Two Bearing-Related Topics:

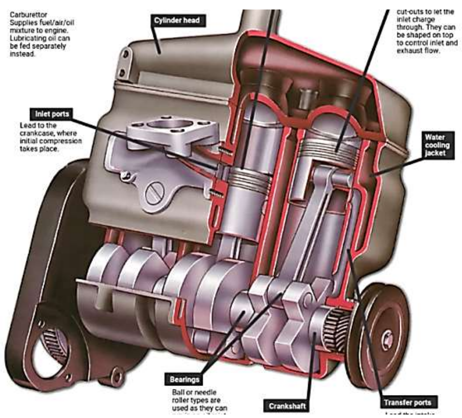
Ch. 11 Rolling-Contact Bearings
Ch. 12 Lubrication and Journal Bearings

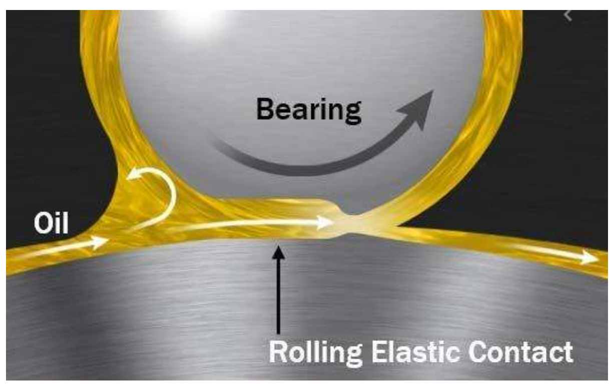
Essential Terminology in Tribology

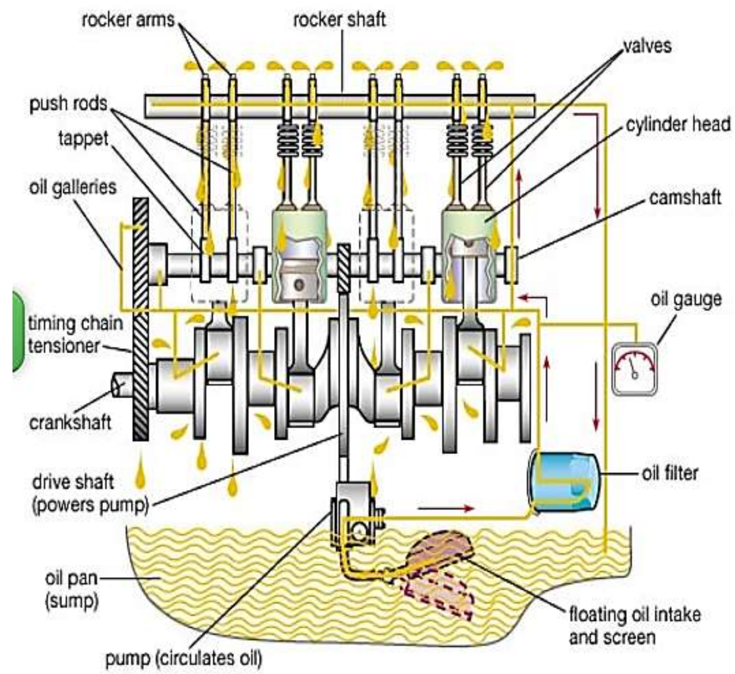
Lubricant: Any interposed substance that reduces friction and wear of a load transmitting surface.

Lubrication: Any active or passive mean/mechanism delivering lubricant to load transmitting interface of a bearing.

Purpose: The synergism of all of the above helps to <u>lower friction</u>, <u>reduce material wear</u>, <u>prolong service life</u>, and <u>improve operation reliability</u> of all mechanically driven equipment.

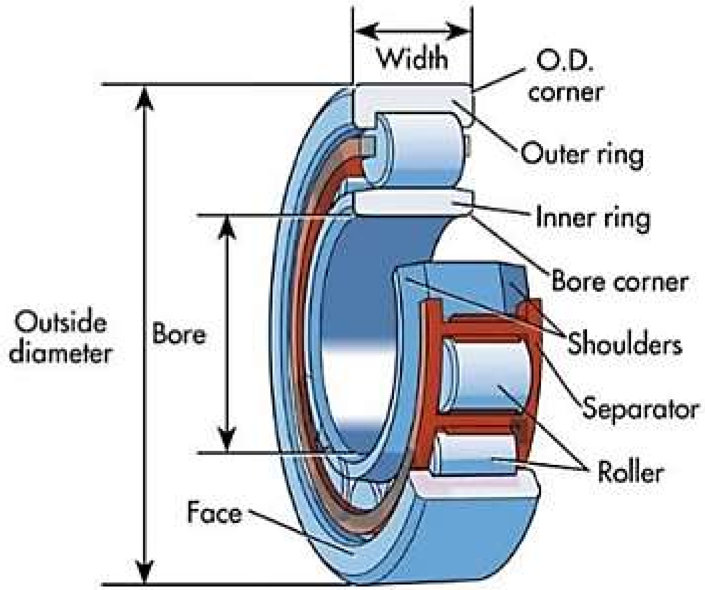




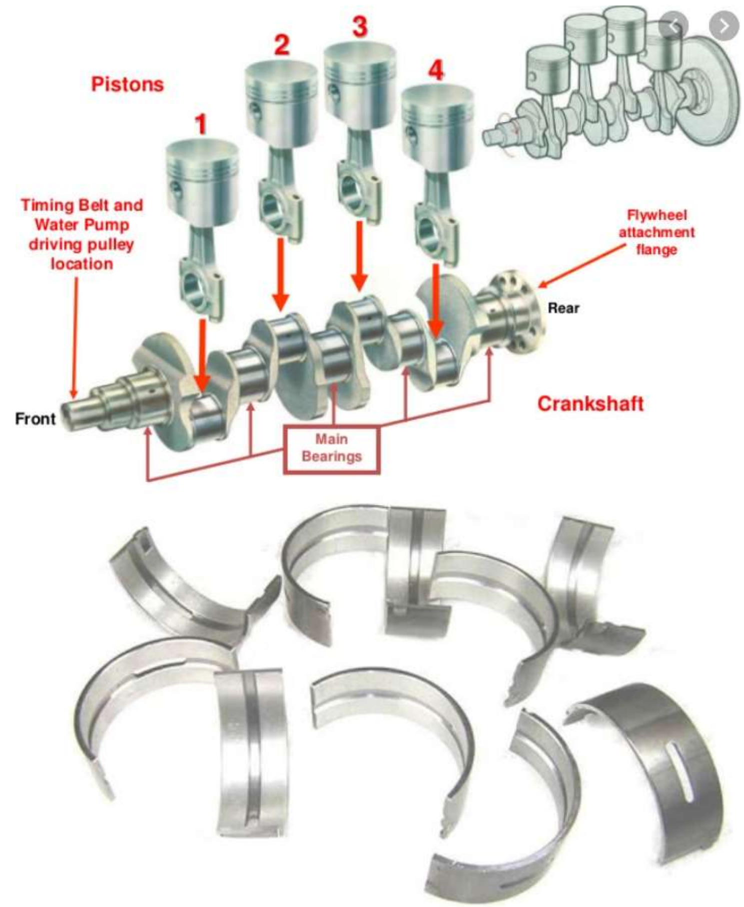


Examples of Bearing Configuration

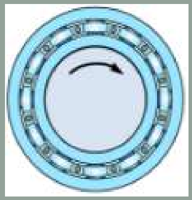
- Cylindrical Roller Bearings

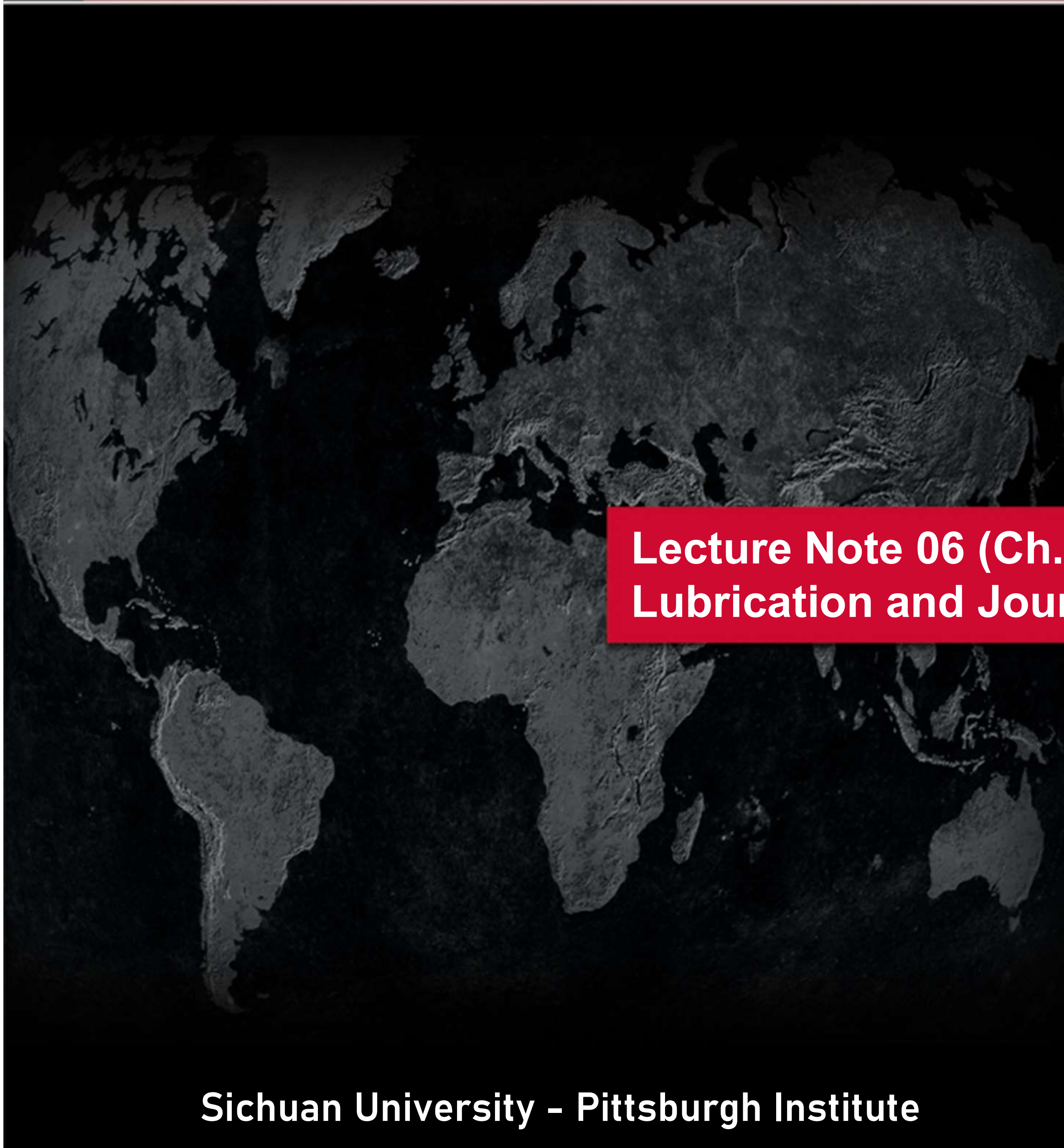


- Crankshaft Radial Bearings



Comparison Between Journal vs Rolling Element Bearings Performance and Operation Characteristics

	Rolling Element Bearings 	Journal Bearings 
Friction Torque, Start	**1**	28-80
Friction Torque, Running	**~0.5**	~1.4
Lubrication (Grease)	**Yes**	No
Lubrication (Oil)	**Yes**	**Yes**
Vertical Mounting	**Yes**	Special Design
High Speed	**Yes**	**Yes**
Low Speed	**Yes**	No
Shock Load Resistance	Weak	**Resilient**
Noise Level	High	**Low**
Space/Weight	High	**Low**
Assembly/Maintenance	Challenging	**Low-Moderate**
Cost	High	**Low**



Lecture Note 06 (Ch.12)
Lubrication and Journal Bearings

Instructor: Ping C. Sui, Ph.D.
ME 1029 Machine Design 2

Fall 2021

Sichuan University - Pittsburgh Institute

Preamble

- The application field for journal bearings is immense:
  - crankshaft and connecting-rod bearings of an automotive engine
  - steam turbines of power-generating stations
  - downhole drilling equipment

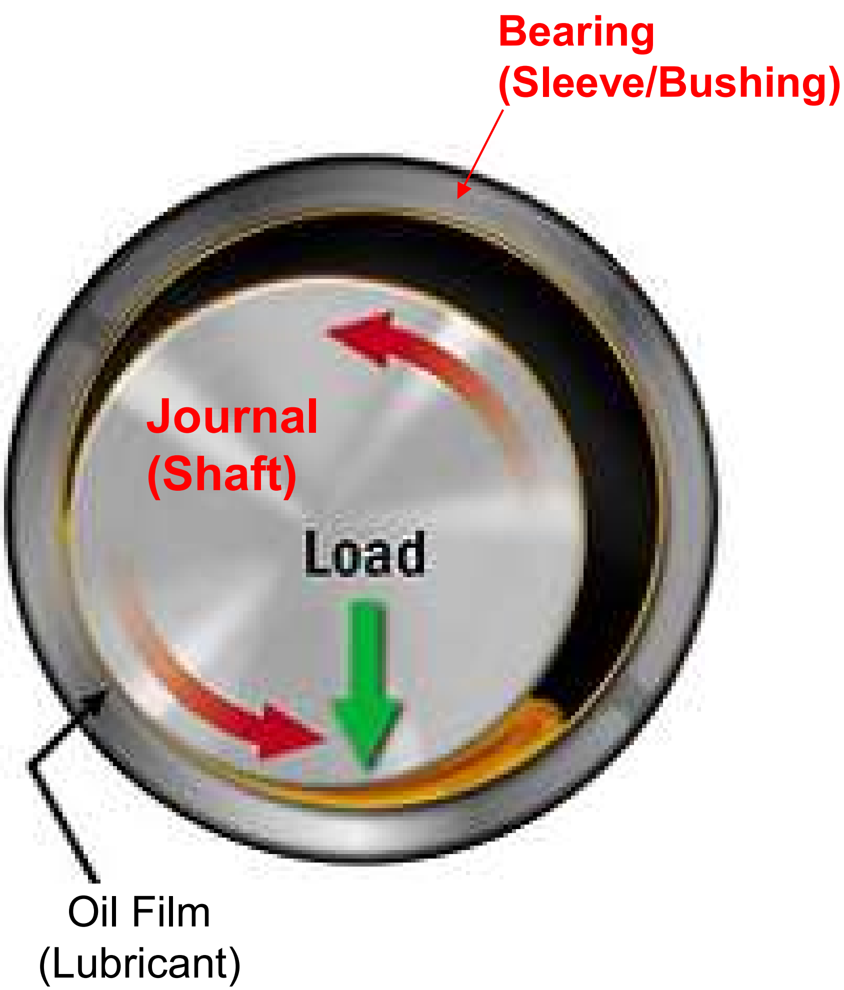
- Much of the area studied thus far has been based on engineering studies, such as statics, dynamics, the mechanics of solids, metal processing, mathematics, and metallurgy.

- In the study of lubrication and journal bearings, additional fundamental studies, such as chemistry, fluid mechanics, thermodynamics, and heat transfer, must be utilized in developing the material.

Elements in a Journal Bearing

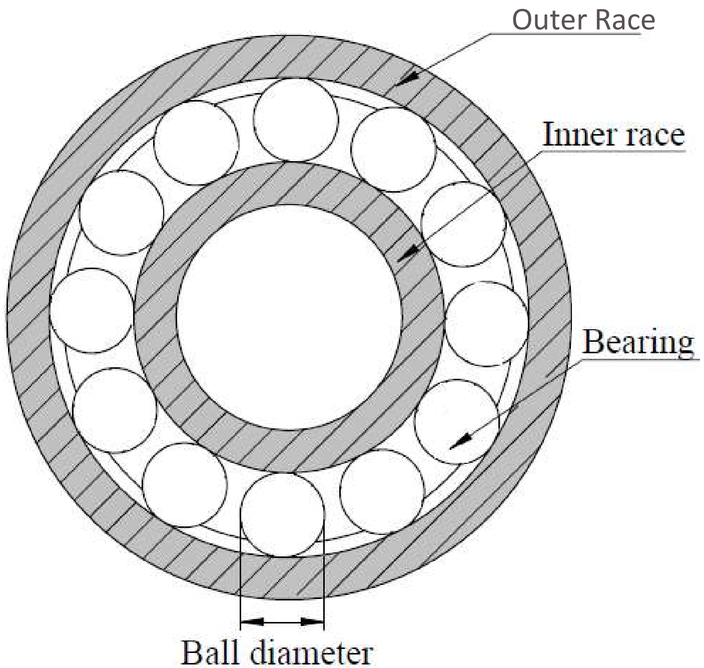
Journal bearing is also known as:

- Sliding bearing;
- Sleeve bearing; and
- Bushing, etc.

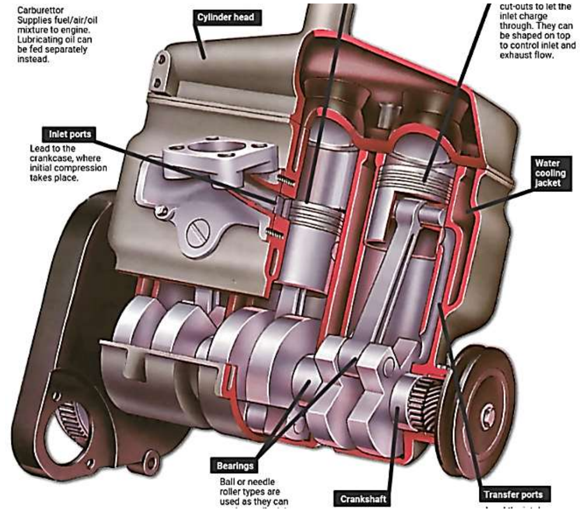


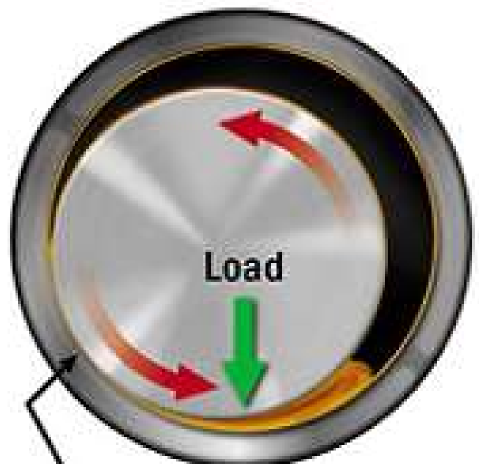
Comparison of Cross-Sectional Geometry

Rolling Element Bearing



- Relative Motion
- Load-Carrying Area

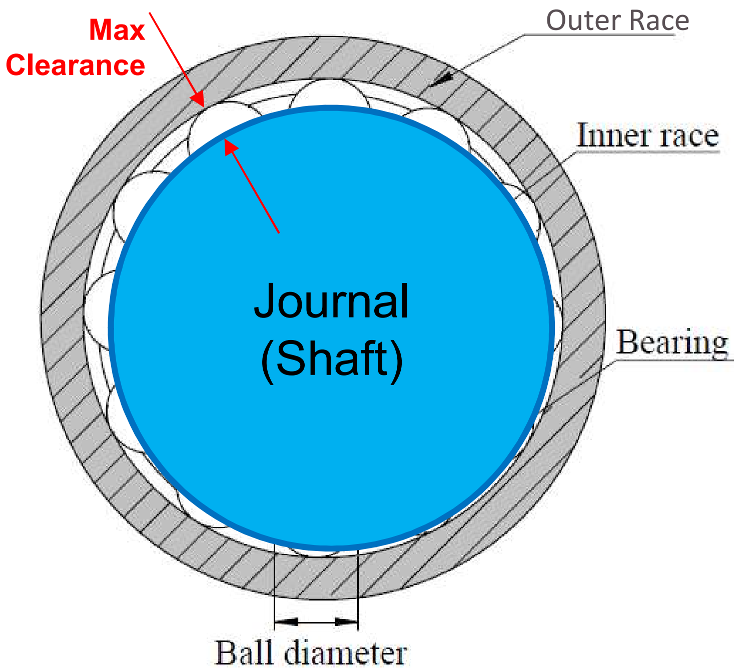




Journal Bearing

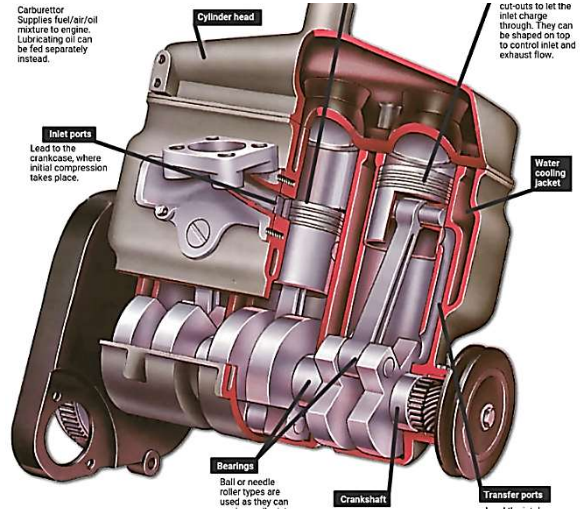
Comparison of Cross-Sectional Geometry

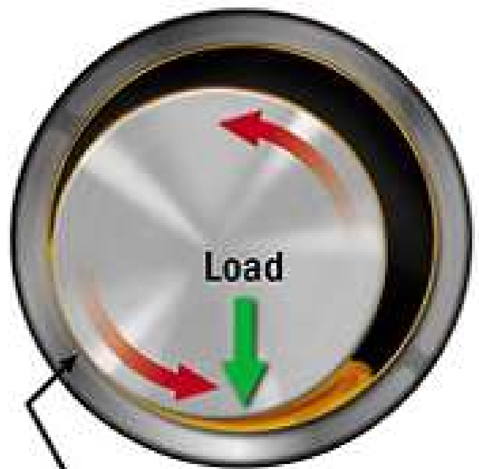
Rolling Element Bearing



Typical journal bearing clearance
is scale of millimeter range.

<u>Question:</u> Which type has a greater bearing area?





Journal Bearing

Topics Covered in This Lecture

- 12–2 Viscosity
- 12–3 Petroff's Equation
- 12–4 Stable Lubrication
- 12–5 Thick-Film Lubrication
- 12–6 Hydrodynamic Theory
- 12–7 Design Considerations
- 12–8 The Relations of the Variables
- 12–9 Steady-State Conditions in Self-Contained Bearings
- 12–10 Clearance
- 12–11 Pressure-Fed Bearings
- 12–12 Loads and Materials
- 12–13 Bearing Types
- 12–14 Thrust Bearings
- 12–15 Boundary-Lubricated Bearings

12-2 Viscosity

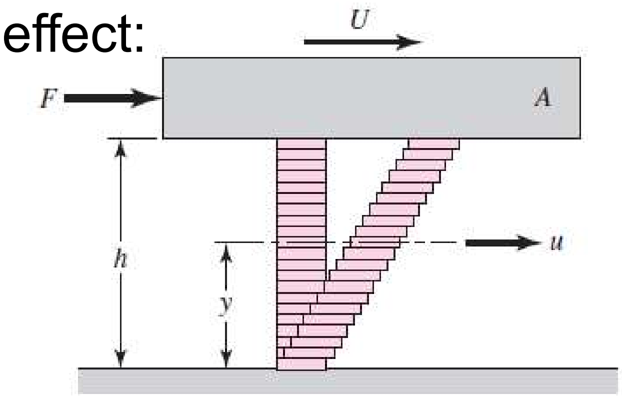
What is Viscosity?

<u>Newton's Postulate</u>

- Plate A moving with a velocity U on a film of lubricant of thickness h.
- Oil molecules were visualized as rolling in layers between flat planes.
  - Bottommost layer does not move at all,
  - uppermost layer moves at velocity of U, and
  - the layers in between move with a velocity proportional to $\frac{y}{h}$.
- Friction between two plates is due solely to the internal friction of the liquid, namely, its viscosity. Newton's viscous effect:

$$F = \mu A \frac{u}{h} \qquad \tau = \frac{F}{A} = \mu \frac{du}{dy}$$

μ : absolute (or dynamic) viscosity in unit of $\left(\frac{F \cdot T}{L^2}\right)$



Viscosity μ is a measure of the **internal frictional resistance** of the fluid.

