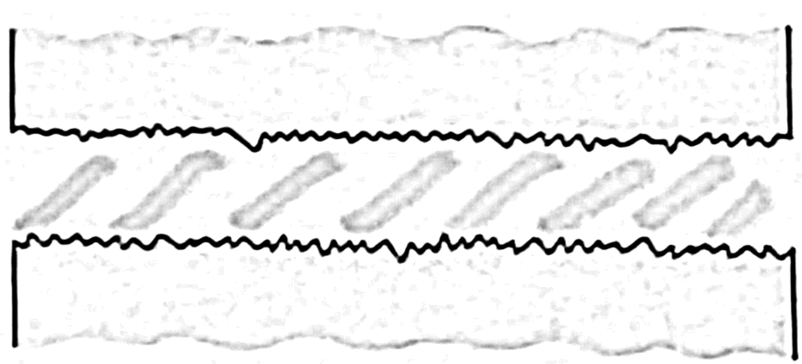
Chapter 12
Lubrication and Journal Bearings

12-1 Types of Lubrication

- **Thick-film (full-film) lubrication:**
  - There is complete separation of contact surfaces by a relatively thick film of lubricant;
  - There is no metal-t-metal contact;
  - Typical minimal film thickness is $0.008 - 0.020\ mm$ or $0.0003 - 0.0008\ in$;
  - Resulting coefficient of friction is $0.002 - 0.01$;



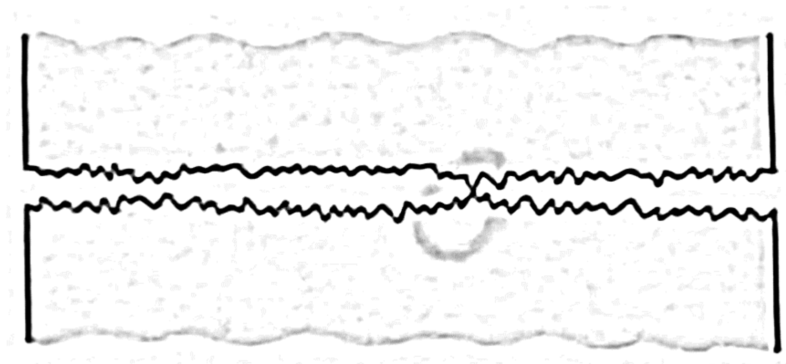
Ways to achieve thick-film lubrication:

- Hydrodynamic lubrication: The journal is lifted by the wedge-action effect of the lubricant; It requires proper lubricant, speed, and clearance; Applicable in situations of high speeds, high

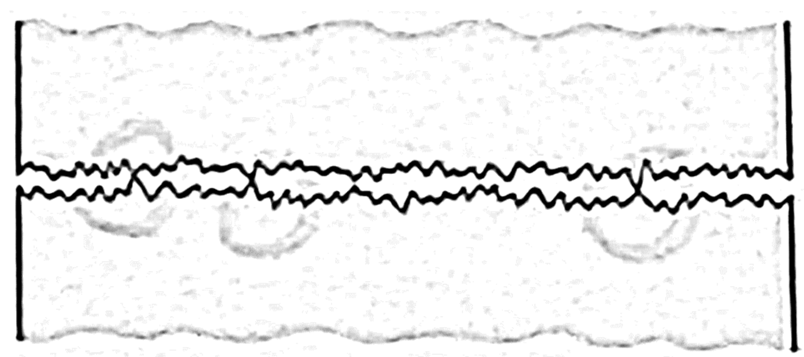
loads, high overloads, and high temperature (e.g., engines, pumps, compressors, turbines, motors, and so on).
    - Hydrostatic lubrication: Lubricant is pressured (force-fed) by external means to separate the two parts; applicable in cases of low speeds and light loads.

- **Elastohydrodynamic lubrication (EHL):**
    - Lubricant is introduced between surfaces that deform elastically;
    - Hard EHL occurs between surfaces in rolling contact, such as mating gears, cams/followers, and rolling-elements and raceways in rolling-contact-bearings;
    - Soft EHL occurs when contact region is relatively large, e.g. brushing.

- **Mixed-film lubrication:**
    - There is only partial full-film lubrication;
    - Surfaces may be in intermittent contact;
    - Coefficient of friction is $0.004 - 0.1$.



- **Boundary lubrication:**
    - It is when 90% or more of surface asperity is in contact;
    - Coefficient of friction is $0.05 - 2$.



- **Solid-film lubrication**
    - Used when bearings must be operated at high to extreme temperatures;
    - Lubricants are of powder form;
    - Graphite, molybdenum disulfide, for example;

Note: above diagrams courtesy of Fundamentals of Machine Components Design, 3rd ed. R.C. Juvinall and K.M. Marshek, Wiley & Sons, 2000.

12-2 Viscosity

- Viscosity is a measure of the internal friction resistance of the fluid.
- Units for absolute or dynamic viscosity μ:
  In ips (inch-pound-second) units: $lb \cdot s/in^2$ or reyn
  In SI units: $N \cdot s/m^2$ or $Pa \cdot s$
- Units used in journal bearing design:
  - In ips: **microreyn ($\mu reyn$)**
    $1\ \mu reyn = 10^{-6}\ reyn$
  - In SI units: **centipoise (cP)**
    $1\ cP = 1\ mPa \cdot s = 10^{-3}\ Pa \cdot s$
- Kinematic viscosity: not used in journal bearing designs

12-3 Petroff's Equation
12-4 Stable Lubrication
12-5 Thick-Film Lubrication
12-6 Hydrodynamic Theory
Available Theories/Solutions of Hydrodynamic Lubrication

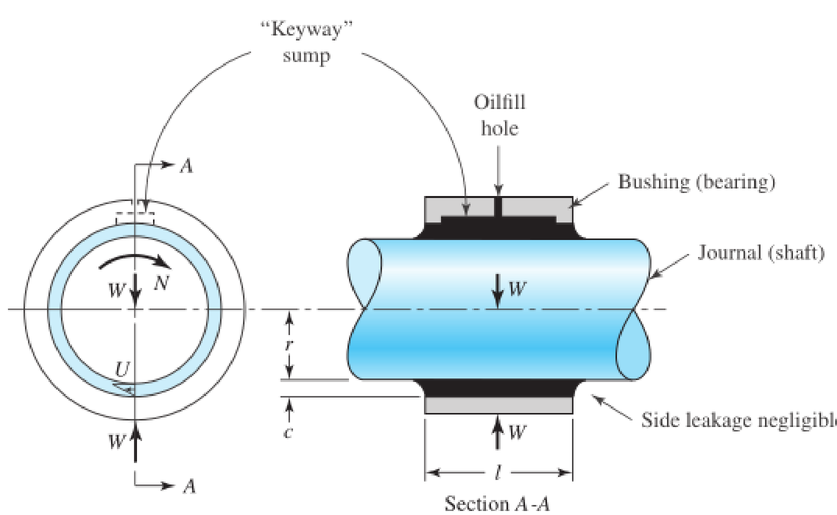
- Petroff's equation for concentric journal bearings (1883)
- Reynolds equation for eccentric journal bearings (1886)
- Sommerfeld's equation for (very) long bearings (1904)
- Ocvirk's solution for (very) short bearings (1952)
- Raimondi & Boyd's numerical solution for finite-length bearings (1958)

Petroff's Equation

- Assume that bearing and shaft are concentric; hence seldom use in design.
- But it was the first to explain the friction phenomenon in bearings.

