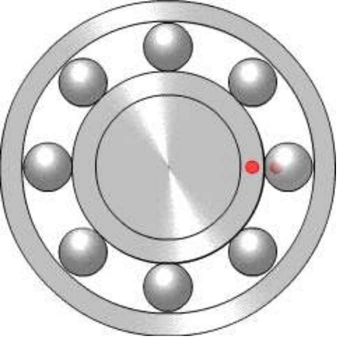


Mechanical Design II

By S.G.Khan

Chapter 12

- Journal Bearings

Lubrication and Journal Bearings

The object of lubrication is to reduce friction, wear, and heating of machine parts which move relative to each other. A lubricant is any substance which, when inserted between the moving surfaces, accomplishes these purposes. In a sleeve bearing, a shaft, or *journal*, rotates or oscillates within a sleeve, or *bushing*, and the relative motion is sliding. In an antifriction bearing, the main relative motion is rolling. A follower may either roll or slide on the cam. Gear teeth mate with each other by a combination of rolling and sliding. Pistons slide within their cylinders. All these applications require lubrication to reduce friction, wear, and heating.

Lubrication and Journal Bearings

The field of application for journal bearings is immense.

The crankshaft and connecting-rod bearings of an automotive engine must operate for thousands of miles at high temperatures and under varying load conditions. The journal bearings used in the steam turbines of power-generating stations are said to have reliabilities approaching 100 percent. At the other extreme there are thousands of applications in which the

Journal:

That part of a shaft that is prepared to accept and support a bearing

Journal Bearing (Oil filmed Bearings) **A journal bearing, simply stated, is a cylinder which surrounds the shaft and is filled with some form of fluid lubricant. In this bearing a fluid is the medium that supports the shaft preventing metal to metal contact. The most common fluid used is oil, with special applications using water or a gas.**

Journal Bearings

Recent metallurgy developments in bearing materials, combined with increased knowledge of the lubrication process, now make it possible to design journal bearings with satisfactory lives and very good reliabilities.

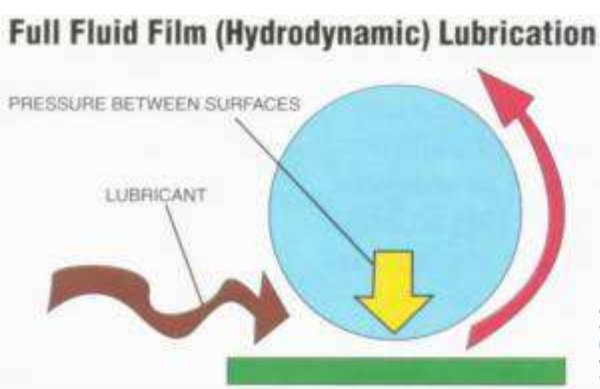
Types of Lubrication

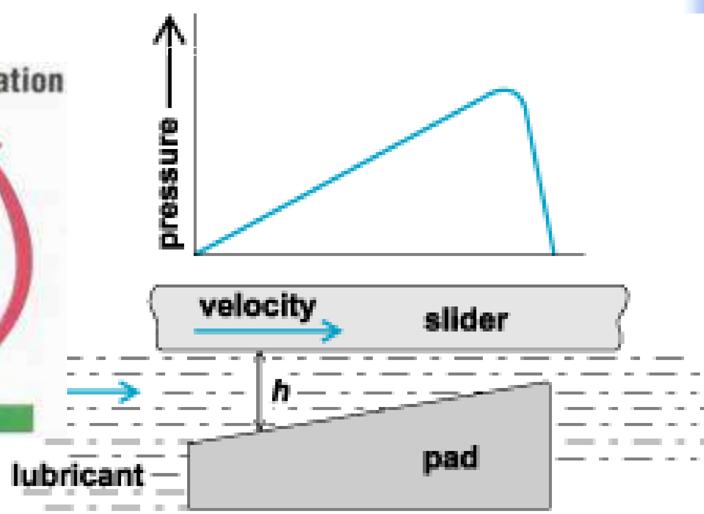
Five distinct forms of lubrication may be identified:

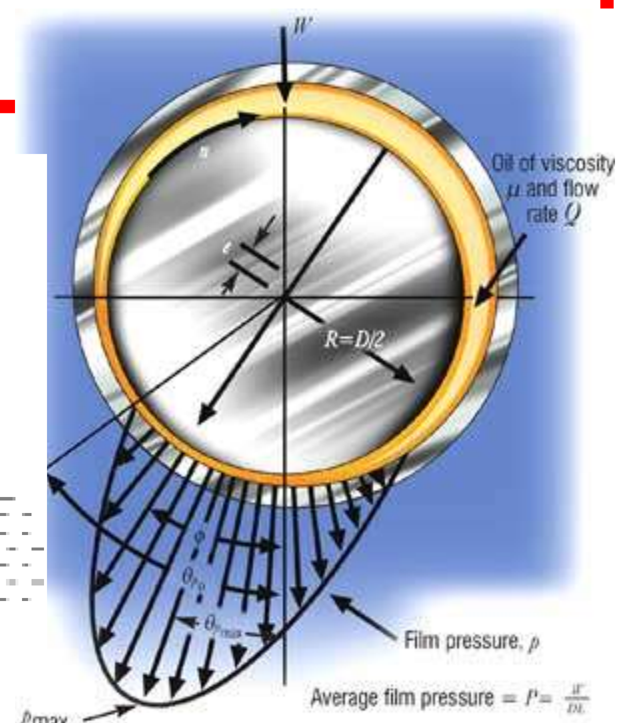
1. Hydrodynamic
2. Hydrostatic
3. Elastohydrodynamic
4. Boundary
5. Solid-film

Hydro Dynamic Lubrication

Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent metal-to-metal contact, and that the stability thus obtained can be explained by the laws of fluid mechanics. Hydrodynamic lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur; but it does require the existence of an adequate supply at all times. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing. Hydrodynamic lubrication is also called *full-film*, or *fluid, lubrication*.

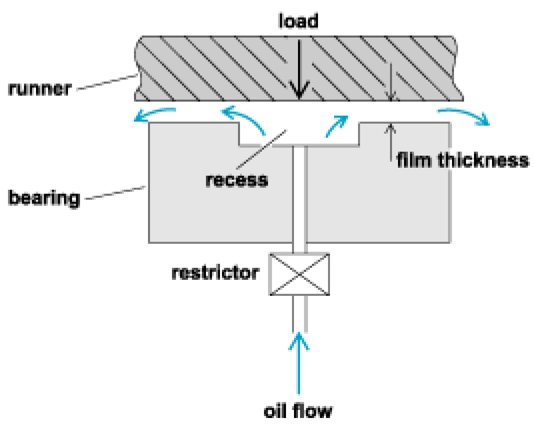


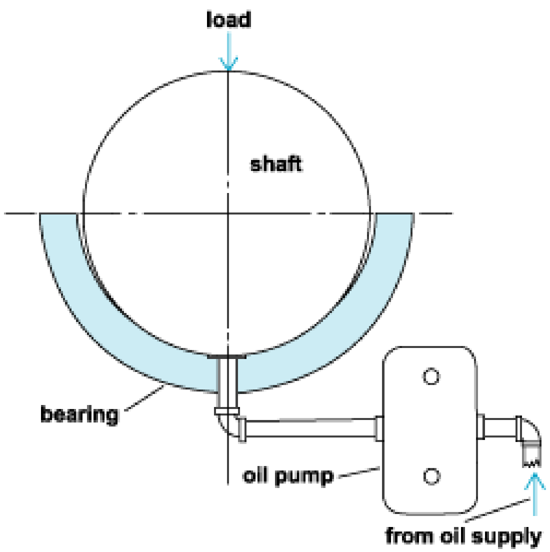




Hydrostatic Lubrication

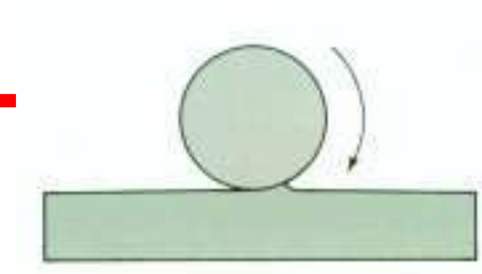
Hydrostatic lubrication is obtained by introducing the lubricant, which is sometimes air or water, into the load-bearing area at a pressure high enough to separate the surfaces with a relatively thick film of lubricant. So, unlike hydrodynamic lubrication, this kind of lubrication does not require motion of one surface relative to another. We shall not deal with hydrostatic lubrication in this book, but the subject should be considered in designing bearings where the velocities are small or zero and where the frictional resistance is to be an absolute minimum.

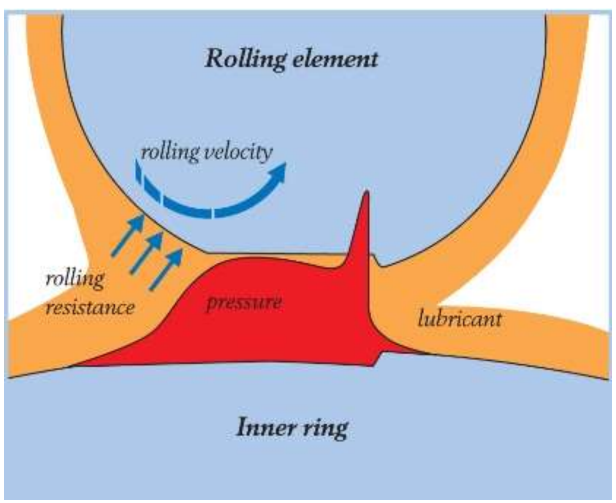




Elasto-Hydro-Dynamic Lubrication

Elastohydrodynamic lubrication is the phenomenon that occurs when a lubricant is introduced between surfaces which are in rolling contact, such as mating gears or rolling bearings. The mathematical explanation requires the Hertzian theory of contact stress and fluid mechanics.

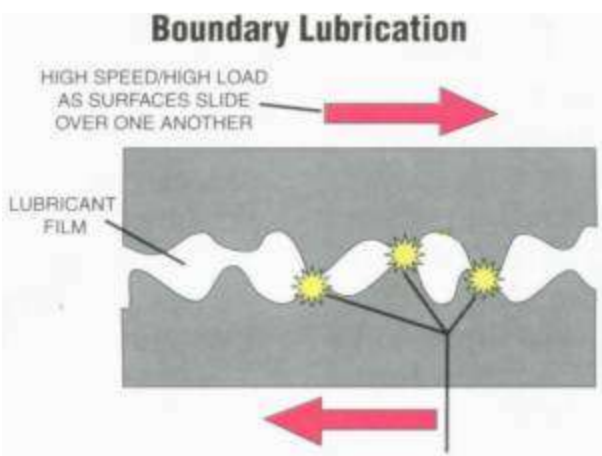




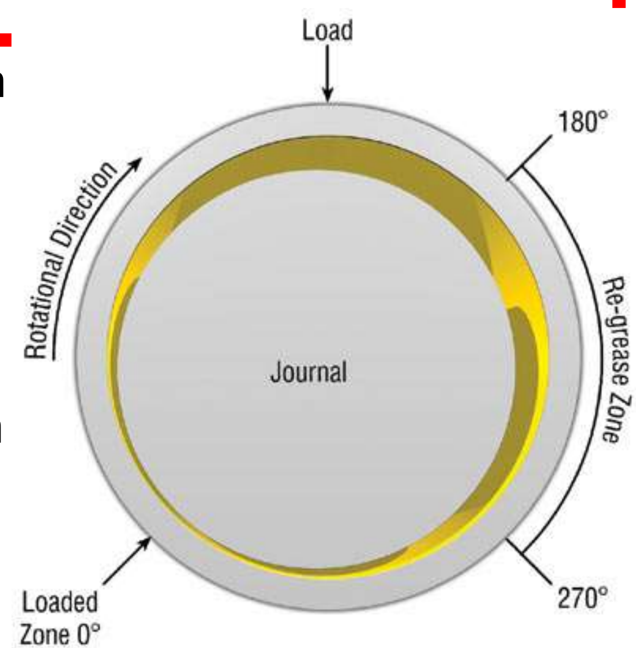
Elastohydrodynamic Lubrication (EHD or EHL) - Condition in which surfaces of rolling elements of high speed and high load components are either completely, or in part, separated by a very thin lubricant film. Mating parts deform elastically, due to incompressibility of the lubricant film under very high pressure and, therefore, trap or create a wedge of lubricant between them. This is similar to "hydroplaning" that cars exhibit on wet roads.

Boundary Lubrication

Insufficient surface area, a drop in the velocity of the moving surface, a lessening in the quantity of lubricant delivered to a bearing, an increase in the bearing load, or an increase in lubricant temperature resulting in a decrease in viscosity—any one of these—may prevent the buildup of a film thick enough for full-film lubrication. When this happens, the highest asperities may be separated by lubricant films only several molecular dimensions in thickness. This is called *boundary lubrication*. The change from hydrodynamic to boundary lubrication is not at all a sudden or abrupt one. It is probable that a mixed hydrodynamic- and boundary-type lubrication occurs first, and as the surfaces move closer together, the boundary-type lubrication becomes predominant. The viscosity of the lubricant is not of as much importance with boundary lubrication as is the chemical composition.