Kinematic Viscosity and Dynamic Viscosity

- Dynamic (absolute) viscosity (μ)
  - force required to maintain the fluid flow at a particular flow rate
  - usually used when the fluid is subjected to any external force
- Kinematic viscosity (ν)
  - a fluid's internal resistance to flow under gravitational forces
  - ratio of dynamic viscosity to density of the fluid: ($\mu = \rho\nu$)

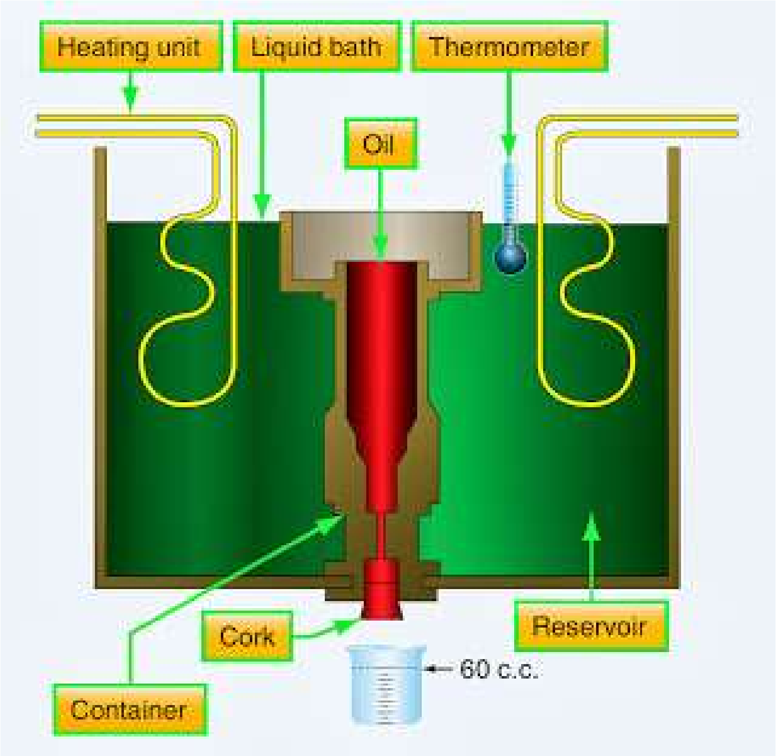
Kinematic Viscosity Units	Dynamics Viscosity Units
cSt (centiStoke) = mm^2/s Stoke (St); 1St = 100 cSt	Poise ($dyne*s/cm^2$) 1 ceterpoise (cP) = 1 mPa·sec $1\,reyn = \frac{1 \cdot lbf \cdot sec}{in^2} = 6890\,Pa \cdot sec$

- Customary to use the **centipoise** (cP) in analysis for convenience. When the viscosity is expressed in cP, it is designated by **Z**.

$$\boxed{\begin{aligned} \mu(\mathrm{Pa\cdot s}) &= (10)^{-3}\mathrm{Z}\ (\mathrm{cP}) \\ \mu(\mathrm{reyn}) &= \frac{\mathrm{Z}\ (\mathrm{cP})}{6.89(10)^6} \\ \mu(\mathrm{mPa\cdot s}) &= 6.89\ \mu'(\mu\mathrm{reyn}) \end{aligned}}$$

Measurement of Kinematic Viscosity

- **Saybolt Universal Viscometer (SUV)** is the ASTM standard method (D2161) for kinematic viscosity (ν) measurement
- Measuring the time in seconds for 60 mL of lubricant at a specified temperature to run through a tube 17.6 mm in diameter and 12.25 mm long.

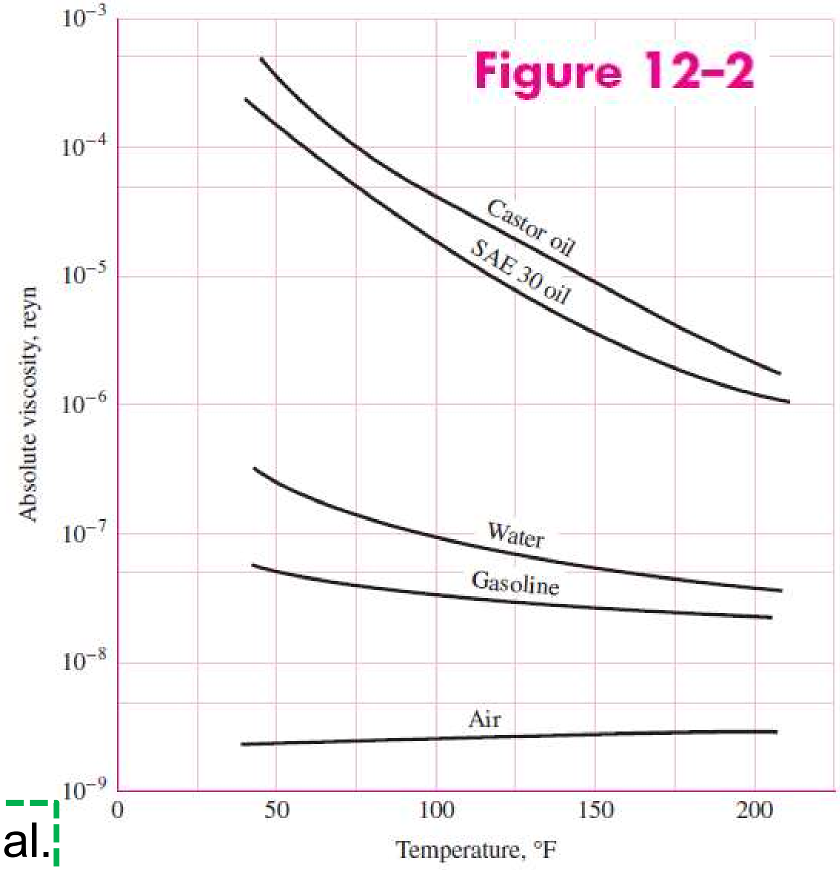


Measurement of Kinematic Viscosity (SUV)

- Kinematic Viscosity (ν) vs. Temperature Relationship

$$\nu = \left(0.22\, t\ - \frac{180}{t}\right)\ 10^{-6}$$

$$\mu = \rho\nu = \rho\left(0.22\, t\ - \frac{180}{t}\right)\ 10^{-6}$$



- Units:
  - ν (m^2/sec)
  - μ (Pa-sec)
  - ρ (kg/m^3)
  - t (sec)
- Generally speaking,
  - Temperature up, viscosity down

$$\tau = \frac{F}{A} = \mu\frac{du}{dy}$$

  - Friction goes down, if all else are equal.
- Lowering viscosity is usually effective in friction reduction.
- Question: Is temperature good for friction?

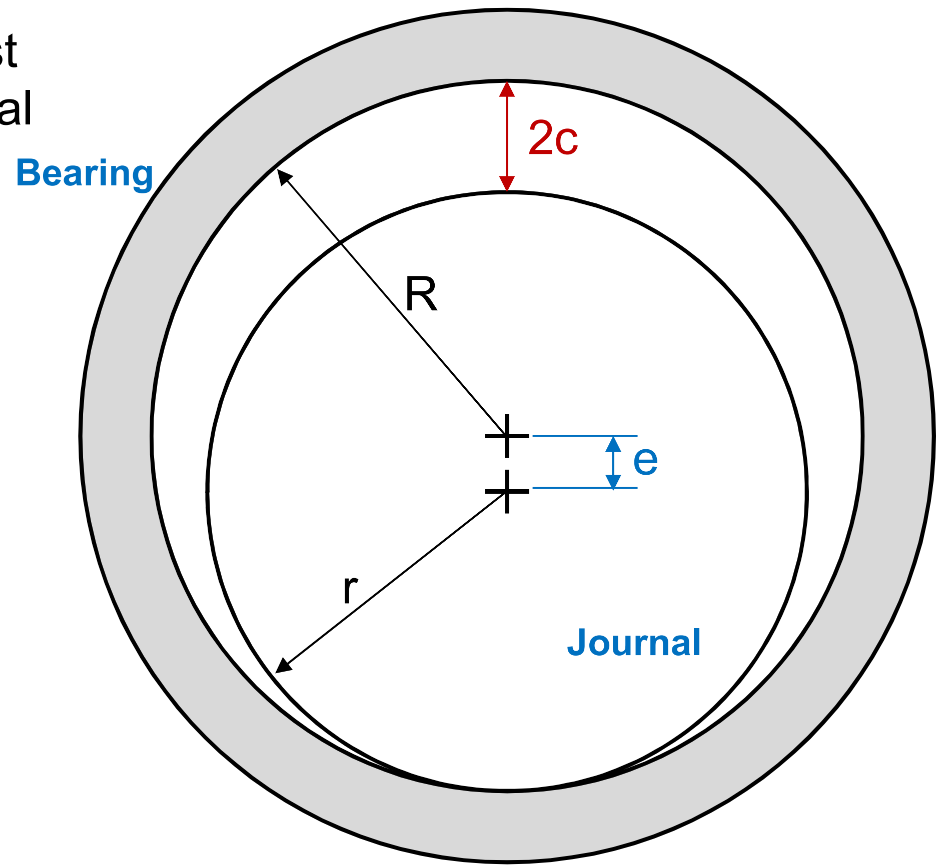
12-5 Thick-Film Lubrication
12-6 Hydrodynamic Theory

Bearing Diametral Clearance and Eccentricity

- Bearing Diametral Clearance (2c)
  - Generally considered as the most critical design parameter in journal bearing design

- Eccentricity (e)
  - Distance between journal center and bearing center

- This is also the configuration of a full bearing, which means the bearing wrapped around the journal in full (360°).



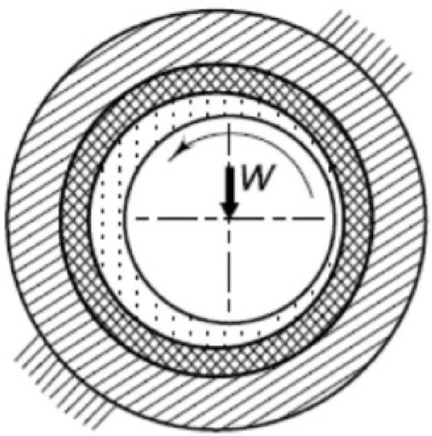
Full Bearing and Partial Bearing

Advantages of partial bearings compared to full journal bearings:

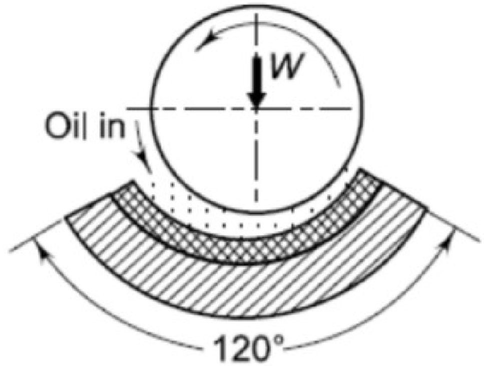
- Simple in construction
- Easy to supply lubricating oil
- Frictional losses are less ► temperature rise is low

Disadvantage of a Partial Bearing:

- Can take load in fixed radial direction, and
- usually in low speed applications.

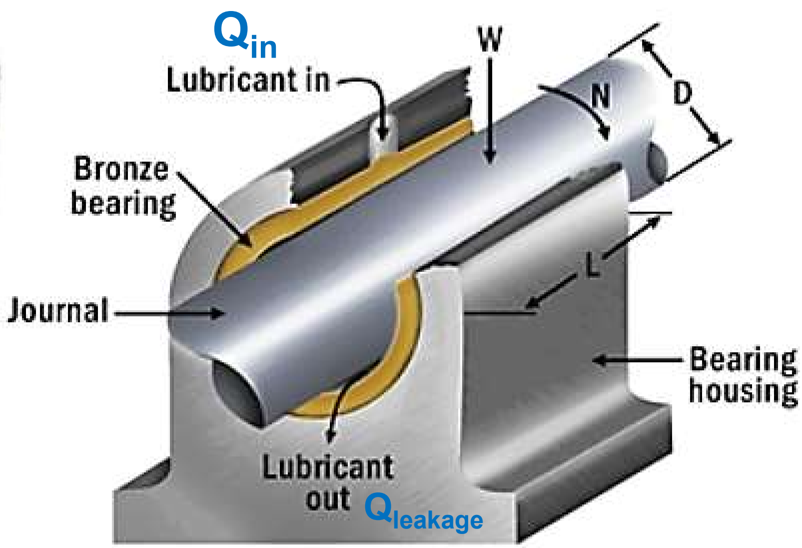


(a) Full bearing



(b) Partial bearing

Operation Principle of Journal Bearings (Hydrodynamic Theory)



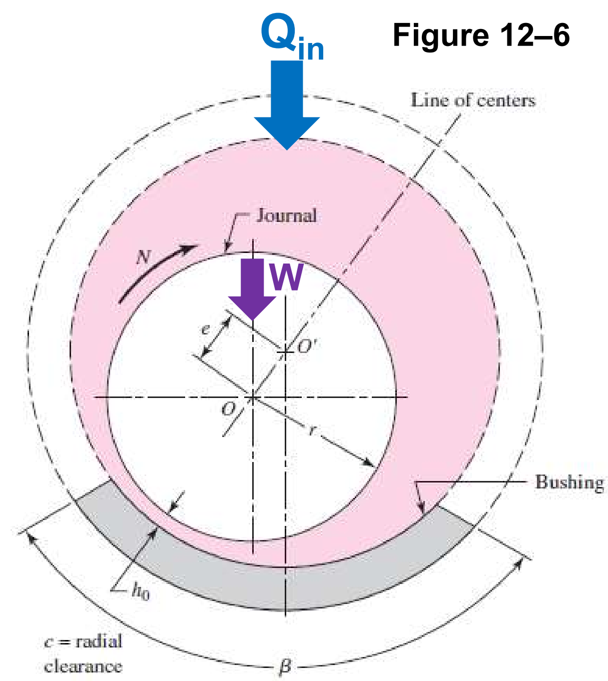


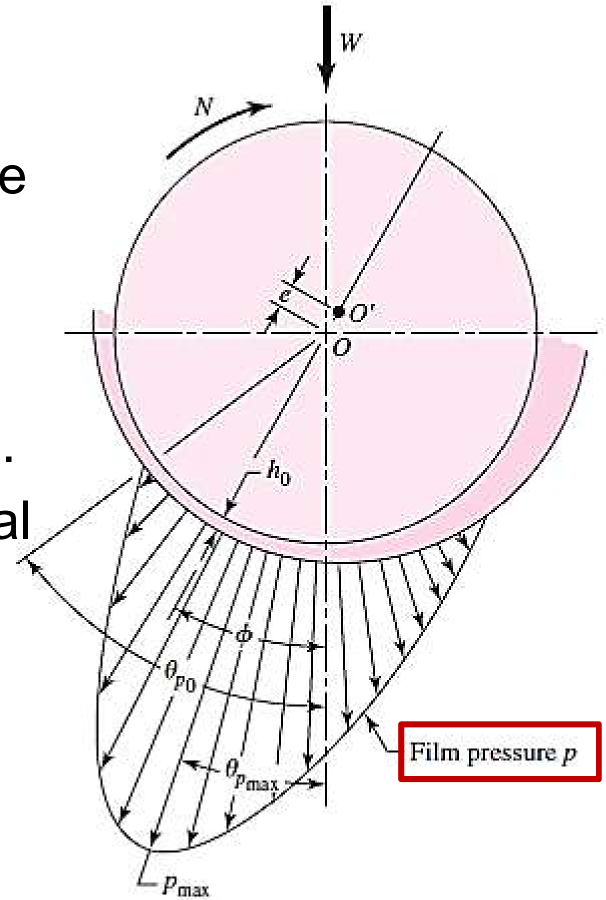
Figure 12–6

- Lubricant adhering to the journal surface is being pulled by the moving surface into a narrowing, wedge-shaped space.
- Shaft rotation has to be fast enough to retain a portion of the inlet flowrate (Q_{in}) within wedge-shaped bearing clearance.
- The rest of Q_{in} is leaked out from two side edges ($Q_{Leakage}$) of bearing, which is called the side leakage.

Film Thickness and Pressure Formation (Hydrodynamic Theory)

When $Q_{in} \geq Q_{leakage}$:

- Lubricant retained in the wedge-shaped space separates the journal from bearing to prevent metal-to-metal contact.
- This action also creates a film pressure of sufficient intensity to support the bearing load.
- The formed pressure is also forcing the journal over to the other side.

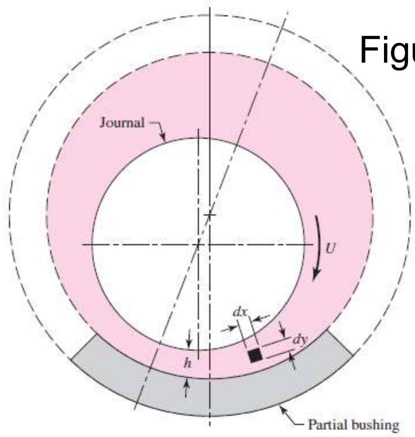


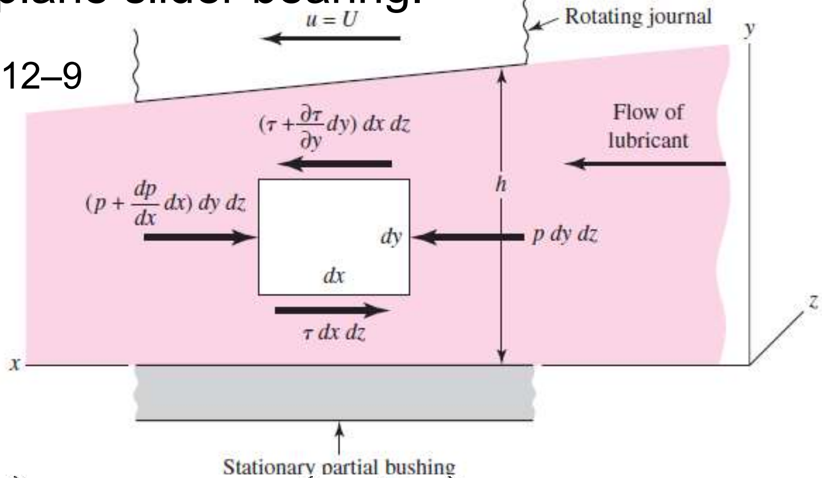
Q: What happen if $Q_{in} < Q_{leakage}$?

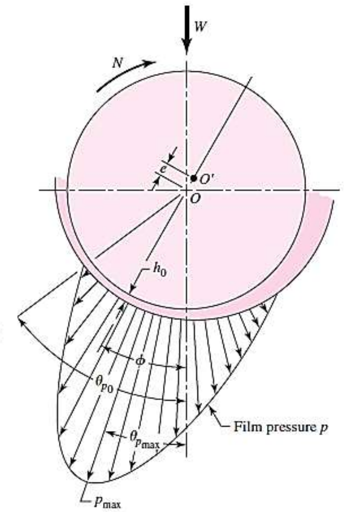
Hydrodynamic Theory (Osborne Reynolds, 1886)

Assumed fluid films were so thin in comparison with the bearing radius that the curvature could be neglected. This enabled replacing the curved partial bearing with a flat bearing, called a plane slider bearing.

Figure 12–9







$$\sum F_x = p\, dy\, dz - \left(p + \frac{dp}{dx} dx\right) dy\, dz - \tau\, dx\, dz + \left(\tau + \frac{\partial \tau}{\partial y} dy\right) dx\, dz = 0$$

$$\boxed{\frac{dP}{dx} = \frac{\partial \tau}{\partial y}}$$

Pressure buildup along the lubricant entraining direction is resulting from shear rate change across the film thickness.

$$\frac{dP}{dx} = \frac{\partial \tau}{\partial y} = \mu \frac{\partial^2 u}{\partial y^2}$$

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

Reynolds equation

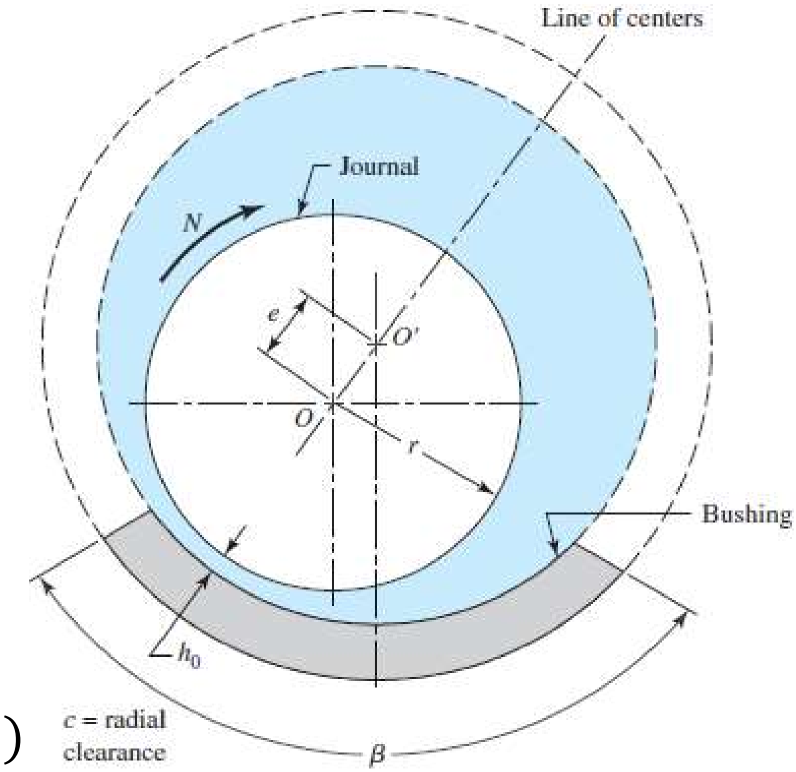
There is <u>no general analytical solution</u> to Reynold's equation.

Journal Bearing Nomenclatures

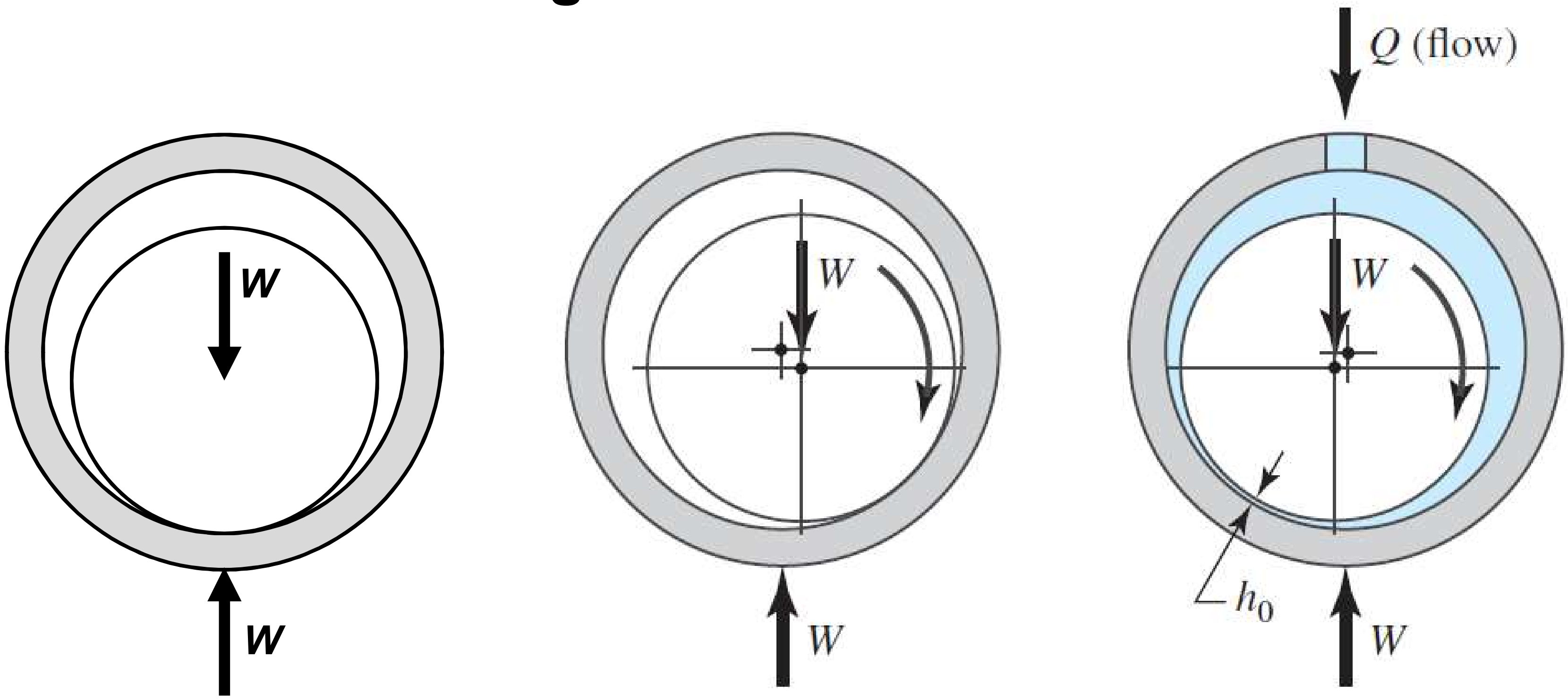
- R: bearing (bushing) radius
- r: journal radius
- $c = R - r$ (Radial Clearance)
- $2c$ (Diametral Clearance)

- O: Center of journal
- O': Center of bearing
- $e = OO'$=eccentricity
- $\epsilon = \frac{e}{c} = eccentricity\ ratio\ (0 \leq \epsilon \leq 1)$
- $h_o = c - e$ (Minimum Film Thickness, MFT)

$$c = h_o + e, \quad \frac{h_o}{c} = 1 - \epsilon$$



Journal and Bearing Contact



Dry,
No Rotation

Dry,
Journal Rotation

Lubricated,
Journal Rotation

Under hydrodynamic lubrication, journal is eccentrically located with respect to the bearing.

12-3 Petroff's Equation: COF Estimation

Petroff in 1883 used a concentric shaft to first explain the phenomenon of bearing friction.

