

PRINCIPLES OF TURBOMACHINERY BEARINGS

by

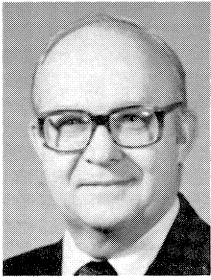
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ABSTRACT

Self-acting fluid film bearings are the predominant choice for bearings used to position the rotors of high power turbomachinery. The present paper outlines the reasons for their selection, discusses the basic phenomenon by which such bearings carry load, describes some typical bearings and points out some common causes of bearing problems.

INTRODUCTION

It is difficult to overemphasize the importance of lubrication in the operation of all kinds of machinery. On a grand scale, a recent ASME report [1] suggests that our nation's energy consumption can be cut 11% by implementing a concerted research and development effort in the field of tribology. The potential yearly dollar savings was estimated at \$16 billion. Tribology, as you may already know, is the relatively new name for the science encompassing lubrication, friction, wear, lubricants and bearing materials, etc. On a personal scale, in 1970 a veteran turbomachinery designer was honored by Product Engineering magazine with the title Master Designer and a substantial cash award [2]. He remarked that sometimes his forty-year turbomachinery career seemed to be one sealing or bearing problem after another.

The presence of lubrication problems and the attempts to control them reach back through hundreds of sequential career lifetimes. The ancient Egyptians were apparently aware of the need for lubricants. A greasy material, probably tallow, has been found on a chariot removed from an Egyptian tomb [3].

The art of lubrication has existed for thousands of years, but the science of lubrication had its beginnings less than a hundred

years ago. The last two decades in particular have witnessed a tremendous growth in lubrication technology, or tribology, if you prefer. This has made possible the development of new machinery operating at new extremes of speed and temperature and with strange lubricants — gases, liquid metal, and liquid oxygen, for example. The growth of tribology has also influenced the rational design of more conventional turbomachinery, such as steam turbines and centrifugal compressors. The effects of bearings upon rotor critical speeds, unbalance response, and stable operation are now better understood. More and better information and computer tools are now available for predicting performance and avoiding rotor dynamics problems in the design stages [4].

It is clearly impossible to cover the general field of lubrication or even the current state of the art of turbomachinery bearings in one paper. What will be attempted is to provide some perspective and a broad over-view of fluid-film bearing technology; what the term means; why such bearings are so widely used; their scientific basis; how they work; what kinds of bearing designs are commonly used; and what can cause bearings to fail.

BEARINGS — DEFINITION AND FUNCTIONS

In a broad sense, a bearing is any element or assembly which guides or positions components subject to relative motion. Wrist pin bearings in an automotive engine are one example where the relative motion is limited to an oscillation. For turbomachinery bearings, of course, continuous rotation is required; bearings are then often classified as radial (journal) or axial (thrust), as shown in Figure 1. As implied, the name derives from the positioning requirement.

The fundamental requirement is that of maintaining the desired relative positions, despite the forces or loads that must be transmitted. It is therefore always important to minimize the wear which may be created by these surfaces under relative

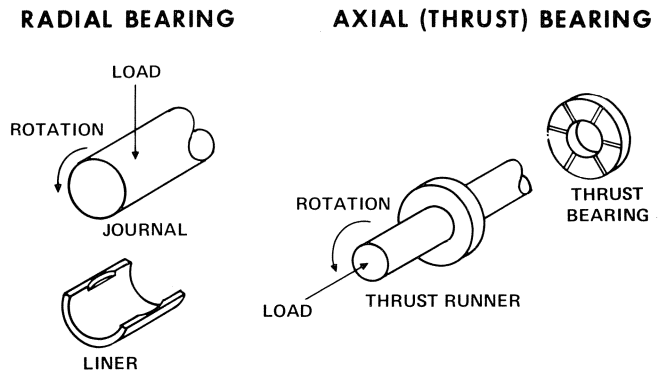


Figure 1. Continuous Rotation Bearings.

motion and load, and desirable to reduce the parasitic energy losses which invariably accompany bearings.

In most turbomachinery, aerodynamic losses are caused by increasing tip clearances; control of these clearances by the bearing is therefore an important function.

Bearings may also be required to provide damping in a rotating machine. In some applications, the damping provided by a bearing is the main reason certain operating speeds can be achieved. Lowered damping would permit destructive amplitudes of vibration at critical speeds.

By their behavior as complex springs and dampers in a mechanical system, bearings may act to amplify or attenuate the forces transmitted through them [5]. They also can greatly alter the response of a rotor itself to inherent disturbances.

There are several sources of bearing load which may be present in a machine. These are generally the dead weight of the rotor, unbalance or inertia forces, those caused by misalignment, unbalanced pressure forces, and those caused by axial or radial changes in the momentum of the fluid passing through the machine. In some situations, load may be transmitted to a turbomachinery driver from its driven equipment by coupling friction; it is possible to transmit both radial and thrust loads even through a gear-type flexible coupling. Estimates of the magnitude and sources of such loads are a necessary first step in designing the bearings. Often these estimates need continued refinement as a new machine design progresses. Axial thrust loads, for example, are often the resultant of two large and opposing forces. Small changes in the individual forces may create a large net force and overload the thrust bearing.

BASIC APPROACHES TO BEARINGS

Two fundamentally different approaches to bearing design have evolved. Among tribologists, it is common to refer to them as rolling element (*not* anti-friction) bearings or fluid film bearings.

Rolling Element Bearings

Rolling element (or R.E.) bearings essentially rely on surfaces-of-revolution and metal-to-metal contact to provide the necessary positioning and load transmission capability. The surfaces of revolution may be balls, rollers, needles, etc. Bearings of this type are found in a wide variety of machinery — automobile wheel bearings, electric motors, centrifugal pumps, fans, etc. They are generally applied in machinery with low power ratings or in high power applications where their advantages are judged to outweigh their drawbacks, such as aircraft gas turbines.

The fundamental advantages of R.E. bearings are these:

1. They are highly developed and standardized [6].
2. They may be designed to provide simultaneous axial and radial load capacities.
3. Compactness — low length/diameter ratios are possible.
4. The load capacity is relatively independent of speed.
5. Low power loss and required lubricant flow rates.
6. The shaft position is relatively fixed and independent of speed.

The fundamental disadvantages of R.E. bearings are:

1. They are inherently fatigue limited even under steady load.

2. The life of a bearing is a statistical phenomenon.
3. They are subject to wear as a result of metal-to-metal contact and sliding.
4. They can be sensitive to thermal gradients, wear debris, and dirt.
5. They have a low damping capacity.

Figure 2 illustrates the reason for the principal handicap of R.E. bearings. Under a *steady* load, motion of the assembly acts to alternately load and unload a specific point on the rolling ball. High alternating sub-surface shear stresses are produced by the concentrated contact area. These will ultimately result in failure by fatigue. R.E. bearings are generally rated by an L-10 (or B-10) life. Given a specific load and operating speed on a large group of identical bearings, the number of operating hours can be estimated at which 10% will have failed *by fatigue*. There is no way to choose a specific bearing among those tested that will be one of those to survive.

