Chapter 12
Lubrication and Journal Bearings

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Chapter Outline

1. Types of Lubrication
2. Viscosity
3. Petroff’s Equation
4. Stable Lubrication
5. Thick-Film Lubrication
6. Hydrodynamic Theory
7. Design Considerations
8. The Relations of the Variables
10. Clearance
11. Pressure-Fed Bearings
12. Loads and Materials
13. Bearing Types
14. Thrust Bearings
15. Boundary-Lubricated Bearings

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Overview

- The purpose of lubrication is to reduce friction, wear and heating of machine parts moving relative to each other.

- In sleeve bearing, a shaft (or Journal) rotates within a sleeve (or bushing) and the relative motion is sliding.

- In ball bearing the relative motion is rolling.

- Journal bearings are applicable for:
  - Extreme operational conditions (high loads & rotational speeds).
  - Low demand applications (without external lubrication) because they are more cost effective than antifriction bearings.

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12–1 Types of Lubrication

1. Hydrodynamic
2. Hydrostatic
3. Elastohydrodynamic
4. Boundary
5. Solid film
12–1 Types of Lubrication

Hydrodynamic (full-film)

- Surfaces of the bearing are separated by a relatively thick film of lubricant (*to prevent metal to metal contact*).

- Film pressure is created by the moving surface forcing the lubricant into a wedge-shaped zone, therefore creating a pressure that separates the sliding surfaces.

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12–1 Types of Lubrication

Hydrostatics

- Lubricant is forced into the bearing at a pressure high enough to separate the surfaces.
- Relative motion of the surfaces is not required in this case.
12–1 Types of Lubrication

Elastohydrodynamic

- Lubricant is introduced between surfaces that are in rolling contact
  - mating gears or rolling bearings
12–1 Types of Lubrication

Boundary

- Special case of hydrodynamic lubrication where film thickness is reduced to be “very thin”

- This may happen because of:
  - increased load
  - reduced lubricant supply
  - reduced rotational speed
  - reduced viscosity
Types of Lubrication

**Solid-film**

- Self-lubricating solid materials such as graphite are used in the bearing.
- Used when bearings must operate at very high temperature.
12–2 Viscosity

Newton’s viscous effect: the shear stress in the fluid is proportional to the rate of change of velocity with respect to $y$.

\[ \tau = \frac{F}{A} = \mu \frac{du}{dy} \]  

(12–1)
12–2 Viscosity

- \( \mu = \text{absolute viscosity (dynamic viscosity)} \)
- \( du/dy = \text{rate of shear, or velocity gradient.} \)
- Viscosity \( \mu = \) a measure of internal frictional resistance of the fluid.
- For most lubricating fluids, rate of shear is constant, and \( du/dy = U/h, \)

\[
\tau = \frac{F}{A} = \mu \frac{U}{h}
\]

(12–2)
12–2 Viscosity

Unit for viscosity “μ”

- In SI system = Pa.s
- In US system = lb.s/in² (psi.s) called “reyn”
- μ’ = μreyn
- 1 reyn (psi.s) = 6895 Pa.s
- poise = dyn . s/cm²
- Centipoise (cP) = Z

\[
\mu(\text{Pa} \cdot \text{s}) = (10)^{-3} Z \ (\text{cP})
\]

\[
\mu(\text{reyn}) = \frac{Z \ (\text{cP})}{6.89(10)^6}
\]

\[
\mu(\text{mPa} \cdot \text{s}) = 6.89 \ \mu' (\mu\text{reyn})
\]
12–2 Viscosity

- Saybolt Universal Viscosimeter
- ASTM standard method for determining viscosity.
- measuring time in seconds for 60 mL of lubricant at a specified temperature to run through a tube 17.6 mm in diameter and 12.25 mm long.
- Result = *kinematic viscosity*,
- unit = cm²/s = *stoke*

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12–2 Viscosity

- Using Hagen-Poiseuille law, kinematic viscosity based upon seconds Saybolt, also called Saybolt Universal viscosity (SUV) in seconds, is

\[ Z_k = \left( 0.22t - \frac{180}{t} \right) \]  

- \( Z_k \) = centistokes (cSt)
- \( t \) = number of seconds Saybolt

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12–2 Viscosity

- In SI, kinematic viscosity $v$ has the unit of $m^2/s$, $v(m^2/s) = 10^{-6}Z_k$ (cSt)

$$v = \left(0.22t - \frac{180}{t}\right) \times 10^{-6}$$  \hspace{1cm} (12–4)

- To convert to dynamic viscosity, multiply $v$ by density in SI units.

- density, $\rho$ with the unit of $kg/m^3$, we have

$$\mu = \rho \left(0.22t - \frac{180}{t}\right) \times 10^{-6}$$  \hspace{1cm} (12–5)

- $\mu$ is in pascal-seconds

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12–2 Viscosity
12–3 Petroff’s Equation

- Petroff equation gives the coefficient of friction in journal bearings.
- It is based on the assumption that the shaft is concentric.
- Though the shaft is not concentric, coefficient of friction predicted by this equation turns out to be quite good.

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12–3 Petroff’s Equation

- It is assumed that:
  - Bearing carries a very small load
  - Clearance space is completely filled with oil
  - Leakage is negligible

Figure 12–3

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12–3 Petroff’s Equation

- The shaft rotates at $N$ rev/s
- Surface velocity is $U = 2\pi r N$ in/s

$$\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c}$$

$$T = (\tau A)(r) = \left( \frac{2\pi r \mu N}{c} \right) (2\pi r l)(r) = \frac{4\pi^2 r^3 l \mu N}{c}$$

- Designate a small force on the bearing by $W$ (lbf)
- $P$ (psi) = pressure in lbf/in$^2$ of projected area $P = W/2rl$
- Frictional force = $f W$
- $f$ = coefficient of friction, and so the frictional torque is

$$T = f W r = (f)(2rlP)(r) = 2r^2 f l P$$

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12–3 Petroff’s Equation

- Petroff’s equation
  \[ f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c} \]  
  \( (12-6) \)

- The bearing characteristic number (Sommerfeld number, is defined by the equation
  \[ S = \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} \]  
  \( (12-7) \)

- \( r/c = \text{radial clearance ratio} \)
  \[ f \frac{r}{c} = 2\pi^2 \frac{\mu N}{P} \left( \frac{r}{c} \right)^2 = 2\pi^2 S \]  
  \( (12-8) \)
12–4 Stable Lubrication

- Petroff’s bearing model (Eq. 12–6):
  - $f$ is proportional to $\mu N/P$, a straight line from origin in first quadrant.
  - It presumes thick-film lubrication, no metal-to-metal contact, the surfaces being completely separated by a lubricant film.