Figure 12–3

Petroff's lightly loaded journal bearing consisting of a shaft journal and a bushing with an axial-groove internal lubricant reservoir. The linear velocity gradient is shown in the end view. The clearance c is several thousandths of an inch and is grossly exaggerated for presentation purposes.



W: bearing load, in lb or newton
l: bearing length, in inch of m

r: shaft/journal radius, in inch or m
$P = \frac{w}{2rl}$: pressure or unit load, in psi or Pa
N: rotating speed, in rps (rotation per second)
μ: absolute viscosity, in reyn or Pa·s
$\frac{\mu N}{P}$: bearing characteristic
c: radial clearance, in inch or m
$\frac{r}{c}$: radial clearance ratio

Then the coefficient of friction, f, is related to the bearing characteristic $\frac{\mu N}{P}$ and radial clearance ratio $\frac{r}{c}$ by the Petroff's equation:

$$f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c} \quad (12\text{–}6)$$

- Another important parameter is the bearings characteristic number of Sommerfeld number, S

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} \quad (12\text{–}7)$$

In terms of the Sommerfeld number, Petroff's equation becomes:

$$f\frac{r}{c} = 2\pi^2 \frac{\mu N}{P}\left(\frac{r}{c}\right)^2 = 2\pi^2 S \quad (12\text{–}8)$$

Stable Lubrication

Figure 12–4

The variation of the coefficient of friction f with $\mu N/P$.

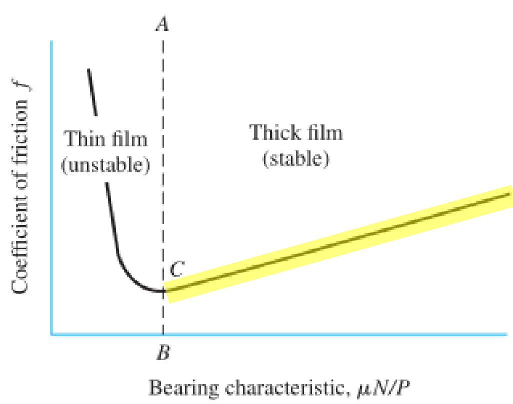


Figure 12-4 is a plot of <u>experimental</u> coefficient of friction in terms of bearing characteristic.

Petroff's equation assumes thick-film lubrication, is presented by the straight line to the right of transition point C. The condition for ensuring thick-film lubrication is $\frac{\mu N}{P} \geq 1.7(10^{-6})$

“Thin film” includes mixed-film and boundary lubrications. Point C represents the transition from metal-to-metal contact to thick film lubrication.

Hydrodynamic Lubrication

- Three aspects or “things” that are required to achieve hydrodynamic lubrication:
  - Relative motion of the surfaces
  - Wedge action
  - A suitable fluid
- Basic assumptions:
  - The lubricant is a Newtonian fluid
  - Inertia forces of the lubricant are negligible
  - The lubricant is an incompressible fluid
  - The viscosity is constant (true if temperature does not change much)
  - There is zero pressure gradient along the length of the bearing (true for very long bearings)
  - The radius of the journal is large compared to the film thickness
- Theory:
  (Eq. 12-10) and (Eq. 12-11) are the Reynolds equation for one-dimensional flow, and two-dimensional flow, respectively.

$$\frac{d}{dx}\left(\frac{h^3}{\mu}\frac{dp}{dx}\right) = 6U\frac{dh}{dx} \qquad (12\text{–}10)$$

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x} \qquad (12\text{–}11)$$

(Eq. 12-12) represents the general form of Sommerfeld solution, which is numerical.

$$\frac{r}{c}f = \phi\left[\left(\frac{r}{c}\right)^2 \frac{\mu N}{P}\right] \qquad (12\text{–}12)$$

Where ϕ represents a function. S is the Sommerfeld number. Sommerfeld found the solution for half and full bearings, under the no side leakage assumption.

12-8 The Relations of the Variable

Between 1951 and 1958, Raimondi and Boyd used the iterative technique to numerically solve the Reynolds equation.

The outcome is represented by the Raimondi-Boyd charts, Figures 12-16 through 12-22, which nomenclatures shown on Figure 12-15. Note that these charts are for full bearings.

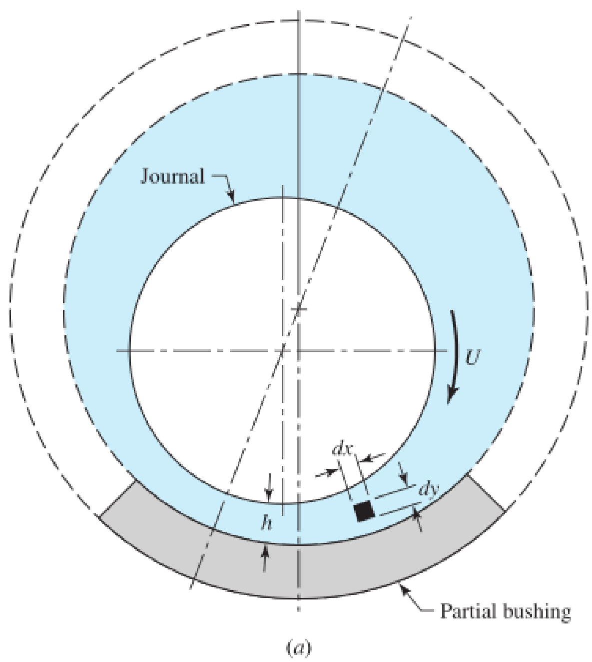
- Sommerfeld number S is the abscissa;
- Four plots in each chart: $\frac{l}{d} = \frac{1}{4}, \frac{1}{2}$, 1, and ∞
- For other $\frac{l}{d}$ ratios, interpolation by (Eq. 12-16)

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

Where $r = \frac{l}{d}$, and $\frac{1}{4} < r < \infty$. y is the desired chart variable, and $y_\infty, y_1, y_{\frac{1}{2}}, y_{\frac{1}{4}}$ are the y values off the corresponding plots.

