Lubrication and Journal Bearings

- Types of Lubrication
- Viscosity
- Petroff’s Equation
- Thick-Film Lubrication
- Hydrodynamic Theory and Lubrication
- Design Considerations
- The Relations of the Variables – Lubrication Charts

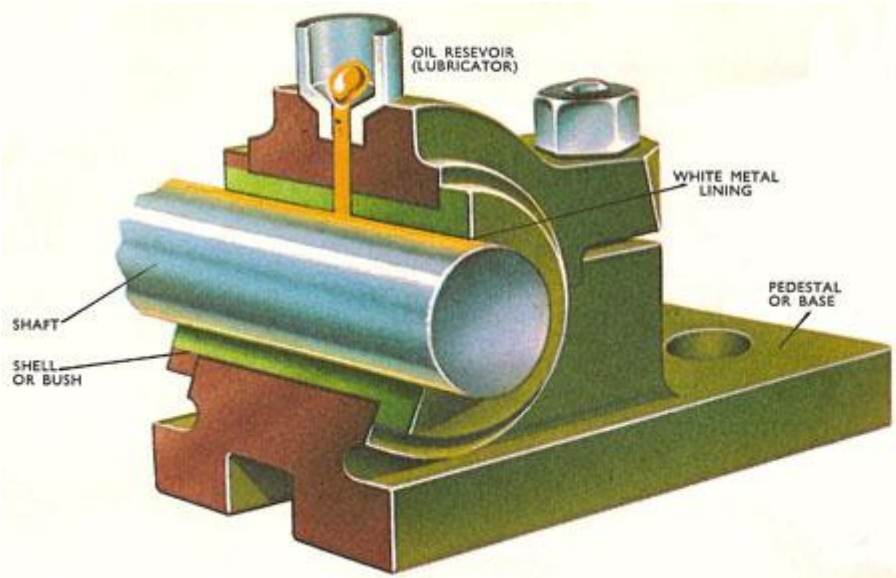
Background

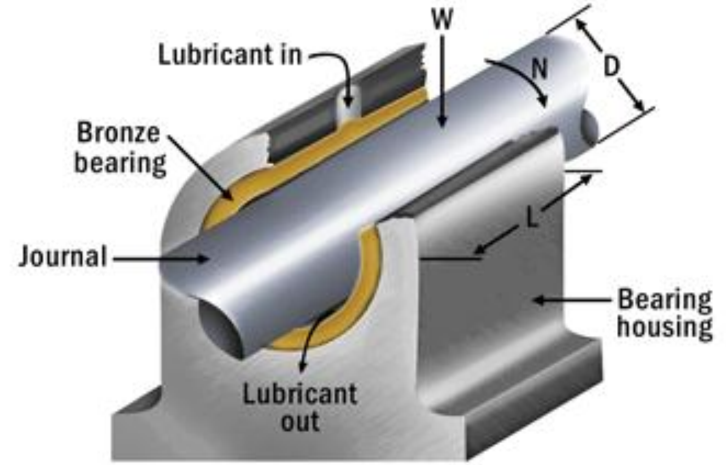
In industry, the **use of journal bearings** is specialized for rotating machinery both low and high speed.

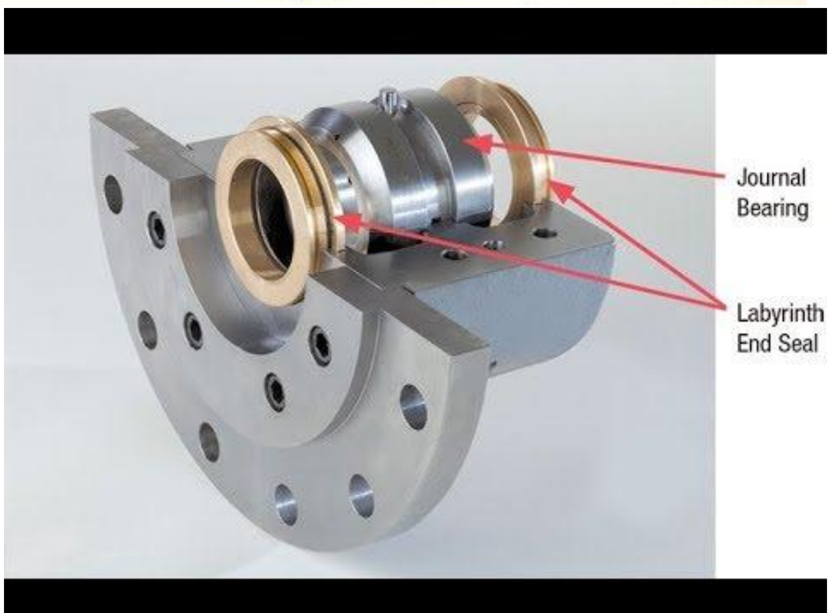
Journal bearings are used widely to support the shafts in industrial machinery involving heavy loads, such as

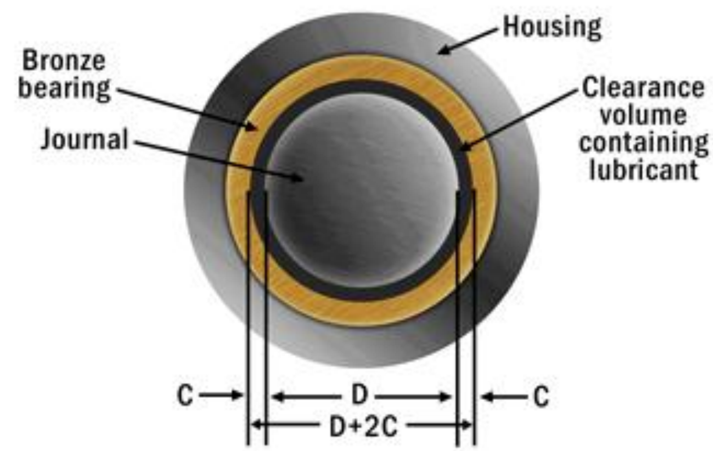
- compressors
- turbines and
- centrifugal pumps.

Background Cont'd







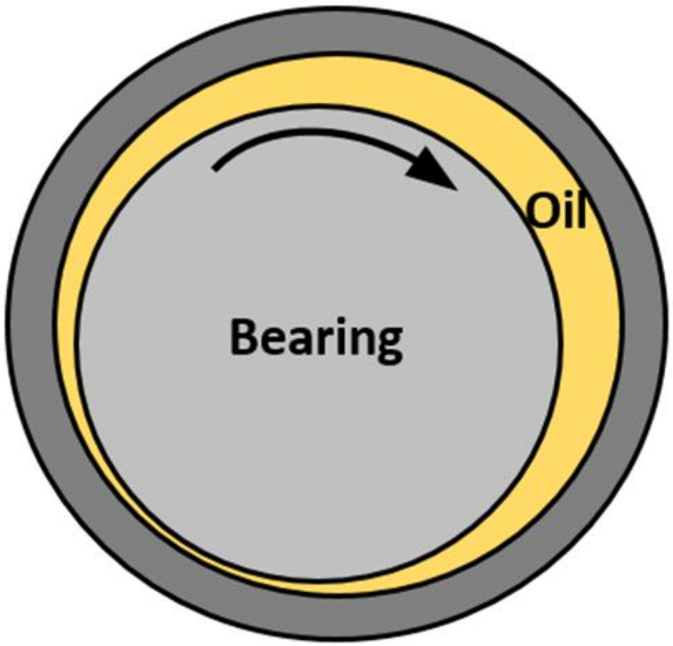


Types of Lubrication

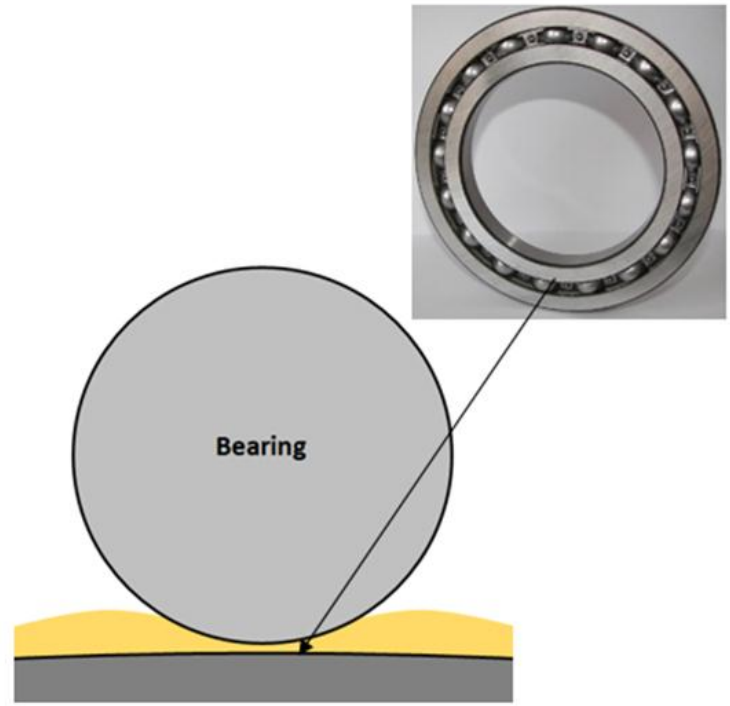
- **Hydrodynamic**
  - Load-Carrying surfaces are separated by a relatively thick film
  - The fluid creates a wedge-shaped zone due to relative motion to separate the parts
  - No metal-to-metal contact
- **Hydrostatic**
  - Lubricant is introduced into the load-bearing zone at a high pressure to separate the parts.
- **Elastohydrodynamic**
  - Occurs when the lubricant is between rolling surfaces such as mating gears.
- **Boundary**
  - Lubricant is just enough to cover the asperities resulting in some degree of metal-to-metal contact at the micro level.
- **Solid film:** Uses solid lubricants such as graphite

Applications – Types of Lubrication

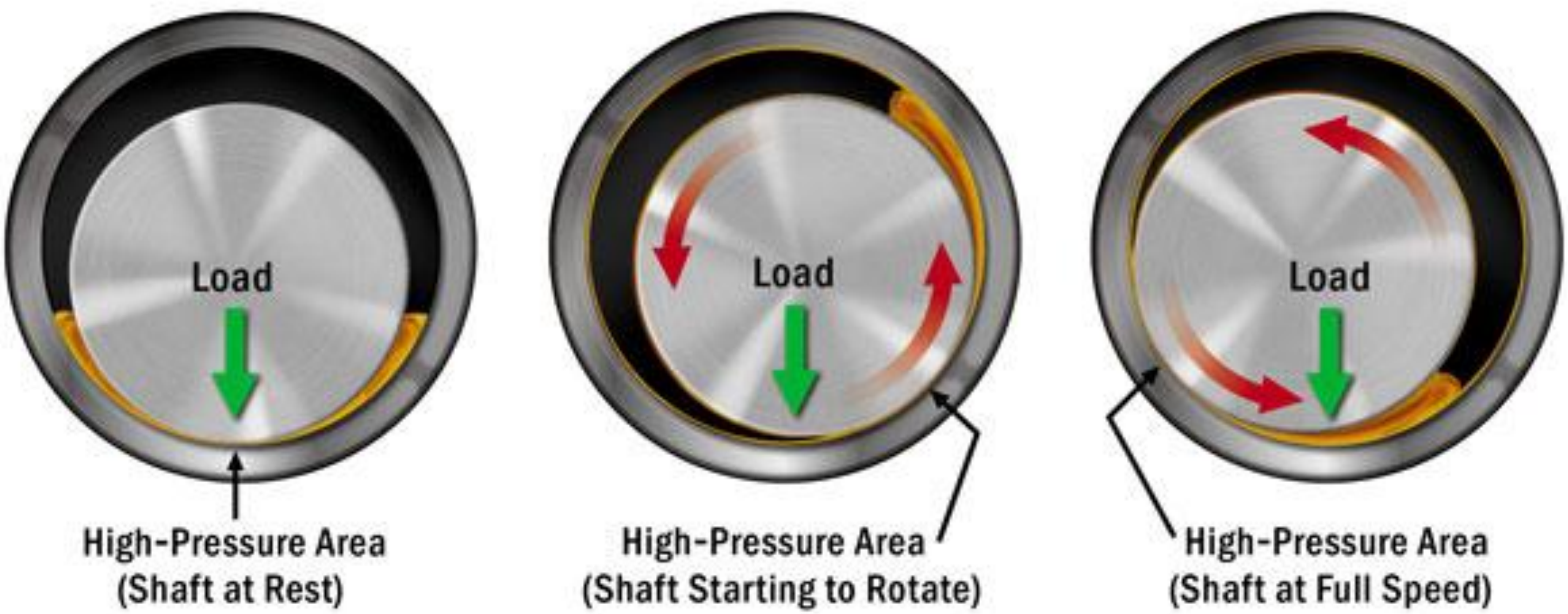
Hydrodynamic Lubrication



Elastohydrodynamic Lubrication



Applications – Types of Lubrication Cont'd



Applications – Types of Lubrication Cont’d

Viscosity

- Shear stress in a fluid is proportional to the rate of change of velocity with respect to *y*

