

# 7

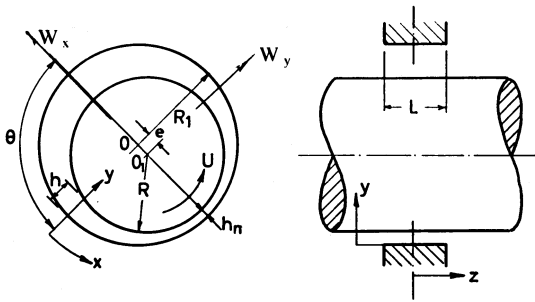
---

## Short Journal Bearings

### 7.1 INTRODUCTION

The term *short journal bearing* refers to a bearing of a short length,  $L$ , in comparison to the diameter,  $D$ , ( $L \ll D$ ). The bearing geometry and coordinates are shown in Fig. 7-1. Short bearings are widely used and perform successfully in various machines, particularly in automotive engines. Although the load capacity, per unit length, of a short bearing is lower than that of a long bearing, it has the following important advantages.

1. In comparison to a long bearing, a short bearing exhibits improved heat transfer, due to faster oil circulation through the bearing clearance. The flow rate of lubricant in the axial direction through the bearing clearance of a short bearing is much faster than that of a long bearing. This relatively high flow rate improves the cooling by continually replacing the lubricant that is heated by viscous shear. Overheating is a major cause for bearing failure; therefore, operating temperature is a very important consideration in bearing design.
2. A short bearing is less sensitive to misalignment errors. It is obvious that short bearings reduce the risk of damage to the journal and the bearing edge resulting from misalignment of journal and bearing bore centrelines.
3. Wear is reduced, because abrasive wear particles and dust are washed away by the oil more easily in short bearings.



**FIG. 7-1** Short hydrodynamic journal bearing.

4. Short bearings require less space and result in a more compact design. The trend in machine design is to reduce the size of machines. Short hydrodynamic journal bearings compete with rolling-element bearings, which are usually short relative to their diameter.

For all these reasons, short bearings are commonly used today. Long bearings were widely used a few decades ago and are still operating, mostly in old machines or in special applications where high load capacity is required.

The analysis of infinitely short hydrodynamic bearings has been introduced by Dubois and Ocvirk (1953). Unlike the analysis of a long bearing with realistic boundary conditions or of a finite bearing, short-bearing analysis results in a closed-form equation for the pressure wave and load capacity and does not require complex numerical analysis. Also, the closed-form expression of the bearing load capacity simplifies the analysis of a short bearing under dynamic conditions, such as unsteady load and speed.

Dubois and Ocvirk assumed that the pressure gradient around the bearing (in the  $x$  direction in Fig. 7-1) is small and can be neglected in comparison to the pressure gradient in the axial direction ( $z$  direction). In short bearings, the oil film pressure gradient in the axial direction,  $dp/dz$ , is larger by an order of magnitude or more in comparison to the pressure gradient,  $dp/dx$ , around the bearing:

$$\frac{dp}{dz} \gg \frac{dp}{dx} \quad (7-1)$$

By disregarding  $dp/dx$  in comparison to  $dp/dz$ , it is possible to simplify the Reynolds equation. This allows a closed-form analytical solution for the fluid film pressure distribution and load capacity. The resulting closed-form expression of the load capacity can be conveniently applied in bearing design as well as in dynamical analysis. One should keep in mind that when the bearing is not very short and the bearing length,  $L$ , approaches the diameter size,  $D$ , the true load capacity is, in fact, higher than that predicted by the short-bearing analysis. If we

use the short-bearing equation to calculate finite-length bearings, the bearing design will be on the safe side, with high safety coefficient. For this reason, the short-journal equations are widely used by engineers for the design of hydrodynamic journal bearings, even for bearings that are not very short, if there is no justification to spend too much time on elaborate calculations to optimize the bearing design.