Or just use this:

$$y = \frac{1}{(l/d)^3}\left[-\frac{1}{8}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_\infty + \frac{1}{3}\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_1 - \frac{1}{4}\left(1-\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_{1/2} + \frac{1}{24}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right)y_{1/4}\right] \quad (12\text{–}16)$$



| Figure 12–9

Figure 12–15

Polar diagram of the film–pressure distribution showing the notation used. *(Raimondi and Boyd.)*

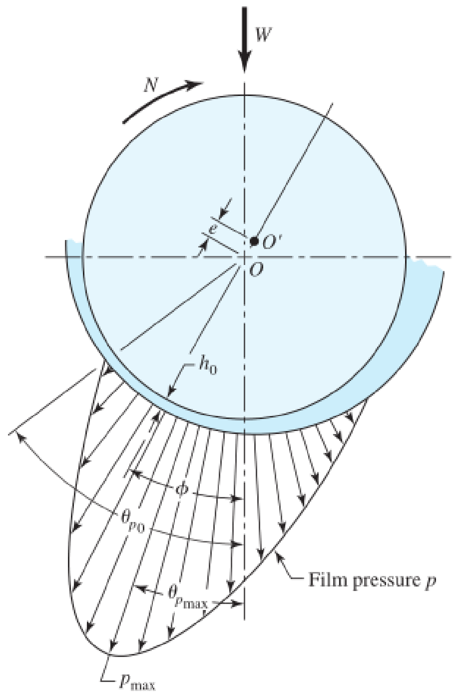
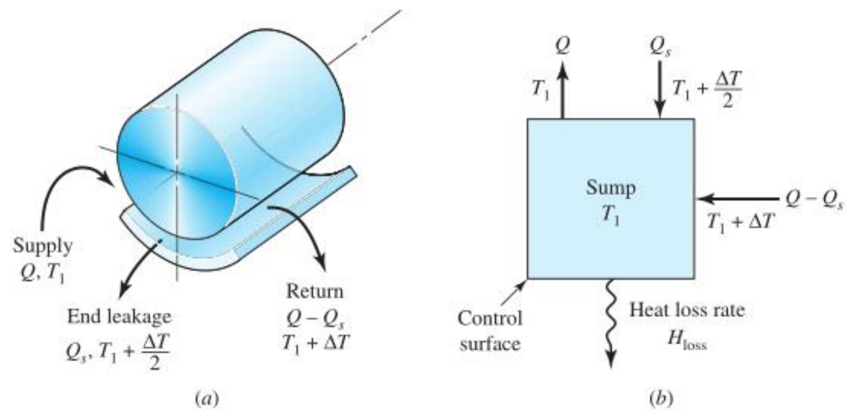


Figure 12–23

Schematic of a journal bearing with an external sump with cooling; lubricant makes one pass before returning to the sump.



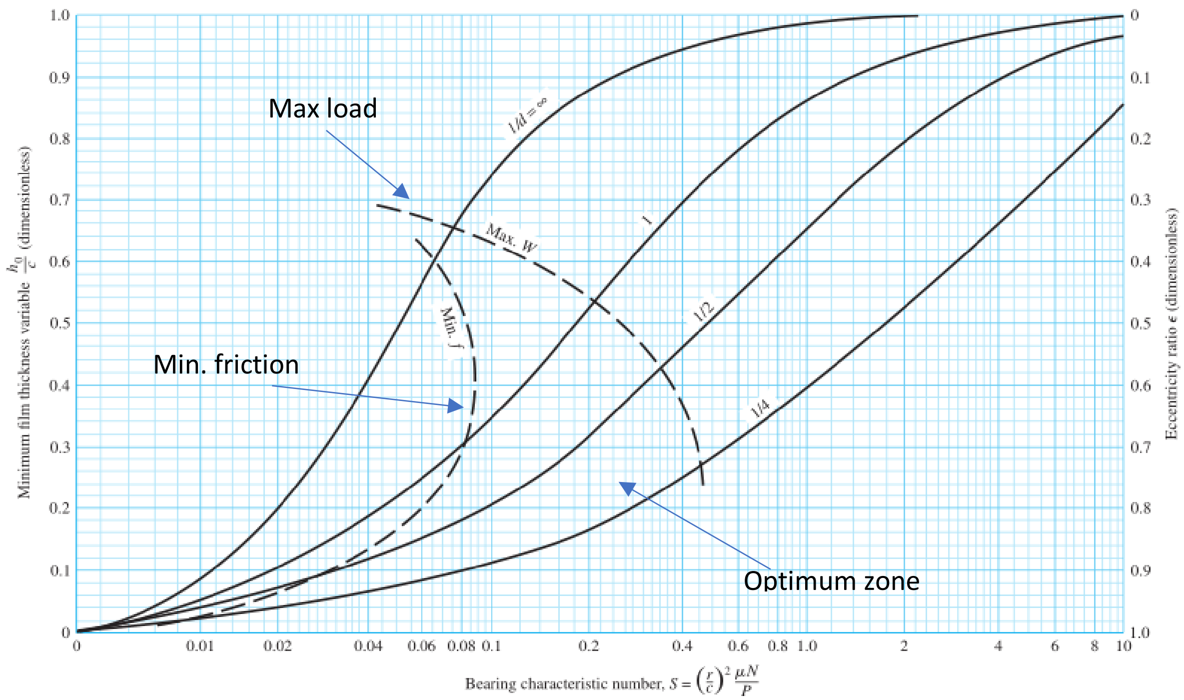


Figure 12–16

Chart for minimum film thickness variable and eccentricity ratio. The left boundary of the zone defines the optimal h_0 for minimum friction; the right boundary is optimum h_0 for load. *(Raimondi and Boyd.)*

Example 1

A journal bearing of 2”-diameter, 2”-length and 0.0015”-radial clearance is to support a steady load of 1000 lv when the shaft rotates at 3000 rpm. The lubricant is SAE 20 oil, supplied at atmospheric pressure. The average temperature of the oil is at 130 $°F$. (1) Estimate h_0, e, Q, Q_s, p_{max}, Φ, θ_{pmax} and θ_{p0}. (2) Estimate the temperature rise ΔT_F.