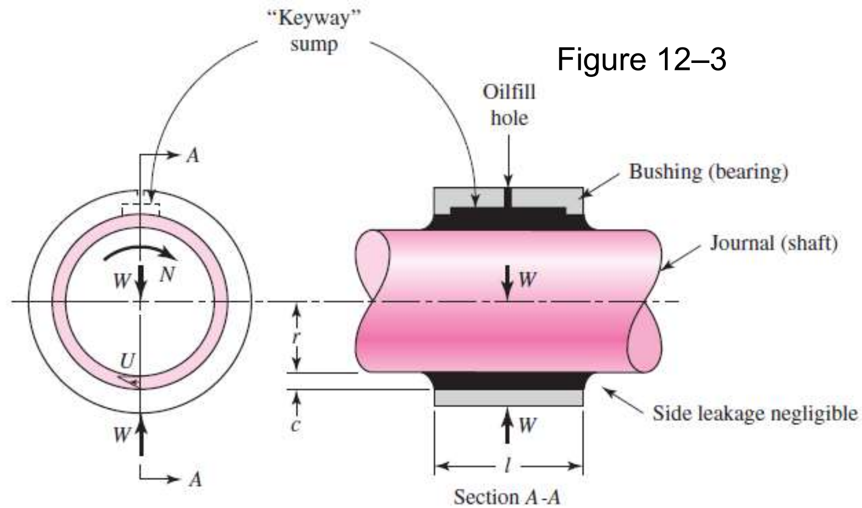


Figure 12–3

Shear stress around shaft: $\tau = \mu \frac{U}{h} = \frac{2\pi r \mu N}{c}$

- r: shaft radius
- c: radial clearance
- l: bearing length
- N: shaft rotational speed (rev/s)
- W: bearing normal force

Shaft torque: $T = \tau A r = \frac{2\pi r \mu N}{c}(2\pi r l) r = \frac{4\pi^2 r^3 l \mu N}{c}$

Shaft average pressure: $P = \frac{W}{2rl}$ (Unit Pressure)

Frictional torque: $T = fWr = 2r^2 f l P$

Solving for COF (f): $\boxed{f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}}$ (Petroff's equation)

Significance of Petroff's equation is that it provides a quick and simple mean of obtaining a reasonable estimate of COF for a lightly loaded bearing.

Definition of Sommerfeld Number

Sommerfeld Number: $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$

- Also known as bearing characteristic number
- Unit: dimensionless

Petroff's equation: $f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}$

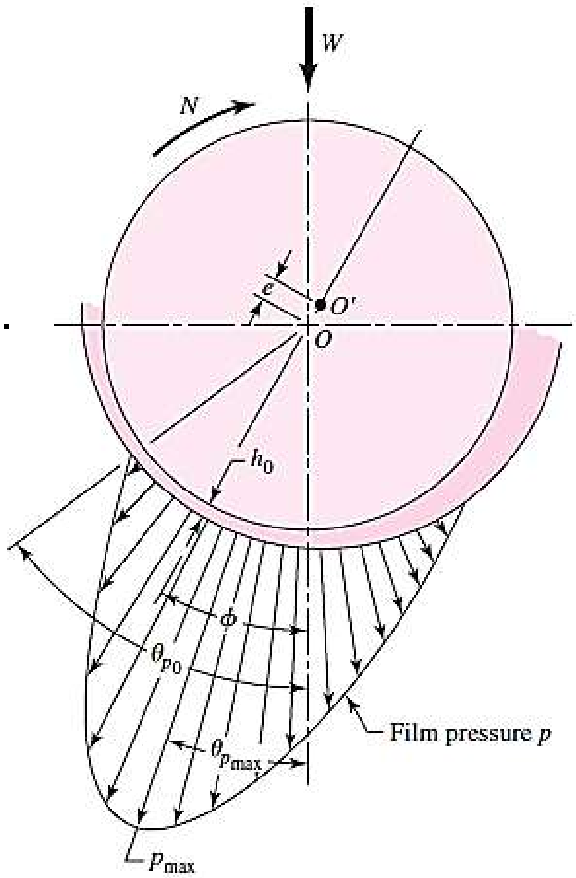
$$f \frac{r}{c} = 2\pi^2 \frac{\mu N}{P} \left(\frac{r}{c}\right)^2 = 2\pi^2 S$$

12-8 The Relations of the Variables

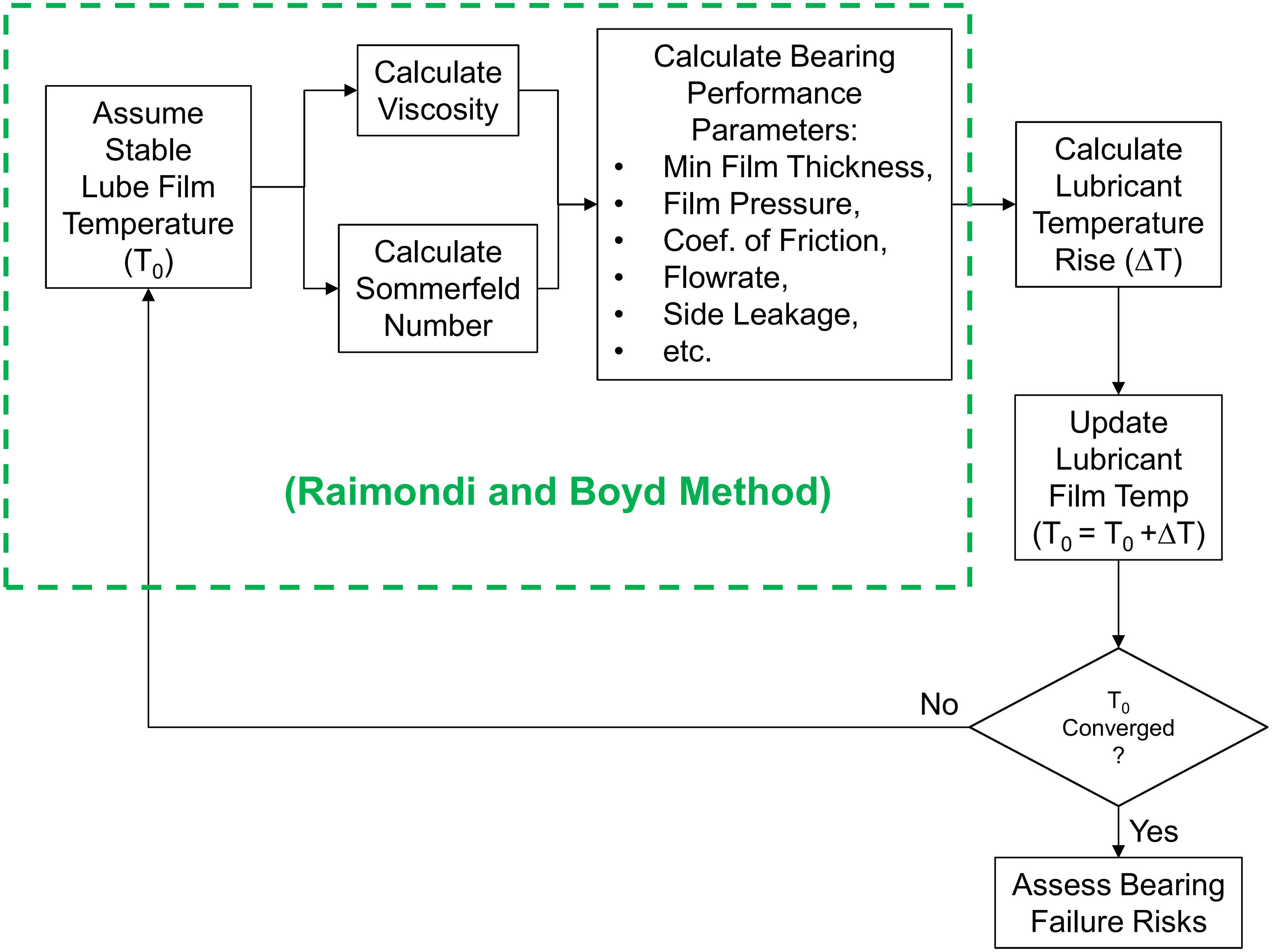
- Raimondi and Boyd used numerical solution to solve the Reynolds equation. They presented their numerical results relating the different variables in the form of charts (Fig. 12-16 to 12-21).
- Charts presented are for **full bearings** and **non-pressure-fed bearings**.

Viscosity Charts (Figs. 12–12 to 12–14)

- Lubricant viscosity depends on entraining lubricant temperature.
- Raimondi-Boyd analysis assumed that lubricant viscosity is constant as it passes through the bearing.
  - Since work is done on the lubricant during this flow; therefore, the temperature of the oil is higher when it leaves the loading zone than it was on entry.
- Since the analysis is based on a constant viscosity, immediate problem is to determine the value of viscosity to be used in the analysis.
- Procedures for estimation of bearing film temperature rise (T_{av}) will be discussed later.



Bearing Design Workflow



Viscosity Charts

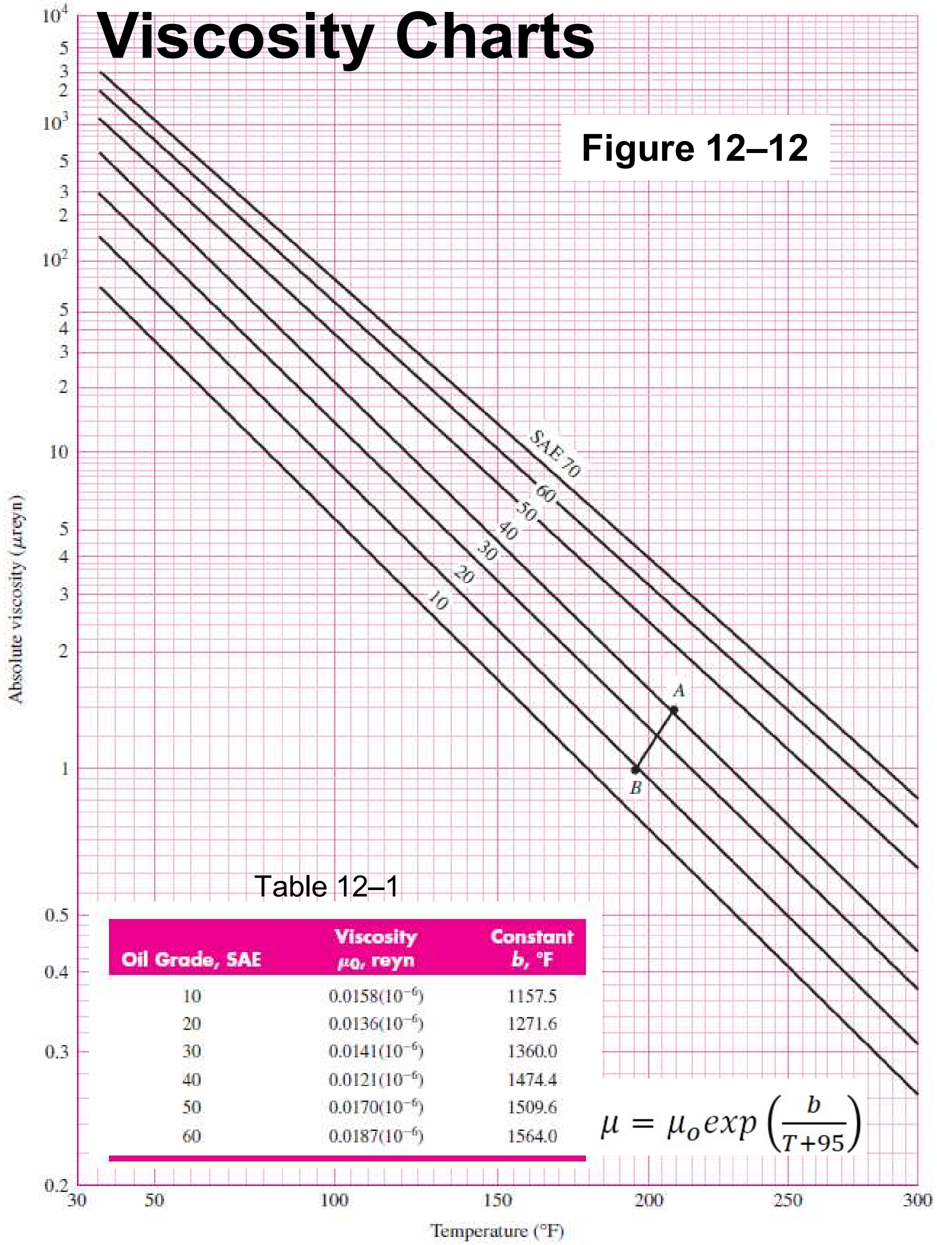


Figure 12–12

Table 12–1

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

$$\mu = \mu_o exp\left(\frac{b}{T+95}\right)$$

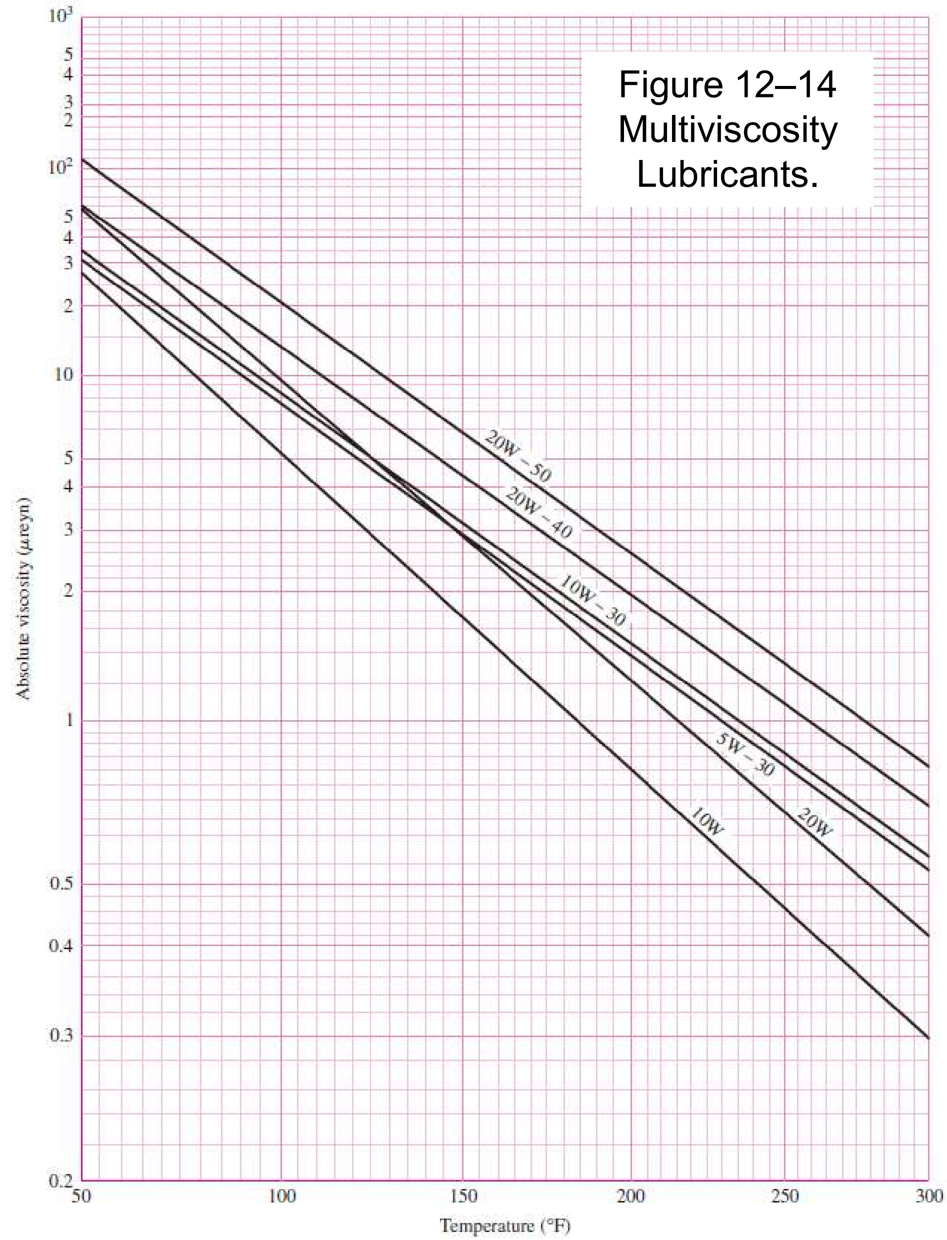


Figure 12–14 Multiviscosity Lubricants.

Viscosity Calculations

- Dynamic Viscosity μ (unit: reyn)

$$\mu = \mu_o \, Exp\left(\frac{b}{T + 95}\right)$$

- Constants μ_o and b are from Table 12-1.
- T: Average Oil Film Temperature Rise (T_{av}, unit: °F)

Table 12–1

Oil Grade, SAE	Viscosity μ_0, reyn	Constant b, °F
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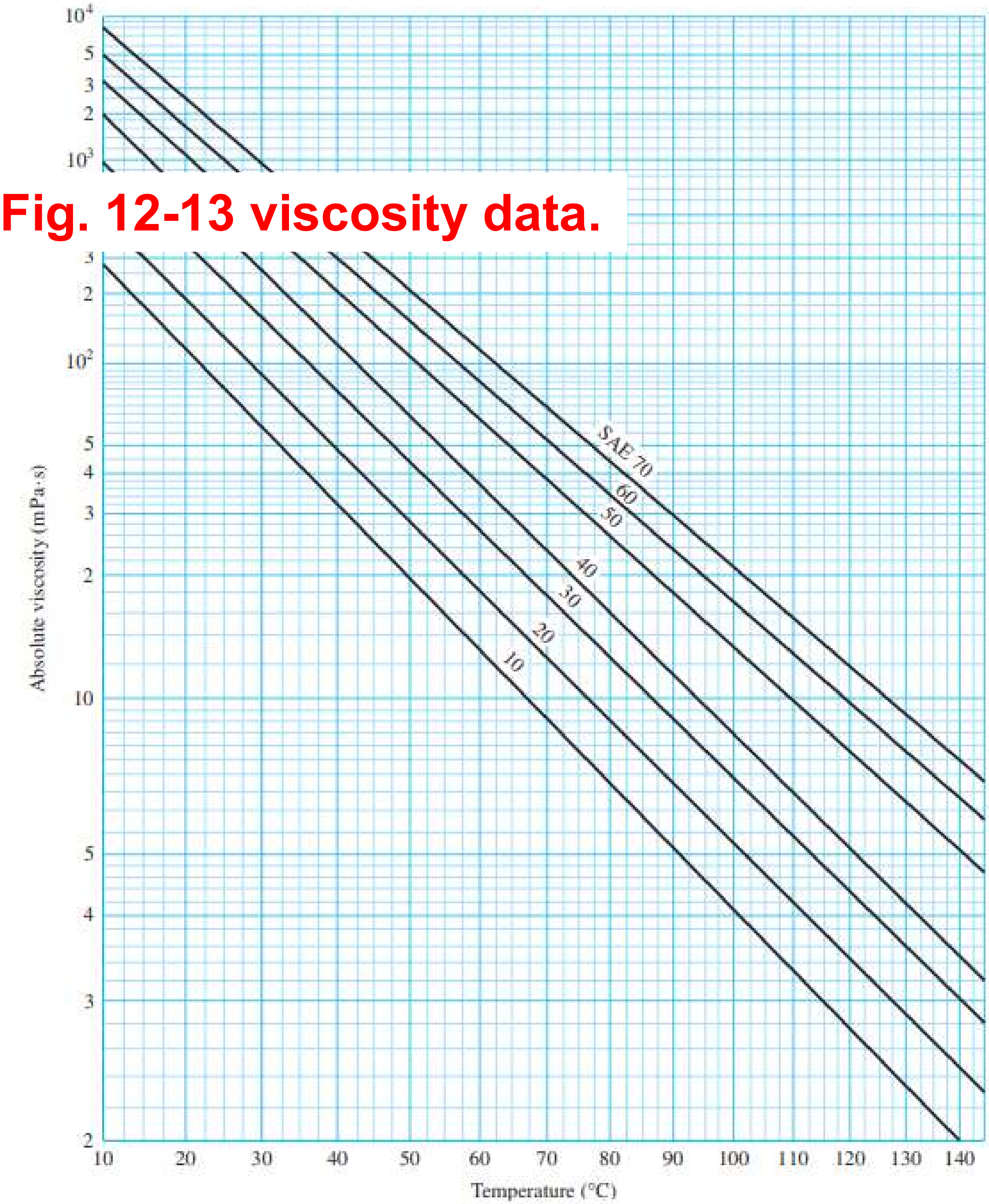
50	$0.0170(10^{-6})$	1509.6
60	$0.0187(10^{-6})$	1564.0

Figure 12–13

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



Minimum Film Thickness (h_o) & Its Angular Position (ϕ)

EXAMPLE 12–1

Determine h_0 and ε using the following given parameters:

Lubricant viscosity μ = 4 μreyn,

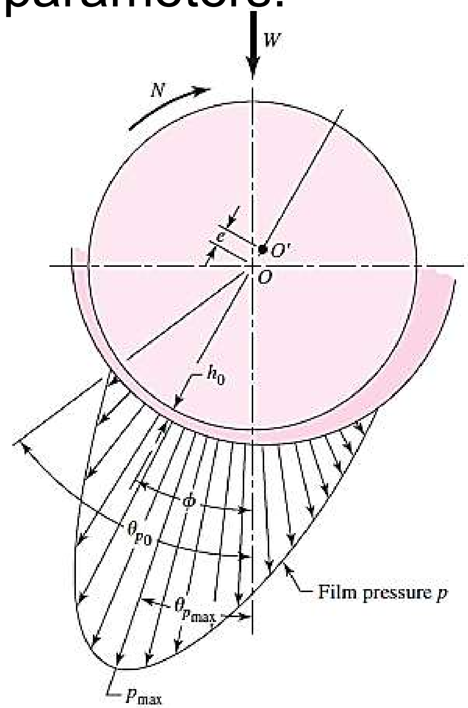
Journal rotational speed N = 30 rev/s,

Bearing Load W = 500 lbf (bearing load),

Journal radius r = 0.75 in,

Bearing radial clearance c = 0.0015 in

Bearing Length l = 1.5 in



Given viscosity is assumed to be at stable temperature.

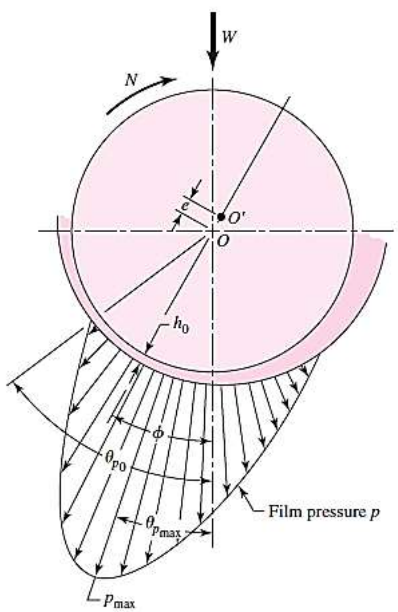
EXAMPLE 12–1 (Cont'd) ►min Film Thickness (Fig. 12-16)

Nominal bearing pressure $P = \frac{W}{2\,r\,l} = \frac{500}{2{\cdot}0.75{\cdot}1.5} = 222\ psi$

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

Since $\frac{l}{d} = \frac{1.5}{2{\cdot}0.75} = 1$, then Fig 12-16 gives $\frac{h_o}{c} = 0.42$ and $\epsilon = 0.58$

