- Maximum film pressure and its angular position (*figs. 12-21 and 12-22*)

Figure 12-21

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

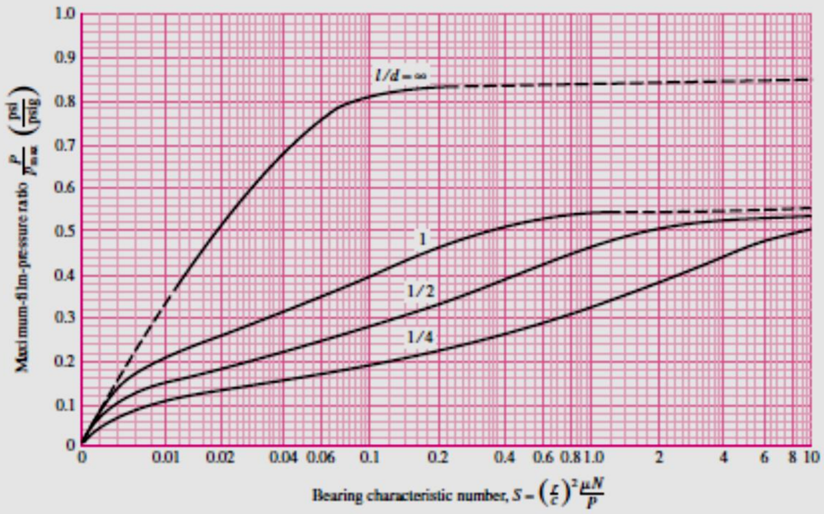


Figure 12-22

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

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 Entering Fig. 12–21 with $S = 0.135$ and $l/d = 1$, we find $P/p_{max} = 0.42$. The maximum pressure p_{max} is therefore

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

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

Lubricant Temperature Rise

The lubricant temperature will increase until a *heat balance* is reached (*heat generated by shearing the lubricant = heat lost to the surroundings*).

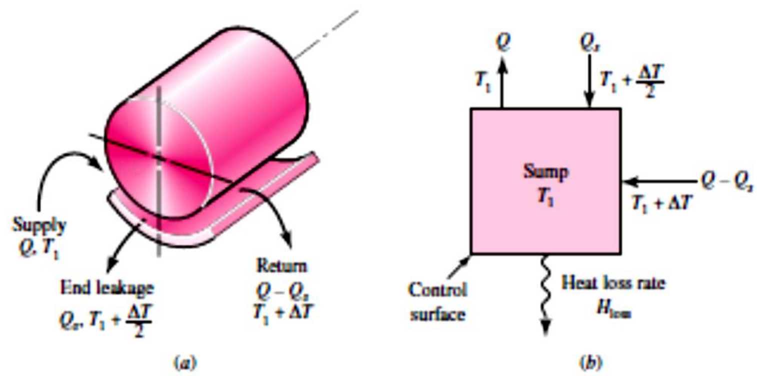


Figure 12-23
Schematic of a journal bearing with an external sump with cooling; lubricant makes one pass before returning to the sump.

Let

Q = volumetric oil-flow rate into the bearing, in^3/s
Qs = volumetric side-flow leakage rate out of the bearing and to the sump, in^3/s
$Q - Qs$ = volumetric oil-flow discharge from annulus to sump, in^3/s
$T1$ = oil inlet temperature (equal to sump temperature Ts), ◦F
ΔT = temperature rise in oil between inlet and outlet, ◦F
ρ = lubricant density, lbm/in^3
Cp = specific heat capacity of lubricant, Btu/(lbm · ◦F)
J = Joulean heat equivalent, in · lbf/Btu
H = heat rate, Btu/s

It can be shown that heat balance calculations (*assuming side flow*) will give the following equation: *(see derivation in text)*

For common petroleum lubricants ρ = 0.0311 lbm/in^3, Cp = 0.42 Btu/(lbm · ◦F), and J = 778(12) = 9336 in · lbf/Btu; therefore the left term of Eq. (c) is

$$\frac{9.70\Delta T_F}{P_{\text{psi}}} = \frac{rf/c}{(1 - \frac{1}{2}Q_s/Q)[Q/(rcN_jl)]} \qquad (12\text{–}15)$$

Or in mm

$$\frac{0.12\Delta T_C}{P_{(MPa)}} = \frac{{}^{r}/_{c}\, f}{\left(1 - \frac{1}{2}\left[{}^{Q_s}/_{Q}\right]\right)\left({}^{Q}/_{rcN_j l}\right)} \qquad (12\text{–}15)$$

- The *RHS* of this equation can be evaluated using charts in *figs. 12-18, 19 & 20* or *easier* using *fig. 12-24* where it combines the three charts together which makes the iterative approach to find ΔT easier.