- Design constraint to keep thick-film lubrication:
  \[
  \frac{\mu N}{P} \geq 1.7 \times 10^{-6}
  \]
12–4 Stable Lubrication

To right of line BA:

- Temperature $\uparrow$, viscosity $(\mu) \downarrow$, $\mu N/P \downarrow$, $f \downarrow$, heat generated by shearing the lubricant $\downarrow$, Temperature $\downarrow$.

- **Self-correcting:** stable lubrication

Figure 12–4
12–4 Stable Lubrication

To left of line BA:

- Temperature $\uparrow$, viscosity $(\mu) \downarrow$, $\mu N/P \downarrow$, $f \uparrow$, (metal to metal contact) more heat is generated, Temperature $\uparrow$.

- Damage: unstable lubrication

Figure 12–4

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12–5 Thick-Film Lubrication

- Suppose the journal starts to rotate in cw direction while it is still dry.
- The journal will roll up the right side of the bearing, as seen in (a).
- Once lubricant is introduced, the rotating journal will pump lubricant around the bearing by forcing into a wedge-shaped space, and this forces the journal to move to the other side (left side) of the bearing, as seen in (b).
12–5 Thick-Film Lubrication

- The “minimum film thickness”, $h_o$, occurs at bottom half of the bearing but slightly to the left (for cw rotation), as seen in (b).
12–5 Thick-Film Lubrication

Figure 12–6: nomenclature of a journal bearing

- **Radial clearance**
  
  \[ c = r_b - r_j \]

- **Eccentricity** “\( e \)”:
  
  distance between centers of bushing & journal.

  \[ e = c - h_o \]

- **Eccentricity ratio**

  \[ \varepsilon = \frac{e}{c} \]

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12–6 Hydrodynamic Theory

Figure 12–7: Schematic representation of partial bearing used by Tower

- A pressure gauge connected to the hole indicated a pressure of more than twice the unit bearing load.
12–6 Hydrodynamic Theory

**Figure 12–8:** Approximate pressure distribution curves obtained by Tower
12–6 Hydrodynamic Theory

- The fluid films were so thin in comparison with the bearing radius that the curvature could be neglected.
- This enabled Reynolds to replace the curved partial bearing with a flat bearing, called a plane slider bearing.
12–6 Hydrodynamic Theory

Reynolds assumptions:

1. Lubricant obeys Newton’s viscous effect, Eq. 12–1
2. Forces due to inertia of lubricant are neglected
3. Lubricant is assumed to be incompressible
4. Viscosity is assumed to be constant throughout the film
5. Pressure does not vary in the axial direction
12–6 Hydrodynamic Theory

Figure 12-9

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12–6 Hydrodynamic Theory

Using Reynolds’ assumption that curvature can be neglected, we fix a right-handed xyz reference system to the stationary bearing.
12–6 Hydrodynamic Theory

Additional assumptions:

6. Bushing & journal extend infinitely in the z direction; no lubricant flow in the z direction.

7. Film pressure is constant in the y direction. Pressure depends only on the coordinate x.

8. Velocity of any particle of lubricant in the film depends only on coordinates x & y.
12–6 Hydrodynamic Theory

- Select an element of lubricant in the film (Fig. 12–9a) of dimensions $dx$, $dy$, and $dz$
- Compute the forces that act on sides of this element.
- Normal forces, due to pressure, act upon right & left sides of element
- Shear forces, due to viscosity and to the velocity, act upon top & bottom sides
12–6 Hydrodynamic Theory

\[ \sum F_x = p \, dy \, dz - \left( p + \frac{dp}{dx} \right) dy \, dz - \tau \, dx \, dz + \left( \tau + \frac{\partial \tau}{\partial y} \right) dx \, dz = 0 \]

\[ \frac{dp}{dx} = \frac{\partial \tau}{\partial y} \quad \tau = \mu \frac{\partial u}{\partial y} \quad \frac{dp}{dx} = \mu \frac{\partial^2 u}{\partial y^2} \]

\[ \frac{\partial u}{\partial y} = \frac{1}{\mu \, dx} \frac{dp}{dy} y + C_1 \]

\[ u = \frac{1}{2\mu} \frac{dp}{dx} y^2 + C_1 y + C_2 \]

At \ y = 0, \ u = 0

At \ y = h, \ u = U

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12–6 Hydrodynamic Theory

- $C_1$ and $C_2$ can be functions of $x$

$$C_1 = \frac{U}{h} - \frac{h}{2\mu} \frac{dp}{dx} \quad C_2 = 0$$

$$u = \frac{1}{2\mu} \frac{dp}{dx}(y^2 - hy) + \frac{U}{h} y$$  \hspace{1cm} (12–9)

- The velocity distribution across the film (from $y = 0$ to $y = h$) is obtained by superposing a parabolic distribution onto a linear distribution.
12–6 Hydrodynamic Theory

When the pressure is maximum, $dp/dx = 0$ and the velocity is

$$u = \frac{U}{h} y$$
12–6 Hydrodynamic Theory

Define $Q$ as volume of lubricant flowing in the $x$ direction per unit time.

By using a width of unity in the $z$ direction,

\[ Q = \int_{0}^{h} u \ dy \]

Substituting the value of $u$ from Eq. (12–9) and integrating gives

\[ Q = \frac{U h}{2} - \frac{h^3}{12\mu} \frac{dp}{dx} \]
12–6 Hydrodynamic Theory

- Using the assumption of an incompressible lubricant and that flow is the same for any cross section. Thus

\[
\frac{dQ}{dx} = 0 \quad \frac{dQ}{dx} = \frac{U}{2} \frac{dh}{dx} - \frac{d}{dx} \left( \frac{h^3}{12\mu} \frac{dp}{dx} \right) = 0
\]

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

- Reynolds equation for 1-D flow.
- It neglects side leakage, flow in z direction
A similar development is used when side leakage is not neglected. The resulting equation is

\[
\frac{\partial}{\partial x} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U \frac{\partial h}{\partial x}
\]  (12–11)
12–6 Hydrodynamic Theory

- No general analytical solution to Eq. (12–11). One of the important solutions is due to Sommerfeld and may be expressed in the form

\[
\frac{r}{c} f = \phi \left[ \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} \right]
\]