Minim film thickness $h_o = c \cdot 0.42 = 0.0015 \cdot 0.42 = 0.00063\ in$



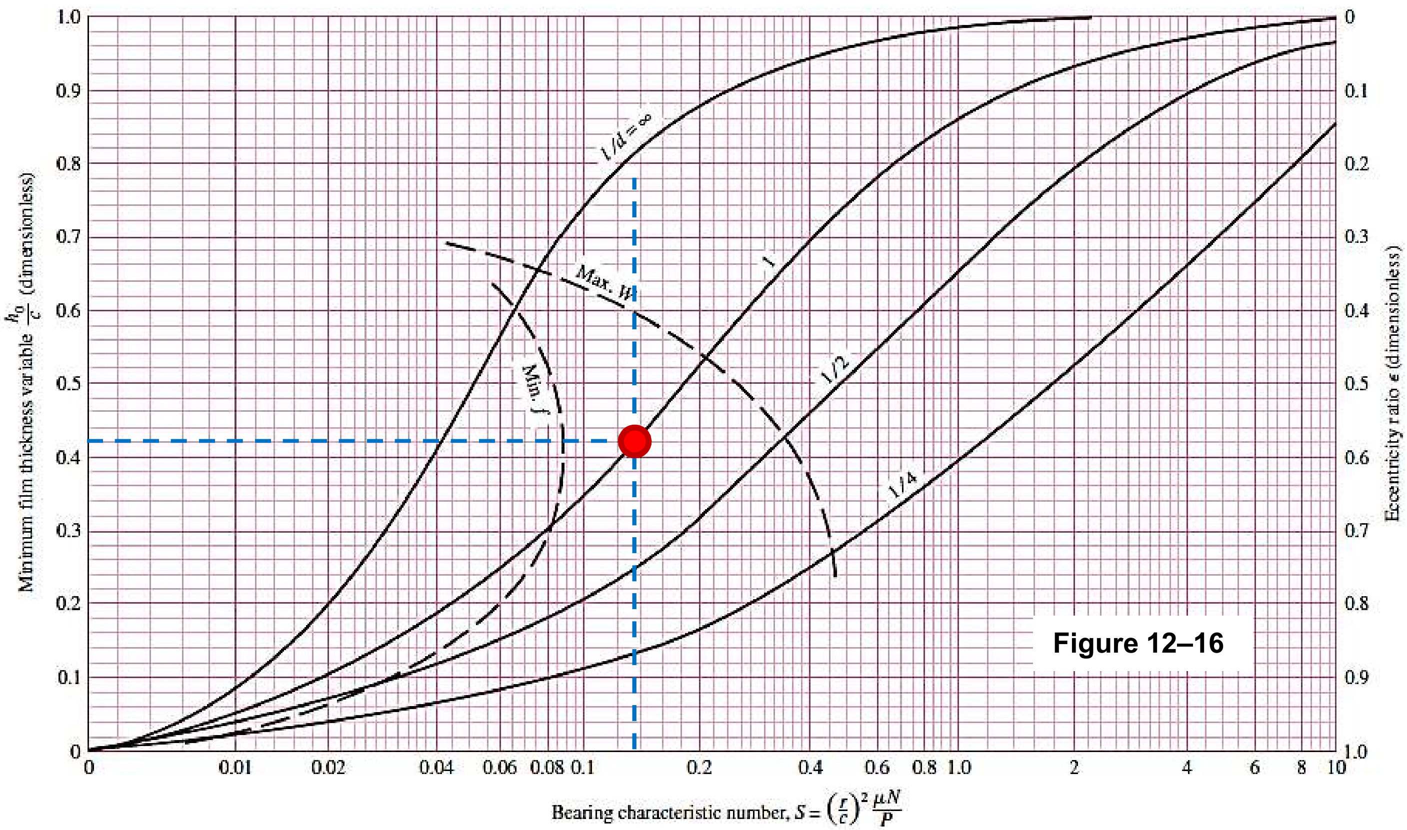


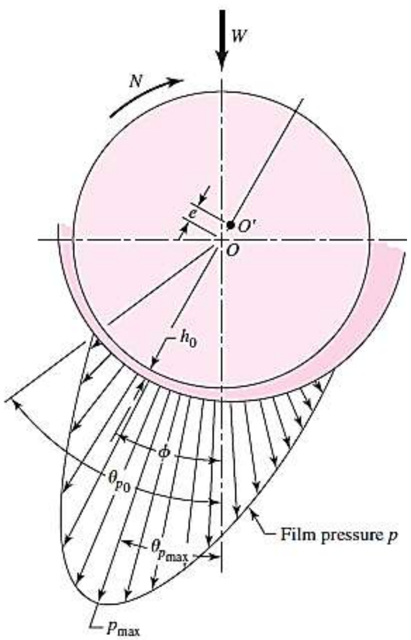
Figure 12–16

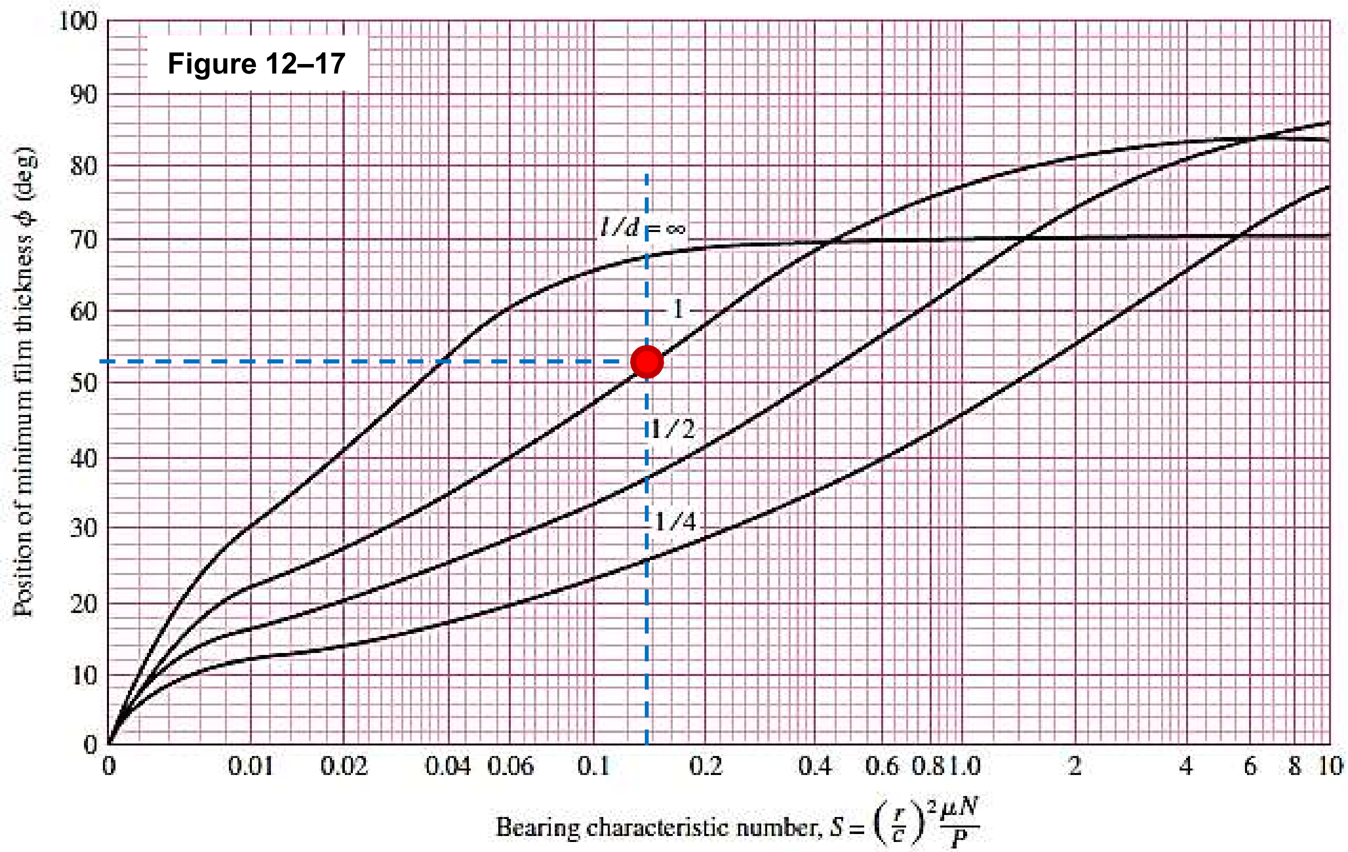
EXAMPLE 12–1 (Cont'd) ► Eccentricity (Fig. 12-17)

Sommerfeld number $S = 0.135$ and $\frac{l}{d} = \frac{1.5}{2 \cdot 0.75} = 1$

Angular location of the minimum film thickness $\phi = 53^o$

Eccentricity ratio $\varepsilon = \frac{e}{c}$, eccentricity $e = \varepsilon \cdot c = 0.58 \cdot 0.0015 = 0.00087\ in$





Coefficient of Friction (Fig. 12-18)

EXAMPLE 12–2 From Example 12–1, determine the coefficient of friction, the torque to overcome friction, and the power loss to friction.

Sommerfeld number $S = 0.135$, $\frac{l}{d} = 1$

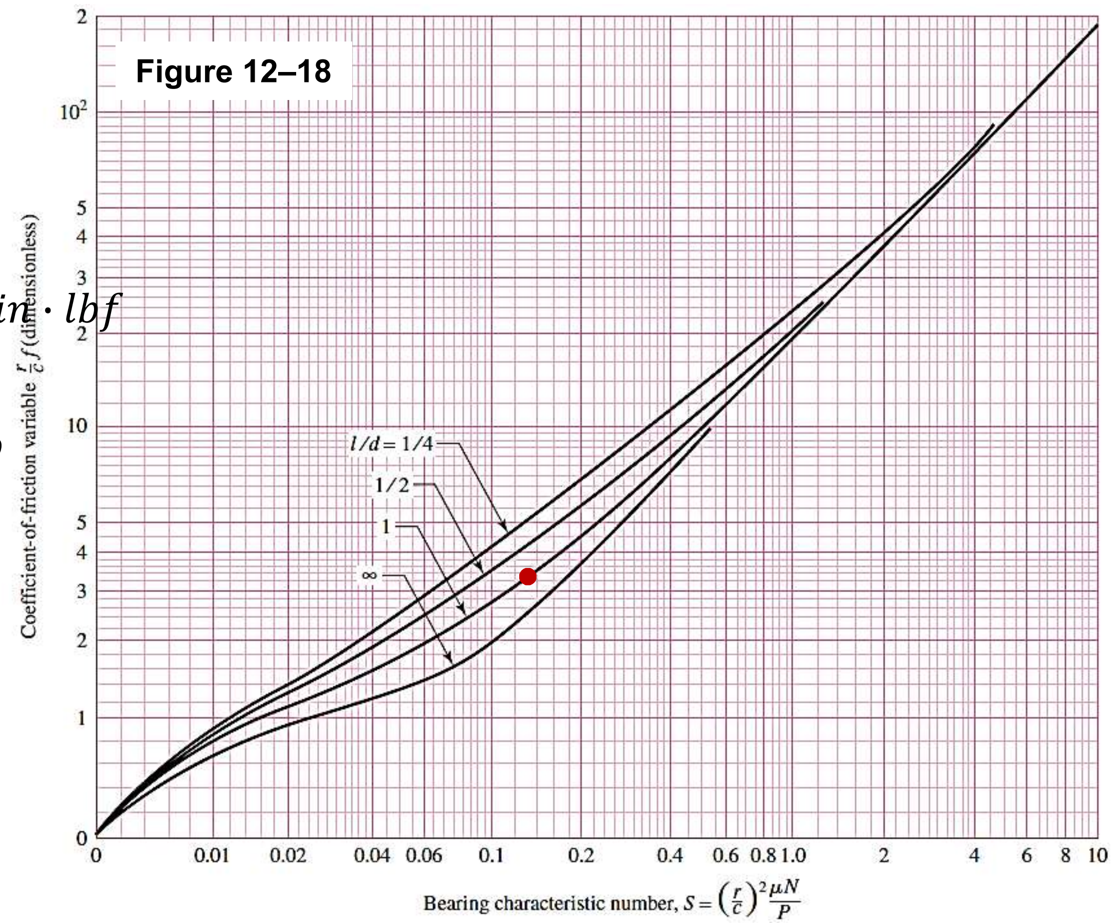
Fig 12-18 gives $\frac{r}{c} f = 3.5$

$$f = 3.5 \cdot \frac{c}{r} = 3.5 \cdot \frac{0.0015}{0.75} = 0.007$$

Journal friction torque:

$$T = fWr = 0.007 \cdot 500 \cdot 0.75 = 2.62\ in \cdot lbf$$

Power loss $= \frac{TN}{1050} = \frac{2.62 \cdot 30}{1050} = 0.075\ hp$



Total Flow Rate (Fig. 12-19), Side Flow Rate (Fig. 12-20)

EXAMPLE 12–3 From Example 12–1, determine the total volumetric flow rate Q and the side flow rate Q_s.

Sommerfeld number $S = 0.135$, $\frac{l}{d} = 1$

Fig 12-19 gives $\frac{Q}{rcNl} = 4.28$, $Q = 4.28 \cdot 0.75 \cdot 0.0015 \cdot 30 \cdot 1.5 = 0.217\, in^3/s$

Fig 12-20 $\frac{Q_s}{Q} = 0.655$, $Q_s = 0.217 \cdot 0.655 = 0.142\, in^3/s$

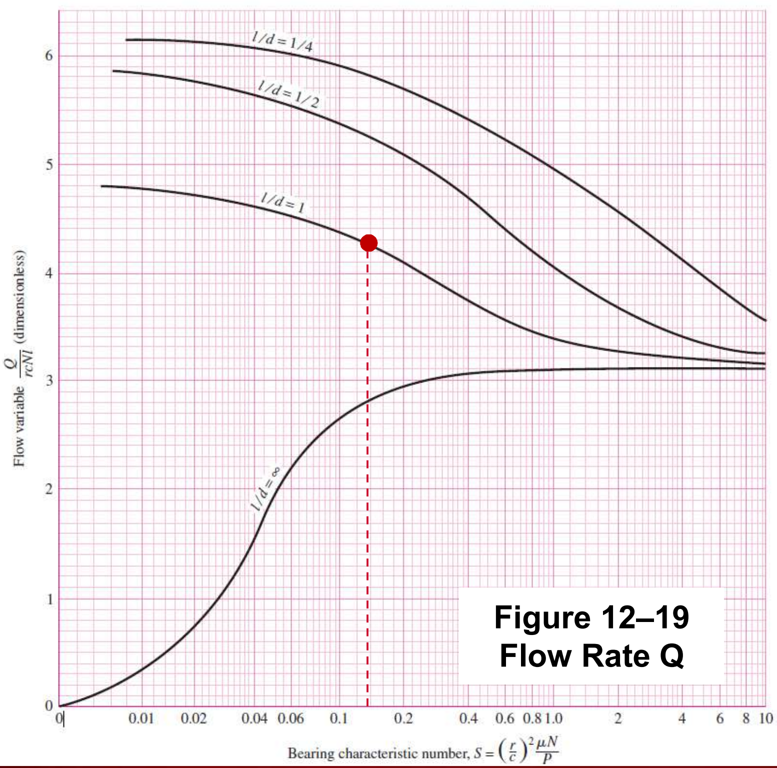


Figure 12–19
Flow Rate Q

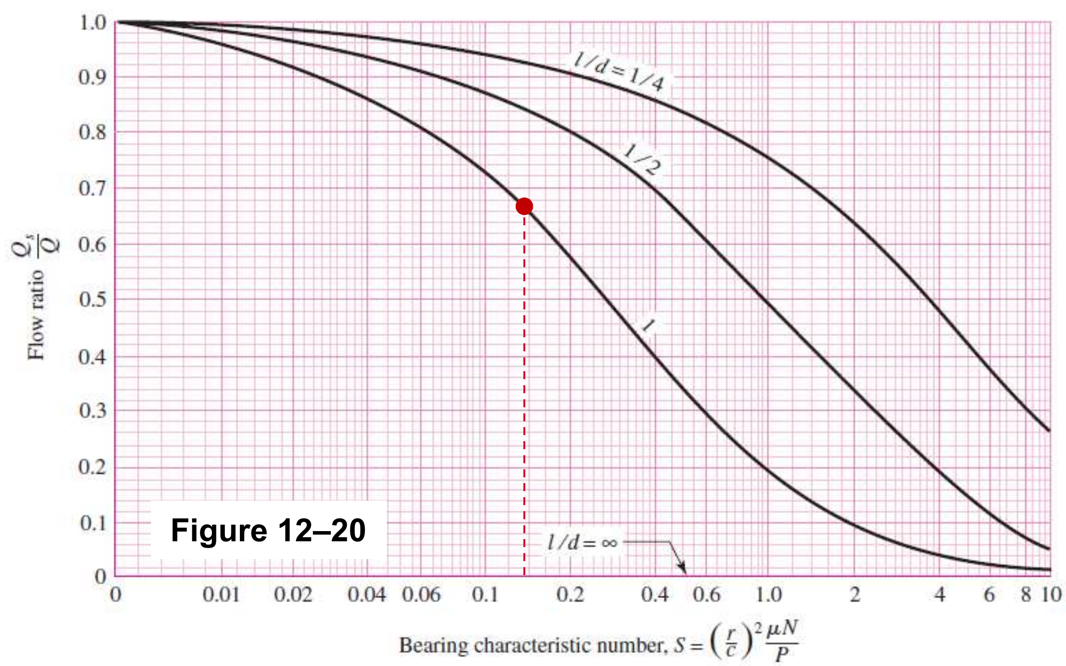


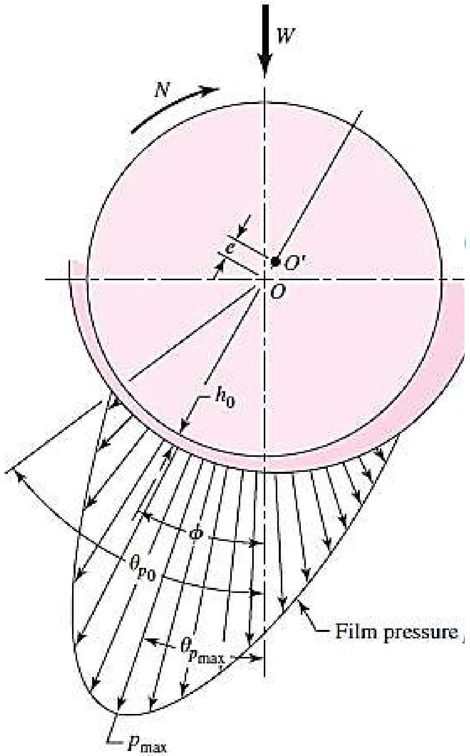
Figure 12–20

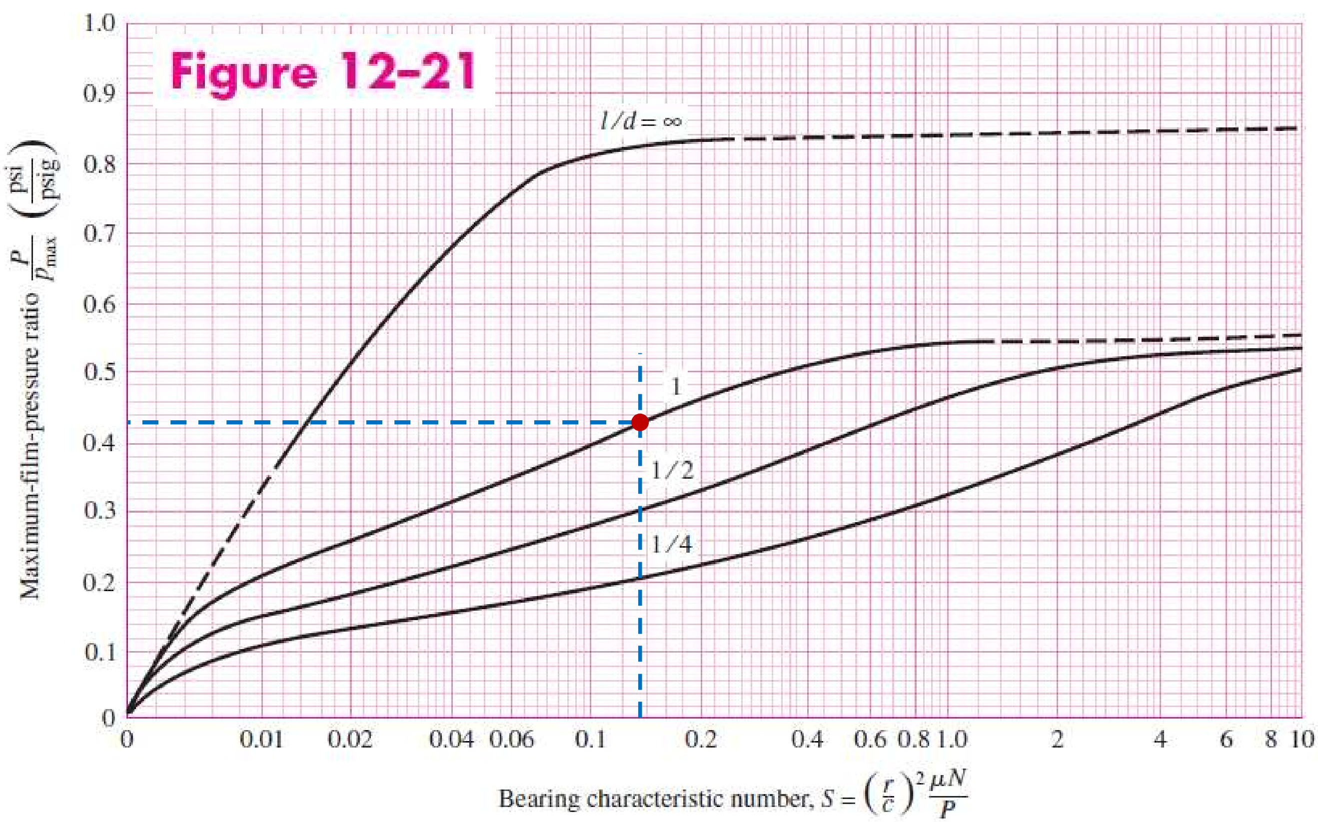
Max Film Pressure (Fig. 12-21)

EXAMPLE 12–4 From Example 12–1, determine the maximum film pressure and the locations of the maximum and terminating pressures.

Sommerfeld number $S = 0.135$, $\frac{l}{d} = 1$

Fig 12-21 gives $\frac{P}{p_{max}} = 0.42$, $p_{max} = \frac{P}{0.42} = \frac{222}{0.42} = 529\,psi$





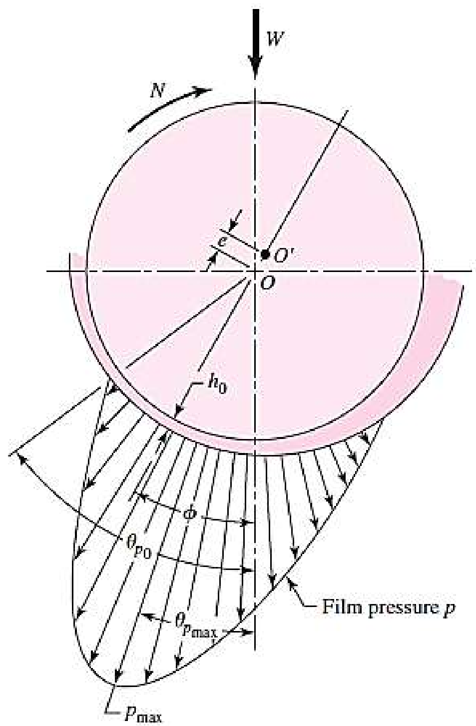
Film Pressure (Fig. 12-21), Peak Prs Location (Fig. 12-22)

EXAMPLE 12–4 From Example 12–1, determine the maximum film pressure and the locations of the maximum and terminating pressures.

Sommerfeld number $S = 0.135$, $\frac{l}{d} = 1$

Fig 12-21 gives $\frac{P}{p_{max}} = 0.42$, $p_{max} = \frac{P}{0.42} = \frac{222}{0.42} = 529\ psi$

Also from Fig. 12–22, $\theta_{pmax} = 18.5^o$ and the terminating position $\theta_{p0} = 75^o$.



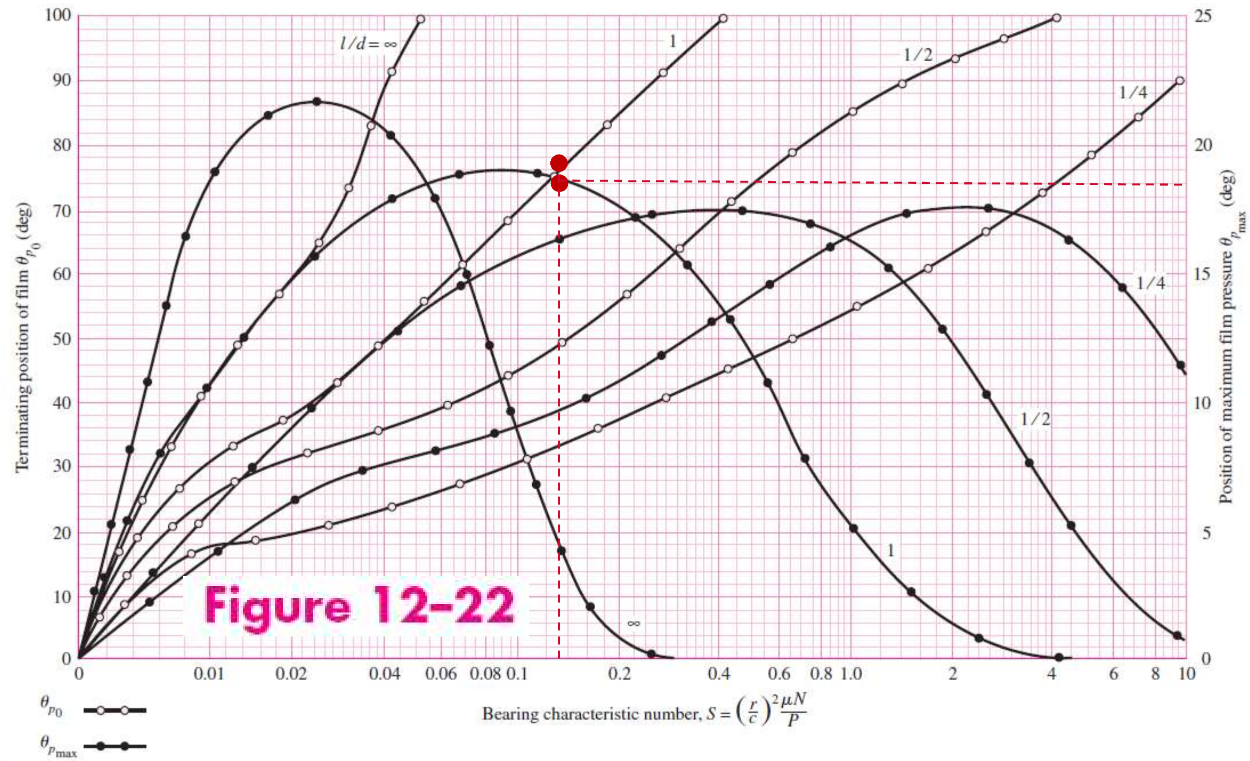


Figure 12–22

TABLE 8-1 Performance Characteristics for a Centrally Loaded 360° Bearing

$\frac{L}{D}$	ε	$\frac{h_n}{C}$	θ_A	$\frac{\alpha}{\beta}$	S	ϕ	$\frac{R}{C}f$	$\frac{Q}{nRCL}$	$\frac{Q_s}{Q}$	$\frac{\rho c}{P}\Delta T$	$\frac{P}{p_{max}}$	$\theta_{p_{max}}$	θ_{p_0}
∞	0	1	0	—	∞	70.92	∞	π	0	∞	—	0	149.38
	0.1	0.9	0	0.308	0.24	69.1	4.8	3.03	0	19.9	0.826	0	137
	0.2	0.8	0	0.314	0.123	67.26	2.57	2.83	0	11.4	0.814	5.6	128
	0.4	0.6	0	0.328	0.0626	61.94	1.52	2.26	0	8.47	0.764	14.4	107
	0.6	0.4	0	0.349	0.0389	54.31	1.2	1.56	0	9.73	0.667	20.8	86
	0.8	0.2	0	0.383	0.021	42.22	0.961	0.76	0	15.9	0.495	21.5	58.8
	0.9	0.1	0	0.412	0.0115	31.62	0.756	0.411	0	23.1	0.358	19	44
	0.97	0.03	0	—	—	—	—	—	0	—	—	—	—
	1	0	0	—	0	0	0	0	0	□□	0	0	0
1	0	1	0	—	∞	85	∞	π	0	∞	—	0	119
	0.1	0.9	0	0.279	1.33	79.5	26.4	3.37	0.15	106	0.54	3.5	113
	0.2	0.8	0	0.294	0.631	74.02	12.8	3.59	0.28	52.1	0.529	9.2	106
	0.4	0.6	0	0.325	0.264	63.1	5.79	3.99	0.497	24.3	0.484	16.5	91.2
	0.6	0.4	0	0.36	0.121	50.58	3.22	4.33	0.68	14.2	0.415	18.7	72.9
	0.8	0.2	0	0.399	0.0446	36.24	1.7	4.62	0.842	8	0.313	18.2	52.3
	0.9	0.1	0	0.426	0.0188	26.45	1.05	4.74	0.919	5.16	0.247	13.8	37.3
	0.97	0.03	0	0.457	0.00474	15.47	0.514	4.82	0.973	2.61	0.152	7.1	20.5
	1	0	0	—	0	0	0	—	1	0	0	0	0
0.5000	0	1	0	—	∞	88.5	∞	π	0	∞	—	0	107
	0.1	0.9	0	0.273	4.31	81.62	85.6	3.43	0.173	343	0.523	5.8	99.2
	0.2	0.8	0	0.292	2.03	74.97	40.9	3.72	0.318	164	0.506	11.9	92.5
	0.4	0.6	0	0.329	0.779	61.45	17	4.29	0.552	68.6	0.441	16.9	78.8
	0.6	0.4	0	0.366	0.319	48.14	8.1	4.85	0.73	33	0.365	17.1	64.3
	0.8	0.2	0	0.408	0.0923	33.31	3.26	5.41	0.874	13.4	0.267	15.3	44.2
	0.9	0.1	0	0.434	0.0313	23.66	1.6	5.69	0.939	6.66	0.206	11	33.8
	0.97	0.03	0	0.462	0.00609	13.75	0.61	5.88	0.98	2.56	0.126	3.8	19.1
	1	0	0	—	0	0	0	—	1	0	0	0	0
0.2500	0	1	0	—	∞	89.5	∞	π	0	∞	—	0	99
	0.1	0.9	0	0.271	16.2	82.31	322	3.45	0.18	1287	0.515	7.4	98.9
	0.2	0.8	0	0.291	7.57	75.18	153	3.76	0.33	611	0.489	13.5	85
	0.4	0.6	0	0.331	2.83	60.86	61.2	4.37	0.567	245	0.415	17.4	70
	0.6	0.4	0	0.37	1.07	46.72	26.7	4.99	0.746	107	0.334	16.4	55.5
	0.8	0.2	0	0.414	0.261	31.04	8.8	5.6	0.884	35.4	0.24	11.5	39.7
	0.9	0.1	0	0.439	0.0736	21.85	3.5	5.91	0.945	14.1	0.18	8.6	27.8
	0.97	0.03	0	0.466	0.0101	12.22	0.922	6.12	0.984	3.73	0.108	4	17.7
	1	0	0	—	0	0	0	—	1	0	0	0	0