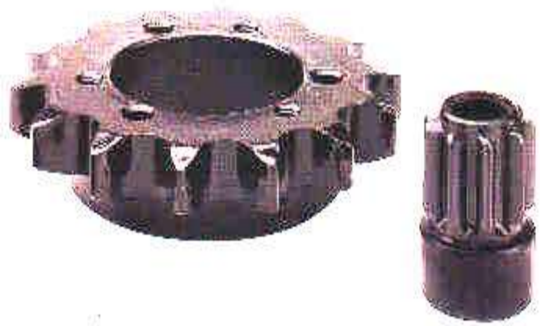


Lubricant film is too thin to provide total surface separation. Contact between surface asperities (or microscopic peaks and valleys) occurs. Friction production and wear production is then only provided through chemical additives.



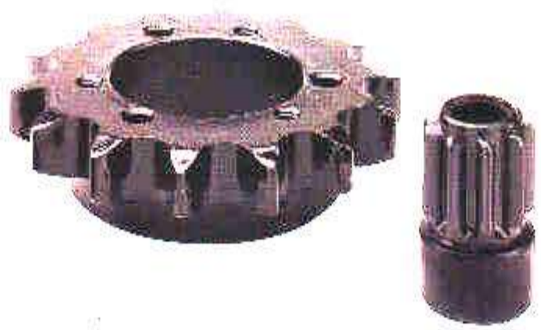
Solid-Film Lubrication

When bearings must be operated at extreme temperatures, a *solid-film lubricant* such as graphite or molybdenum disulfide must be used because the ordinary mineral oils are not satisfactory. Much research is currently being carried out in an effort, too, to find composite bearing materials with low wear rates as well as small frictional coefficients.

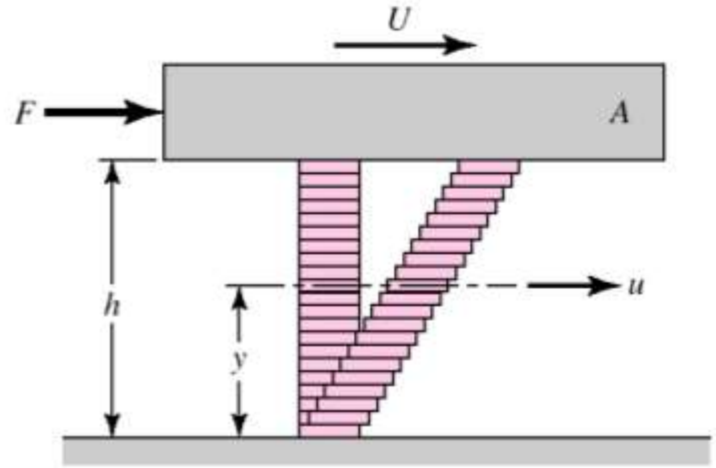


Solid Film Lube Over Gears

there are thousands of applications in which the loads are light and the service relatively unimportant; a simple, easily installed bearing is required, using little or no lubrication. In such cases an antifriction bearing might be a poor answer because of the cost, the elaborate enclosures, the close tolerances, the radial space required, the high speeds, or the increased inertial effects. Instead, a nylon bearing requiring no lubrication, a powder-metallurgy bearing with the lubrication "built in," or a bronze bearing with ring oiling, wick-feeding, or solid-lubricant film or grease lubrication might be a very satisfactory solution. Recent metallurgy developments in



Viscosity



Newton's viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to y. Thus

$$\tau = \frac{F}{A} = \mu \frac{du}{dy} \tag{12–1}$$

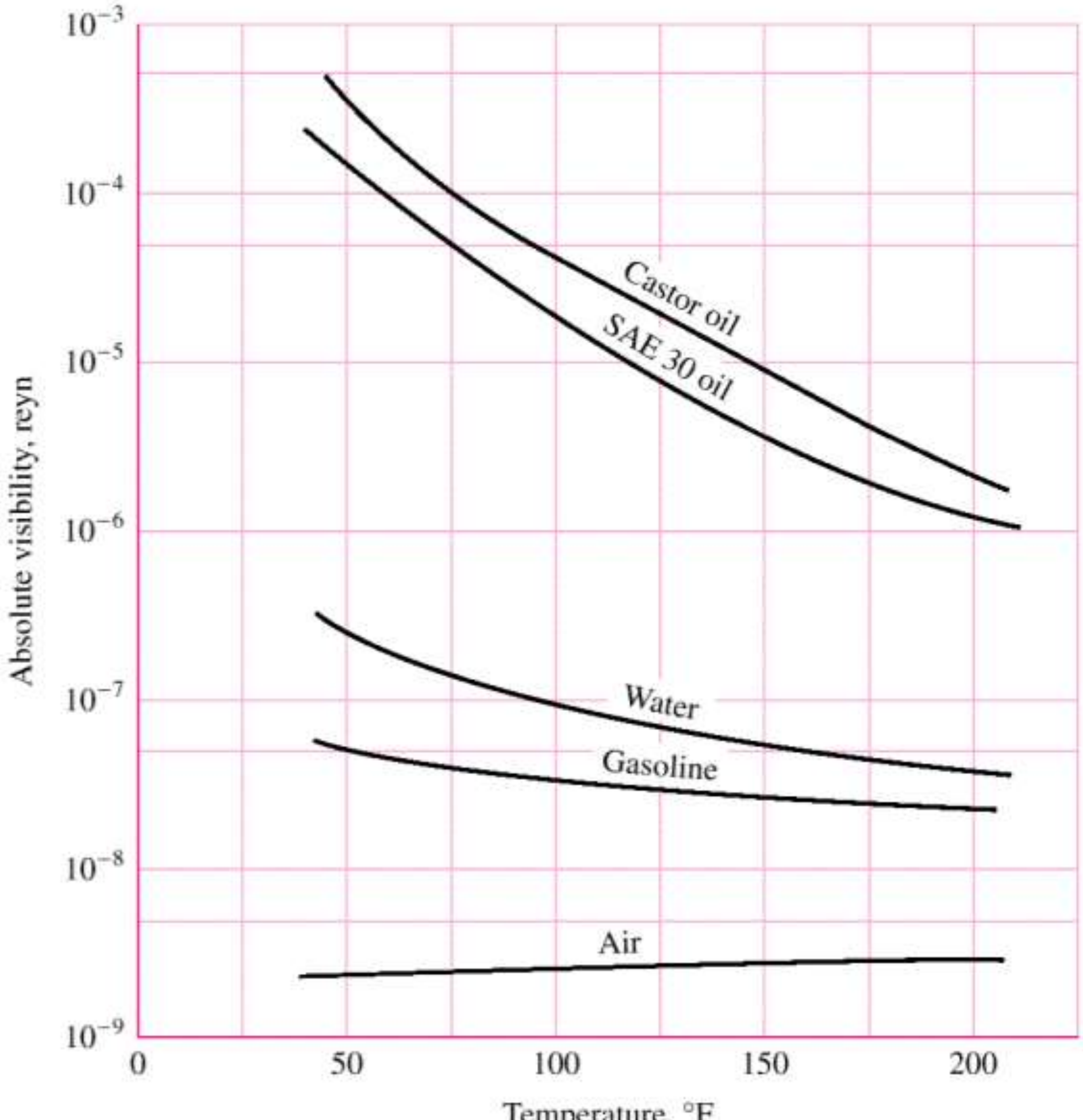
where μ is the constant of proportionality and defines *absolute viscosity,* also called *dynamic viscosity*. The derivative du/dy is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient. The viscosity μ is thus a measure of the internal frictional resistance of the fluid. If the assumption is made that the rate of shear is a constant, then $du/dy = U/h$, and from Eq. (12–1),

$$\tau = \frac{F}{A} = \mu \frac{U}{h} \tag{12–2}$$

The unit of viscosity in the ips system is seen to be the pound-force-second per square inch; this is the same as stress or pressure multiplied by time. The ips unit is called the *reyn,* in honor of Sir Osborne Reynolds.

The absolute viscosity is measured by the pascal-second ($\mathrm{Pa} \cdot \mathrm{s}$) in SI; this is the same as a Newton-second per square meter. The conversion from ips units to SI is the same as for stress. For example, multiply the absolute viscosity in reyns by 6890 to convert to units of $\mathrm{Pa} \cdot \mathrm{s}$.

Comparisons of Viscosities



A comparison of the viscosities of various fluids.

WHAT EVERY LUBRICANT MUST DO

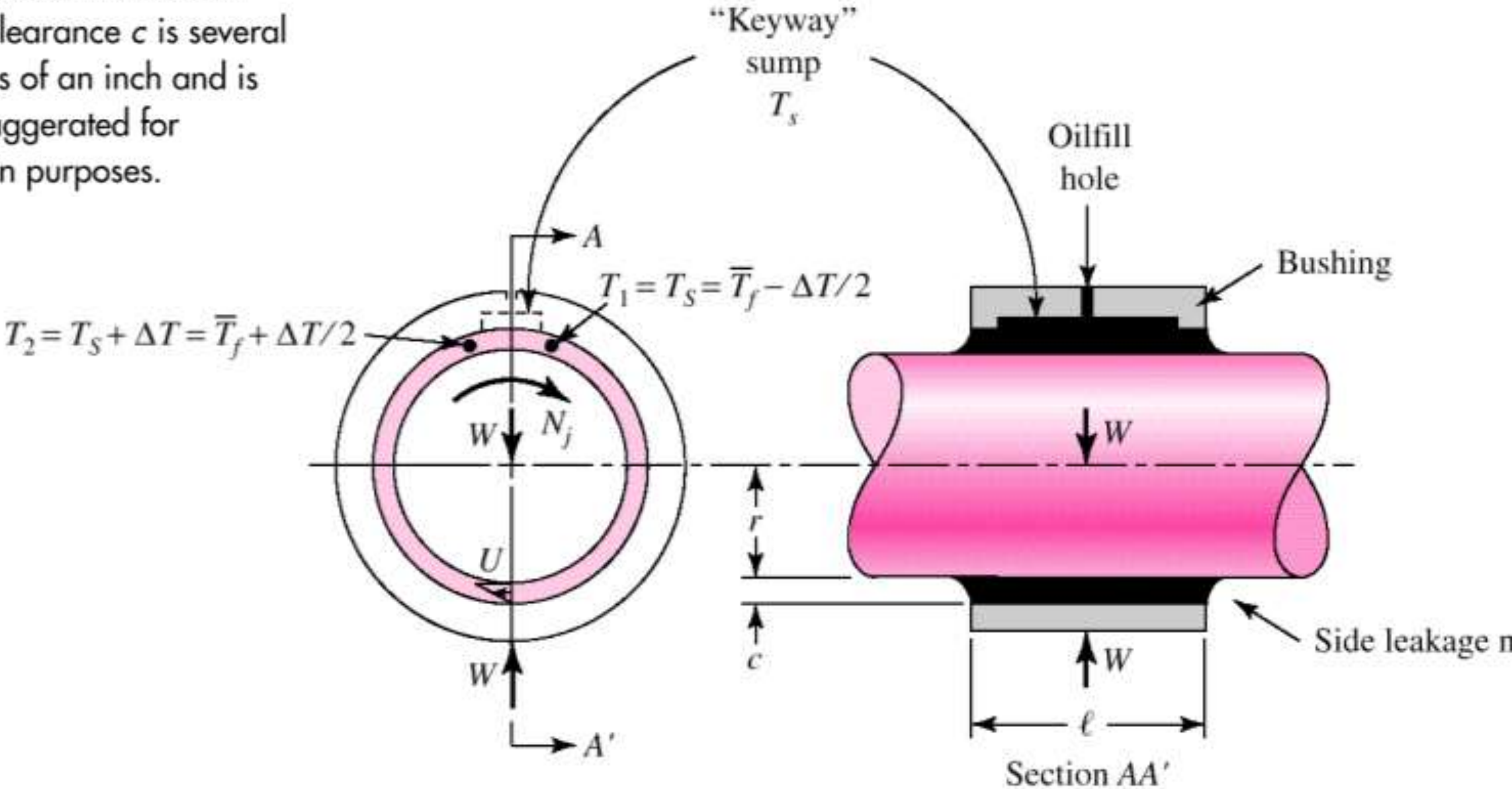
Through the ability to **minimize friction is the number one function of a lubricant**, there are other major functions that must be considered. A lubricant is likely to also be required to:

- **CLEAN**
- **COOL MOVING ELEMENTS**
- **PREVENT CONTAMINATION**
- **DAMPEN SHOCK**
- **TRANSFER ENERGY .**
- **PREVENT CORROSION**

Petroff's Equation

The phenomenon of bearing friction was first explained by Petroff using the assumption that the shaft is concentric. Though we shall seldom make use of Petroff's method of analysis in the material to follow, it is important because it defines groups of dimensionless parameters and because the coefficient of friction predicted by this law turns out to be quite good even when the shaft is not concentric.

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.



Let us now consider a vertical shaft rotating in a guide bearing. It is assumed that the bearing carries a very small load, that the clearance space c is completely filled with oil, and that leakage is negligible (Fig. 12–3). We denote the radius of the shaft by r, the radial clearance by c, and the length of the bearing by l, all dimensions being in inches. If the shaft rotates at N r/s, then its surface velocity is $U = 2\pi r N$ in/s. Since the shearing stress in the lubricant is equal to the velocity gradient times the viscosity, from Eq. (12–2) we have

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

where the radial clearance c has been substituted for the distance h. The force required to shear the film is the stress times the area. The torque is the force times the lever arm r. Thus

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

If we now designate a small force on the bearing by W, in pounds-force, then the pressure P, in pounds-force per square inch of projected area, is $P = W/2rl$. The frictional force is fW, where f is the coefficient of friction, and so the frictional torque is

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

Substituting the value of the torque from Eq. (c) in Eq. (b) and solving for the coefficient of friction, we find

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

Petroff's Equation

Equation (12–6) is called *Petroff's equation* and was first published in 1883. The two quantities $\mu N/P$ and r/c are very important parameters in lubrication. Substitution of the appropriate dimensions in each parameter will show that they are dimensionless.