It is now fairly well accepted that most R.E. bearings fail earlier by reasons other than fatigue [7,8]. Thus, only a small proportion of R.E. bearings will actually reach the limit predicted by fatigue theory. Despite a great deal of research and improvements in rolling element bearings, this fundamental situation prevails.

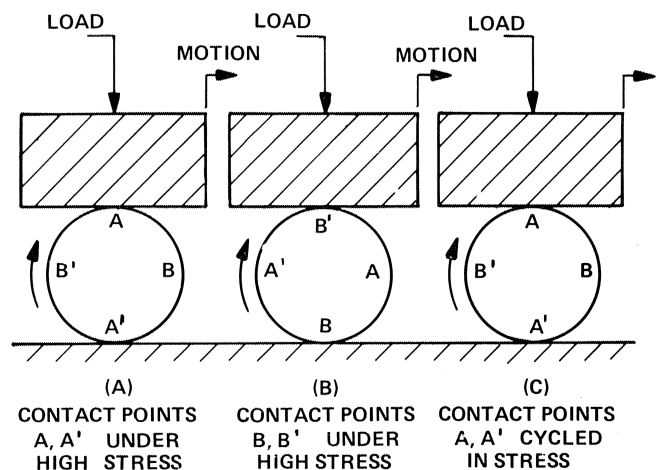


Figure 2. Rolling Element Bearing Element Under Steady Load.

Fluid-Film Bearings

The second basic approach to bearing design is termed fluid-film bearings. The objective, as the name implies, is to transmit all or part of the force through a film of fluid separating the two bearing surfaces. With full fluid-film bearings (Figure 3) a lubricating film is created which is greater than the asperities of the bearing surfaces. With boundary lubricated bearings (Figure 4), only part of the load is carried by the film; the remainder is carried by mechanical contact of the asperities.

In small machinery, e.g., low horsepower electric motors, both rolling element and fluid-film bearings are frequently found. In large turbomachinery such as steam and industrial gas turbines, centrifugal compressors, etc., the fluid-film bearing is the predominant choice of designers. In such machinery, the choice of fluid-film bearings over rolling element bearings is based on these factors:

1. Metal-to-metal contact and wear can be eliminated at design operating speeds.
2. Under steady loads, life is not fatigue limited.
3. Stiffness and damping may be significantly greater.
4. There is reduced sensitivity to dirt and thermal gradients in the shaft and the bearing housing.
5. The need for higher lubricant flow rates is not significant.
6. The higher power loss of fluid-film bearings is not of critical importance, compared to the overall machine rating.

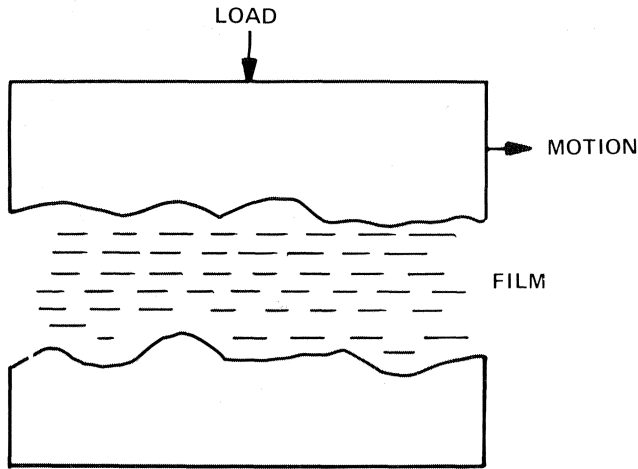


Figure 3. Full Fluid-Film Lubrication.

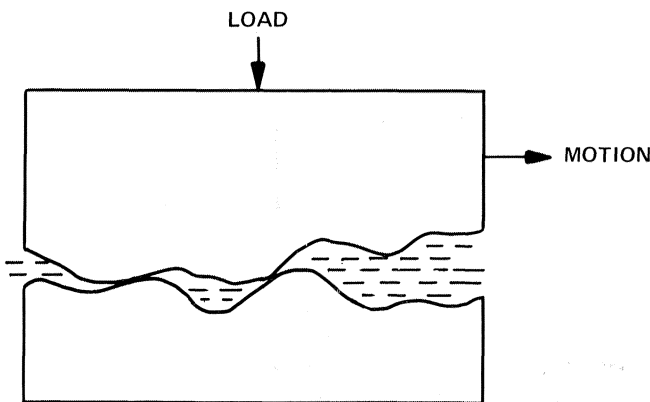


Figure 4. Boundary Lubrication.

With regards to wear and life of fluid-film bearings, it is not uncommon for fluid-film bearings to operate for twenty-five years without difficulty. The first Kingsbury thrust bearing was installed in 1912 and inspected in 1950. It required no maintenance and was reassembled for continued operation [9]. This bearing is reported to be still operating today.

A further subdivision of fluid-film bearings exists, depending upon how the first of the previous factors is achieved. Self-acting (hydrodynamic) bearings create the required film separation without the need for a high pressure supply of lubricant; externally pressurized (hydrostatic) types require such a

pressure source, typically operating at a factor two or three times larger than the average bearing unit load.

The conditions which favor the choice of a self-acting bearing over an externally pressurized type are:

1. Light start-up loads, or loads which increase with speed;
2. Moderate or high operating speeds;
3. High viscosity lubricant available at low supply pressure; and
4. Infrequent starts, stops, slow speed operation.

The converse conditions favor externally-pressurized bearings.

Most turbomachinery uses self-acting, fluid-film bearings. Some examples may be found, however, where externally pressurized bearings are employed. They tend to be those where the process fluid (water) is used as the lubricant. In some instances, externally pressurized bearings features are included in a self-acting bearing design. The objectives are usually to reduce wear at start-up or slow-speed running conditions, or to reduce start-up friction torque.

HOW FLUID-FILM BEARINGS WORK

Discovery of Fluid-Film Bearing Phenomenon

The scientific basis for lubrication began with the experiments of Beauchamp Tower in 1883 on railway car bearings and the subsequent analysis [10] by Osborne Reynolds to explain them. Tower's experiments demonstrated that a pressure was created in the lubricant film of a plain journal bearing. The pressure measured was dependent on the position of the radial pressure tap in the bearing liner; it was highest in the mid-plane, and dropped off on either side. Figure 5 is a reproduction of the illustration from one of Tower's reports, showing the measured pressure distribution. The peak pressure measured was about twice the average unit load (load divided by cross section area); however, the average fluid pressure was found equal to the average unit load. Thus, the load was being carried by a film of fluid, rather than by mechanical contact.

Reynolds' analysis helped explain the phenomenon of pressure generation. The equations which he developed have been the starting point for most fluid-film bearing analysts.