(12–12)

- Sommerfeld found the functions for half bearings and full bearings by using the assumption of no side leakage.
12–7 Design Considerations

The variables involved in the design of sliding bearings may be divided in two groups:

1. Independent (design) variables
2. Dependent variables
12–7 Design Considerations

Independent \((design)\) variables

- controlled directly by the designer which include:
  1. viscosity “\(\mu\)”
  2. load per unit projected area “\(p\)”
  3. angular speed “\(N\)”
  4. bearing dimensions: \(r, c, l, \beta\)
Design Considerations

Dependent variables

- controlled indirectly by changing one or more of the design variables, Includes:
  1. coefficient of friction “f”
  2. temperature rise “ΔT”
  3. oil flow rate “Q”
  4. minimum film thickness “h₀”

Performance factors

- The designer may impose limitations on those variables to ensure satisfactory performance
12–7 Design Considerations

**Significant Angular Speed**

\[ N = |N_j + N_b - 2N_f| \]  \hspace{1cm} (12–13)

- \( N_j \) = journal angular speed, rev/s
- \( N_b \) = bearing angular speed, rev/s
- \( N_f \) = load vector angular speed, rev/s
12–7 Design Considerations

**Significant Angular Speed**

\[ N = \left| N_j + N_b - 2N_f \right| \]  \hspace{1cm} (12–13)

- \( N_b = 0, \ N_f = 0 \)
  \[ N = \left| N_j + 0 - 2(0) \right| = N_j \]
- \( N_b = 0, \ N_f = N_j \)
  \[ N = \left| N_j + 0 - 2N_j \right| = N_j \]
- \( N_b = 0, \ N_f = \frac{N_j}{2} \)
  \[ N = \left| N_j + 0 - 2N_j/2 \right| = 0 \]
- \( N_b = N_j, \ N_f = 0 \)
  \[ N = \left| N_j + N_j - 2(0) \right| = 2N_j \]
12–7 Design Considerations

**Trumpler’s Design Criterion**

**Minimum film thickness**

- When bearing starts rotation some debris are generated because of metal to metal contact and it moves with the lubricant.
- Min film thickness is kept thick enough such that debris can pass and will not block lubricant flow.

\[ h_0 \geq 0.00508 + 0.00004d \quad \text{mm} \]
12–7 Design Considerations

Trumpler’s Design Criterion

Maximum lubricant temperature

- When temperature increases beyond a certain limit, lighter components of lubricant starts to evaporate which increases viscosity and thus friction.

\[ T_{max} \leq 121 \, ^\circ C \]
12–7 Design Considerations

Trumpler’s Design Criterion

Starting load

- Journal bearing usually consist of a steel journal and a bushing of softer material.
- If starting load is high, the bushing will be damaged because of metal to metal contact.
- Starting load is usually smaller than running load

\[
\frac{W_{st}}{ld} \leq 2068 \quad \text{kPa}
\]
12–7 Design Considerations

Trumpler’s Design Criterion

Running load design factor

- To account for external vibrations, a design factor is to be used;

\[ n_d \geq 2 \]

- For running load not starting load
12–8 The Relations of the Variables

- The charts are for full journal bearings ($\beta = 360^\circ$) only.
- **Viscosity Charts (Figs. 12–12 to 12–14)**
- viscosity of lubricant is constant as it passes through the bearing.
- Temperature of oil is higher when it leaves the loading zone than it was on entry.
- Viscosity drops off significantly with a rise in temperature.
- Viscosity is determined at the average of inlet & outlet temperatures,

$$T_{av} = T_1 + \frac{\Delta T}{2}$$
12–8 The Relations of the Variables

- **Viscosity Charts**
- **Figure 12–12**

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12–8 The Relations of the Variables

- Viscosity Charts
- Figure 12–13

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12-8 The Relations of the Variables

- **Viscosity Charts**
- **Figure 12-14:** Chart for multi-viscosity lubricants. This chart was derived from known viscosities at two points, 100 and 210°F, and the results are believed to be correct for other temperatures.

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12–8 The Relations of the Variables

- **Viscosity Charts**

- **Table 12–1**: Curve Fits* to Approximate Viscosity versus Temperature Functions for SAE Grades 10 to 60

<table>
<thead>
<tr>
<th>Oil Grade, SAE</th>
<th>Viscosity ( \mu_0 ), reyn</th>
<th>Constant ( b ), °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.0158(10^{-6})</td>
<td>1157.5</td>
</tr>
<tr>
<td>20</td>
<td>0.0136(10^{-6})</td>
<td>1271.6</td>
</tr>
<tr>
<td>30</td>
<td>0.0141(10^{-6})</td>
<td>1360.0</td>
</tr>
<tr>
<td>40</td>
<td>0.0121(10^{-6})</td>
<td>1474.4</td>
</tr>
<tr>
<td>50</td>
<td>0.0170(10^{-6})</td>
<td>1509.6</td>
</tr>
<tr>
<td>60</td>
<td>0.0187(10^{-6})</td>
<td>1564.0</td>
</tr>
</tbody>
</table>

\[ \mu = \mu_0 \exp \left[ \frac{b}{(T + 95)} \right], \ T \text{ in °F} \]
12–8 The Relations of the Variables

Figure 12–15: Polar diagram of film–pressure distribution showing the notation used. (*Raimondi and Boyd.*)
12–8 The Relations of the Variables

Figure 12–16: min film thickness variable & eccentricity ratio
12–8 The Relations of the Variables

Figure 12–17: position of min film thickness $h_0$
The Relations of the Variables

Figure 12–18: coefficient of friction variable

Coefficient-of-friction variable \( \frac{f}{f_c} \) (dimensionless)

Bearing characteristic number, \( S = \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} \)
12–8 The Relations of the Variables

Figure 12–19: flow variable. Note: Not for pressure-fed bearings.
12–8 The Relations of the Variables

Figure 12–20: ratio of side flow to total flow

Flow ratio $\frac{Q_s}{Q_t}$

Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$

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12–8 The Relations of the Variables

**Figure 12–21:** max film pressure.

*Note:* Not for pressure-fed bearings.
12–8 The Relations of the Variables

Figure 12–22: Terminating position of lubricant film & position of max film pressure
EXAMPLE 12–1

Determine $h_0$ and $e$ using the following given parameters: $\mu = 4$ $\mu$reyn, $N = 30$ rev/s, $W = 500$ lbf (bearing load), $r = 0.75$ in, $c = 0.0015$ in, and $l = 1.5$ in.

\[
P = \frac{W}{2rl} = \frac{500}{2(0.75)1.5} = 222 \text{ psi}
\]

\[
S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N}{P}\right) = \left(\frac{0.75}{0.0015}\right)^2 \left[\frac{4(10^{-6})30}{222}\right] = 0.135
\]

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EXAMPLE 12–2

Using the parameters given in Ex. 12–1, determine the coefficient of friction, the torque to overcome friction, and the power loss to friction.