Sommerfeld Number

The *bearing characteristic number,* or the *Sommerfeld number,* is defined by the equation

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

where S = bearing characteristic number
r = journal radius, in
c = radial clearance, in
μ = absolute viscosity, reyn
N = significant speed, r/s
P = load per unit of projected bearing area, psi

The Sommerfeld number is very important in lubrication analysis because it contains many of the parameters which specified by the designer. Note also that it is dimensionless. The quantity r/c is called the *radial clearance ratio*. If we multiply both sides of Eq. (12–6) by this ratio, we obtain the interesting relation

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

The dimensionless friction variable fr/c is a function of the Sommerfeld number. Rarely is the journal angular speed N in Eq. (12–8) subscripted. It should be. We will use N_j from now on. w

we examine Petroff's equation.

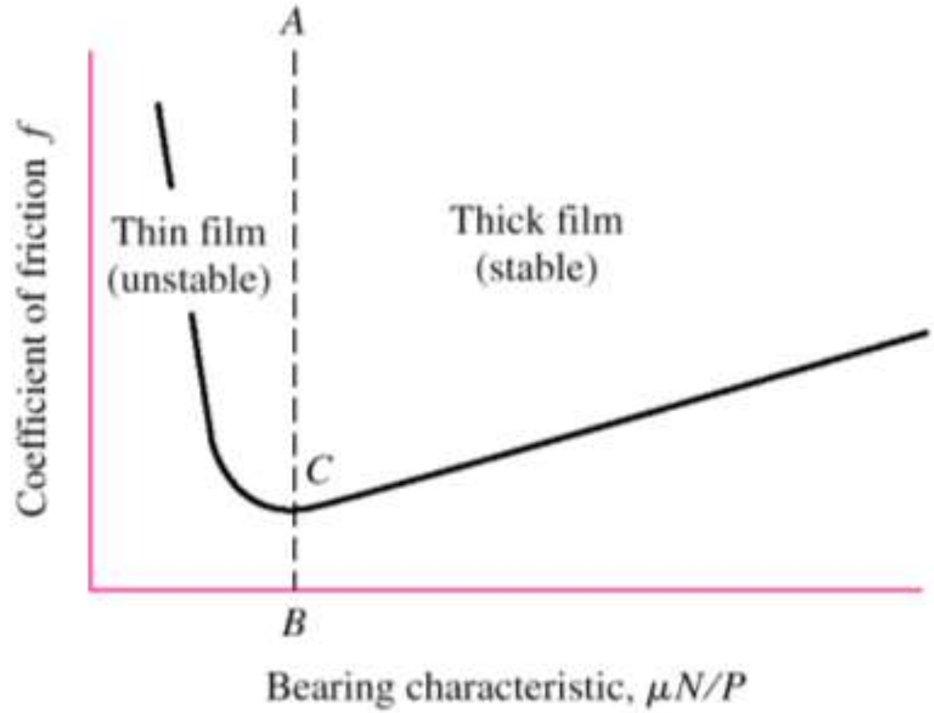
First, there are three dimensionless groups present:

$$f\frac{r}{c} \qquad \frac{\mu N_j}{P} \qquad \frac{r}{c}$$

Stable Lubrication

Petroff's bearing model in the form of Eq. (12–6) predicts

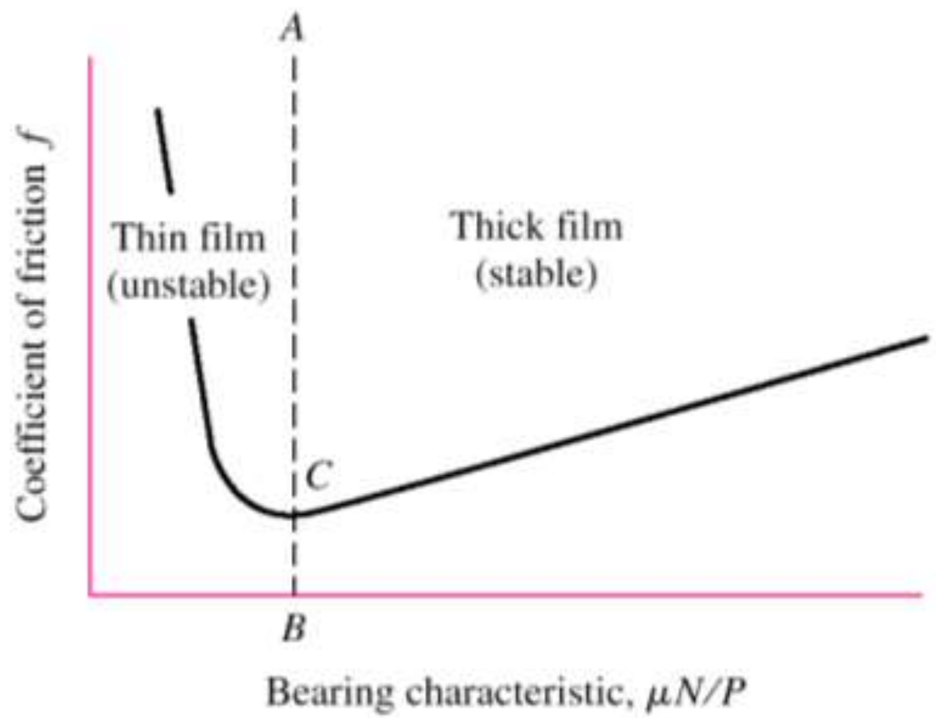
$$f = 2\pi^2 \left(\frac{\mu N_j}{P}\right)\left(\frac{r}{c}\right)$$



for a given r and c, and that f is proportional to $\mu N_j / P$, that is, a straight line from the origin in the first quadrant. On the coordinates of Fig. 12–4 the locus to the right of point C is an example. Petroff's model presumes thick-film lubrication, that is, no metal-to-metal contact, the surfaces being completely separated by a lubricant film. Conditions under which the thick film would fail were reported in 1932.

The difference between boundary and hydrodynamic lubrication can be explained by reference to Fig. 12–4. This plot of the change in the coefficient of friction versus the bearing characteristic $\mu N/P$ was obtained by the McKee brothers in an actual test of friction.[5] The plot is important because it defines stability of lubrication and helps us to understand hydrodynamic and boundary, or thin-film, lubrication.

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



$$\boxed{\frac{\mu N_j}{P} \geq 17(10^{-6})}$$

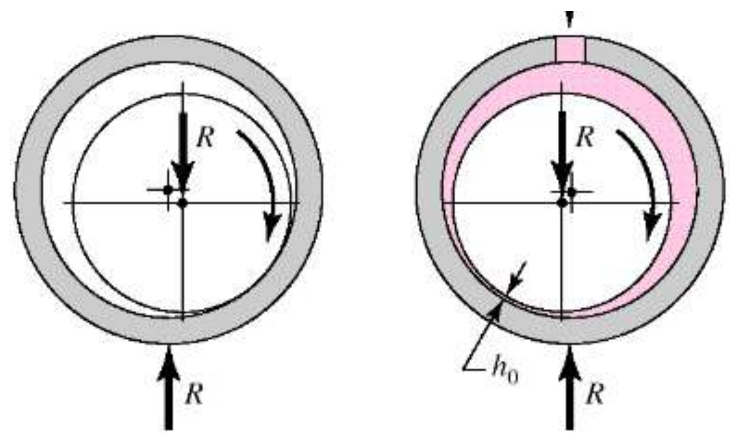
One of the design Constraint to keep thick film lubrication

Thick-Film Lubrication

Let us now examine the formation of a lubricant film in a journal bearing. Figure 12–5*a* shows a journal which is just beginning to rotate in a clockwise direction. Under starting conditions, the bearing will be dry, or at least partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig. 12–5*a*. Under the conditions of a dry bearing, equilibrium will be obtained when the friction force is balanced by the tangential component of the bearing load.

Now suppose a lubricant is introduced into the top of the bearing as shown in Fig. 12–5*b*. The action of the rotating journal is to pump the lubricant around the bearing in a

Formation of a film.

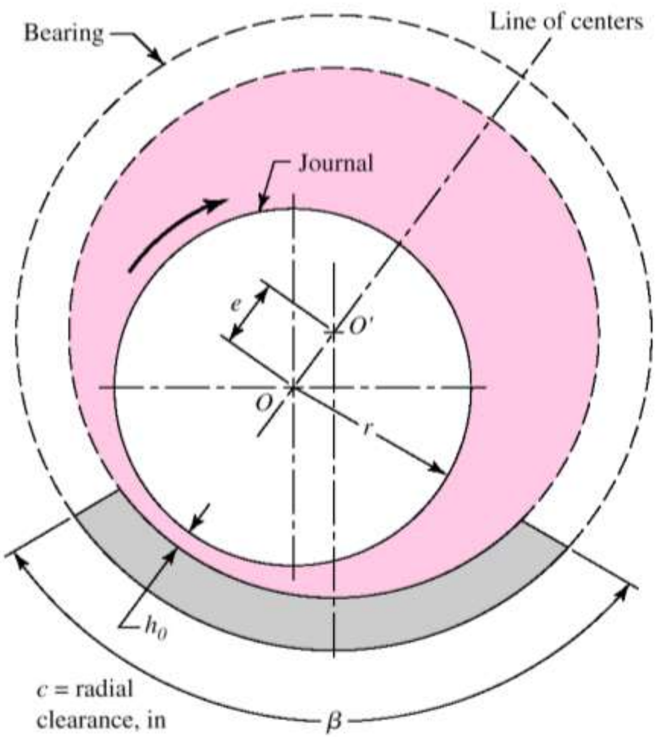


(*a*) Dry (*b*) Lubricated

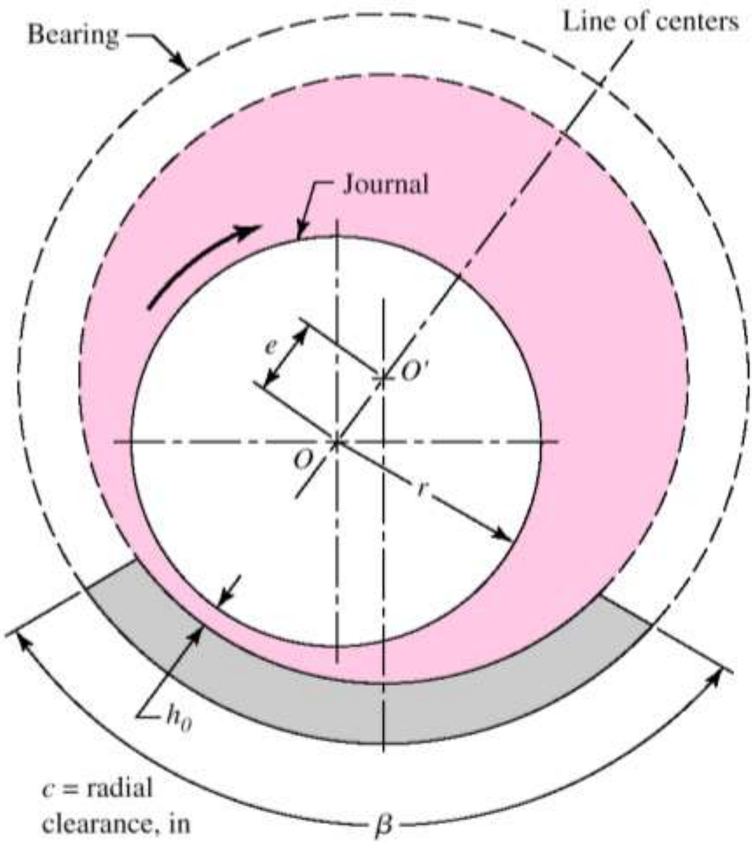
Nomenclature of Journal Bearing

The nomenclature of a journal bearing is shown in Fig. 12–6. The dimension c is the *radial clearance* and is the difference in the radii of the bushing and journal. In Fig. 12–6 the center of the journal is at O and the center of the bearing at O'. The distance between these centers is the *eccentricity* and is denoted by e. The *minimum film thickness* is designated by h_0, and it occurs at the line of centers. The film thickness at any other point is designated by h. We also define an *eccentricity ratio* ϵ as