## 7.2 SHORT-BEARING ANALYSIS

The starting point of the derivation is the Reynolds equation, which was discussed in [Chapter 5](#). Let us recall that the Reynolds equation for incompressible Newtonian fluids is

$$\frac{\partial}{\partial x} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6(U_1 - U_2) \frac{\partial h}{\partial x} + 12(V_2 - V_1) \quad (7-2)$$

Based on the assumption that the pressure gradient in the  $x$  direction can be disregarded, we have

$$\frac{\partial}{\partial x} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) \approx 0 \quad (7-3)$$

Thus, the Reynolds equation (7-2) reduces to the following simplified form:

$$\frac{\partial}{\partial z} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6(U_1 - U_2) \frac{\partial h}{\partial x} + 12(V_2 - V_1) \quad (7-4)$$

In a journal bearing, the surface velocity of the journal is not parallel to the  $x$  direction along the bore surface, and it has a normal component  $V_2$  (see diagram of velocity components in [Fig. 6-2](#)). The surface velocity components of the journal surface are

$$U_2 \approx U; \quad V_2 \approx U \frac{\partial h}{\partial x} \quad (7-5)$$

On the stationary sleeve, the surface velocity components are zero:

$$U_1 = 0; \quad V_1 = 0 \quad (7-6)$$

After substituting Eqs. (7-5) and (7-6) into the right-hand side of Eq. (7-4), it becomes

$$6(U_1 - U_2) \frac{\partial h}{\partial x} + 12(V_2 - V_1) = 6(0 - U) \frac{\partial h}{\partial x} + 12U \frac{\partial h}{\partial x} = 6U \frac{\partial h}{\partial x} \quad (7-7)$$

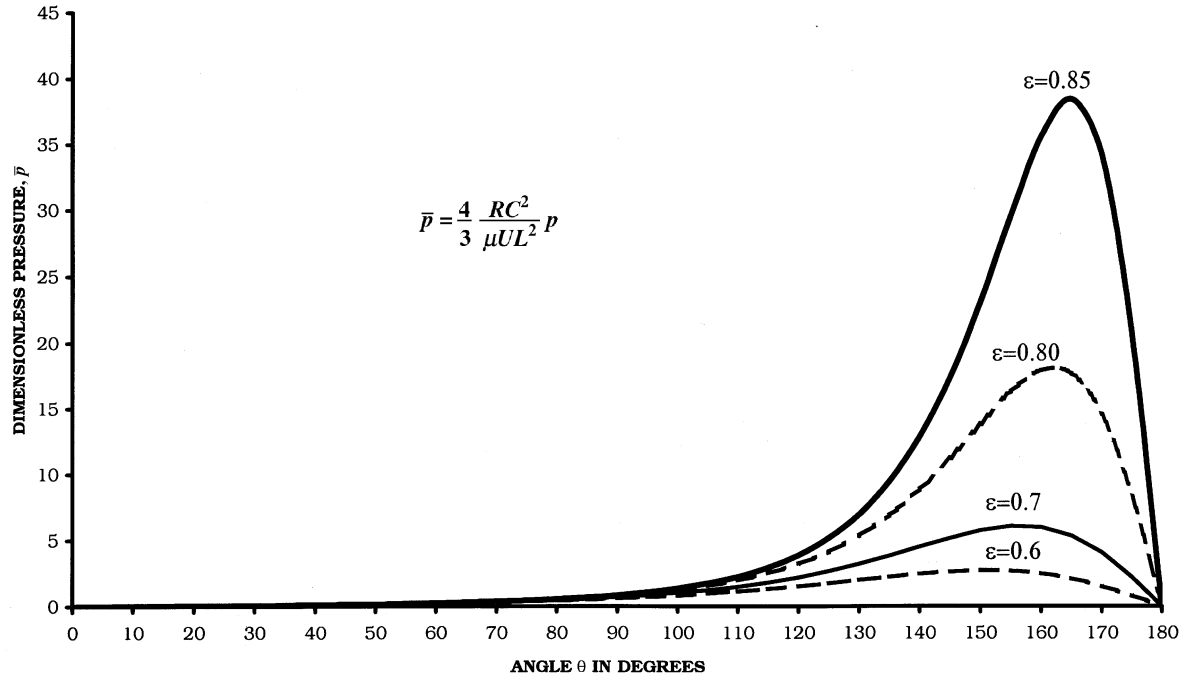


FIG. 7-2 Pressure wave at  $z = 0$  in a short bearing for various values of  $\epsilon$ .

