Sliding Bearings

The goal of a bearing is to provide relative positioning and rotational freedom while transmitting a load between two parts, commonly a shaft and its housing. The object of lubrication is to reduce the friction, wear, and heating between two surfaces moving relative to each other. This is done by inserting a substance, called lubricant, between the moving surfaces.

Lubrication and Journal Bearings

Journal bearings support loads perpendicular to the shaft axis by pressure developed in the liquid. A journal bearing is a typical sliding bearing requiring sliding of the load carrying member on its support. **Sleeve thrust bearings** support loads in the direction of the shaft axis. Lubricants may be Liquid, solid or gas.

Types of Journal Bearings

The journal bearing or sleeve bearing supports a load in the radial direction. It has two main parts: a shaft called the journal and a hollow cylinder or sleeve that carries the shaft, called the bearing as shown in figure (1). The journal bearings may be full or partial journal bearing.

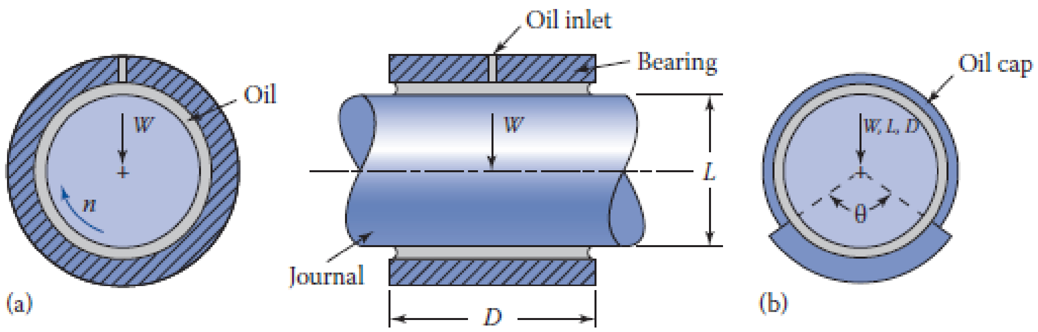


Fig.(1) (a) Full-journal bearing and (b) partial-journal bearing

Forms of Lubrication

Lubrications commonly are classified according to the degree with which the lubricant separates the sliding surfaces. Five distinct forms or types of lubrication occur in bearings: boundary, mixed, elasto-hydrodynamic, hydrodynamic, and hydrostatic as shown in figure (2). The bearings are often designated according to the form of lubrication used.

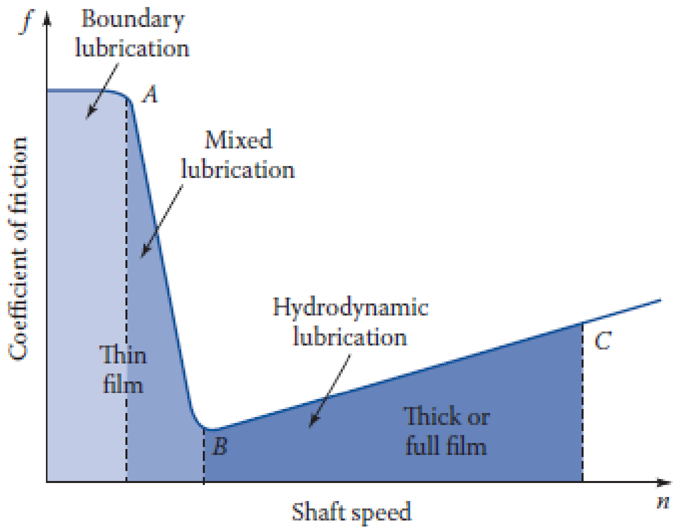


Fig.(2) The change in the coefficient of friction (f) with shaft speed (n) in a journal bearing.

In hydrodynamic lubrication (thick film lubrication) the coefficients of friction (f) commonly range from 0.002 to 0.010.

In mixed lubrication Typical values of the coefficient of friction are 0.004–0.10. This lubricating condition may also be present where lubrication is deficient, viscosity is too low, bearing is overloaded, clearance is too tight, bearing speed is too low, and bearing assembly is misaligned.

In boundary lubrication the coefficient of friction is about 0.1. boundary lubrication is less desirable than the other types, inasmuch as it allows the surface asperities to contact and wear rapidly. Design for this

type of lubrication is largely empirical. Electric motor shaft bearings, office machinery bearings, power screw support bearings, and electric fan bearings represent some examples of boundary lubrication bearings.

Elasto-hydrodynamic lubrication is concerned with the interrelation between the hydrodynamic action of full-fluid films and the elastic deformation of the supporting materials. It occurs when the lubricant is introduced between surfaces in rolling contact, such as mating gears and rolling bearings.

Units of Viscosity

In SI, viscosity is measured in newton-second per square meter ($N \cdot s/m^2$) or Pascal-seconds.

The U.S. customary unit of viscosity is the pound-force-second per square inch ($lb \cdot s/in.^2$), called the **Reyn.**

The conversion between the two units is the same as stress:

1 reyn = 6890 Pa . s

The reyn and pascal-second are such large units that micro-reyn (μreyn) and milli Pascal second (mPa·s) are more commonly used.

Effects of Temperature and Pressure

The viscosity of a liquid varies inversely with temperature and directly with pressure, both nonlinearly. Figure (3) shows the absolute viscosity of various fluids and how they vary.

The Society of Automotive Engineers (SAE) and the International Standards Organization (ISO) classify oils according to viscosity. Viscosity–temperature curves for typical SAE numbered oils are given in Figure (4).