Derivations of Reynolds' equations and the assumptions involved may be found in a number of lubrication texts, e.g., [11, 12, 13]. The equation is nothing more than the formal statement of force equilibrium and mass continuity on an elemental flow volume.

Reynolds Equation

Consider an element of fluid with dimensions dx , dy , and dz which is between two surfaces such as pictured in Figure 6. The two surfaces have relative velocities, U_1 and U_2 in the x direction; the lower surface is also approaching the upper one in the y direction at a velocity V . No velocity components exist in the z direction.

For a fluid of constant density, the dynamic Reynolds equation becomes:

$$\frac{\partial}{\partial x} \left[\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right] = 6(U_1 - U_2) \frac{\partial h}{\partial x} + 6h \frac{\partial}{\partial x} (U_1 + U_2) + 12V \quad (1)$$

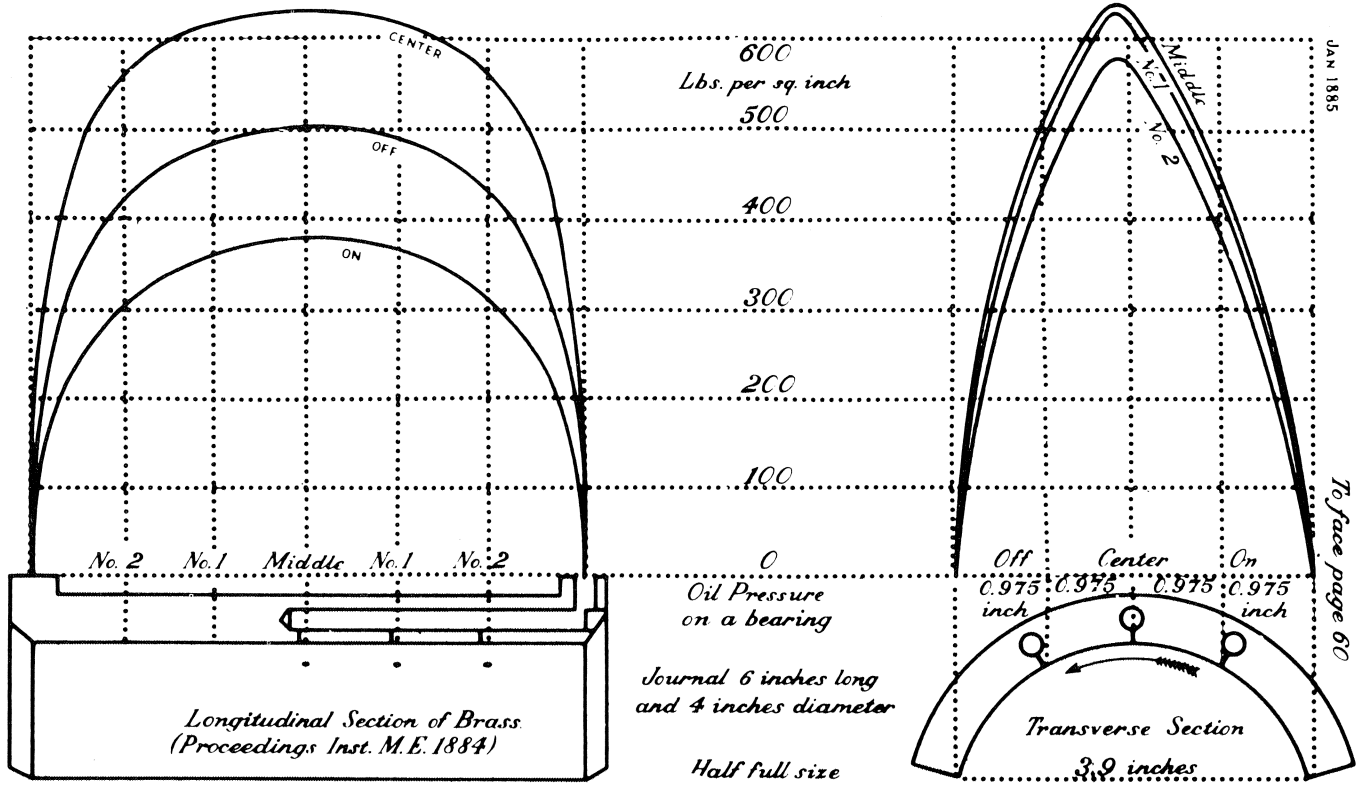


Figure 5. Experimental Results of Beauchamp Tower.

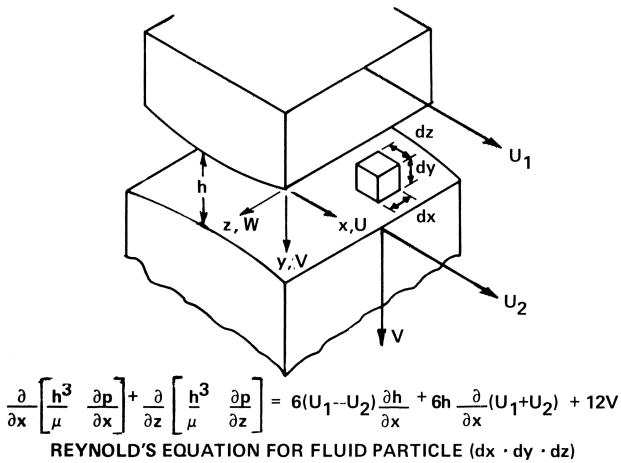


Figure 6. Fluid Element Between Two Moving Surfaces.

where μ is the fluid viscosity

p is the pressure at any position x, z in the flow field

h is the separation between the two surfaces at any position x, z and at a given instant of time.

The above equation (1) does not have a general closed-form solution except for two limiting cases of cylindrical journal bearings:

1. The infinitely wide (Sommerfeld) bearing when the pressure gradient

$$\frac{\partial p}{\partial z} = 0$$

2. The infinitely short (Ocvirk) bearing, when the term

$$\frac{\partial}{\partial x} \left[\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right] = 0$$

The Ocvirk journal bearing solution [14] has been demonstrated to represent a less radical simplification of the basic Reynolds equation than the Sommerfeld bearing. The pressure generated is still a function of the two variable x and z for the Ocvirk approximation. Good agreement has been obtained between experiments and predictions at journal bearing length/diameter ratios of one and less [15].

Computer solutions of Reynolds equations are now widely used for bearings of finite length [16]. Finite difference or finite element techniques are able to remove the need for simplifying assumptions, such as constant viscosity. Nevertheless, rotor dynamics analysts continue to use the Ocvirk approximation because of the advantages inherent in a closed-form solution.

PRESSURE DEVELOPMENT IN FLUID-FILM BEARINGS

Returning to the right-hand side of equation (1), we can examine each of the terms independently to assess their impact on the pressure generating mechanism. Figure 7 illustrates that the three right-hand terms be described in terms of (1) a wedge effect, (2) a stretch effect, and (3) a squeeze effect.

