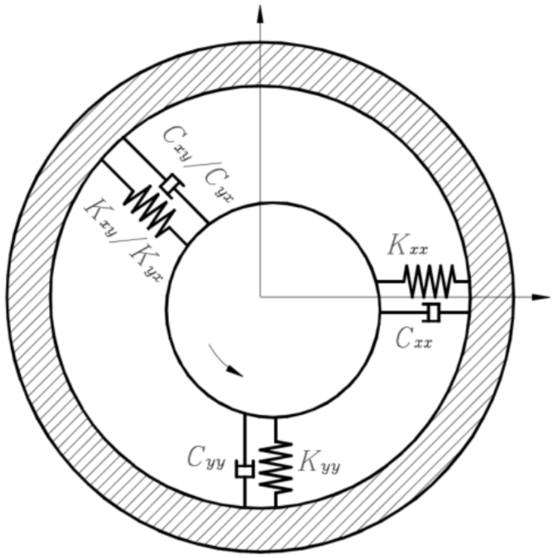

Figure 6. Schematic representation of bearing stiffness and damping terms [3].

Cyx,), which are known as the cross-coupled stiffness and damping coefficients, will react to perturbations in one direction with a force in the other direction. These cross-coupled terms explain the attitude angle (φ) shown in Figure 2, which is measured between the line of centers and the applied load. In that case, both a vertical and horizontal displacement of the journal is required to react to the purely vertical gravity load (W).

The direct stiffness and damping terms are important with regard to a machine's vibration performance, and they govern the location and amplification of the rotor's critical speeds. There is an optimum amount of bearing damping that is a function of bearing and shaft stiffness [3].

The cross-coupled stiffness coefficients are particularly relevant to bearing dynamic stability. When a dynamic force (such as that due to rotor imbalance) perturbs the journal, these cross-coupled stiffness terms tend to generate forces that encourage a subsynchronous forward whirling motion at approximately half of the shaft speed. This phenomenon is referred to as *oil whirl*, which typically occurs at lightly loaded conditions (that is, low journal eccentricity). If damping is insufficient to dissipate the energy added by cross-coupled stiffness, and when the oil whirl frequency coincides with the system's natural frequency, the shaft will resonate, leading to the more serious instability of shaft whip. If left uncorrected, shaft whip may cause catastrophic failure in a relatively short period of time. For more information on how bearing dynamic properties influence a machine's rotodynamics, the reader is referred to API Standard 684 [4].

Some bearing designs have been developed that have much better dynamic characteristics compared with the simple cylindrical bore journal bearing we have assumed in this discussion, so far. However, before we move on to more exotic geometries, a brief introduction to bearing design will illustrate the general approach and terminology used on the field.

General Design Approach

This section will introduce the basic tools for designing hydrodynamic journal bearings. Not meant as a tutorial on the process, this section is intended only to familiarize the reader with the nomenclature and general approach of journal bearing design.

Using the same principles of conservation of mass and momentum that were invoked to demonstrate the principle of hydrodynamic lubrication earlier in this report, the classical Reynolds equation can be derived (Equation 3), which relates the pressure field, p(x,z), to the film thickness, h(x):

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6U\frac{\partial h}{\partial x} \qquad \textit{(Eq. 3)}$$

Note that this equation does not assume an infinitely long journal, and it accommodates leakage of the lubricant in the z-direction (along the length of the journal).

The Reynolds equation does not have a general solution but can be solved numerically for specific bearing designs. Some of these results have been plotted as functions of the so called, Sommerfeld number, S, which is a non-dimensional group that encompasses all of the key bearing design variables [5]. See Equation 4.

$$S=\left(\frac{r}{c}\right)^2\frac{\mu N}{P} \qquad \textit{(Eq. 4)}$$

where r = Journal radius
c = Radial clearance
μ = Absolute viscosity
N = Shaft speed in rps
P = Unit load (load divided by journal projected area)