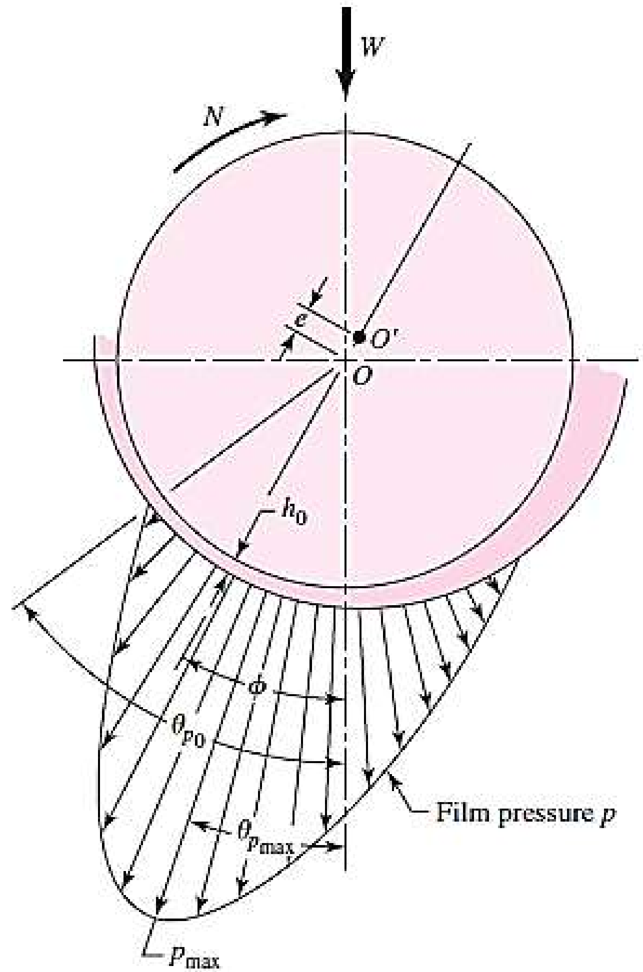
Harnoy, A., 2003, Bearing Design in Machinery, 1st ed., Marcel Dekker, Inc., USA

Review of Bearing Performance

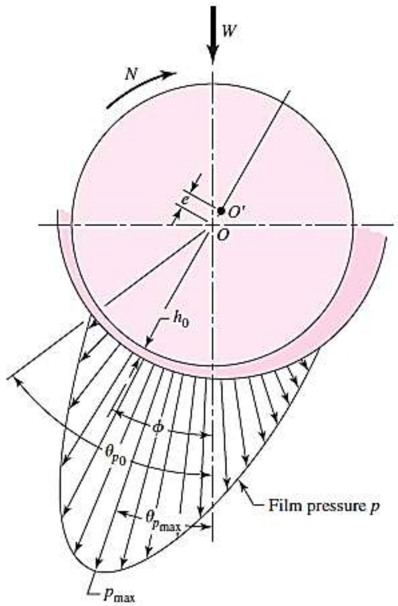
Film Thickness and Pressure Formation

Lubricant adhering to the journal surface is being pulled by the moving surface into a narrowing, wedge-shaped space and forcing the journal over to the other side. This <u>action</u> creates a <u>film pressure</u> of sufficient intensity to support the bearing load.

Q: What is the name of this action?



Raimondi-Boyd Bearing Design Charts (Figs. 12–12 to 12–14)



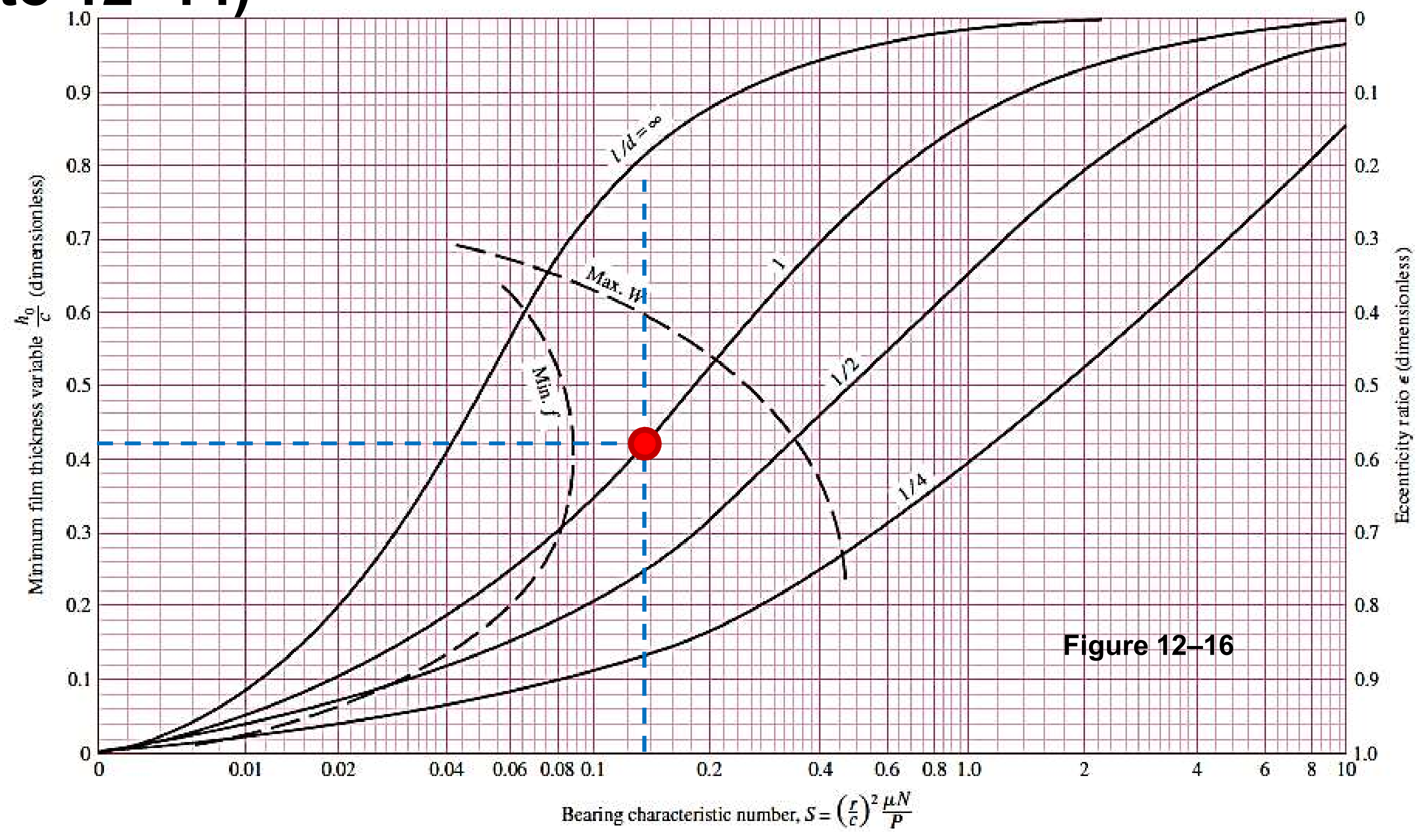


Figure 12–16

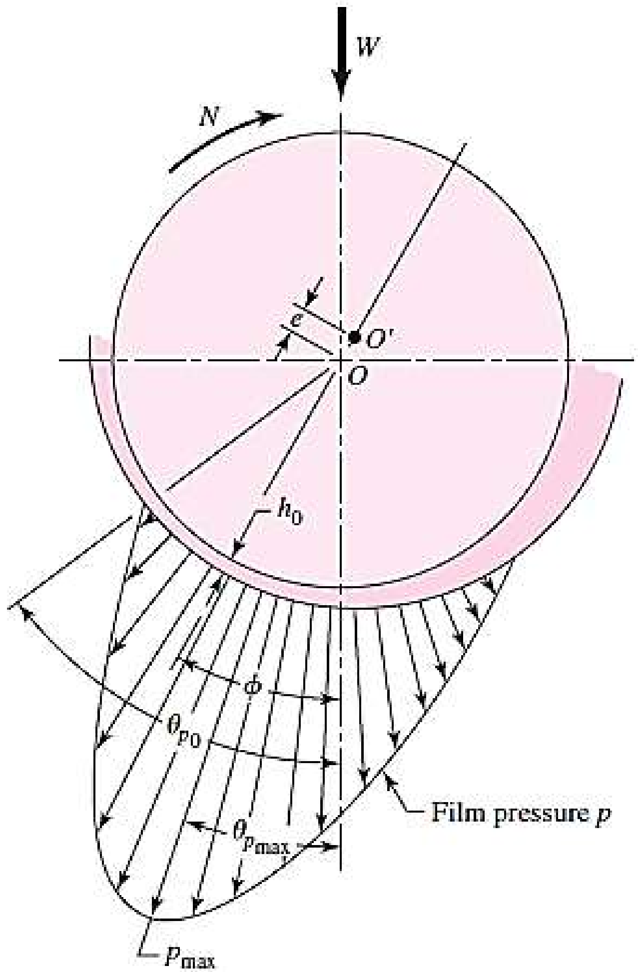
- Raimondi-Boyd analysis assumed <u>lubricant viscosity is constant</u>
- Viscosity depends on lubricant temperature rise.
- Procedures for estimation of bearing average film temperature rise (T_{av}) needs to be addressed.

Discussions

For bearing protection, increase or decrease min film thickness (MFT) is the preferred action?

A: Increase MFT

B. Decrease MFT



Discussions

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

How to increase MFT?

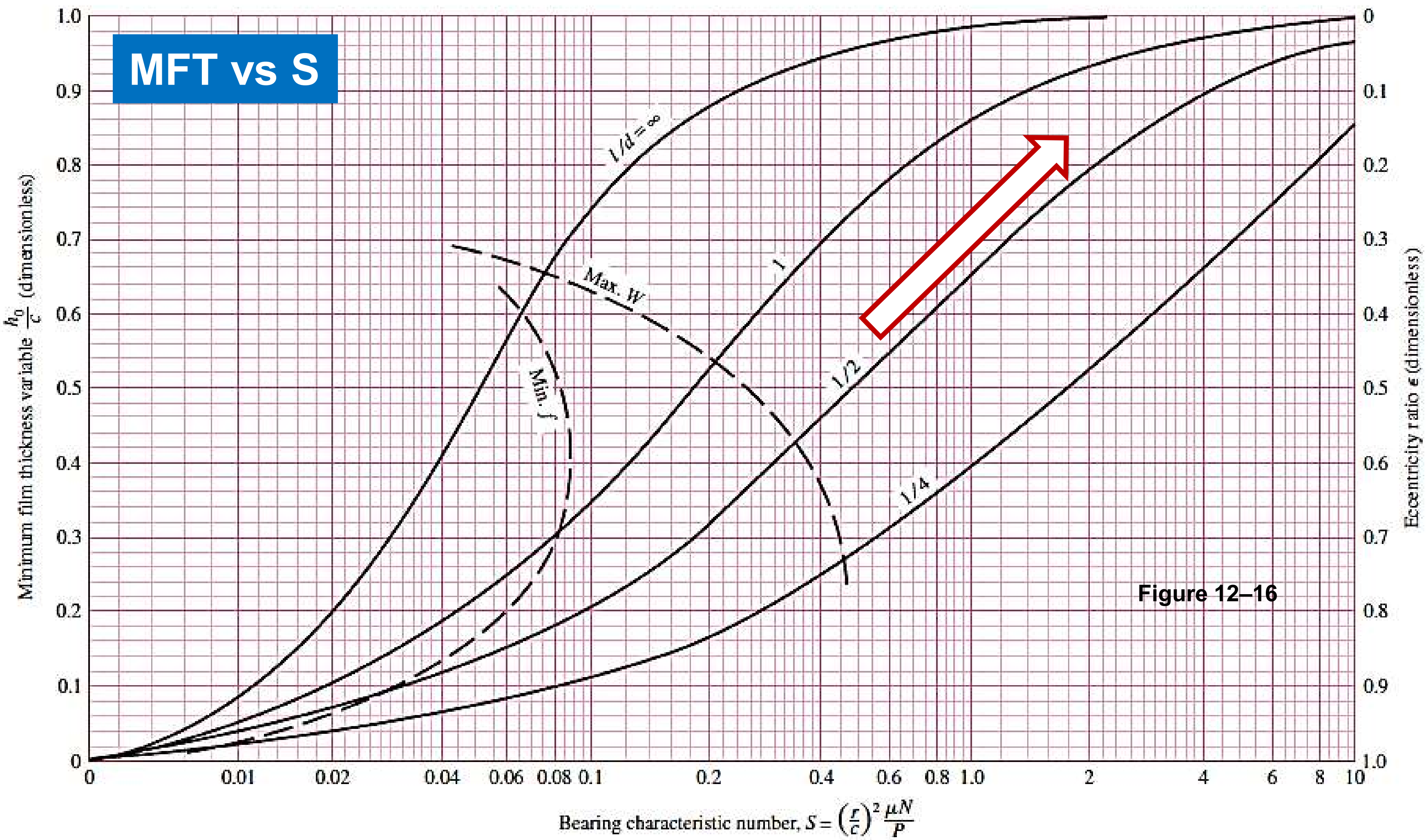


Figure 12–16

Discussions

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

How to decrease COF?

COF vs S

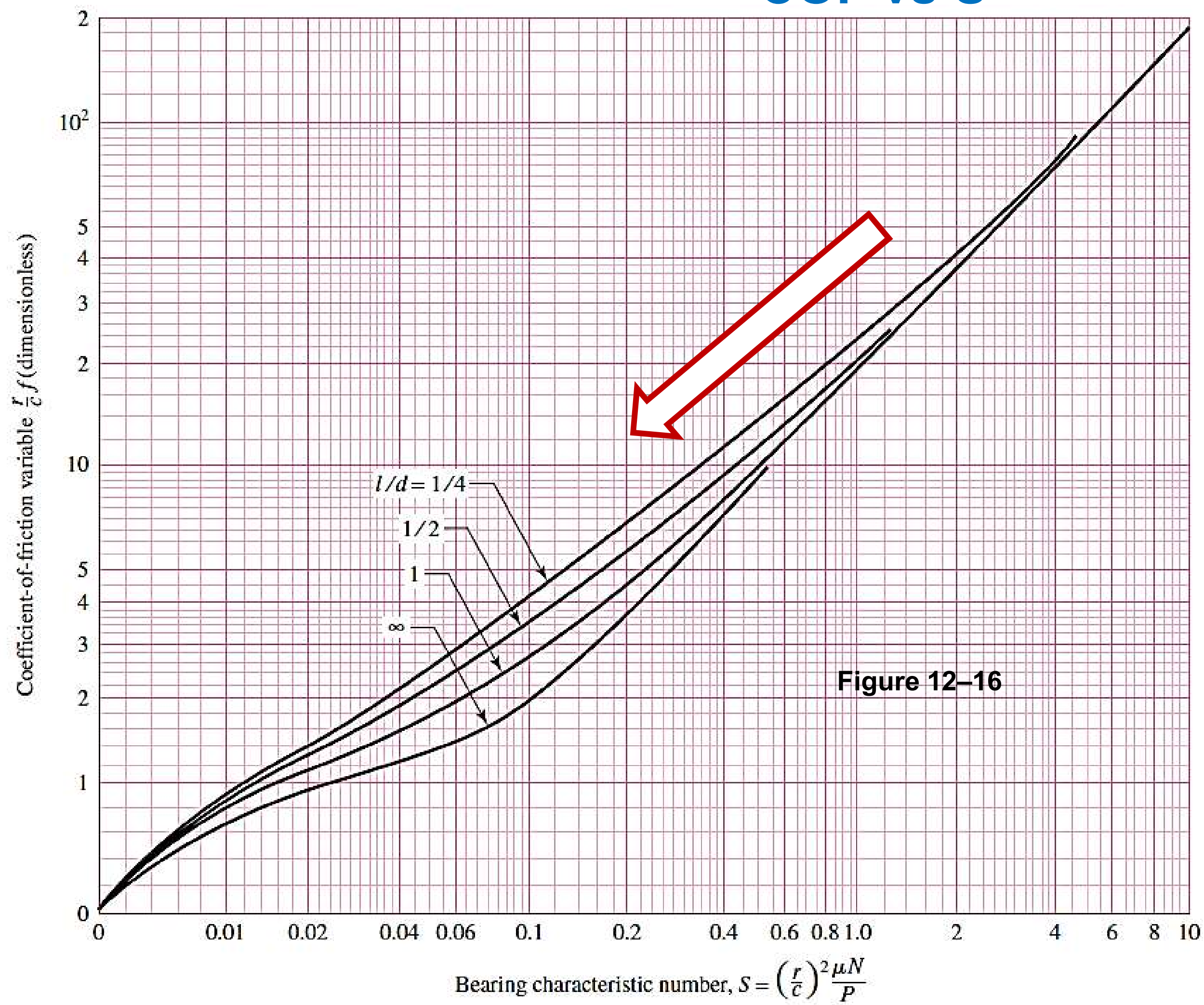
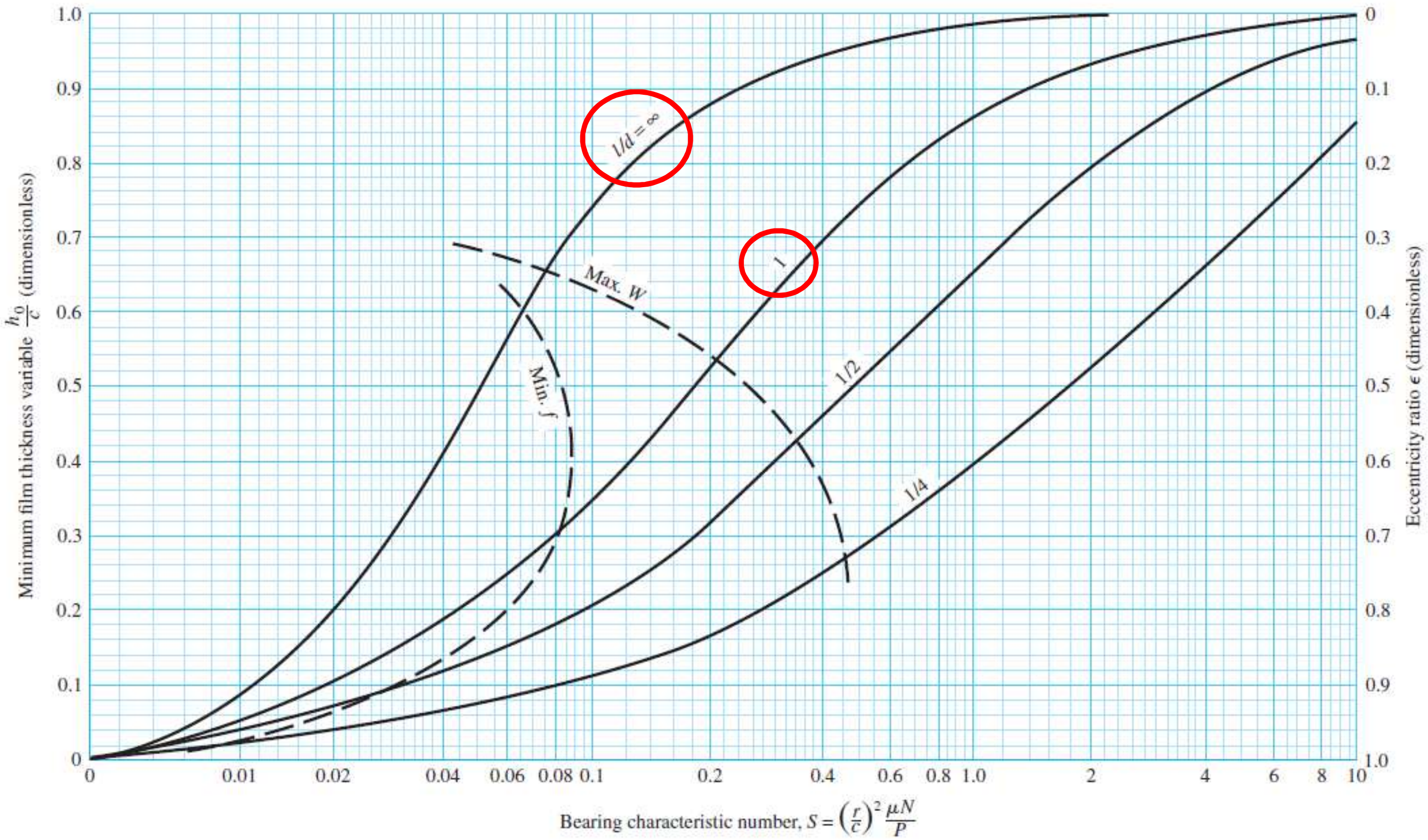


Figure 12–16

Curves on Design Charts Requires L/D Ratio



Questions:

- L/D= ∞, What does it mean?
- Where is L/D=2 on chart?