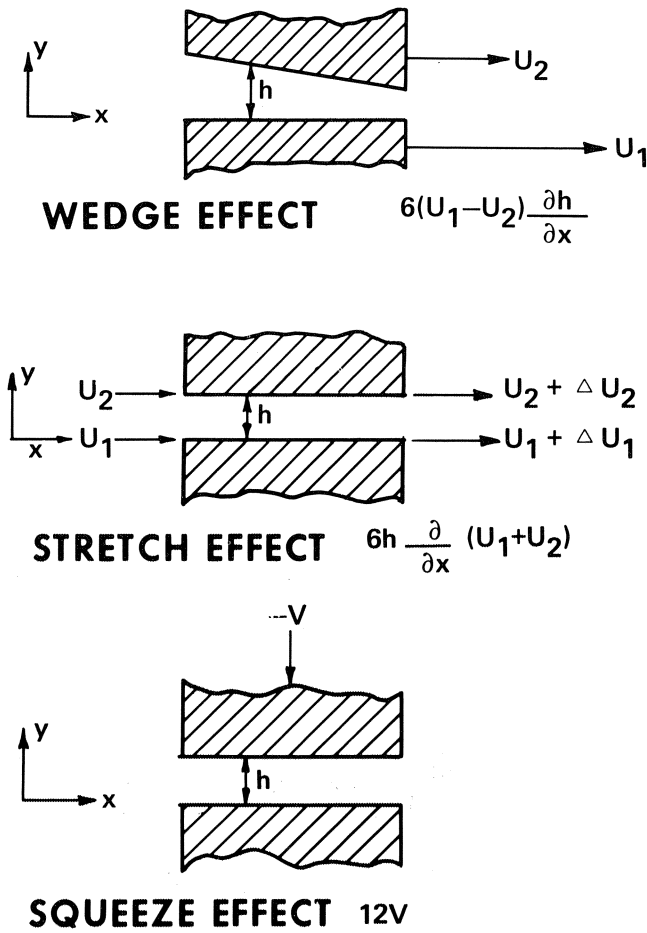


Figure 7. Pressure Development in Fluid-Film Bearings.

For the sake of simplicity in examining these three effects, we will adopt the Sommerfeld approximation. Pressure variation in the z direction (where there is no relative motion between the surfaces) is zero for any given x value. The left-hand side of Reynolds equation then is simply:

$$\frac{d}{dx} \left[\frac{h^3}{\mu} \frac{dp}{dx} \right]$$

Wedge Effect

In this case, we will consider sliding surfaces with no normal motion V and assume that the stretch effect is zero. The simplified Reynolds equation then becomes simply:

$$\frac{d}{dx} \left[\frac{h^3}{\mu} \frac{dp}{dx} \right] = 6(U_1 - U_2) \frac{dh}{dx} \tag{2}$$

The right-hand side of equation (2) will be zero under either of two conditions (a) if $U_1 = U_2$ and (b) if $dh/dx = 0$.

Condition (a) states that no relative motion exists and condition (b) implies that the surfaces are parallel. For either of these two conditions, a solution of equation (2) predicts no pressure rise will occur over the inlet ambient pressure. Assuming that the lubricant is not pressurized at the inlet, no load capacity will exist for a slider bearing under those conditions.

If the velocity U_2 is zero, and the film thickness h decreases in the direction of motion, a pressure build-up will

occur. This need not be a simple taper. A wide variety of converging surfaces can create a pressure rise [17]. Some of these are pictured in Figure 8.

For a simple taper on the bearing surfaces, however, the slope dh/dx is fixed. Assuming constant viscosity, an infinitely wide bearing, and velocity $U_2 = 0$, equation (2), can be integrated directly for the pressure. Figure 9 illustrates the geometric parameters of the slider.

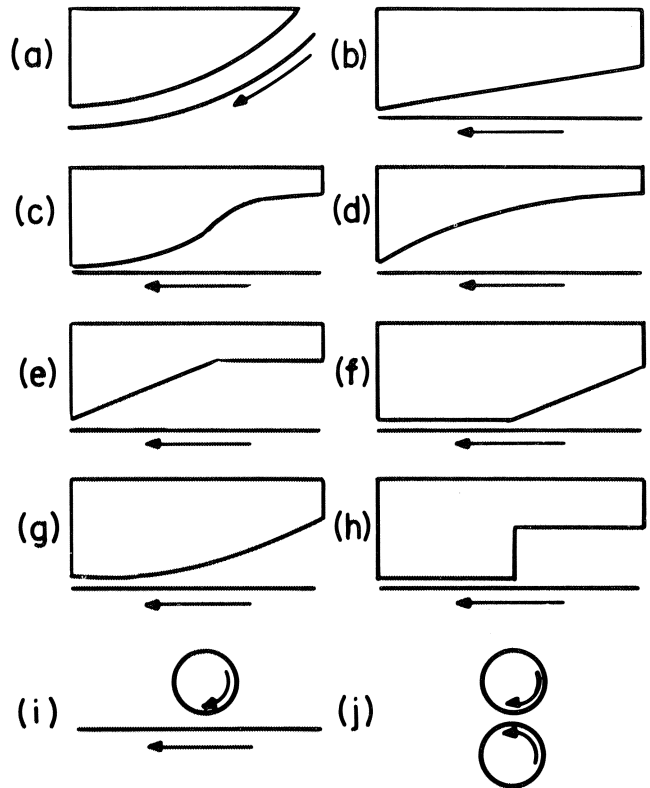


Figure 8. Sliding Surfaces Producing Wedge Effect [17].

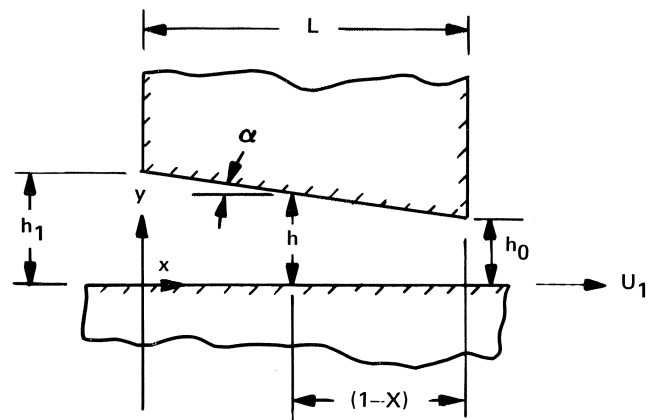


Figure 9. Geometric Parameters of a Simple Tapered Slider.