Why there is no ($l/d = \infty$) curve in fig. 12-24 ?

Steady-State Conditions in Self Contained Bearings

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. This type of bearings is also called *pillow-block* bearings. In this type, the sump is expanded peripherally in the top half of the bearing and the bushing covers the lower half($\beta = 180°$). As the oil film exits the lower half of the bearing it mixes with sump contents, then heat is transferred to the surroundings.

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

$$\boxed{H_{loss} = \hbar_{CR} A (T_b - T_\infty)} \qquad (12\text{–}17)$$

where

H_{loss} = heat dissipated, Btu/h
h_{CR} = combined overall coefficient of radiation and convection heat transfer, Btu/(h · ft^2 · ◦F)
A = surface area of bearing housing, ft^2
T_b = surface temperature of the housing, ◦F
T_∞ = ambient temperature, ◦F

- Some representative values of $\hbar_{CR}$ are given as:

$$\hbar_{CR} = \begin{cases} 11.4 \;\; W/(m^2.°C) & for\ still\ air \\ 15.3 \;\; W/(m^2.°C) & for\ shaft - stirred\ air \\ 33.5 \;\; W/(m^2.°C) & for\ air\ moving\ at\ 25.4\ m/s \end{cases} \qquad (12\text{–}18)$$

- Similar expression can be written for the temperature difference between the lubricant film and housing surface ($T_f - T_b$). If we define $\bar{T}_f$ as the *average film temperature* between the inlet T_s and the outlet ($T_s + \Delta T$), then the following proportionality can be observed:

$$\boxed{\bar{T}_f - T_b = \alpha(T_b - T_\infty)} \qquad (a)$$

Where α depends on lubrication system and housing geometry

- Table 12-2 gives representative values of α .

| Table 12-2

Lubrication System	Conditions	Range of α
Oil ring	Moving air	1–2
	Still air	$\frac{1}{2}$–1
Oil bath	Moving air	$\frac{1}{2}$–1
	Still air	$\frac{1}{5}$–$\frac{2}{5}$

- Solving for T_b we get: $$\boxed{T_b = \bar{T}_f + \alpha T_\infty/(1+\alpha)} \qquad (12\text{–}19b)$$

- Substituting in the heat loss equation, we get:

$$\boxed{H_{loss} = \frac{\hbar_{CR} A}{1+\alpha}(\bar{T}_f - T_\infty)} \qquad (12\text{–}19a)$$

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

- The heat generated can be found as:

$$H_{gen} = T(2\pi N)$$

But, $$T = 4\pi^2 r^3 l\mu N/c$$

Thus,

$$H_{gen} = \frac{248\mu N^2 l r^3}{c} \qquad (b)$$

And in inches;

$$\boxed{H_{\text{gen}} = \frac{95.69\mu N^2 l r^3}{c}}$$

- In steady-state analysis, the average film temperature $\bar{T}_f$ is unknown and therefore the viscosity is unknown. Thus, a trial value of $\bar{T}_f$ is used (*the corresponding μ is found*) and both H_{loss}& H_{gen} are evaluated. Then, iterations continue until we get = H_{loss}& H_{gen} .
- Equating H_{loss}& H_{gen} and solving for $\bar{T}_f$ we get,

$$\boxed{\bar{T}_f = T_\infty + 248(1+\alpha)\frac{\mu N^2 l r^3}{\hbar_{CR} A c}}$$

Note that:
$\bar{T}_f = T_s + \Delta T/2$

(12–20)

And in inches;

$$\bar{T}_f = T_\infty + 95.69(1+\alpha)\frac{\mu N^2 l r^3}{\hbar_{CR} A c} \qquad (12\text{–}20)$$

EXAMPLE 12–5 Consider a pillow-block bearing with a keyway sump, whose journal rotates at 900 rev/min 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.

