Lubrication and Journal Bearings

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, where in ball bearing the relative motion is rolling.

Journal bearings are more applicable for extreme operational conditions (*high loads and rotational speeds*). Also they are used for low demand applications (*without external lubrication*) because they are more cost effective than antifriction bearings.

Types of Lubrication

Five forms of lubrication can be identified:

1. Hydrodynamic
2. Hydrostatic
3. Elastohydrodynamic
4. Boundary
5. Solid film

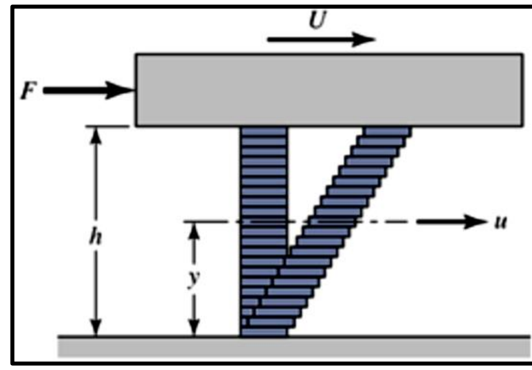
- ***Hydrodynamic (or full-film)***: in this type, the surfaces of the bearing are separated by a relatively thick film of lubricant (*to prevent metal to metal contact*). The film pressure is created by the moving surface forcing the lubricant into a wedge-shaped zone, therefore creating a pressure that separates the sliding surfaces.
- ***Hydrostatics***: in this type, the 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*).
- ***Elastohydrodynamic***: in this type, the lubricant is introduced between surfaces that are in rolling contact (*such as mating gears or rolling bearings*).
- ***Boundary***: this type is special case of hydrodynamic lubrication where the film thickness is reduced to be "very thin". This may happen because of increased load, reduced lubricant supply, reduced rotational speed, reduced viscosity, etc.
- ***Solid-film***: in this type, self-lubricating solid materials such as graphite are used in the bearing. This is used when bearings must operate at very high temperature.

Viscosity

- When we have a film of lubricant of thickness " " trapped between two plates where one is moving with velocity " " and the other is stationary, the lubricant film will be sheared such that the layer in contact with the moving plate will move at the same

velocity of the plate and the layer in contact with the stationary plate will stay stationary.

- The intermediate layers will have the velocities proportional to their distance " " from the stationary plate.



- *Newton viscous effect* states that the shear stress in the fluid is proportional to the velocity gradient. Thus we can write:

$$\boxed{\tau = \frac{F}{A} = \mu \frac{du}{dy}} \qquad (12\text{–}1)$$

where "**μ**", the dynamic *(or absolute)* viscosity, is the constant of proportionality.

- For *Newtonian fluids* the velocity gradient is constant; $\frac{du}{dy} = \frac{U}{h}$ *(i.e., linear increase)*

Thus,

$$\boxed{\tau = \frac{F}{A} = \mu \frac{U}{h}} \qquad (12\text{–}2)$$

- The unit for viscosity "μ " in *SI* system is *(Pa.s)*.
- In *US* system the unit is *lb.s/in2 (psi.s)* and it is called "*reyn*".
- The conversion factor is: 1 *reyn* (psi.s) = 6895 *Pa.s*

Petroff's Equation

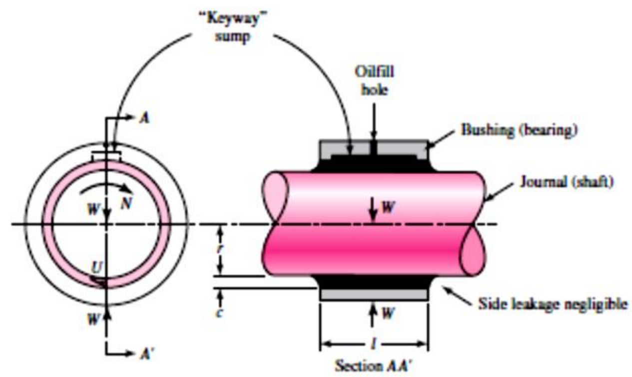
The *Petroff* equation gives the coefficient of friction in journal bearings. It is based on the assumption that the shaft is concentric. Though the shaft is not concentric but the coefficient of friction predicted by this equation turns out to be quite good.

- Consider a shaft of radius "r " rotating inside a bearing with rotational speed " ", and the clearance between the shaft and sleeve " " is filled with oil (*leakage is negligible*).

- From *Newton's* viscosity equation we get:

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.



$$\boxed{\tau = \mu \frac{U}{h} = \mu \frac{2\pi r N}{c}} \qquad \text{(a)}$$

The force needed to shear the film is $F = \tau A$ (where $A = 2\pi r l$) and the torque $T = Fr = \tau A r$.