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

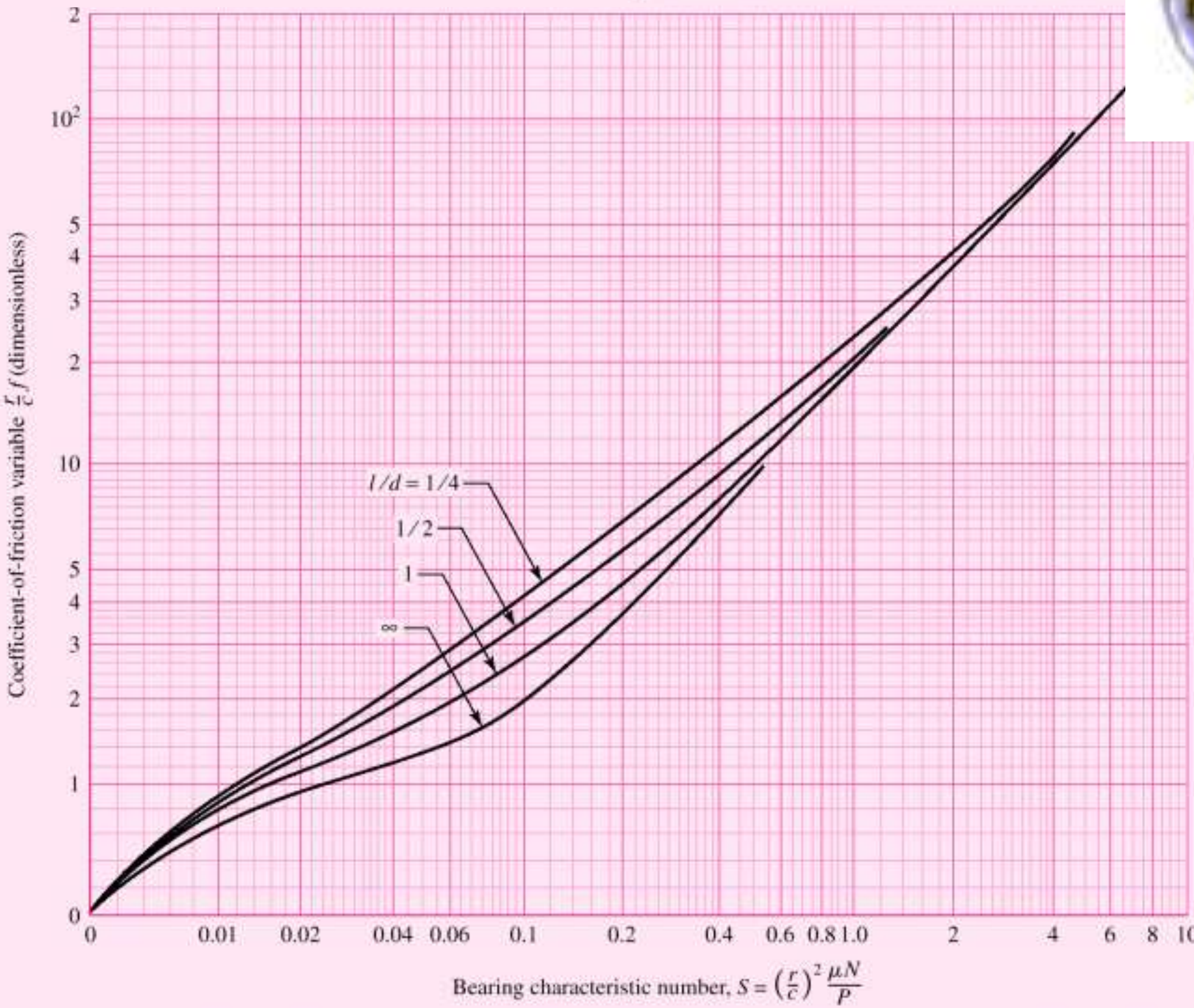


Nomenclature of Journal Bearing



The bearing shown in the figure is known as a *partial bearing*. If the radius of the bushing is the same as the radius of the journal, it is known as a *fitted bearing*. If the bushing encloses the journal, as indicated by the dashed lines, it becomes a *full bearing*. The angle β describes the angular length of a partial bearing. For example, a 120° partial bearing has the angle β equal to 120°.

Chart for coefficient-of-friction variable;



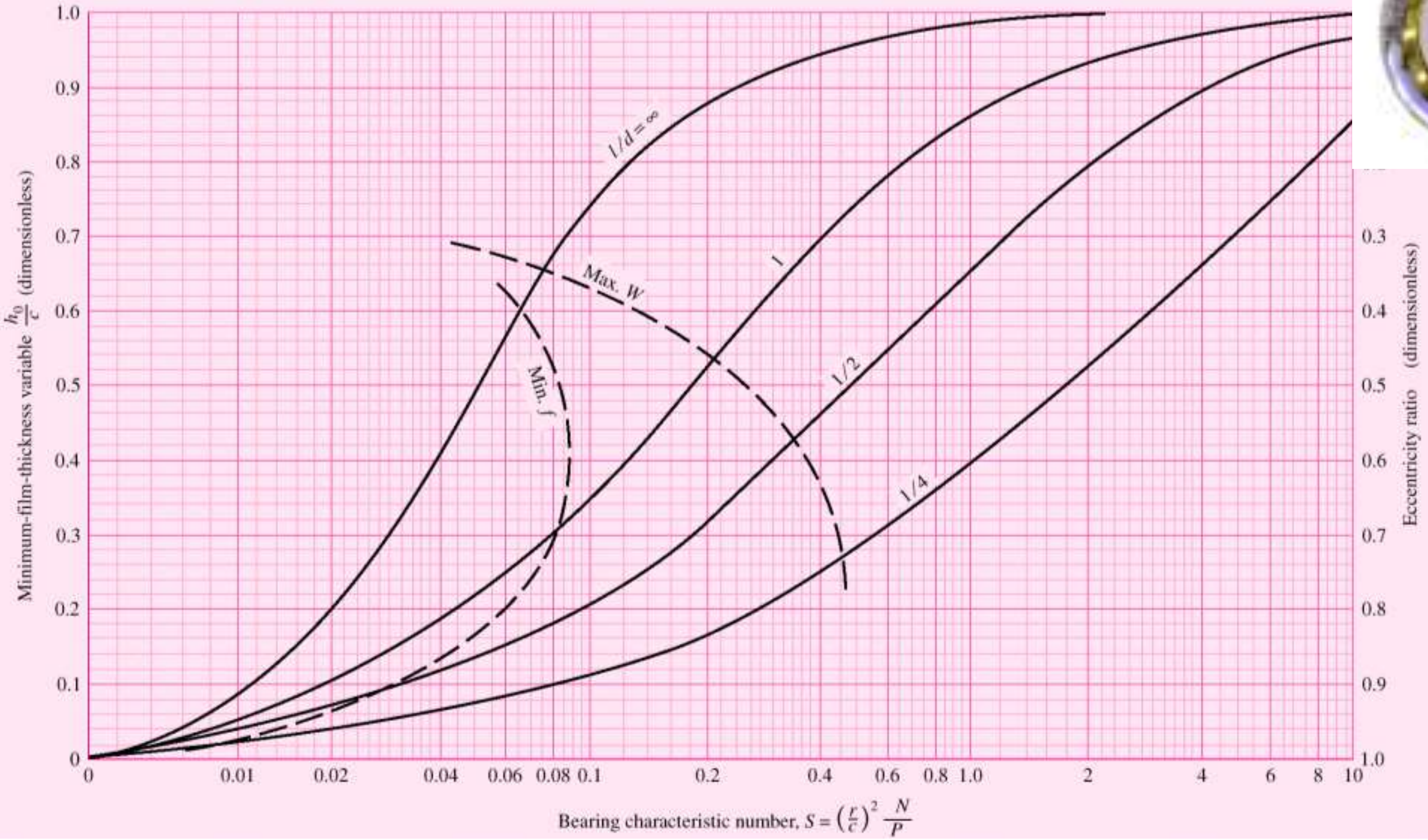
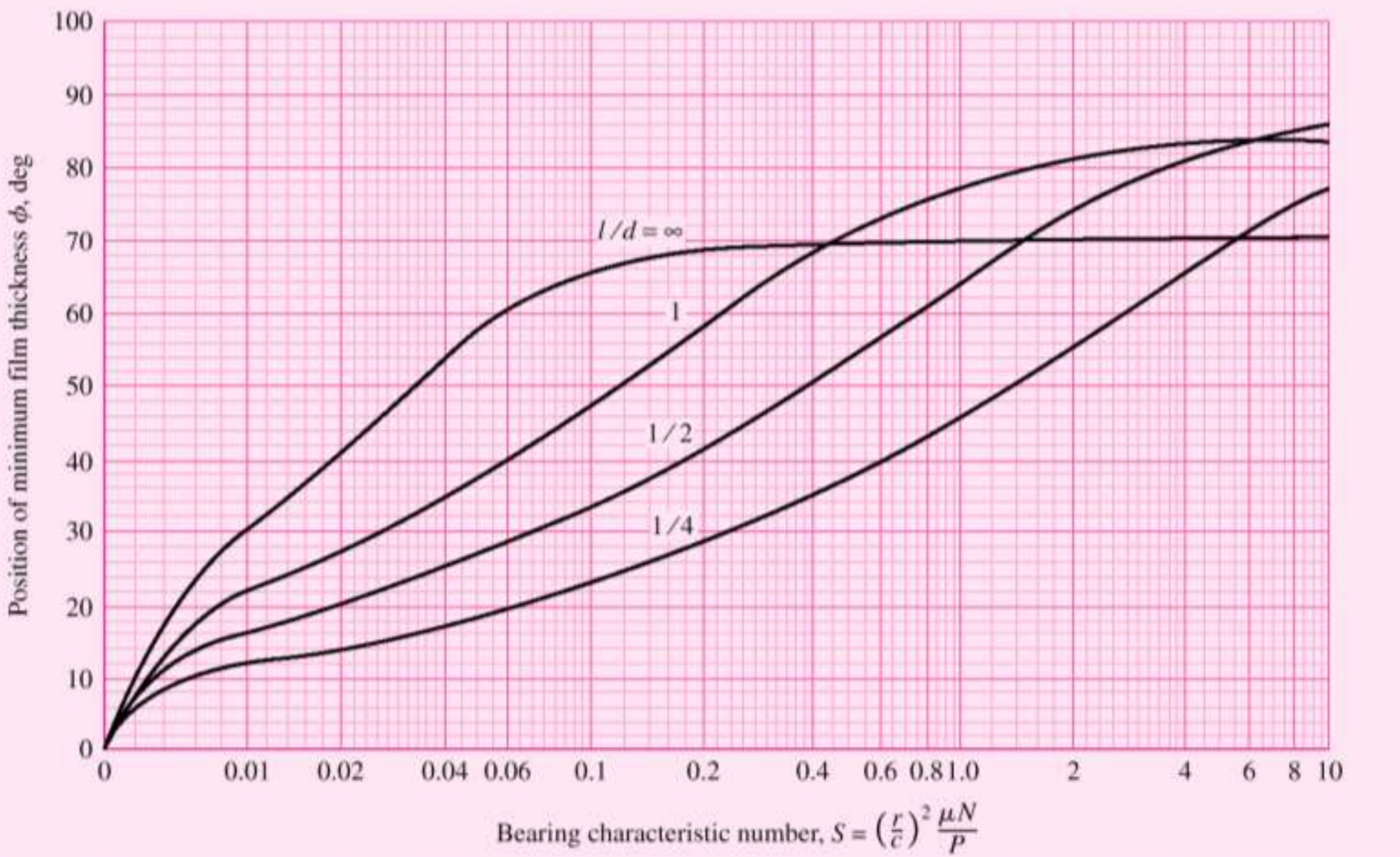


Chart for minimum film-thickness variable and eccentricity ratio. The left boundary of the zone defines the optimal h_0 for minimum friction; the right boundary is optimum h_0 for load. *(Raimondi and Boyd.)*

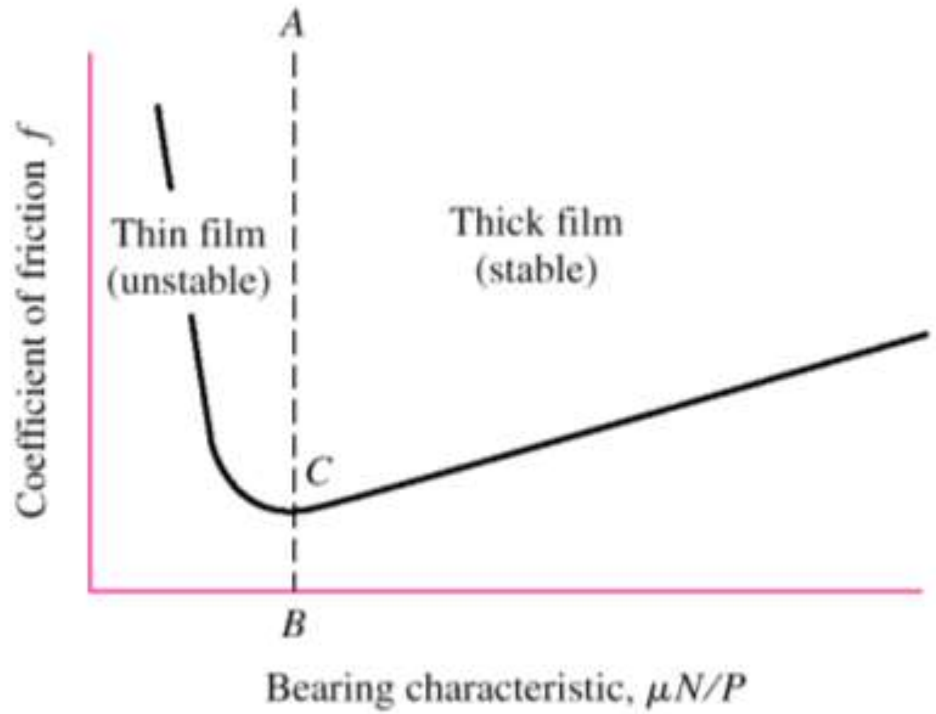
Chart for determining the position of the minimum film thickness h_0. For location of the origin, see Fig. 12–16. *(Raimondi and Boyd.)*



Stability

The difference between boundary and hydrodynamic lubrication can be explained by reference to Fig. 12–4. This plot of the change in the coefficient of friction versus the bearing characteristic $\mu N/P$ was obtained by the McKee brothers in an actual test of friction.[5] The plot is important because it defines stability of lubrication and helps us to understand hydrodynamic and boundary, or thin-film, lubrication.