The Reynolds equation for a short journal bearing is finally simplified to the form

$$\frac{\partial}{\partial z} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U \frac{\partial h}{\partial x} \quad (7-8)$$

The film thickness  $h$  is solely a function of  $x$  and is constant for the purpose of integration in the  $z$  direction. Double integration results in the following parabolic pressure distribution, in the  $z$  direction, with two constants, which can be obtained from the boundary conditions of the pressure wave:

$$p = -\frac{6\mu U}{h^3} \frac{dh}{dx} \frac{z^2}{2} + C_1 z + C_2 \quad (7-9)$$

At the two ends of the bearing, the pressure is equal to the atmospheric pressure,  $p = 0$ . These boundary conditions can be written as

$$p = 0 \quad \text{at } z = \pm \frac{L}{2} \quad (7-10)$$

Solving for the integration constants, and substituting the function for  $h$  in a journal bearing,  $h(\theta) = C(1 + \varepsilon \cos \theta)$ , the following expression for the pressure distribution in a short bearing (a function of  $\theta$  and  $z$ ) is obtained:

$$p(\theta, z) = \frac{3\mu U}{RC^2} \left( \frac{L^2}{4} - z^2 \right) \frac{\varepsilon \sin \theta}{(1 + \varepsilon \cos)^3} \quad (7-11)$$

In a short journal bearing, the film thickness  $h$  is converging (decreasing  $h$  vs.  $\theta$ ) in the region ( $0 < \theta < \pi$ ), resulting in a viscous wedge and a positive pressure wave. At the same time, in the region ( $\pi < \theta < 2\pi$ ), the film thickness  $h$  is diverging (increasing  $h$  vs.  $\theta$ ). In the diverging region ( $\pi < \theta < 2\pi$ ), Eq. (7-11) predicts a negative pressure wave (because  $\sin \theta$  is negative). The pressure according to Eq. (7-11) is an antisymmetrical function on the two sides of  $\theta = \pi$ .

In an actual bearing, in the region of negative pressure ( $\pi < \theta < 2\pi$ ), there is fluid cavitation and the fluid continuity is breaking down. There is fluid cavitation whenever the negative pressure is lower than the vapor pressure. Therefore, Eq. (7-11) is no longer valid in the diverging region. In practice, the contribution of the negative pressure to the load capacity can be disregarded. Therefore in a short bearing, only the converging region with positive pressure ( $0 < \theta < \pi$ ) is considered for the load capacity of the oil film (see Fig. 7-2).

Similar to a long bearing, the load capacity is solved by integration of the pressure wave around the bearing. But in the case of a short bearing, the pressure is a function of  $z$  and  $\theta$ . The following are the two equations for the integration

for the load capacity components in the directions of  $W_x$  and  $W_y$  of the bearing centerline and the normal to it:

$$W_x = -2 \int_0^\pi \int_0^{L/2} p \cos \theta R d\theta dz = -\frac{\mu UL^3}{2C^2} \int_0^\pi \frac{\varepsilon \sin \theta \cos \theta}{(1 + \varepsilon \cos \theta)^3} d\theta \quad (7-12a)$$

$$W_y = 2 \int_0^\pi \int_0^{L/2} p \sin \theta d\theta dz = \frac{\mu UL^3}{2C^2} \int_0^\pi \frac{\varepsilon \sin^2 \theta}{(1 + \varepsilon \cos \theta)^3} d\theta \quad (7-12b)$$

The following list of integrals is useful for short journal bearings.