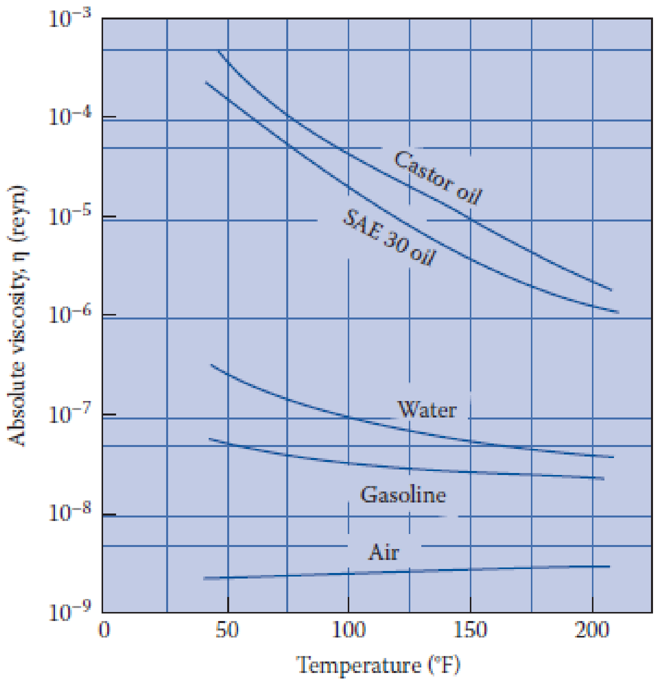


Fig.(3) Variation in viscosity with temperature of several fluids.

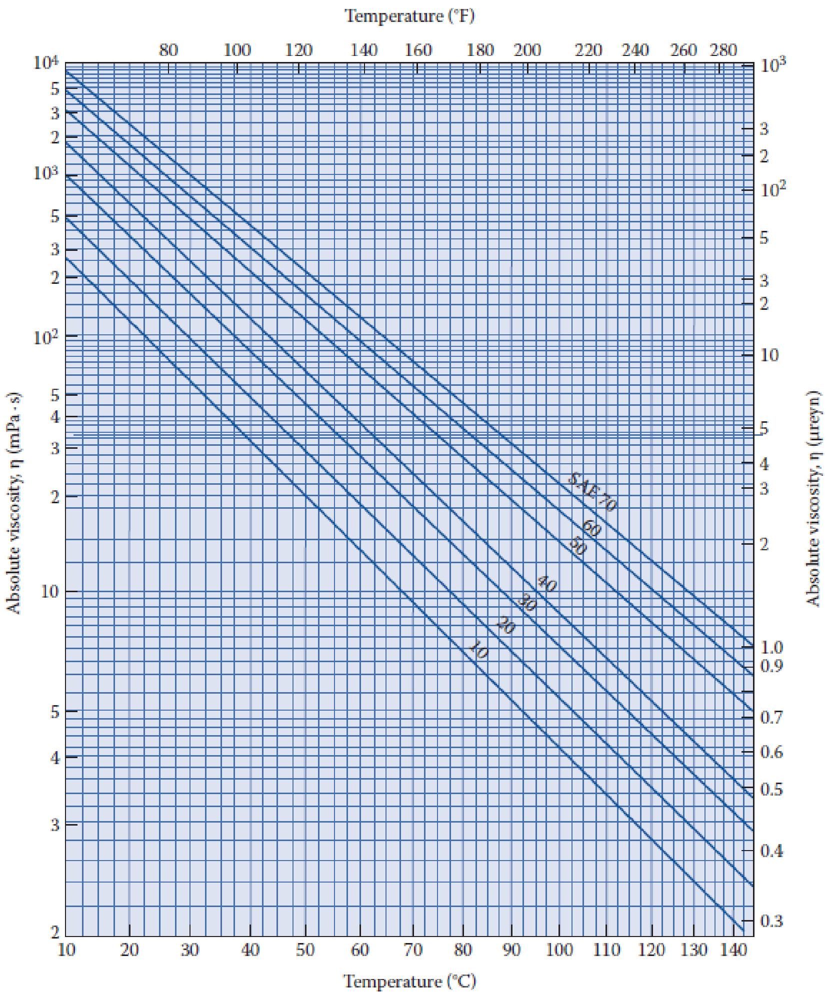


Fig.(4) Viscosity versus temperature curve for typical SAE-graded oils.

Design of Journal Bearings

The design of journal bearings usually involves two suitable combinations of variables:

variables under control (viscosity, load, radius and length of bearing, and clearance) and dependent variables or performance factors (coefficients of friction, temperature rise, oil flow, and minimum oil-film thickness). Essentially, in bearing design, limits for the latter group of variables are defined. Then, the former group is decided on so that these limitations are not exceeded. The following is a brief discussion of the quantities under control.

Lubricants

Recall that lubricants are characterized by their viscosity (η). Their choice is based on such factors as type of machine, method of lubrication, and load features.

Bearing Load

Usually, the load acting on a bearing is particularized. The value of the load per projected area, P, depends on the length and diameter of the bearing. Obviously, the smaller is P, the greater the bearing life.

Length–Diameter Ratio

Various factors are considered in choosing proper length-to-diameter ratios, or L/D values. Bearings with a length-to-diameter ratio less than 1 (short bearings) accommodate the shaft deflections and misalignments that are expected to be severe. Long bearings ($L/D > 1$) must be used in applications where shaft alignment is important.

Clearance

The effects of varying dimensions and clearance ratios are very significant in a bearing design. The radial clearance c (Figure5) is contingent to some extent on the desired quality. Suitable values to be

used for radial bearing clearance rely on factors that include materials, manufacturing accuracy, load-carrying capacity, minimum film thickness, and oil flow. Furthermore, the clearance may increase because of wear. The clearance ratios (c/r) typically vary from 0.001 to 0.002 and occasionally as high as 0.003. It would seem that large clearances increase the flow that reduces film temperature and hence increase bearing life. However, very large clearances result in a decrease in minimum film thickness.

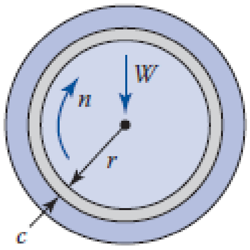


Fig.(5) Lightly loaded journal bearing

Design Charts

A.A. Raimondi and J. Boyd applied digital computer techniques toward the solution of Reynolds's equation and present the results in the form of design charts and tables. These provide accurate results for bearings of all proportions. Most charts utilize the bearing characteristic number, or the **Sommerfeld number**:

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

where

S = the bearing characteristic number, dimensionless

r = the journal radius

c = the radial clearance

η = the viscosity

n = the relative speed between journal and bearing, rps

P = the load per projected area

The principle of operation of the journal bearing can be seen in figure (6). Notations used in this figure are the center of the journal is shown at O and the center of the bearing is at O′. The minimum oil-film thickness h_0 occurs at the line of centers. The distance between these centers represents the eccentricity, denoted by *e*.

