9

Practical Applications of Journal Bearings

9.1 INTRODUCTION

A hydrodynamic journal bearing operates effectively when it has a full fluid film without any contact between the asperities of the journal and bearing surfaces. However, under certain operating conditions, this bearing has limitations, and unique designs are used to extend its application beyond these limits.

The first limitation of hydrodynamic bearings is that a certain minimum speed is required to generate a full fluid film of sufficient thickness for complete separation of the sliding surfaces. When the bearing operates below that speed, there is only mixed or boundary lubrication, with direct contact between the asperities. Even if the bearing is well designed and successfully operating at the high-rated speed, it can be subjected to excessive friction and wear at low speed, during starting and stopping of the machine. In particular, hydrodynamic bearings undergo severe wear during start-up, when the journal accelerates from zero speed, because static friction is higher than dynamic friction. In addition, there is a limitation on the application of hydrodynamic bearings in machinery operating at variable speed, because the bearing has high wear rate when the machine operates in the low-speed range.

The second important limitation of hydrodynamic journal bearings is the low stiffness to radial displacement of the journal, particularly under light loads and high speed, when the eccentricity ratio, ε, is low. Low stiffness rules out the application of hydrodynamic bearings for precision applications, such as machine tools and measurement machines. In addition, under dynamic loads, the low stiffness of the hydrodynamic bearings can result in dynamic instability, referred to as *bearing whirl*. It is important to prevent bearing whirl, which often causes bearing failure. It is possible to demonstrate bearing whirl in a variable-speed testing machine for journal bearings. When the speed is increased, it reaches the critical whirl speed, where noise and severe vibrations are generated.

In a rotating system of a rotor supported by two hydrodynamic journal bearings, the stiffness of the shaft combines with that of the hydrodynamic journal bearings (similar to the stiffness of two springs in series). This stiffness and the distributed mass of the rotor determine the *natural frequencies*, also referred to as the *critical speeds* of the rotor system. Whenever the force on the bearing oscillates at a frequency close to one of the critical speeds, bearing instability results (similar to resonance in dynamic systems), which often causes bearing failure. An example of an oscillating force is the centrifugal force due to imbalance in the rotor and shaft unit.

9.2 HYDRODYNAMIC BEARING WHIRL

In addition to resonance near the critical speeds of the rotor system, there is a failure of the oil film in hydrodynamic journal bearings under certain dynamic conditions. The stiffness of long hydrodynamic bearings is not similar to that of a spring support. The bearing reaction force increases with the radial displacement, o–o_1, of the journal center (or eccentricity, e). However, the reaction force is not in the same direction as the displacement. There is a component of cross-stiffness, namely, a reaction-force component in a direction perpendicular to that of the displacement. In fact, the bearing force based on the Sommerfeld solution is only in the normal direction to the radial displacement of the journal center.

The cross-stiffness of hydrodynamic bearings causes the effect of the *half-frequency whirl*; namely, the journal bearing loses its load capacity when the external load oscillates at a frequency equal to about half of the journal rotation speed. It is possible to demonstrate this effect by computer simulation of the trajectory of the journal center of a long bearing under external oscillating force. If the frequency of the dynamic force is half of that of the journal speed, the eccentricity increases very fast, until there is contact of the bearing and journal surfaces. In practice, hydrodynamic bearing whirl is induced at relatively high speed under light, steady loads superimposed on oscillating loads. In actual machinery, oscillating loads at various frequencies are always present, due to imbalance in the various rotating parts of the machine.

Several designs have been used to eliminate the undesired half-frequency whirl. Since the bearing whirl takes place under light loads, it is possible to prevent it by introducing internal preload in the bearing. This is done by using a

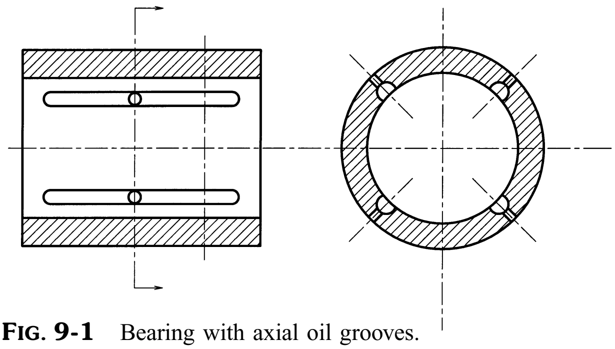

FIG. 9-1 Bearing with axial oil grooves.