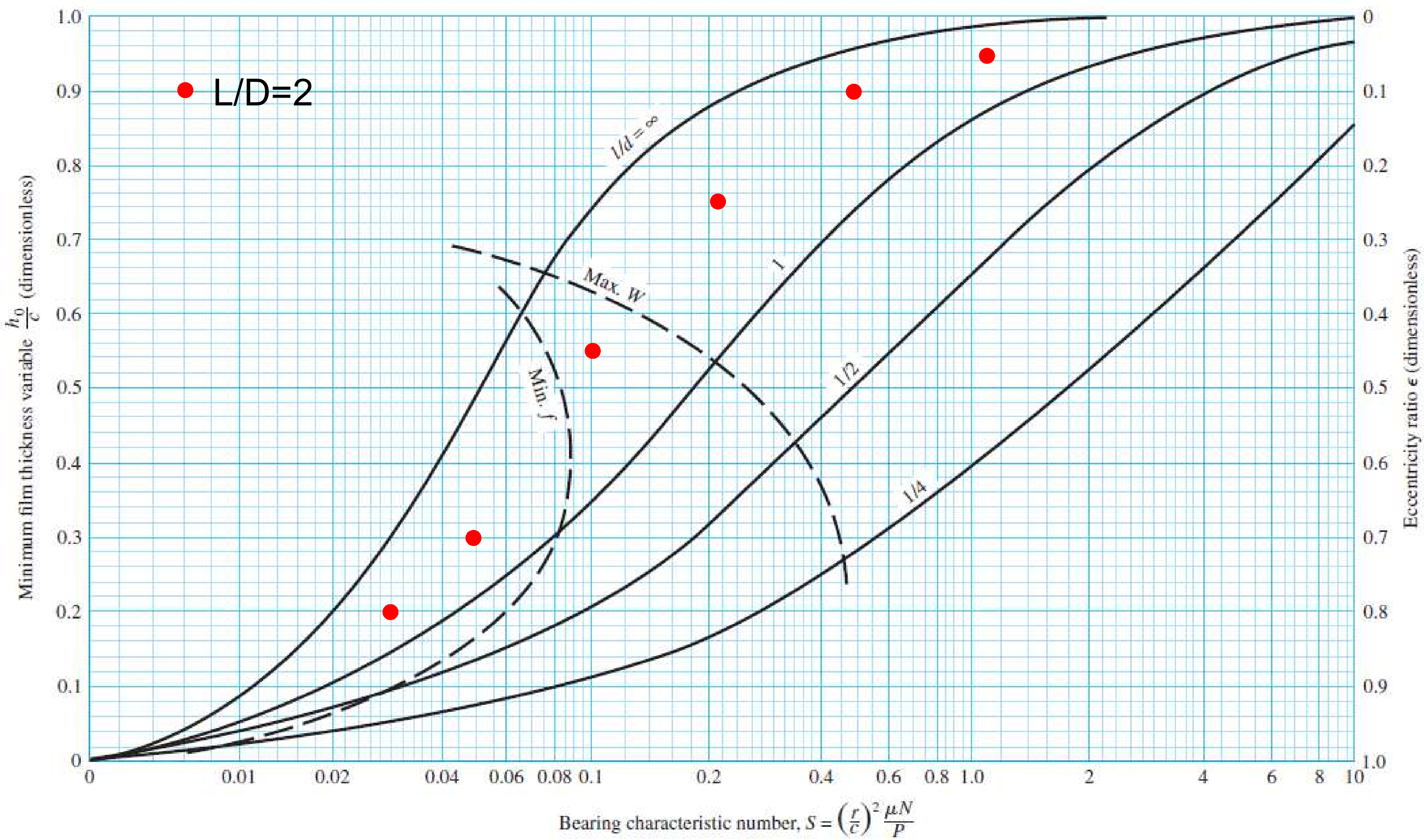


Figure 12–16

Naffin, R.K. and Chang, L., April 2010, “An Analytical Model for the Basic Design Calculations of Journal Bearings”, ASME J. of Tribology, Vol. 132,

Interpolate Bearing Performance Parameter by Equation 12-16

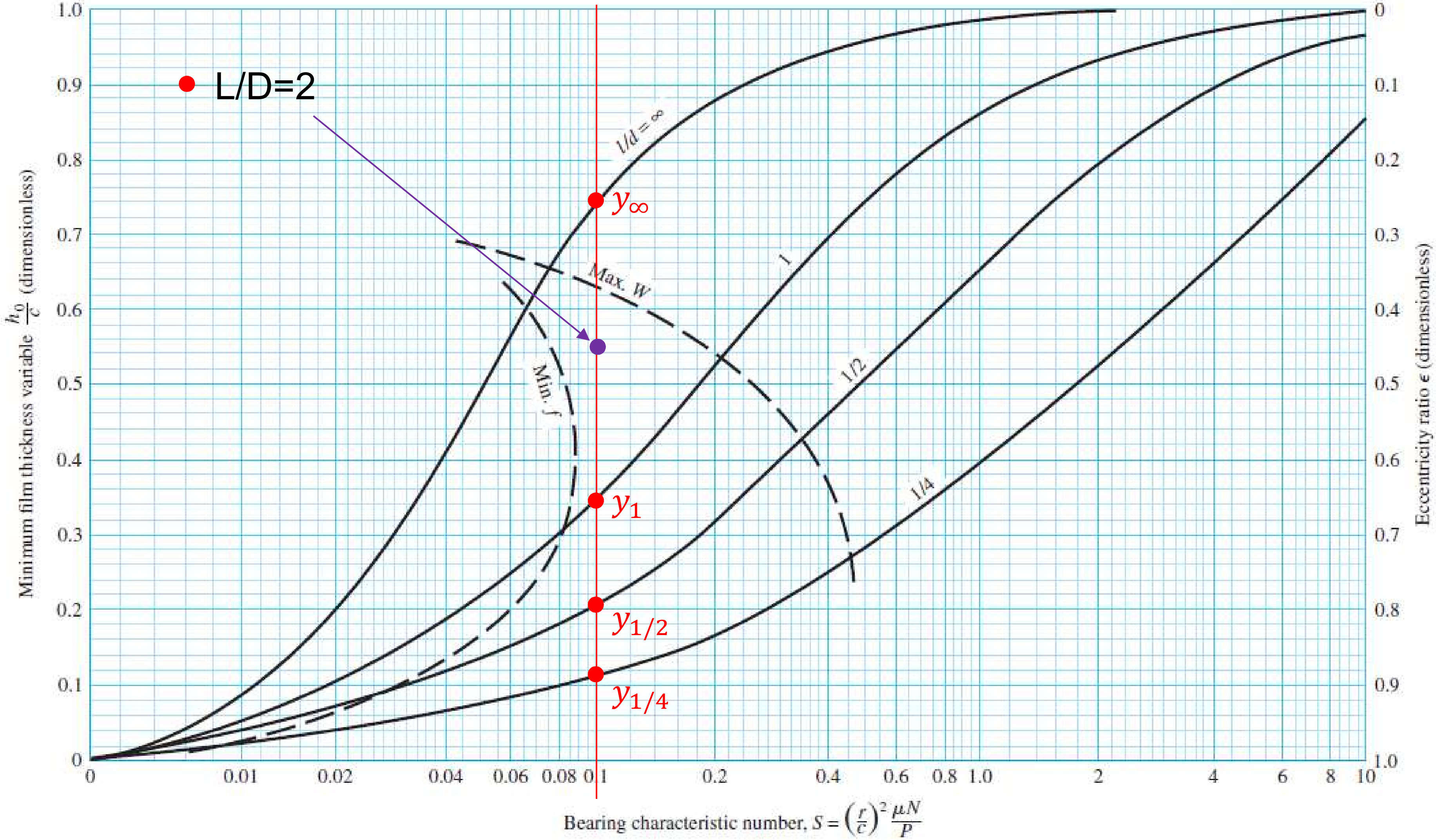


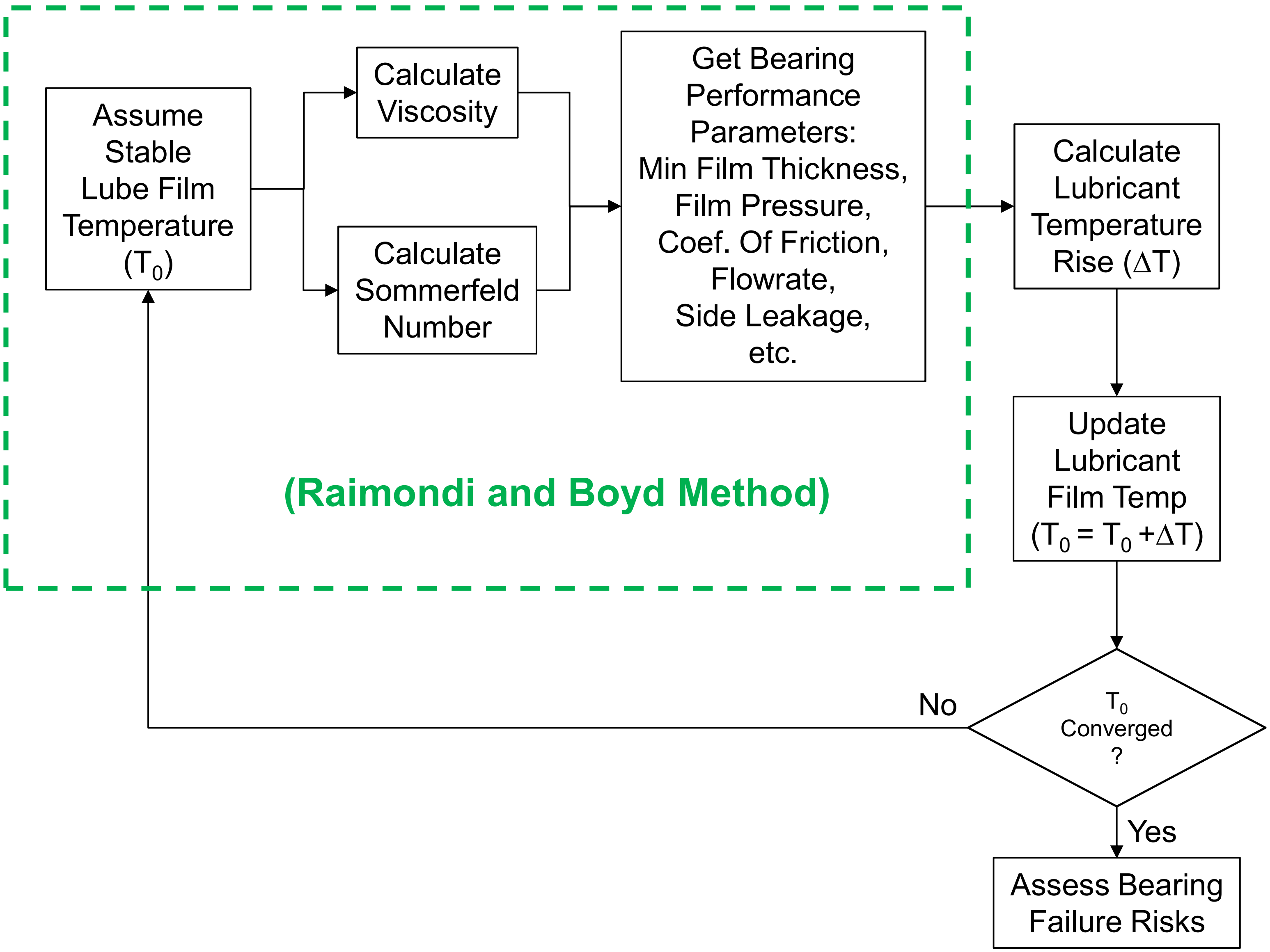
Figure 12–16

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

Bearing Thermal Analysis

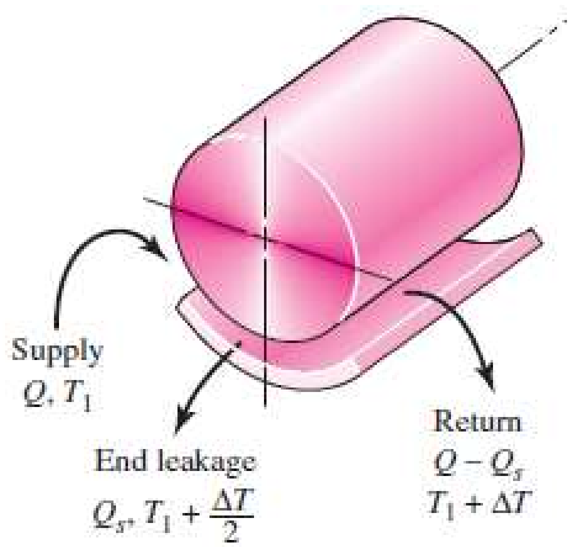
- 12-8 The Relations of the Variables
  - Lubricant Temperature Rise (Bearing Lubrication With Sump)
- 12-9 Steady-State Conditions in Self-Contained Bearings (Bearing Lubrication Without Sump)

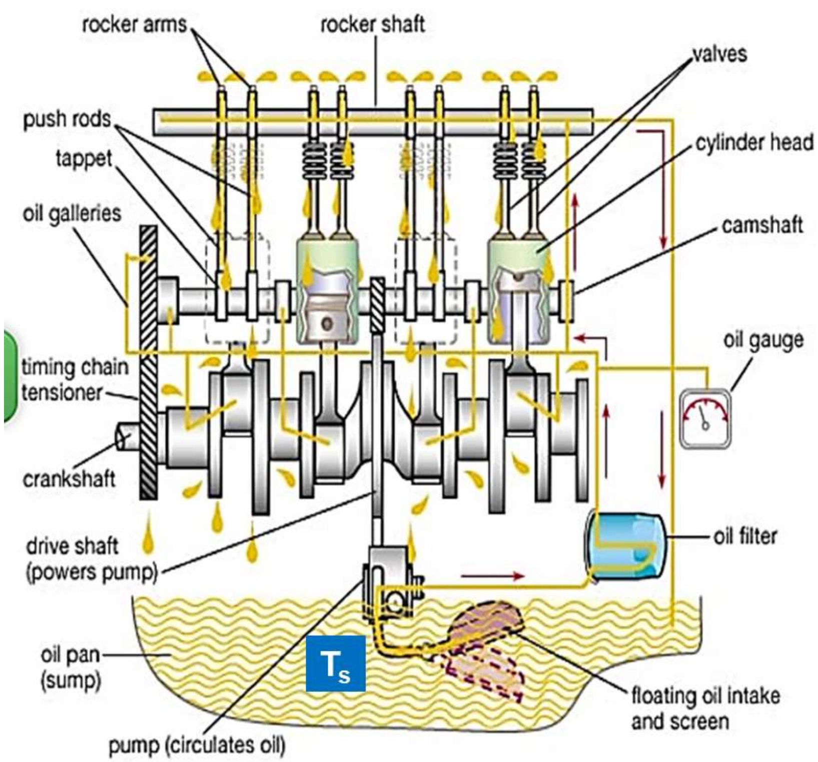
Bearing Design Workflow



Lubricant Temperature Rise (With Sump)

- Raimondi-Boyd analysis (Figs. 12–12 to 12–14) assumed that lubricant viscosity is constant as it passes through the bearing.
- Lubricant temperature rise is resulting from fluid shear between journal and lubricant film. It is a frictional heating phenomenon.





Average Temperature for Viscosity Calculation

- Assume inlet temperature T_1 = oil sump temperature T_s
- In determining the viscosity to be used, employ a temperature that is the average of the inlet and outlet temperatures (T_{av}).

$$T_{av} = T_1 + \frac{\Delta T}{2}$$

  – T_1: inlet temperature (= T_s, Oil Sump Temperature)
  – ΔT: temperature rise of the lubricant from inlet to outlet

- Viscosity can be found using Figs. 12–12 to 12–14, or $\mu = \mu_o exp\left(\frac{b}{T+95}\right)$

Figure 12–23

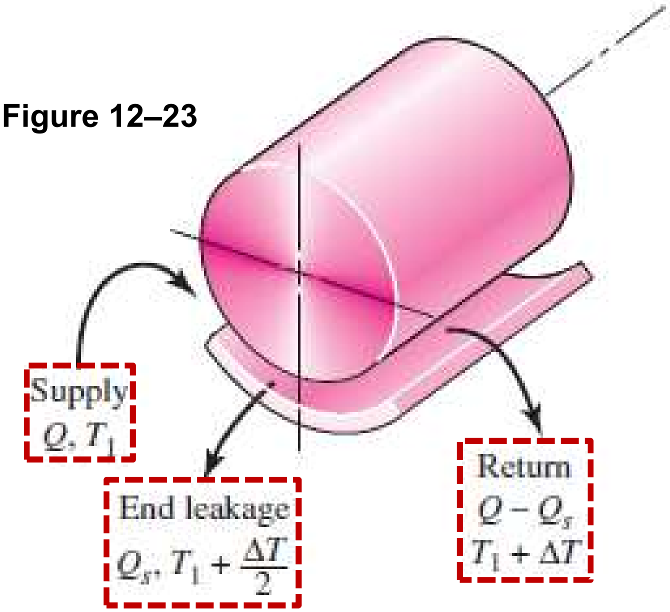


Table 12–1

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.

Balance of Heat Generation and Heat Removal

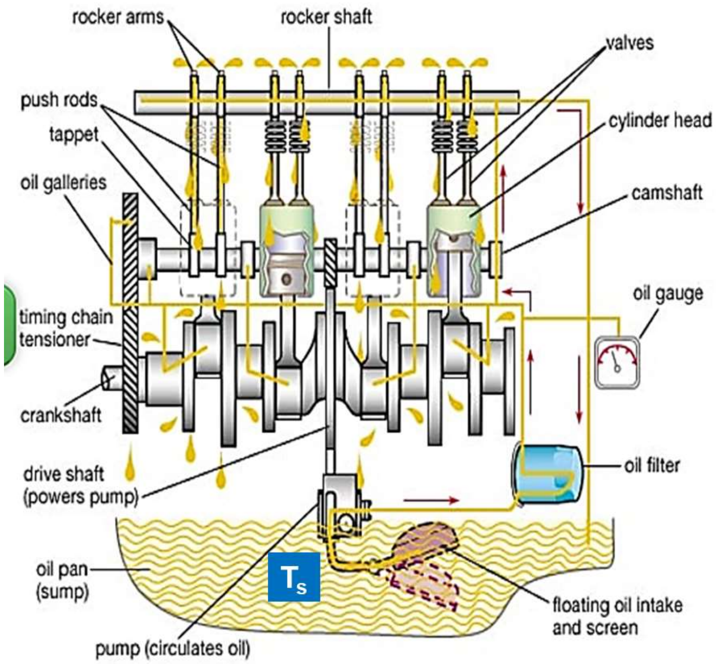
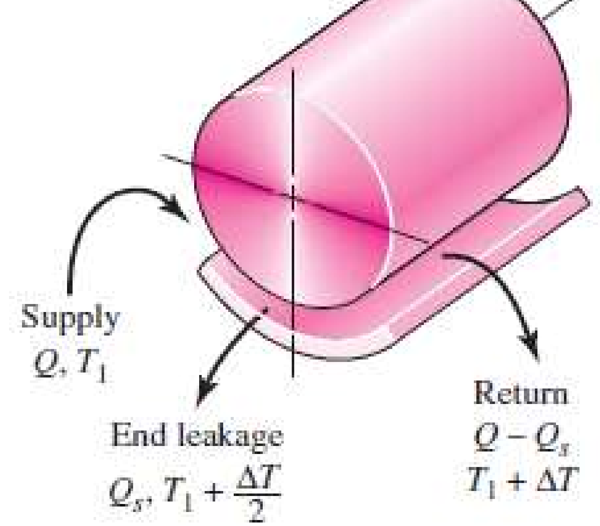
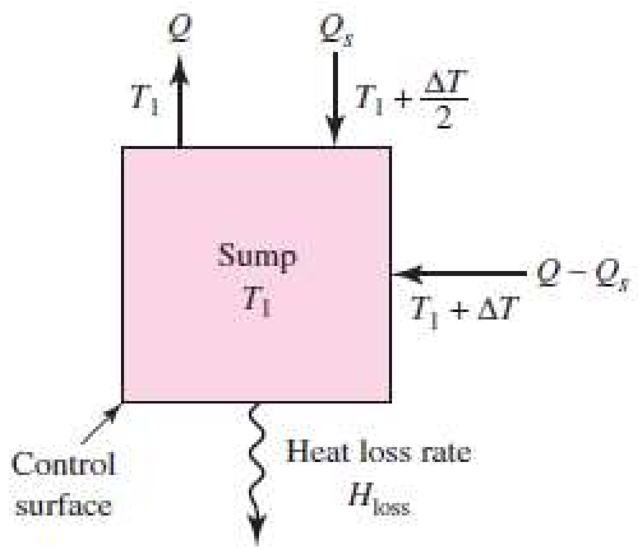


Figure 12–23





Temperature rise in oil sump continues until a balance is reached between heat generation rate by fluid shear and heat removal rate from sump to the greater surroundings.

Thermal Analysis of Lubricant Temperature Rise (Bearing lubrication With Sump)

Use sump as the control volume and let

$$H_{loss} = \rho C_p Q_s \frac{\Delta T}{2} + \rho C_p (Q - Q_s)\Delta T = \rho C_p Q \Delta T \left(1 - \frac{1}{2}\frac{Q_s}{Q}\right)$$

Q = oil flowrate into the bearing, in^3/s

Q_s = side leakage flowrate rate out of bearing and to sump, in^3/s

$Q - Q_s$ = discharge flowrate from annulus to sump, in^3/s

T_1 = oil inlet temperature (=T_s), ◦F

ΔT = temperature rise in oil between inlet and outlet, ◦F

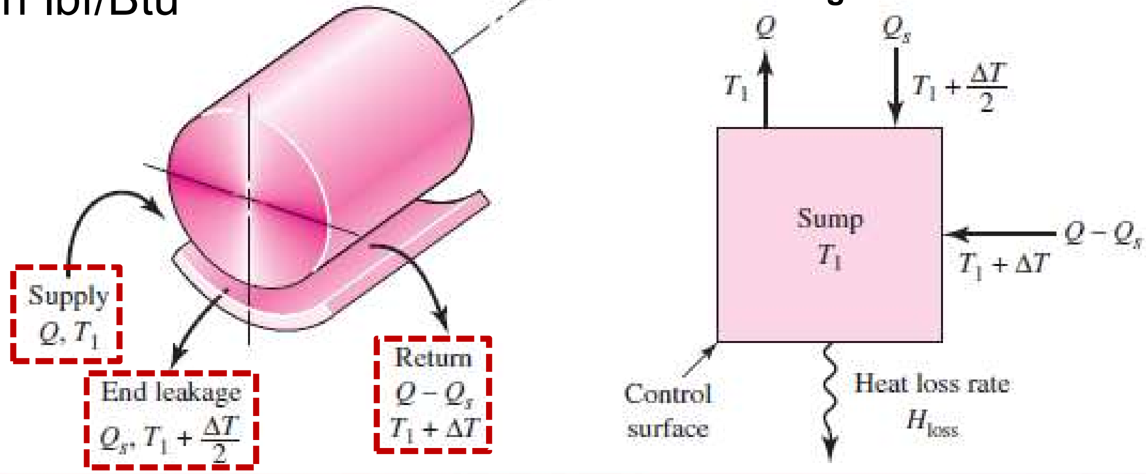
ρ = lubricant density, lbm/in^3

C_p = specific heat capacity of lubricant, Btu/(lbm · °F)

J = Joulean heat equivalent, in·lbf/Btu

H = heat rate, Btu/s

Figure 12–23



Lubricant Temperature Rise (Cont'd)

Heat removed from sump:

$$H_{loss} = \rho C_p Q_s \frac{\Delta T}{2} + \rho C_p (Q - Q_s)\Delta T = \rho C_p Q \Delta T \left(1 - \frac{1}{2}\frac{Q_s}{Q}\right)$$

Bearing frictional heating:

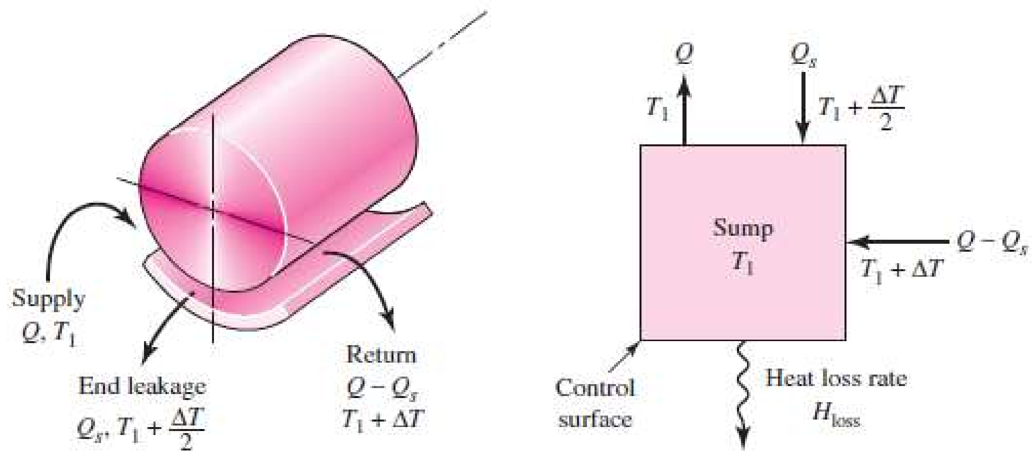
$$H_{bfh} = \frac{2\pi T N}{J} = \frac{2\pi N \cdot (fWr)}{J} = \frac{2\pi N \cdot (f \cdot 2Prl \cdot r)}{J} = \frac{4\pi PrlNc}{J}\frac{rf}{c}$$

H_{loss} should equal to generation rate of bearing frictional heating (H_{bfh}).

$$H_{loss} = \rho C_p Q \Delta T \left(1 - \frac{1}{2}\frac{Q_s}{Q}\right) = H_{bfh} = \frac{4\pi PrlNc}{J}\frac{rf}{c}$$

$$\boxed{\frac{J\rho C_p \Delta T}{4\pi P} = \frac{rf}{c}\frac{1}{\left(1 - \frac{1}{2}\frac{Q_s}{Q}\right)\left(\frac{Q}{rcNl}\right)}}$$

Figure 12–23



Lubricant Temperature Rise (Cont'd)

Given Sommerfeld Number S:

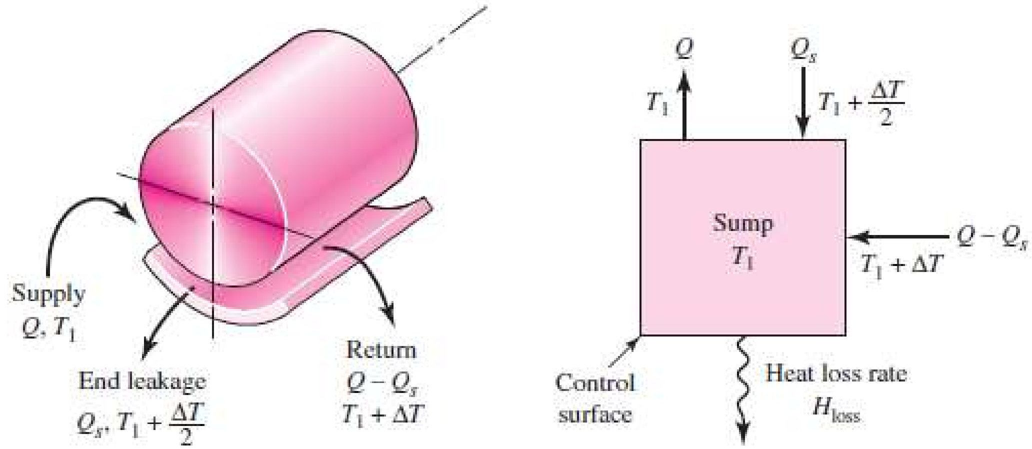
$$\frac{J\rho C_p \Delta T}{4\pi P} = \boxed{\frac{rf}{c}}\,\frac{1}{\left(1-\frac{1}{2}\boxed{\frac{Q_s}{Q}}\right)\boxed{\left(\frac{Q}{rcNl}\right)}}$$