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

- μ is *absolute viscosity*, also called *dynamic viscosity*

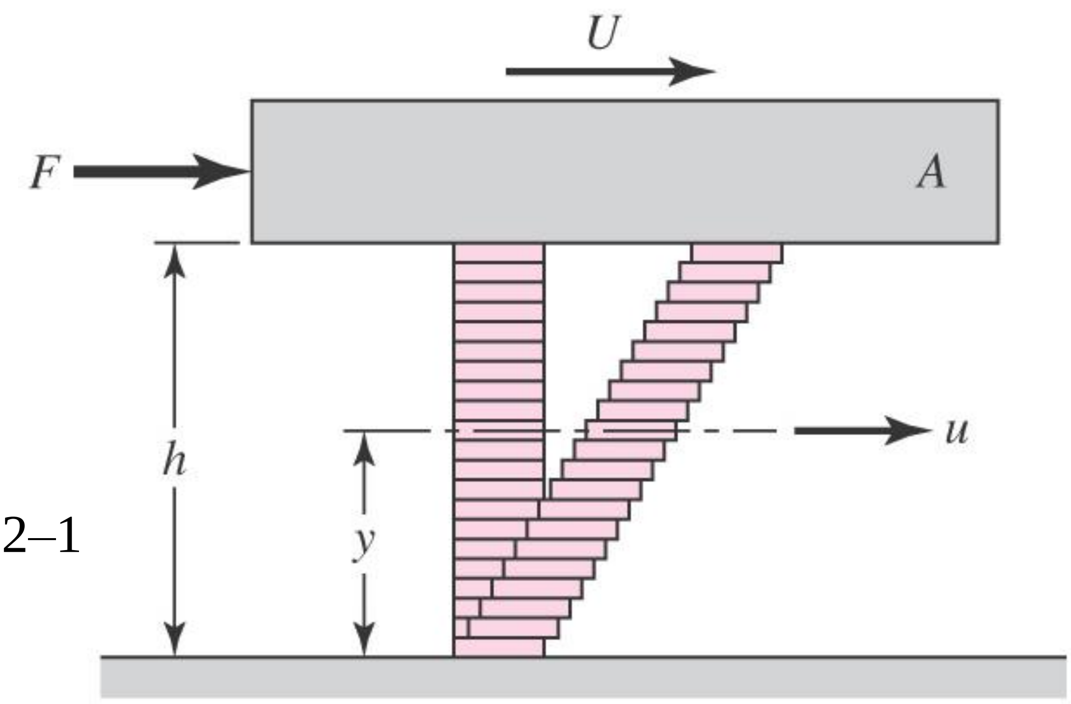


Fig. 12–1

Viscosity

- For most lubricating fluids, the rate of shear is constant, thus

$$du/dy = U/h$$

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

- Fluids exhibiting this characteristic are called *Newtonian fluids*

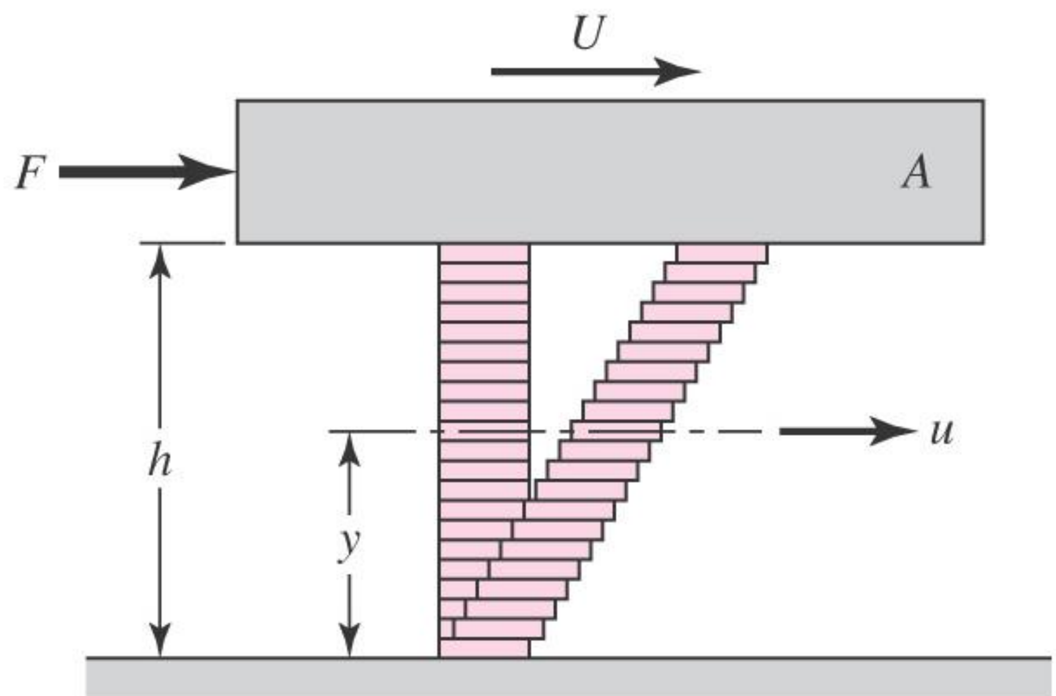


Fig. 12–1

Units of Viscosity

- Units of absolute viscosity
  - ips units: reyn = lbf·s/in^2
  - **SI units:** Pa·s = N·s/m^2 length (metre), mass (kilogram), and time (sec).
  - cgs units: Poise =dyn·s/cm^2
- cgs (centimeters, grams, seconds) units are discouraged, but common historically in lubrication
- Viscosity in cgs is often expressed in centipoise (cP), designated by *Z*
- Conversion from cgs to SI and ips:

$$\mu(\mathrm{Pa}\cdot\mathrm{s}) = (10)^{-3} Z\ (\mathrm{cP})$$

$$\mu(\mathrm{reyn}) = \frac{Z\ (\mathrm{cP})}{6.89(10)^6}$$

$$\mu(\mathrm{mPa}\cdot\mathrm{s}) = 6.89\ \mu'(\mu\mathrm{reyn})$$

Units of Viscosity

- In ips units, the microreyn (μreyn) is often convenient.
- The symbol μ' is used to designate viscosity in μreyn

$$\mu = \mu'/(10^6)$$

Measurement of Viscosity

- *Saybolt Universal Viscosimeter* used to measure viscosity
- Measures time in seconds for 60 mL of lubricant at specified temperature to run through a tube 17.6 mm in diameter and 12.25 mm long
  - Result is *kinematic viscosity*
  - Unit is stoke = cm^2/s
- Using *Hagen-Poiseuille law* kinematic viscosity based on seconds Saybolt, also called *Saybolt Universal viscosity* (SUV) in seconds is

$$Z_k = \left(0.22t - \frac{180}{t}\right) \qquad (12\text{–}3)$$

where Z_k is in centistokes (cSt) and t is the number of seconds Saybolt

Measurement of Viscosity

- In SI, kinematic viscosity ν has units of m²/s
- Conversion is $\nu(\mathrm{m^2/s}) = 10^{-6} Z_k\ (\mathrm{cSt})$
- Eq. (12–3) in SI units,