bearing made of several segments; each segment is a partial hydrodynamic bearing. In this way, each segment has hydrodynamic force, in the direction of the bearing center, that is larger than the external load. The partial bearings can be rigid or made of tilting pads. Elliptical bearings are used that consist of only two opposing partial pads. However, for most applications, at least three partial pads are desirable. An additional advantage is improved oil circulation, which reduces the bearing operating temperature.

Some resistance to oil whirl is obtained by introducing several oil grooves, in the axial direction of the internal cylindrical bore of the bearing, as shown in Fig. 9-1. The oil grooves are along the bearing length, but they are not completely open at the two ends, as indicated in the drawing. It is important that the oil grooves not be placed at the region of minimum film thickness, where it would disturb the pressure wave. Better resistance to oil whirl is achieved by designs that are described in the following sections.

9.3 ELLIPTICAL BEARINGS

The geometry of the basic elliptical bearing is shown in Fig. 9-2a. The bore is made of two arcs of larger radius than for a circular bearing. It forms two pads with opposing forces. In order to simplify the manufacturing process, the bearing bore is machined after two shims are placed at a split between two halves of a round sleeve. After round machining, the two shims are removed. In fact, the shape is not precisely elliptical, but the bearing has larger clearances on the two horizontal sides and smaller clearance in the upper and bottom sides. In this way, the bearing operates as a two-pad bearing, with action and reaction forces in opposite directions.

The additional design shown in Fig. 9-2b is made by shifting the upper half of the bearing, relative to the lower half, in the horizontal direction. In this way,

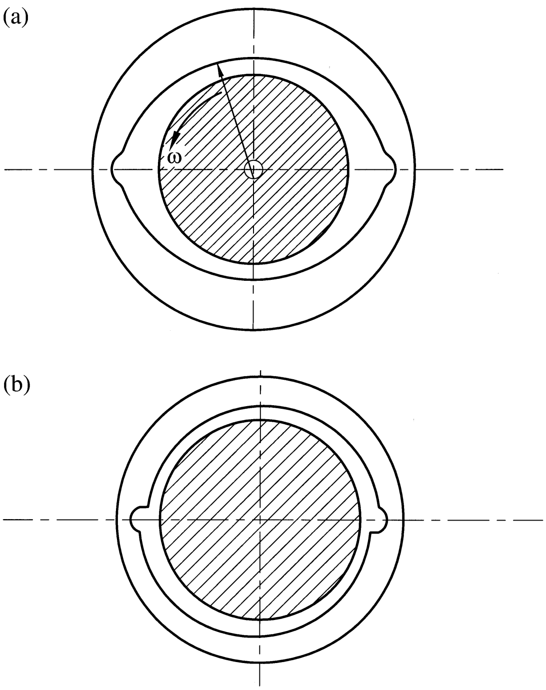

FIG. 9-2 Elliptical bearing: (a) basic bearing design; (b) shifted bearing.

each half has a converging fluid film of hydrodynamic action and reaction forces in opposite directions.

Elliptical and shifted bearings offer improved resistance to oil whirl at a reasonable cost. They are widely used in high-speed turbines and generators and other applications where the external force is in the vertical direction. The circulation of oil is higher in comparison to a full circular bearing with equivalent minimum clearance.