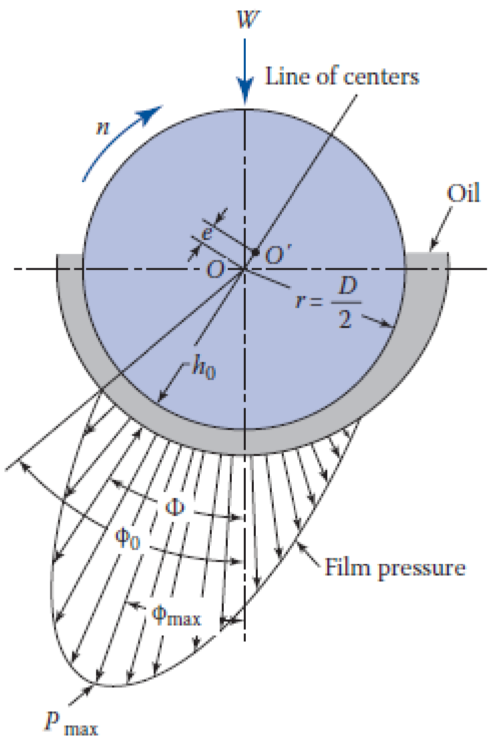


Fig.(6) Radial pressure distribution in a journal bearing.

The eccentricity ratio ε is defined by

$$\boldsymbol{\varepsilon = \frac{e}{c}}$$

The minimum film thickness is then

$$\boldsymbol{h_o = c - e = c(1 - \varepsilon)}$$

Or

$$\boldsymbol{\varepsilon = 1 - \frac{h_o}{c}}$$

As depicted in the figure, the angular location of the minimum oil-film thickness is designated by Φ. The terminating position and position of maximum film pressure p_{max} of the lubricant are denoted by ϕ_0 and ϕ_{max}, respectively. Load per projected area, the average pressure or the so-called unit loading, is

$$\mathrm{P} = \frac{\mathrm{W}}{\mathrm{DL}}$$

where,

W = the load

D = the journal diameter also referred to bearing diameter.

L = the journal length also referred to bearing length.

Design charts by Raimondi and Boyd provide solutions for journal bearings having various length–diameter (L/D) ratios. The design charts can be shown in Figures (7) through (12), for full bearings. All charts give the plots of dimensionless bearing parameters as functions of the dimensionless Sommerfeld variable, S. Note that the S scale on the charts is logarithmic except for a linear portion between 0 and 0.01.