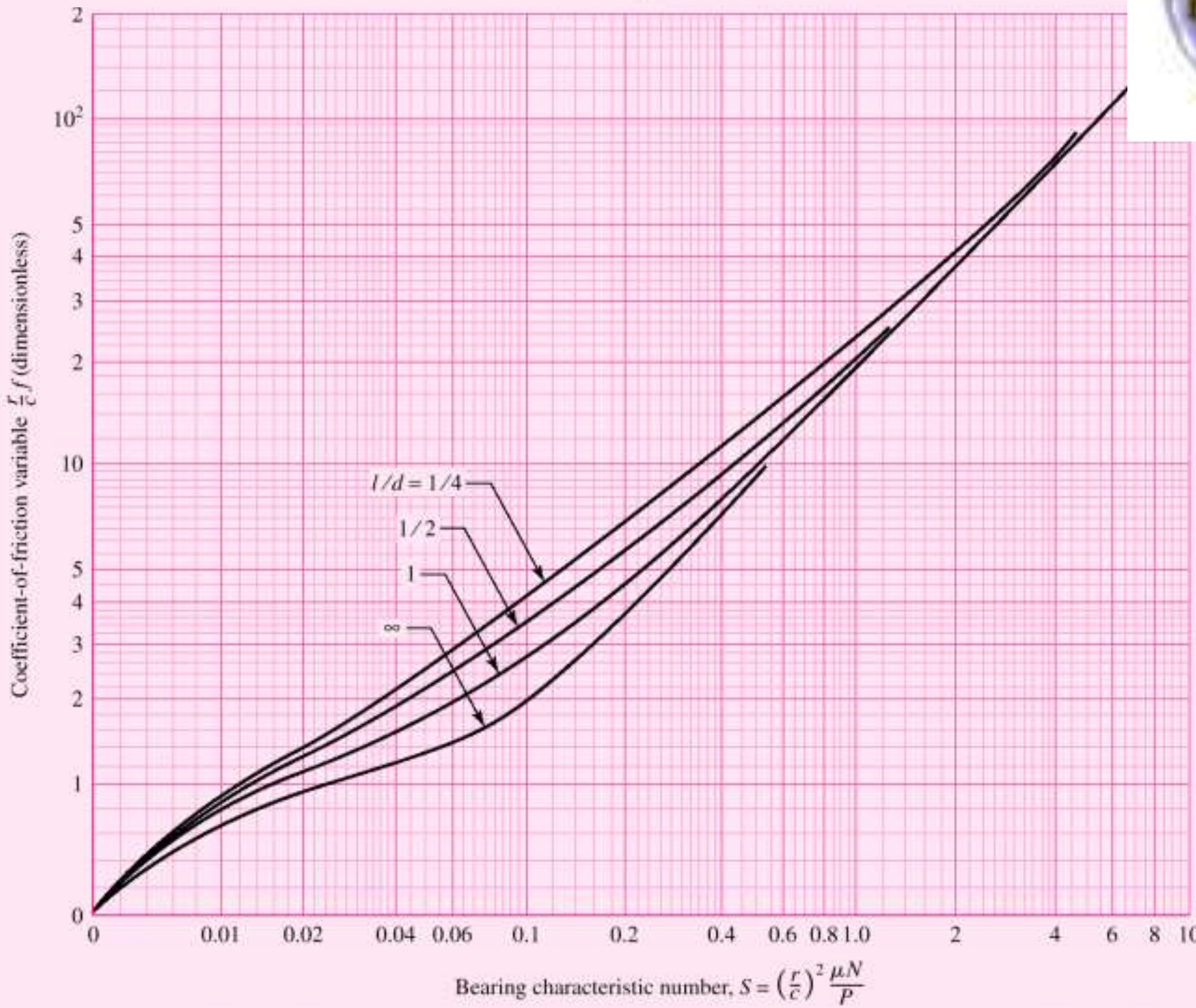
The variation of the coefficient of friction f with $\mu N/P$.

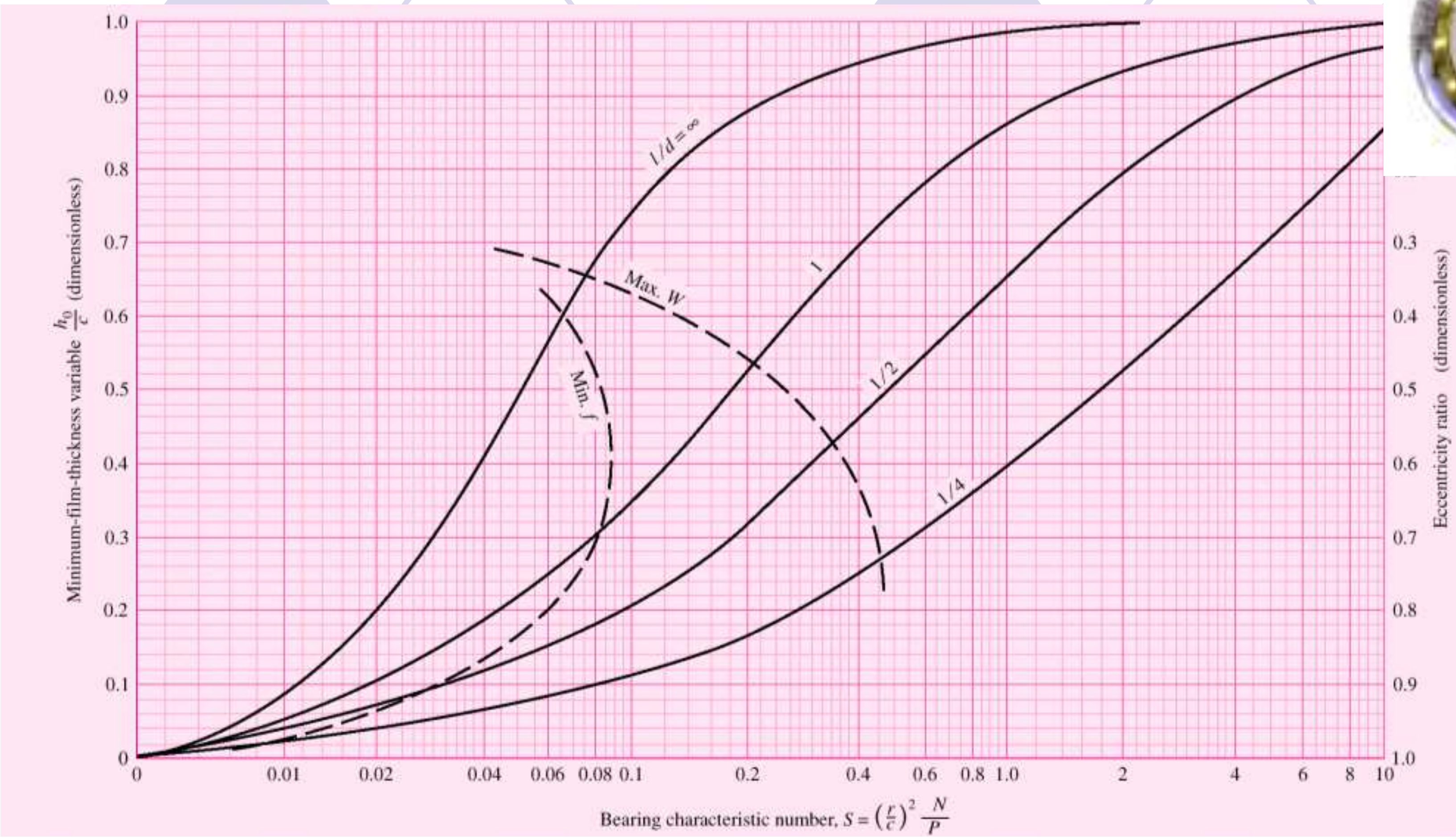


$$\frac{\mu N_j}{P} \geq 17(10^{-6})$$

One of the design Constraint to keep thick film lubrication

Chart for coefficient-of-friction variable;





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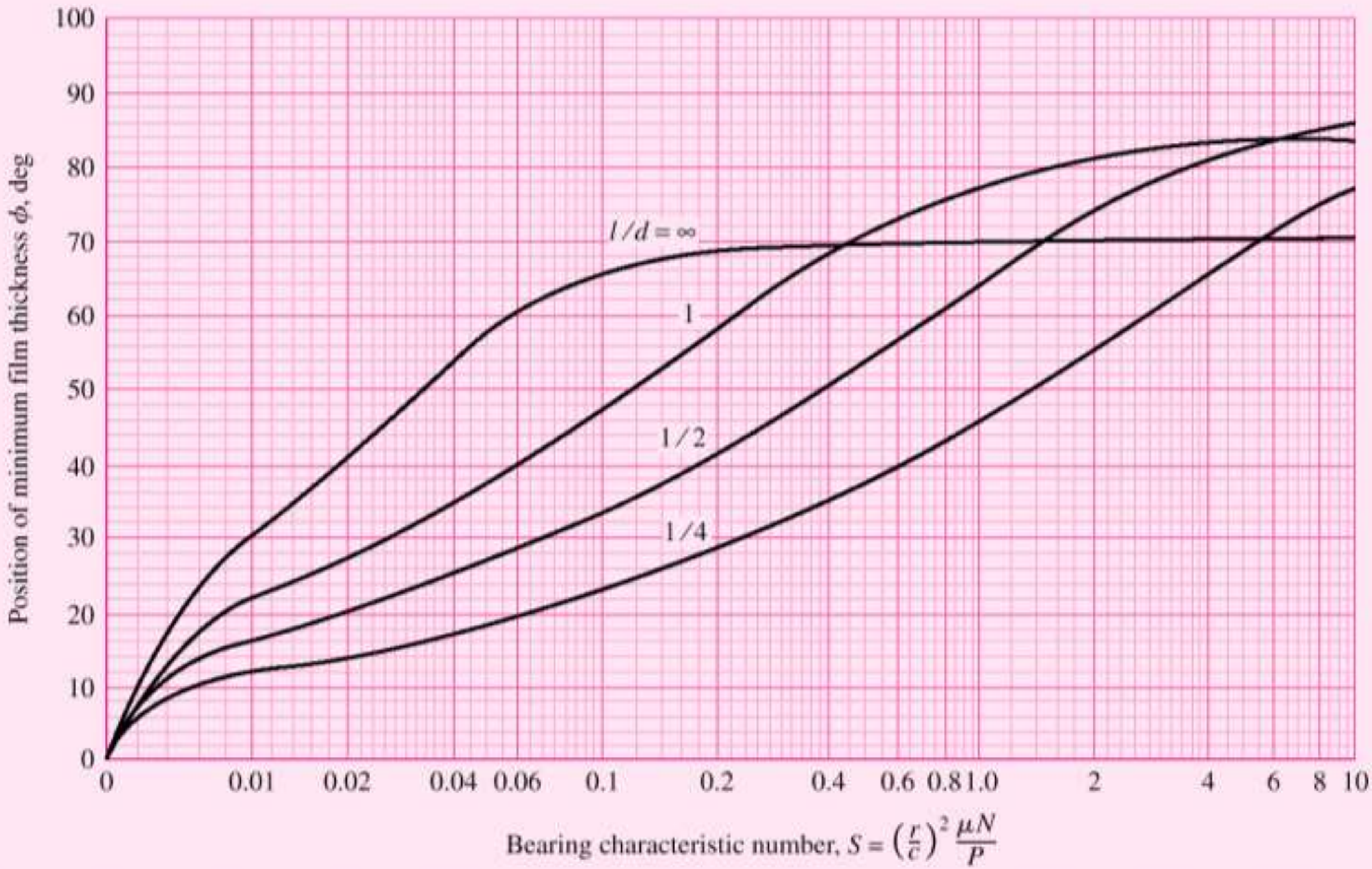


