Lubrication and Journal Bearing

ME 423: Machine Design
Instructor: Ramesh Singh

Outline

- Types of Lubrication
- Viscosity
- Petroff's Equation
- Thick Film Lubrication
- Hydrodynamic Theory

Introduction

- Lubrication reduces friction, wear and heating of mating parts
- A shaft (or Journal) rotates within a sleeve (or bushing) and the relative motion is sliding.
- Journal bearings are more applicable for extreme operational conditions (high loads and rotational speeds)

from "Tribology Handbook" by P.B. NEALE

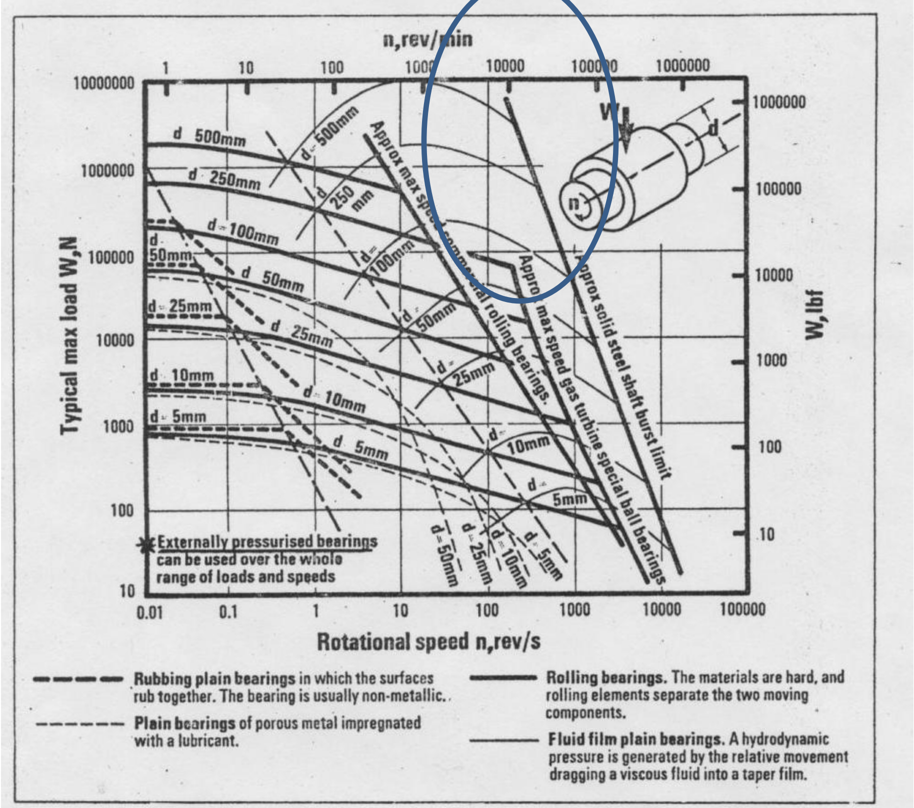


Figure 1. Radial Bearing Selection Guide

Types of Lubrication

- Hydrodynamic
  - The surfaces of the bearing are separated by a relatively thick film of lubricant (to prevent metal to metal contact). The fluid flow due to moving surface is in a converging gap which generates pressure separating the sliding surfaces
- Hydrostatic
  - The lubricant is forced into the bearing at a pressure high enough to separate the surfaces
- Elastohydrodynamic
  - The lubricant is introduced between surfaces that are in rolling contact (such as mating gears or rolling bearings). There is elastic deformation of the surfaces with a fluid film in between the contacting surfaces

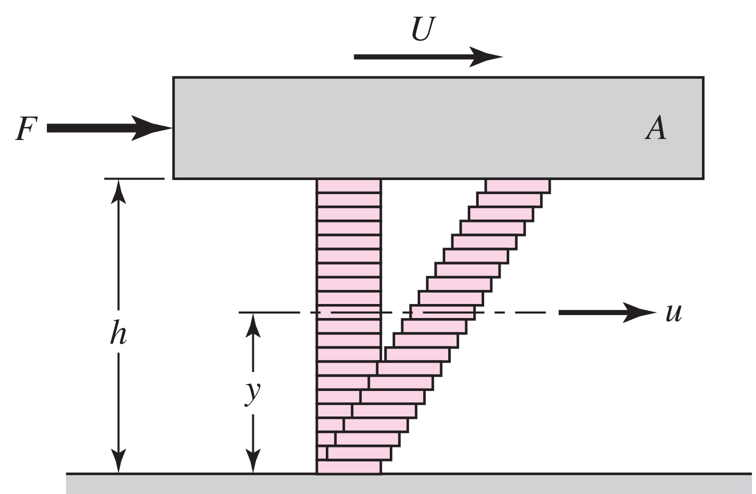
Types of Lubrication

- Boundary
  - There is a very thin film and there is an effect of surface roughness of the contacting surfaces. In mixed lubrication there is asperity to asperity contact and a very thin fluid film.
- Solid-film
  - self-lubricating solid materials such as graphite are used in the bearing. This is used when bearings must operate at very high temperature.