Fig. 12-18

Fig. 12-20

Fig. 12-19

Figure 12–23



Constant J is used to normalize the LHS of the equation, which

$1\ Btu = 9338\ in \cdot lbf$

Lubricant Temperature Rise (Cont'd)

For common lubricants:

ρ = 0.0311 lbm/in^3, C_p = 0.42 Btu/(lbm-$^\circ$F), and J=9336 in·lbf/Btu

$$\frac{J\rho C_p \Delta T}{4\pi P} = \frac{9336 \cdot 0.0311 \cdot 0.42 \cdot \Delta T_F}{4\pi P_{psi}} = 9.7\frac{\Delta T_F}{P_{psi}}$$

Thus $\boxed{9.7\frac{\Delta T_F}{P_{psi}} = \frac{rf}{c}\frac{1}{\left(1-\frac{1}{2}\frac{Q_s}{Q}\right)\left(\frac{Q}{rcNl}\right)}}$

ΔT_F : temperature rise in $\circ$F

P_{psi}: bearing pressure in psi

or

$$\boxed{0.120\frac{\Delta T_C}{P_{MPa}} = \frac{rf}{c}\frac{1}{\left(1-\frac{1}{2}\frac{Q_s}{Q}\right)\left(\frac{Q}{rcNl}\right)}}$$

ΔT_C : temperature rise in $^\circ$C

P_{MPa}: bearing pressure in MPa

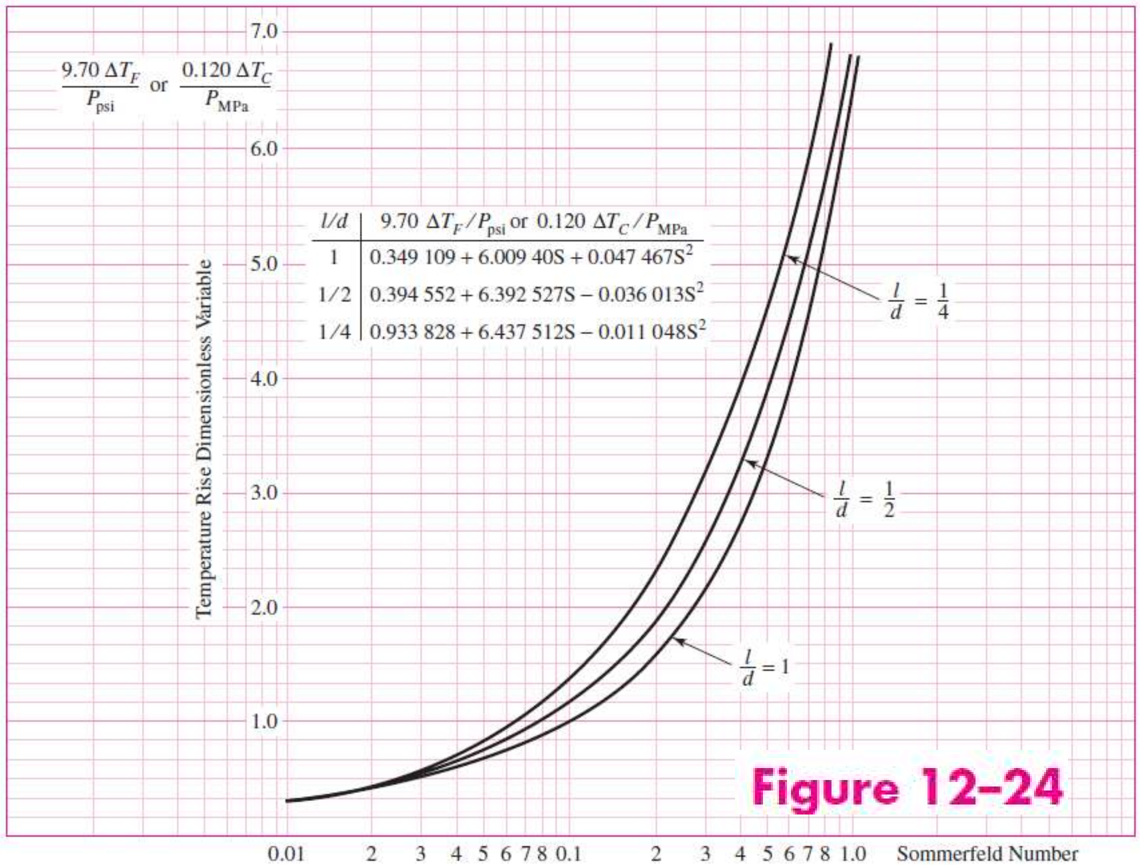


Figure 12–24

Guidelines for Journal Bearing Design

Potential Failure Modes

Abrasive Wear

- Foreign particles from operating environment, oxidized wear particles or grit may cause scraping or scratching on the mating element surfaces.

Adhesive Wear

- Occurs when a sliding bearing carries a heavy load or the supplied lubricant is insufficient, the lubricant film may be so thin that metal-to-metal contact occurs at the bearing interface. In serious cases, adhesive wear results in scoring, scuffing, galling or even seizure.

Fatigue

- When a bearing is subject to a cyclic load or a cyclic small amplitude sliding, cracks may generate and fatigue or fretting wear may occur.

Corrosion

- Acid formation during oxidation of a lubricant may induce unacceptable corrosion on bearing surfaces

Bearing Design Parameters

- N: Speed
- W: Bearing Load

Operating Parameters

- μ: Viscosity
- R: bearing radius
- r: journal radius
- L: Bearing Length
- c: <u>radial</u> clearance $(c = R - r)$
- β: bearing inclusion angle

(β =360°: full bearing; β <360° :partial bearing)

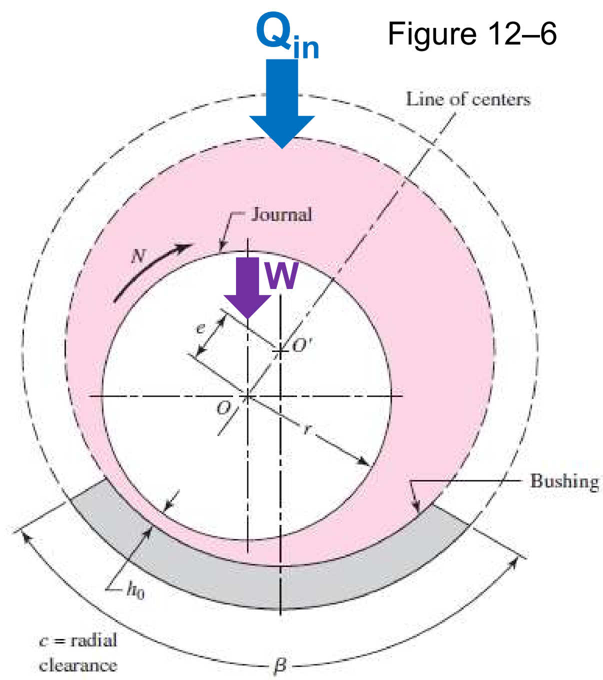
Design Parameters

- h_o: minimum film thickness
- h: film thickness @any angular position
- e: eccentricity $(e = OO')$
- ε: eccentricity ratio $\left(\epsilon = \frac{e}{c}\right)$
- f: coefficient of friction
- T: temperature rise
- Q: volume flow rate of oil

Responding (Performance) Variables

Note that: $c = h_o + e$, $h_o = c - e$, $\frac{h_o}{c} = 1 - \epsilon$

Figure 12–6



Journal rotates in CW direction

Guidelines for Journal Bearing Design

Clearance Ratio

$$\mathrm{CR} = \left(\frac{2c}{d}\right)$$

- For journals 25 to 150 mm in diameter, the clearance ratio is usually of the order of 0.001, particularly for precision bearings.

- For less precise bearings, this ratio tends to be higher
  - up to about 0.002 for general machinery bearings, and
  - 0.004 for rough-service machinery.

Guidelines for Journal Bearing Design

Bearing Length/Diameter Ratio (L/D)

- Often shaft diameter is determined by strength and deflection requirements, and then bearing length is determined in order to provide adequate bearing capacity

- L/D ratio high
  - Bearing load carrying capacity increases
  - friction coefficient and oil flow rate (bearing leakage) decreases
- L/D ratio low
  - greater flow of oil out of the ends, thus keeping the bearing cooler and thus more favorable for bearings operate with high annulus velocity

- L/D ratios of 0.25 to 0.75 are most commonly used now
- Older machinery having ratios averaging closer to unity.

Guidelines for Journal Bearing Design

- Unit (or Average) Pressure Loading

$$P_{avg} = \frac{W}{L \cdot d}$$

- Suggested Safety Factor $n_d \geq 2$

  (steady running load)

- Bearing Start-up/Shut-Down Load (W_{st})

$$\frac{W_{st}}{l \cdot d} \leq 300\, psi\ (\sim 2MPa)$$

Table 12-5

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

Design Criteria for Journal Bearings

Pass/Fail Criteria Suggested by Trumpler, assuming SF=2 on load.

- Minimum Film Thickness h_o (d: journal diameter)

$$h_o \geq 0.0002 + 0.00004 \cdot d \; (Unit{:}\ in)$$

$$h_o \geq 0.005 + 0.00004 \cdot d \; (Unt{:} mm)$$

- Maximum Film Temperature

$$T_{max} \leq 250^o F \; (\sim 121^o C)$$

12-11 Pressure-Fed Bearings

Reasons for Pressure-Fed Bearings

- Heat-dissipation capability of the bearing is one primary limiting factor for better bearing performance.
- Heating is generated within bearings. Two ways can be implemented to improve heat-dissipation capability:
  - <u>In-situ Cooling</u>: Spray cooled-down lubricant at location or near vicinity of bearing, usually difficult due to space limitation.
  - <u>Pressure-Fed Cooling</u>: Use an external pump to force more lubricant into and flush through heated bearings.
- Example of Pressure-Fed Bearing
  - use a circumferential groove at the center of the bearing, with an oil-supply hole located opposite the load-bearing zone

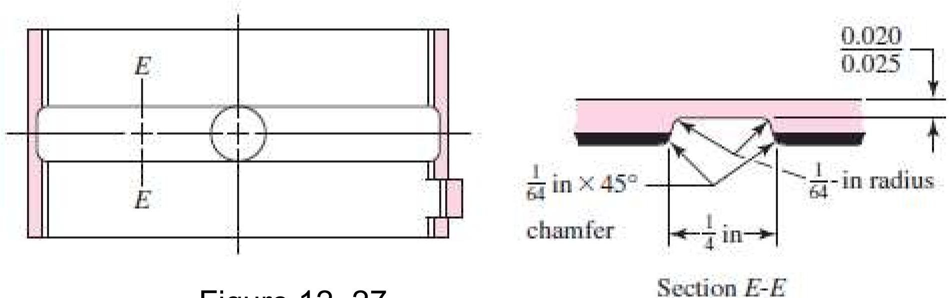


Figure 12–27

Groove Patterns for Pressure-Fed Bearings

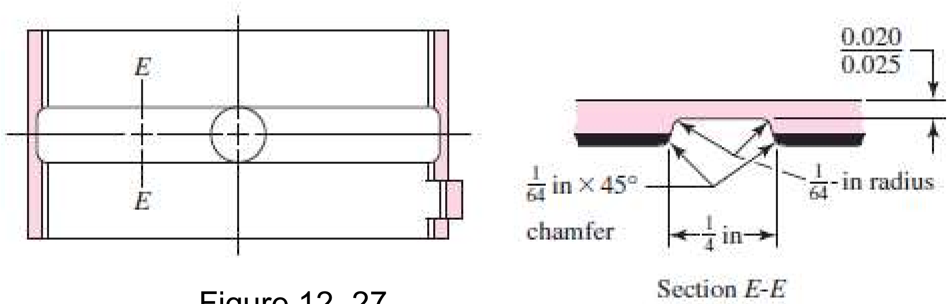
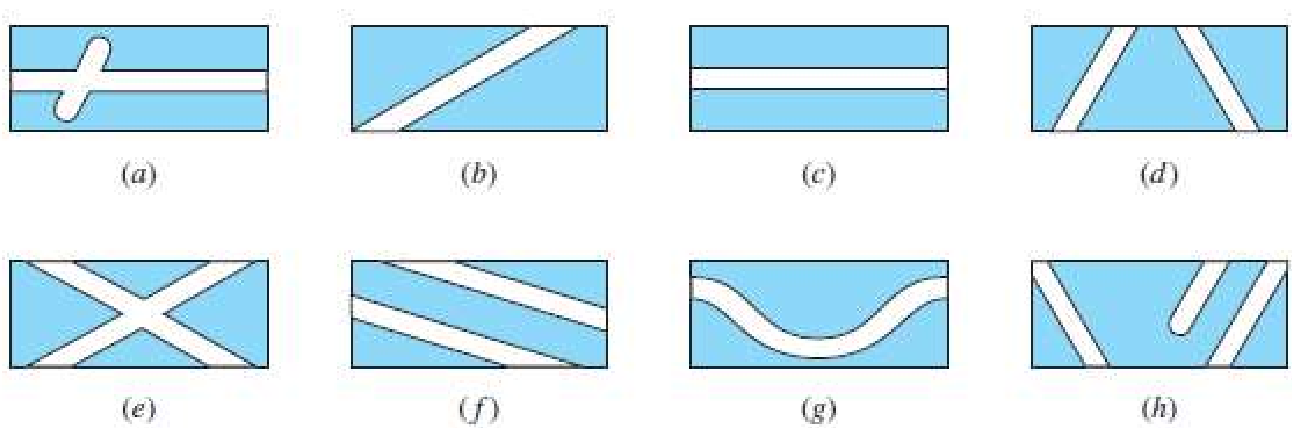


Figure 12–27

Figure 12–34



Side Flowrate of Pressure-Fed Bearings

Laminar flow is assumed, with the pressure varying linearly from p = p_s at x = 0, to p = 0 at x = l'

$$-2y(p+dp)+2yp+2\tau dx=0$$

Which gets $\tau = y\frac{dp}{dx} = \mu\frac{du}{dy}$

Integrate to get velocity profile u with BC's $u=0 \;\; @y=\pm\frac{h}{2}$

, and linear variation of pressure $p(x) = p_s - \frac{p_s}{l'}x$

Velocity $u = \frac{p_s}{8\mu l'}(h^2-4y^2)$, which is a profile of parabola.

Max velocity $u_{max} = \frac{p_s \cdot h^2}{8\mu l'}$

Replace c with film thickness $h = c - e\cos\theta$

Average velocity $u_{av} = \frac{2}{3}\frac{p_s \cdot h^2}{8\mu l'} = \frac{p_s}{12\mu l'}(c-e\cos\theta)^2$

At any position θ, the flow out of the bearing ends is

$$dQ_s = 2u_{av}dA = 2u_{av}(rh\,d\theta) = \frac{p_s}{6\mu l'}(c-e\cos\theta)^3\,d\theta$$

$$\boxed{\text{Total side flow } Q_s = \int dQ_s = \frac{\pi\, p_s\, r\, c^3}{3\mu l'}(1+1.5\epsilon^2)}$$

*(Flowrate from charts using Figs. 12–19 and 12–20 do not apply to pressure-fed bearings)

Figure 12–28

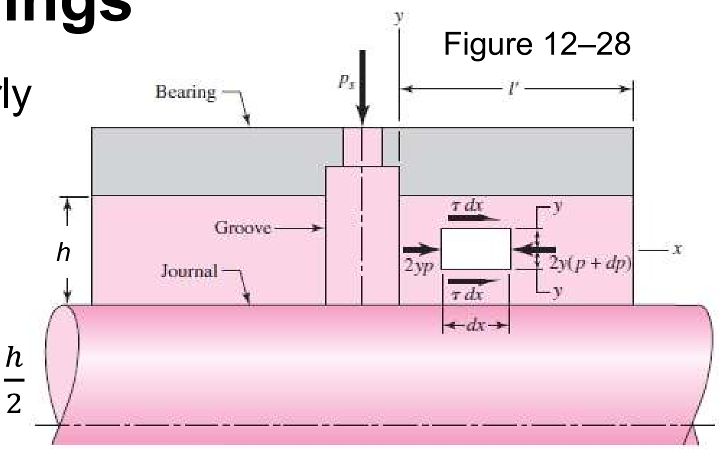


Figure 12–29
Velocity Profile

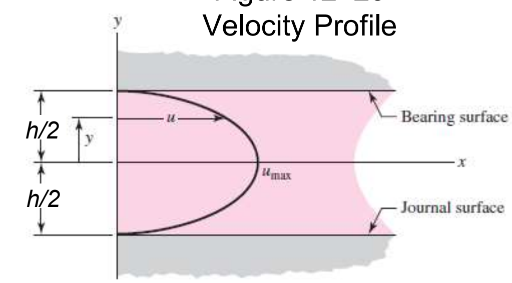
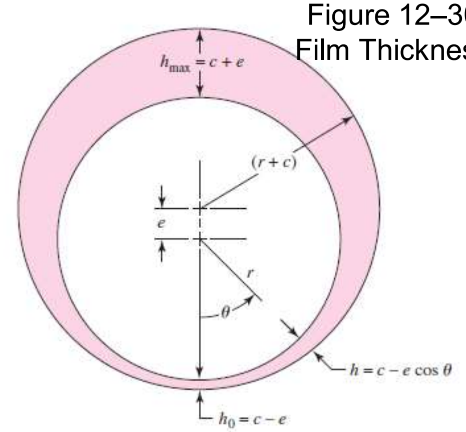


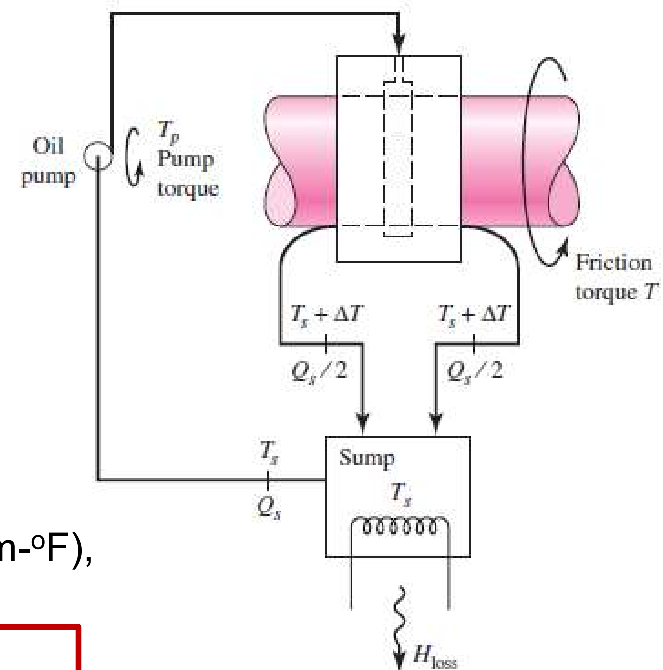
Figure 12–30
Film Thickness



Temperature Rise of Pressure-Fed Bearings

Temperature rise of a pressure-fed centrally located full annular-groove journal bearing with external, coiled lubricant sump can be calculated as followed:

Figure 12–30



Average bearing pressure $P = \frac{W/2}{2rl'} = \frac{W}{4rl'}$

Heat gain passing thru the bearing $H_{gain} = \rho C_p Q_s \Delta T$

Bearing frictional heating $H_f = \frac{2\pi}{J} = \frac{2\pi N \cdot (fWr)}{J} = \frac{2\pi WNc}{J}\frac{rf}{c}$

Since $H_{gain} = H_f$ and $Q_s = \frac{\pi\, p_s\, r\, c^3}{3\mu l'}(1 + 1.5\epsilon^2)$

$$\Delta T = \frac{2\pi WN}{J\rho C_p Q_s}\frac{rf}{c} = \frac{2\pi}{J\rho C_p}WNc\frac{rf}{c}\frac{3\mu l'}{\pi\, p_s\, r\, c^3(1+1.5\epsilon^2)}$$

Sommerfeld number $S = \left(\frac{r}{c}\right)^2\left(\frac{\mu N}{P}\right) = \left(\frac{r}{c}\right)^2\frac{4\, r\, l'\mu N}{W}$

For common lubricants: ρ = 0.0311 lbm/in^3, C_p = 0.42 Btu/(lbm-^{o}F), and J=9336 in·lbf/Btu

$$\boxed{\Delta T_F = \frac{3SW^2}{2J\rho C_p p_s r^4}\left(\frac{fr}{c}\right)\frac{1}{(1+1.5\epsilon^2)} = 0.0123\left(\frac{fr}{c}\right)\frac{SW^2}{(1+1.5\epsilon^2)p_s r^4}}$$

For SI unit where W in kN, p_s in Kpa, and journal radius r in mm:

$$\boxed{\Delta T_C = 978\cdot 10^6\left(\frac{fr}{c}\right)\frac{SW^2}{(1+1.5\epsilon^2)p_s r^4}}$$

EXAMPLE 12–6

Circumferential-groove pressure-fed bearing

SAE grade 20 oil supplied at a gauge pressure of 30 psi

Journal diameter d_j=1.750 in, with a unilateral tolerance of −0.002 in.

Bushing diameter d_b=1.753 in, with a unilateral tolerance of +0.004 in.

$\frac{l'}{d}$ ratio of the two “half-bearings” =1/2

Journal angular speed=3000 rev/min

Radial steady load = 900 lbf.

External sump temperature =120^{o}F as long as the necessary heat transfer < 800 Btu/h.

Determine

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.

EXAMPLE 12-6 (Cont'd)

Min bearing radial clearance:

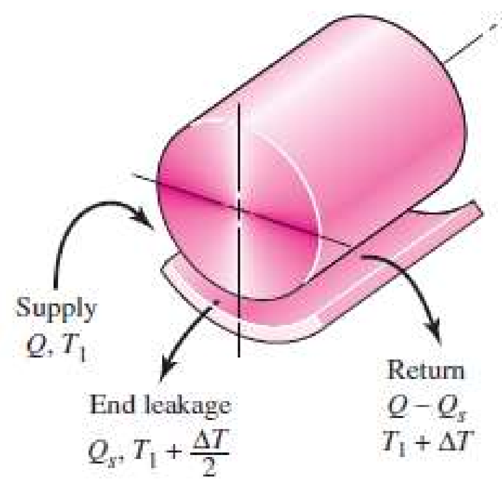
$$c_{min} = \frac{(d_b)_{min} - (d_j)_{max}}{2} = \frac{1.753 - 1.750}{2} = 0.0015\ in$$

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

$\frac{l'}{d} = \frac{1}{2}$, $l' = \frac{d}{2} = 0.875\ in$

Average bearing pressure $P = \frac{W}{4\ r\ l'} = \frac{900}{4 \cdot 0.875 \cdot 0.875} = 294\ psi$

Sommerfeld number $\boxed{S = \left(\frac{r}{c}\right)^2 \left(\frac{\mu N}{P}\right) = \left(\frac{0.875}{0.0015}\right)^2 \left(\frac{\mu' \cdot 10^{-6} \cdot 50}{294}\right) = 0.0579\mu'}$



Initial guess: average film temperature $T_F = 170°\text{F}$

SAE 20 oil viscosity

$$\mu = \mu_o exp\left(\frac{b}{T_F + 95}\right) = 0.0136 \cdot exp\left(\frac{1271.6}{170 + 95}\right) = 1.650\ \mu reyn$$

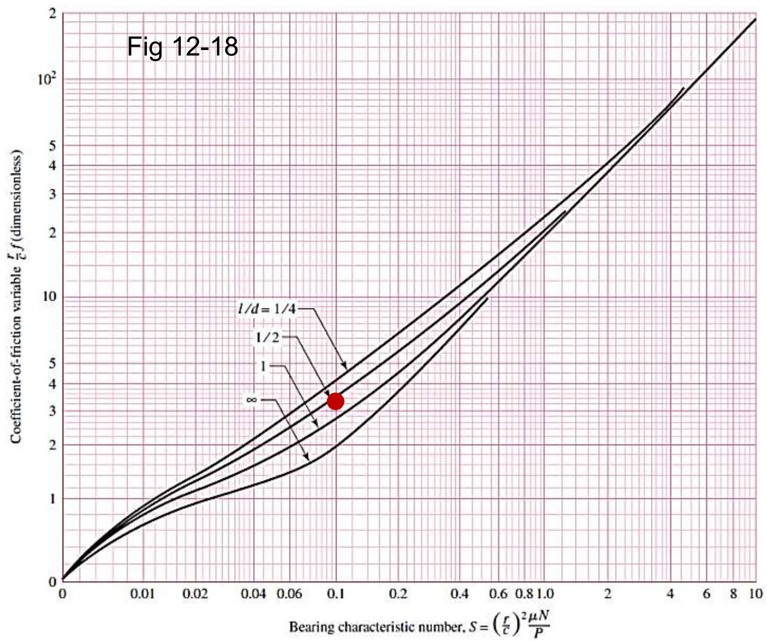
Table 12–1

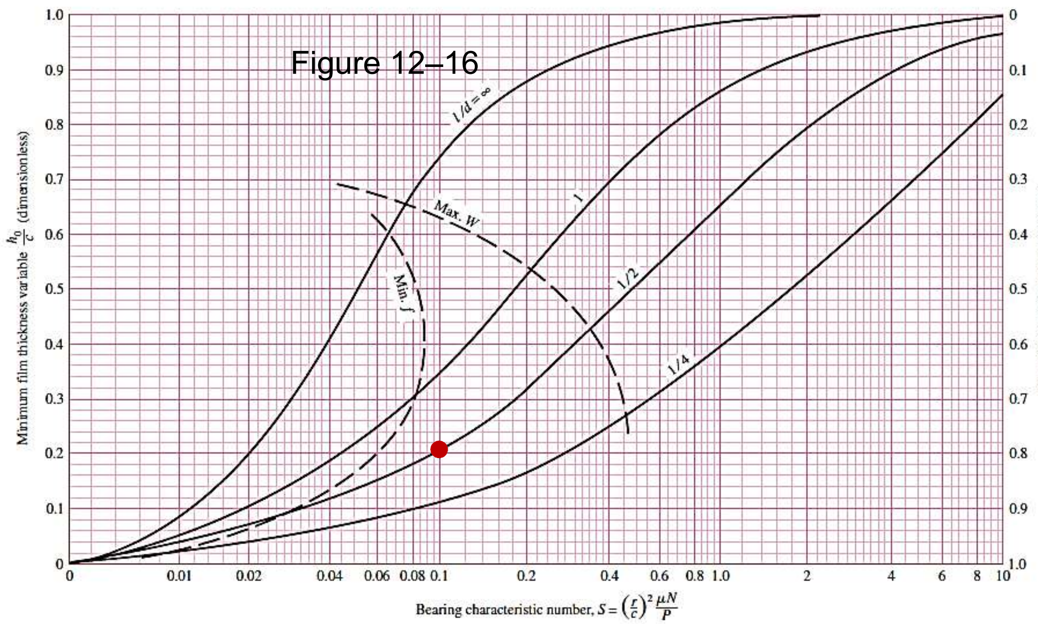
Oil Grade, SAE	Viscosity μ_0, reyn	Constant b, °F
10	$0.0158(10^{-6})$	1157.5
20	$0.0136(10^{-6})$	1271.6
30	$0.0141(10^{-6})$	1360.0
40	$0.0121(10^{-6})$	1474.4
50	$0.0170(10^{-6})$	1509.6
60	$0.0187(10^{-6})$	1564.0

EXAMPLE 12-6 (Cont'd)

S=0.0955

From Fig 12-18, $\frac{fr}{c} = 3.3$ From Fig. 12-16, $\epsilon = 1 - \frac{h_o}{c} = 0.8$





EXAMPLE 12-6 (Cont'd)

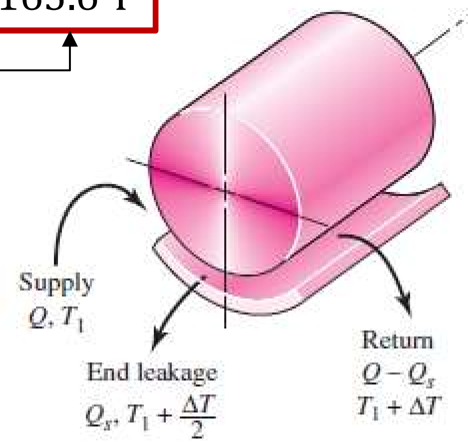
$$\Delta T_F = 0.0123\left(\frac{fr}{c}\right)\frac{SW^2}{(1+1.5\epsilon^2)p_s r^4} = 0.0123(3.3)\frac{0.0955\cdot 900^2}{(1+1.5\cdot 0.8^2)\cdot 30\cdot 0.875^4} = 91.1°\text{F}$$

Estimated side flow oil temperature: $T_{av} = T_s + \frac{\Delta T_F}{2} = 120 + \frac{91.1}{2} = \boxed{165.6°\text{F}}$

<u>2nd Iteration</u>: revised film temperature: $\boxed{T_F = 168.5°\text{F}}$

Table below summarizes the iterative results

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



After two iterations, T_{av} is in agreement with the guesstimated T_F

More iterations will be needed in case discrepancy between T_{av} and T_F had not closed.