$$J_{11} = \int_0^\pi \frac{\sin^2 \theta}{(1 + \varepsilon \cos \theta)^3} d\theta = \frac{\pi}{2(1 - \varepsilon^2)^{3/2}} \quad (7-13a)$$

$$J_{12} = \int_0^\pi \frac{\sin \theta \cos \theta}{(1 + \varepsilon \cos \theta)^3} d\theta = \frac{-2\varepsilon}{(1 - \varepsilon^2)^2} \quad (7-13b)$$

$$J_{22} = \int_0^\pi \frac{\cos^2 \theta}{(1 + \varepsilon \cos \theta)^3} d\theta = \frac{\pi(1 + 2\varepsilon^2)}{2(1 - \varepsilon^2)^{5/2}} \quad (7-13c)$$

The load capacity components are functions of the preceding integrals:

$$W_x = -\frac{\mu UL^3}{2C^2} \varepsilon J_{12} \quad (7-14)$$

$$W_y = \frac{\mu UL^3}{2C^2} \varepsilon J_{11} \quad (7-15)$$

Using Eqs. (7-13) and substitution of the values of the integrals results in the following expressions for the two load components:

$$W_x = \frac{\mu UL^3}{C^2} \frac{\varepsilon^2}{(1 - \varepsilon^2)^2} \quad (7-16a)$$

$$W_y = \frac{\mu UL^3}{4C^2} \frac{\pi \varepsilon}{(1 - \varepsilon^2)^{3/2}} \quad (7-16b)$$

Equations (7-16) for the two load components yield the resultant load capacity of the bearing,  $W$ :

$$W = \frac{\mu UL^3}{4C^2} \frac{\varepsilon}{(1 - \varepsilon^2)^2} [\pi^2(1 - \varepsilon^2) + 16\varepsilon^2]^{1/2} \quad (7-17)$$

The attitude angle,  $\phi$ , is determined from the two load components:

$$\tan \phi = \frac{W_y}{W_x} \quad (7-18)$$

Via substitution of the values of the load capacity components, the expression for the attitude angle of a short bearing becomes

$$\tan \phi = \frac{\pi(1 - \varepsilon^2)^{1/2}}{4\varepsilon} \quad (7-19)$$

### 7.3 FLOW IN THE AXIAL DIRECTION

The velocity distribution of the fluid in the axial,  $z$ , direction is

$$w = 3U \frac{zh'}{Rh^3}(y^2 - hy) \quad (7-20)$$

Here,

$$h' = \frac{dh}{dx} \quad (7-21)$$

The gradient  $h'$  is the clearance slope (wedge angle), which is equal to the fluid film thickness slope in the direction of  $x = R\theta$  (around the bearing). This gradient must be negative in order to result in a positive pressure wave as well as positive flow,  $w$ , in the  $z$  direction.

Positive axial flow is directed outside the bearing (outlet flow from the bearing). There is positive axial flow where  $h$  is converging (decreasing  $h$  vs.  $x$ ) in the region ( $0 < \theta < \pi$ ). At the same time, there is inlet flow, directed from outside into the bearing, where  $h$  is diverging (increasing  $h$  vs.  $\theta$ ) in the region ( $\pi < \theta < 2\pi$ ). In the diverging region,  $h' > 0$ , there is fluid cavitation that is causing deviation from the theoretical axial flow predicted in Eq. (7-20). However, in principle, the lubricant enters into the bearing in the diverging region and leaves the bearing in the converging region. In a short bearing, there is much faster lubricant circulation relative to that in a long bearing. Fast lubricant circulation reduces the peak temperature of the lubricant. This is a significant advantage of the short bearing, because high peak temperature can cause bearing failure.

### 7.4 SOMMERFELD NUMBER OF A SHORT BEARING

The definition of the dimensionless Sommerfeld number for a short bearing is identical to that for a long bearing; however, for a short journal bearing, the expression of the Sommerfeld number is given as,

$$S = \frac{\mu n}{P} \left(\frac{R}{C}\right)^2 = \left(\frac{D}{L}\right)^2 \frac{(1 - \varepsilon^2)^2}{\pi \varepsilon [\pi^2(1 - \varepsilon^2) + 16\varepsilon^2]^{1/2}} \quad (7-22)$$

Let us recall that the Sommerfeld number of a long bearing is only a function of  $\varepsilon$ . However, for a short bearing, the Sommerfeld number is a function of  $\varepsilon$  as well as the ratio  $L/D$ .