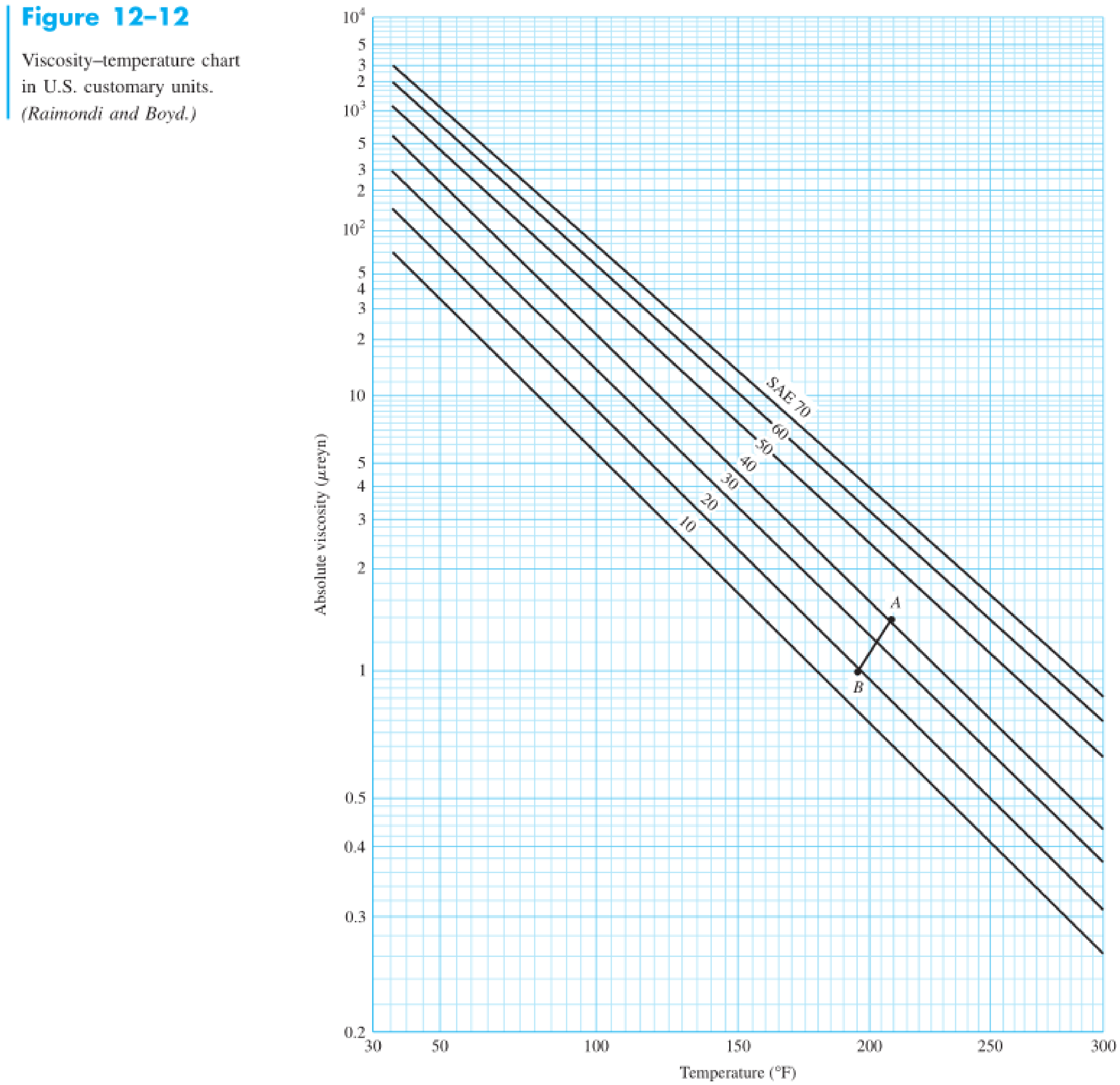
Solution:

Knowns:

$d = 2", l = 2", c = 0.0015', w = 1000\ lb, N = \frac{3000}{60} = 50\ rps$

So, the unit load is $P = \frac{W}{ld} = 1000/(2 * 2) = 250\ psi$

Figure 12-12: $\mu = 3.7\ \mu reyn$



Sommerfeld number (or bearing characteristic number)

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = \left(\frac{1}{0.0015}\right)^2 \frac{(3.7)(10^{-6})(50)}{250} = 0.33$$

(1) Estimate the list of variables

Figure 12-16: minimum film thickness and eccentricity

$h_0/c = 0.65$, $\varepsilon = \frac{e}{c} = 0.35$

So, $h_0 = (0.65)c = 0.000975"$, $e = (0.35)c = 0.000525"$

Note: sum of these two ratios is always unity;

Also, the bearing, as represented by $(S, h_0/c)$ is outside of the optimal zone.

Figure 12-17: position of the minimum film thickness $\Phi = 70°$

Figure 12–17

Chart for determining the position of the minimum film thickness h_0. *(Raimondi and Boyd.)*