Oil outlet temperature: $T_{max} = 120 + 97.1 = 211.1°\text{F}$

Revised oil viscosity $\mu = \mu_o exp\left(\frac{b}{T_F+9}\right) = 0.0136\cdot exp\left(\frac{1271.6}{168.5+95}\right) = 1.693\ \mu reyn$

Revised Sommerfeld number $S = 0.0579\mu' = 0.0982$

Revised eccentricity $\epsilon = 0.792$ from Fig 12-16.

Revised $\left(\frac{fr}{c}\right) = 3.39$

EXAMPLE 12-6 (Cont'd)

Assessment per Trumpler criteria:

- Min film thickness
  - $h_o = (1 - \epsilon)c = 0.000312\ in$
  - required $h_o \geq 0.0002 + 0.00004 \cdot d = 0.000270\ in$ (Pass)
- Oil outlet temperature:
  - $T_{max} = 120 + 97.1 = 211.1°\text{F} \ < 250°\text{F}$ (Pass)

- Average bearing pressure
  - For starting load: $P = \frac{W}{4\,r\,l'} = \frac{900}{4{\cdot}0.875\ \ .875} = 294\ psi$ < 300 psi (Pass)
  - For running load: $n_d = \frac{300}{294} = 1.02 < 2$ (Fail, required min SF =2)

Side flowrate $Q_s = \frac{\pi\,p_s\,r\,c^3}{3\mu l'}(1 + 1.5\epsilon^2) = \frac{\pi{\cdot}30{\cdot}\ \ .875{\cdot}\ 0.0015^3}{3(1.693{\cdot}\ \ ^{-6})0.875}(1 + 1.5 \cdot 0.792^2) = 0.122\ \frac{in^3}{s}$

Heat removed by oil flow:

$H_{loss} = \rho C_p Q_s \Delta T = 0.0311 \cdot 0.042 \cdot 0.122 \cdot 97.1 = 0.156\ \frac{Btu}{s} \approx 562\ \frac{Btu}{hr} < 800\frac{Btu}{hr}$ (Pass)

Parasitic frictional torque $T = fWr = \left(\frac{fr}{c}\right)Wc = 3.39 \cdot 900 \cdot 0.0015 = 4.58\ in \cdot lbf$

EXAMPLE 12-6 (Cont'd)

Sommerfeld number S=0.0982 , $\frac{l'}{d} = \frac{1}{2}$

Fig 12-21 gives $\frac{P}{p_{max}} = 0.28$, $p_{max} = \frac{P}{0.28} = \frac{294}{0.28} = 1050\, psi$

Total film pressure $p_{total} = p_{max} + p_s = 1080\, psi$

For pressure-fed bearing, maximum film pressure must be increased by the oil supply pressure p_s to obtain the total film pressure.

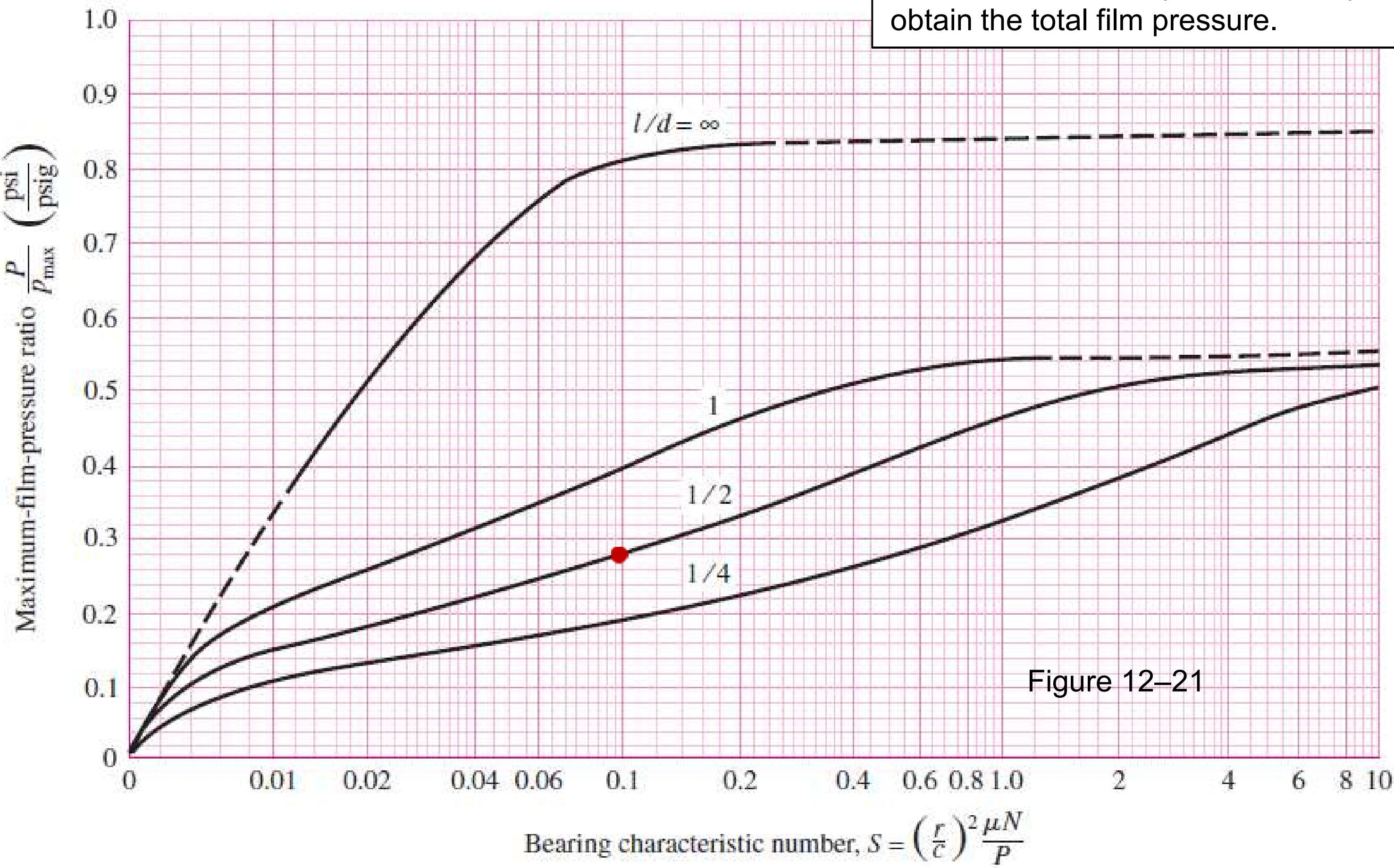


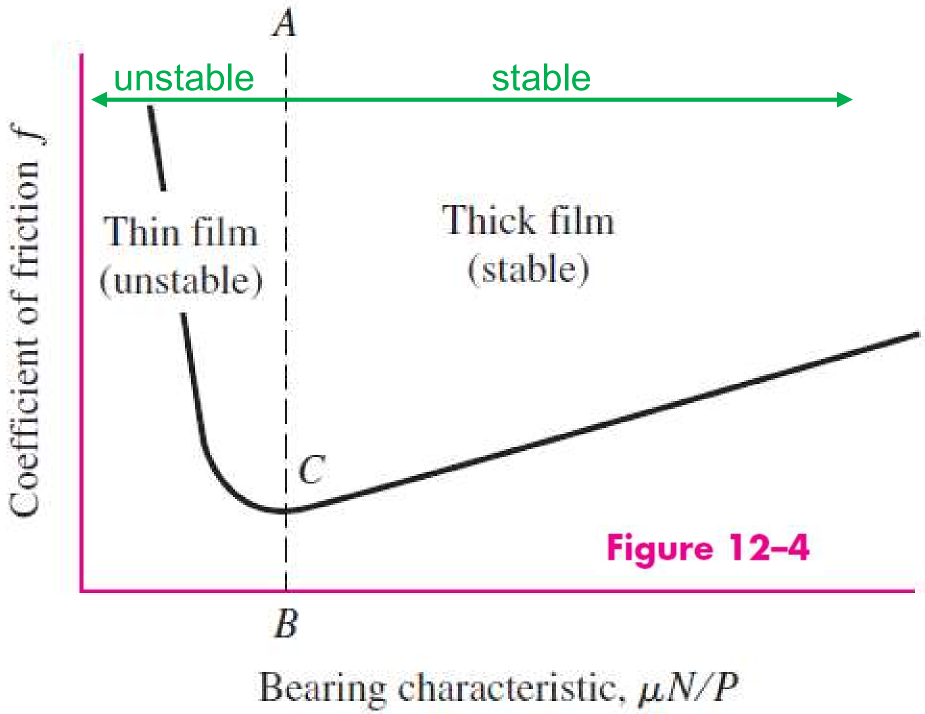
Figure 12–21

SCUPI

12-4 Stable Lubrication

Change in coefficient of friction versus bearing characteristic

- Region to the right of line BA defines stable lubrication because variations are self-correcting
- Region to the left of line BA represents unstable lubrication.



Point B = $30\,\frac{cP\cdot rev}{min\cdot psi}\left(\frac{ZN}{P}\right)$

Point B = $1.7(10^{-6})\,\frac{reyn\cdot rev}{sec\cdot psi}\left(\frac{\mu N}{P}\right)$

S. A. McKee and T. R. McKee, “Journal Bearing Friction in the Region of Thin Film Lubrication,” *SAE J.*, vol. 31, 1932, pp. (T)371–377.

12-10 Clearance Effect on Bearing Performance

- Bearing radial clearance c is the most critical parameter in bearing design
- Clearance is difficult to hold accurate during manufacture, and it may increase because of wear.

Clearance Effect on Bearing Performance

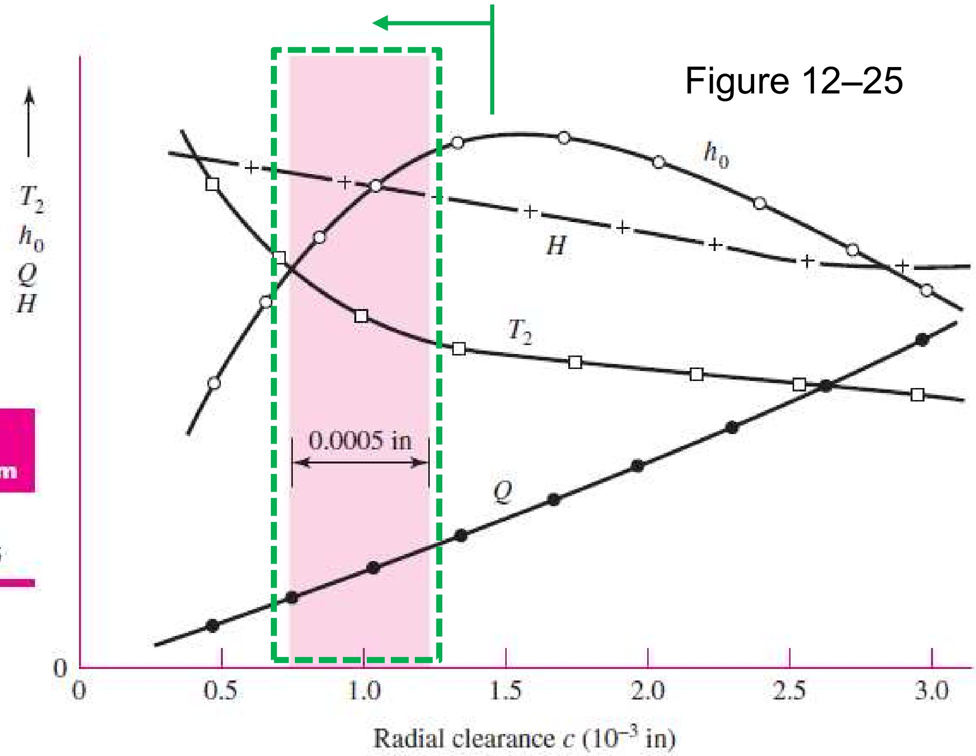
- clearance too tight ► temperature increases ► lower min film thickness
  - High temperatures may cause the bearing to fail by fatigue.
  - Oil film too thin, dirt particles may be unable to pass without scoring or may embed themselves in the bearing.
  - Either event will cause excessive wear and friction, resulting in high temperatures and possible seizing.
- clearance too loose ► noisy bearing ► bearing starvation ► min film thickness too low

A window exists for satisfactory and balanced bearing performance

Table 12-3

Type of Fit	Symbol	Clearance c, in 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

Figure 12–25



Clearance Effect on Bearing Performance

- Best compromise is a clearance range slightly to the left of the top of the min film thickness curve. In this way, future wear will move the operating point to the right and increase the film thickness and decrease the operating temperature.
- Fig. 12-26 gives temperature limits for mineral oils.

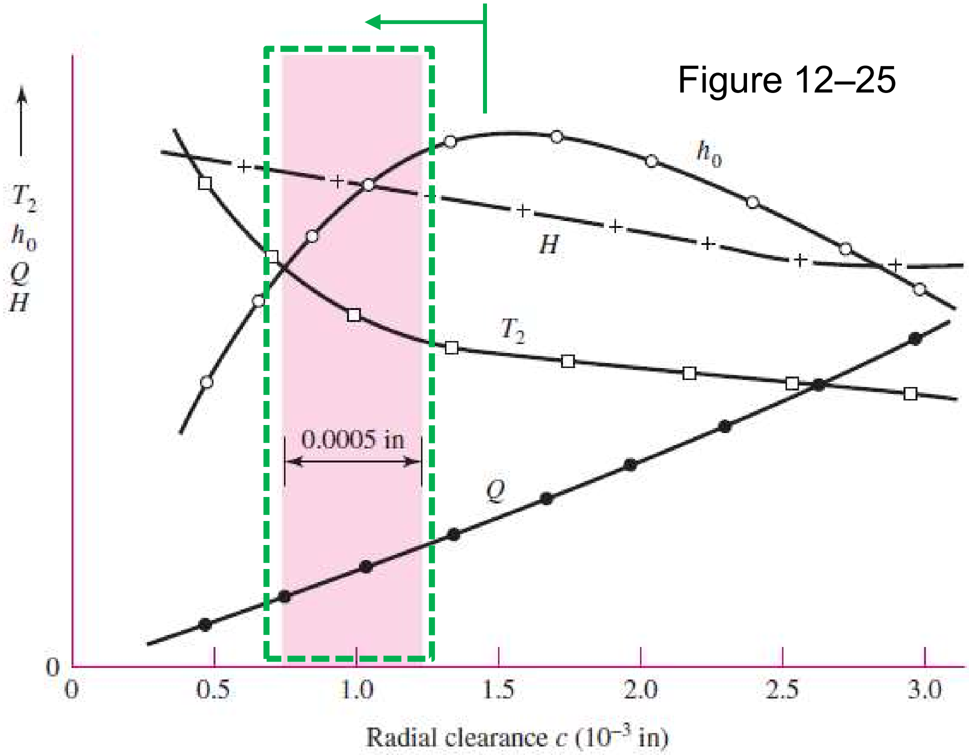


Figure 12–25

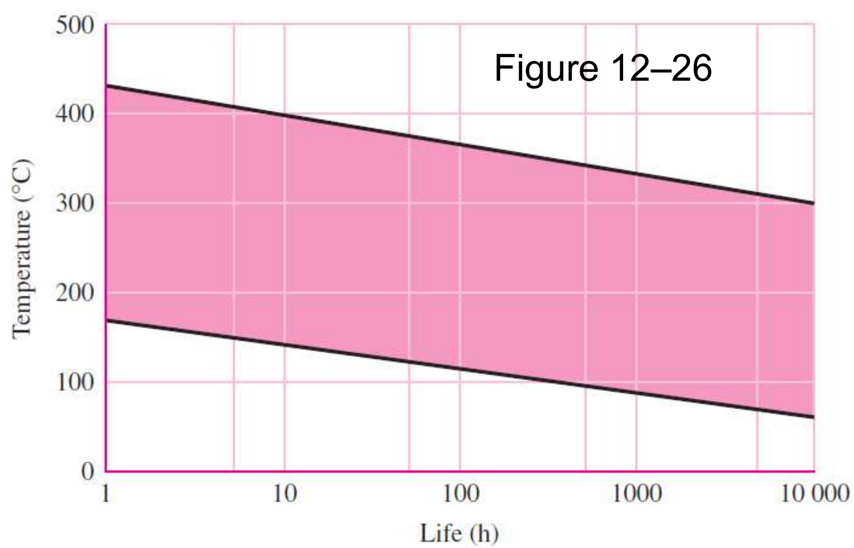
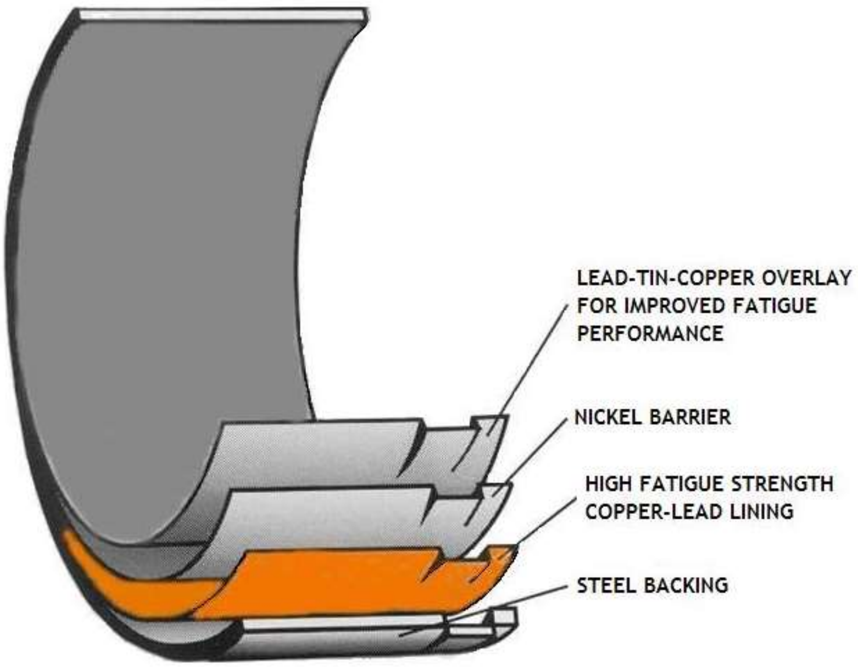


Figure 12–26

12-12 Loads and Materials



Bearing Material Selection

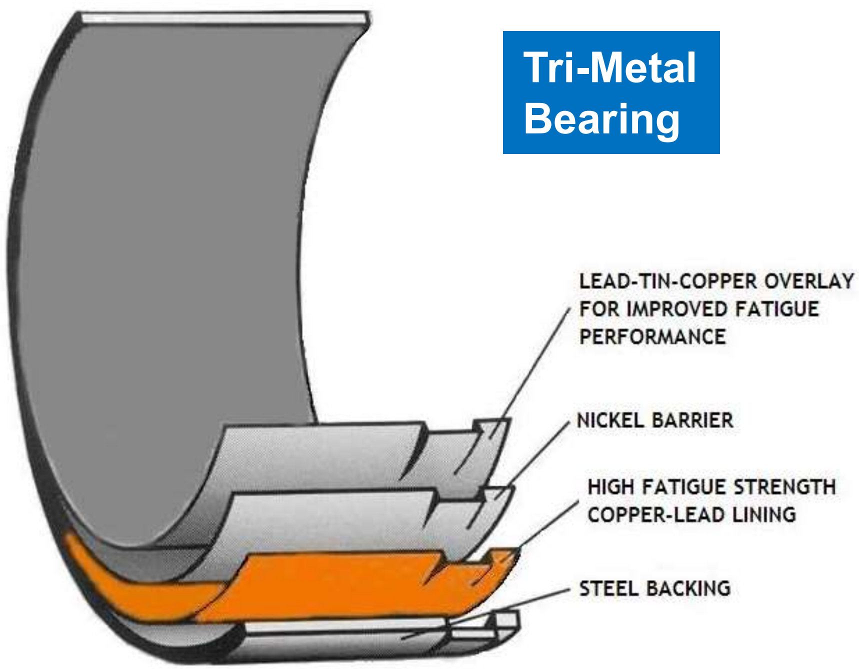
Material properties is a balance among the following concerns:

- Satisfactory compressive and fatigue strength to resist applied loads
- Must be soft and a low modulus of elasticity
  - necessary to permit the material to wear or break in, since the material can then conform to slight irregularities and absorb and release foreign particles.
- Resistance to wear and the coefficient of friction
  - all bearings must operate, at least for part of the time, with thin-film or boundary lubrication.
- Ability to resist corrosion
- Cost of producing the bearing

Types of Bearing Material

- Bearing life can be increased very substantially by depositing a layer of <u>babbitt</u>, or other <u>white metal</u>, in thicknesses from 0.001 to 0.014 in over steel backup material.
- A copper-lead layer on steel to provide strength, combined with a babbitt overlay to enhance surface conformability and corrosion resistance, makes an excellent bearing.

Tri-Metal Bearing



Bi-Metal Bearing

EXAMPLE 12-6 (Cont'd)

Table 12–5
Range of Unit Loads in
Current Use for Sleeve Bearings

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

Table 12–6 Characteristics of Bearing Alloys

Alloy Name	Thickness, in	SAE Number	Clearance Ratio r/c	Load Capacity	Corrosion Resistance
Tin-base babbitt	0.022	12	600–1000	1.0	Excellent
Lead-base babbitt	0.022	15	600–1000	1.2	Very good
Tin-base babbitt	0.004	12	600–1000	1.5	Excellent
Lead-base babbitt	0.004	15	600–1000	1.5	Very good
Leaded bronze	Solid	792	500–1000	3.3	Very good
Copper-lead	0.022	480	500–1000	1.9	Good
Aluminum alloy	Solid		400–500	3.0	Excellent
Silver plus overlay	0.013	17P	600–1000	4.1	Excellent
Cadmium (1.5% Ni)	0.022	18	400–500	1.3	Good
Trimetal 88*				4.1	Excellent
Trimetal 77†				4.1	Very good

*This is a 0.008-in layer of copper-lead on a steel back plus 0.001 in of tin-base babbitt.
†This is a 0.013-in layer of copper-lead on a steel back plus 0.001 in of lead-base babbitt.