We enter Fig. 12–18 with \( S = 0.135 \) and \( l/d = 1 \), find \( (r/c) f = 3.50 \).

\[
f = 3.50 \frac{c}{r} = 3.50\left(\frac{0.0015}{0.75}\right) = 0.0070
\]

\[
T = fWr = 0.007(500)0.75 = 2.62 \text{ lbf} \cdot \text{in}
\]

\[
(hp)_{\text{loss}} = \frac{T N}{1050} = \frac{2.62(30)}{1050} = 0.075 \text{ hp}
\]
EXAMPLE 12–3

Continuing with the parameters of Ex. 12–1, determine the total volumetric flow rate \( Q \) and the side flow rate \( Q_s \).

✓ Fig. 12–19: \( S = 0.135 \) and \( l/d = 1 \) to obtain \( Q/(rcNI) = 4.28 \).

✓ Total volumetric flow rate \( Q = 4.28rcNI = 4.28(0.75)0.0015(30)1.5 = 0.217 \) in\(^3\)/s

✓ Fig. 12–20: \( Q_s/Q = 0.655 \)

✓ \( Q_s = 0.655Q = 0.655(0.217) = 0.142 \) in\(^3\)/s
EXAMPLE 12–4

Using the parameters given in Ex. 12–1, determine the maximum film pressure and the locations of the maximum and terminating pressures.

Solution

Fig. 12–21: with $S = 0.135$ and $l/d = 1$, we find $P/p_{\text{max}} = 0.42$.

\[
p_{\text{max}} = \frac{P}{0.42} = \frac{222}{0.42} = 529 \text{ psi}
\]

With $S = 0.135$ and $l/d = 1$, from Fig. 12–22, $\theta_{p_{\text{max}}} = 18.5^\circ$ and the terminating position $\theta_{p_0}$ is $75^\circ$.  

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Lubricant Temperature Rise

Lubricant temperature will increase until a heat balance is reached (heat generated by shearing the lubricant = heat lost to surroundings)
12–8 The Relations of the Variables

**Lubricant Temperature Rise**

\[
\frac{9.70 \Delta T_F}{P_{\text{psi}}} = \frac{rf/c}{(1 - \frac{1}{2} Q_s/Q) [Q/(rcN_jl)]}
\]

\[
\frac{0.12 \Delta T_c}{P_{(\text{MPa})}} = \frac{r/cf}{\left(1 - \frac{1}{2} \left[Q_s/Q \right] \right) \left( Q/rcN_jl \right)}
\]

(12–15)
12–8 The Relations of the Variables

**Lubricant Temperature Rise**

\[
\frac{9.70 \Delta T_E}{P_{\text{psig}}} \quad \text{or} \quad \frac{0.120 \Delta T_C}{P_{\text{MPa}}}
\]

\[
\begin{array}{c|c}
 l/d & \frac{9.70 \Delta T_E}{P_{\text{psig}}} \text{ or } \frac{0.120 \Delta T_C}{P_{\text{MPa}}} \\
 \hline
 1 & 0.349109 + 6.009408S + 0.047467S^2 \\
 1/2 & 0.304552 + 6.392527S - 0.036013S^2 \\
 1/4 & 0.938828 + 6.437512S - 0.011048S^2 \\
\end{array}
\]

\[
\frac{L}{d} = \frac{1}{4}
\]

\[
\frac{L}{d} = \frac{1}{2}
\]

\[
\frac{L}{d} = 1
\]

**Figure 12–24:**

Figures 12–18, 12–19, & 12–20 combined in one
12–8 The Relations of the Variables

Interpolation

For \( l/d \) ratios other than the ones given in the charts, Raimondi and Boyd have provided the following interpolation equation

\[
y = \frac{1}{(l/d)^3} \left[ -\frac{1}{8} \left( 1 - \frac{l}{d} \right) \left( 1 - 2 \frac{l}{d} \right) \left( 1 - 4 \frac{l}{d} \right) y_\infty + \frac{1}{3} \left( 1 - 2 \frac{l}{d} \right) \left( 1 - 4 \frac{l}{d} \right) y_1 \\
- \frac{1}{4} \left( 1 - \frac{l}{d} \right) \left( 1 - 4 \frac{l}{d} \right) y_{1/2} + \frac{1}{24} \left( 1 - \frac{l}{d} \right) \left( 1 - 2 \frac{l}{d} \right) y_{1/4} \right]
\]

(12–16)
12–9 Steady-State Conditions in Self-Contained Bearings

- The lubricant stays within the bearing housing and it is cooled within the housing by dissipating the heat to the surroundings.

- Called *pillow-block* bearings.

- As the oil film exits the lower half of the bearing it mixes with sump contents, then heat is transferred to the surroundings.

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12–9 Steady-State Conditions in Self-Contained Bearings

- The heat lost from housing to surroundings can be estimated as:

\[
H_{loss} = h_{CR} A (T_b - T_\infty)
\]

- \(H_{loss}\) = Dissipated heat, J/s or W
- \(h_{CR}\) = Combined coefficient of radiation & convection, W/(m\(^2\) °C)
- \(A\) = Bearing surface area, m\(^2\)
- \(T_b, T_\infty\) = Housing surface temperature & ambient temp, °C
12–9 Steady-State Conditions in Self-Contained Bearings

\[ \dot{h}_{CR} = \begin{cases} 
11.4 \ W/(m^2.\ ^{\circ}C) & \text{for still air} \\
15.3 \ W/(m^2.\ ^{\circ}C) & \text{for shaft – stirred air} \\
33.5 \ W/(m^2.\ ^{\circ}C) & \text{for air moving at 25.4 m/s} 
\end{cases} \]
12–9 Steady-State Conditions in Self-Contained Bearings

- $\bar{T}_f = \text{average film temperature between inlet } T_s \text{ & outlet } (T_s + \Delta T)$,
  \[ \bar{T}_f = T_s + \Delta T/2 \]
  \[ \bar{T}_f - T_b = \alpha (T_b - T_\infty) \quad T_b = \bar{T}_f + \alpha T_\infty/(1 + \alpha) \]