Thus the torque can be written as:

$$\boxed{T = \frac{4\pi^2 r^3 l \mu N}{c}} \qquad \text{(b)}$$

- The pressure on the projected area is $P = \frac{W}{2rl} \rightarrow W = 2rlP$ and the torque created by the frictional force "***fW***" is:

$$\boxed{T = fWr = (f)(2rlP)(r) = 2r^2 flP} \qquad \text{(c)}$$

- Equating 1 & 2 and solving for "***f***" gives:

The coefficient of friction $\Rightarrow$ $$\boxed{f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}}$$ Pitroff's equation (12–6)

- The quantities $\left(\frac{\mu N}{P}\right)$ & $\left(\frac{r}{c}\right)$ are non-dimensional, and they are very important parameters in lubrication.
- They are used to give the "*bearing characteristic number*" or the "*Sommerfeld number*" (also non-dimensional) which is:

$$\boxed{S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}} \qquad \boxed{N \text{ is in } (rev/s)} \qquad (12\text{–}7)$$

And then;

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

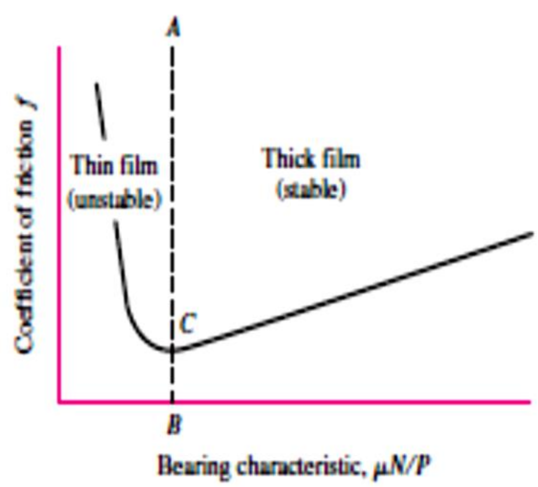
- The *Sommerfeld* number is very important in lubrication analysis because it contains many of the parameters specified by the designer.

Stable Lubrication

The difference between boundary (*thin film*) and hydrodynamic (*thick film*) lubrication can be explained by the figure (*which was obtained from testing*).

Figure 12–4

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



- Suppose we are operating to the right of point "***C*** " and something happens and increases the temperature of lubricant:

⇨ Temperature ↑ , viscosity "μ" , friction "f"↓ , heat generated by shearing the lubricant ↓ , temperature ↓ *Self-correcting * stable lubrication **

- If we operate to the left of point "***C***" and the temperature increased
- Temperature↑, viscosity "μ" ↓ , friction "f" ↑, (*metal to metal contact*) more heat is generated which increases the temperature more.

⇨ *Damage * unstable lubrication **

- To ensure thick-film lubrication we should have:

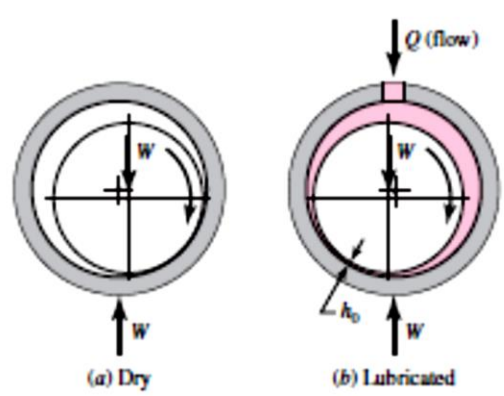
$$\boxed{\frac{\mu N}{P} \geq 1.7(10^{-6})} \qquad (a)$$

Thick-Film Lubrication

- Suppose the journal starts to rotate in clockwise direction while it is still dry, the journal will roll up the right side of the bearing, as seen in *(a)*.
- Once the lubricant is introduced, the rotating journal will pump the 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)*.

Figure 12-5

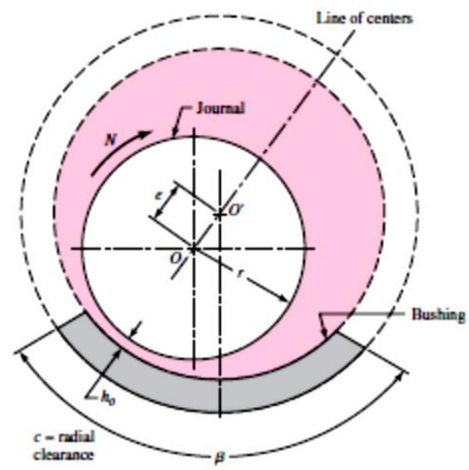
Formation of a film.



- The "minimum film thickness", h_o , occurs at the bottom half of the bearing but slightly to the left (*for clockwise rotation*), as seen in *(b)*.