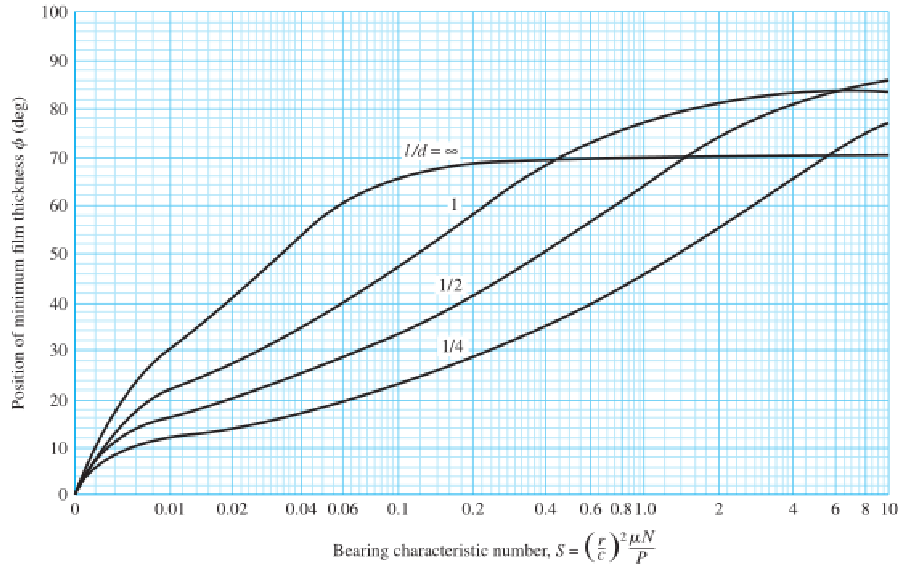


Figure 12-18: coefficient of friction

$$\frac{r}{c} f = 7; so\ f = 7\left(\frac{0.0015}{1}\right) = 0.0105$$

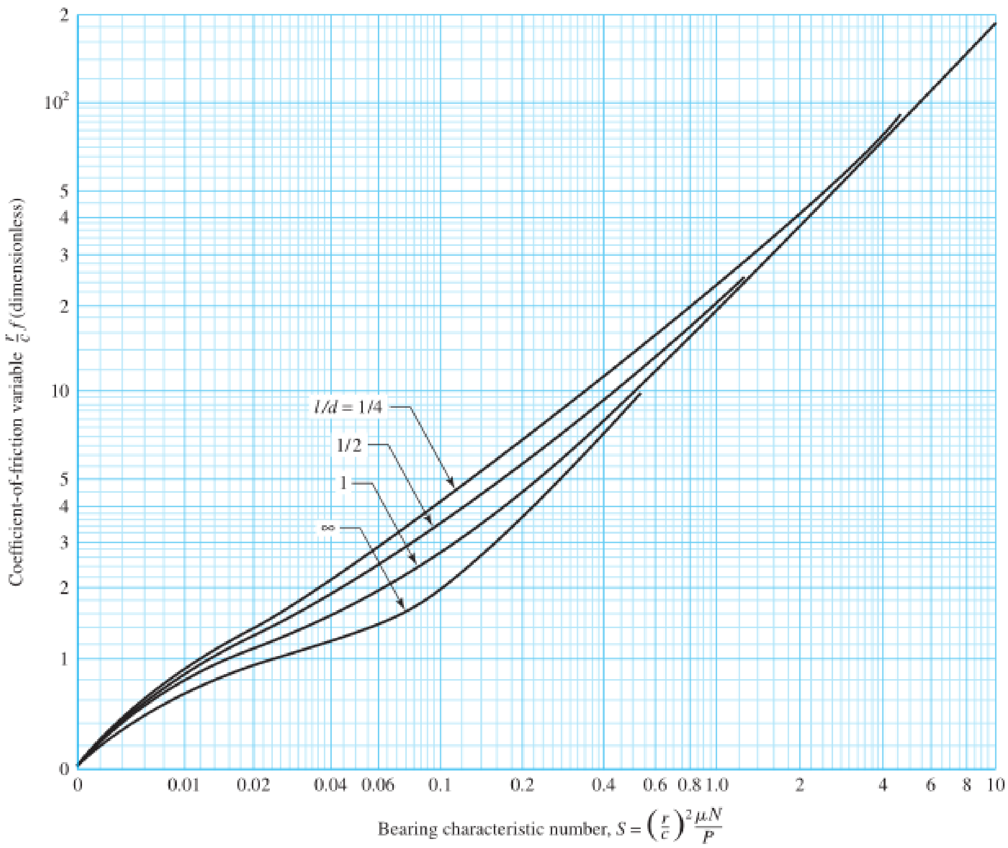


Figure 12–18

Chart for coefficient-of-friction variable; note that Petroff's equation is the asymptote. *(Raimondi and Boyd.)*

Figure 12-19 and 12-20: total flow rate and side flow rate of lubricant.

$\frac{Q}{rcNl} = 3.85$, so $Q = (3.85)(1)(0.0015)(50)(2) = 0.5775\ in^3/s$

$\frac{Q_s}{Q} = 0.45$, so $Q_s = (0.45)(0.5775) = 0.2599\ in^3/s$

Figure 12–19

Chart for flow variable. *Note:* Not for pressure-fed bearings. *(Raimondi and Boyd.)*

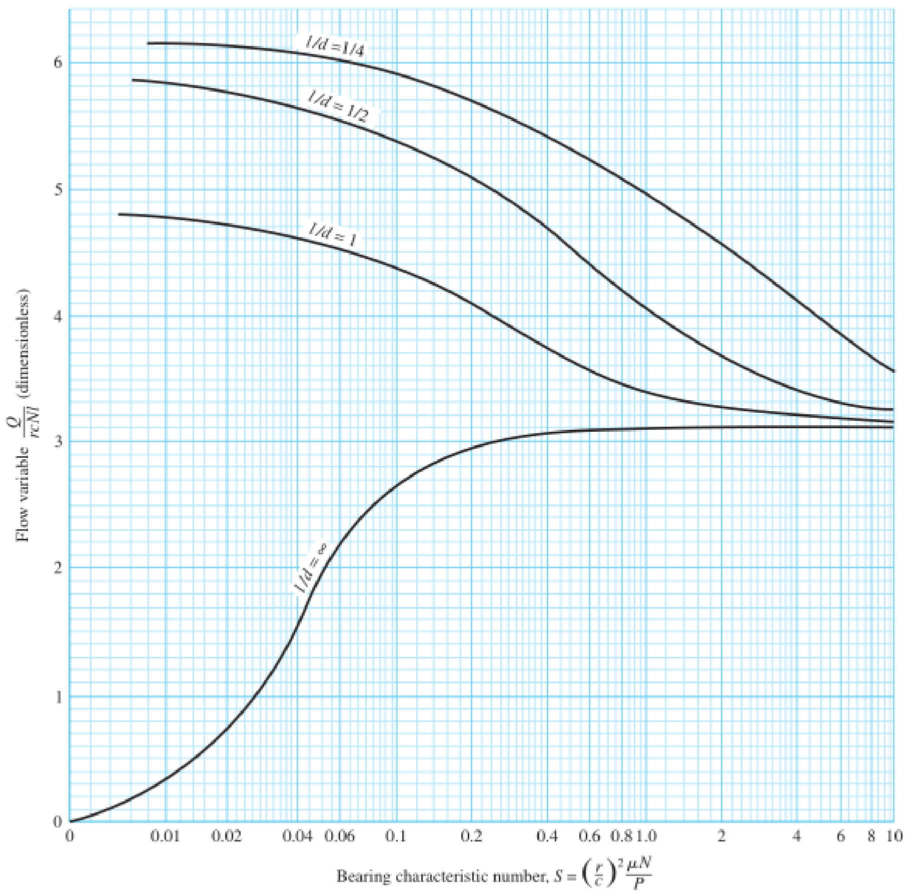
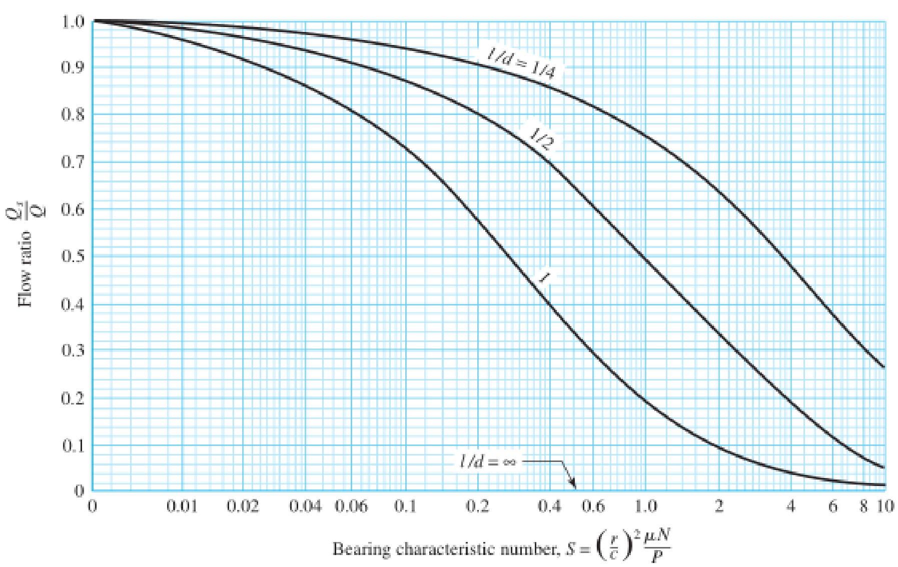


Figure 12–20

Chart for determining the ratio of side flow to total flow. *(Raimondi and Boyd.)*



Figures 12-21 and 12-22: maximum film pressure and its position, and film's termination position:

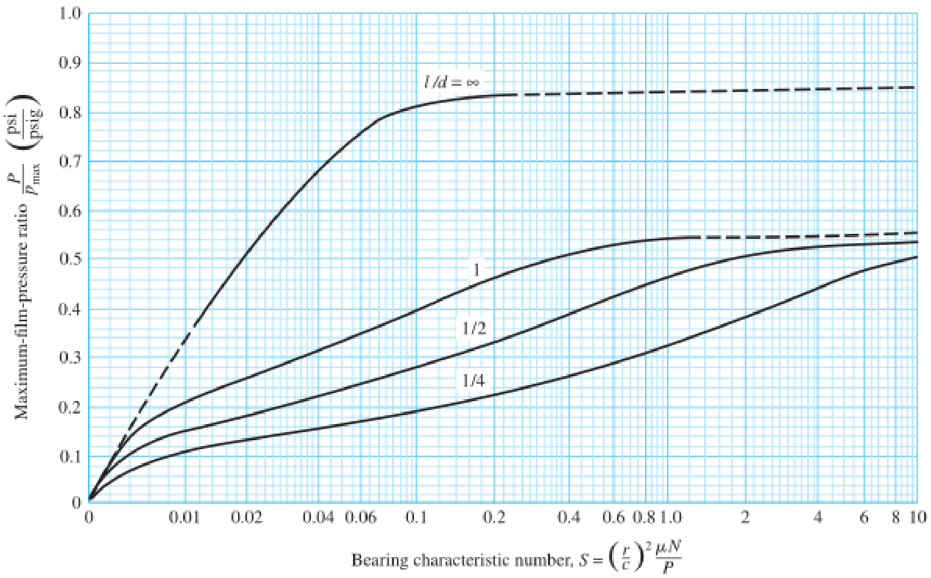
$$\frac{P}{p_{max}} = 0.5, so\ p_{max} = \frac{250}{0.5} = 500\ psi$$

$$\theta_{pmax} = 15°$$

$$\theta_{p0} = 90°$$

Figure 12–21

Chart for determining the maximum film pressure. *Note:* Not for pressure-fed bearings. *(Raimondi and Boyd.)*



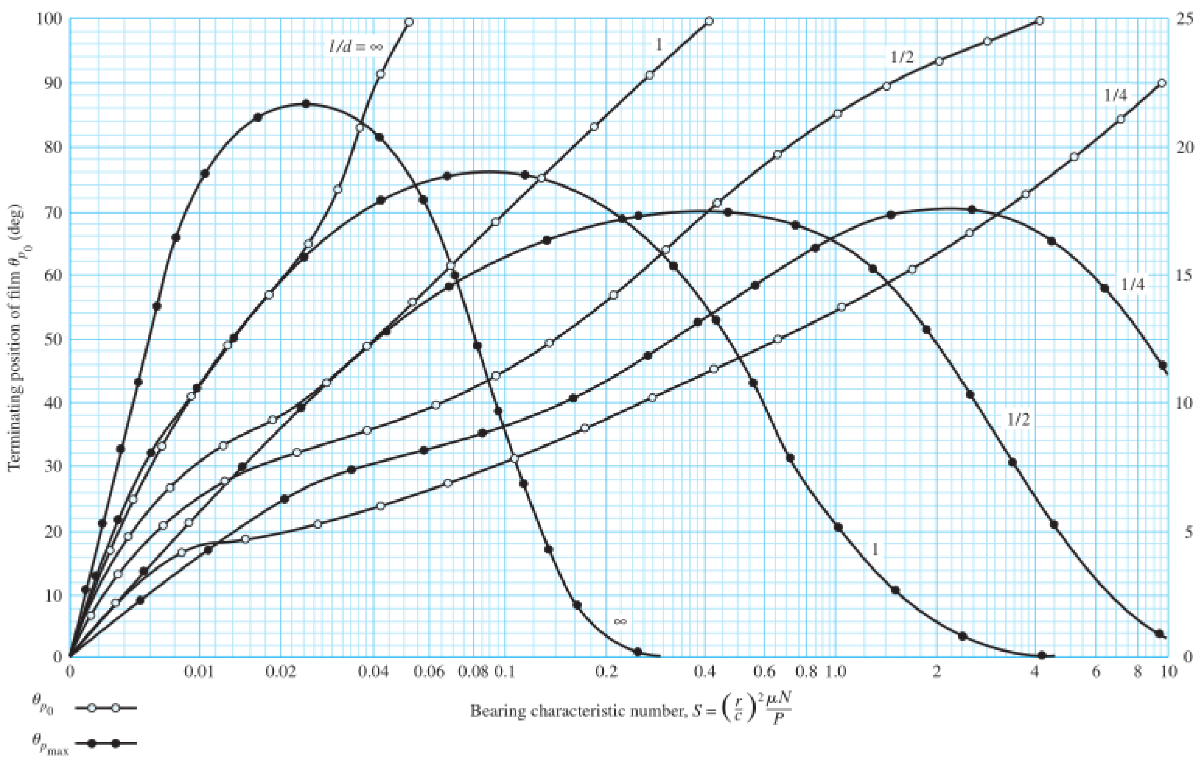


Figure 12–22

Chart for finding the terminating position of the lubricant film and the position of maximum film pressure. *(Raimondi and Boyd.)*

(2) There are two ways to determine temperature rise.

$$\frac{9.70\Delta T_F}{P_{psi}} = \frac{1}{1-0.5\left(\frac{Q_s}{Q}\right)}\frac{\frac{r}{c}f}{\frac{Q}{rcNl}} = \frac{1}{1-(0.5)(0.45)}\frac{7}{3.85} = 2.3460$$

$So, \Delta T_F = (2.3460)(250)/(9.70) = 60.5\ °F$

Or, Figure 12-24 (noting $S = 0.33$)