The pressure p is a function of the sliding velocity U, taper α , fluid viscosity μ , minimum clearance h_0 , slider length L and

distance along slider x . As shown in reference [17], the pressure rise can be written as:

$$p = \frac{6\mu UL}{h_0^2} K_p \quad (3)$$

where K_p is a function depending only upon the position along the slider, x/L , and the film thickness ratio,

$$m_1 = \frac{h_1 - h_0}{L}$$

The non-dimensional pressure rise

$$\bar{p} = \frac{p h_0^2}{6\mu UL} = K_p \quad (4)$$

A plot showing the pressure rise for some assumed conditions is given in Figure 10.

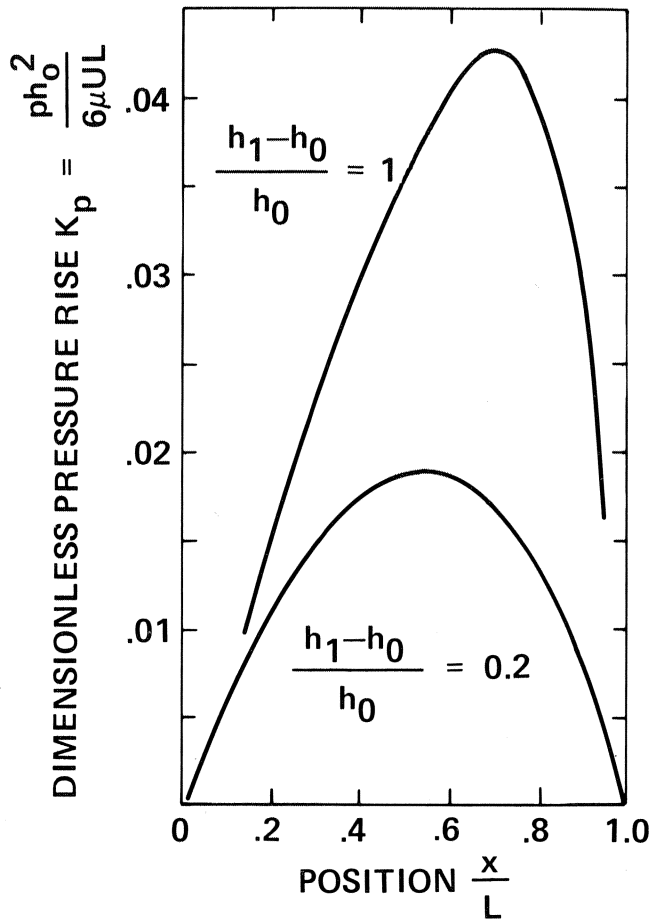


Figure 10. Pressure Generation from Wedge Effect for an Infinitely Wide Slider [17].

A word of caution needs to be expressed in applying the conclusions from the simplified Reynolds equation (2) to journal bearings. Using the coordinate system described, it is not possible to eliminate the velocity V for a journal bearing. Owing to rotation, a given point on the shaft surface will have a

normal velocity component. When this fact is taken into account, it can be shown that a journal bearing will double its load carrying capacity if the velocities U_1 and U_2 are equal. In the slider bearing, the same doubling effect occurs if the velocity U_1 is equal but opposite to U_2 .

Stretch Effect

The second term on the right-hand side of equation (1), i.e.,

$$6h \frac{\partial}{\partial x} (U_1 + U_2),$$

is often termed the stretch effect. It implies that the velocities of the two surfaces are changing in the x direction. At first glance, this phenomenon would appear to require a rubber-like material. A physical interpretation can, however, be applied without requiring such a material. One need only visualize a journal bearing in which the instantaneous center of rotation does not coincide with the journal geometric center. Points on the journal surface are not equidistant from the instant center. Hence, the tangential velocities at different points will vary. As a practical matter, however, these variations in surface velocities can be shown to be of second-order importance. The so-called stretch effect can be safely ignored for either slider or journal bearings.

Squeeze Effect

The third term on the right-hand side of equation (1) represents the velocity V normal to the surfaces. With a fixed coordinate system in the lower surface, a downward motion of the upper surface represents a negative velocity. As discussed previously, for parallel surfaces ($dh/dx = 0$), the wedge effect is absent and the stretch effect can be ignored. Under this set of assumptions, and for a bearing of infinite width, equation (1) becomes:

$$\frac{d}{dx} \left[\frac{h^3}{\mu} \frac{dp}{dx} \right] = -12V \quad (5)$$

This can be integrated directly for the pressure over the bearing length L

$$p = \frac{6\mu VL^2}{h^3} \left(\frac{x}{L} \right) \left[1 - \frac{x}{L} \right] \quad (6)$$

or, non-dimensionalizing the pressure p due to normal velocity V ,

$$\bar{p} = \frac{ph^3}{6\mu VL^2} = \frac{x}{L} \left[1 - \frac{x}{L} \right] \quad (7)$$

A plot of the above expression is given in Figure 11.

The squeeze film phenomenon is extremely important in many applications. For wrist pin bearings in automotive applications, it provides the load carrying capacity. For turbomachinery using damper bearings which also do not rotate, the same phenomenon prevails.

The Short Bearing Approximation

The previous discussion has described how pressures, and therefore load capacity, can be generated in bearing configurations. It can be particularly instructive to illustrate how this has been applied to the very short (Ocvirk) journal bearing with no motion of the shaft center (Figure 12).

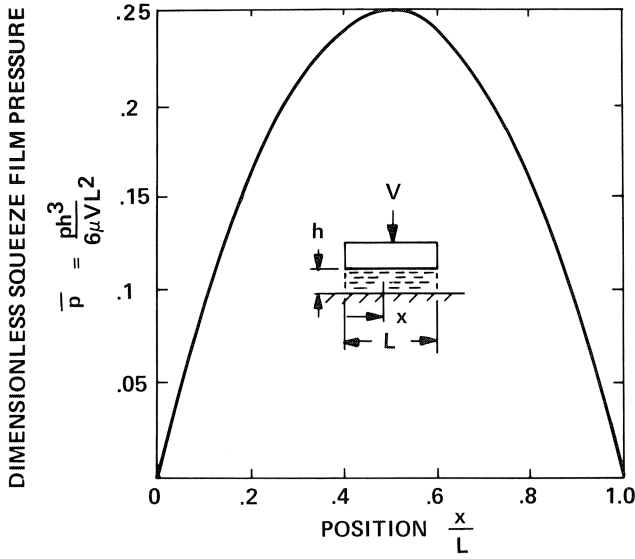


Figure 11. Squeeze Film Pressure Generation for an Infinitely Wide Slider.

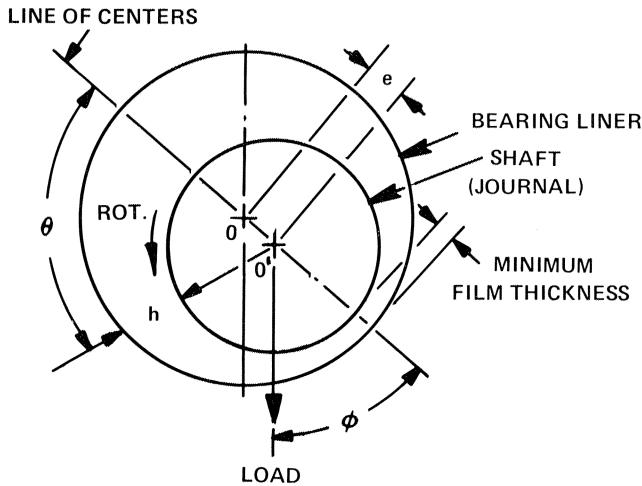


Figure 12. Journal Bearing Schematic.