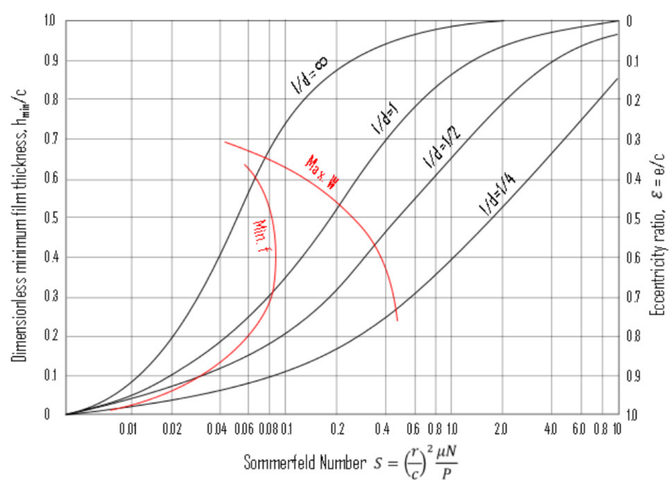

Figure 7. Chart for minimum film thickness variable and eccentricity ration versus Sommerfeld number (Adapted from [5]).

Plotted against the Sommerfeld number are the dependent variables, such as the minimum film thickness (h_{min}), coefficient of friction (f), and volume flow rate of oil (Q). Figure 7 is one such plot, showing minimum film thickness and eccentricity ratio versus S for different length-to-diameter ratios. The area bounded by designs yielding minimum friction coefficient (Min f.) and maximum load (Max. W) can be considered as a desirable space for a design point.

In the preliminary design phase, the designer can use such plots to obtain the performance parameters of a bearing, given a set of design inputs. Note that the process is generally an iterative one because of the very strong dependence of lubricant viscosity on operating temperature. For a detailed design, or for troubleshooting purposes, all aspects of the bearing can be modeled numerically, including hydrodynamic pressure, heat transfer, elastic deformation, and dynamic coefficients.

Journal Bearing Geometries

Journal bearing geometries used in real machines can be much more complex than the simple schematic shown in Figure 2. Even the basic cylindrical bore geometry has multiple variations (Figure 8). There are also a number of more complicated designs, that all have some advantages with regard to stability, cooling, and operation. These are briefly introduced as follows [6]:

Two-axial groove bearing. This classical cylindrical bore design is made of two halves (or pads), with grooves for introducing and distributing lubricant positioned axially, as shown in Figure 8a.

Circumferential groove bearing. A more uniform oil distribution can be achieved with a circumferential oil groove, which essentially splits the bearing into two shorter bearings. This also adds stability due to the resulting higher loading and greater operating eccentricity.

Cylindrical overshot bearing. This design has a wide circumferential groove extending over its upper half (see Figure 8b). This helps reduce viscous dissipation and improve oil flow, allowing for cooler operation.

Pressure dam bearing. Like an overshot bearing but with the groove on the upper half of the bearing abruptly truncated about 45° beyond the vertical direction (see Figure 8c). This design generates positive pressure on the upper half of the bearing, maintaining a higher eccentricity of the journal during low-load operation, thereby contributing to the bearing stability.

Elliptical bearing. Also referred to as *lemon-bore*, this has a vertical offset between the centers of curvature of the upper and lower pads (Figure 9). This difference between the geometric and assembled diameter accentuates the converging wedge, making the bearing more stable. This is referred to as *preloading*. The amount of preload is defined in Equation 5:

$$m = 1 - \frac{c_b}{c_p} \qquad \text{(Eq. 5)}$$

where cb = (Assembled radius) - (Journal radius)
cp = (Pad radius) - (Journal radius)

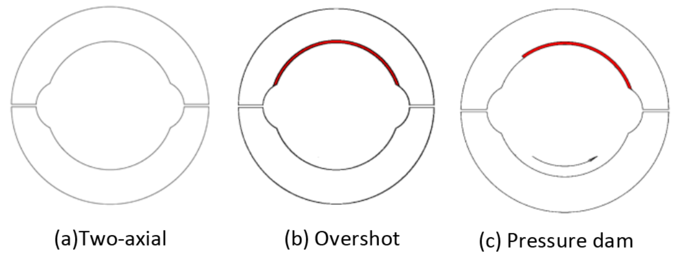

Figure 8. Cylindrical bore bearing designs.