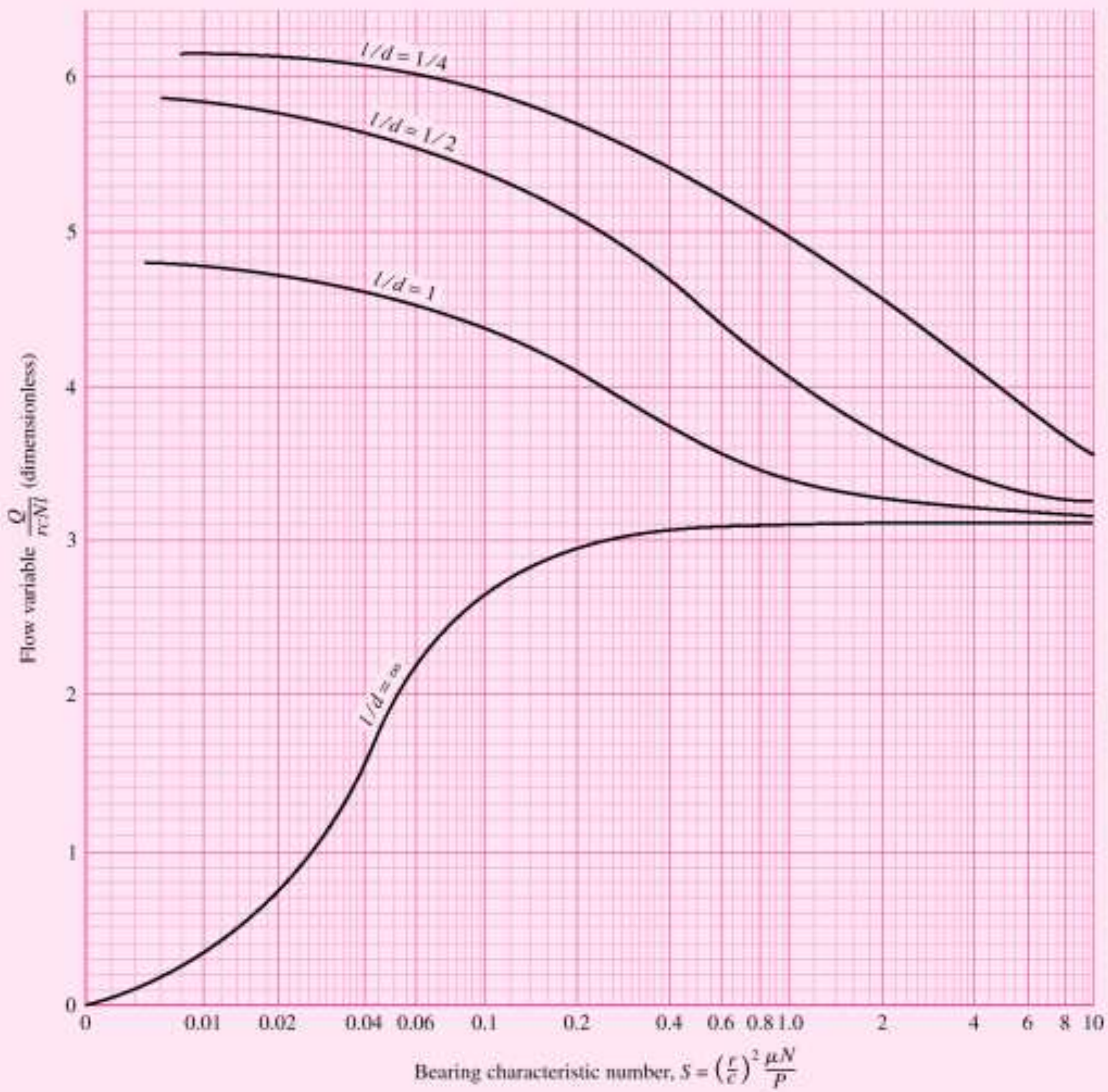


Figure 12–18

Chart for flow variable. *(Raimondi and Boyd.)*

Figure 12–19

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

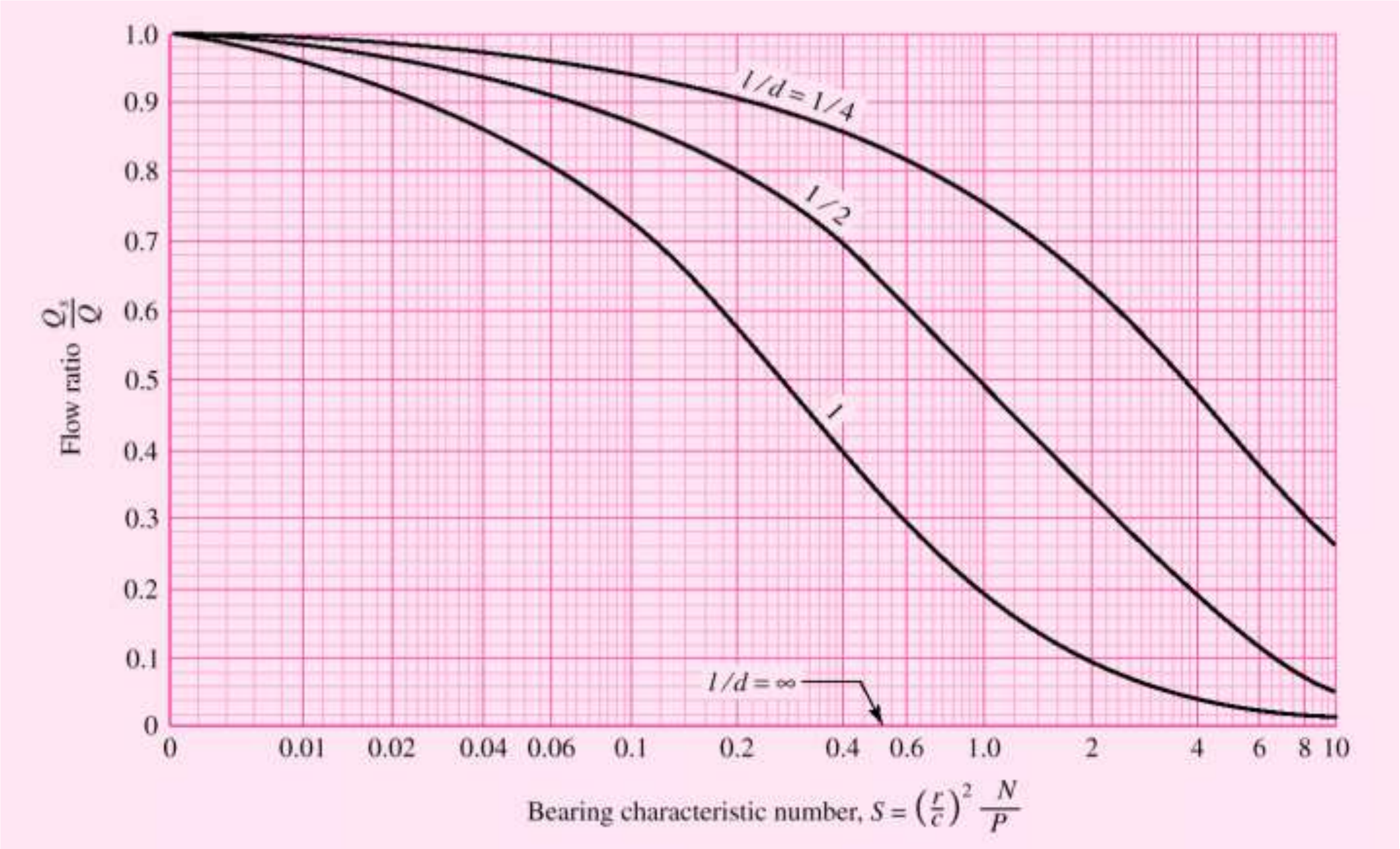


Figure 12–20

Chart for determining the maximum film pressure. *(Raimondi and Boyd.)*

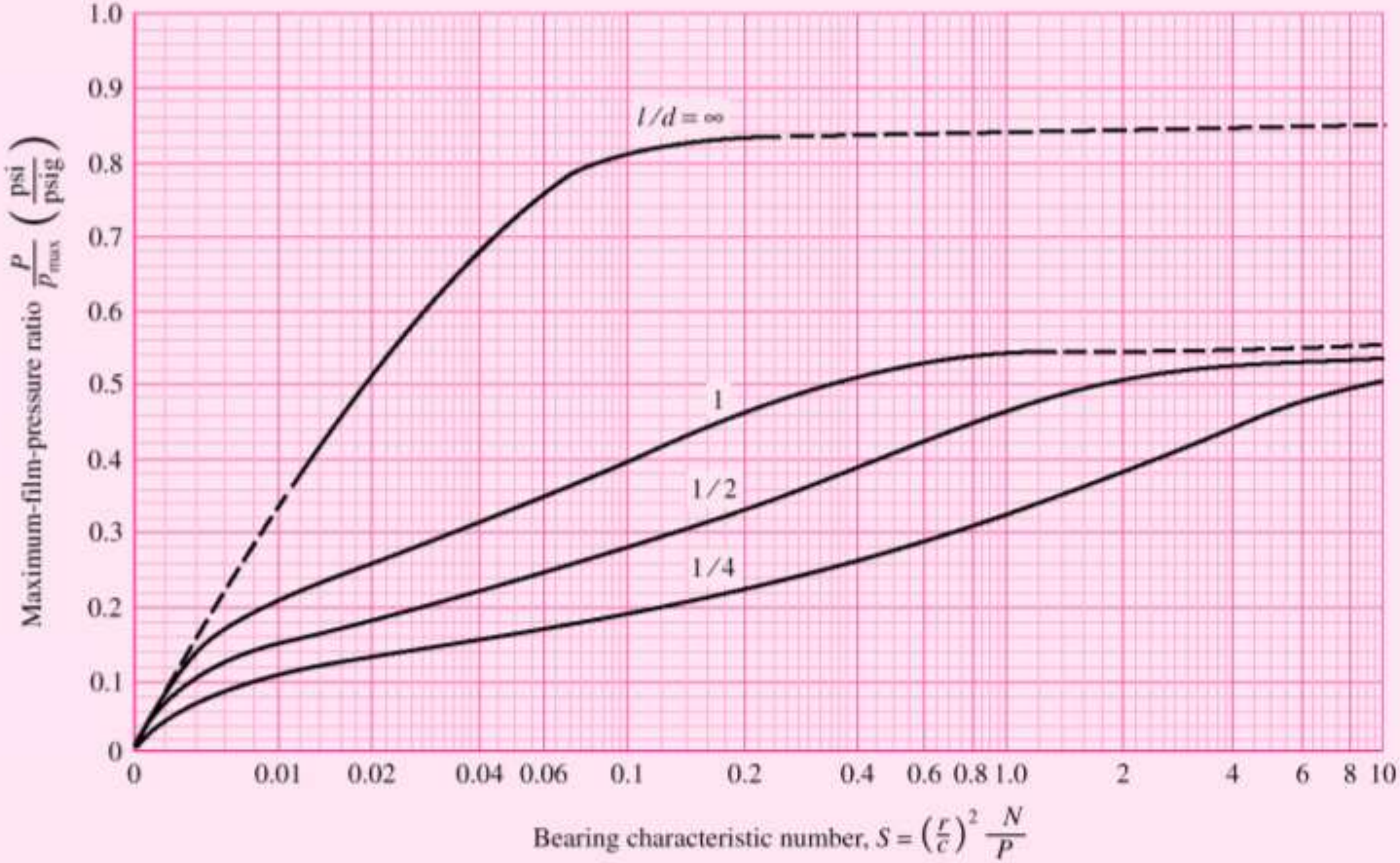
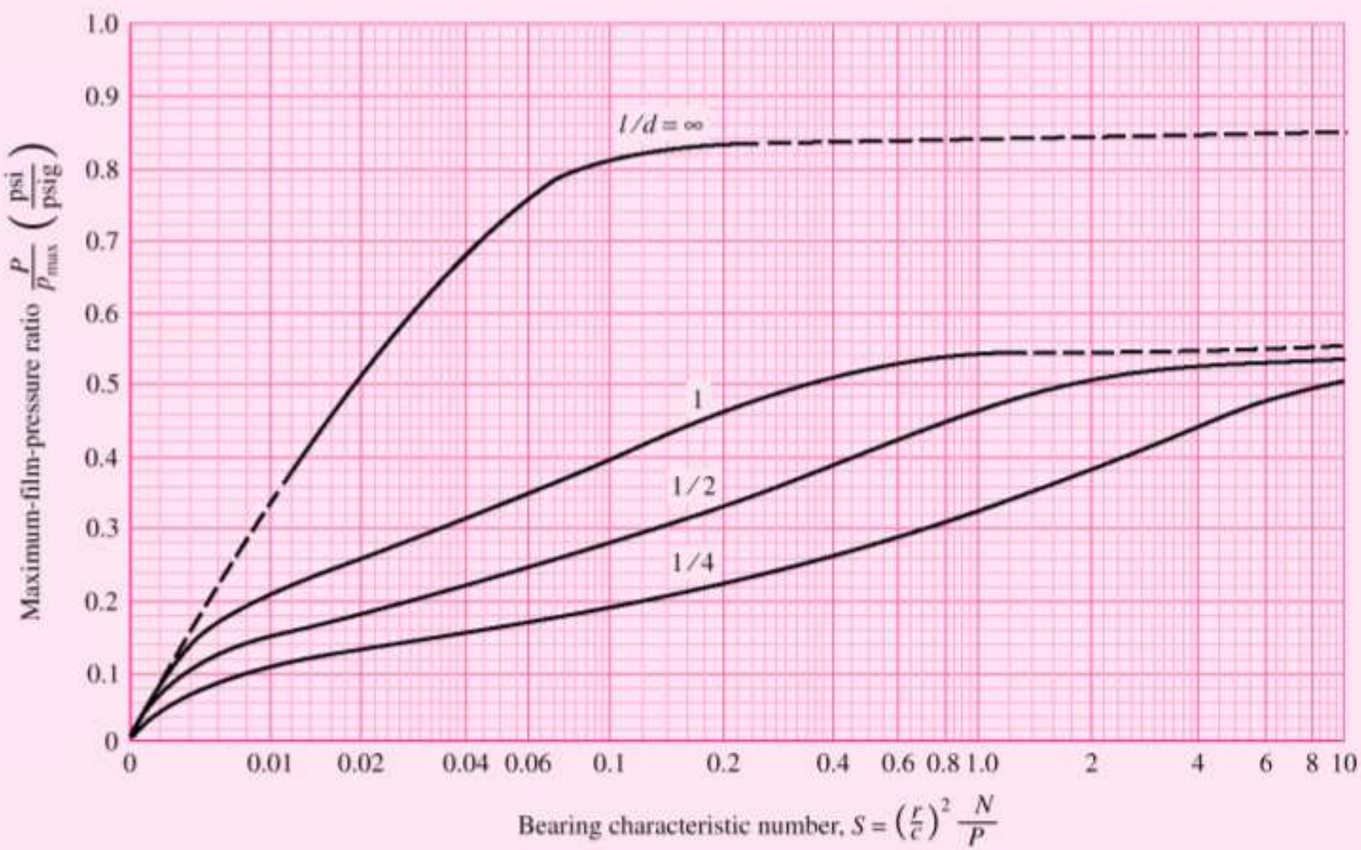