Solution The minimum radial clearance, $c_{\min}$, is

$$c_{\min} = \frac{2.002 - 2.000}{2} = 0.001 \text{ in}$$

$$P = \frac{W}{ld} = \frac{100}{(2)2} = 25 \text{ psi}$$

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = \left(\frac{1}{0.001}\right)^2 \frac{\mu'(15)}{10^6(25)} = 0.6\,\mu'$$

where μ' is viscosity in μreyn. The friction horsepower loss, $(\text{hp})_f$, is found as follows:

$$(\text{hp})_f = \frac{fWrN}{1050} = \frac{WNc}{1050}\frac{fr}{c} = \frac{100(900/60)0.001}{1050}\frac{fr}{c} = 0.001\,429\frac{fr}{c} \text{ hp}$$

The heat generation rate H_{gen}, in Btu/h, is

$$H_{gen} = 2545(hp)_f = 2545(0.001\,429) fr/c = 3.637\, fr/c \text{ Btu/h}$$

From Eq. (12–19*a*) with $\hbar_{CR} = 2.7$ Btu/(h · ft^2 · °F), the rate of heat loss to the environment H_{loss} is

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

Build a table as follows for trial values of $\bar{T}_f$ of 190 and 195°F:

Trial $\bar{T}_f$	μ'	S	fr/c	H_{gen}	H_{loss}
190	1.15	0.69	13.6	49.5	45.0
195	1.03	0.62	12.2	44.4	46.9

The temperature at which $H_{gen} = H_{loss} = 46.3$ Btu/h is 193.4°F. Rounding $\bar{T}_f$ to 193°F we find $\mu' = 1.08$ μreyn and $S = 0.6(1.08) = 0.65$. From Fig. 12–24, $9.70\Delta T_F/P = 4.25$°F/psi and thus

$$\Delta T_F = 4.25P/9.70 = 4.25(25)/9.70 = 11.0°\text{F}$$

$$T_1 = T_s = \bar{T}_f - \Delta T/2 = 193 - 11/2 = 187.5° \text{ F}$$

$$T_{max} = T_1 + \Delta T_F = 187.5 + 11 = 198.5°\text{F}$$

From Eq. (12–19*b*)

$$T_b = \frac{T_f + \alpha T_\infty}{1+\alpha} = \frac{193 + (1)70}{1+1} = 131.5°\text{F}$$

with $S = 0.65$, the minimum film thickness from Fig. 12–16 is

$$h_0 = \frac{h_0}{c}c = 0.79(0.001) = 0.000\,79 \text{ in}$$

The coefficient of friction from Fig. 12–18 is

$$f = \frac{fr}{c}\frac{c}{r} = 12.8\frac{0.001}{1} = 0.012\,8$$

The parasitic friction torque T is

$$T = fWr = 0.012\,8(100)(1) = 1.28 \text{ lbf} \cdot \text{in}$$

Clearance

When designing journal bearings for thick film lubrication, the designer selects the lubricant and suitable values for the bearing parameters to give satisfactory performance. However, the clearance "C" is difficult to hold accurate during manufacturing. Also, clearance increases with time because of wear.

- The figure shows the effect of wide range of clearances on the performance of a bearing (*for bearing parameters bgiven in examples 12-1 to 12-4*). It can be seen from the figure that lubricant flow increases with increased clearance and this decreases the generated heat and outlet temperature. The minimum film thickness "h_0" increases with clearance then it starts to decrease.
- If the clearance is too small, dirt (debris) may block the oil flow and therefore cause overheating and failure.

Figure 12–25

A plot of some performance characteristics of the bearing of Exs. 12–1 to 12–4 for radial clearances of 0.0005 to 0.003 in. The bearing outlet temperature is designated T_2. New bearings should be designed for the shaded zone, because wear will move the operating point to the right.

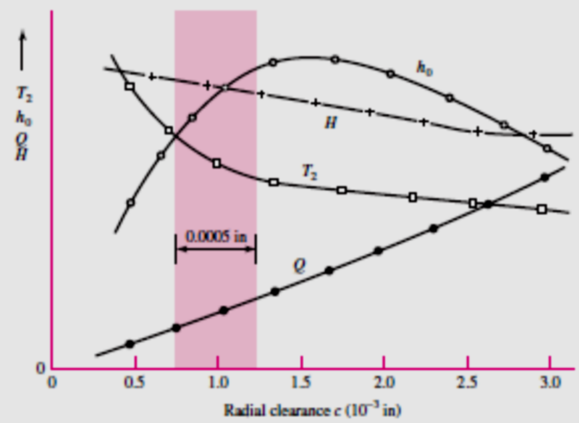


Table 12–3

Maximum, Minimum, and Average Clearances for 1.5-in-Diameter Journal Bearings Based on Type of Fit