## 7.5 VISCOUS FRICTION

The friction force around a bearing is obtained by integration of the shear stresses. The shear stress in a short bearing is

$$\tau = \mu \frac{U}{h} \quad (7-23)$$

For the purpose of computing the friction force, the shear stresses are integrated around the complete bearing. The fluid is present in the diverging region ( $\pi < \theta < 2\pi$ ), and it is contributing to the viscous friction, although its contribution to the load capacity has been neglected. The friction force,  $F_f$ , is obtained by integration of the shear stress,  $\tau$ , over the complete surface area of the journal:

$$F_f = \int_{(A)} \tau \, dA \quad (7-24)$$

Substituting  $dA = LR \, d\theta$ , the friction force becomes

$$F_f = RL \int_0^{2\pi} \tau \, d\theta \quad (7-25)$$

To solve for the friction force, we substitute the expression for  $h$  into Eq. (7-23) and substitute the resulting equation of  $\tau$  into Eq. (7-25). For solving the integral, the following integral equation is useful:

$$J_1 = \int_0^{2\pi} \frac{1}{1 + \cos \theta} \, d\theta = \frac{2\pi}{(1 - \varepsilon^2)^{1/2}} \quad (7-26)$$

Note that  $J_1$  has the limits of integration  $0-2\pi$ , while for the first three integrals in Eqs. (7-13), the limits are  $0-\pi$ . The final expression for the friction force is

$$F_f = \frac{\mu LRU}{C} \int_0^{2\pi} \frac{d\theta}{1 + \varepsilon \cos \theta} = \frac{\mu LRU}{C} \frac{2\pi}{(1 - \varepsilon^2)^{1/2}} \quad (7-27)$$

The bearing friction coefficient  $f$  is defined as

$$f = \frac{F_f}{W} \quad (7-28)$$

The friction torque  $T_f$  is

$$T_f = F_f R; \quad T_f = \frac{\mu LR^2 U}{C} \frac{2\pi}{(1 - \varepsilon^2)^{1/2}} \quad (7-29)$$

The energy loss per unit of time,  $\dot{E}_f$  is determined by the following:

$$\dot{E}_f = F_f U \quad (7-30a)$$

Substituting Eq. (7-27) into Eq. (7-30a) yields the following expression for the power loss on viscous friction:

$$\dot{E}_f = \frac{\mu L R U^2}{C} \frac{2\pi}{(1 - \varepsilon^2)^{1/2}} \quad (7-30b)$$

## 7.6 JOURNAL BEARING STIFFNESS

Journal bearing stiffness,  $k$ , is the rate of increase of load  $W$  with displacement  $e$  in the same direction,  $dW/de$  (similar to that of a spring constant). High stiffness is particularly important in machine tools, where any displacement of the spindle centerline during machining would result in machining errors. Hydrodynamic journal bearings have low stiffness at low eccentricity (under light load).

The displacement of a hydrodynamic bearing is not in the same direction as the force  $W$ . In such cases, the journal bearing has *cross-stiffness* components. The stiffness components are presented as four components related to the force components  $W_x$  and  $W_y$  and the displacement components in these directions.

In a journal bearing, the load is divided into two components,  $W_x$  and  $W_y$ , and the displacement of the bearing center,  $e$ , is divided into two components,  $e_x$  and  $e_y$ . The two components of the journal bearing stiffness are

$$k_x = \frac{dW_x}{de_x}; \quad k_y = \frac{dW_y}{de_y} \quad (7-31)$$

and the two components of the cross-stiffness are defined as

$$k_{xy} = \frac{dW_x}{de_y}; \quad k_{yx} = \frac{dW_y}{de_x} \quad (7-32)$$

Cross-stiffness components cause instability, in the form of an oil whirl in journal bearings.