- Table 12-2: representative values of $\alpha$

<table>
<thead>
<tr>
<th>Lubrication System</th>
<th>Conditions</th>
<th>Range of $\alpha$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil ring</td>
<td>Moving air</td>
<td>1–2</td>
</tr>
<tr>
<td></td>
<td>Still air</td>
<td>$\frac{1}{2}$–1</td>
</tr>
<tr>
<td>Oil bath</td>
<td>Moving air</td>
<td>$\frac{1}{2}$–1</td>
</tr>
<tr>
<td></td>
<td>Still air</td>
<td>$\frac{1}{5}$–$\frac{2}{5}$</td>
</tr>
</tbody>
</table>

$H_{loss} = \frac{h_{CR} A}{1 + \alpha} (\bar{T}_f - T_\infty)$
12–9 Steady-State Conditions in Self-Contained Bearings

- Because of the shearing of lubricant film heat is generated.
- In steady-state condition,
  - heat generated in lubricant film = heat dissipated from housing to surrounding

\[
H_{gen} = \frac{248\mu N^2 lr^3}{c}
\]

\[
\bar{T}_f = T_\infty + 248(1 + \alpha) \frac{\mu N^2 lr^3}{h_{CR} Ac}
\]
EXAMPLE 12–5

Consider a pillow-block bearing with a keyway sump, whose journal rotates at **900 rpm** in **shaft-stirred air** at **70°F** with \( \alpha = 1 \). The lateral area of the bearing is **40 in\(^2\)**. The lubricant is **SAE grade 20** oil. The gravity radial load is **100 lbf** and the \( l/d \) ratio is **unity**. The bearing has a journal diameter of **2.000 + 0.000/−0.002** in, a bushing bore of **2.002 + 0.004/−0.000** in. For a minimum clearance assembly estimate the steady-state temperatures as well as the minimum film thickness and coefficient of friction.
EXAMPLE 12–5

Solution:

The friction horsepower loss, \((\text{hp})_f\), is found as follows:

\[
(\text{hp})_f = \frac{fr}{1050} \left( \frac{W}{1050} \frac{N}{c} \right) = \frac{fr}{1050} \left( \frac{100\times900/60}{1050} \right) \times 0.001 \times \frac{fr}{c} = 0.001 \times 429 \times \frac{fr}{c} \text{ hp}
\]

The heat generation rate \(H_{\text{gen}}\), in Btu/h, is

\[
H_{\text{gen}} = 2545(\text{hp})_f = 2545(0.001 \times 429) \times \frac{fr}{c} = 3.637 \times \frac{fr}{c} \text{ Btu/h}
\]
EXAMPLE 12–5

From Eq. (12–19a) with \( h_{CR} = 2.7 \) Btu/(h \cdot \text{ft}^2 \cdot ^\circ \text{F})\), the rate of heat loss to the environment \( H_{\text{loss}} \) is

\[
H_{\text{loss}} = \frac{h_{CR}A}{\alpha + 1} (\bar{T}_f - 70) = \frac{2.7(40/144)}{(1 + 1)} (\bar{T}_f - 70) = 0.375(\bar{T}_f - 70) \text{ Btu/h}
\]

<table>
<thead>
<tr>
<th>Trial ( \bar{T}_f )</th>
<th>( \mu' )</th>
<th>( S )</th>
<th>( fr/c )</th>
<th>( H_{\text{gen}} )</th>
<th>( H_{\text{loss}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>190</td>
<td>1.15</td>
<td>0.69</td>
<td>13.6</td>
<td>49.5</td>
<td>45.0</td>
</tr>
<tr>
<td>195</td>
<td>1.03</td>
<td>0.62</td>
<td>12.2</td>
<td>44.4</td>
<td>46.9</td>
</tr>
</tbody>
</table>

Temperature at which \( H_{\text{gen}} = H_{\text{loss}} = 46.3 \text{ Btu/h} \) is 193.4°F.

Rounding \( \bar{T}_f \) to 193°F we find \( \mu' = 1.08 \mu\text{reyn} \) and \( S = 0.6(1.08) = 0.65 \).

From Fig. 12–24, \( 9.70 \Delta T_f/P = 4.25 \) °F/psi and thus
EXAMPLE 12–5

\[ \Delta T_F = \frac{4.25P}{9.70} = \frac{4.25(25)}{9.70} = 11.0^\circ F \]

\[ T_1 = T_s = \bar{T}_f - \Delta T/2 = 193 - 11/2 = 187.5^\circ F \]

\[ T_{\text{max}} = T_1 + \Delta T_F = 187.5 + 11 = 198.5^\circ F \]

\[ T_b = \frac{T_f + \alpha T_{\infty}}{1 + \alpha} = \frac{193 + (1)70}{1 + 1} = 131.5^\circ F \]

with \( S = 0.65 \), minimum film thickness from Fig. 12–16 is

\[ h_0 = \frac{h_0}{c} = 0.79(0.001) = 0.00079 \text{ in} \]

The coefficient of friction from Fig. 12–18 is

\[ f = \frac{f r c}{c r} = 12.8 \frac{0.001}{1} = 0.0128 \]

The parasitic friction torque \( T \) is

\[ T = f W r = 0.0128 \times 100 \times 1 = 1.28 \text{ lbf \cdot in} \]
12.10 Clearance

- **Figure 12–25**: effect of wide range of clearances on performance of a bearing (examples 12-1 to 12-4).

- Lubricant flow increases with increased clearance & this decreases generated heat and outlet temperature.
12.10 Clearance

- Minimum film thickness “h₀” increases with clearance then it starts to decrease.
- If clearance is too small, dirt (debris) may block oil flow and therefore cause overheating and failure.
- If clearance is too high, bearing becomes noisy and h₀ decreases.
- Optimum range of clearances is shown by shaded area in the figure.
- If clearance value is within this range the performance of bearing will improve with wear.
12–11 Pressure-Fed Bearings

- Load carrying capacity of self-contained bearings is limited because of limited heat-dissipating capability.
- To increase heat-dissipation, an external pump is used to increase lubricant flow through the bearing.

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The pump supplies bearing with lubricant of high pressure, therefore increasing lubricant flow and heat dissipation.

Figure 12–31
12–11 Pressure-Fed Bearings

- A circumferential groove at the center of bearing, with an oil-supply hole located opposite to the load zone, is usually used to feed the lubricant.