- The nomenclature of a journal bearing is:
- *Radial clearance* "***c***" = $r_b - r_j$
- *Eccentricity* " ***e*** ": the distance between the centers of the bushing and journal.

Figure 12-6
Nomenclature of a partial journal bearing.



$$e = c - h_o$$

- We also define an "*eccentricity ratio*", ϵ, as:

$$\epsilon = \frac{e}{c}$$

- *Full bearing*: the bushing encloses the journal.
- *Partial bearing*: the angle β describes the angular length of the partial bushing.

Hydrodynamic Theory

The present hydrodynamic theory is based on the experiments conducted by *Tower*, which later *Reynolds* developed a mathematical model to explain it.

- *Reynolds'* work was based on some assumptions which are:
- The film thickness is very small compared with the bearing radius; therefore the curvature could be neglected.
- The lubricant is Newtonian, incompressible and its viscosity is constant.
- Forces due to inertia of the lubricant are neglected.
- The film pressure is constant in the axial and vertical directions.
- The bearing has infinite length in the axial direction (*no side leakage*).

- The velocity of any particle in the lubricant depends only on the x and y coordinates.
- *Reynolds* equation for one-dimensional flow (*with negligible side leakage*) is: *"See derivation in the text"*

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

- When side leakage is not neglected, the equation becomes:

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

- There is no general analytical solution for this equation; one of the important solutions (*numerical*) was introduced by *Sommerfeld* which is:

$$\boxed{\frac{r}{c}f = \emptyset\left[\underbrace{\left(\frac{r}{c}\right)^2\frac{\mu N}{P}}_{\text{Sommerfeld number}}\right]} \qquad (12\text{–}12)$$

where φ indicates a functional relationship. Sommerfeld found the functions for half bearings and full bearings by using the assumption of no side leakage.

Design Considerations

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

In the first group are those whose values either are given or are under the control of the designer. These Independent (or *design*) variables; might be controlled directly by the designer which include:

1 The viscosity μ
2 The load per unit of projected bearing area, *P*
3 The speed *N*
4 The bearing dimensions *r*, *c*, β, and *l*

(though speed "N " and sometimes viscosity "μ " may be forced on the designer)

In the second group are the dependent variables. The designer cannot control these except indirectly by changing one or more of the first group. These are:

1 The coefficient of friction *f*
2 The temperature rise _*T*
3 The volume flow rate of oil *Q*
4 The minimum film thickness *h*0

These variables tell about the performance of the bearing, and may be called the "*performance factors*" (*the designer may impose limitations on those variables to ensure satisfactory performance*).

Significant angular speed:

The rotational speed " N" that is used in the *Sommerfeld* number depends on the rotation of the journal, the bearing and the load. It can be found as:

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

where
Nj = journal angular speed, rev/s
Nb = bearing angular speed, rev/s
Nf = load vector angular speed, rev/s

Figure 12–11 shows several situations for determining *N*.

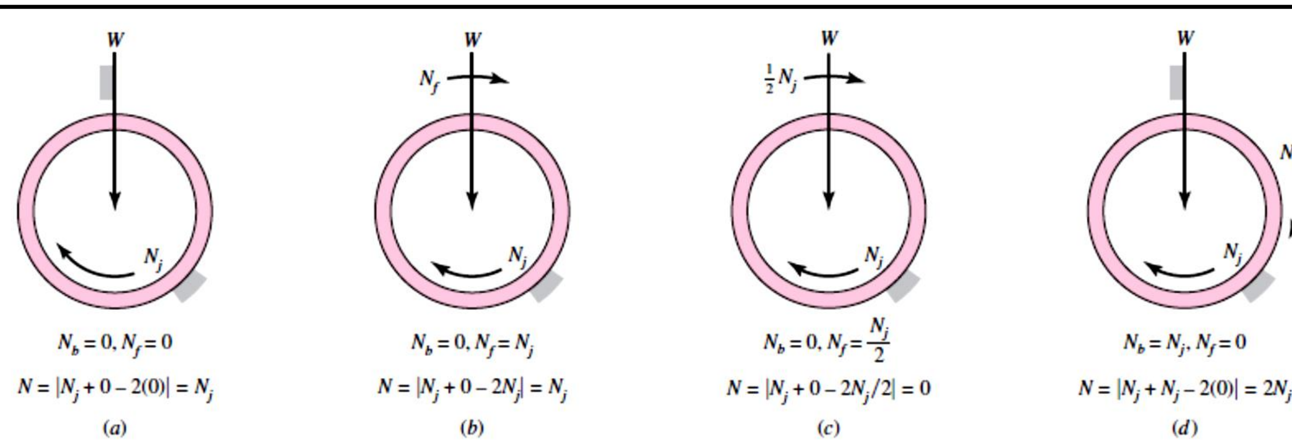


Figure 12–11

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.