Viscosity

- A fluid film of thickness his sheared between two plates one stationary and other moving at a velocity U, assuming that there is no slip condition, means the fluid and the plate velocities are same. Newton's viscous effect states that the shear stress in the fluid is proportional to the rate of change of velocity with respect to y.

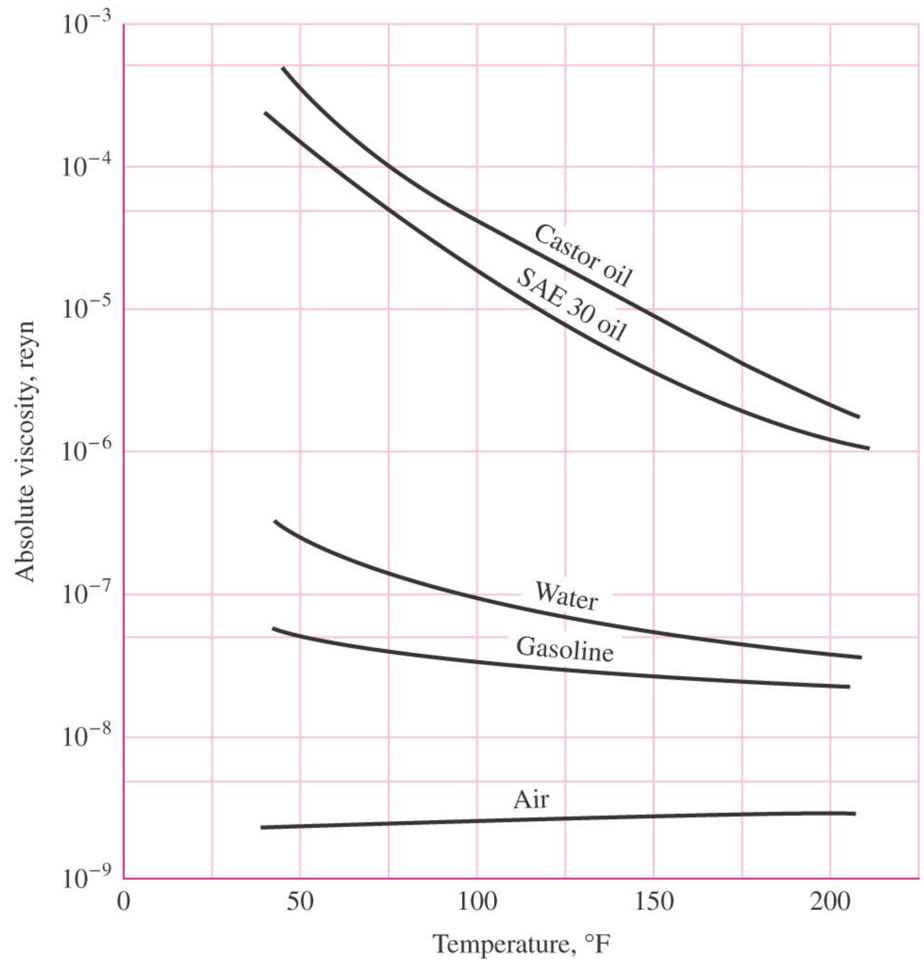
$$\tau = \mu \frac{du}{dy}$$



Viscosity

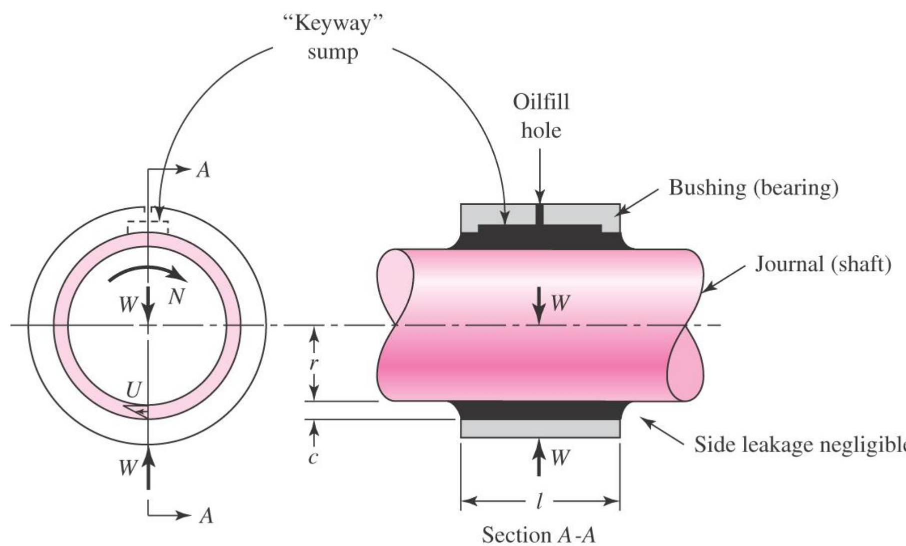
- where μ is the constant of proportionality and defines absolute viscosity, also called dynamic viscosity. The derivative du/dy is the rate of change of velocity with distance and may be called the rate of shear, or the velocity gradient