Figure 12–27
12–11 Pressure-Fed Bearings

Figure 12–28
12–11 Pressure-Fed Bearings

- When determining lubricant flow, eccentricity is first neglected, then a correction factor for eccentricity is applied.
- Also, rotation of shaft is neglected.
12–11 Pressure-Fed Bearings

- For lubricant supplied at pressure “$P_s$”, average velocity of lubricant flowing towards ends of bearing can be found as:

$$u_{av} = \frac{P_s}{12\mu l'} (c - e \cos \theta)^2$$

- $l'$ = length of each half of bearing
12–11 Pressure-Fed Bearings

- Minimum film thinness “\( h_0 \)” is assumed to be at the bottom of bearing (fig. 12-30) because rotation is neglected.

- Lubricant flow out of both ends of the bearing:

\[
Q_s = \frac{\pi P_s r c^3}{3 \mu l'} (1 + 1.5 \epsilon^2)
\]

- Pressure per projected area for each half of bearing:

\[
P = \frac{w/2}{2rl'} = \frac{w}{4rl'}
\]

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12–11 Pressure-Fed Bearings

- **Sommerfeld number;**
  \[
  S = \left( \frac{r}{c} \right)^2 \frac{\mu N}{P} = \left( \frac{r}{c} \right)^2 \frac{4rl'\mu N}{W}
  \]

- The heat generated & heat loss:
  \[
  H_{\text{loss}} = \rho C_p Q_s \Delta T
  \]
  \[
  H_{\text{gen}} = 2\pi WNc \frac{fr}{c}
  \]
  ✓ \( \rho = \text{density} \)
  ✓ \( C_p = \text{heat capacity} \)
12–11 Pressure-Fed Bearings

Equating $H_{\text{loss}}$ & $H_{\text{gen}}$, the temperature rise can be found as:

$$\Delta T_c = \frac{978 \times 10^6 \left( fr/c \right) SW^2}{1 + 1.5 \epsilon^2 \frac{P_s r^4}{W}}$$

- W = KN
- $P_s$ = Kpa
- $r$ = mm
EXAMPLE 12–6

A circumferential-groove pressure-fed bearing is lubricated with SAE grade 20 oil supplied at a gauge pressure of 30 psi. The journal diameter \( d_j \) is 1.750 in, with a unilateral tolerance of −0.002 in. The central circumferential bushing has a diameter \( d_b \) of 1.753 in, with a unilateral tolerance of +0.004 in. The \( l'/d \) ratio of the two “half Bearings” that constitute the complete pressure-fed bearing is 1/2. The journal angular speed is 3000 rpm (50 rev/s), and the radial steady load is 900 lbf. The external sump is maintained at 120°F as long as the necessary heat transfer does not exceed 800 Btu/h.

a. Find the steady-state average film temperature.

b. Compare \( h_0, T_{\text{max}}, \) and \( P_{\text{st}} \) with the Trumpler criteria.

c. Estimate the volumetric side flow \( Q_s \), the heat loss rate \( H_{\text{loss}} \), and the parasitic friction torque.

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12–12 Loads & Materials

Since diameter & length of a bearing depend upon unit load, these tables will help the designer to establish a starting point in the design.

<table>
<thead>
<tr>
<th>Application</th>
<th>Unit Load</th>
<th>psi</th>
<th>MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel engines:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>900–1700</td>
<td>6–12</td>
<td></td>
</tr>
<tr>
<td>Crankpin</td>
<td>1150–2300</td>
<td>8–15</td>
<td></td>
</tr>
<tr>
<td>Wristpin</td>
<td>2000–2300</td>
<td>14–15</td>
<td></td>
</tr>
<tr>
<td>Electric motors</td>
<td>120–250</td>
<td>0.8–1.5</td>
<td></td>
</tr>
<tr>
<td>Steam turbines</td>
<td>120–250</td>
<td>0.8–1.5</td>
<td></td>
</tr>
<tr>
<td>Gear reducers</td>
<td>120–250</td>
<td>0.8–1.5</td>
<td></td>
</tr>
<tr>
<td>Automotive engines:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>600–750</td>
<td>4–5</td>
<td></td>
</tr>
<tr>
<td>Crankpin</td>
<td>1700–2300</td>
<td>10–15</td>
<td></td>
</tr>
<tr>
<td>Air compressors:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Main bearings</td>
<td>140–280</td>
<td>1–2</td>
<td></td>
</tr>
<tr>
<td>Crankpin</td>
<td>280–500</td>
<td>2–4</td>
<td></td>
</tr>
<tr>
<td>Centrifugal pumps</td>
<td>100–180</td>
<td>0.6–1.2</td>
<td></td>
</tr>
</tbody>
</table>

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12–12 Loads & Materials

<table>
<thead>
<tr>
<th>Alloy Name</th>
<th>Thickness, in</th>
<th>SAE Number</th>
<th>Clearance Ratio $r/c$</th>
<th>Load Capacity</th>
<th>Corrosion Resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tin-base babbitt</td>
<td>0.022</td>
<td>12</td>
<td>600–1000</td>
<td>1.0</td>
<td>Excellent</td>
</tr>
<tr>
<td>Lead-base babbitt</td>
<td>0.022</td>
<td>15</td>
<td>600–1000</td>
<td>1.2</td>
<td>Very good</td>
</tr>
<tr>
<td>Tin-base babbitt</td>
<td>0.004</td>
<td>12</td>
<td>600–1000</td>
<td>1.5</td>
<td>Excellent</td>
</tr>
<tr>
<td>Lead-base babbitt</td>
<td>0.004</td>
<td>15</td>
<td>600–1000</td>
<td>1.5</td>
<td>Very good</td>
</tr>
<tr>
<td>Leaded bronze</td>
<td>Solid</td>
<td>792</td>
<td>500–1000</td>
<td>3.3</td>
<td>Very good</td>
</tr>
<tr>
<td>Copper-lead</td>
<td>0.022</td>
<td>480</td>
<td>500–1000</td>
<td>1.9</td>
<td>Good</td>
</tr>
<tr>
<td>Aluminum alloy</td>
<td>Solid</td>
<td>400–500</td>
<td></td>
<td>3.0</td>
<td>Excellent</td>
</tr>
<tr>
<td>Silver plus overlay</td>
<td>0.013</td>
<td>17P</td>
<td>600–1000</td>
<td>4.1</td>
<td>Excellent</td>
</tr>
<tr>
<td>Cadmium (1.5% Ni)</td>
<td>0.022</td>
<td>18</td>
<td>400–500</td>
<td>1.3</td>
<td>Good</td>
</tr>
<tr>
<td>Trimetal 88*</td>
<td></td>
<td></td>
<td></td>
<td>4.1</td>
<td>Excellent</td>
</tr>
<tr>
<td>Trimetal 77†</td>
<td></td>
<td></td>
<td></td>
<td>4.1</td>
<td>Very good</td>
</tr>
</tbody>
</table>

*This is a 0.008-in layer of copper-lead on a steel back plus 0.001 in of tin-base babbitt.
†This is a 0.013-in layer of copper-lead on a steel back plus 0.001 in of lead-base babbitt.