		Clearance c, in		
Type of Fit	**Symbol**	**Maximum**	**Average**	**Minimum**
Close-running	H8/f7	0.001 75	0.001 125	0.000 5
Free-running	H9/d9	0.003 95	0.002 75	0.001 55

Table 12–4

Performance of 1.5-in-Diameter Journal Bearing with Various Clearances. (SAE 20 Lubricant, $T_1 = 100°F$, $N = 30$ r/s, $W = 500$ lbf, $l = 1.5$ in)

c, in	T_2, °F	h_0, in	f	Q, in³/s,	H, Btu/s
0.000 5	226	0.000 38	0.011 3	0.061	0.086
0.001 125	142	0.000 65	0.009 0	0.153	0.068
0.001 55	133	0.000 77	0.008 7	0.218	0.066
0.001 75	128	0.000 76	0.008 4	0.252	0.064
0.002 75	118	0.000 73	0.007 9	0.419	0.060
0.003 95	113	0.000 69	0.007 7	0.617	0.059

- If the clearance is too high, the bearing becomes noisy h_0 and decreases.
- Thus, the optimum range of clearances is shown by shaded area in the figure. If clearance value is within this range the performance of the bearing will improve with wear.

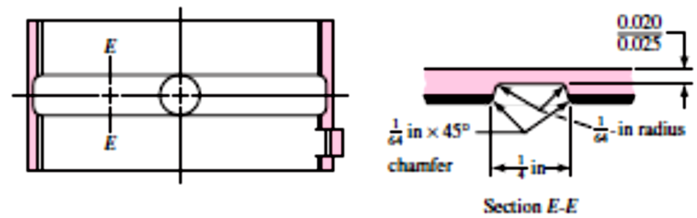
Pressure-Fed Bearings

The load carrying capacity of self-contained bearings is limited because of the limited heat-dissipating capability. To increase the heat-dissipation, an external pump is used to increase the lubricant flow through the bearing. The pump supplies the bearing with lubricant of high pressure therefore increasing the lubricant flow and heat dissipation. The lubricant sump may also be cooled with water to reduce temperature further.

- A circumferential groove at the center of the bearing, with an oil-supply hole located opposite to the load zone, is usually used to feed the lubricant (*fig. 12-27*).

Figure 12-27

Centrally located full annular groove. (Courtesy of the Cleveland Graphite Bronze Company, Division of Clevite Corporation.)



- The oil flows from the groove in the middle towards the ends of the bearing.
- Note that in this type of bearings, the lubricant is supplied at high pressure (*pressure is not created by the sliding of the journal surface*).
- When determining the lubricant flow, eccentricity is first neglected, then a correction factor for eccentricity is applied. Also, rotation of the shaft is neglected.
- For lubricant supplied at pressure "P_s", the average velocity of lubricant flowing towards the ends of the bearing can be found as *(see derivation in text)*:

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

Average velocity for any angular position

(i)

Figure 12-29

Parabolic distribution of the lubricant velocity.

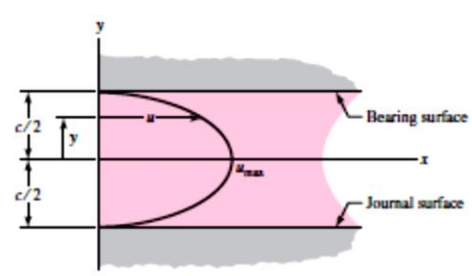
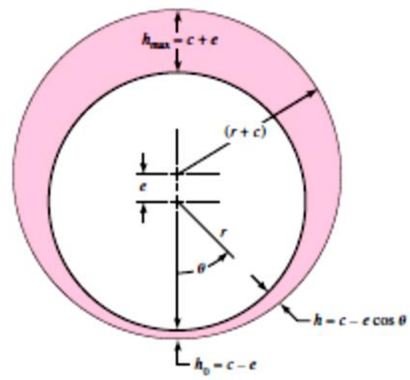
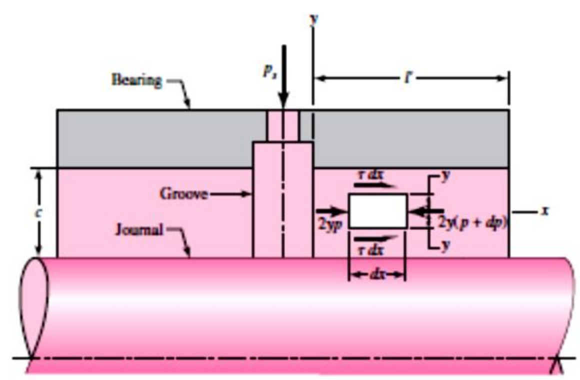


Figure 12–30



where is length of each half of the bearing (*see fig. 12-28*)

Figure 12–28
Flow of lubricant from a pressure-fed bearing having a central annular groove.