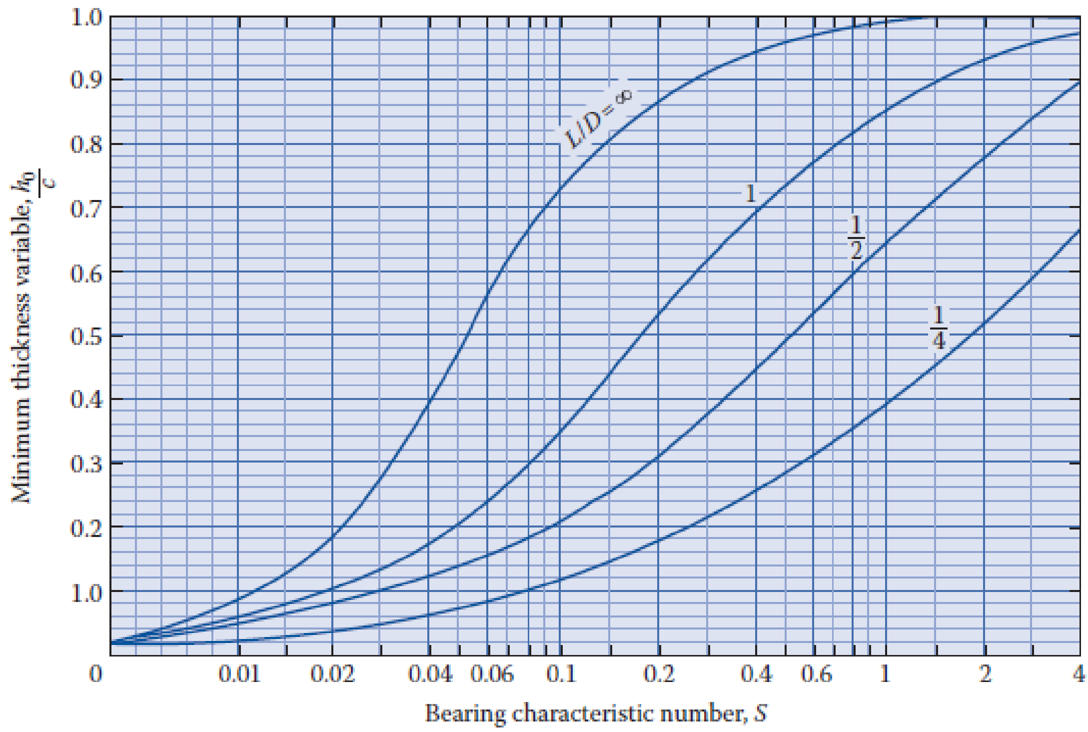


Fig.(7) Chart for minimum film-thickness variable

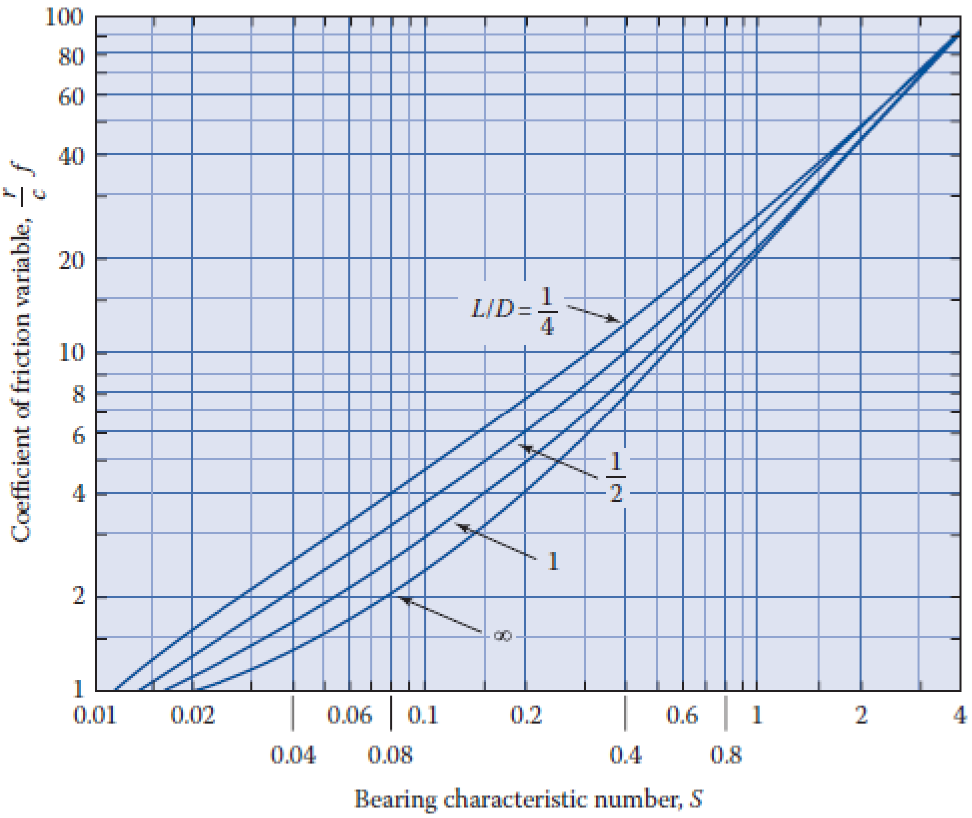


Fig.(8) Chart for coefficient of friction variable.

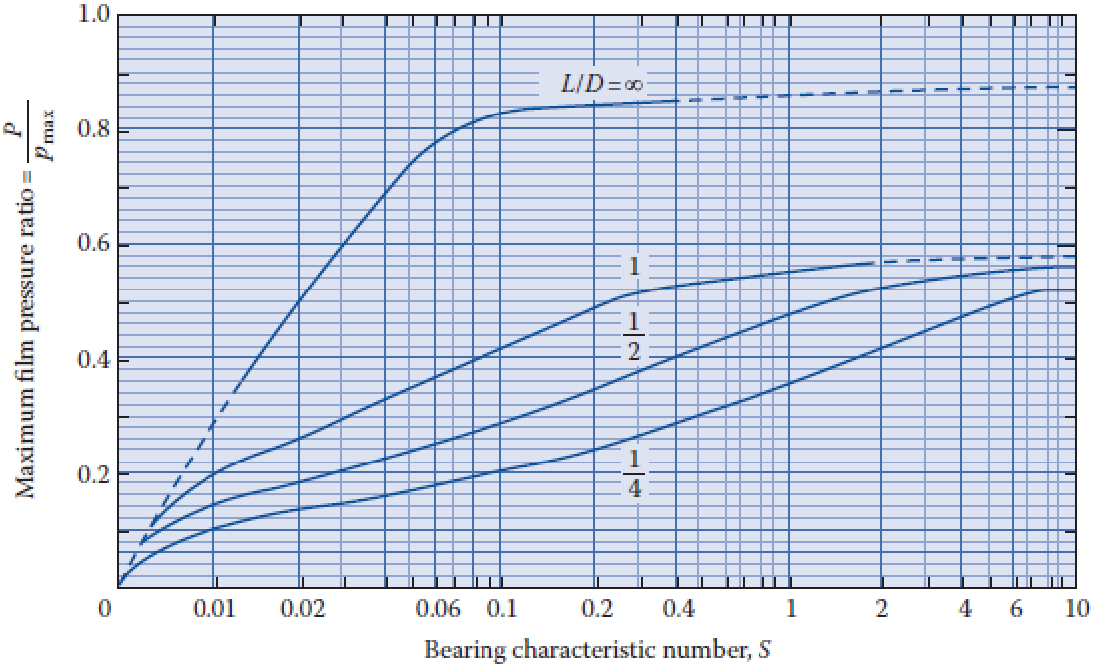


Fig.(9) Chart for film maximum pressure.