$$\nu = \left(0.22t - \frac{180}{t}\right)(10^{-6}) \qquad (12\text{–}4)$$

- To convert to dynamic viscosity, multiply ν by density in SI units

$$\mu = \rho\left(0.22t - \frac{180}{t}\right)(10^{-6}) \qquad (12\text{–}5)$$

where ρ is in kg/m³ and μ is in pascal-seconds

Comparison of Absolute Viscosities of Various Fluids

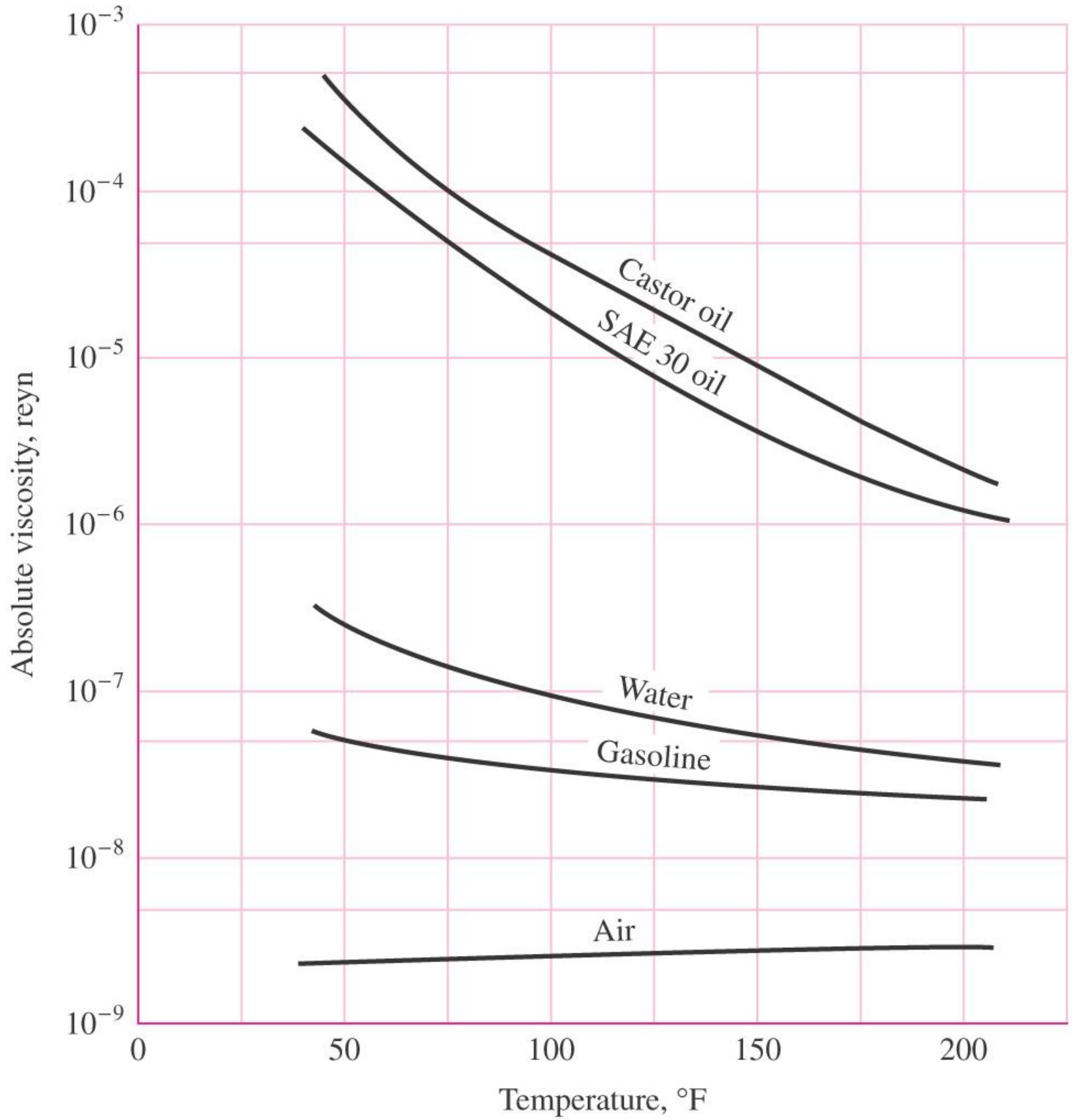


Fig. 12–2

Petroff's Lightly Loaded Journal Bearing

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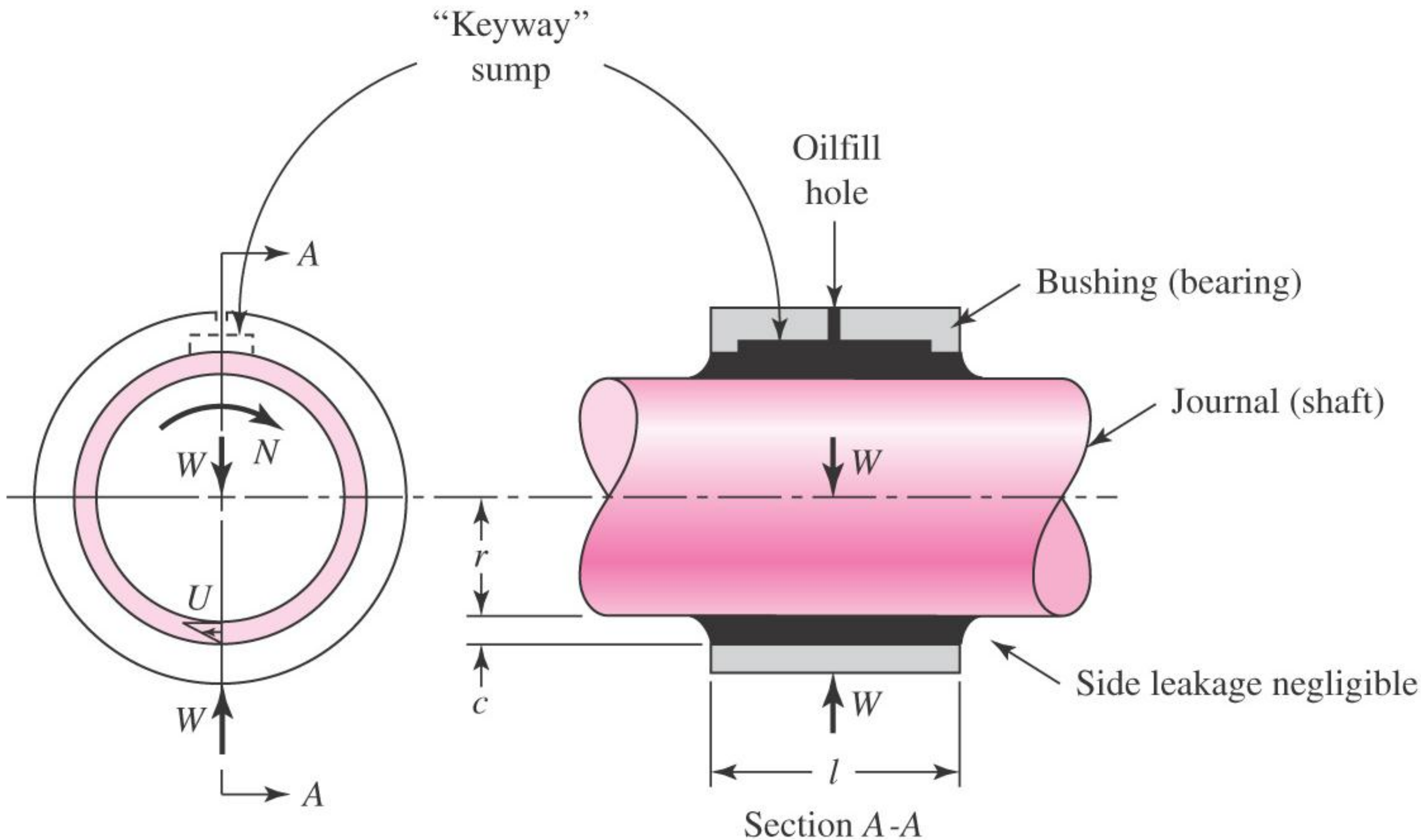


Fig. 12–3

Stable Lubrication

- To the right of *AB*, changes in conditions are self-correcting and results in stable lubrication
- To the left of *AB*, changes in conditions tend to get worse and results in unstable lubrication
- Point *C* represents the approximate transition between metal-to-metal contact and thick film separation of the parts
- Common design constraint for point *B*, $\dfrac{\mu N}{P} \geq 1.7(10^{-6})$

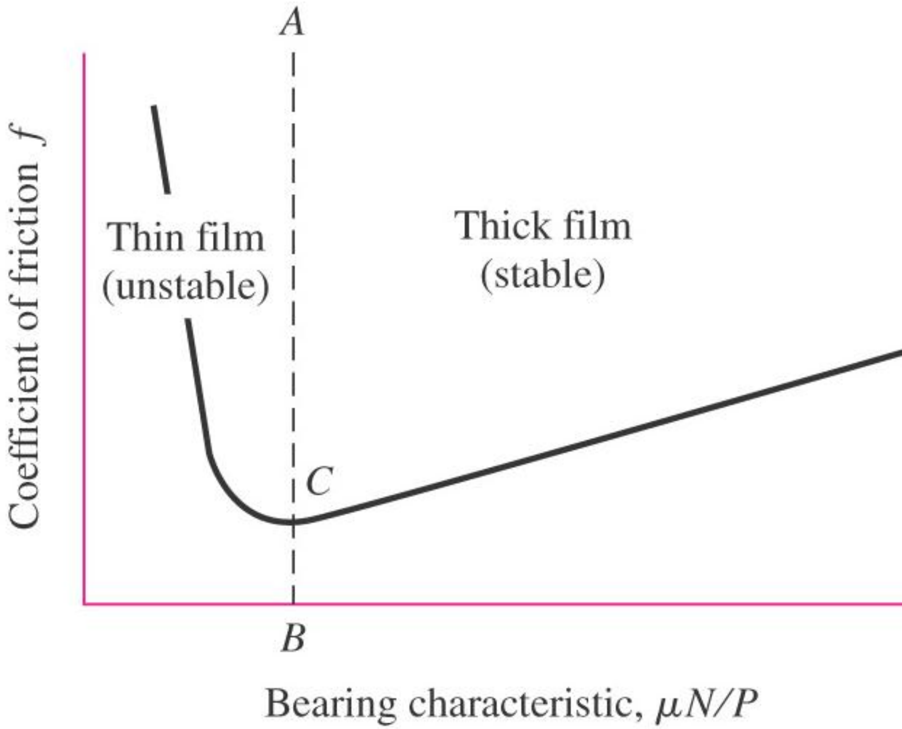
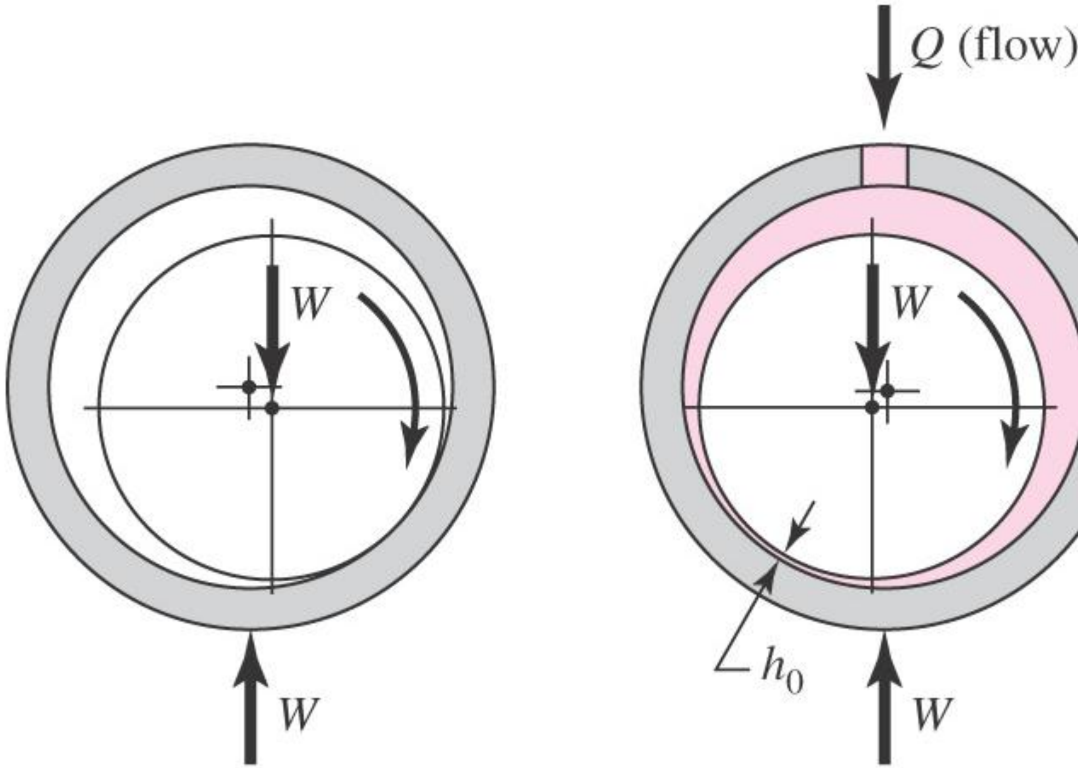


Fig. 12–4

Thick Film (Hydrodynamic) Lubrication

- Formation of a film



(a) Dry
(b) Lubricated

Fig. 12–5

Nomenclature of a Journal Bearing

- Center of journal at *O*
- Center of bearing at *O'*
- *Eccentricity e*
- *Minimum film thickness* h_0 occurs at line of centers
- Film thickness anywhere is *h*
- *Eccentricity ratio*

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

- *Partial bearing* has $\beta < 360$
- *Full bearing* has $\beta = 360$
- *Fitted bearing* has equal radii of bushing and journal

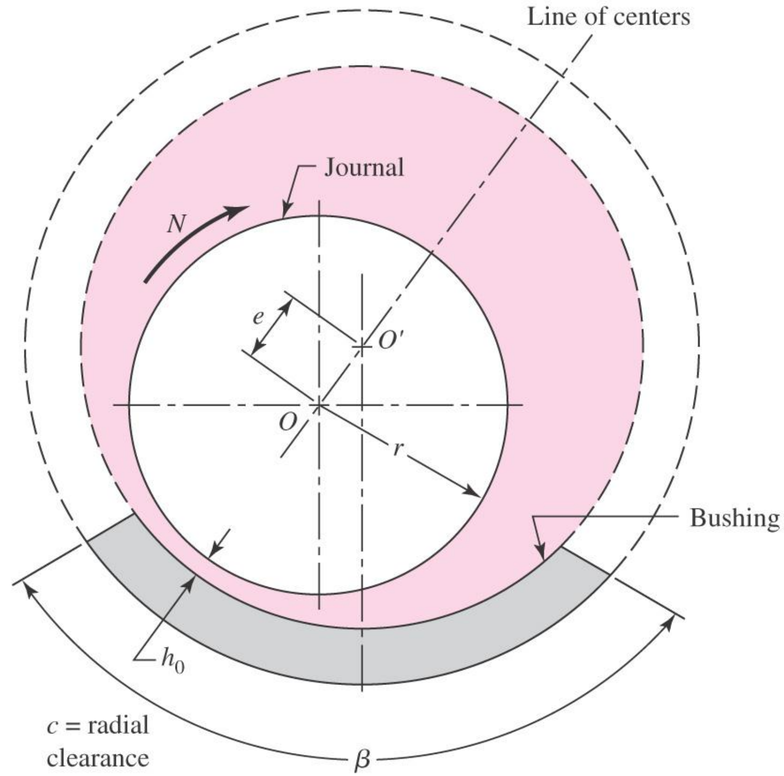


Fig. 12–6

Design Considerations

- **Variables either given or under control of designer**

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

- **Dependent variables, or *performance factors***

1. The coefficient of friction f
2. The temperature rise ΔT
3. The volume flow rate of oil Q
4. The minimum film thickness h_0

Trumpler's Design Criteria

Trumpler, a well-known bearing designer, recommended a set of design criteria.

- **Minimum film thickness** to prevent accumulation of ground off surface particles

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

- **Maximum temperature** to prevent vaporization of lighter lubricant components

$$T_{\max} \leq 250^\circ\text{F} \qquad (120^\text{o}\text{ C}) \qquad (b)$$

- **Maximum starting load** to limit wear at startup when there is metal-to-metal contact

$$\frac{W_{st}}{lD} \leq 300 \text{ psi} \qquad (2 \text{ MPa}) \qquad (c)$$

- **Minimum design factor** on running load

$$n_d \geq 2 \qquad (d)$$

Reynolds Equation

- Classical Reynolds equation for one-dimensional flow, neglecting side leakage,

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

- With side leakage included,

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

- No general analytical solutions
- One important approximate solution by Sommerfeld,

$$\frac{r}{c}f = \phi\left[\left(\frac{r}{c}\right)^2\frac{\mu N}{P}\right] \qquad (12\text{–}12)$$