9.4 THREE-LOBE BEARINGS

Various designs have been developed to prevent the undesired effect of bearing whirl. An example of a successful design is the *three-lobe journal bearing* shown in Fig. 9-3. It has three curved segments that are referred to as *lobes*. During operation, the geometry of the three lobes introduces preload inside the bearing. This design improves the stability because it increases the bearing stiffness and reduces the magnitude of the cross-stiffness components. The preferred design

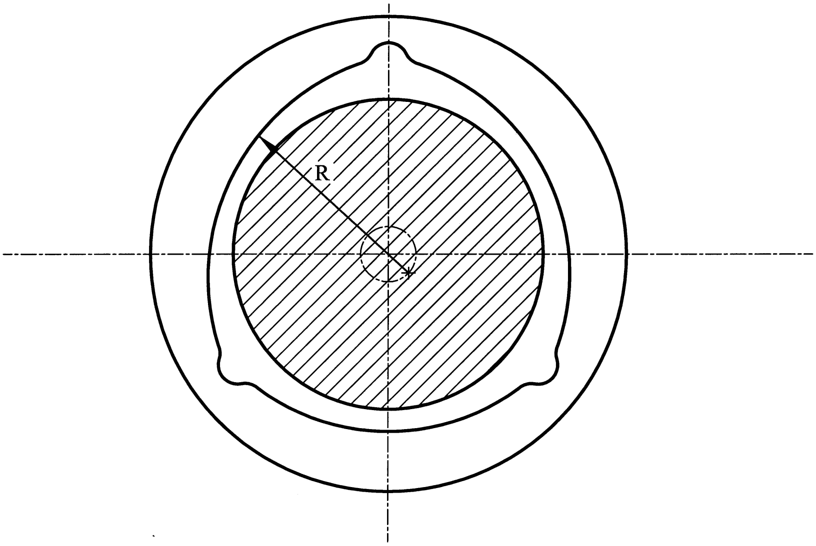

FIG. 9-3 Three-lobe bearing.

for optimum stability is achieved if the center of curvature of each lobe lies on the journal center trajectory. This trajectory is the small circle generated by the journal center when the journal is rolling in contact with the bearing surface around the bearing. According to this design, the journal center is below the center of each of the three lobes, and the load capacity of each lobe is directed to the bearing center.

The calculation of the load capacity of each lobe is based on a simplifying assumption that the journal is running centrally in the bearing. This assumption is justified because this type of bearing is commonly used at low loads and high speeds, where the shaft eccentricity is very small.

An additional advantage of the three-lobe bearing is that it has oil grooves between the lobes. The oil circulation is obviously better than for a regular journal bearing (360°). This bearing can carry higher loads when the journal center is over an oil groove rather than over the center of a lobe.

9.5 PIVOTED-PAD JOURNAL BEARING

Figure 9-4 shows a *pivoted-pad bearing*, also referred to as *tilt-pad bearing*, where a number of tilting pads are placed around the circumference of the journal. The best design is a universal self-aligning pad; namely, each pad is free to align in both the tangential and axial directions. These two degrees of freedom

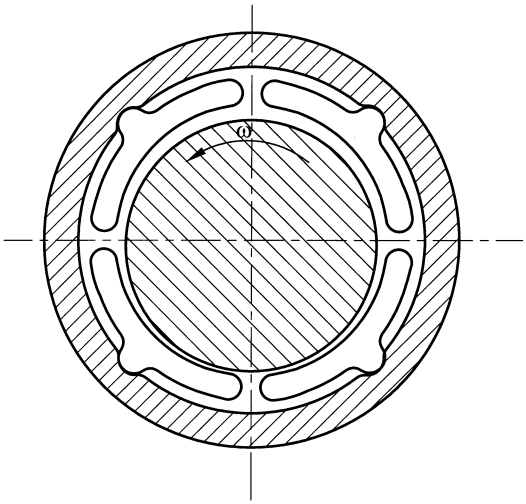

FIG. 9-4 Pivoted-pad journal bearing.

allow a full adjustment for any misalignment between the journal and bearing. This design has clear advantages in comparison to a rigid bearing, and it is used in critical applications where continuous operation of the machine, without failure, is essential. This is particularly important in large bearings that have large tolerances due to manufacturing errors, such as a propeller shaft of a ship. In many cases, a pivoted-pad journal bearing is used for this application. During a storm, the ship experiences a large elastic deformation, resulting in considerable misalignment between the bearings that are attached to the body of the ship and the propeller shaft. Self-aligning of pivoted-pad journal bearings can prevent excessive wear due to such misalignment.