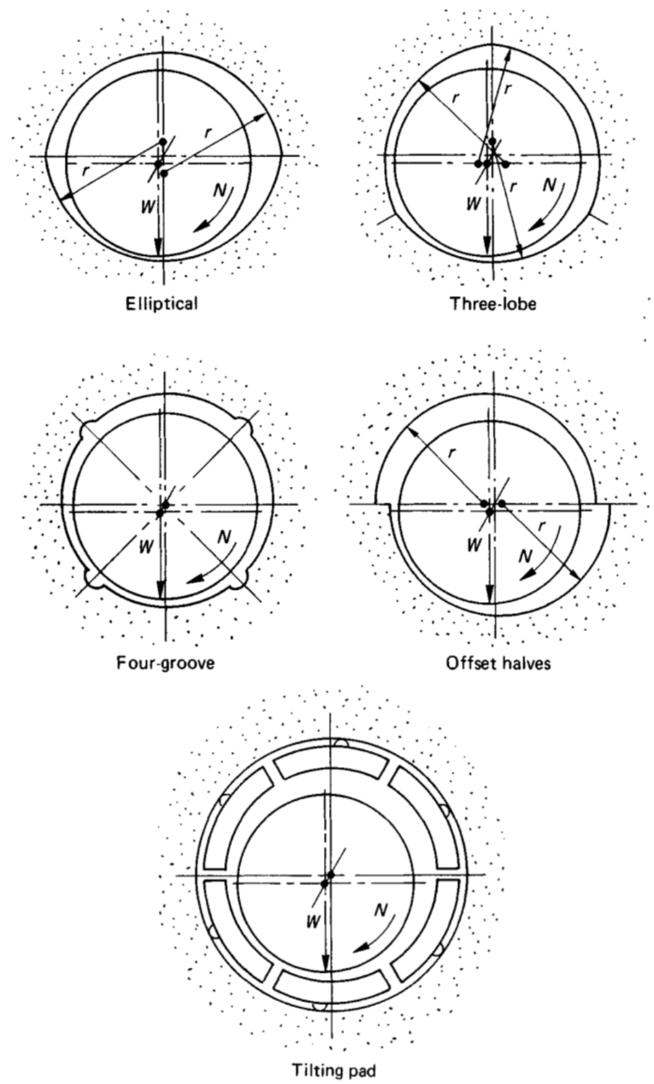

Figure 9. Alternative journal bearing designs [7].

Three-lobed bearing. Similar to the elliptical bearing but with three preloaded pads and three axial oil grooves (see Figure 9).

Multi-groove bearing. More grooves can be added to a cylindrical bore bearing, such as the four-groove design in Figure 9. This increases oil flow for cooler operation, and the interruptions of fluid film give some improvement in stability.

Displaced elliptical bearing. This has both vertical and horizontal offsets in the upper and lower pads (see "Offset halves" in Figure 9). This provides good stiffness in both directions, as well as improved oil flow.

Tilting pad bearing. So far, all of the bearing designs discussed have been of the fixed-geometry type. Tilting pad bearings (see Figure 9) have three or more pivoted pads (or shoes), which passively tilt during operation to form individual wedge-converging wedges. These bearings are very stable and hardly have any cross-coupled stiffness.

Bearing Materials

A lubricated bearing assembly has four main components, each with its own function and material requirements. These are described in the following:

Journal. The journal is part of the rotor shaft and will be constructed of a material according to the function and constrains of the rotor. However, the choice of material may affect the potential damage mechanisms to which the bearing is susceptible (see "wire-wool" or "black scab damage" in the next section).

Housing. The housing is the robust structure that clamps around the bearing and provides structural support.

Bearing shell. The bearing shell (or backing) fits directly into the housing and is the structural part of the bearing itself. It is typically constructed of low carbon steel of copper alloy.

Bearing liner. The bearing liner is a layer of particular liner material that is metallurgically bonded to the bearing shell. The liner material is chosen for its resistance to galling or scuffing, its ability to embed foreign particles and conform to the mating journal surface, and for its corrosion resistance. The most common liner materials are tin and lead-based babbitts, also known as *white metals*. Copper and aluminum alloys are used in applications requiring better fatigue resistance and corrosion resistance [8].