Chart for determining the maximum film pressure. *(Raimondi and Boyd.)*



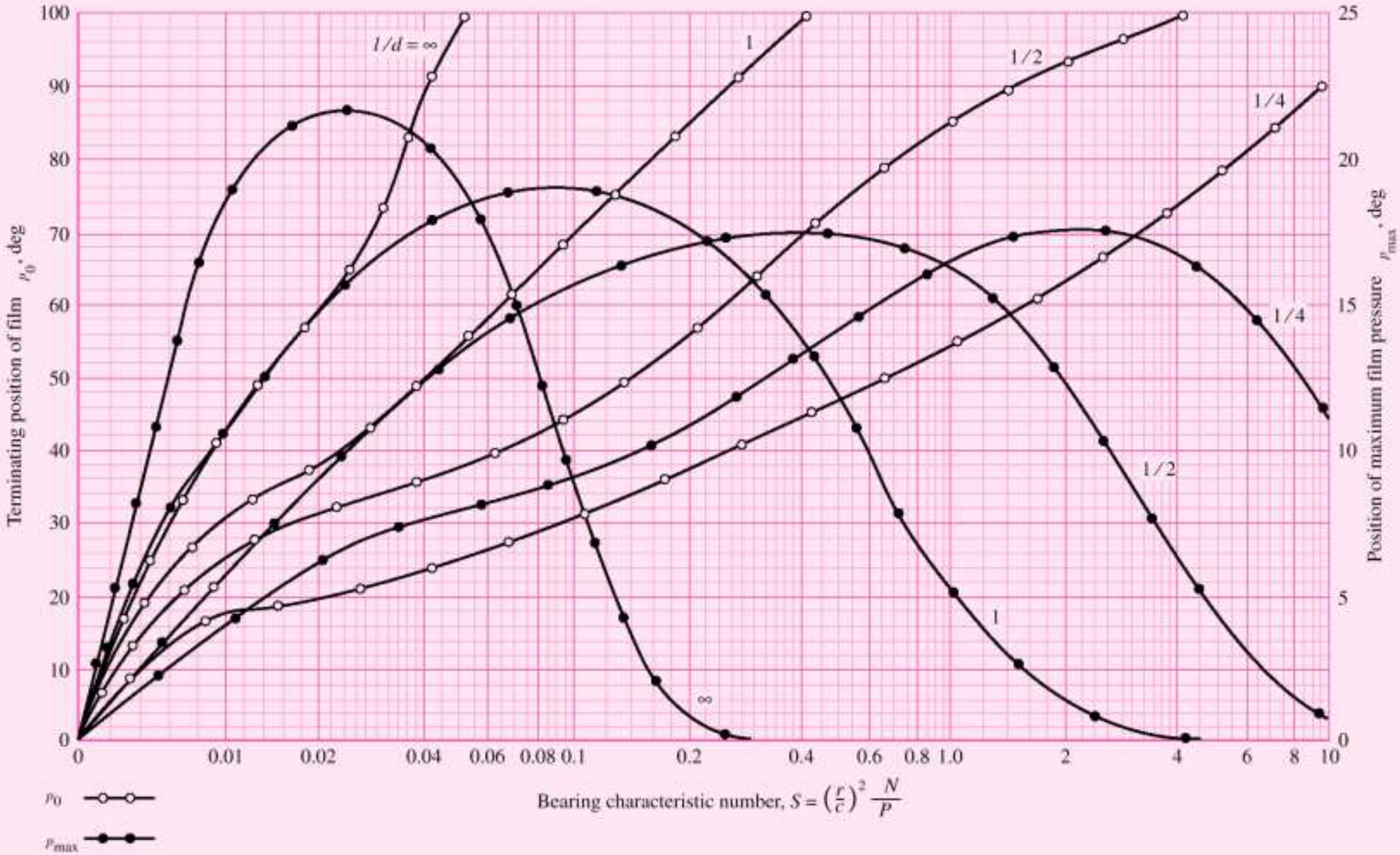


Figure 12–21

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

Example 12-1

Determine h_0 and e using the following given parameters:

$$P = \frac{W}{2rl}$$

$$\mu = 4\ \mu\text{reyns}$$
$$N_j = 30\ \text{r/s}$$
$$W = 500\ \text{lb (bearing load)}$$

$$r = 0.75\ \text{in}$$
$$c = 0.0015\ \text{in}$$
$$l = 1.5\ \text{in}$$

The Sommerfeld number is,

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N_j}{P}\right)$$

Example 12-2

Using the parameters example 12-1. determine the coefficient of friction , the torque to overcome friction and the power loss to friction

$\mu = 4\ \mu\text{reyns}$

$N_j = 30\ \text{r/s}$

$W = 500\ \text{lb (bearing load)}$

$r = 0.75\ \text{in}$

$c = 0.0015\ \text{in}$

$l = 1.5\ \text{in}$

$f = 3.50c/r$

l/d

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N_j}{P}\right)$$

$$T = fWr$$

$$(hp)_{\text{loss}} = \frac{T N_j}{1050}$$

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

$$\mu = 4\ \mu\text{reyns} \qquad r = 0.75\ \text{in}$$
$$N_j = 30\ \text{r/s} \qquad c = 0.0015\ \text{in}$$
$$W = 500\ \text{lb (bearing load)} \qquad l = 1.5\ \text{in}$$

Example 12-4

Continuing with the parameters of EX. 12-1 Determine the Maximum film pressure and the locations of the maximum film pressure and terminating pressures

$\mu = 4\ \mu\text{reyns}$

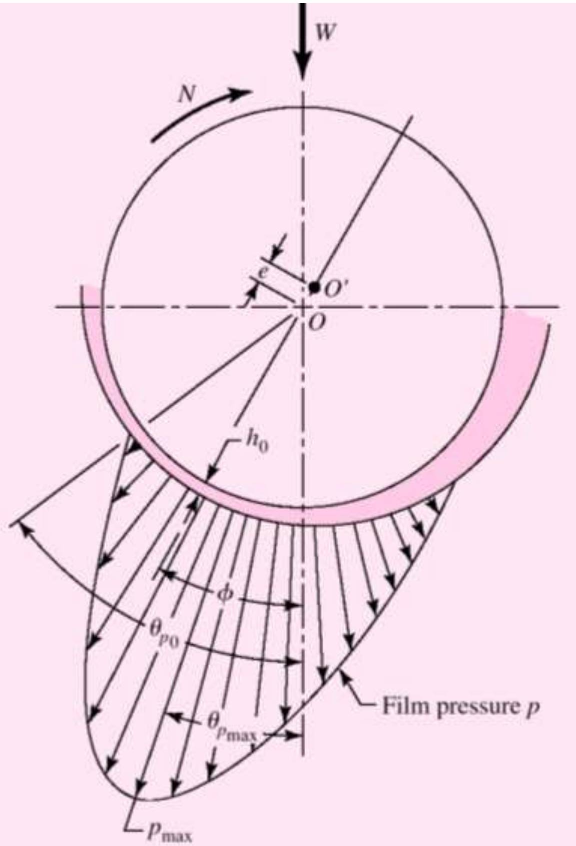
$N_j = 30\ \text{r/s}$

$W = 500\ \text{lb (bearing load)}$

$r = 0.75\ \text{in}$

$c = 0.0015\ \text{in}$

$l = 1.5\ \text{in}$



Problem 12- 1)

Journal Bearing Lubrications)

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A full journal bearing has a journal diameter of 1.000 in, with a unilateral tolerance of −0.0015 in. The bushing bore has a diameter of 1.0015 in and a unilateral tolerance of 0.003 in. The l/d ratio is unity. The load is 250 lbf and the journal runs at 1100 rev/min. If the average viscosity is 8 μreyn, find the minimum film thickness, the power loss, and the side flow for the minimum clearance assembly.

Formulas:

$$T = fWr$$

$$S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N_j}{P}\right)$$

$$P = \frac{W}{2rl}$$

$$(hp)_{\text{loss}} = \frac{TN_j}{1050}$$

Solution:

The power loss in Btu/s is $H = \dfrac{2\pi fWrN}{778(12)} =$

Given $d_{max} = 1.000$ in and $b_{min} = 1.0015$ in, the minimum radial clearance is

$$c_{min} = \frac{b_{min} - d_{max}}{2} = \frac{1.0015 - 1.000}{2} = 0.000\,75 \text{ in}$$

Also

$$l/d = 1$$

$$r \doteq 1.000/2 = 0.500$$

$$r/c = 0.500/0.000\,75 = 667$$

$$N = 1100/60 = 18.33 \text{ rev/s}$$

$$P = W/(ld) = 250/[(1)(1)] = 250 \text{ psi}$$

Eq. (12-7):

$$S = (667^2)\left[\frac{8(10^{-6})(18.33)}{250}\right] = 0.261$$

Fig. 12-16: $$h_0/c = 0.595$$

Fig. 12-19: $$Q/(rcNl) = 3.98$$

Fig. 12-18: $$fr/c = 5.8$$

Fig. 12-20: $$Q_s/Q = 0.5$$

$$h_0 = 0.595(0.000\,75) = 0.000\,466 \text{ in} \quad Ans.$$

$$f = \frac{5.8}{r/c} = \frac{5.8}{667} = 0.0087$$

The power loss in Btu/s is

$$H = \frac{2\pi f W r N}{778(12)} = \frac{2\pi(0.0087)(250)(0.5)(18.33)}{778(12)}$$

$$= 0.0134 \text{ Btu/s} \quad Ans.$$

$$Q = 3.98rcNl = 3.98(0.5)(0.000\,75)(18.33)(1) = 0.0274 \text{ in}^3\text{/s}$$

$$Q_s = 0.5(0.0274) = 0.0137 \text{ in}^3\text{/s} \quad Ans.$$

12-2 A full journal bearing has a journal diameter of 1.250 in, with a unilateral tolerance of −0.001 in. The bushing bore has a diameter of 1.252 in and a unilateral tolerance of 0.003 in. The bearing is 2.5 in long. The journal load is 400 lbf and it runs at a speed of 1150 rev/min. Using an average viscosity of 10 μreyn find the minimum film thickness, the maximum film pressure, and the total oil-flow rate for the minimum clearance assembly.

Design Considerations

We may distinguish between two groups of variables in the design of sliding bearings In the first group are those whose values either are given or are under the control of the designer. These are:

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

Of these four variables, the designer usually has no control over the speed, because it is specified by the overall design of the machine. Sometimes the viscosity is specified in advance, as, for example, when the oil is stored in a sump and is used for lubricating and cooling a variety of bearings. The remaining variables, and sometimes the viscosity, may be controlled by the designer and are therefore the *decisions* he or she makes. In other words, when these four decisions are made, the design is complete.

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 ΔT
3 The flow of oil Q
4 The minimum film thickness h_0

This group of variables tells us how well the bearing is performing, and hence we may regard them as *performance factors*. Certain limitations on their values must be imposed by the designer to ensure satisfactory performance. These limitations are specified by the characteristics of the bearing materials and of the lubricant. The fundamental problem in bearing design, therefore, is to define satisfactory limits for the second group of variables and then to decide upon values for the first group such that these limitations are not exceeded.

Trumpler's Design Criteria for Journal Bearings

Because the bearing assembly creates the lubricant pressure to carry a load, it reacts to loading by changing its eccentricity, which reduces the minimum film thickness h_0 until the load is carried. What is the limit of smallness of h_0?

$$h_0 \geq 0.0002 + 0.000\,04D \text{ in}$$

where D is the journal diameter in inches.

Trumpler limits the maximum film temperature T_{max} to

$$T_{max} \leq 250°\text{F}$$

The starting load divided by the projected area is limited to

$$\frac{W_{st}}{lD} \leq 300 \text{ psi}$$

$$n_d \geq 2$$

Viscosity Charts

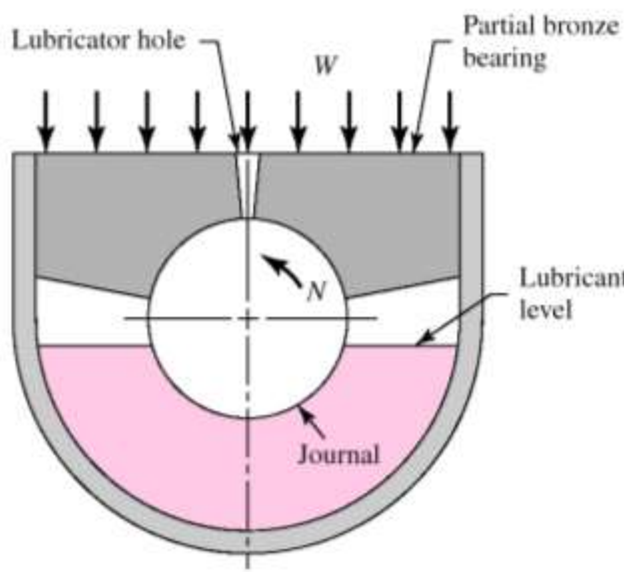
$$T_{av} = T_1 + \frac{\Delta T}{2}$$ Eq. (12–13),

where T_1 is the inlet temperature and ΔT is the temperature rise of the lubricant from inlet to outlet. Of course, the viscosity used in the analysis must correspond to T_{av}.

Hydrodynamic Theory

The present theory of hydrodynamic lubrication originated in the laboratory of Beauchamp Tower in the early 1880s in England. Tower had been employed to study the friction in railroad journal bearings and learn the best methods of lubricating them. It

Schematic representation of the partial bearing used by Tower.