- Note that minimum film thinness "h_0" is assumed to be at the bottom of the bearing (*fig. 12-30*) because rotation is neglected.
- Thus, lubricant flow out of both ends of the bearing is:

$$\boxed{Q_s = \frac{\pi P_s r c^3}{3\mu l'}(1 + 1.5\epsilon^2)} \qquad (12\text{–}22)$$

- In this type of bearings, the pressure per projected area for each half of the bearing is:

$$\boxed{P = \frac{w/2}{2rl'} = \frac{w}{4rl'}} \qquad (12\text{–}23)$$

- Therefore, the *Sommerfeld* number is;

$$\boxed{S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = \boxed{\left(\frac{r}{c}\right)^2 \frac{4rl'\mu N}{W}}}$$

- The heat generated and heat loss can be found as:

$$H_{loss} = \rho C_p Q_s \Delta T$$

Where ρ: density
C_p: heat capacity

$$H_{gen} = 2\pi W N c \frac{fr}{c}$$

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

$$\boxed{\Delta T_c = \frac{978(10^6)}{1+1.5\epsilon^2}\frac{(fr/c)SW^2}{P_s r^4}} \quad \text{for } \Delta T \text{ in } ^\circ C \qquad (12\text{–}25)$$

Where: W (kN)
P_s (kPa)
r (mm)

Or

$$\boxed{\Delta T_F = \frac{0.0123(fr/c)SW^2}{(1+1.5\epsilon^2)P_s r^4}} \quad \text{for } \Delta T \text{ in } ^\circ F \qquad (12\text{–}24)$$

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 rev/min, or 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_{max}, and P_{st} with the Trumpler criteria.

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

Solution (*a*)

$$r = \frac{d_j}{2} = \frac{1.750}{2} = 0.875 \text{ in}$$

$$c_{min} = \frac{(d_b)_{min} - (d_j)_{max}}{2} = \frac{1.753 - 1.750}{2} = 0.0015 \text{ in}$$

Since $l'/d = 1/2$, $l' = d/2 = r = 0.875$ in. Then the pressure due to the load is

$$P = \frac{W}{4rl'} = \frac{900}{4(0.875)0.875} = 294 \text{ psi}$$

The Sommerfeld number S can be expressed as

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} = \left(\frac{0.875}{0.0015}\right)^2 \frac{\mu'}{(10^6)} \frac{50}{294} = 0.0579\mu' \qquad (1)$$

We will use a tabulation method to find the average film temperature. The first trial average film temperature $\bar{T}_f$ will be 170°F. Using the Seireg curve fit of Table 12–1, we obtain

$$\mu' = 0.0136 \exp[1271.6/(170 + 95)] = 1.650 \ \mu\text{reyn}$$

From Eq. (1)

$$S = 0.0579\mu' = 0.0579(1.650) = 0.0955$$

From Fig. (12–18), $fr/c = 3.3$, and from Fig. (12–16), $\epsilon = 0.80$. From Eq. (12–24),

$$\Delta T_F = \frac{0.0123(3.3)0.0955(900^2)}{[1 + 1.5(0.80)^2]30(0.875^4)} = 91.1°\text{F}$$

$$T_{av} = T_s + \frac{\Delta T}{2} = 120 + \frac{91.1}{2} = 165.6°\text{F}$$

We form a table, adding a second line with $\bar{T}_f = 168.5$°F:

Trial $\bar{T}_f$	μ'	S	fr/c	ϵ	ΔT_F	T_{av}
170	1.65	0.0955	3.3	0.800	91.1	165.6
168.5	1.693	0.0980	3.39	0.792	97.1	168.5

If the iteration had not closed, one could plot trial $\bar{T}_f$ against resulting T_{av} and draw a straight line between them, the intersection with a $\bar{T}_f = T_{av}$ line defining the new trial $\bar{T}_f$.