Self-aligning pivoted-pad journal bearings are widely used in high-speed machines that have a relatively low radial load, resulting in low eccentricity. For example, the pivoted-pad journal bearing is used in high-speed centrifugal compressors. This design offers stability of operation and resistance to oil whirl by increasing the bearing stiffness.

The pivoted-pad journal bearing has the advantages of high radial stiffness of the bearing and low cross-stiffness. These advantages are important in applications where it is necessary to resist bearing whirl or where high precision is required. Better precision results from higher stiffness that results in lower radial run-out (eccentricity) of the journal center. The reaction force of each pad is in the radial direction. The forces of all the pads preload the journal at the bearing center and tend to increase the stiffness and minimize the eccentricity.

9.6 BEARINGS MADE OF COMPLIANT MATERIALS

Pivoted pad bearings are relatively expensive. For many applications, where only a small alignment is required, low cost bearings that are made of elastic materials such as an elastomer (rubber) can align the contact surface to the journal. Of course, the alignment is much less in comparison to that of the tilting pad. Rubber-to-metal bonding techniques have been developed with reference to compliant surface bearings; see a report by Rightmire (1967). Water-lubricated rubber bearings can be used in boats, see Orndorff and Tiedman (1977).

Bearings made of plastic materials are also compliant, although to a lesser degree than elastomer materials. Plastics have higher elasticity than metals, since their modulus of elasticity, E, is much lower.

In rolling-element bearings or gears there is a theoretical point or line contact resulting in very high maximum contact pressure. When the gears or rolling elements are made of soft compliant materials, such as plastics, the maximum pressure is reduced because there is a larger contact area due to more elastic deformation. Even steel has a certain elastic deformation (compliance) that plays an important role in the performance of elastohydrodynamic lubrication in gears and rollers.

Similar effects take place in journal bearings. The journal has a smaller diameter than the bearing bore, and for a rigid surface under load there is a theoretical line contact resulting in a peak contact pressure that is much higher than the average pressure. Engineers realized that in a similar way to gears and rollers, it is possible to reduce the high peak pressure in rigid bearings by using compliant bearing materials. Although the initial application of hydrodynamic bearings involved only rigid materials, the later introduction of a wide range of plastic materials has motivated engineers to test them as alternative materials that would result in a more uniform pressure distribution. In fact, plastic materials demonstrated successful performance in light-duty applications under low load and speed (relatively low PV). The explanations for the improved performance are the self-lubricating properties and compliant surfaces of plastic materials. In fact, biological joints, such as the human hip joint, have soft compliant surfaces that are lubricated by synovial fluid. The superior performance of the biological bearings suggested that bearings in machinery could be designed with compliant surfaces with considerable advantages.

Plastic materials have a low dry-friction coefficient against steel. In addition, experiments indicated that bearings made of rubber or plastic materials have a low friction coefficient at the boundary or mixed lubrication region. This is explained by the surface compliance near the minimum film thickness, where the high pressure forms a depression in the elastic material. In the presence of lubricant, the depression is a puddle of lubricant under pressure.

Another important advantage of compliant materials is that they have a certain degree of elastic self-aligning. The elastic deformation compensates for misaligning or other manufacturing errors of the bearing or sleeve. In contrast, metal bearings are very sensitive to any deviation from a perfect roundness of the bearing and journal. For hydrodynamic metal bearings, high precision as well as perfect surface finish is essential for successful performance with minimum contact between the surface asperities. In comparison, plastic bearings can be manufactured with lower precision due to their compliance characteristic. The advantage of surface compliance is that it relaxes the requirement for high precision, which involves high cost.

Moreover, compliant surfaces usually have better wear resistance. Elastic deformation prevents removal of material due to rubbing of rough and hard surfaces. Compliant materials allow the rough asperities to pass through without tearing. In addition, it has better wear resistance in the presence of abrasive particles in the lubricant, such as dust, sand, and metal wear derbies. Rubber sleeves are often used with slurry lubricant in pumps. Embedding of the abrasive particles in the sleeve is possible by means of elastic deformation. Later, elastic deformation allows the abrasive particles to roll out and leave the bearing.