### Example Problem 7-1

A short bearing is designed to operate with an eccentricity ratio  $\varepsilon = 0.8$ . The journal diameter is 60 mm, and its speed is 1500 RPM. The journal is supported by a short hydrodynamic bearing of length  $L/D = 0.5$ , and clearance ratio  $C/R = 10^{-3}$ . The radial load on the bearing is 1 metric ton (1 metric ton = 9800 [N]).

- a. Assume that infinitely-short-bearing theory applies to this bearing, and find the Sommerfeld number.

- b. Find the minimum viscosity of the lubricant for operating at  $\varepsilon = 0.8$ .
- c. Select a lubricant if the average bearing operating temperature is  $80^\circ\text{C}$ .

### Solution

#### a. Sommerfeld Number

The Sommerfeld number is

$$S = \frac{\mu n}{P} \left(\frac{R}{C}\right)^2 = \left(\frac{D}{L}\right)^2 \frac{(1 - \varepsilon^2)^2}{\pi \varepsilon [\pi^2(1 - \varepsilon^2) + 16\varepsilon^2]^{1/2}}$$

From the right-hand side of the equation,

$$S = (2)^2 \frac{(1 - 0.8^2)^2}{\pi \cdot 0.8 [\pi^2(1 - 0.8^2) + 16 \times 0.8^2]^{1/2}} = \frac{0.36^2}{\pi \times 0.8 \times 3.71} = 0.0139$$

#### b. Minimum Viscosity

The average pressure is

$$P = \frac{W}{LD} = \frac{9800}{0.06 \times 0.03} = 5.44 \times 10^6 \text{ Pa}$$

The load in SI units (newtons) is 9800 [N], and the speed is  $n = 1500/60 = 25$  RPS.

The viscosity is determined by equating:

$$S = \frac{\mu n}{P} \left(\frac{R}{C}\right)^2 = 0.0139$$

$$\mu = \frac{5.44 \times 10^6 \times 0.0139}{25 \times 10^6} = 0.0030 \text{ [N-s/m}^2\text{]}$$

#### c. Lubricant

For lubricant operating temperature of  $80^\circ\text{C}$ , mineral oil SAE 10 has suitable viscosity of  $0.003 \text{ [N-s/m}^2\text{]}$  (Fig. 2-3). This is the minimum required viscosity for the operation of this bearing with an eccentricity ratio no higher than  $\varepsilon = 0.8$ .

### Example Problem 7-2

A journal of 75-mm diameter rotates at 3800 RPM. The journal is supported by a short hydrodynamic bearing of length  $L = D/4$  and a clearance ratio  $C/R = 10^{-3}$ . The radial load on the bearing is 0.5 metric ton, (1 metric ton = 9800 [N]). The lubricant is SAE 40, and the operating temperature of the lubricant in the bearing is  $80^\circ\text{C}$ .

- Assume infinitely-short-bearing theory, and find the eccentricity ratio,  $\varepsilon$ , of the bearing (use a graphic method to solve for  $\varepsilon$ ) and the minimum film thickness,  $h_n$ .
- Derive the equation for the pressure wave around the bearing, at the center of the width (at  $z = 0$ ).
- Find the hydrodynamic friction torque and the friction power losses (in watts).

## Solution

The following conversion is required for calculation in SI units:

Speed of shaft:  $n = 3800/60 = 63.3$  [RPS]

Radial load:  $W = 0.5 \times 9800 = 4900$  [N]

Axial length of shaft:  $L = D/4 = 0.075/4 = 0.01875$  [m]

$C/R = 10^{-3}$ ; hence  $R/C = 10^3$ .