Bearing Damage

The final topic in this report deals with bearing damage that can occur over the course of bearing operation. The main modes of damage are briefly defined below. A small selection of photographs is also included for illustrative purposes. For additional information, refer to the Electric Power Research Institute study, *Manual of Bearing Failures and Repair in Power Plant Rotating Equipment, 2011 Update* (1021780) [9].

Abrasion. Scoring, gouging, and scratching of the bearing liner and journal surfaces caused by particles (such as sand or weld spatter) of a size on the order of the minimum film thickness (Figure 10).

Bond failure. Separation of the bearing liner from the bearing shell (due to poor original adhesion of bearing alloy), buildup of intermetallic layer due to tin migration, or mechanical flexing of the bearing shell.

Corrosion. Chemical attack on metal surfaces by electrolytes and organic acids. Corrosion may produce either removal of bearing liner material or buildup of a deposit on the bearing surface.

Electrical pitting. Intermittent arcing of electrical current across the oil film. The source of the voltage buildup that drives this current may be chronic to the generator or incidental, such as an electrically charged lubricant. Mild voltages as low as 1 V are sufficient to cause small pits and give the bearing and journal a frosted appearance (Figure 11).

Cavitation erosion. Bearing damage that results from the impact of a high-velocity fluid carrying suspended foreign particles, or the large implosive forces associated with the collapse of cavitation bubbles.

Fatigue. Cracking and fracture of metals resulting from an excessive number of cycling stresses. Babbitt has a notoriously poor endurance limit.

Fretting. Fretting is a form of wear damage occurring between contacting surfaces that experience small amplitude relative motion (such as the result of a poor interference fit).

Wire-wool damage (black scab damage). This damage is unique to journals with moderate chromium content (3%–20%). It is initiated by abrasive contaminants but self-propagates as debris from the journal and embeds into the bearing liner as a sort of "wire-wool." The journal becomes severely scored, while damage to the bearing liner appears as a hard "black scab" (Figure 12).

Hydrogen blisters. Blistering of the babbitt due to migration of residual hydrogen within the steel shell toward the bond line of the babbitt.

Overheating. High temperatures can cause plastic flow of the babbitt, cracking due to thermal gradients, and varnish buildup from lubricant damage.

Seizure. Loss of clearance between the shaft and the bearing can be caused by insufficient lubricant or uneven thermal expansion of the journal and bearing during fast startups.

Surface wear. Gradual removal of the bearing liner surface due to insufficient film thickness (such as during machine startup), causing changes over time in the dimensions of the bearing bore.

Tin oxide damage. Alteration of the babbitt surface by the formation of tin oxide. The damage arises from the presence of the hard tin oxide on the babbitt surface.

Wiping. Bearing liner material is displaced or removed by direct contact with the journal (Figure 13).

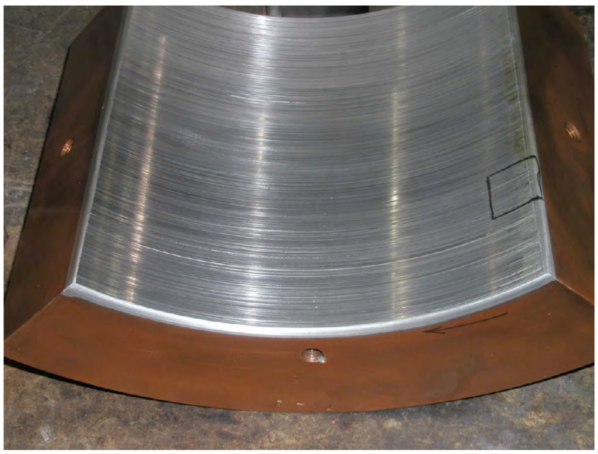

Figure 10. Heavily scored lower tilt pad bearing [9].