Trumpler's Design Criterion

Based on his experience, *Trumpler* introduced some limitation for the design of journal bearings, which are:

- *Minimum film thickness* " h_0 "

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

Therefore *Trumpler* suggest that:

$$\boxed{h_0 \geq 0.00508 + 0.00004d} \quad mm \qquad (a)$$

Or in inches;

$$\boxed{h_0 \geq 0.0002 + 0.000\,04d \text{ in} \qquad (a)}$$

Where d is the journal diameter in "*mm*".

- *Maximum lubricant temperature*

When temperature increases beyond a certain limit, lighter components of the lubricant starts to evaporate which increases viscosity and thus friction. For light oils, *Trumpler* suggests:

$$\boxed{T_{max} \leq 121} \ °C \qquad (b)$$

Or in inches;

$$\boxed{T_{max} \leq 250°\text{F} \qquad (b)}$$

- ***Starting load***

Journal bearing usually consist of a steel journal and a bushing of softer material. If the starting load is high, the bushing will be damaged because of the metal to metal contact. Thus, it is suggested that the starting load divided by the projected area is:

$$\boxed{\frac{W_{st}}{ld} \leq 2068} \quad kPa \qquad (c)$$

Note that starting load is usually smaller than running load

Or in inches;

$$\frac{W_{st}}{lD} \leq 300 \text{ psi} \qquad \text{(c)}$$

- ***Running load design factor***

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

$$\boxed{n_d \geq 2} \qquad \boxed{\begin{array}{l}\textit{For running load}\\ \underline{\textit{not}}\ \textit{starting load}\end{array}} \qquad \text{(d)}$$

The Relation of the Variables

Raimondi and *Boyd* used numerical solution to solve the *Reynolds'* equation. They presented their numerical results relating the different variables in the form of charts.

- The charts presented in the text are for full bearing $(\beta = 360°)$
- Viscosity charts (*figs. 12-12 to 12-14*).

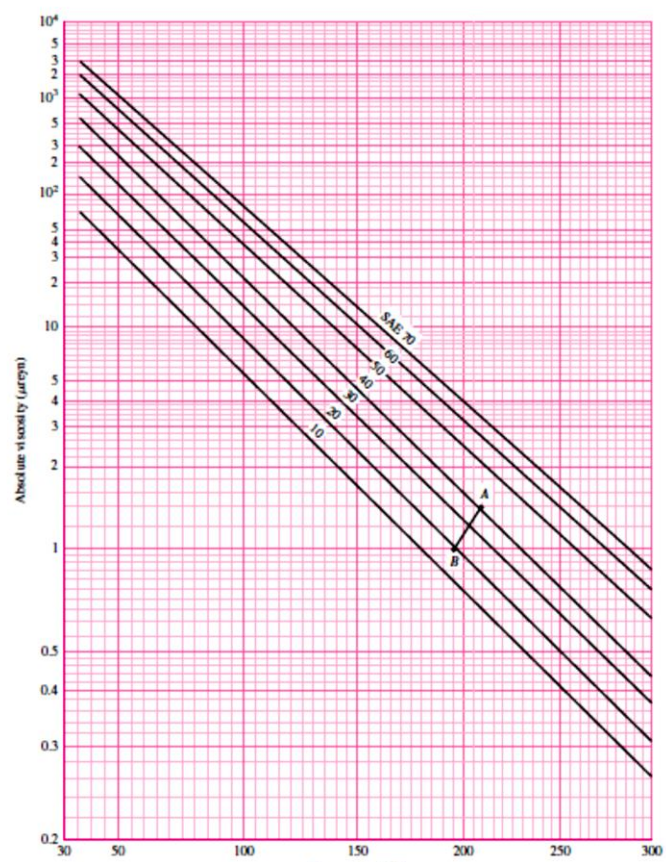


Figure 12-12 Viscosity–temperature chart in U.S. customary units. (*Raimondi and Boyd.*)