$$\tau = \frac{F}{A} = \mu \frac{U}{h}$$



Petroff's Equation

- The Petroff equation gives the coefficient of friction in journal bearings.
- It assumes that the shaft (journal) and the bushing are concentric.
- In reality, the shaft is not concentric with the bearing but the coefficient of friction predicted is quite good.



Petroff's Equation

- The shaft is operating at a speed of n
- $\tau = \mu \frac{U}{h} = \frac{\mu 2\pi r N}{c}$
- $T = (\tau A) r = \left(\frac{\mu 2\pi r N}{c}\right)(2\pi r l) r$
- $T = fWr$

Where f is the coefficient of friction

The coefficient of friction $\Rightarrow$ $\boxed{f = 2\pi^2 \frac{\mu N}{P} \frac{r}{c}}$

Pitroff's equation

Petroff's Equation

- The non-dimensional quantities $\left(\frac{\mu N}{P}\right)$ & $\left(\frac{r}{c}\right)$ are important

- The Sommerfeld number or bearing characteristics number contains many important

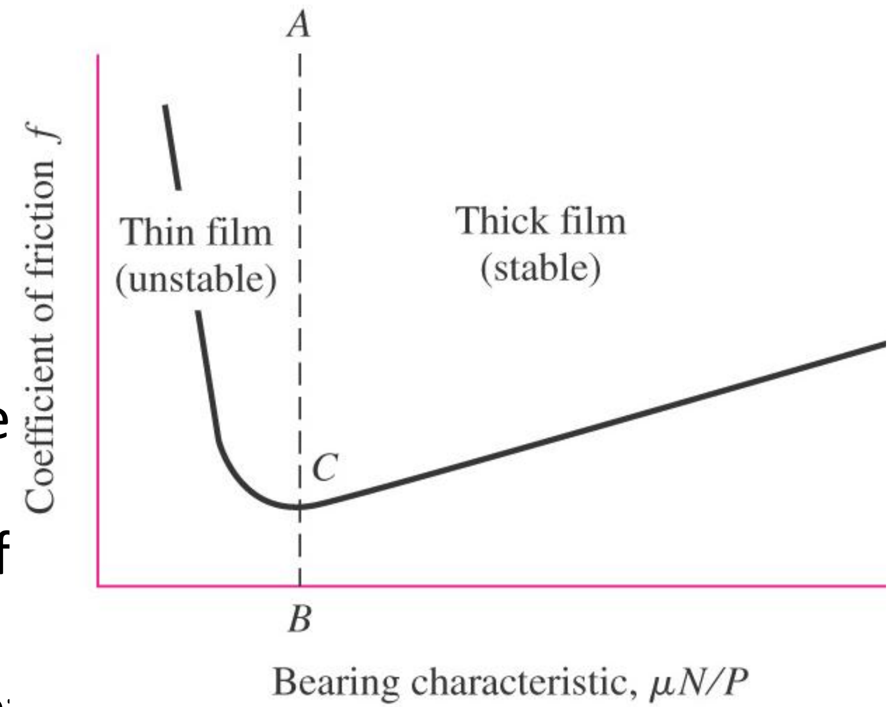
$$\boxed{S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}}$$