### a. Eccentricity Ratio and Minimum Film Thickness

For an operating temperature of  $T = 80^\circ\text{C}$ , the viscosity of SAE-40 oil is obtained from the viscosity–temperature chart:  $\mu = 0.0185$  [N-s/m<sup>2</sup>]. The equation for the load capacity is applied to solve for the eccentricity ratio,  $\varepsilon$ , the only unknown in the following equation:

$$W = \frac{\mu UL^3}{4C^2} \frac{\varepsilon}{(1 - \varepsilon^2)^2} [\pi^2(1 - \varepsilon^2) + 16\varepsilon^2]^{1/2} \quad (7-32)$$

To simplify the mathematical derivation of  $\varepsilon$ , the following substitution is helpful:

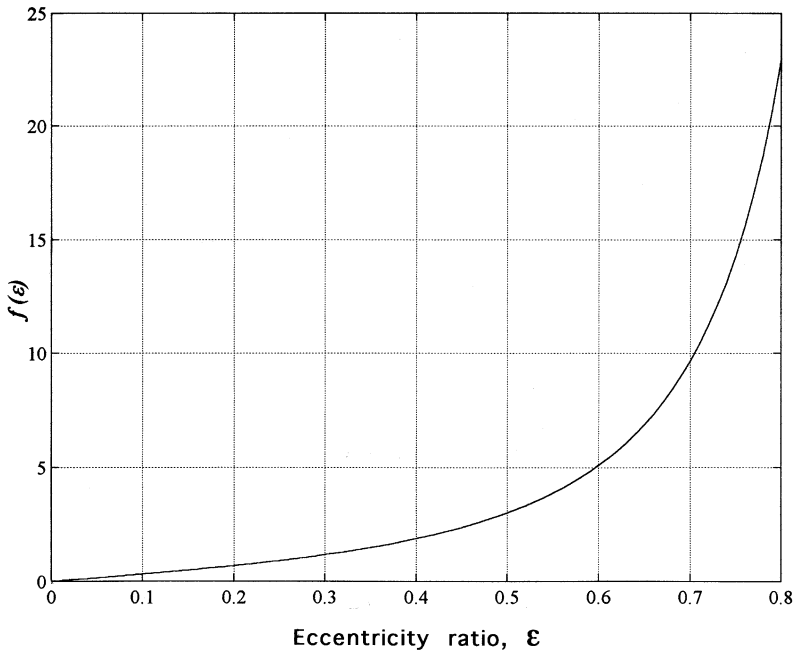
$$f(\varepsilon) = \frac{\varepsilon}{(1 - \varepsilon^2)^2} [\pi^2(1 - \varepsilon^2) + 16\varepsilon^2]^{1/2} \quad (7-33)$$

First, we can solve for  $f(\varepsilon)$ , and later we can obtain the value of  $\varepsilon$  from the graph of  $f(\varepsilon)$  vs.  $\varepsilon$  (Fig. 7-3).

Equations (7-32) and (7-33) yield

$$f(\varepsilon) = \frac{4WC^2}{\mu UL^3} = \frac{4W(10^{-3} \times D/2)^2}{\mu \pi n D (D/4)^3} = \frac{4 \times 16 \times 4900 \times 10^{-6}}{0.0185 \pi \times 63.3 (0.075)^2} = 15.15$$

According to the curve of  $f(\varepsilon)$  vs.  $\varepsilon$ , for  $f(\varepsilon) = 15.15$ , the eccentricity of the bearing is  $\varepsilon = 0.75$ .



**FIG. 7-3** Graph of  $f(\varepsilon)$  vs.  $\varepsilon$ , describing Eq. (7-33).

We find the minimum film thickness,  $h_{\min}$ , as follows:

$$\begin{aligned}
 h_{\min} &= C(1 - \varepsilon) = 10^{-3} \times R(1 - 0.75) = 10^{-3} \left( \frac{0.075}{2} \right) 0.25 \\
 &= 9.4 \times 10^{-6} \text{ m} \\
 h_{\min} &= 9.4 \text{ } \mu\text{m}
 \end{aligned}$$