The Reynolds equation for the above situation is reduced to

$$\frac{d}{dz} \left[\frac{h^3}{\mu} \frac{dp}{dz} \right] = 6\mu U \frac{dh}{dx} \quad (8)$$

If the shaft center is displaced from the bearing center by an amount as shown in Figure 12, and the film thickness is not a function of axial distance z , the circumferential variation of film thickness can be expressed as:

$$h = C_r + \epsilon \cos \theta \quad (9)$$

or

$$h = C_r (1 + \epsilon \cos \theta) \quad (10)$$

where C_r is the radial clearance

ϵ is the eccentricity ratio of the journal with respect to the bearing.

θ is the angular location of the film thickness h from the maximum clearance, measured in the direction of rotation.

Again, since the film thickness variation is a simple expression, the pressure rise can be readily calculated. The pressure p is given by:

$$p = \frac{3\mu UL^2}{R C_r^2} \left[\frac{1}{4} - \left(\frac{z}{L} \right)^2 \right] \frac{\epsilon \sin \theta}{(1 + \epsilon \cos \theta)^3} \quad (11)$$

In the expression above, z is the axial distance measured from the bearing midplane, L is axial length of the bearing, R the journal radius, U the tangential velocity, μ the fluid viscosity, and θ the angular displacement from the position of maximum clearance, measured in the direction of rotation.

It will be seen that the maximum pressure occurs in the bearing midplane ($z=0$), dropping to zero at the bearing ends where $z/L = \pm 1/2$. Figure 13 shows the approximate pressure distribution given by equation (11). Negative pressures are predicted in the diverging portion of the film clearance. These are generally accepted as physically unrealistic and are therefore taken out of consideration in analysis of load capacity.

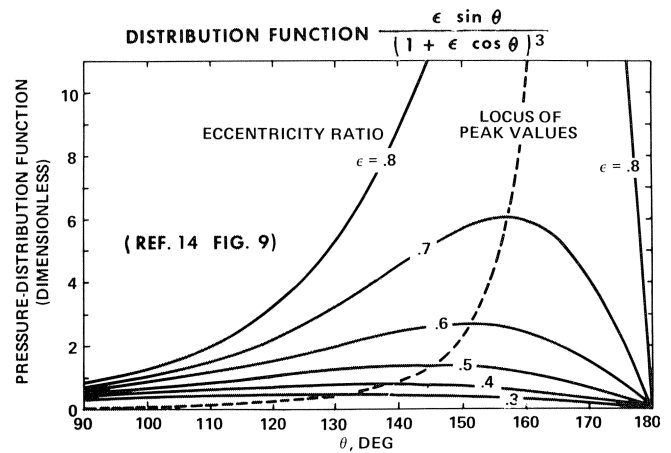


Figure 13. Circumferential Pressure Distribution in Short Journal Bearings.

Pressure distribution measurements have been made on journal bearings confirming that the short bearing analysis is a reasonable approximation. Figure 14 shows a set of pressure measurements [18]; there is a strong similarity to the first measurements made by Tower (Figure 5).

In 1952 Ocvirk was able to integrate equation (11) through the use of Sommerfeld's substitutions. Thus, the relationship between the bearing projected or averaged psi load P and the other variables was expressed as:

$$\frac{\mu N}{P} \left(\frac{D}{C_d} \right)^2 \left(\frac{L}{D} \right)^2 = \frac{(1 - \epsilon^2)^2}{\pi \epsilon} \left[\frac{1}{\pi^2(1 - \epsilon^2) + 16\epsilon^2} \right]^{1/2} \quad (12)$$

In essence, the above expression relates film thickness of a cylindrical bearing to the imposed operating conditions of

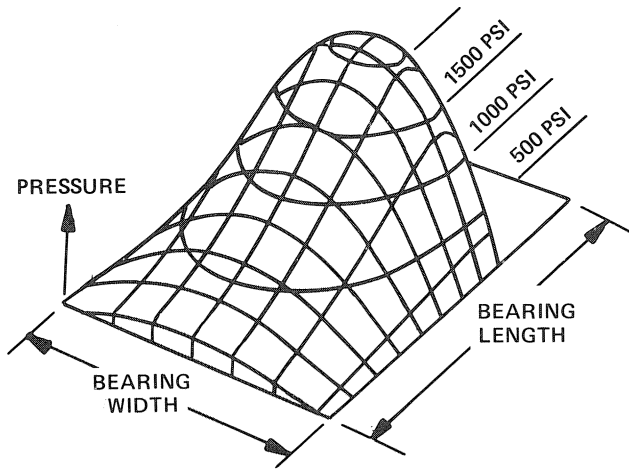


Figure 14. Oil Film Pressure Distribution in an Aligned Bearing [18].

speed N , average pressure P , fluid viscosity μ and the bearing geometric factors of length L , diameter D and radial clearance C_d .

Ocvirk referred to the combination of variables on the left-hand side of equation (12) as the capacity number. In his honor, however, many fluid-film bearing researchers now refer to this dimensionless value as the Ocvirk number. Figure 15 is a plot of the eccentricity ratio versus the Ocvirk number.

The short bearing solution also yields a simple expression for the attitude angle ϕ (Figure 12) of the bearing; i.e., the angular position of the line of centers of the shaft and bearing with respect to the load. This relationship is:

$$\tan \phi = \frac{\pi}{4} \frac{(1 - \epsilon^2)^{1/2}}{\epsilon} \quad (13)$$

A polar plot of attitude angle versus eccentricity ratio is given in Figure 16. The other closed-form solution of Reynolds equation, i.e., the Sommerfeld solution for an infinitely long journal bearing, predicts an attitude angle of 90° for all eccentricity ratios. This is not in agreement with experimental data; in fact, a bearing with such a response would be inherently unstable [19].

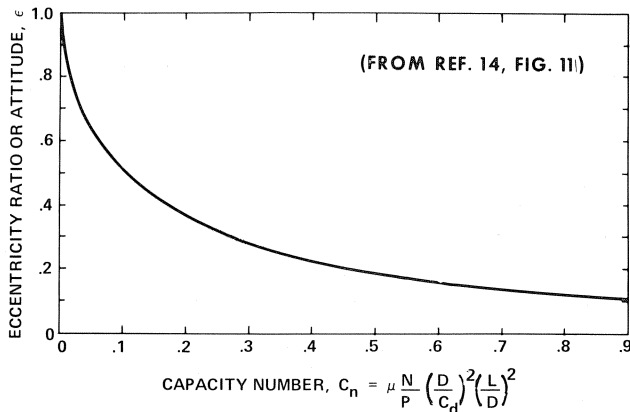


Figure 15. Eccentricity Ratio vs Capacity Number for Short Journal Bearings [14].