Assumptions

Reynolds pictured the lubricant as adhering to both surfaces and being pulled by the moving surface into a narrowing, wedge-shaped space so as to create a fluid or film pressure of sufficient intensity to support the bearing load. One of the important simplifying assumptions resulted from Reynolds' realization that the fluid films were so thin in comparison with the bearing radius that the curvature could be neglected. This enabled him to replace the curved partial bearing with a flat bearing, called a *plane slider*

Other assumptions made were:

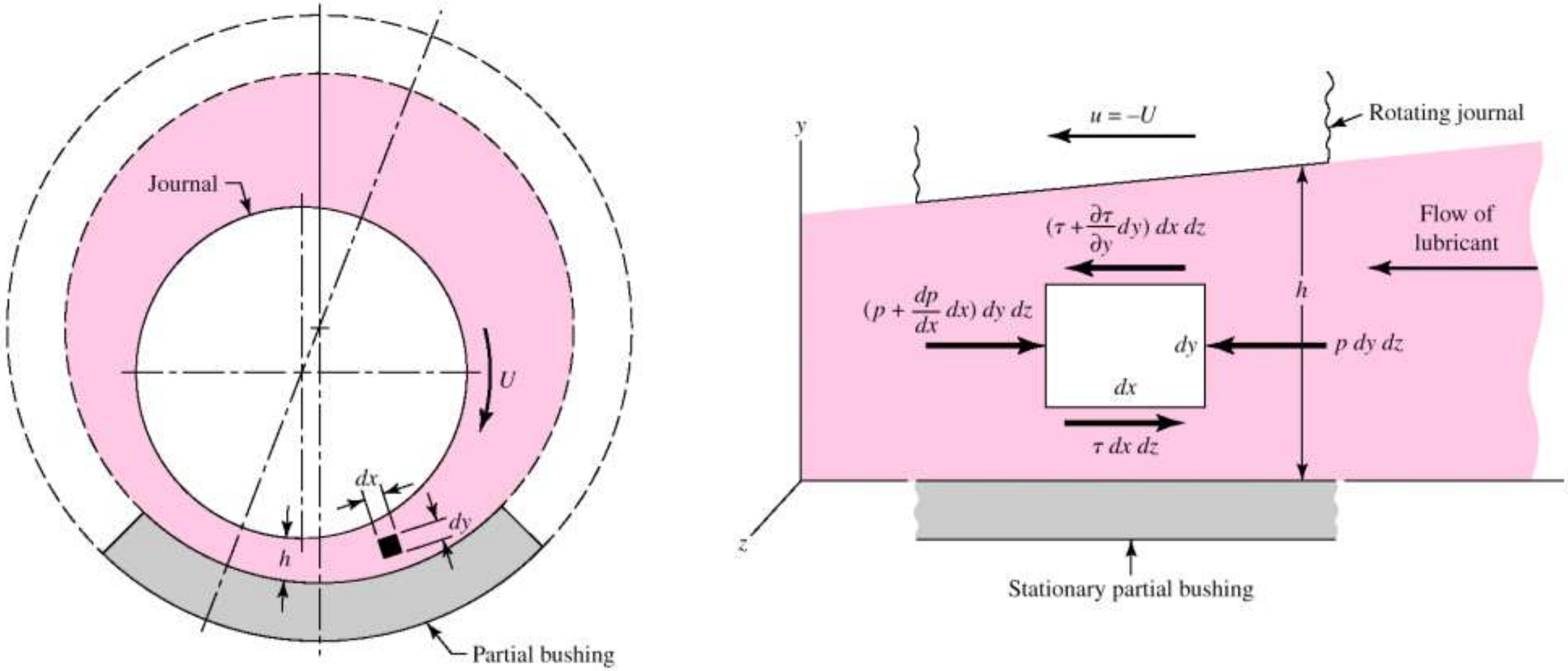
1. The lubricant obeys Newton's viscous effect, Eq. (12–1).
2. The forces due to the inertia of the lubricant are neglected.
3. The lubricant is assumed to be incompressible.
4. The viscosity is assumed to be constant throughout the film.
5. The pressure does not vary in the axial direction.

We now make the following additional assumptions:

6 The bushing and journal extend infinitely in the z direction; this means there can be no lubricant flow in the z direction.

7 The film pressure is constant in the y direction. Thus the pressure depends only on the coordinate x.

8 The velocity of any particle of lubricant in the film depends only on the coordinates x and y.


Journal
U
dx
dy
h
Partial bushing
u = −U
Rotating journal
y
x
z
$(\tau + \frac{\partial \tau}{\partial y} dy)\, dx\, dz$
Flow of lubricant
$(p + \frac{dp}{dx} dx)\, dy\, dz$
dy
dx
h
$p\, dy\, dz$
$\tau\, dx\, dz$
Stationary partial bushing

(a) (b)

Design Considerations

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$$h_0 \geq 0.0002 + 0.000\,04D \text{ in}$$

where D is the journal diameter in inches.

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$$T_{max} \leq 250°\text{F}$$

The starting load divided by the projected area is limited to

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$$n_d \geq 2$$

$$\frac{fr}{c} \qquad \frac{h_0}{c} \qquad \phi \qquad \frac{Q}{rcNl} \qquad \frac{Q_s}{Q} \qquad \frac{P}{p_{\max}} \qquad \theta_P \qquad \frac{C_p \Delta T}{P}$$

Viscosity Charts

$$T_{av} = T_1 + \frac{\Delta T}{2} \quad \text{Eq. (12–13)},$$

where T_1 is the inlet temperature and ΔT is the temperature rise of the lubricant from inlet to outlet. Of course, the viscosity used in the analysis must correspond to T_{av}.

To illustrate, suppose we have decided to use SAE 30 oil in an application in which the oil inlet temperature is $T_1 = 180°\text{F}$. We begin by estimating that the temperature rise will be $\Delta T = 30°\text{F}$. Then, from Eq. (12–13),

$$T_{av} = T_1 + \frac{\Delta T}{2} = 180 = \frac{30}{2} = 195°\text{F}$$

This corresponds to point *A* on Fig. 12–11, which is above the SAE 30 line and indicates that the viscosity used in the analysis was too high.

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

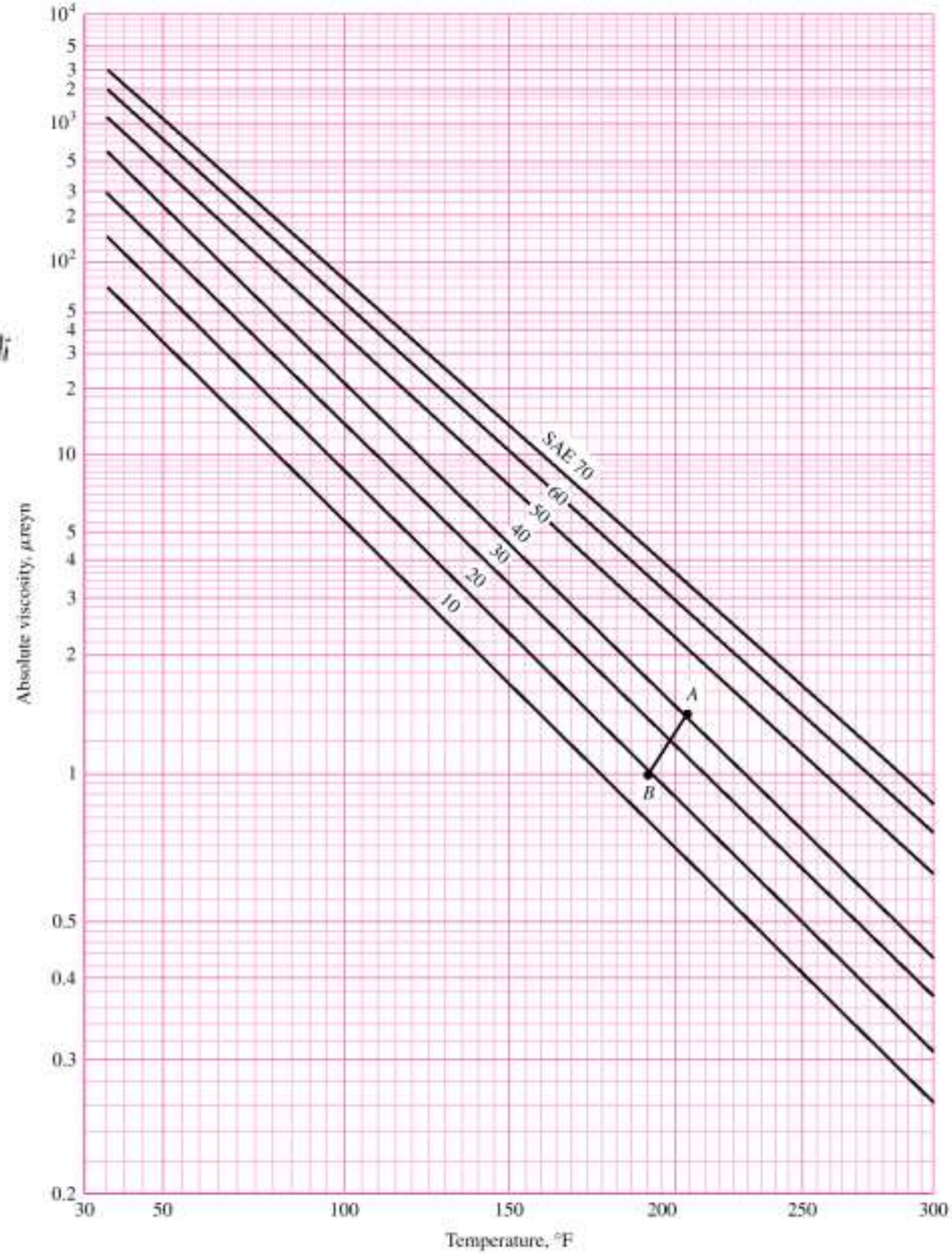
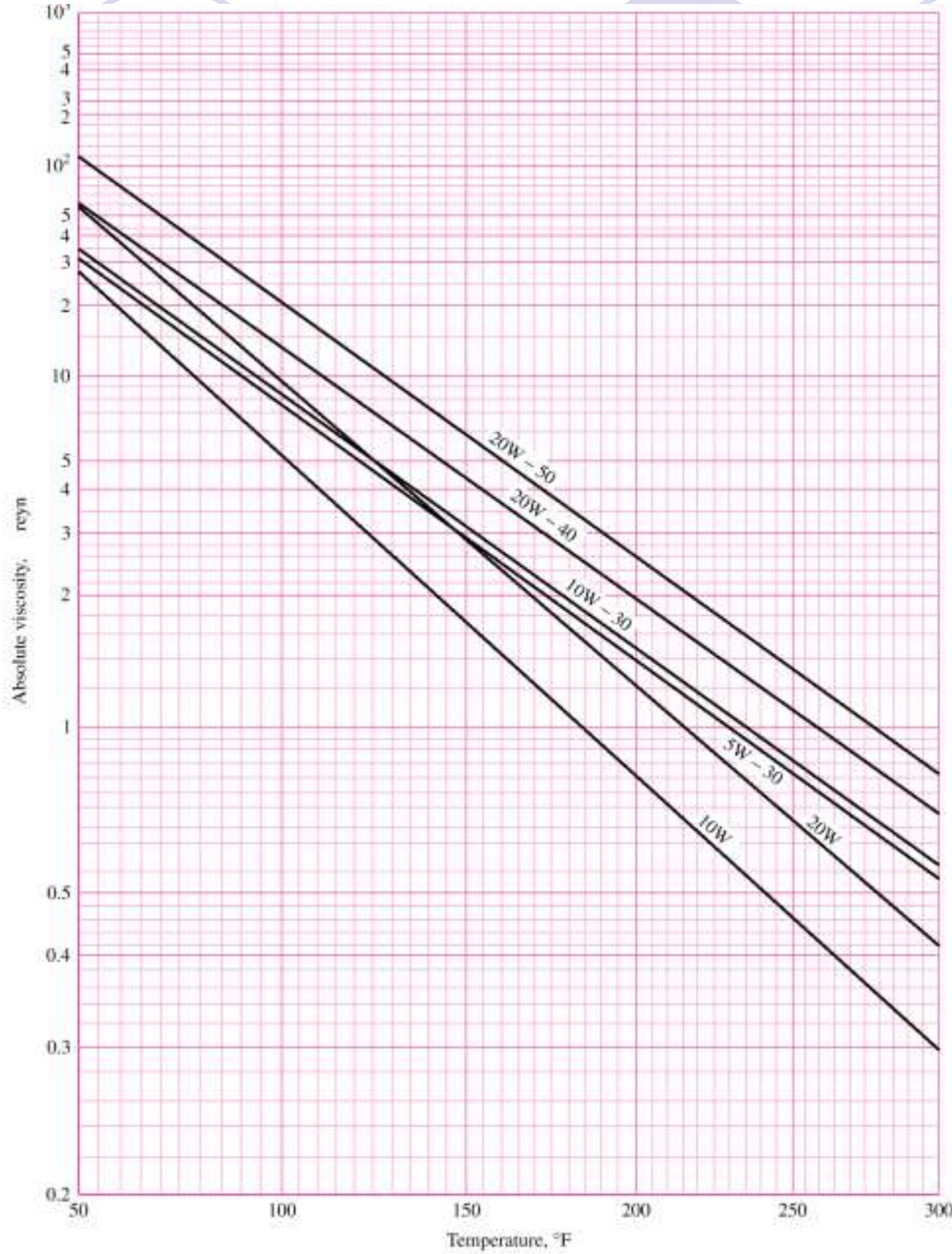


Chart for multiviscosity lubricants. This chart was derived from known viscosities at two points, 100 and 210°F, and the results are believed to be correct for other temperatures.



Home Work

- 12.3,12.4.12.5,12.6

Thanks