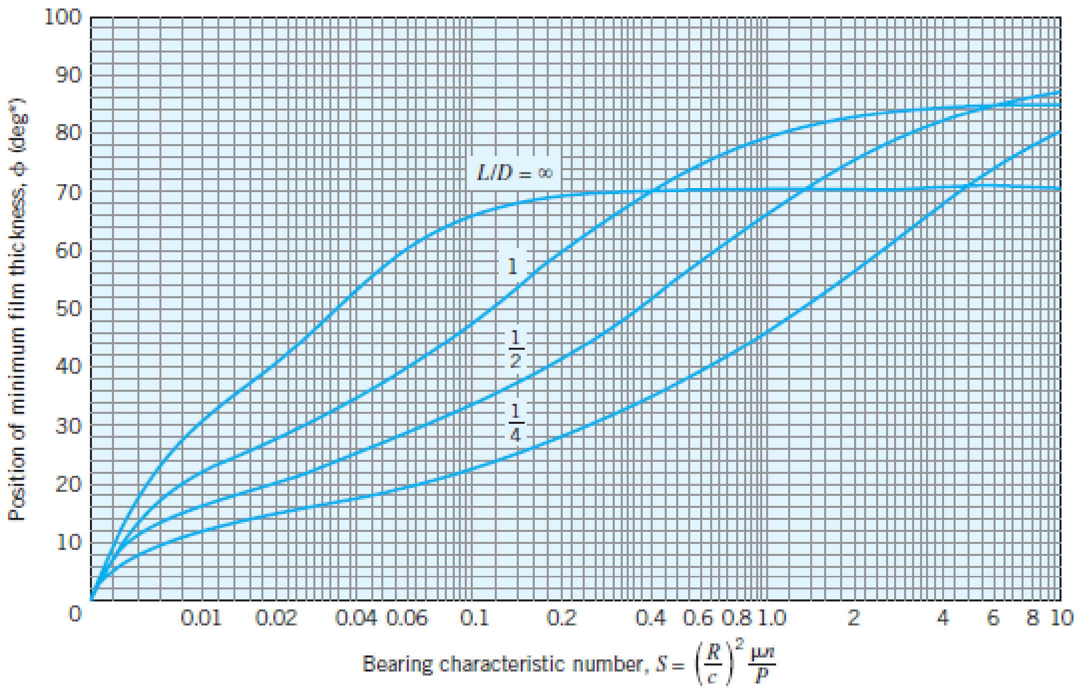


Fig.(10) Chart for determining the position of the minimum film thickness h_0

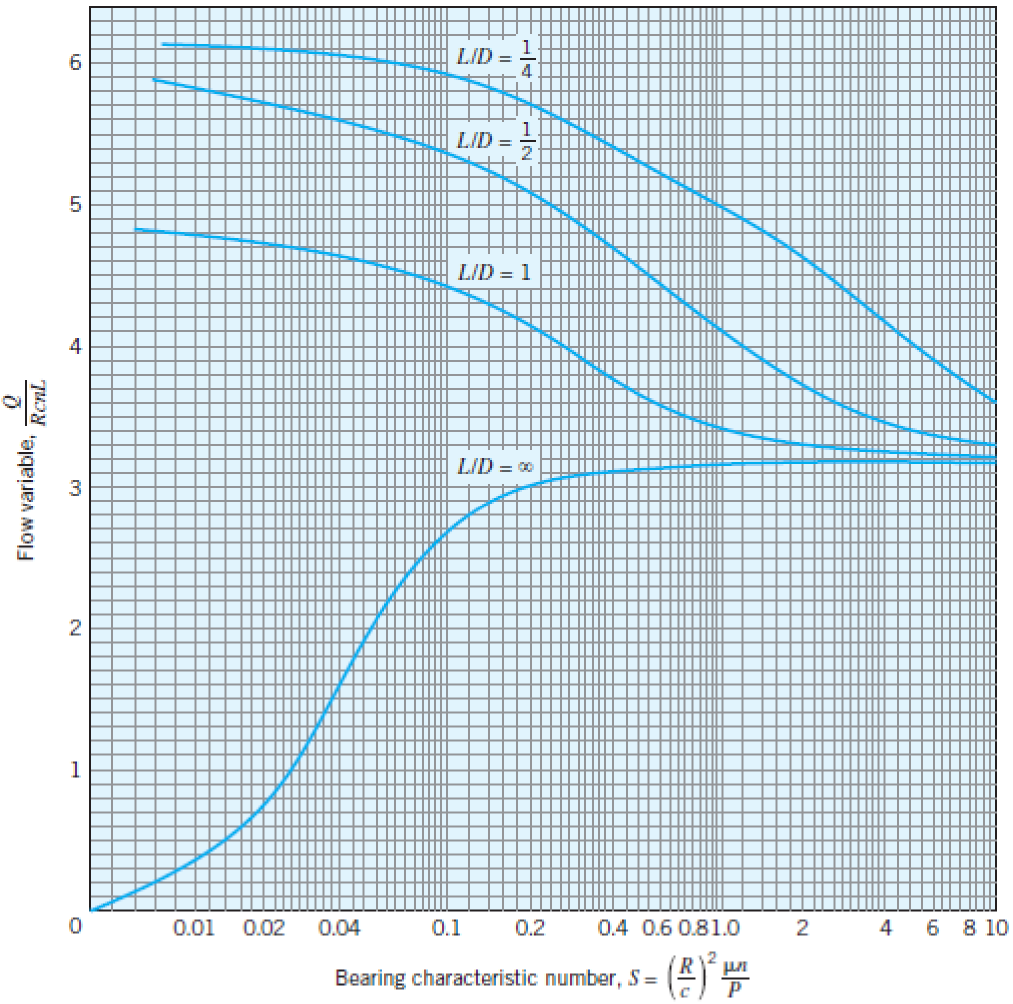


Fig.(11) Chart for flow variable

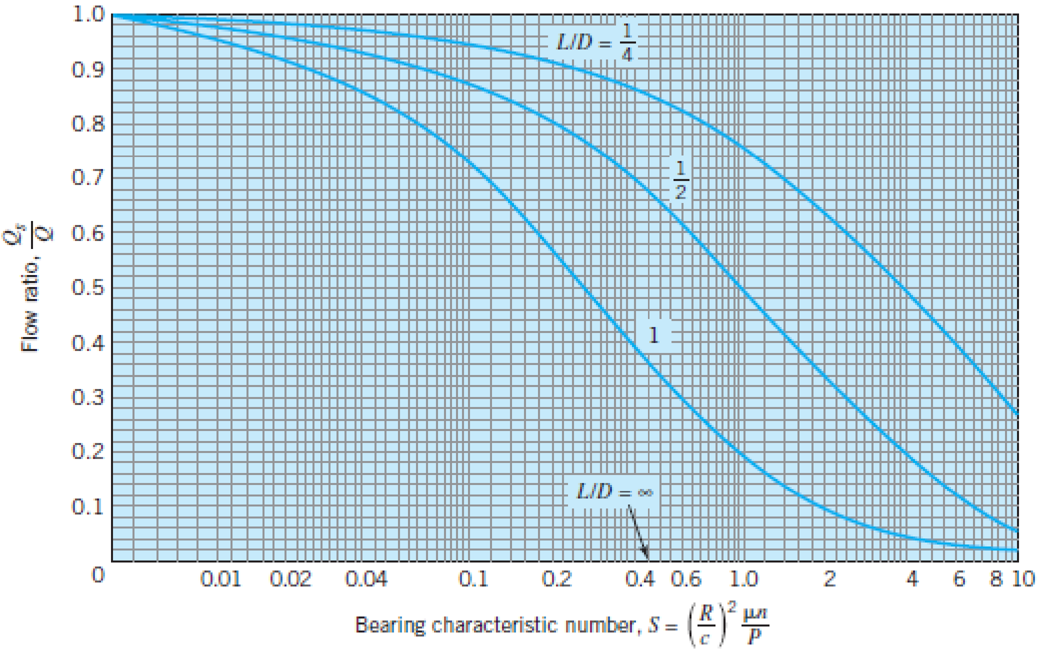


Fig.(12) Chart for the ratio of side flow to total flow

Note: You can use Table (1) to define the dimensionless performance parameters rather than using the above figures

Table(1): Dimensionless performance parameters for full journal bearing.