The Relations of the Variables

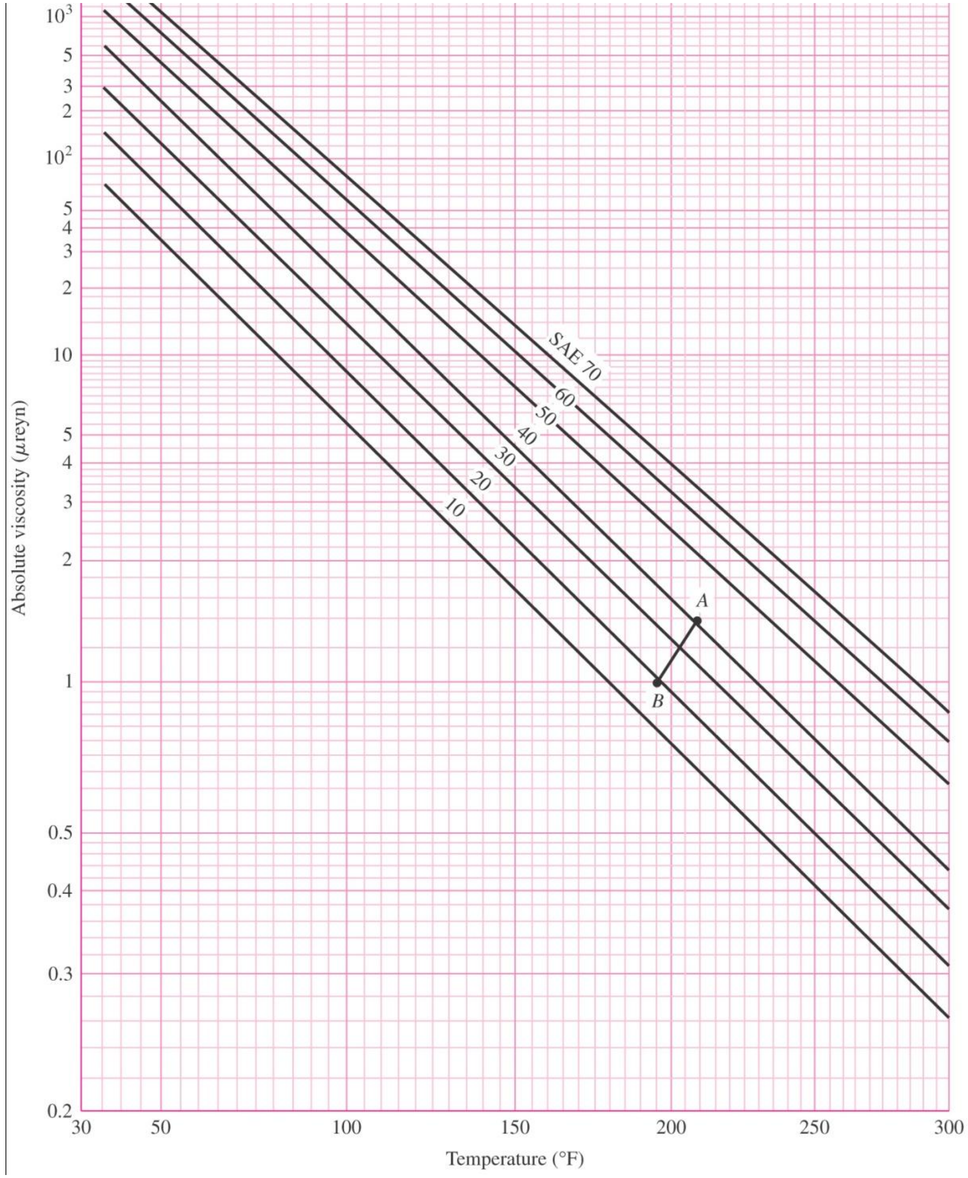
- Albert Raymondi and John Boyd used an iteration technique to solve Reynolds' equation.
- Published 45 charts and 6 tables
- This text includes charts from Part III of Raymondi and Boyd
  - Assumes infinitely long bearings, thus no side leakage
  - Assumes full bearing
  - Assumes oil film is ruptured when film pressure becomes zero

Viscosity Charts

- Viscosity is clearly a function of temperature
- Viscosity charts of common lubricants are given in Figs. 12–12 through 12–14
- Raymondi and Boyd assumed constant viscosity through the loading zone
  - Not completely true since temperature rises as work is done on the lubricant passing through the loading zone
  - Use average temperature to find a viscosity

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

Viscosity-Temperature Chart in U.S. Customary Units



Raimondi and Boyd.

Fig. 12–12

Curve Fits for Viscosity-Temperature Chart

- Approximate curve fit for Fig. 12–12 is given by

$$\mu = \mu_0 \exp\left[b/(T + 95)\right], \; T \text{ in } ^\circ\text{F}.$$

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

Table 12–1

A.S. Seireg and S. Dandage, “Empirical Design Procedure for the Thermodynamic Behavior of Journal Bearings,” J. Lubrication Technology, vol. 104, April 1982, pp. 135-148.

Viscosity-Temperature Chart in Metric Units

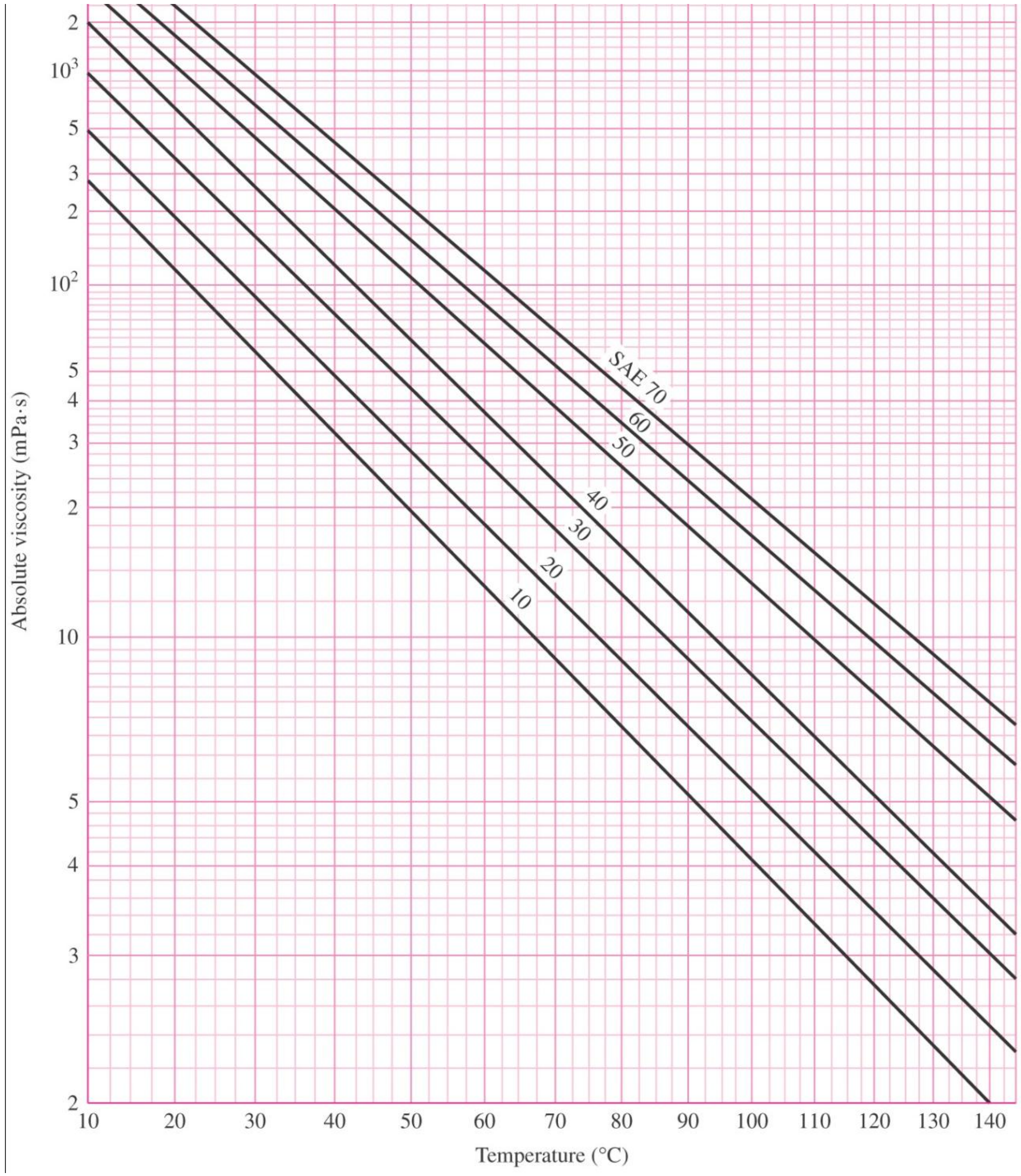


Fig. 12–13

Viscosity-Temperature Chart for Multi-viscosity Lubricants

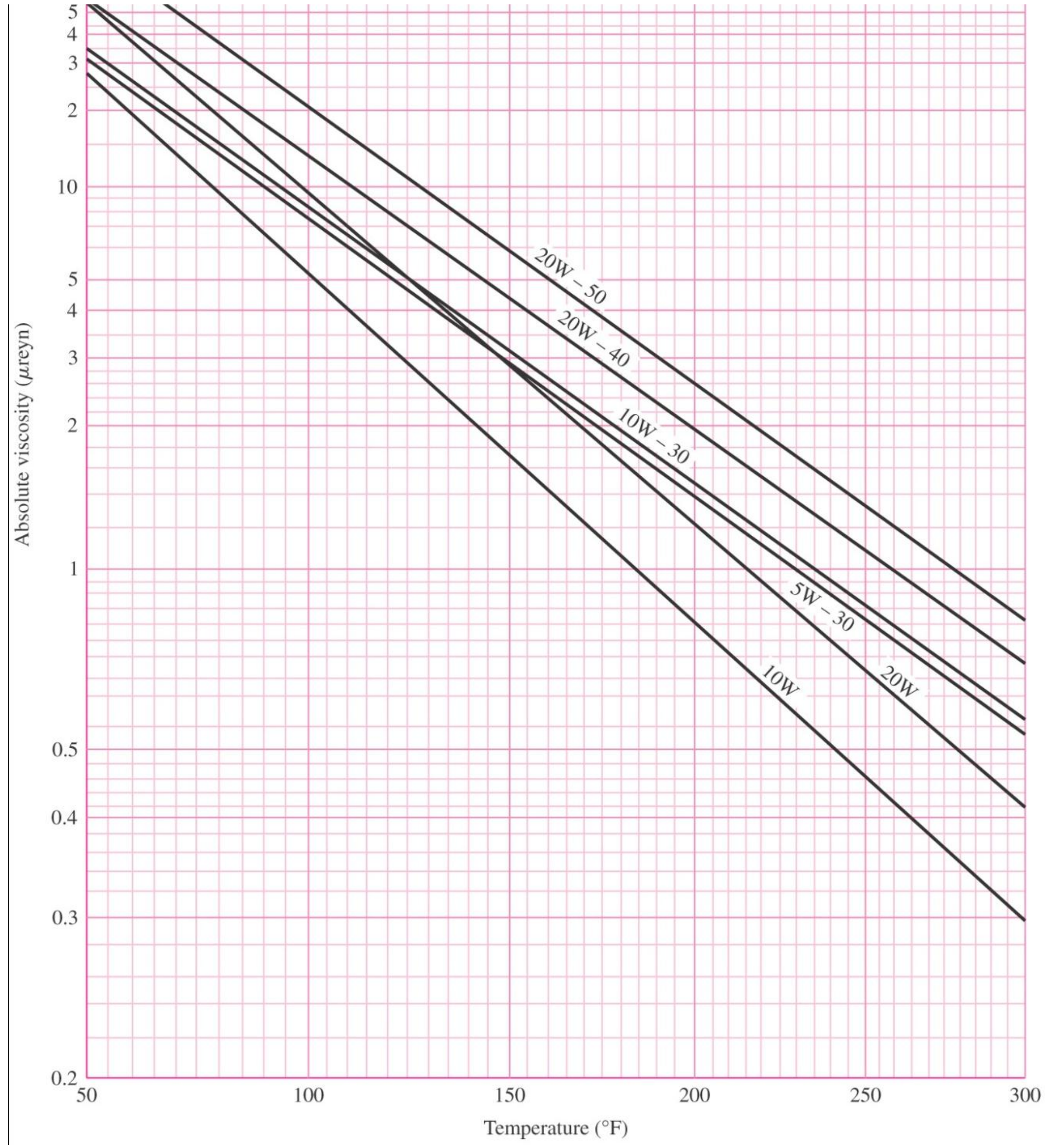
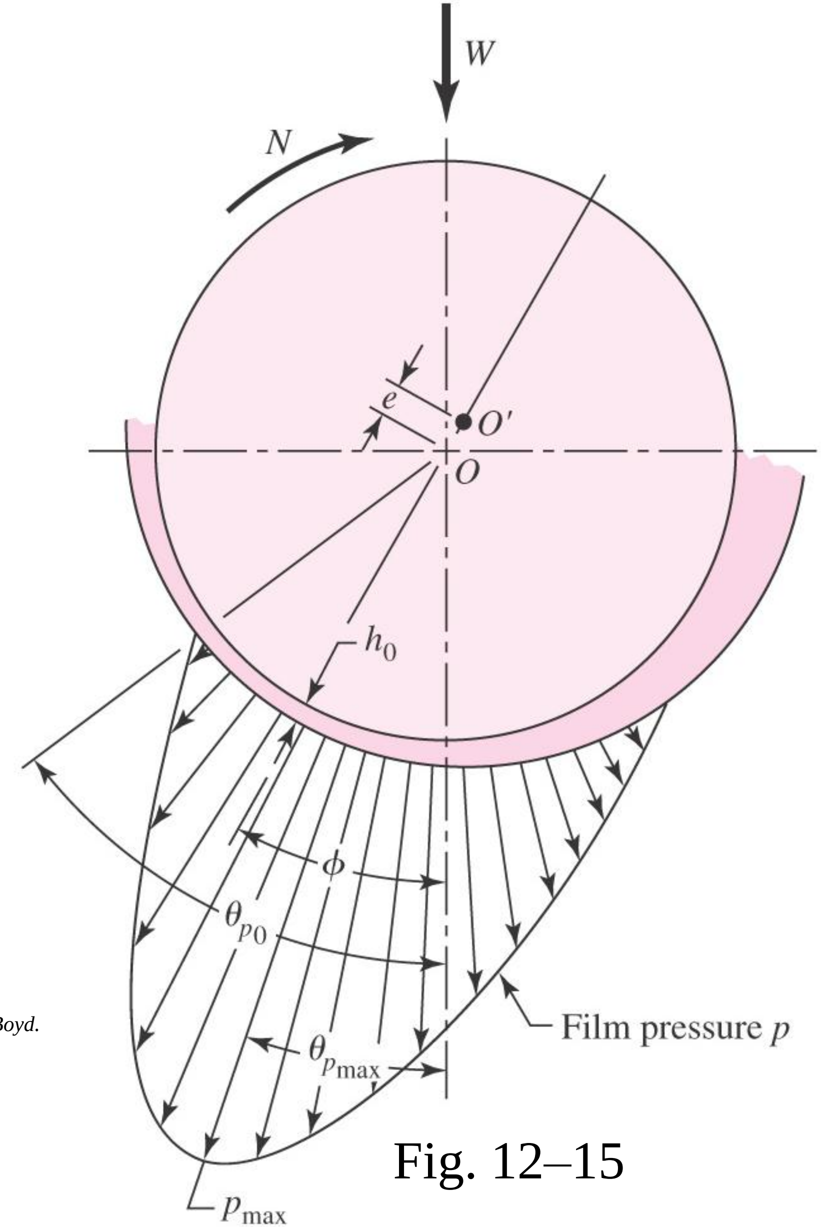


Fig. 12–14

Notation of Raimondi and Boyd

- Polar diagram of the film pressure distribution showing notation used by Raimondi and Boyd



Raimondi and Boyd.