Figure 12–13

Viscosity–temperature chart in SI units. *(Adapted from Fig. 12–12.)*

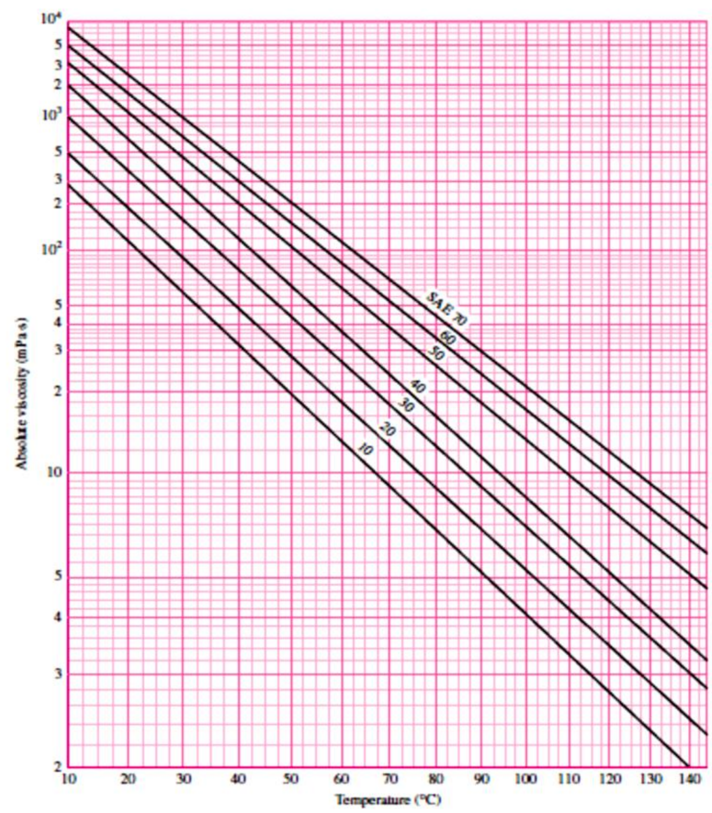
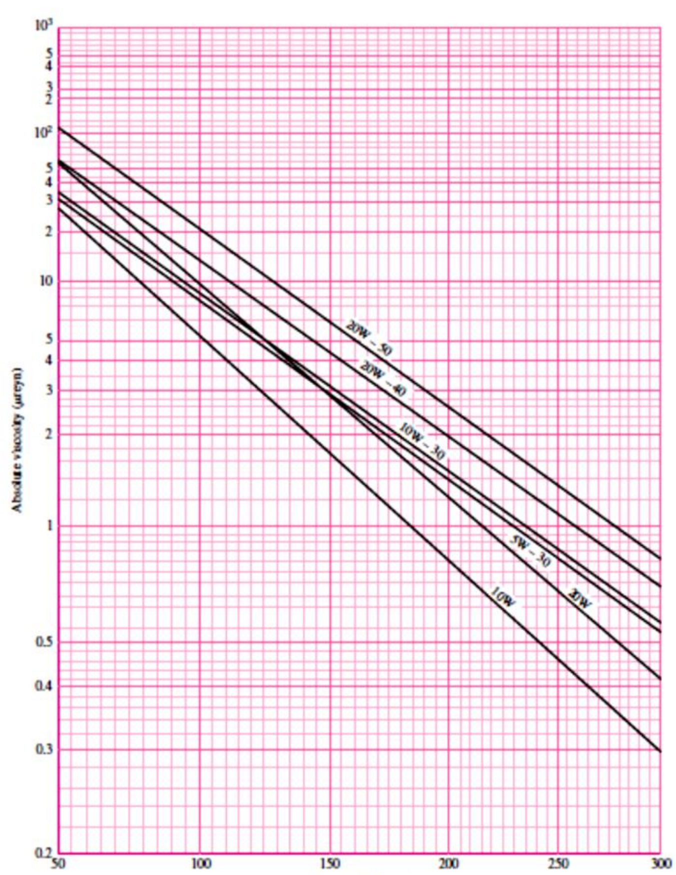


Figure 12–14

Chart for multiviscosity 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.



- *Table 12-1* gives the viscosity *vs*. temperature in table from using curve fits.

Table 12-1

Curve Fits* to Approximate the Viscosity versus Temperature Functions for SAE Grades 10 to 60

Source: A. S. Seireg and S. Dandage, "Empirical Design Procedure for the Thermodynamic Behavior of Journal Bearings," *J. Lubrication Technology*, vol. 104, April 1982, pp. 135–148.

Oil Grade, SAE	Viscosity μ_0, reyn	Constant b, °F
10	$0.0158(10^{-6})$	1157.5
20	$0.0136(10^{-6})$	1271.6
30	$0.0141(10^{-6})$	1360.0
40	$0.0121(10^{-6})$	1474.4
50	$0.0170(10^{-6})$	1509.6
60	$0.0187(10^{-6})$	1564.0

*$\mu = \mu_0 \exp[b/(T + 95)]$, T in °F.