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q_s}{Q}\right)$	$\left(\frac{p}{p_{max}}\right)$
∞	0	1.0	∞	(70.92)	∞	π	0	—
	0.1	0.9	0.240	69.10	4.80	3.03	0	0.826
	0.2	0.8	0.123	67.26	2.57	2.83	0	0.814
	0.4	0.6	0.0626	61.94	1.52	2.26	0	0.764
	0.6	0.4	0.0389	54.31	1.20	1.56	0	0.667
	0.8	0.2	0.021	42.22	0.961	0.760	0	0.495
	0.9	0.1	0.0115	31.62	0.756	0.411	0	0.358
	0.97	0.03	—	—	—	—	0	—
	1.0	0	0	0	0	0	0	0
1	0	1.0	∞	(85)	∞	π	0	—
	0.1	0.9	1.33	79.5	26.4	3.37	0.150	0.540
	0.2	0.8	0.631	74.02	12.8	3.59	0.280	0.529
	0.4	0.6	0.264	63.10	5.79	3.99	0.497	0.484
	0.6	0.4	0.121	50.58	3.22	4.33	0.680	0.415
	0.8	0.2	0.0446	36.24	1.70	4.62	0.842	0.313
	0.9	0.1	0.0188	26.45	1.05	4.74	0.919	0.247
	0.97	0.03	0.00474	15.47	0.514	4.82	0.973	0.152
	1.0	0	0	0	0	0	1.0	—
$\left(\frac{1}{2}\right)$	0	1.0	∞	(88.5)	∞	π	0	—
	0.1	0.9	4.31	81.62	85.6	3.43	0.173	0.523
	0.2	0.8	2.03	74.94	40.9	3.72	0.318	0.506
	0.4	0.6	0.779	61.45	17.0	4.29	0.552	0.441
	0.6	0.4	0.319	48.14	8.10	4.85	0.730	0.365
	0.8	0.2	0.0923	33.31	3.26	5.41	0.874	0.267
	0.9	0.1	0.0313	23.66	1.60	5.69	0.939	0.206
	0.97	0.03	0.00609	13.75	0.610	5.88	0.980	0.126
	1.0	0	0	0	0	—	1.0	0

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q_s}{Q}\right)$	$\left(\frac{P}{p_{max}}\right)$
$\left(\frac{1}{4}\right)$	0.0	1.0	∞	(89.5)	∞	π	0	—
	0.1	0.9	16.2	82.31	322.0	3.45	0.180	0.515
	0.2	0.8	7.57	75.18	153.0	3.76	0.330	0.489
	0.4	0.6	2.83	60.86	61.1	4.37	0.567	0.415
	0.6	0.4	1.07	46.72	26.7	4.99	0.746	0.334
	0.8	0.2	0.261	31.04	8.8	5.60	0.884	0.240
	0.9	0.1	0.0736	21.85	3.50	5.91	0.945	0.180
	0.97	0.03	0.0101	12.22	0.922	6.12	0.984	0.108
	1.0	0	0	0	0	—	1.0	0

The oil film temperature rise can be calculated as

$$\Delta t = \frac{8.3 \times P \times (\frac{r}{c})f}{Q/rcnl}$$

Ex:

A full-journal bearing shown in figure (13) of diameter 60mm, length 30mm, with a radial clearance 0.05mm, carries a load of 3.6kN at a speed of 30rps. It is lubricated by SAE 30 oil, supplied at atmospheric pressure, and the average temperature of the oil film is 60^{o}C .Using the design charts, analyze the bearing.

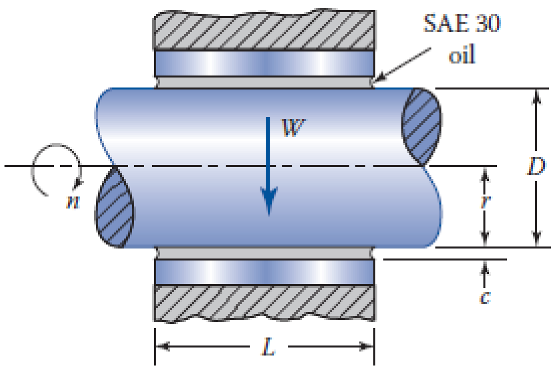


Fig.(13)

Solution

The pressure generated in the clearance gap can be calculated as:

$$P = \frac{W}{LD} = \frac{3600}{(0.03)(0.06)} = 2MPa$$

From figure(4) η=27mPa.s

Then the Somerfeld number can be calculated as

$$S = (\frac{r}{c})^2 \left(\frac{\eta n}{P}\right) = \left(\frac{30}{0.05}\right)^2 \frac{(0.027)(30)}{2 \times 10^6} = 0.146$$

After evaluating the Somerfeld number different journal parameters can be evaluated such as:

Minimum film thickness can be calculated by using figure (7). Use S = 0.146 with L/D = 1/2 to enter the chart in this figure:

$$\frac{h_o}{c} = 0.25 \quad or\ h_o = 0.25 \times c = 0.25 \times 0.05 = 0.0125mm$$

Then the eccentricity can be evaluated by using the following equation,

$e = c - h_o = 0.05 - 0.0125 = 0.0375$ mm

The eccentricity ratio can be evaluated as:

$$\varepsilon = \frac{e}{c} = \frac{0.0375}{0.05} = 0.75$$

The coefficient of friction can be evaluated by using figure (8). Use S = 0.146 with L/D = 1/2. Hence, from the chart in this figure,

$$\frac{r}{c} f = 4.8\ or\ f = 4.8\frac{c}{r} = 4.8 \times \frac{0.05}{30} = 0.008$$

The friction torque is then calculated by applying the following equation,

$$T_f = fWr = 0.008(3600)(0.03) = 0.864N.m$$

The frictional power lost in the bearing can be evaluated as:

$$Power = T_f \times \omega = 0.86 \times 2 \times \pi \times 30 = 162.1W\ or\ 0.162kW$$