For all these advantages, bearings made of plastic material are widely used. However, their application is limited to light loads and moderate speeds. For heavy-duty applications, metal bearings are mostly used, because they have better heat conduction. Plastic bearings would fail very fast at elevated temperatures.

9.7 FOIL BEARINGS

The foil bearing has the ultimate bending compliance, and its principle is shown in Fig. 9-5. The foil is thin and lacks any resistance to bending. The flexible foil stretches around the journal. In the presence of lubricant, a thin fluid film is formed, which separates the foil from the journal. At high speeds, air can perform as a lubricant, and a thin air film prevents direct contact between the rotating shaft and the foil. An air film foil bearing has considerable advantages at high speeds. Air film can operate at much higher temperatures in comparison to oils, and, of course, air lubrication is much simpler and less expensive than oil lubrication.

Lubricant flow within the foil bearing has a converging region, which generates bearing pressure, and a parallel region, of constant clearance, h_0, supports the load. Foil bearings have important applications wherever there is a requirement for surface compliance at elevated temperature. Mineral oils or synthetic oils deteriorate very fast at high temperature; therefore, several designs have already been developed for foil bearings that operate as air bearings.

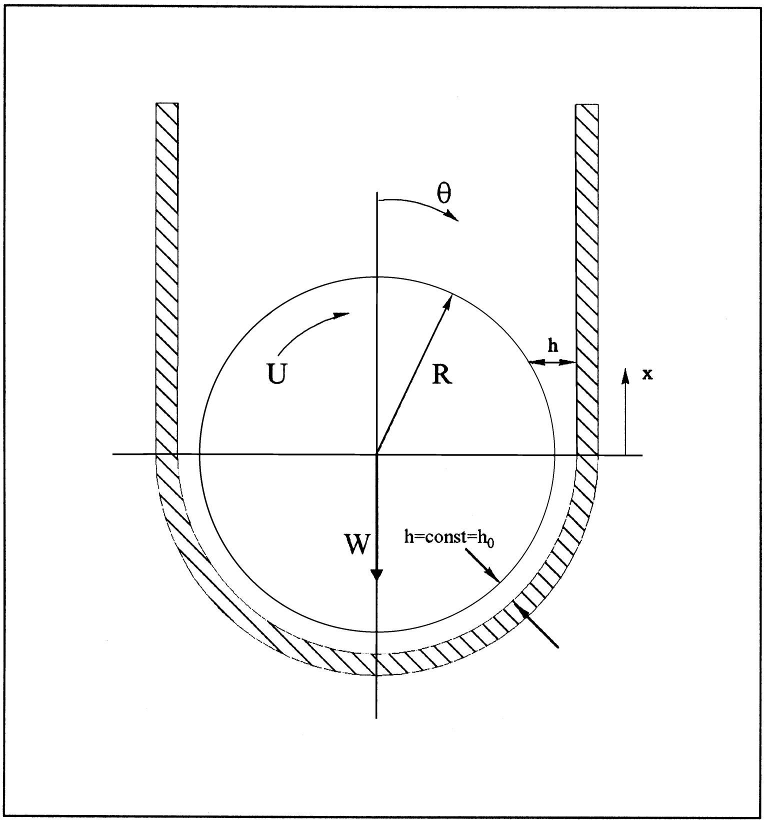

FIG. 9-5 Foil bearing.

9.8 ANALYSIS OF A FOIL BEARING

The following analysis is presented to illustrate the concept of the hydrodynamic foil bearing. The foil is stretched around a journal of radius R rotating at constant speed, and resulting in a tangential velocity U at the journal surface. A vertical external load F is applied to the journal that is equal to the load capacity W of the fluid film.

The following assumptions are made for solving the load capacity W of the foil bearing.

1. The foil sleeve is parallel to the journal surface (constant clearance) along the lower half of the journal, due to its high flexibility. In turn, there is no localized pressure concentration.

2. The contributions to load-carrying capacity of the converging and diverging ends are neglected.
3. The simplified equation for an infinitely long bearing can be applied.