- In their solution, *Raimondi-Boyd*, assumed the temperature of the lubricant to stay constant as it passes through the bearing. In reality, temperature increases because of the work done on the lubricant. Thus, when finding viscosity from the chart we should use an average temperature value which is:

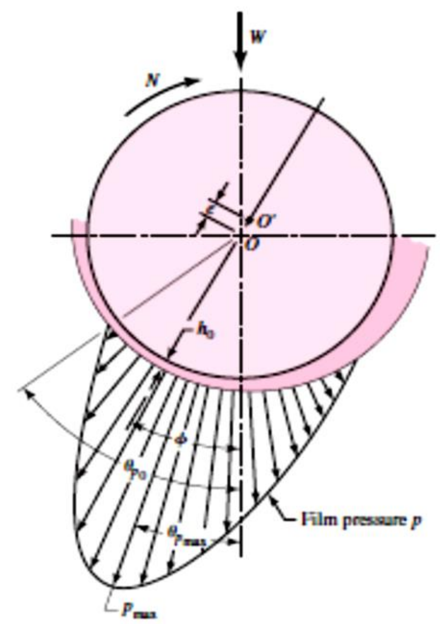
$$\boxed{T_{av} = T_1 + \frac{\Delta T}{2}} \qquad (12\text{–}14)$$

T_1: Inlet temperature
ΔT: Temperature rise

- When we know the oil inlet temperature and need to find the outlet temperature we have to use trial-and-error where we assume the temperature rise and find viscosity from chart then use it to compute a temperature rise. If it does not match the assumed ΔT then another value is tried and so on.
- The charts give the variables against the *Sommerfeld* number "S " for different $^{l}/_{d}$ ratios.
- The figure explains the notation used in the charts.
- Minimum film thickness and its angular position (*figs. 12-16 and 12-17*)

Figure 12–15

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



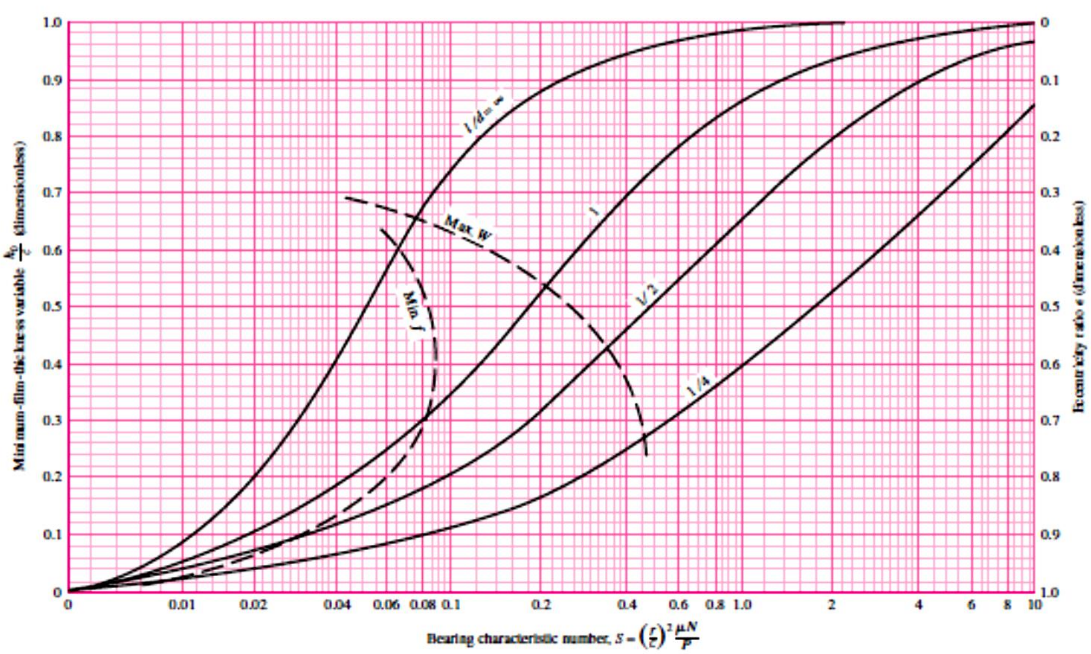
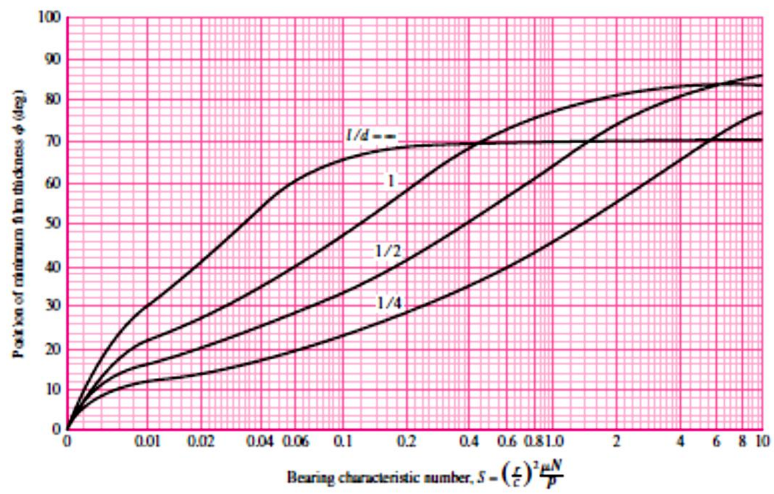


Figure 12–16

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.)*

Figure 12–17

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



- note that $h_0 = c - e$ ➔ $\frac{h_0}{c} = 1 - \epsilon$

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.