Answer The result of this tabulation is $\bar{T}_f = 168.5$, $\Delta T_F = 97.1$°F, and $T_{max} = 120 + 97.1 = 217.1$°F

(*b*) Since $h_0 = (1 - \epsilon)c$,

$$h_0 = (1 - 0.792)0.0015 = 0.000\,312 \text{ in}$$

The required four Trumpler criteria, from "Significant Angular Speed" in Sec. 12–7 are

$$h_0 \geq 0.0002 + 0.000\,04(1.750) = 0.000\,270 \text{ in} \qquad \text{(OK)}$$

Answer $$T_{max} = T_s + \Delta T = 120 + 97.1 = 217.1°\text{F} \qquad \text{(OK)}$$

$$P_{st} = \frac{W_{st}}{4rl'} = \frac{900}{4(0.875)0.875} = 294 \text{ psi} \qquad \text{(OK)}$$

The factor of safety on the load is approximately unity. (Not OK.)

(*c*) From Eq. (12–22),

Answer $$Q_s = \frac{\pi(30)0.875(0.0015)^3}{3(1.693)10^{-6}(0.875)}[1 + 1.5(0.80)^2] = 0.123 \text{ in}^3/\text{s}$$

$$H_{loss} = \rho C_p Q_s \Delta T = 0.0311(0.42)0.123(97.1) = 0.156 \text{ Btu/s}$$

or 562 Btu/h or 0.221 hp. The parasitic friction torque T is

Answer $$T = fWr = \frac{fr}{c}Wc = 3.39(900)0.0015 = 4.58 \text{ lbf} \cdot \text{in}$$

Thrust Bearings

Journal bearings are designed to take radial loads only, if there is a thrust component, a sliding thrust bearing can be used.

- The working principle of thrust bearings is similar to that of journal bearings. The figure shows a fixed-pad thrust bearing consisting of a runner sliding over a fixed pad. The lubricant is brought into the radial grooves and pumped into the wedge shaped spaces (*see fig. 12-36*).

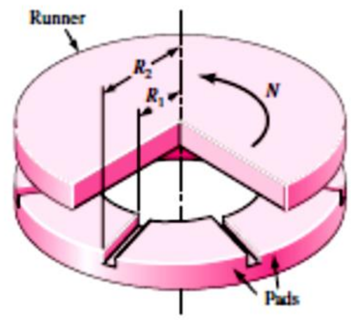


Figure 12-35
Fixed-pad thrust bearing.
(Courtesy of Westinghouse Electric Corporation.)

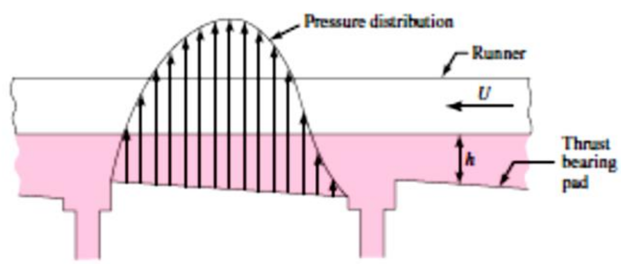


Figure 12-36
Pressure distribution of lubricant in a thrust bearing.
(Courtesy of Copper Research Corporation.)

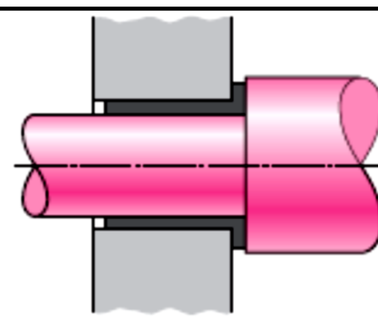


Figure 12-37
Flanged sleeve bearing takes both radial and thrust loads.

- Hydrodynamic lubrication is obtained if the speed of the runner is continuous and sufficiently high and lubricant is available in sufficient quantity.

Boundary-Lubricated Bearings

When two surfaces slide relative to each other with only partial lubricant film between them, boundary (*or thin-film*) lubrication is said to exist.

- Boundary lubrication occurs in hydrodynamically lubricated bearings on starting or stopping condition, or when load increases or if lubricant supply or rotational speed decreases.