Table 12–6: Some Characteristics of Bearing Alloys
12-12 Loads & Materials

- $l/d$ ratio of a bearing depends upon whether it is expected to run under thin-film-lubrication conditions.

- A long bearing (large $l/d$) reduces coefficient of friction & side flow of oil:
  - Desirable where thin-film or boundary-value lubrication is present.
12–12 Loads & Materials

- Where forced-feed or positive lubrication is present, $l/d$ ratio should be relatively small.
  - Short bearing length results in a greater flow of oil out of ends, thus keeping bearing cooler.

- Current practice is to use $l/d$ ratio of about unity,
  - Increase ratio if thin-film lubrication is likely to occur
  - Decrease it for thick-film lubrication or high temperatures.
12–12 Loads & Materials

- If shaft deflection is likely to be severe, a short bearing should be used to prevent metal-to-metal contact at the ends of the bearings.

- You should always consider use of a partial bearing if high temperatures are a problem, because relieving the non-load-bearing area of a bearing can very substantially reduce the heat generated.
12–12 Loads & Materials

- The two conflicting requirements of a good bearing material are that:
  1. it must have a satisfactory compressive & fatigue strength to resist the externally applied loads
  2. it must be soft and have a low melting point and a low modulus of elasticity.
12–12 Loads & Materials

- The 2nd set of requirements is necessary to permit the material to wear or break in, since the material can then conform to slight irregularities and absorb and release foreign particles.

- The resistance to wear and the coefficient of friction are also important because all bearings must operate, at least for part of the time, with thin-film or boundary lubrication.
12–12 Loads & Materials

- Additional considerations in the selection of a good bearing material:
  - its ability to resist corrosion
  - cost of producing the bearing
12–12 Loads & Materials

- Small bushings & thrust collars are often expected to run with thin-film or boundary lubrication.

- A powder-metallurgy bushing is porous and permits the oil to penetrate into the bushing material.

  ✓ Sometimes such a bushing may be enclosed by oil-soaked material to provide additional storage space.

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12–12 Loads & Materials

- Bearings are frequently ball-indentented to provide small basins for the storage of lubricant while the journal is at rest. This supplies some lubrication during starting.

- Another method of reducing friction is to indent the bearing wall and to fill the indentations with graphite.
12–13 Bearing Types

Figure 12–32: Sleeve bushings

(a) Solid bushing

(b) Lined bushing
12–13 Bearing Types

Figure 12–33: Two-piece bushings

(a) Flanged  (b) Straight
12–14 Thrust Bearings

- **Figure 12–35:** Fixed-pad thrust bearing

- Lubricant is brought into the radial grooves & pumped into the wedge-shaped space by the motion of the runner.

- Film lubrication is obtained if:
  - speed of runner is continuous and sufficiently high
  - Lubricant has the correct viscosity
  - It is supplied in sufficient quantity
12–14 Thrust Bearings

Figure 12–35: Pressure distribution of lubricant in a thrust bearing
12–15 Boundary-Lubricated Bearings

- **Boundary lubrication**: When two surfaces slide relative to each other with only a partial lubricant film between them.

- Boundary or thin-film lubrication occurs in hydrodynamically lubricated bearings when:
  - they are starting or stopping
  - the load increases
  - the supply of lubricant decreases
The coefficient of friction for boundary-lubricated surfaces may be greatly decreased by the use of animal or vegetable oils mixed with the mineral oil or grease.
12–15 Boundary-Lubricated Bearings

- When a bearing operates partly under hydrodynamic conditions and partly under dry or thin-film conditions, a *mixed-film lubrication* exists.

- If the lubricant is supplied by hand oiling, by drop or mechanical feed, or by wick feed, the bearing is operating under mixed-film conditions.
Mixed-film conditions may be present when:

- The viscosity is too low
- The bearing speed is too low
- The bearing is overloaded
- The clearance is too tight
- Journal and bearing are not properly aligned
12–15 Boundary-Lubricated Bearings

- Relative motion between surfaces in contact in the presence of a lubricant is called boundary lubrication.
- This condition is present in hydrodynamic film bearings during starting, stopping, overloading, or lubricant deficiency.
- Table 12–7 gives some properties of a range of bushing materials.
# 12–15 Boundary-Lubricated Bearings

## Table 12–7: Some Materials for Boundary-Lubricated Bearings and Their Operating Limits

<table>
<thead>
<tr>
<th>Material</th>
<th>Maximum Load, psi</th>
<th>Maximum Temperature, °F</th>
<th>Maximum Speed, fpm</th>
<th>Maximum PV Value*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast bronze</td>
<td>4 500</td>
<td>325</td>
<td>1 500</td>
<td>50 000</td>
</tr>
<tr>
<td>Porous bronze</td>
<td>4 500</td>
<td>150</td>
<td>1 500</td>
<td>50 000</td>
</tr>
<tr>
<td>Porous iron</td>
<td>8 000</td>
<td>150</td>
<td>800</td>
<td>50 000</td>
</tr>
<tr>
<td>Phenolics</td>
<td>6 000</td>
<td>200</td>
<td>2 500</td>
<td>15 000</td>
</tr>
<tr>
<td>Nylon</td>
<td>1 000</td>
<td>200</td>
<td>1 000</td>
<td>3 000</td>
</tr>
<tr>
<td>Teflon</td>
<td>500</td>
<td>500</td>
<td>100</td>
<td>1 000</td>
</tr>
<tr>
<td>Reinforced Teflon</td>
<td>2 500</td>
<td>500</td>
<td>1 000</td>
<td>10 000</td>
</tr>
<tr>
<td>Teflon fabric</td>
<td>60 000</td>
<td>500</td>
<td>50</td>
<td>25 000</td>
</tr>
<tr>
<td>Delrin</td>
<td>1 000</td>
<td>180</td>
<td>1 000</td>
<td>3 000</td>
</tr>
<tr>
<td>Carbon-graphite</td>
<td>600</td>
<td>750</td>
<td>2 500</td>
<td>15 000</td>
</tr>
<tr>
<td>Rubber</td>
<td>50</td>
<td>150</td>
<td>4 000</td>
<td></td>
</tr>
<tr>
<td>Wood</td>
<td>2 000</td>
<td>150</td>
<td>2 000</td>
<td>15 000</td>
</tr>
</tbody>
</table>