Solution The nominal bearing pressure (in projected area of the journal) is

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

The Sommerfeld number is, from Eq. (12–7), where $N = N_j = 30$ rev/s,

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

Also, $l/d = 1.50/[2(0.75)] = 1$. Entering Fig. 12–16 with $S = 0.135$ and $l/d = 1$ gives $h_0/c = 0.42$ and $\epsilon = 0.58$. The quantity h_0/c is called the *minimum film thickness variable*. Since $c = 0.0015$ in, the minimum film thickness h_0 is

$$h_0 = 0.42(0.0015) = 0.000\,63 \text{ in}$$

We can find the angular location ϕ of the minimum film thickness from the chart of Fig. 12–17. Entering with $S = 0.135$ and $l/d = 1$ gives $\phi = 53°$.

The eccentricity ratio is $\epsilon = e/c = 0.58$. This means the eccentricity e is

$$e = 0.58(0.0015) = 0.000\,87 \text{ in}$$

- The coefficient of friction variable (fig. 12-18) has the friction variable *(r/c) f* plotted against Sommerfeld number S with contours for various values of the *l/d* ratio

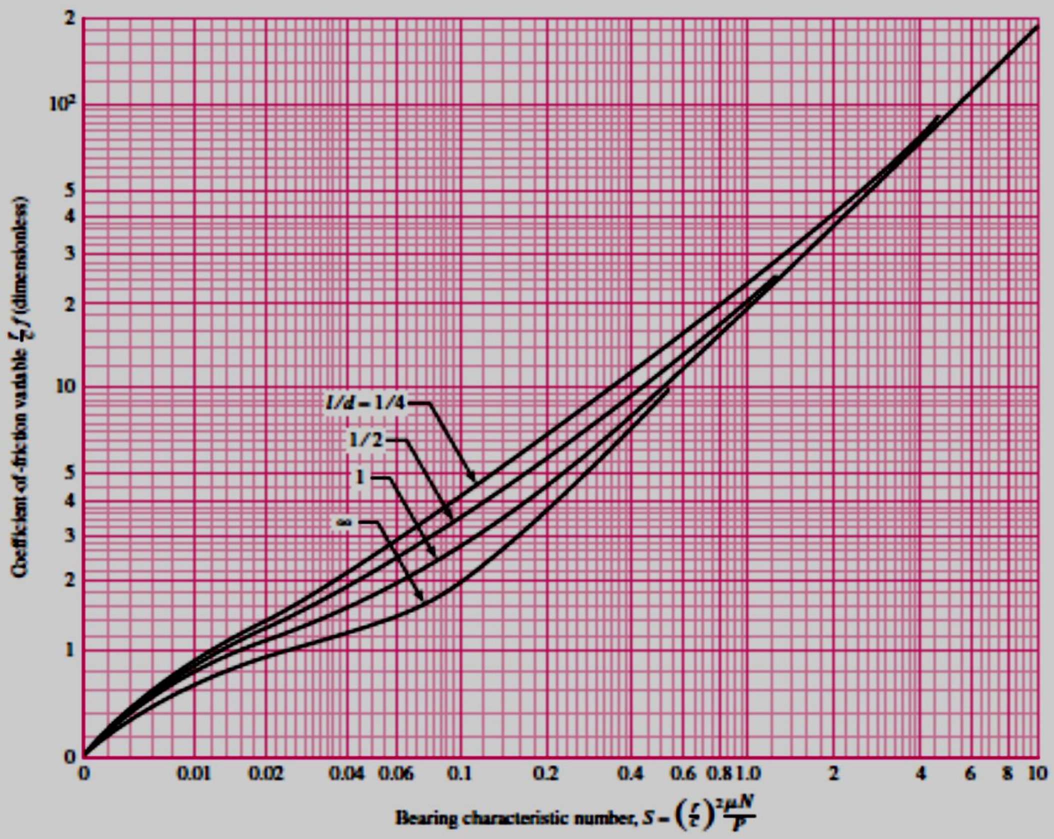


Figure 12–18

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

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.