$$\frac{9.70\Delta T_F}{P_{psi}} = 0.349109 + 6.00940S + 0.047467S^2 = 2.3374$$

So, $\Delta T_F = (2.3374)(250)/(9.70) = 60.2\ °F$

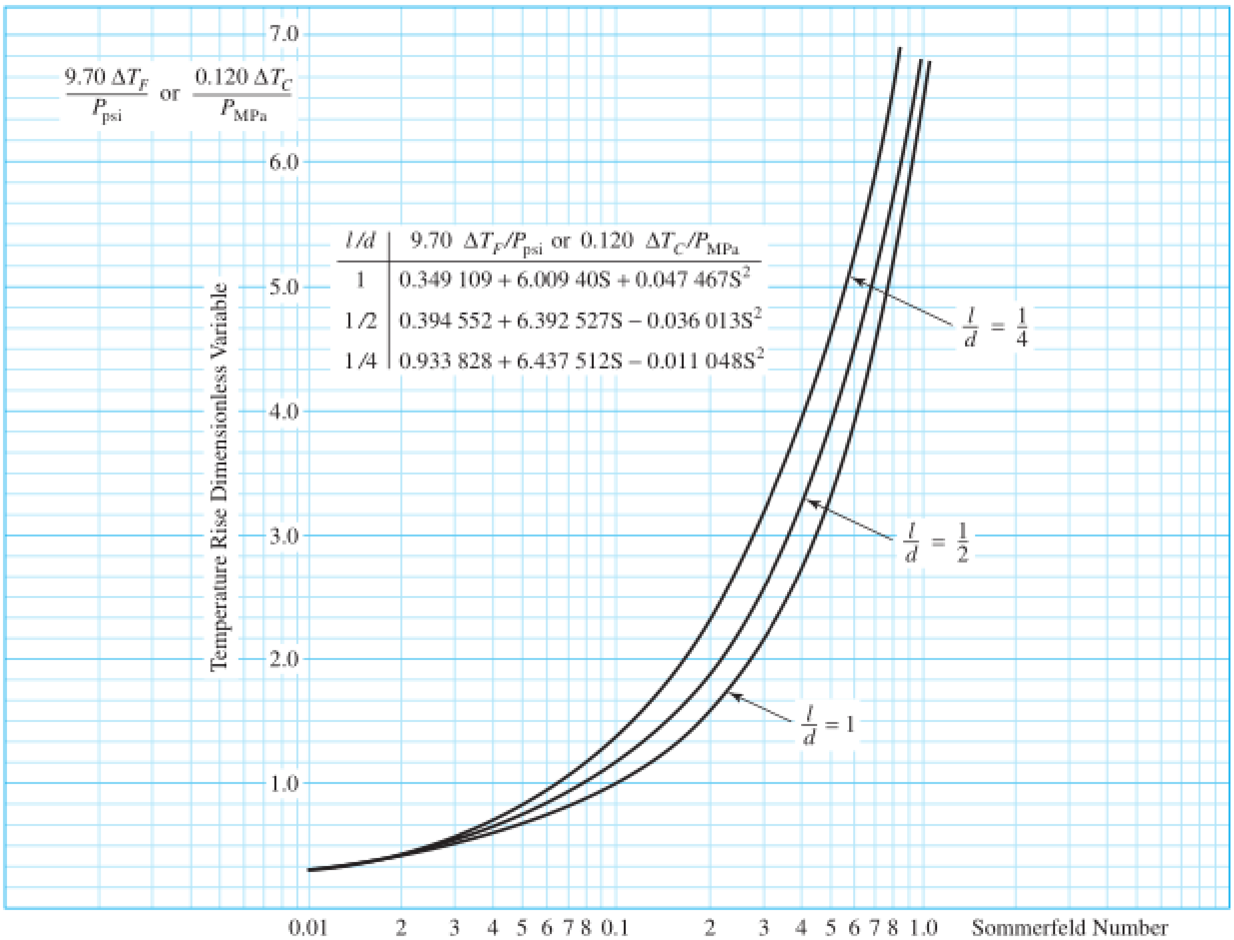


Figure 12–24

Figures 12–18, 12–19, and 12–20 combined to reduce iterative table look-up. *(Source: Chart based on work of Raimondi and Boyd boundary condition (2), i.e., no negative lubricant pressure developed. Chart is for full journal bearing using single lubricant pass, side flow emerges with temperature rise $\Delta T/2$, thru flow emerges with temperature rise ΔT, and entire flow is supplied at datum sump temperature.)*

Note: Figure 12-24 combines Figures 12-18, 12-19, 12-20, to facilitate the evaluation of ΔT_F or ΔT_C, which in turn is to expedite the iterative process of journal bearing design.

The metric version of the temperature rise formula is:

$$\frac{0.120\Delta T_C}{P_{MPa}} = \frac{1}{1-0.5\left(\frac{Q_s}{Q}\right)} \frac{\frac{r}{c}f}{\frac{Q}{rcNl}}$$

12-12 Loads and Materials

12-13 Bearing Types

Unit Loads in Typical Applications: Table 12-5.

Table 12-5

Range of Unit Loads in Current Use for Sleeve Bearings

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

Bearing Materials

Table 12-6 lists bearing alloys for applications involving high speed, high temperature, and high varying loads:

- Automotive engines (e.g., connecting rod, crankshaft)
- Turbo machinery

Why Alloys?

Babbitt metals are tin- or lead-based bearing materials, named after Babbitt, who came up with some tin-based alloys (~80% tin, ~10% antimony, ~10% copper).

Lead-based: ~80% lead, ~15% antimony, ~5% tin

Lead-based Babbittt can compensate for reasonable shaft misalignment and deflections.

Tin-based and Lead-based Babbitt metals allow foreign particles to become embedded not the bearing to prevent scratching of journal and bearing.

Trimetals typically have layers of materials such as Babbitt, copper or lead, and steel backing.

Trimetals have high fatigue strength to support compressive cyclic loading. In addition, metals like lead, copper and so on, have high thermal conductivity to remove heat rapidly from the bearing.

Bearing Types:
See Figures 12-32 to 12-34, from (Very simple) solid bushing to two-piece designs with elaborate groove pattern.

12-7 Design Variables

- **Controlling Variables** (variables under the control of the designer)
  Viscosity of the lubricant μ;
  Unit load $P = W/(2rl)$; ($W = radial\ load, d = 2r$)
  Speed N (rps);
  Bearing dimensions: r (journal radius), c (radial clearance), and l (bearing length).
- **Dependent Variables** (variables determined by the charts; they indicate how well the bearing performs, hence performance variables)
  The coefficient of friction;
  The temperature rise;
  The maximum film pressure and location;
  The flow rate of lubricant:
  The minimum film thickness and location; and so on

Recommended Bearing Dimensions

- $l/(2r) = 0.25 \sim 1.5$ (not to exceed 2.0);
  Shorter bearings place less stringent requirement on shaft deflection and misalignment; longer bearings on the other hand have less end leakage.
- $c/r = 0.001 \sim 0.0015$;
  The lower end value 0.001 is for precision bearings; the higher end value 0.0015 is for less precise bearings.

Minimum Film Thickness

- $h_{min} = 0.0002 + 0.000\ 04d$ (British units, d and h_{min} in mm)
  or
  $h_{min} = 0.005 + 0.000\ 04d$ (Metric units, d and h_{min} in mm)
- The first term on the RHS represents the peak-valley roughness of finely ground journal surface. Smaller h_{min} would require more expensive manufacturing.
- The second term on the RHS represents the influence of size (tolerance increases with size)

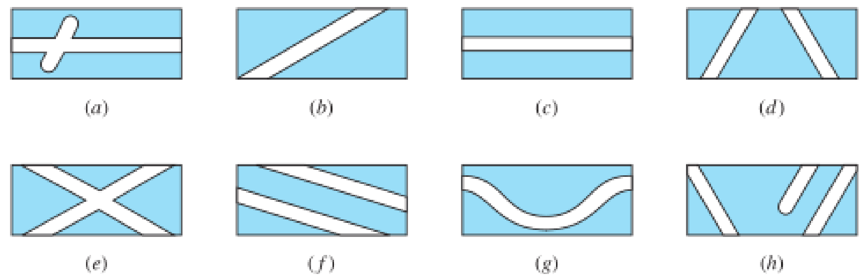
What Constitutes a Good Design?