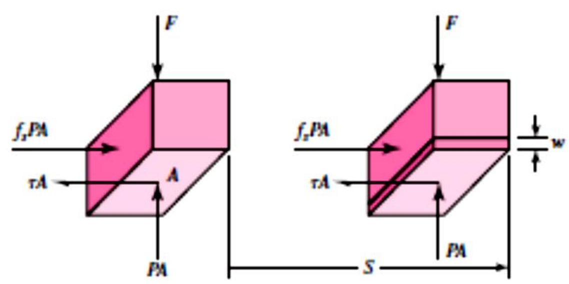
- To reduce friction in boundary lubricated bearings, animal or vegetable oils can be mixed with mineral oils because they form a soap film that sticks to metallic surfaces.
- "*Mixed-film lubrication*" is said to exist if the bearing operates partly under hydrodynamic conditions and partly under boundary lubrication conditions. Mixed-film lubrication condition occurs if lubricant is supplied manually using drop or mechanical feed, viscosity is too low, bearing speed is too low, the bearing is overloaded, the clearance is too tight, or the journal is not properly aligned.

Linear sliding wear:

Considering a block with surface area "A " sliding over a fixed surface with contact pressure "P " where the coefficient of sliding friction "f_s " and define a linear wear measure "w ".

Figure 12-38

Sliding block subjected to wear.



- The work done by the frictional force "f_sPA" during displacement "S" is "f_sPAS " or "f_sPAVt " (where "V" velocity and "t " time).

- The volume of material removed because of wear "wA " is proportional to the work ($wA \propto f_sPAVt$), or we can write:

$$\boxed{wA = KPAVt}$$

Where "K" is the proportionality factor which includes the coefficient of friction "f_s ". The unit of "f_s " in *SI* units is: $m^3.s/(N.m.s)$

- *Table 12-8* gives the values of "K " (which are determined from testing).

Table 12-8

Wear Factors in U.S. Customary Units*

Source: Oiles America Corp., Plymouth, MI 48170.

Bushing Material	Wear Factor K	Limiting PV
Oiles 800	$3(10^{-10})$	18 000
Oiles 500	$0.6(10^{-10})$	46 700
Polyactal copolymer	$50(10^{-10})$	5 000
Polyactal homopolymer	$60(10^{-10})$	3 000
66 nylon	$200(10^{-10})$	2 000
66 nylon + 15% PTFE	$13(10^{-10})$	7 000
+ 15% PTFE + 30% glass	$16(10^{-10})$	10 000
+ 2.5% MoS_2	$200(10^{-10})$	2 000
6 nylon	$200(10^{-10})$	2 000
Polycarbonate + 15% PTFE	$75(10^{-10})$	7 000
Sintered bronze	$102(10^{-10})$	8 500
Phenol + 25% glass fiber	$8(10^{-10})$	11 500

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

- Cancelling "A " from both sides, " w" can be expressed as:

$$\boxed{w = KPVt} \qquad (12\text{–}26)$$

- Additional correction factors f_1 & f_2 can be included such that:

$$\boxed{w = f_1 f_2 KPVt} \qquad (12\text{–}27)$$

where, f_1 correctio factor for motion type; ***Table 12-10***

f_2 correctio factor for enviroment; ***Table 12-11***

Table 12-10

Motion-Related Factor f_1

Mode of Motion	Characteristic Pressure P, psi		Velocity V, ft/min	f_1*
Rotary	720 or less		3.3 or less	1.0
			3.3–33	1.0–1.3
			33–100	1.3–1.8
	720–3600		3.3 or less	1.5
			3.3–33	1.5–2.0
			33–100	2.0–2.7
Oscillatory	720 or less	>30°	3.3 or less	1.3
			3.3–100	1.3–2.4
		<30°	3.3 or less	2.0
			3.3–100	2.0–3.6
	720–3600	>30°	3.3 or less	2.0
			3.3–100	2.0–3.2
		<30°	3.3 or less	3.0
			3.3–100	3.0–4.8
Reciprocating	720 or less		33 or less	1.5
			33–100	1.5–3.8
	720–3600		33 or less	2.0
			33–100	2.0–7.5

*Values of f_1 based on results over an extended period of time on automotive manufacturing machinery.