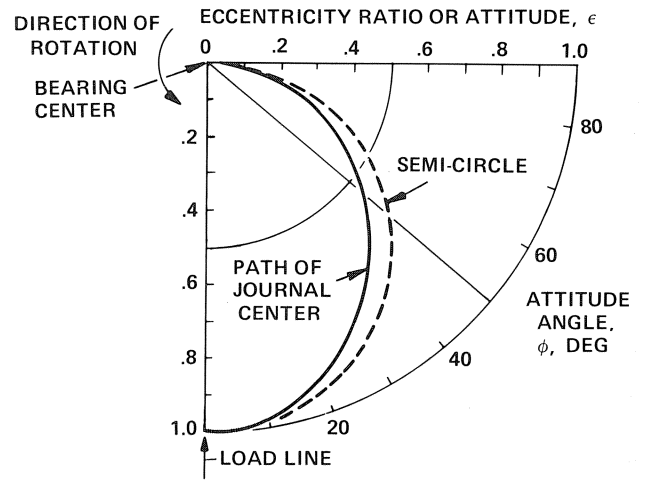


Figure 16. Eccentricity Ratio vs Attitude Angle of Short Bearings [14].

Dynamic Bearing Coefficients

It will be seen from Figure 15 that any change in the average steady-state bearing load P will produce some change in the eccentricity ratio of the bearing. Similarly, from equation (13), any change in the eccentricity ratio will also produce, in general, a new attitude angle for the shaft center. Thus, a change in the vertical load acting on a bearing will generally produce both a vertical and horizontal displacement of the shaft center. By analogy with a mechanical spring, the relationship between the change in force and the change in displacement of the bearing can be described in terms of direct and cross-coupled spring coefficients. The cross-coupling is a result of the fact that a displacement in, say, the vertical direction alone requires a change in both the vertical and horizontal forces.

In the presence of rapid variations of the shaft center position, additional forces will exist on the shaft which are velocity dependent. By analogy with a mechanical system, these forces are referred to as damping coefficients. As with the spring coefficients, cross-coupling effects exist. Schematically, the shaft bearing system may be regarded as shown in Figure 17. Mathematically, the spring and damping coefficients

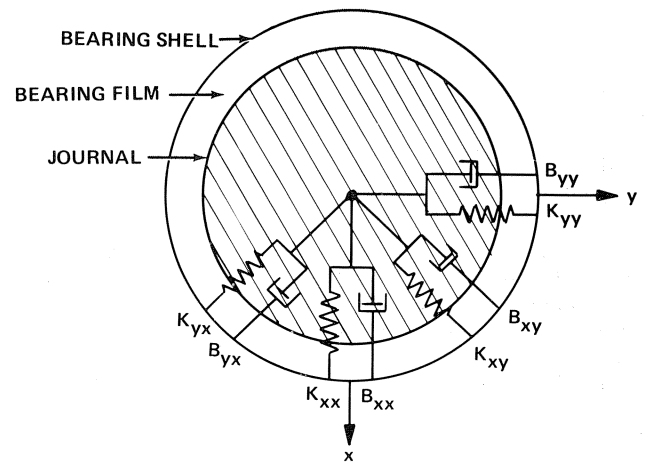


Figure 17. Dynamic Representation of a Bearing by Spring and Damping Coefficients.

coefficients are the first order terms used in linearizing the change in bearing force with displacement and velocity. Details of this derivation may be found, for example, in reference [20].

A total of eight coefficients generally exist for a given bearing design at each specified eccentricity. Since eccentricity is related to the Sommerfeld number, values of spring and damping coefficients are sometimes plotted against the Sommerfeld number. Some recent data for selected bearings is given by Lund [21].

The dynamic bearing coefficients are widely used today in predicting stability and unbalance response of turbomachinery. Figure 18 shows the results of some predictions and measurements of dynamic force transmitted from an unbalanced rotor [22]. Reasonable agreement was demonstrated between the forces predicted by the linearized coefficients and those actually measured, using a standard production turbine bearing.

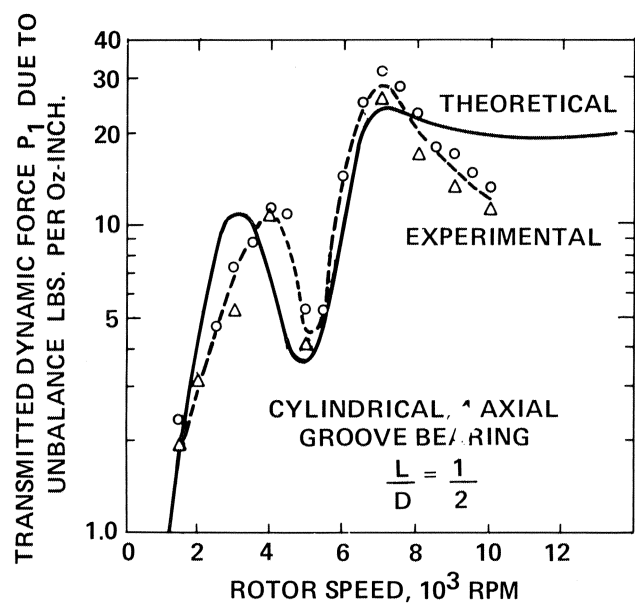


Figure 18. Comparison of Theoretical and Experimental Dynamic Force Transmitted from Unbalanced Rotor Through Fluid-Film Bearing.

SOME TYPICAL FLUID-FILM BEARINGS

Journal Bearings [23]

Understandably, turbomachinery manufacturers are prone to continue applying bearing designs which have proven reliable in their own experience. Journal bearing designs have therefore evolved into a variety of forms. Generally, however, one can place a given design into one of the following categories, pictured in Figure 19:

1. plain cylindrical or axial groove
2. elliptical (or lemon)
3. pressure dam
4. multi-lobe
5. tilting pad