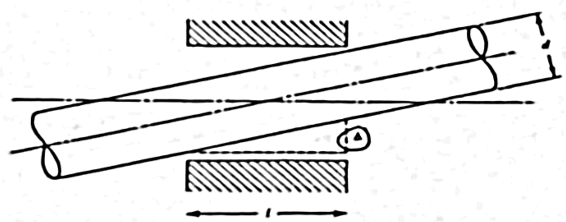
- Minimum film thickness: $h_0 \geq h_{min}$
- Friction: as low as possible ($f < 0.01$)
- Maximum temperature: $T_{max} \leq 250°F$
- In the “optimal zone” (as shown in Figure 12-16)

- Minimum film thickness is greater than shaft deflection across the length of bearing (to prevent binding between shaft and bearing; see diagram below)
- Using a suitable bearing material (see Sec. 12-12)
- Able to accommodate changes in clearance, temperature, viscosity, etc. (this will be discussed in Sec. 12-10)
- Allowing lubricant to be distributed over the surface (see Figure 12-34 regarding groove pattern)

Figure 12–34

Developed views of typical groove patterns. *(Courtesy of the Cleveland Graphite Bronze Company, Division of Clevite Corporation.)*





Δ: shaft deflection across the bearing; *l*: length of bearing.

Courtesy of "Machine Design Databook", Ch.23, K. Lingaiah, 2nd Ed.,

Procedure

- There is not a *typical* procedure.
- The common theme amongst the procedures: most involve assumptions and iterations, and do not guarantee convergence in the end, not to mention a solution/design in the optimal zone and meeting other requirements.

The first part of the process includes,

1. Choose bearing dimensions r, l, and c, making reference to Table 12-5: unit load for various applications.
2. Determine significant angular speed N by (Eq. 12-13); and
3. Select a lubricant

$$N = |N_j + N_b - 2N_f| \qquad (12\text{–}13)$$

where N_j = journal angular speed, rev/s

N_b = bearing angular speed, rev/s

N_f = load vector angular speed, rev/s

Where N_j is the speed of the journal; N_b is the speed of the bushing and N_f is the speed of load W as a vector. And all speeds are in rps. Figure 12-11 shows examples of applying (Eq. 12-13)

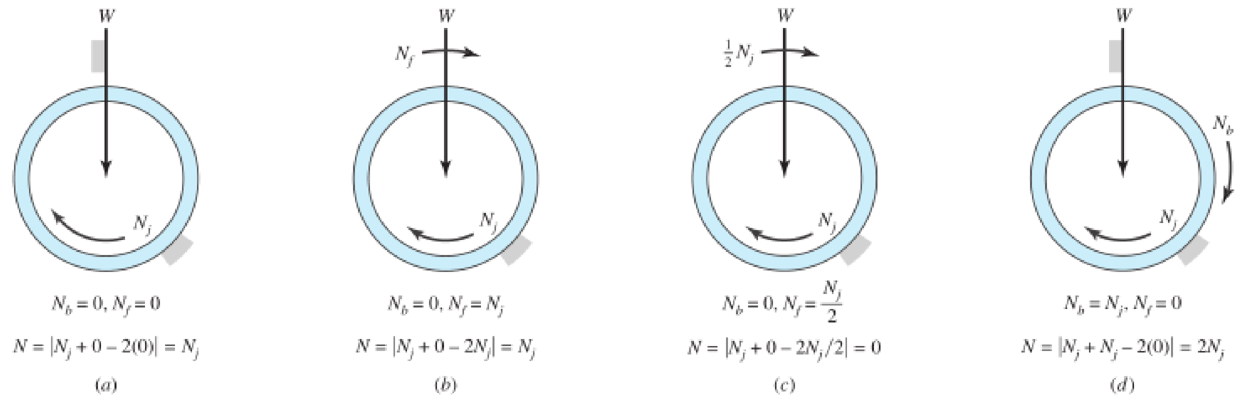


Figure 12–11

How the significant speed varies. (*a*) Common bearing case. (*b*) Load vector moves at the same speed as the journal. (*c*) Load vector moves at half journal speed, no load can be carried. (*d*) Journal and bushing move at same speed, load vector stationary, capacity halved.

The second part involves assumptions, iterations and so on. It ends when a converged solution/design is found. For example,

> Assume a temperature rise ΔT; determine the lubricant's temperature rise ΔT_F (in Fahrenheit) or ΔT_C (in Celsius); compare the assumed value with the calculated value; and if necessary, assume a new ΔT or select a different lubricant, iterate until the assumed ΔT and calculated ΔT_F or ΔT_C are close; (p. 624 of text, last paragraph)

For other ways to iterate for a converged solution/design, it's recommended that you reference other sources.

The last part is to evaluate the performance variables via Figures 12-16 through 12-22, and check against design requirements such as, in the optimal zone, $T_{max} \leq 250°F$, and so on. Typically, power loss due to friction, or rate of heat loss, is also determined.

Power loss H_{loss}

British units:

$$H_{loss} = \frac{fWrN}{1050}$$

Where H_{loss} is in hp; W is in lb; r is in inch; and N is in rps.

Metric units:

$$H_{loss} = \frac{fWrN}{9549}$$

Where H_{loss} is in kW; W is in newton; r is in m; and N is in rpm.

Example 2

A journal bearing has $d = l = 1.5"$, $c = 0.0015"$, $W = 500\ lb$, and $N = 30\ rps$. Lubricant is SAE20 oil with an inlet temperature of $100°F$. Design and evaluate the bearing.

Solution:

The first part of the process is not necessary since d, l, c, and N are given and lubricant is chosen. So,

$$P = \frac{W}{ld} = \frac{500}{1.5 \cdot 1.5} = 222.2\ psi, N = 30\ rps, \frac{l}{d} = 1, and\frac{r}{c} = \frac{0.75}{0.0015} = 500$$

Also, $T_1 = 100°F$

The second part is completed as follows:

Select $\mu = 4\ \mu reyn$

Sommerfeld number is:

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = (500)^2 \frac{(4)(10^{-6})(30)}{222.2} = 0.135$$

Temperature rise is, by Figure 12-24

$$\frac{9.70\Delta T_F}{P_{psi}} = 0.349109 + 6.00940S + 0.047467S^2 = 1.161$$

$$\Delta T_F = \frac{(1.161)(222.2)}{9.70} = 26.6°F$$

And the average temperature is $T_{av} = T_1 + \frac{\Delta T_F}{2} = 113.3°F$

On Figure 12-12, plot ($T_{av},\ \mu$). It is located below the SAE20 line.

Select $\mu = 7\ \mu reyn$

Sommerfeld number is:

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = (500)^2 \frac{(7)(10^{-6})(30)}{222.2} = 0.236$$

Temperature rise is, by Figure 12-24

$$\frac{9.70\Delta T_F}{P_{psi}} = 1.773$$

$$\Delta T_F = \frac{(1.773)(222.2)}{9.70} = 40.6°F$$

And the average temperature is $T_{av} = T_1 + \frac{\Delta T_F}{2} = 120.3°F$

Plot the second point ($T_{av},\ \mu$), making sure it is above the SAE20 line. Otherwise, assume a different viscosity until ($T_{av},\ \mu$) is above the line.