Fig. 12–15

Minimum Film Thickness and Eccentricity Ratio

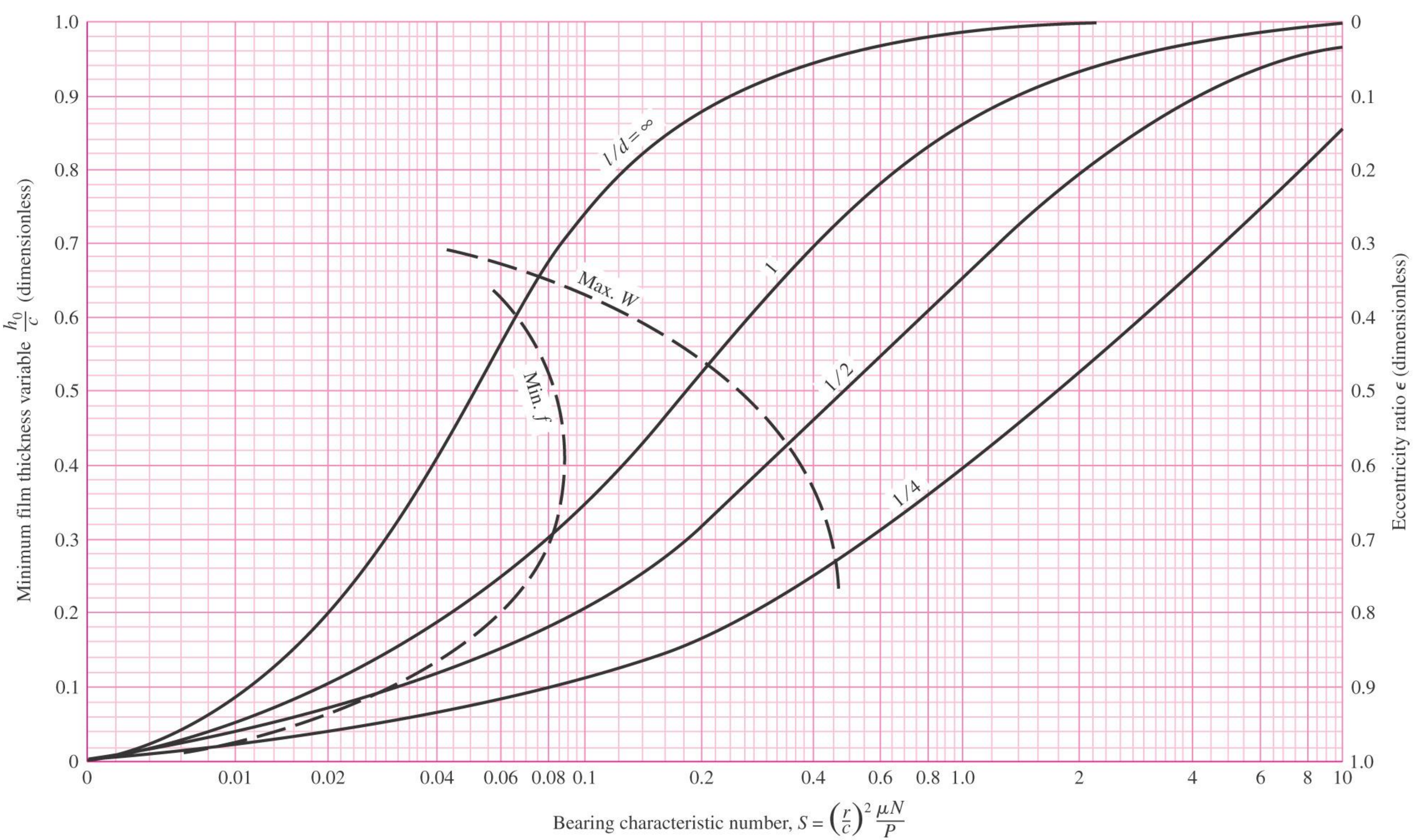
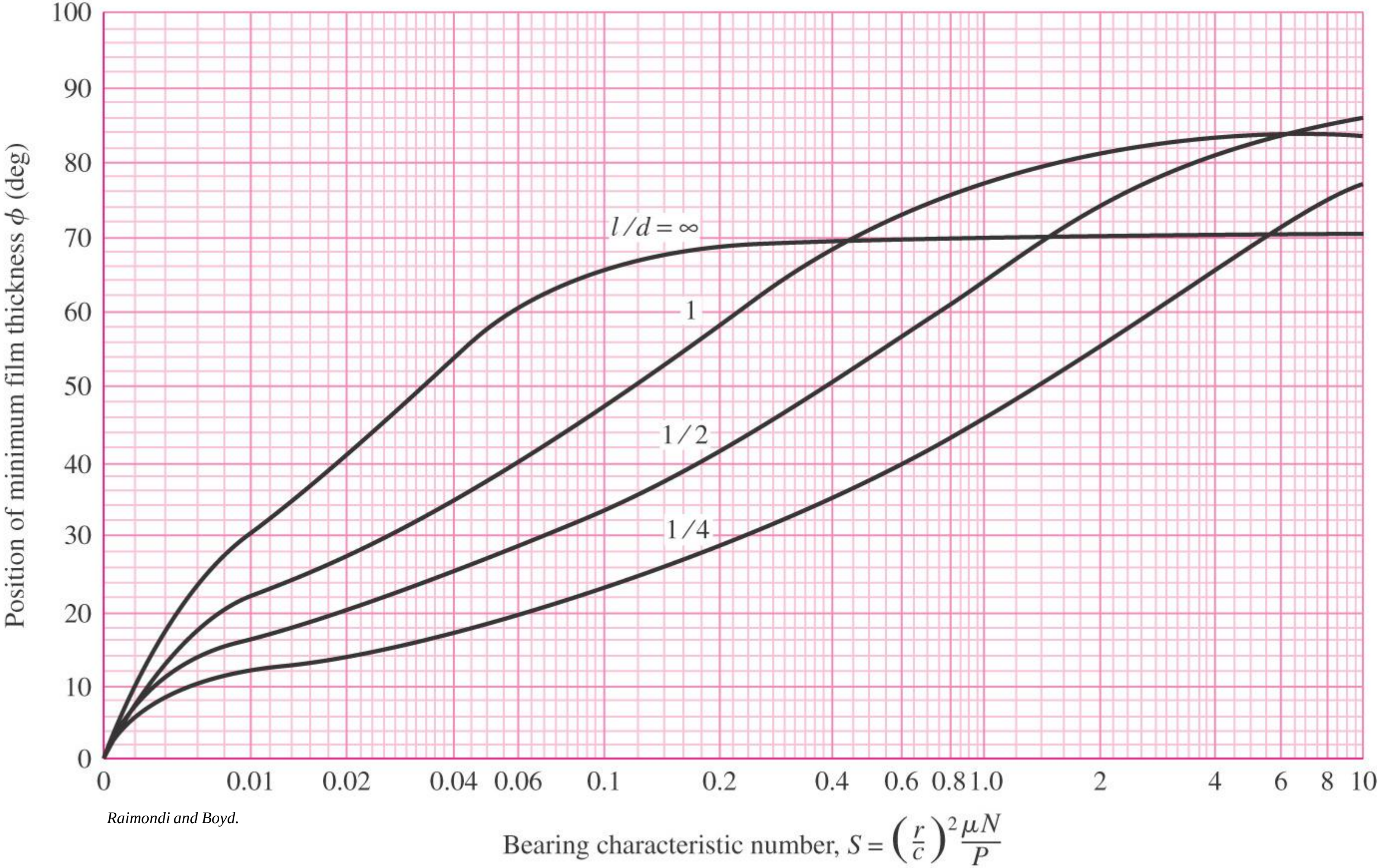


Fig. 12–16

Position of Minimum Film Thickness



Raimondi and Boyd.

Fig. 12–17

Coefficient of Friction Variable

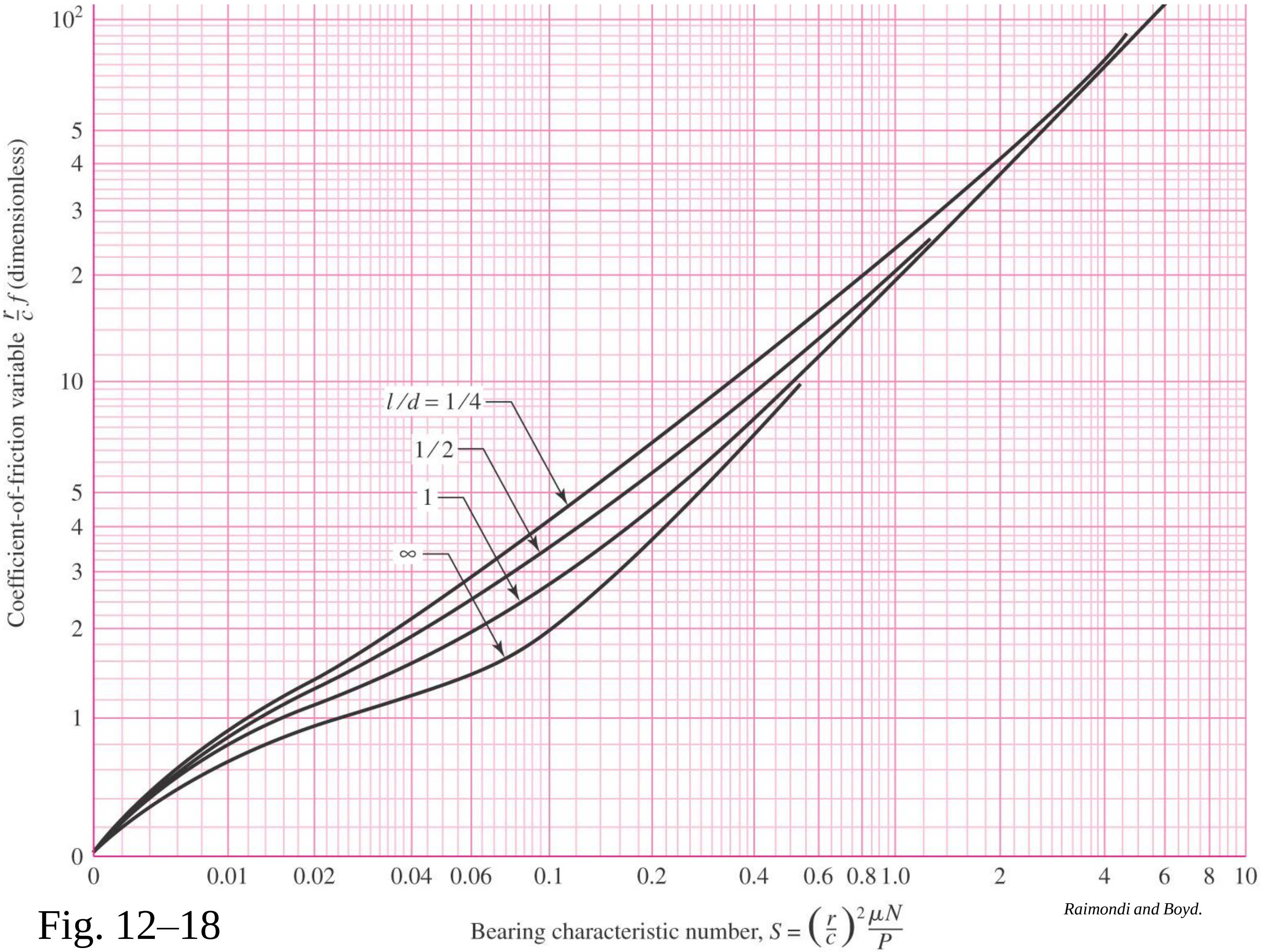


Fig. 12–18

Raimondi and Boyd.