This problem is similar to that of the sled of Example problem 4-4. In the sled problem, there is also a converging clearance between a flat plate and a cylinder. Let us recall that the film thickness in the converging region between a flat plate and a cylinder is given by the following equation [see equation (4-24)]:

$$h(\theta) = h_0 + R(1 - \cos\theta) \tag{9-1}$$

Here, θ is measured from the line $x = 0$. The expression for the film thickness h is approximated by (see Sec. 4.10.1)

$$h(x) = h_0 + \frac{x^2}{2R} \tag{9-2}$$

This approximation is valid within the relevant range of the converging region. The boundary conditions of lubricant pressure is $p = 0$ at $x = \infty$. The Reynolds equation for an infinitely long bearing is

$$\frac{dp}{dx} = 6\mu U \frac{h_0 - h}{h^3} \tag{9-3}$$

In the parallel region, $h = h_0$. So it follows from Eq. (9-3) that $dp/dx = 0$ (parallel region). This means that the pressure is constant within the parallel region. The pressure acting over the entire lower parallel region is constant and solely responsible for carrying the journal load.

The force dW on an elementary area $dA = LR\,d\theta$ is

$$dW = p_0 LR\, d\theta \tag{9-4}$$

This force is in the direction normal to the journal surface. The vertical component of this elementary force is

$$dW = LRp_0 \sin\theta\, d\theta \tag{9-5}$$

Integration of the vertical elementary force component over the parallel region would result in the load capacity W, according to the equation

$$W = LR \int_{\pi/2}^{3\pi/2} p_0 \sin\theta\, d\theta \tag{9-6}$$

Here, L is the bearing length (in the axial direction, z) and p_0 is the constant pressure in the parallel region.

Upon integration, this equation reduces to

$$W = 2LRp_0 \tag{9-7}$$

Similar to the case of the sled, the fluid flow in the converging region generates the pressure, which is ultimately responsible for supporting the journal load. The viscous fluid is dragged into the foil–journal convergence, due to the journal rotation, generating the pressure supplied to the parallel region.

In a similar way to the parallel sled problem in Chapter 4, by substituting the value of h into Eq. (9-3) and integrating, the equation of pressure distribution becomes

$$p(x) = 24\mu UR^2 \int_{\infty}^{x} \frac{x^2}{(2Rh_0 + x^2)^3}\, dx \tag{9-8}$$

The following example is a numerical solution of the pressure wave at the inlet to the foil bearing. This solution is in dimensionless form. The advantage of a dimensionless solution is that it is universal, in the sense that it can be applied to any design parameters.

Example Problem 9-1

Find the expression of the film thickness, h_0, as a function of the load capacity, W.

Solution

For conversion of Eq. (9-8) to a dimensionless form, the following dimensionless terms are introduced:

$$\bar{x} = \frac{x}{\sqrt{2Rh_0}} \qquad \bar{h} = \frac{h}{h_0} \qquad \bar{h} = 1 + \bar{x}^2$$

Similar to the sled problem in Chapter 4, the expression for the dimensionless pressure becomes

$$\bar{p} = \frac{h_0^2}{\sqrt{2Rh_0}} \frac{1}{6\mu U} \int_0^p dp = \int_{\infty}^{x} \frac{\bar{x}^2}{(1 + \bar{x}^2)^3}\, d\bar{x}$$

Here, the dimensionless pressure is defined as

$$\bar{p} = \frac{h_0^2}{\sqrt{2Rh_0}} \frac{1}{6\mu U} p$$

The dimensionless pressure can be integrated analytically or numerically. The pressure becomes significant only near $x = 0$. Therefore, for numerical integration, the infinity is replaced by a finite number, such as 4. Numerical integration results in a dimensionless pressure distribution, as shown in Fig. 4-9.