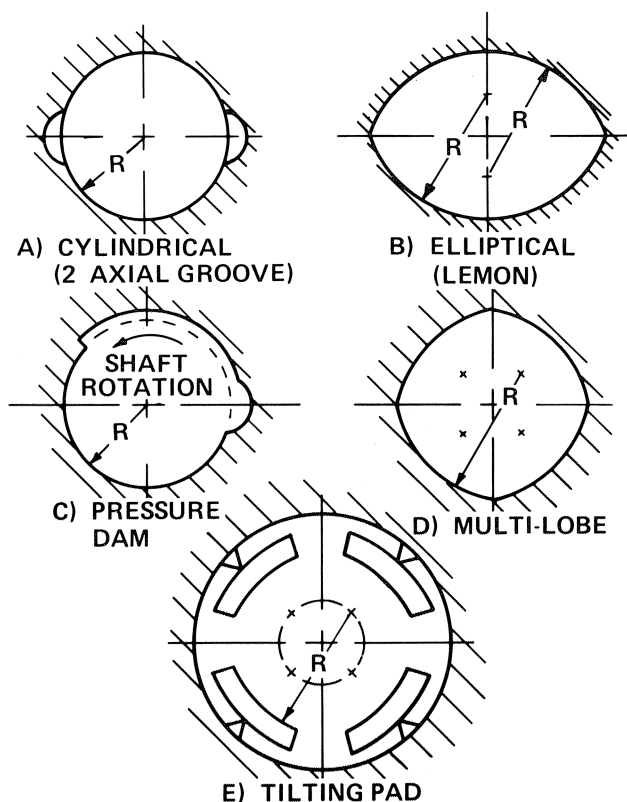


Figure 19. Schematic Diagrams of Typical Journal Bearings.

Some provision must be made for introducing the lubricant into the bearing-shaft interface. The plain cylindrical bearing, which is defined as one having a full circular bore, is often designed with two or more axial slots for this purpose. The slots may be placed at the horizontal split line of the bearing, or occasionally within the lower half. The plain cylindrical bearing without such slots is more likely to be used when the direction of load is unknown. The objective is to avoid placing any grooves within the normal load carrying pressure zone.

The elliptical (or lemon) bearing has a non-circular bore. The upper and lower halves of this bearing are circular segments which do not have a common center. This design is sometimes achieved by machining a circular bore bearing from two halves with a shim in place on the horizontal centerline. Removal of the shim produces the "elliptical" shape. A shaft centered in such a bearing will have different clearances in the vertical and horizontal directions. One objective of such a design is frequently to place a converging lubricant wedge on the top half of the liner to assist in stable operation.

The pressure-dam bearing is also chosen for the same purpose. In this configuration, a groove is placed in the middle portion of the top half, extending over perhaps one-half, or more, the top width. The groove terminates in a step or dam. The objective here is to create a hydrodynamic load on the top half of the journal.

Multi-lobe bearings are more generalized examples of the elliptical bearing. The bearing inner surface is composed of three or more arc segments. The center of these arcs are displaced from one another and from the common bearing center. The objective of such a design is again to produce multiple converging wedges around the shaft and to stabilize it.

The tilting pad bearing is a design which has come into prominence within the last decade, as computer programs have been created for predicting its performance. The bearing consists of a number of circular arc segments, free to pivot in a tangential (and frequently axial) direction. A large number of design variables exist for the tilting pad journal bearing. A discussion of these is given in reference [24]. The tilting pad bearing is frequently selected because of its superior stability characteristics, particularly at light loads or high speeds.

Thrust Bearings

Compared to journal bearings, the variety of basic designs of thrust bearings in common use is much reduced. The most common designs are shown in Figure 20. They are (1) the fixed or tapered land, (2) the pivoted shoe, and (3) the pivoted shoe with a load equalizing linkage. The theory of the tapered wedge has been discussed earlier; it should be emphasized, however, that the required taper is very small — on the order of 0.001 inches per inch of circumferential length. A ring of such pads is placed on the stationary surface adjacent to the thrust collar on the shaft.

The tilting pad bearing concept was apparently conceived about the same time by Kingsbury in the U.S. and Michell in Europe, and development carried on separately. Thus, these thrust bearings tend to be called by different manner in different countries. As used within the United States, the tilting pad bearing is often used with a load-leveling linkage to distribute the thrust load more uniformly among the several pads. It has been common practice to use a 50% pivot location for tilting pad thrust designs, although theory has long predicted the advantages of an offset design. Discussions of thrust bearing designs may be found in references [25], [26], and [27].

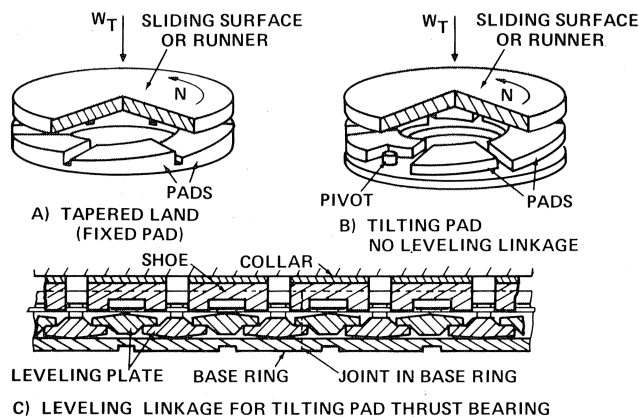


Figure 20. Schematic Diagrams of Typical Thrust Bearings.

WHAT CAN CAUSE FLUID-FILM BEARING FAILURES

In a broad sense, one can classify the reasons why fluid-film bearings fall into three categories. Some of the possible sources of trouble will be described in each category:

A. Design Deficiencies.

1. The minimum film thickness under steady state conditions may be too small, in view of possible misalignment, dirt in lubricant, etc.

2. The ability of the bearing to dissipate the heat generated by shearing losses is inadequate.
3. Insufficient lubricant is being supplied.
4. The rotor-bearing system is unstable.
5. Structural stresses and deflections are excessive.
6. Incompatible bearing-journal materials for withstanding occasional rubs.

B. Manufacturing/Assembly Errors

1. Deviations from design clearance values (too small or too large).
2. Installing bearing liners misaligned.
3. Non-cylindrical journals (tapered, barrel, saddle-shaped).
4. Excessive roughness or scratches in journals or liners.
5. Excessive run-out of thrust collars.
6. Dirt or debris from inadequate cleaning of supply lines.
7. Inadequate bonding of babbitt to steel backing.

C. Operational Mistreatment

1. Failure to provide clean supply of lubricant meeting required specifications.
2. Operation without lubricant, even for short periods.
3. Operation under excessive vibratory loadings.
4. Operation with excessive lube oil inlet temperature.
5. Placing additional loads on bearings through improper alignment of driver-driven equipment.

The above listings are by no means exhaustive, but include some of the more common causes of difficulties. A useful guide to troubleshooting sliding bearing problems is given in reference [1].

SUMMARY

The objective of this paper was to provide a broad overview and perspective on fluid-film bearings. It discussed the relative merits of rolling-element and fluid-film bearings; described the different basic phenomena by which fluid-film bearings create a load carrying capacity; outlined how the characteristics of fluid-film bearings affect the behavior of a rotor; described some typical bearings, and pointed out some of the reasons why these bearings can fail. For those who wish to dig deeper into the topic, a number of useful references have been cited. The impact of tribology on turbomachinery design is still growing. Your time spent in understanding the basic phenomena will be well rewarded, whether your responsibility is in design, operation or maintenance of turbomachinery.

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