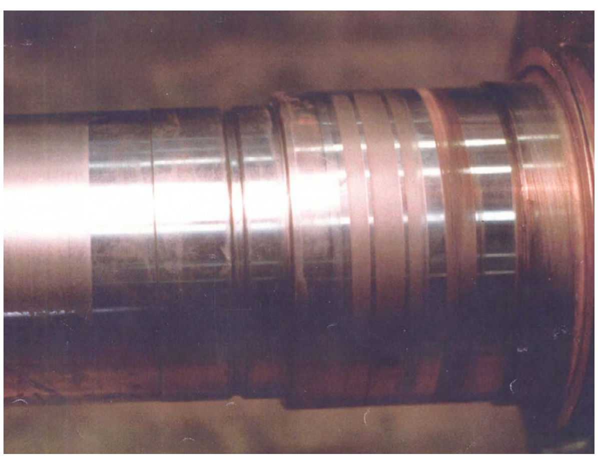

Figure 11. Frosting from electrical pits on journal and seal areas (Used with permission from Sohre Turbomachinery) [9].

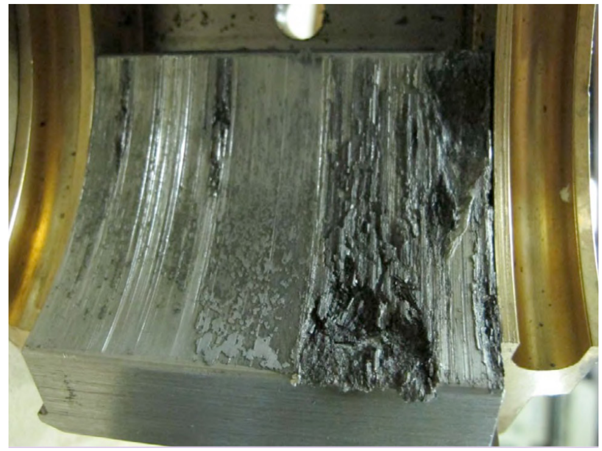

Figure 12. Black scab damage to loaded pad of tilting-pad bearing [9].

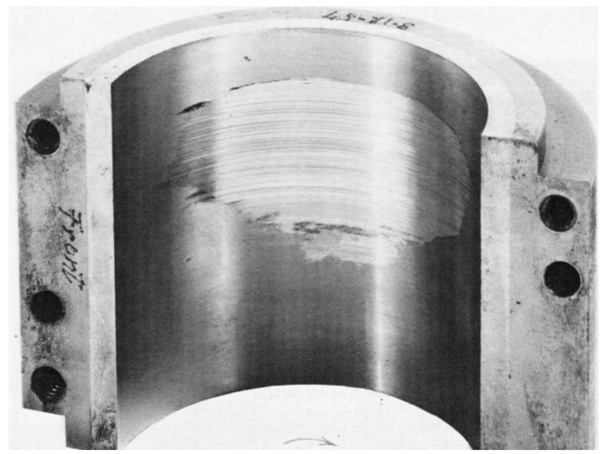

Figure 13. Light wipe on loaded half of bearing due to a temporary loss of lubricant supply (Used with permission from McGraw-Hill) [9].

Summary

Fluid film journal bearings are critical components in rotating equipment for power generation. This report introduced the reader to major aspects of journal bearing principles, design, and operation. Although each topic was touched upon only briefly, the reader should walk away with the following key points:

- The requirements for hydrodynamic lubrication are (1) a thin film of viscous lubricant, (2) a converging wedge geometry, and (3) relative motion between shaft and bearing.
- There are three regimes of fluid film bearing operation: stable full film lubrications, unstable mixed lubrication, and boundary lubrication.
- Journal bearings have direct stiffness and damping that affect the machine's vibration characteristics, as well as cross-coupled stiffness that can cause instability. (Tilting-pad bearings have very little cross-coupled stiffness, which is one of the reasons they are superior to fixed geometry designs.)
- The design of bearings is based on the Reynolds equation. Preliminary design can be based on performance charts cast in terms of the Sommerfeld number.
- More complicated geometries, including cylindrical pressure dam, multi-lobe preloaded, and tilting pad designs, can provide superior performance in terms of cooling and dynamic characteristics.
- Bearing liner material (typically, babbitt) is chosen for its resistance to galling, its embedability and conformability, and for its corrosion resistance.
- There are more than a dozen modes of bearing damage, most of which can be avoided through proper operation, design, and maintenance.

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