The maximum film pressure can be evaluated by using figure(9) for S=0.146 and L/D=0.5.

$$\frac{P}{P_{max}} = 0.32$$

Hence $P_{max} = \frac{P}{0.32} = \frac{2}{0.32} = 6.25MPa$

Ex:

A journal bearing shown in figure (14) of 2-in. diameter, 1-in. length, and 0.0015-in. radial clearance supports a fixed load of 1000 lb when the shaft rotates 3000 rpm. It is lubricated by SAE 20 oil, supplied at atmospheric pressure. The average temperature of the oil film is estimated at 130°F. Using the Raimondi–Boyd charts, estimate the minimum oil film thickness, bearing coefficient of friction, maximum pressure within the oil film, and total oil flow rate through the bearing; the fraction of this flow rate that is recirculated oil flow; and the fraction of new flow that must be introduced to make up for side leakage

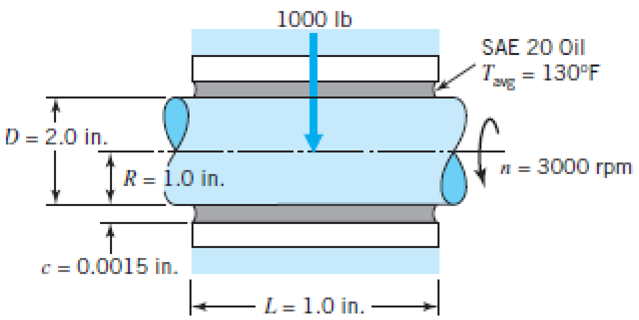


Fig.(14)

Solution

$$\boldsymbol{P = \frac{W}{LD} = \frac{1000}{(1)(2)} = 500psi}$$

$$\boldsymbol{\eta = 4 \times 10^{-6} reyn}$$

$$\boldsymbol{S = \left(\frac{R}{c}\right)^2 \left(\frac{\eta n}{P}\right) = (\frac{1}{0.0015})^2 \frac{(4 \times 10^{-6})}{500} = 0.18}$$

Use S = 0.18, L/D = 0.5 to enter all charts, and use units of inch-pound seconds consistently: From Figure (7) $h_0/c = 0.3$, hence $h_0 = 0.00045$ in.

From figure(8) $\left(\frac{R}{c}\right)\boldsymbol{f = 5.4, hence\ f = 0.008}$

From figure(9) $\frac{P}{p_{max}} = 0.32, hence\ p_{max} = 1562psi$

From figure(10), Φ=40^{o}

From figure(11),$\frac{Q}{RcL} = 5.15, hence\ Q = 0.39in^3/s$

From figure(12), $\frac{Q_s}{Q} = 0.81$, hence side leakage that must be made up by new oil represents 81% of the flow , the remaining 19%is recirculated.

Heat Dissipation and Equilibrium Oil Film Temperature:

The time rate of heat dissipation from oil can be evaluated as:

$$H = CA(t_o - t_a)$$

Where

H = time rate of heat dissipation (watts)

C = overall heat transfer coefficient (watts per hour per square meter per degree centigrade)

A = exposed housing surface area (square meters)

t_o = average oil film temperature (°C)

t_a = air temperature in the vicinity of the bearing housing (°C).

Values of *C* for representative conditions are given in Table (2). Values of *A* for pillow block bearings sometimes estimated as 20 times the bearing projected area (i.e., 20*DL*).

Table(2) Rough Estimates of Heat Transfer Coefficient *C* for Self-Contained Bearings

Bearing Type	C, W/(m²·°C)[Btu/(hr·ft²·°F)][a] Still Air	Average Air Circulation	Air Moving at 500 fpm
Oil ring or collar	7.4 (1.3)	8.5 (1.5)	11.3 (2.0)
Oil bath	9.6 (1.7)	11.3 (2.0)	17.0 (3.0)

The average oil film temperature can be found by the heat equilibrium as temperature rise can be evaluated as follows:

$$H = fw(2 \times \pi \times r \times n)$$

Or

$$CA(t_o - t_a) = fw(2 \times \pi \times r \times n)$$

The average oil film temperature can be evaluated approximately as

$$t_o = t_i + \frac{\Delta t}{2}$$

Where t_i= inlet oil film temperature (^{o}C)

The unit load capacity for the hydrodynamic journal bearings used for different industrial applications has been summarized in table (2).

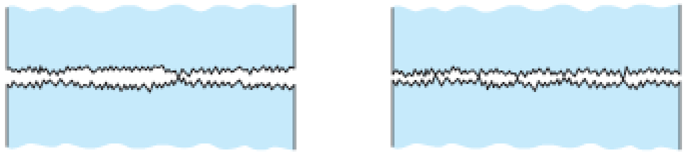
Table (3) Representative Unit Sleeve Bearing Load Capacities in Current Practice

Application	Unit Load Capacity, $P = W_{max}/LD$ MPa	psi
Relatively steady loads		
Electric motors	0.8–1.5	120–250
Steam turbines	1.0–2.0	150–300
Gear reducers	0.8–1.5	120–250
Centrifugal pumps	0.6–1.2	100–180
Rapidly fluctuating loads		
Diesel engines		
Main bearings	6–12	900–1700
Connecting rod bearings	8–15	1150–2300
Automotive gasoline engines		
Main bearings	4–5	600–750
Connecting rod bearings	10–15	1700–2300

•

Boundary and Mixed-Film Lubrication:

In boundary and mixed lubrication the sliding surfaces were separated by non-sufficient oil film, hence metal to metal contact occurs at the tip of the asperities as shown in figure (15).



(a) Mixed lubrication (b) Boundary lubrication

Fig.(15) Mixed and boundary lubricated bearings.

Boundary lubricated machine elements such as porous bearings and non-metal bearings are selected on the bases of (PV). Table (3) gives Operating Limits of Boundary-Lubricated Porous Metal while table (4) gives corresponding design recommendations for nonmetallic bearings.