*P = load, psi; V = speed, fpm.
12–15 Boundary-Lubricated Bearings

**Linear sliding wear**

- The thickness of material removed because of wear is given by:

\[ w = f_1 f_2 K P V t \]

- **Table 12-8**: values of K
- **Table 12-10**: \( f_1 \) = correction factor for motion type
- **Table 12-11**: \( f_2 \) = correction factor for environment

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Table 12–8: Wear Factors in U.S. Customary Units*

<table>
<thead>
<tr>
<th>Bushing Material</th>
<th>Wear Factor $K$</th>
<th>Limiting $PV$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oiles 800</td>
<td>$3 \times 10^{-10}$</td>
<td>18 000</td>
</tr>
<tr>
<td>Oiles 500</td>
<td>$0.6 \times 10^{-10}$</td>
<td>46 700</td>
</tr>
<tr>
<td>Polyactal copolymer</td>
<td>$50 \times 10^{-10}$</td>
<td>5 000</td>
</tr>
<tr>
<td>Polyactal homopolymer</td>
<td>$60 \times 10^{-10}$</td>
<td>3 000</td>
</tr>
<tr>
<td>66 nylon</td>
<td>$200 \times 10^{-10}$</td>
<td>2 000</td>
</tr>
<tr>
<td>66 nylon + 15% PTFE</td>
<td>$13 \times 10^{-10}$</td>
<td>7 000</td>
</tr>
<tr>
<td>+ 15% PTFE + 30% glass</td>
<td>$16 \times 10^{-10}$</td>
<td>10 000</td>
</tr>
<tr>
<td>+ 2.5% MoS$_2$</td>
<td>$200 \times 10^{-10}$</td>
<td>2 000</td>
</tr>
<tr>
<td>6 nylon</td>
<td>$200 \times 10^{-10}$</td>
<td>2 000</td>
</tr>
<tr>
<td>Polycarbonate + 15% PTFE</td>
<td>$75 \times 10^{-10}$</td>
<td>7 000</td>
</tr>
<tr>
<td>Sintered bronze</td>
<td>$102 \times 10^{-10}$</td>
<td>8 500</td>
</tr>
<tr>
<td>Phenol + 25% glass fiber</td>
<td>$8 \times 10^{-10}$</td>
<td>11 500</td>
</tr>
</tbody>
</table>

*dim[$K$] = in$^3$ · min/(lbf · ft · h), dim [PV] = psi · ft/min.
# Boundary-Lubricated Bearings

**Table 12–10:** Motion Related Factor $f_1$

<table>
<thead>
<tr>
<th>Mode of Motion</th>
<th>Characteristic Pressure $P$, psi</th>
<th>Velocity $V$, ft/min</th>
<th>$f_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotary</td>
<td></td>
<td>3.3 or less</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–33</td>
<td>1.0–1.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>33–100</td>
<td>1.3–1.8</td>
</tr>
<tr>
<td></td>
<td>720–3600</td>
<td>3.3 or less</td>
<td>1.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–33</td>
<td>1.5–2.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>33–100</td>
<td>2.0–2.7</td>
</tr>
<tr>
<td>Oscillatory</td>
<td></td>
<td>&gt;30°</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–100</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>&lt;30°</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–100</td>
<td>2.0–3.6</td>
</tr>
<tr>
<td></td>
<td>720–3600</td>
<td>&gt;30°</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–100</td>
<td>2.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>&lt;30°</td>
<td>3.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3.3–100</td>
<td>3.0–4.8</td>
</tr>
<tr>
<td>Reciprocating</td>
<td>720 or less</td>
<td>33 or less</td>
<td>1.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>33–100</td>
<td>1.5–3.8</td>
</tr>
<tr>
<td></td>
<td>720–3600</td>
<td>33 or less</td>
<td>2.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>33–100</td>
<td>2.0–7.5</td>
</tr>
</tbody>
</table>

*Values of $f_1$ based on results over an extended period of time on automotive manufacturing machinery.*
# 12–15 Boundary-Lubricated Bearings

## Table 12–11: Environmental Factor $f_2$

<table>
<thead>
<tr>
<th>Ambient Temperature, °F</th>
<th>Foreign Matter</th>
<th>$f_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>140 or lower</td>
<td>No</td>
<td>1.0</td>
</tr>
<tr>
<td>140 or lower</td>
<td>Yes</td>
<td>3.0–6.0</td>
</tr>
<tr>
<td>140–210</td>
<td>No</td>
<td>3.0–6.0</td>
</tr>
<tr>
<td>140–210</td>
<td>Yes</td>
<td>6.0–12.0</td>
</tr>
</tbody>
</table>
12–15 Boundary-Lubricated Bearings

Bushing wear:

- For the case of journal bearing of diameter “D” and Length “L” rotating at speed “N”, the wear of the bushing “w” is given by

\[
w = f_1 f_2 K \frac{4 F \pi DN t}{\pi DL} \frac{12}{12} = \frac{f_1 f_2 KFN t}{3L}
\]  

(12–32)

- In designing a bushing, it is recommended that the length/diameter ratio be in the range

\[
0.5 \leq L/D \leq 2
\]

(12–33)

\[
P_{\max} = \frac{4 F}{\pi DL}
\]

(12–31)

\[
V = \frac{\pi DN}{12}
\]

(12–29)
EXAMPLE 12–7

An Oiles SP 500 alloy brass bushing is 1 in long with a 1-in bore and operates in a clean environment at 70°F. The allowable wear without loss of function is 0.005 in. The radial load is 700 lbf. The peripheral velocity is 33 ft/min. Estimate the number of revolutions for radial wear to be 0.005 in. See Fig. 12–40 and Table 12–12 from the manufacturer.
3rd Exam

On Wednesday 20/11/2013 at 11:00

Tested Material: Chapter 12

Practice Problems: 14, 17, 19, 27, 28, 41