Chapter 12: Hydrodynamic and Hydrostatic Bearings

A cup of tea, standing in a dry saucer, is apt to slip about in an awkward manner, for which a remedy is found in introduction of a few drops of water, or tea, wetting the parts in contact.

Lord Rayleigh (1918)

A Kingsbury Bearing.
Density wedge and stretch

Figure 12.1 Density wedge mechanism.

Figure 12.2 Stretch mechanism.
Physical Wedge and Normal Squeeze

Figure 12.3 Physical wedge mechanism.

Figure 12.4 Normal squeeze mechanism.
Translation Squeeze and Local Expansion

Figure 12.5 Translation squeeze mechanism.

Figure 12.6 Local expansion mechanism.
Velocity Profiles in Slider Bearings

Figure 12.7 Velocity profiles in a parallel-surface slider bearing.

Figure 12.8 Flow within a fixed-incline slider bearing (a) Couette flow; (b) Poiseuille flow; (c) resulting velocity profile.
Figure 12.9 Thrust slider bearing geometry.
Figure 12.10 Force components and oil film geometry in a hydrodynamically lubricated thrust slider bearing.

Figure 12.11 Side view of fixed-incline slider bearing.
Design Procedure for Fixed-Incline Thrust Bearing

1. Choose a pad length-to-width ratio. A square pad \( \lambda = 1 \) is generally thought to give good performance. If it is known whether maximum load or minimum power loss is more important in a particular application, the outlet film thickness ratio \( H_o \) can be determined from Fig. 12.13.

2. Once \( \lambda \) and \( H_o \) are known, Fig. 12.14 can be used to obtain the bearing number \( B_t \).

3. From Fig. 12.15 determine the temperature rise due to shear heating for a given \( \lambda \) and \( B_t \).

The volumetric specific heat \( C_s = \rho C_p \), which is the dimensionless temperature rise parameter, is relatively constant for mineral oils and is equivalent to \( 1.36 \times 10^6 \) N/(m\(^2\)°C).

4. Determine lubricant temperature. Mean temperature can be expressed as

\[
\tilde{t}_m = t_{mi} + \frac{\Delta t_m}{2}
\]

where \( t_{mi} \) = inlet temperature, °C. The inlet temperature is usually known beforehand.

Once the mean temperature \( t_m \) is known, it can be used in Fig. 8.17 to determine the viscosity of SAE oils, or Fig. 8.16 or Table 8.4 can be used. In using Table 8.4 if the temperature is different from the three temperatures given, a linear interpolation can be used.

(continued)
Design Procedure for Fixed-Incline Thrust Bearing

5. Make use of Eqs. (12.34) and (12.68) to get the outlet (minimum) film thickness \( h_0 \) as

\[
h_0 = H_o l \sqrt{\frac{\eta_0 u_b w_t}{W_z B_t}}
\]

Once the outlet film thickness is known, the shoulder height \( s_h \) can be directly obtained from \( s_h = \frac{h_0}{H_o} \). If in some applications the outlet film thickness is specified and either the velocity \( u_b \) or the normal applied load \( W_z \) is not known, Eq. (12.72) can be rewritten to establish \( u_b \) or \( W_z \).

6. Check Table 12.1 to see if the outlet (minimum) film thickness is sufficient for the pressurized surface finish. If \( h_o \) from Eq. (12.72) \( \geq h_o \) from Table 12.1, go to step 7. If \( h_o \) from Eq. (12.72) \(< h_o \) from Table 12.1, consider one or both of the following steps:
   a. Increase the bearing speed.
   b. Decrease the load, the surface finish, or the inlet temperature. Upon making this change return to step 3.

7. Evaluate the other performance parameters. Once an adequate minimum film thickness and a proper lubricant temperature have been determined, the performance parameters can be evaluated. Specifically, from Fig. 12.16 the power loss, the coefficient of friction, and the total and side flows can be determined.
Slider Bearings: Configuration and Film Thickness

Figure 12.12 Configuration of multiple fixed-incline thrust slider bearing

Figure 12.13 Chart for determining minimum film thickness corresponding to maximum load or minimum power loss for various pad proportions in fixed-incline bearings.
### Film Thickness for Hydrodynamic Bearings