Table(4): Operating Limits of Boundary-Lubricated Porous Metal Bearings

	Static *P*		Dynamic *P*		*V*		*PV*	
Material	**MPa**	**(ksi)**	**MPa**	**(ksi)**	**m/s**	**(fpm)**	**MPa · m/s**	**(ksi · fpm)**
Bronze	55	(8)	14	(2)	6.1	(1200)	1.8	(50)
Lead–bronze	24	(3.5)	5.5	(0.8)	7.6	(1500)	2.1	(60)
Copper–iron	138	(20)	28	(4)	1.1	(225)	1.2	(35)
Hardenable copper–iron	345	(50)	55	(8)	0.2	(35)	2.6	(75)
Iron	69	(10)	21	(3)	2.0	(400)	1.0	(30)
Bronze–iron	72	(10.5)	17	(2.5)	4.1	(800)	1.2	(35)
Lead–iron	28	(4)	7	(1)	4.1	(800)	1.8	(50)
Aluminum	28	(4)	14	(2)	6.1	(1200)	1.8	(50)

Table (5): Operating Limits of Boundary-Lubricated Nonmetallic Bearings

	P		Temperature		*V*		*PV*	
Material	**MPa**	**(ksi)**	**°C**	**(°F)**	**m/s**	**(fpm)**	**MPa · m/s**	**(ksi · fpm)**
Phenolics	41	(6)	93	(200)	13	(2500)	0.53	(15)
Nylon	14	(2)	93	(200)	3.0	(600)	0.11	(3)
TFE	3.5	(0.5)	260	(500)	0.25	(50)	0.035	(1)
Filled TFE	17	(2.5)	260	(500)	5.1	(1000)	0.35	(10)
TFE fabric	414	(60)	260	(500)	0.76	(150)	0.88	(25)
Polycarbonate	7	(1)	104	(220)	5.1	(1000)	0.11	(3)
Acetal	14	(2)	93	(200)	3.0	(600)	0.11	(3)
Carbon (graphite)	4	(0.6)	400	(750)	13	(2500)	0.53	(15)
Rubber	0.35	(.05)	66	(150)	20	(4000)	—	—
Wood	14	(2)	71	(160)	10	(2000)	0.42	(12)

Ex:

Following data is given for a 360^{o}hydrodynamic bearing

Radial load=3.2kN

Journal speed =1490rpm

Journal diameter=50mm

Bearing length =50mm

Radial clearance =0.05mm

Viscosity of lubricant=25cP

Assuming that the total heat generated in the bearing is carried by the total oil flow in the bearing, calculate

1. The coefficient of friction
2. Power lost in friction
3. Minimum oil film thickness
4. Flow requirement in l/min.
5. Temperature rise

Note: cP is centi-Poise=$\frac{dyn.s}{cm^2}$ to convert it to $\frac{N.s}{mm^2}$ the cP must be divided by 10^9

Solution:

$$\mathrm{P} = \frac{\mathrm{W}}{\mathrm{ld}} = \frac{3.2 \times 10^3}{(50)(50)} = 1.28\mathrm{N/mm^2}$$

$$S = (\frac{r}{c})^2 \frac{\eta n}{P}$$

$$S = \left(\frac{25}{0.05}\right)^2 \left(\frac{25}{10^9}\right)\left(\frac{1490}{60}\right)\left(\frac{1}{1.28}\right) = 0.121$$

$$\frac{l}{d} = \frac{50}{50} = 1$$

From table (1)

$\left(\frac{r}{c}\right) f = 3.22$ $\quad\left(\frac{h_o}{c}\right) = 0.4$ $\quad\left(\frac{Q}{rcnl}\right) = 4.33$

Therefore

$$f = 3.22\left(\frac{c}{r}\right) = 3.22\left(\frac{0.05}{25}\right) = 0.00644$$

The power loss can be evaluated as

$$(kW)_f = \frac{2 \times \pi \times n \times f \times W \times r}{10^6}$$

$$= \frac{2\pi(1490/60)(0.00644)(3.2 \times 10^3)(25)}{10^6} = 0.08$$

$$h_o = 0.4c = 0.4(0.05) = 0.02mm$$

$$Q = 4.33rcnl = 4.33(25)(0.05)\left(\frac{1490}{60}\right)(50) = 6720mm^3/s$$

Or

$$Q = (6720)(10^{-6} \times 60) l/min$$

$$\Delta t = \frac{8.3 \times P \times (\frac{r}{c})f}{Q/rcnl}$$

$$\Delta t = \frac{8.3 \times 1.28 \times 3.22}{4.33} = 7.9c^{o}$$

Ex: A 50mm diameter hardened and ground steel journal rotates at 1440rpm in a lathe turned bronze bushing which is 50mm long. For hydrodynamic lubrication, the minimum oil film thickness should be five times sum of surface roughness (c.l.a values) of journal and bearing. The data about machining method is as follows:

	Machining method	Roughness (C.L.a)
Shaft	grinding	1.6micron
Bearing	Turning/boring	0.8micron

The class of fit is H8d8 and the viscosity of the lubricant is 18cP.

Determine the maximum radial load that the journal can carry and still operate under hydrodynamic conditions.

Solution:

$$h_o = 5(1.6 + 0.8) = 12 micron\ or\ 0.012mm$$

The limits for the hole and the shaft are:

Hole limits $50^{+0.0,+0.039} mm$

Shaft limits $50^{-0.08,-0.119} mm$

Solution:

When the process is centered of the average size of the hole and the shaft will be:

Hole diameter=50+0.039/2=50.0195mm

Shaft diameter=50+(-0.08-0.119)/2=49.9005mm

Hence the diametral clearance= 50.0195-49.9005=0.119mm

The radial clearance is

$$c = \frac{0.119}{2} = 0.0595mm$$

And

$$\frac{h_o}{c} = \left(\frac{0.012}{0.0595}\right) = 0.2$$

$$\frac{l}{d} = \frac{50}{50} = 1$$

From table (1)

S=0.0446

Since

$$S = (\frac{r}{c})^2 \frac{\eta n}{P}$$

Hence

$$P = (\frac{r}{c})^2 \frac{\eta n}{S} = (\frac{25}{0.0595})^2 \times \left(\frac{18}{10^9}\right) \times \left(\frac{1440}{60}\right) \times \left(\frac{1}{0.0446}\right)$$
$$= 1.71N/mm^2$$

$$W = Pld = (1.71)(50)(50) = 4275N$$