Table 12-11

Environmental Factor f_2

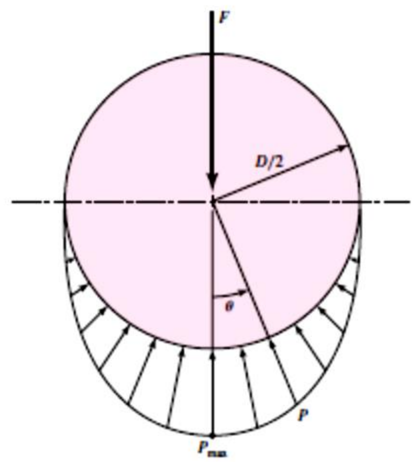
Source: Oiles America Corp., Plymouth, MI 48170.

Ambient Temperature, °F	Foreign Matter	f_2
140 or lower	No	1.0
140 or lower	Yes	3.0–6.0
140–210	No	3.0–6.0
140–210	Yes	6.0–12.0

***Bushing wear*:**

Figure 12-39

Pressure distribution on a boundary-lubricated bushing.



For the case of journal bearing of diameter "D " and Length " L" rotating at speed " N" , the wear of the bushing " W" can be found from the prvious equation knowing that:

$$P_{max} = \frac{4}{\pi} P = \frac{4}{\pi} \frac{F}{DL} \quad \& \quad F = W = \underline{\text{radial load}} \qquad (12\text{–}31)$$

Thus, the wear is found to be:

$$w = f_1 f_2 K \frac{4}{\pi} \frac{F}{DL} \frac{\pi D N t}{12} = \frac{f_1 f_2 K F N t}{3L} \qquad (12\text{–}32)$$

And the same;

$$w = f_1 f_2 K \frac{4}{\pi} \frac{F}{DL} V t = \frac{4 f_1 f_2 K F N t}{L}$$

Where
$V(m/s)$
$N(rev/s)$
$t(s)$

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.

Solution From Table 12–8, $K = 0.6(10^{-10})$ in^3 · min/(lbf · ft · h); Tables 12–10 and 12–11, $f_1 = 1.3$, $f_2 = 1$; and Table 12–12, $PV = 46\,700$ psi · ft/min, $P_{max} = 3560$ psi, $V_{max} = 100$ ft/min. From Eqs. (12–31), (12–29), and (12–30),

$$P_{max} = \frac{4}{\pi}\frac{F}{DL} = \frac{4}{\pi}\frac{700}{(1)(1)} = 891 \text{ psi} < 3560 \text{ psi} \qquad \text{(OK)}$$

$$P = \frac{F}{DL} = \frac{700}{(1)(1)} = 700 \text{ psi}$$

$$V = 33 \text{ ft/min} < 100 \text{ ft/min} \qquad \text{(OK)}$$

$$PV = 700(33) = 23\,100 \text{ psi} \cdot \text{ft/min} < 46\,700 \text{ psi} \cdot \text{ft/min} \qquad \text{(OK)}$$

Equation (12–32) with Eq. (12–29) is

$$w = f_1 f_2 K \frac{4}{\pi}\frac{F}{DL}\frac{\pi D N t}{12} = f_1 f_2 K \frac{4}{\pi}\frac{F}{DL} V t$$

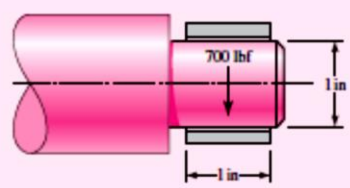


Figure 12–40
Journal/bushing for Ex. 12–7.

Table 12–12
Oiles 500 SP (SPBN · SPWN) Service Range and Properties
Source: Oiles America Corp., Plymouth, MI 48170.

Service Range	Units	Allowable
Characteristic pressure P_{max}	psi	<3560
Velocity V_{max}	ft/min	<100
PV product	(psi)(ft/min)	<46 700
Temperature T	°F	<300
Properties	**Test Method, Units**	**Value**
Tensile strength	(ASTM E8) psi	>110 000
Elongation	(ASTM E8) %	>12
Compressive strength	(ASTM E9) psi	49 770
Brinell hardness	(ASTM E10) HB	>210
Coefficient of thermal expansion	(10^{-5}) °C	>1.6
Specific gravity		8.2

Solving for t gives

$$t = \frac{\pi D L w}{4 f_1 f_2 K V F} = \frac{\pi(1)(1)0.005}{4(1.3)(1)0.6(10^{-10})33(700)} = 2180 \text{ h} = 130\,770 \text{ min}$$

The rotational speed is

$$N = \frac{12V}{\pi D} = \frac{12(33)}{\pi(1)} = 126 \text{ r/min}$$

Answer Cycles $= Nt = 126(130\,770) = 16.5(10^6)$ rev