A comparison of numerical and analytical solutions, in Chapter 4, resulted in the following same solution for the constant pressure in the parallel clearance:

$$\bar{p} = \int_0^{p_0} dp = \int_\infty^0 \frac{\bar{x}^2}{(1+\bar{x}^2)^3} d\bar{x}; \qquad \bar{p}_0 = 0.196$$

Minimum Film Clearance: Neglecting the pressure in the entrance and outlet regions, the equation for the load capacity as a function of the uniform pressure, p_0, in the clearance, h_0, is given in Eq. (9-7)

$$W = p_0 DL$$

Using the value of dimensionless pressure, $\bar{p}_0 = 0.196$, and converting to pressure p_0, the following equation is obtained for the clearance, h_0, as a function of the pressure, p_0:

$$\frac{h_0}{R} = 4.78\left(\frac{\mu n}{p_0}\right)^{2/3}$$

Here, n is the journal speed, in RPS, and the pressure, p_0, is determined by the load:

$$p_0 = \frac{W}{LD}$$

9.9 FOIL BEARINGS IN HIGH-SPEED TURBINES

It has been realized that a compliant air bearing can offer considerable advantages in high-speed gas turbines that operate at very high temperatures. Unlike compliant bearings made of polymers or elastomer materials, foil air bearings made of flexible metal foils can operate at relatively high temperature because metals that resist high temperature can be selected. In addition, the viscosity of air increases with temperature and it does not deteriorate at elevated temperature like oils. In addition, a foil bearing does not require precision manufacturing with close tolerances, as does a conventional rigid journal bearing.

Hydrodynamic bearings have a risk of catastrophic failure by seizure, because, in the case of overheating due to direct contact, the journal has larger thermal expansion than the sleeve. This problem is eliminated in compliant foil because the clearance can adjust itself. An additional advantage is that the foil bearing has very good resistance to whirl instability.

The foil air bearing concept has already been applied in high-speed gas turbines, with a few variations. Two types of compliant air bearings made of

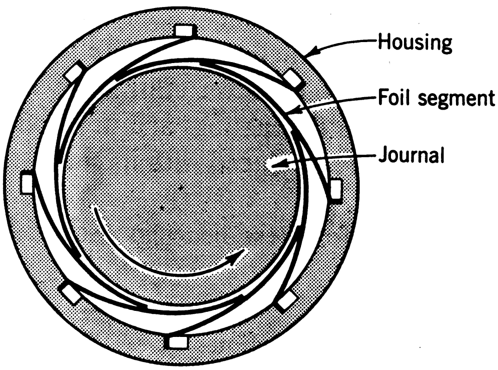

FIG. 9-6 Compliant journal bearing A. (From Suriano et al., 1983.)

flexible metal have been extensively investigated and tested. One problem is the high friction and wear during the start-up or occasional contact. The thin flexible metal strips and journal undergo severe wear during the start-up. Several coatings, such as titanium nitride, have been tested in an attempt to reduce the wear. Experimental investigation of the dynamic characteristics of a turborotor simulator of gas-lubricated foil bearings is described by Licht (1972).

The two designs are shown in Figs. 9-6 and 9-7. The first design [see Suriano et al. (1983)] is shown in Fig. 9-6, and a second design [see Heshmat et

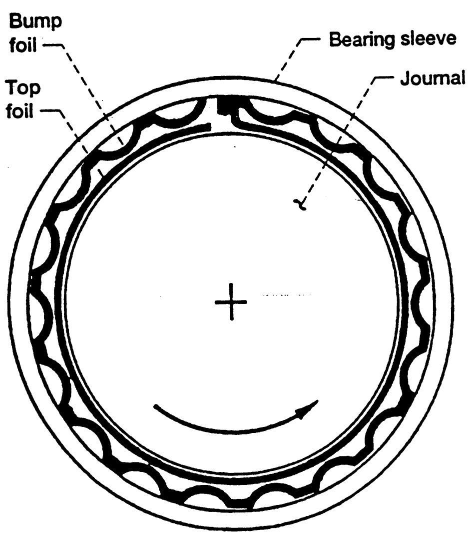

FIG. 9-7 Compliant journal bearing B. (From Heshmat et al., 1982.)

al. (1982)] is shown in Fig. 9-7. The tests indicated that foil bearings have adequate load capacity for gas turbines. In addition, foil bearings demonstrated very good whirl stability at very high speeds. Foil bearings have already been tested successfully in various applications. Additional information about research and development of foil bearings for turbomachines is included in a report by the Air Force Aero Propulsion Laboratory (1977).