*b. Pressure Wave*

The pressure is a function of  $\theta$  and  $z$ , according to the equation

$$p = \frac{3\mu U}{h^3} \frac{dh}{dx} \left( Z^2 - \frac{L^2}{4} \right)$$

where  $h = C(1 + \varepsilon \cos \theta)$ ,  $x = R\theta$ , and  $dx = R d\theta$ . The clearance slope is

$$\frac{dh}{dx} = \frac{dh}{d\theta} \frac{d\theta}{dx} = -\frac{C\varepsilon \sin \theta}{R}$$

The pressure wave at the width center,  $z = 0$ , is

$$p(\theta) = \frac{3\mu UL^2 C\varepsilon \sin \theta}{4h^3 R}$$

$$p(\theta) \text{ (at } z = 0) = \frac{3 \times 0.0185 \times \pi \times 63.3 \times 0.075 \times 0.075^2 \times 0.75}{64 \times 10^3}$$

$$\times \frac{\sin \theta}{(1 + 0.75 \cos \theta)^3}$$

$$p(\theta) \text{ (at } z = 0) = \frac{5.5 \times 10^{-8} \sin \theta}{(1 + 0.75 \cos \theta)^3}$$

*c. Friction Torque and Friction Power Loss*

We find the friction torque as follows:

$$T_f = F_f R = \frac{\mu LR^2 U}{C} \frac{2\pi}{(1 - \varepsilon^2)^{1/2}}$$

$$= \frac{0.0185 \times 0.01875 \times (\pi \times 63.3 \times 0.075)}{10^{-3}} \frac{2\pi}{(1 - 0.75^2)^{1/2}}$$

$$T_f = 1.80 \text{ N-m}$$

The friction power loss is found as follows:

$$\dot{E}_f = T_f \omega$$

$$\dot{E}_f = \frac{\mu LR U^2}{C} \frac{2\pi}{(1 - \varepsilon^2)^{1/2}}$$

$$\dot{E}_f = \frac{0.0185 \times 0.01875 \times (\pi \times 63.3 \times 0.075)^2}{10^{-3}} \frac{2\pi}{(1 - 0.75^2)^{1/2}} = 733 \text{ [W]}$$

## Problems

- 7-1 A short bearing is designed to operate with an eccentricity ratio of  $\varepsilon = 0.7$ . Find the journal diameter if the speed is 30,000 RPM and the radial load on the bearing is 8000 N. The bearing length ratio  $L/D = 0.6$ , and the clearance ratio is  $C/R = 10^{-3}$ . The lubricant is SAE 30 and the average operating temperature in the bearing is 70°C. Assume that infinitely-short-bearing theory applies.
- 7-2 Plot the dimensionless pressure distribution (function of  $\theta$ ) at the bearing center,  $z = 0$ , in Example Problem 7-2.
- 7-3 A short bearing is designed to operate with an eccentricity ratio of  $\varepsilon = 0.75$ . The journal is 80 mm in diameter, and its speed is

3500 RPM. The journal is supported by a short hydrodynamic bearing of length  $D/L = 4$  and a clearance ratio of  $C/R = 10^{-3}$ . The radial load on the bearing is 1000 N.

- a. Assume that infinitely-short-bearing theory applies to this bearing, and find the minimum viscosity of the lubricant.
- b. Select a lubricant for an average operating temperature in the bearing of  $60^\circ\text{C}$ .

7-4 The journal speed of a 100 mm diameter journal is 2500 RPM. The journal is supported by a short hydrodynamic bearing of length  $L = 0.6D$  and a clearance ratio of  $C/R = 10^{-3}$ . The radial load on the bearing is 10,000 [N]. The lubricant is SAE 30, and the operating temperature of the lubricant in the bearing is  $70^\circ\text{C}$ .

- a. Assume infinitely-short-bearing theory, and find the eccentricity ratio,  $\varepsilon$ , of the bearing and the minimum film thickness,  $h_n$  (use a graphic method to solve for  $\varepsilon$ ).
- b. Derive the equation and plot the pressure distribution around the bearing, at the center of the width (at  $z = 0$ ).
- c. Find the hydrodynamic friction torque and the friction power losses (in watts) for each bearing.