Flow Variable

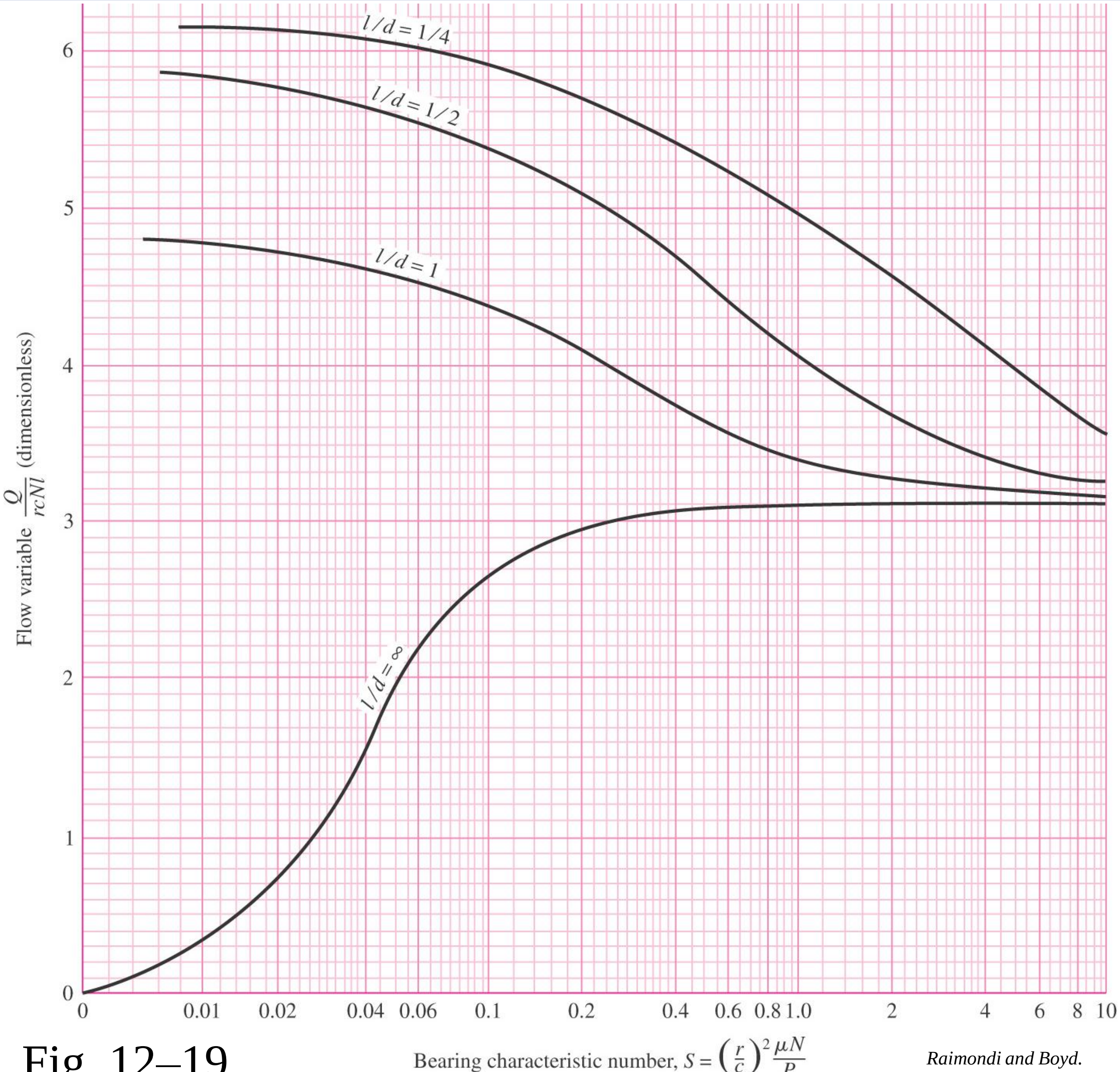


Fig. 12–19

Raimondi and Boyd.

Flow Ratio of Side Flow to Total Flow

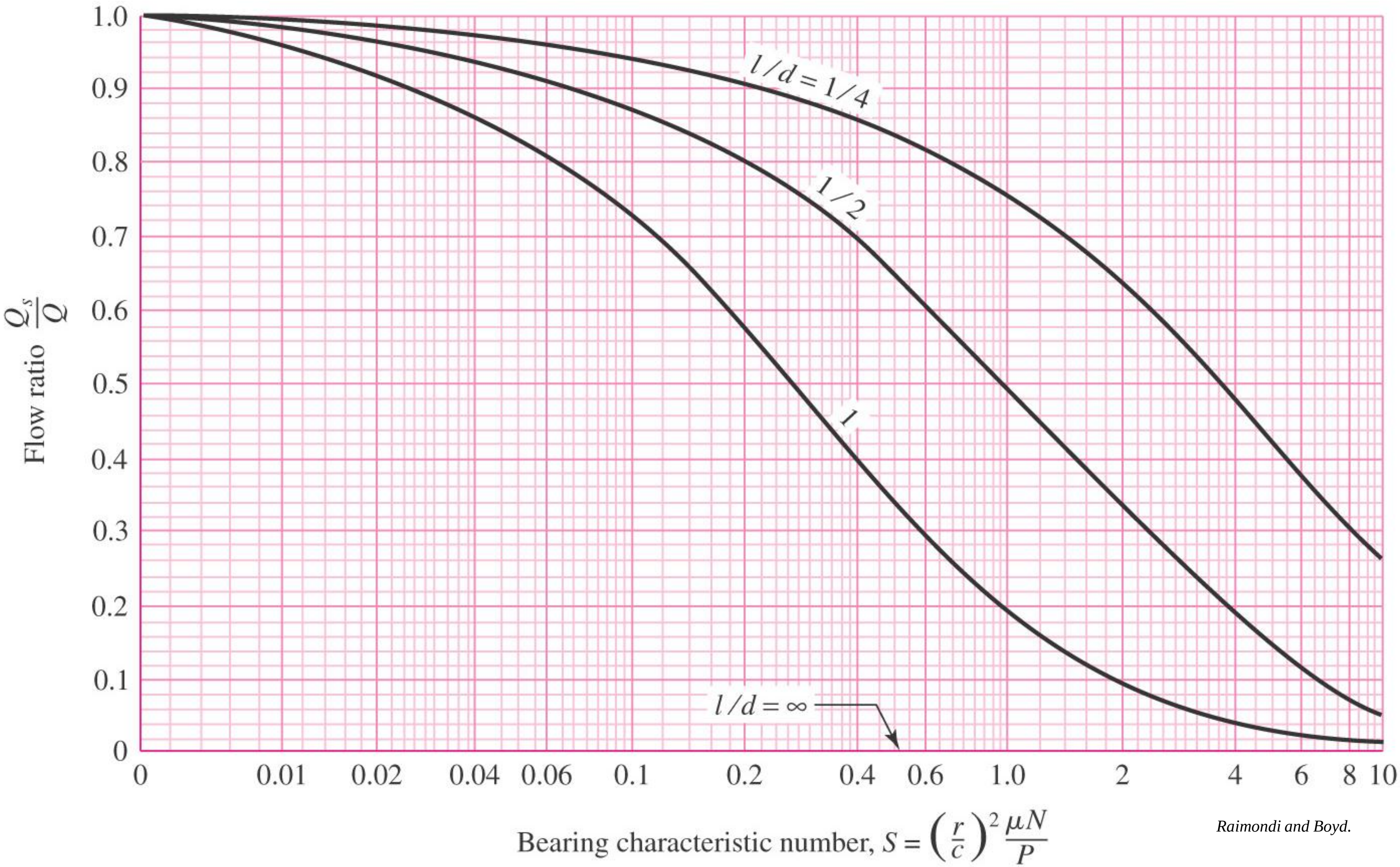
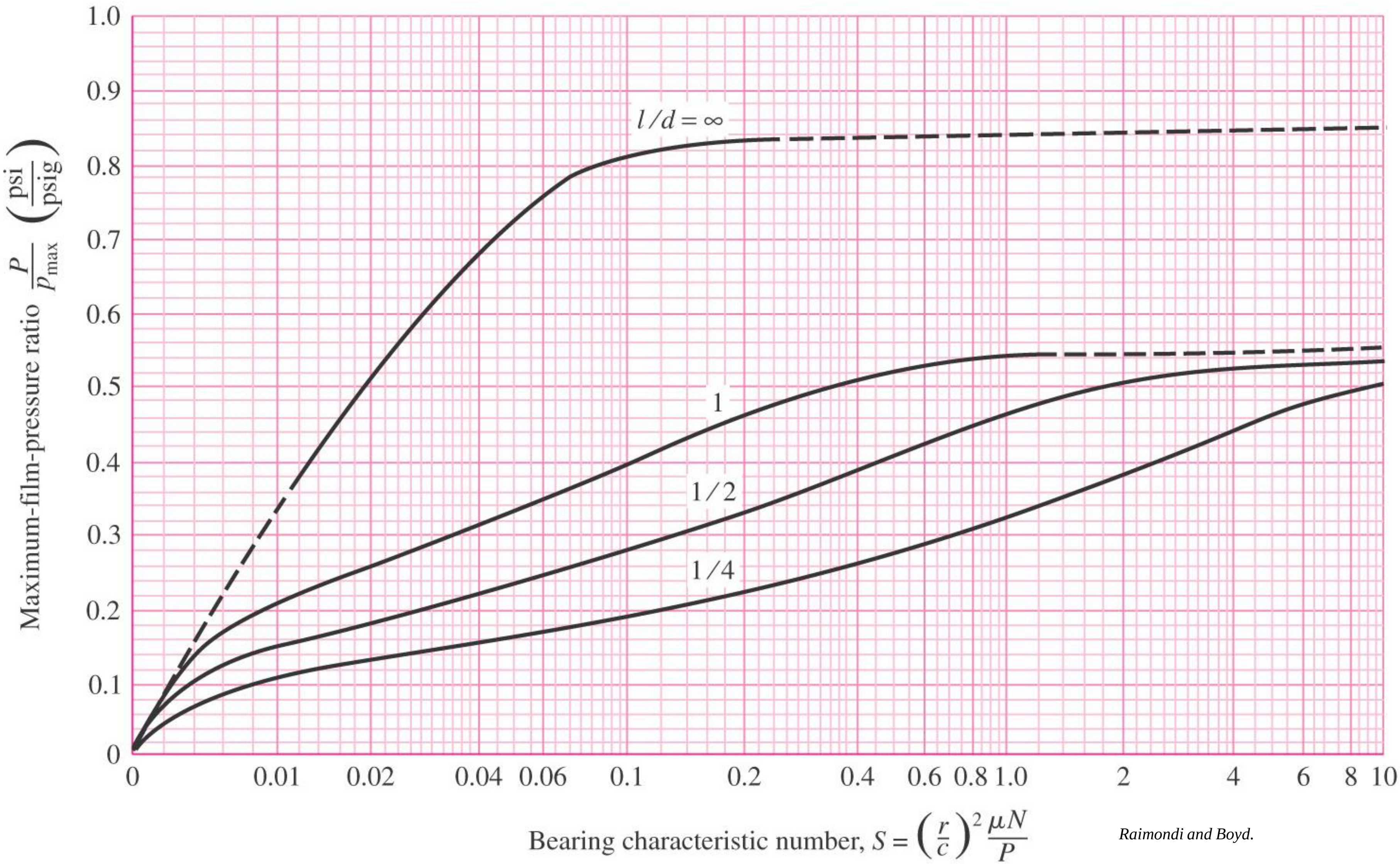