Extensive research and development are still conducted in order to improve the performance of the compliant air bearing for potential use in critical applications such as aircraft turbines.

9.10 DESIGN EXAMPLE OF A COMPLIANT BEARING

In Sec. 9.5, the advantages of the pivoted-pad bearing were discussed. In addition to self-aligning, it offers high stiffness, which is essential for high-speed operation. However, in many cases, compliance can replace the tilting action. A compliant-pad bearing has been suggested by the KMC Company [see Earles et al. (1989)]. A compliant-pad journal bearing is shown in Fig. 9-8. The most important advantage is that it is made of one piece, in comparison to the pivoted pad, which consists of many parts (Fig. 9-9). In turn, the compliant pad is easier

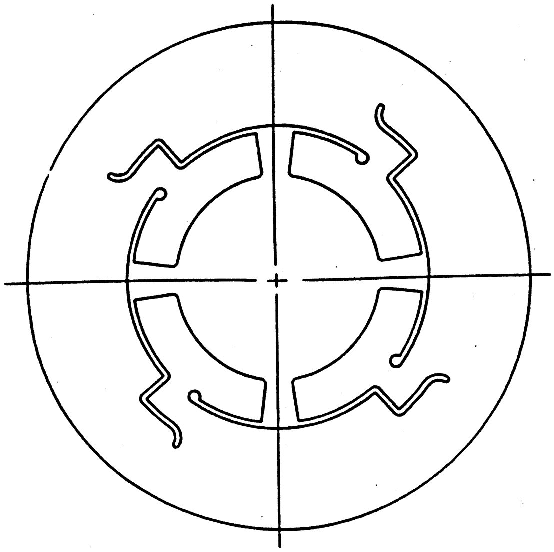

FIG. 9-8 Compliant-pad journal bearing. (With permission from KMC Co.)

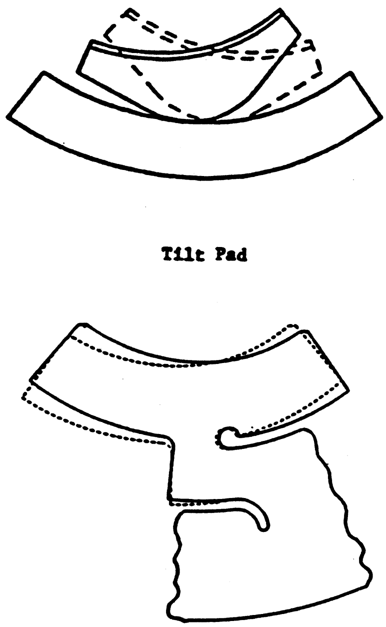

FIG. 9-9 Comparison of a compliant and a pivoted pad. (With permission from KMC Co.)

to manufacture and assemble. The bearing is made from one solid bronze cylinder by an electrical discharge machining process. In addition, tests indicated that the flexural compliance of the pads improves the bearing characteristics at high speed, such as in centrifugal compressors.

Problems

9-1 A foil bearing operates as shown in Fig. 9-5. Find the uniform film thickness, h_0, around the journal of a foil bearing. The bearing has the following design parameters:

$R = 50$ mm $\qquad \mu = 0.015$ N-s/m^2

$L = 100$ mm $\qquad W = 10{,}000$ N

$N = 5000$ RPM

9-2 Find the uniform film thickness around the journal of a foil bearing, as shown in Fig. 9-5, that is floating on a hydrodynamic air film. The temperature of the air is 20°C. The bearing has the following design parameters:

$$D = 120 \text{ mm} \qquad \mu_{air} = 184 \times 10^{-7} \text{ N-s/m}^2$$
$$L = 60 \text{ mm} \qquad N = 25{,}000 \text{ RPM}$$

a. Find the load capacity, W, for a fluid film thickness around the bearing of 50 μm.
b. Use the infinitely-short-bearing equation and compare the load capacity with that of a short bearing having the given design parameters, air viscosity, and minimum film thickness. The radial clearance is $C/R = 0.001$ ($R = 60$ mm).