<table>
<thead>
<tr>
<th>Surface finish (centerline average), $R_a$</th>
<th>Description of surface</th>
<th>Examples of manufacturing methods</th>
<th>Approximate relative costs</th>
<th>Allowable outlet (minimum) film thickness$^a$, $h_o$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$m</td>
<td>$\mu$m.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1-0.2</td>
<td>4-8</td>
<td>Mirror-like surface without toolmarks; close tolerances</td>
<td>Grind, lap, and superfinish</td>
<td>17-20</td>
</tr>
<tr>
<td>.2-.4</td>
<td>8-16</td>
<td>Smooth surface without scratches; close tolerances</td>
<td>Grind and lap</td>
<td>17-20</td>
</tr>
<tr>
<td>.4-.8</td>
<td>16-32</td>
<td>Smooth surfaces; close tolerances</td>
<td>Grind, file, and lap</td>
<td>10</td>
</tr>
<tr>
<td>.8-1.6</td>
<td>32-63</td>
<td>Accurate bearing surface without toolmarks</td>
<td>Grind, precision mill, and file</td>
<td>7</td>
</tr>
<tr>
<td>1.6-3.2</td>
<td>63-125</td>
<td>Smooth surface without objectionable toolmarks; moderate tolerances</td>
<td>Shape, mill, grind and turn</td>
<td>5</td>
</tr>
</tbody>
</table>

$^a$The values of film thickness are given only for guidance. They indicate the film thickness required to avoid metal-to-metal contact under clean oil conditions with no misalignment. It may be necessary to take a larger film thickness than that indicated (e.g., to obtain an acceptable temperature rise). It has been assumed that the average surface finish of the pads is the same as that of the runner.

**Table 12.1** Allowable outlet (minimum) film thickness for a given surface finish.
Figure 12.14 Chart for determining minimum film thickness for fixed-incline thrust bearings.
Figure 12.14 Chart for determining dimensionless temperature rise due to viscous shear heating of lubricant for fixed-incline thrust bearings.
Thrust Bearings - Friction Coefficient

Figure 12.16a Chart for determining friction coefficient for fixed-incline thrust bearings.
Figure 12.16b Chart for determining power loss for fixed-incline thrust bearings.
Figure 12.16c Chart for determining lubricant flow for fixed-incline thrust bearings.
Figure 12.16d Chart for determining lubricant side flow for fixed-incline thrust bearings.
Figure 12.17 Pressure distribution around a journal bearing.
Figure 12.18 Concentric Journal Bearing

Figure 12.19 Developed journal bearing surfaces for a concentric journal bearing.

**Concentric Journal Bearing**

\[ h = c = \frac{2\pi r}{u} \]

**Hamrock • Fundamentals of Machine Elements**
Typical Radial Load for Journal Bearings

<table>
<thead>
<tr>
<th>Application</th>
<th>Average radial load per area, $W_r^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>psi</td>
</tr>
<tr>
<td>Automotive engines:</td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>600-750</td>
</tr>
<tr>
<td>Connecting rod bearing</td>
<td>1700-2300</td>
</tr>
<tr>
<td>Diesel engines:</td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>900-1700</td>
</tr>
<tr>
<td>Connecting rod bearing</td>
<td>1150-2300</td>
</tr>
<tr>
<td>Electric motors</td>
<td>120-250</td>
</tr>
<tr>
<td>Steam turbines</td>
<td>150-300</td>
</tr>
<tr>
<td>Gear reducers</td>
<td>120-250</td>
</tr>
<tr>
<td>Centrifugal pumps</td>
<td>100-180</td>
</tr>
<tr>
<td>Air compressors:</td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>140-280</td>
</tr>
<tr>
<td>Crankpin</td>
<td>280-500</td>
</tr>
<tr>
<td>Centrifugal pumps</td>
<td>100-180</td>
</tr>
</tbody>
</table>

Table 12.2  Typical radial load per area $W_r^*$ in use for journal bearings.
Figure 12.20  Effect of bearing number on minimum film thickness for four diameter-to-width ratios.
Figure 12.21  Effect of bearing number on attitude angle for four different diameter-to-width ratios.
Figure 12.21 Effect of bearing number on coefficient of friction for four different diameter-to-width ratios.
Figure 12.23  Effect of bearing number on dimensionless volume flow rate for four different diameter-to-width ratios.
Figure 12.21 Effect of bearing number on side-flow leakage for four different diameter-to-width ratios.
Figure 12.25 Effect of bearing number on dimensionless maximum film pressure for four different diameter-to-width ratios.
Figure 12.21  Effect of bearing number on location of terminating and maximum pressure for four different diameter-to-width ratios.
Figure 12.27 Effect of radial clearance on some performance parameters for a particular case.
Figure 12.21 Parallel-surface squeeze film bearing

\[ w = - \frac{\partial h}{\partial t} \]
Figure 12.29 Formation of fluid film in hydrostatic bearing system. (a) Pump off; (b) pressure buildup; (c) pressure times recess area equals normal applied load; (d) bearing operation; (e) increased load; (f) decreased load.
Figure 12.30 Radial flow hydrostatic thrust bearing with circular step pad.