Fig. 12–20

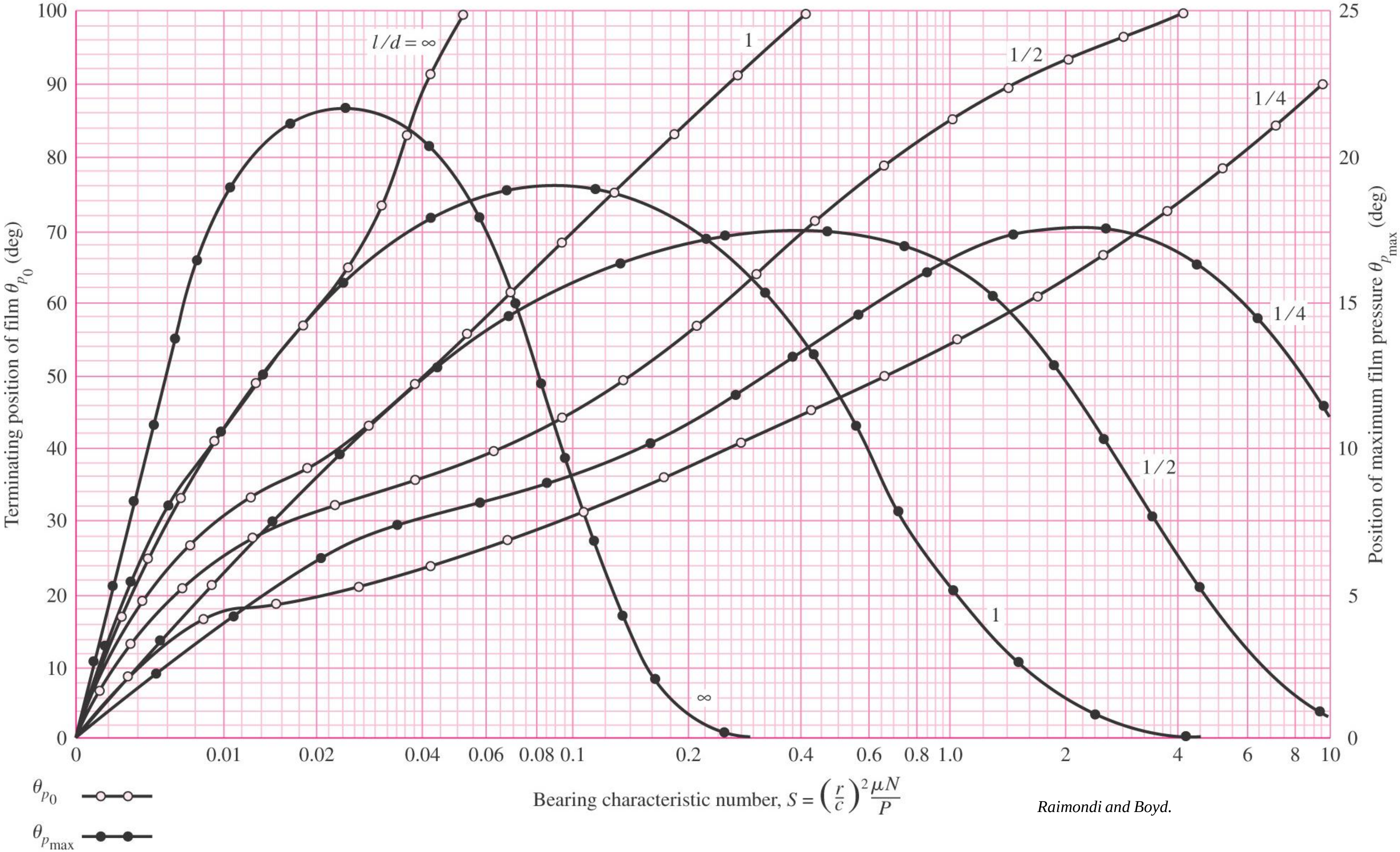
Maximum Film Pressure



Raimondi and Boyd.

Fig. 12–21

Terminating Position of Film



Raimondi and Boyd.

Fig. 12–22

Example 1 (12.1)

A full journal bearing has a journal diameter of 25 mm, with a unilateral tolerance of 0.03 mm.

The bushing bore has a diameter of 25.03 mm and a unilateral tolerance of 0.04 mm. The *l/d* ratio is 1/2. The load is 1.2 kN and the journal runs at 1100 rev/min. If the average viscosity is 55 mPa.s, find:

a. the minimum film thickness,
b. the power loss, and
c. the total flow and the side flow for the minimum clearance assembly
d. Maximum film pressure
e. Terminating position of minimum film

Example 2(12.8)

A journal bearing has a shaft diameter of 75.00 mm with a unilateral tolerance of 0.02 mm.
The bushing bore has a diameter of 75.10 mm with a unilateral tolerance of 0.06 mm. The bushing is 36 mm long and supports a load of 2 kN. The journal speed is 720 rev/min.
For the minimum clearance assembly find:

a. the minimum film thickness,
b. the heat loss rate, and
c. The maximum lubricant pressure

Perform the analyses for SAE 20 and SAE 40 lubricants operating at an average film temperature of 60°C.

Sample Problem

A full journal bearing is 28 mm long. The shaft journal has a diameter of 56 mm with a unilateral tolerance of 0.012 mm. The bushing bore has a diameter of 56.05 mm with a unilateral tolerance of 0.012 mm. The load is 2.4 kN and the journal speed is 900 rev/min.

The operating temperature is 65°C and SAE 40 lubricating oil is used.

For the minimum clearance assembly find:

a. the minimum oil-film thickness,
b. the power loss due to friction
c. Total oil flow rate and the side flow rate
d. Maximum pressure within the fluid
e. Terminal location of the minimum film thickness

Pressure-Fed Bearings

- Temperature rise can be reduced with increased lubricant flow
- *Pressure-fed bearings* increase the lubricant flow with an external pump
- Common practice is to use circumferential groove at center of bearing
- Effectively creates two half-bearings

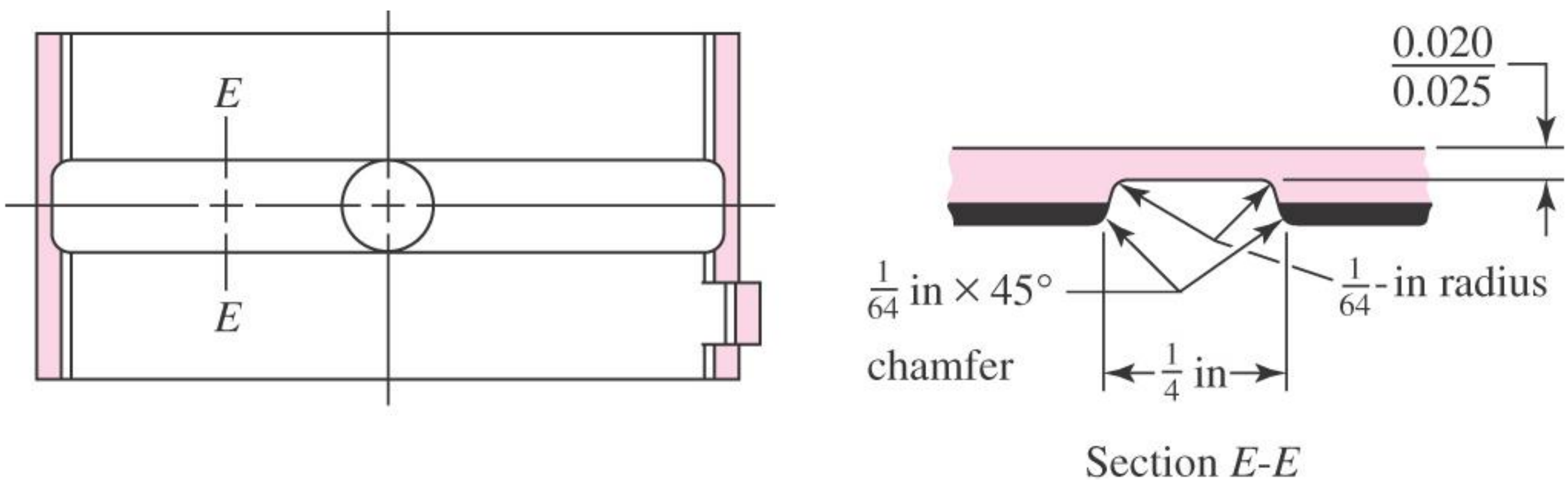


Fig. 12–27

Flow of Lubricant From Central Groove

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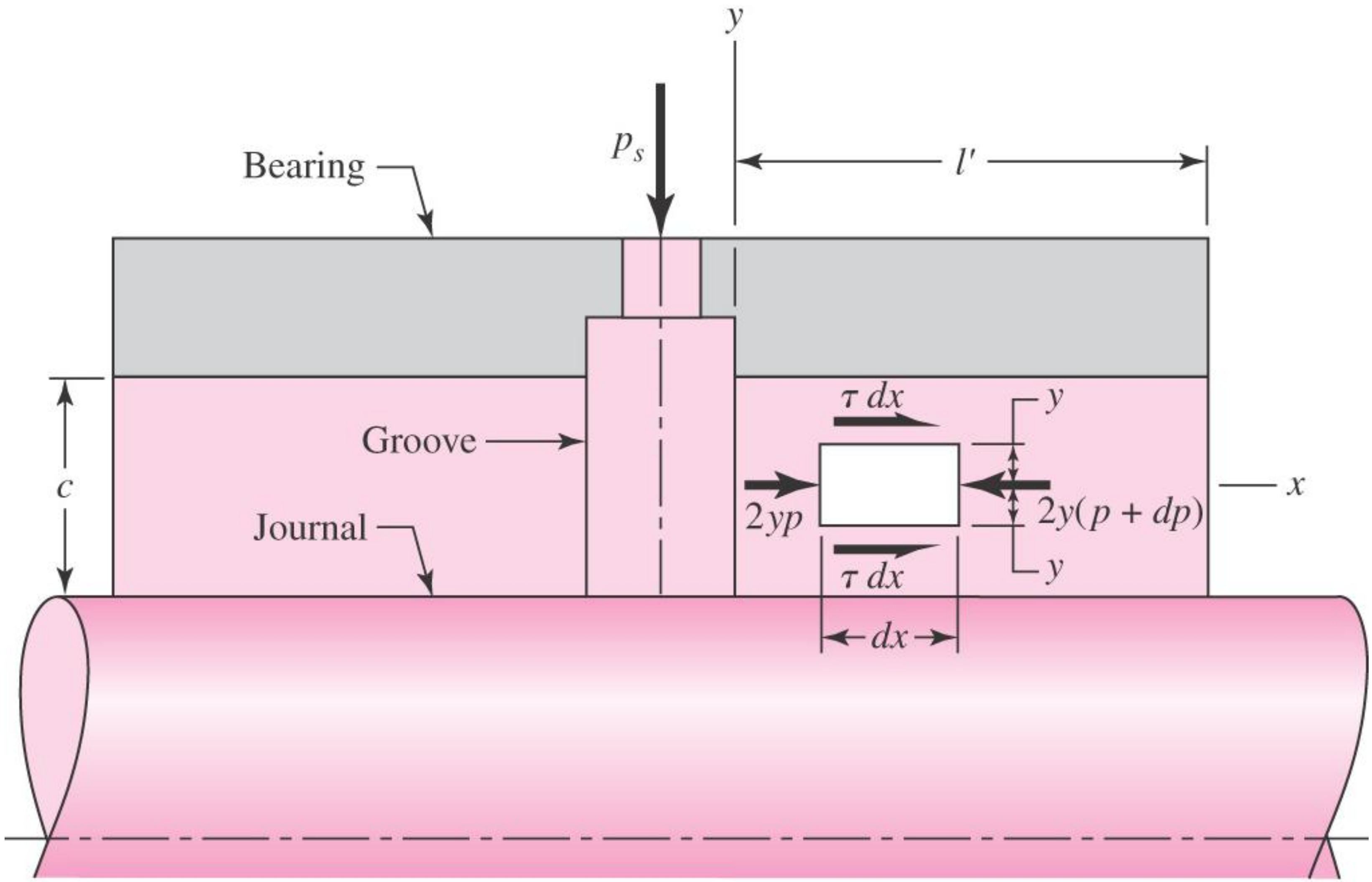


Fig. 12–28

Courtesy of the Cleveland Graphite Bronze Company, Division of Clevite Corporation.

Typical Range of Unit Loads for Sleeve Bearings

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

Some Characteristics of Bearing Alloys

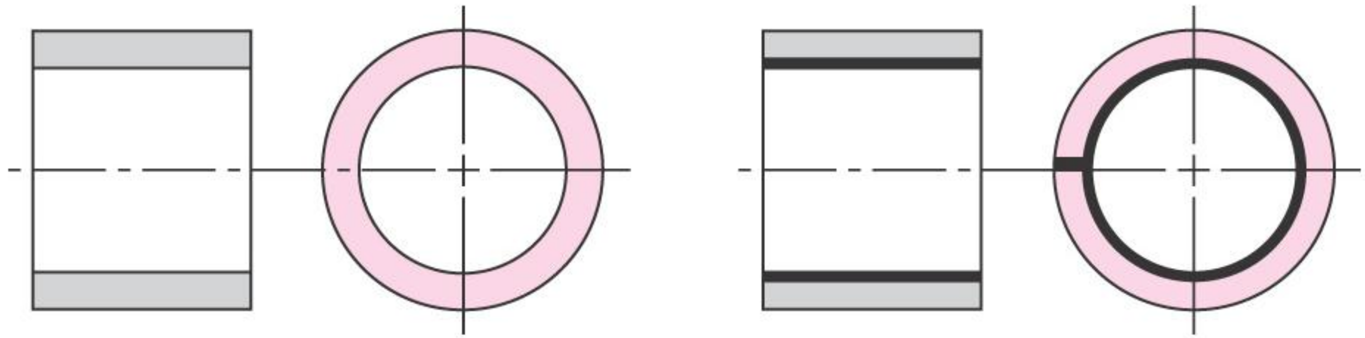
Alloy Name	Thickness, in	SAE Number	Clearance Ratio r/c	Load Capacity	Corrosion Resistance
Tin-base babbitt	0.022	12	600–1000	1.0	Excellent
Lead-base babbitt	0.022	15	600–1000	1.2	Very good
Tin-base babbitt	0.004	12	600–1000	1.5	Excellent
Lead-base babbitt	0.004	15	600–1000	1.5	Very good
Leaded bronze	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.