Solution We enter Fig. 12–18 with $S = 0.135$ and $l/d = 1$ and find $(r/c)f = 3.50$. The coefficient of friction f is

$$f = 3.50\,c/r = 3.50(0.0015/0.75) = 0.0070$$

The friction torque on the journal is

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

The power loss in horsepower is

$$(hp)_{\text{loss}} = \frac{TN}{1050} = \frac{2.62(30)}{1050} = 0.075 \text{ hp}$$

or, expressed in Btu/s,

$$H = \frac{2\pi TN}{778(12)} = \frac{2\pi(2.62)30}{778(12)} = 0.0529 \text{ Btu/s}$$

- Lubricant flow and side flow ratio (*figs. 12-19 and 12-20*)

Figure 12–19

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

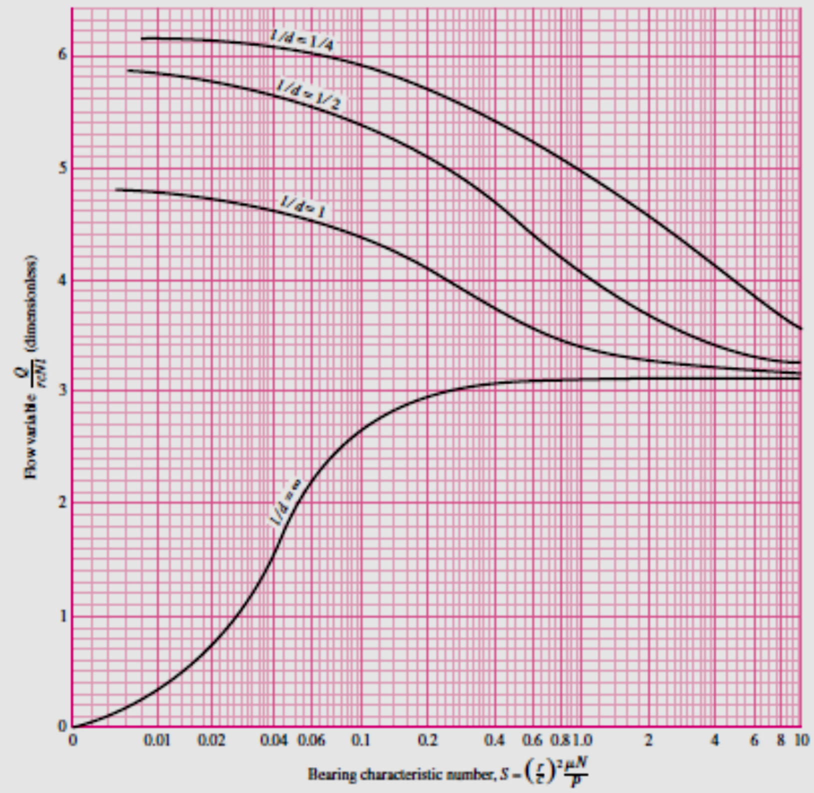
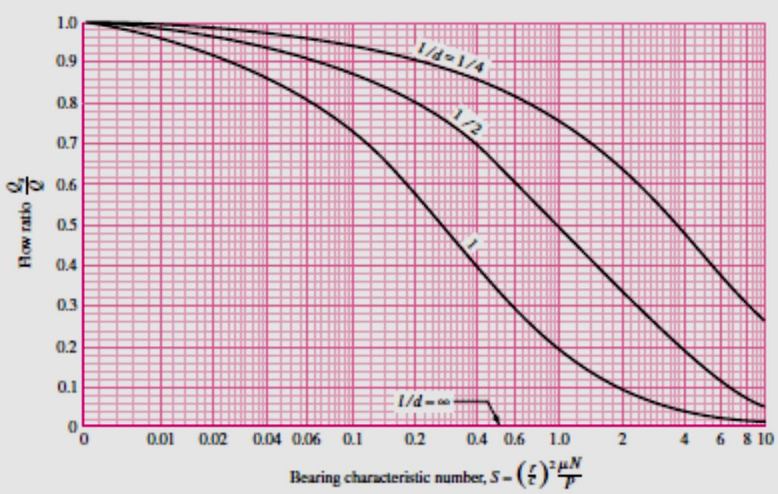


Figure 12–20

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



- *Why there is no ($l/d = \infty$) curve in fig. 12-20?*

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.

Solution To estimate the lubricant flow, enter Fig. 12–19 with $S = 0.135$ and $l/d = 1$ to obtain $Q/(rcNl) = 4.28$. The total volumetric flow rate is

$$Q = 4.28rcNl = 4.28(0.75)0.0015(30)1.5 = 0.217 \text{ in}^3/\text{s}$$

From Fig. 12–20 we find the *flow ratio* $Q_s/Q = 0.655$ and Q_s is

$$Q_s = 0.655Q = 0.655(0.217) = 0.142 \text{ in}^3/\text{s}$$