Table 12–6

Bearing Types



(*a*) Solid bushing Fig. 12–32 (*b*) Lined bushing

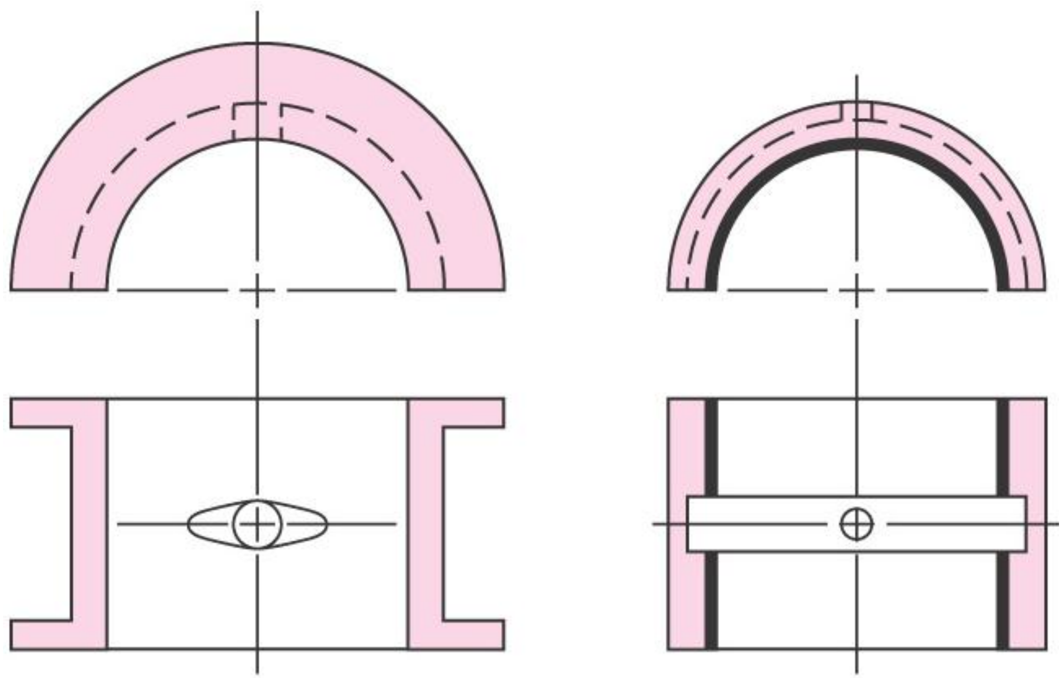


Fig. 12–33

(*a*) Flanged (*b*) Straight

Typical Groove Patterns

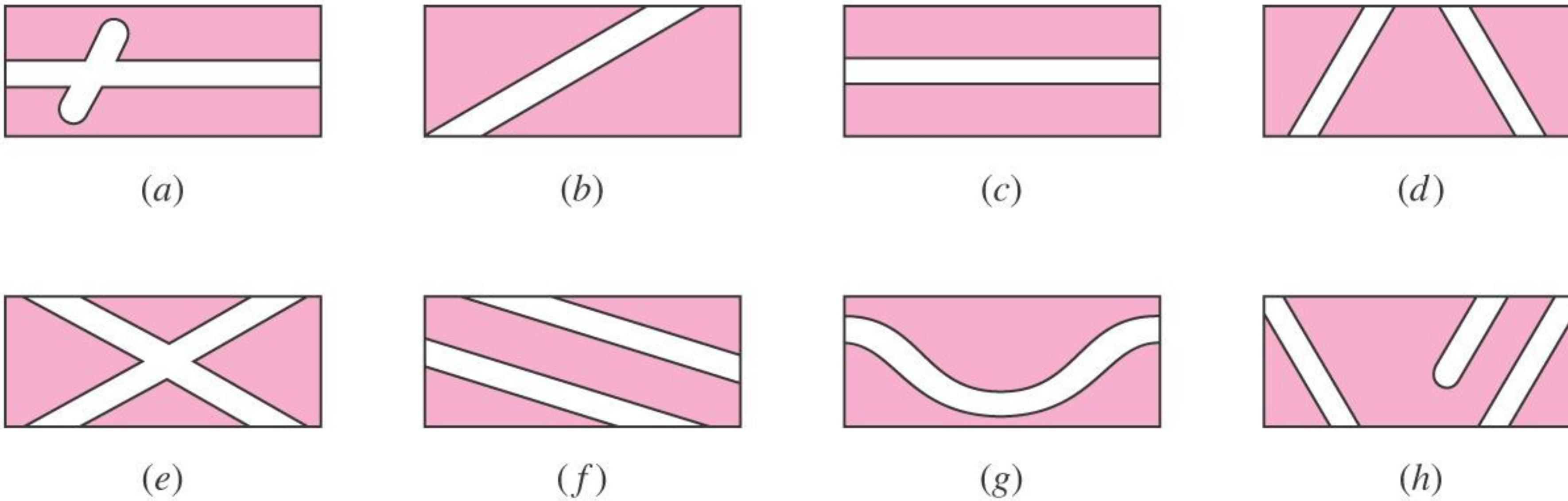


Fig. 12–34

Courtesy of the Cleveland Graphite Bronze Company, Division of Clevite Corporation.

Thrust Bearings

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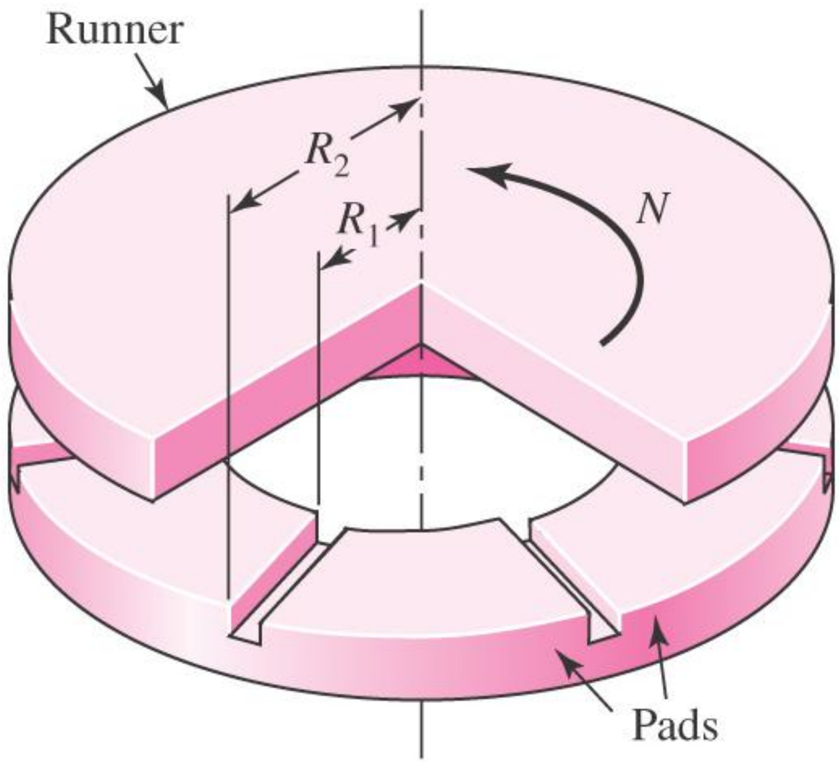

Courtesy of Westinghouse Electric Corporation.

Fig. 12–35

Pressure Distribution in a Thrust Bearing

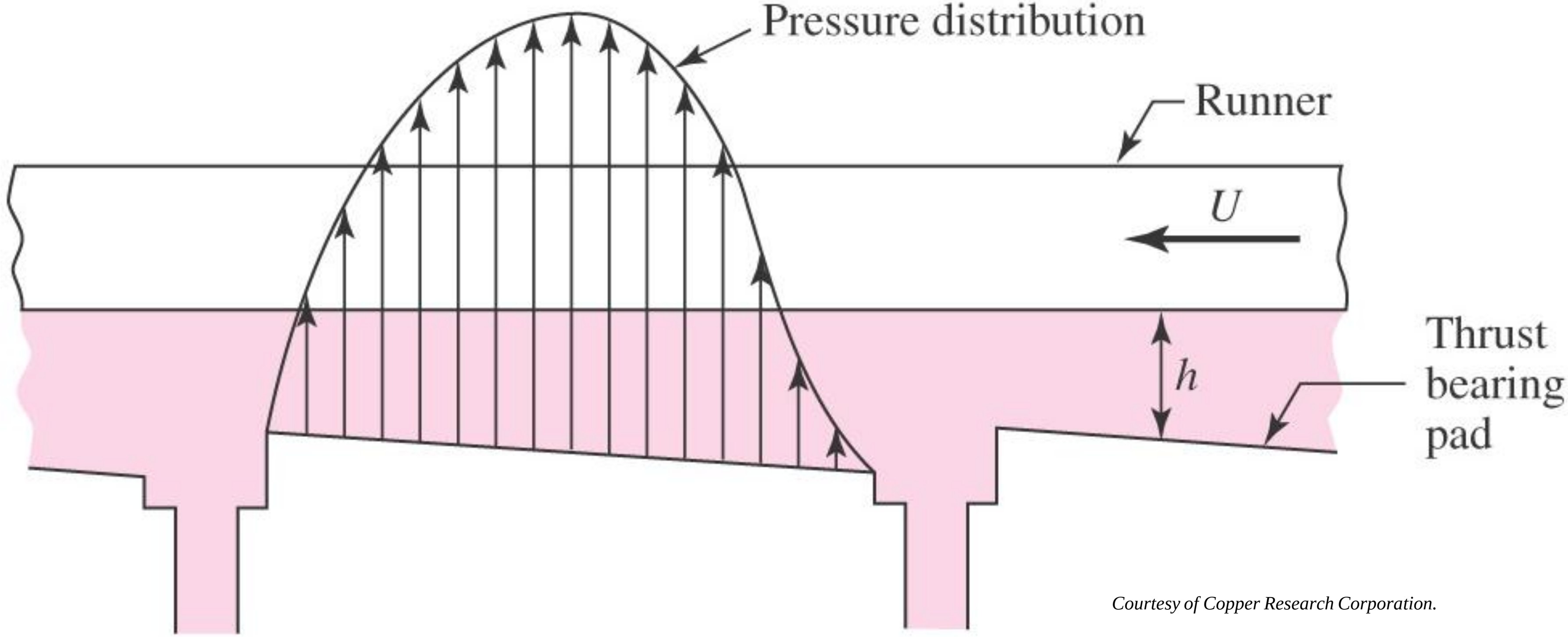

Fig. 12–36

Flanged Sleeve Bearing

- Flanged sleeve bearing can take both radial and thrust loads
- Not hydrodynamically lubricated since clearance space is not wedge-shaped

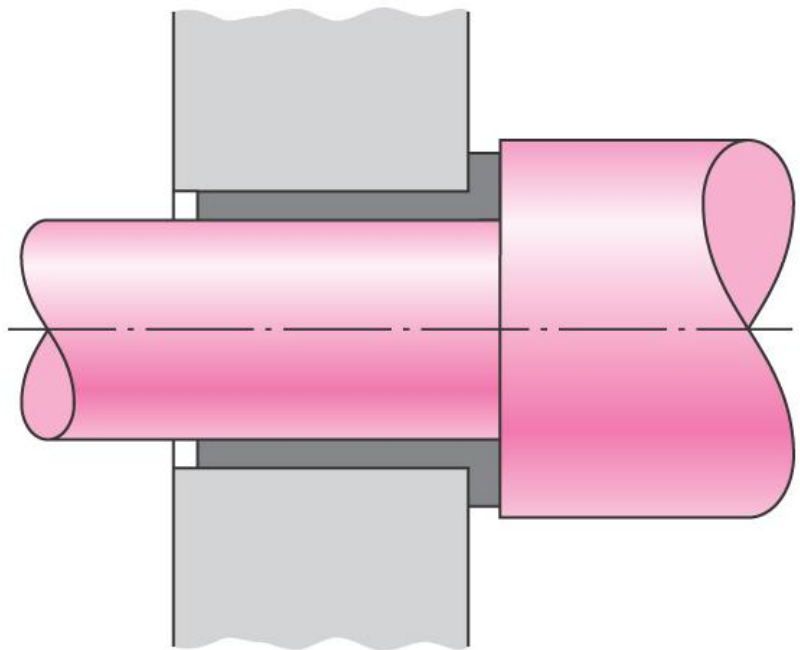

Fig. 12–37

Boundary-Lubricated Bearings

- Relative motion between two surfaces with only a partial lubricant film (not hydrodynamic) is called *boundary lubrication* or *thin-film lubrication.*
- Even hydrodynamic lubrication will have times when it is in thin-film mode, such as at startup.
- Some bearings are boundary lubricated (or dry) at all times.
- Such bearings are much more limited by load, temperature, and speed.

Limits on Some Materials for Boundary-Lubricated Bearings

Material	Maximum Load, psi	Maximum Temperature, °F	Maximum Speed, fpm	Maximum *PV* Value*
Cast bronze	4 500	325	1 500	50 000
Porous bronze	4 500	150	1 500	50 000
Porous iron	8 000	150	800	50 000
Phenolics	6 000	200	2 500	15 000
Nylon	1 000	200	1 000	3 000
Teflon	500	500	100	1 000
Reinforced Teflon	2 500	500	1 000	10 000
Teflon fabric	60 000	500	50	25 000
Delrin	1 000	180	1 000	3 000
Carbon-graphite	600	750	2 500	15 000
Rubber	50	150	4 000	
Wood	2 000	150	2 000	15 000

*P = load, psi; V = speed, fpm.

Table 12–7