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

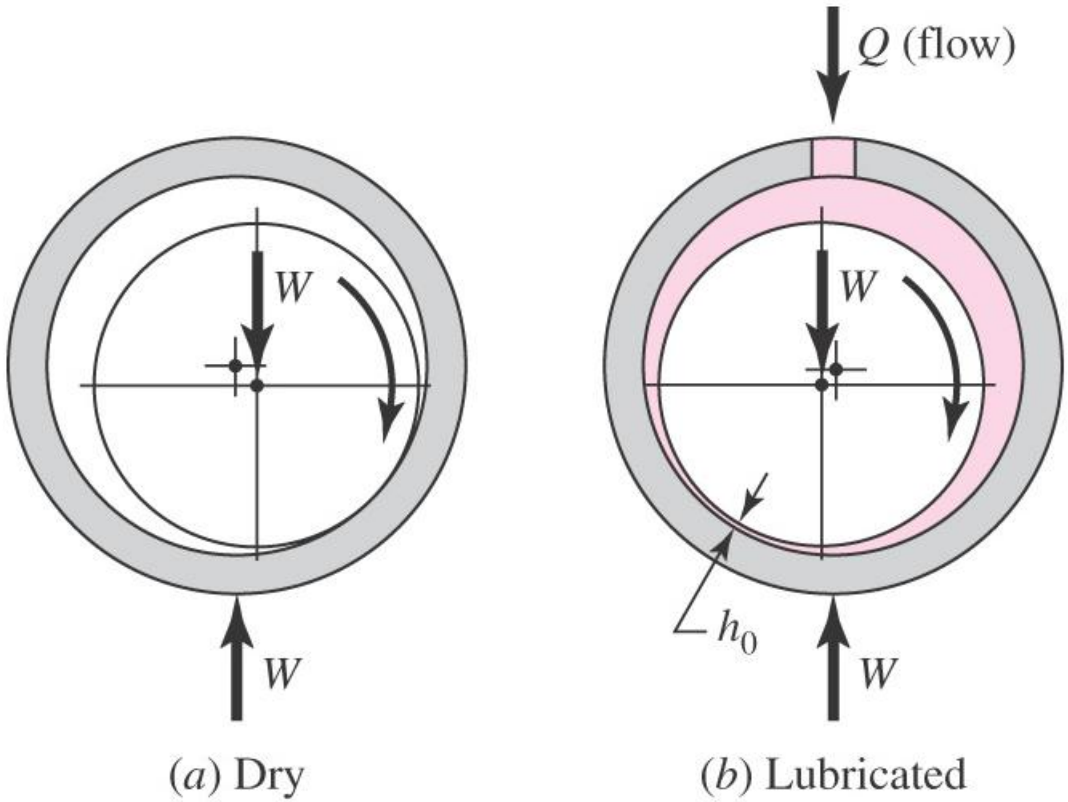
Stable Lubrication

- On he right of AB, change sin 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,

$$\frac{\mu N}{P} \geq 1.7(10^{-6})$$

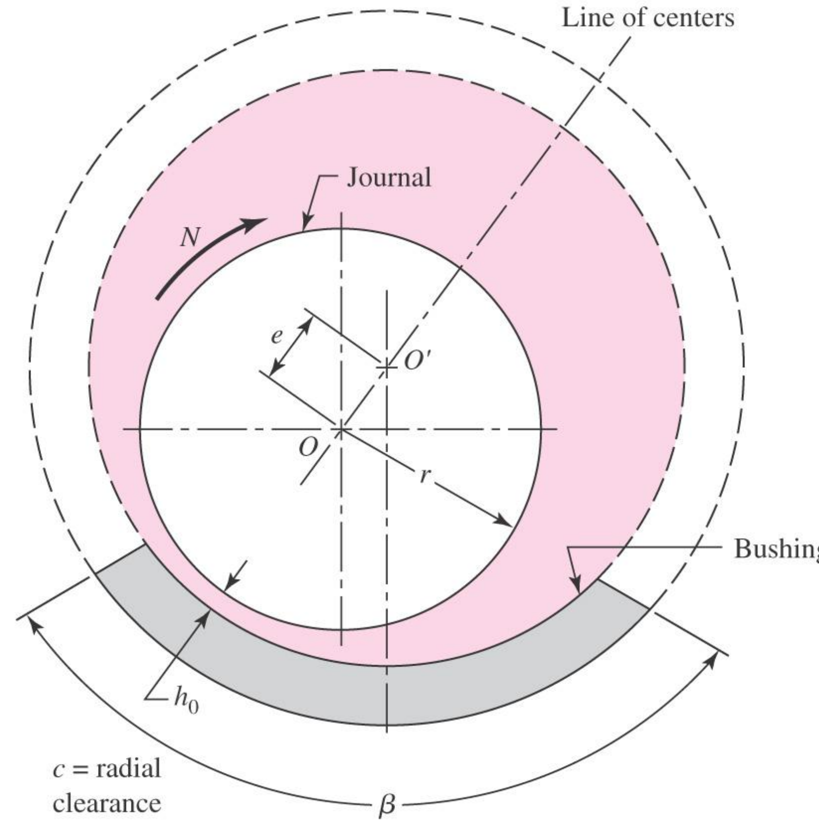


Thick Film Lubrication



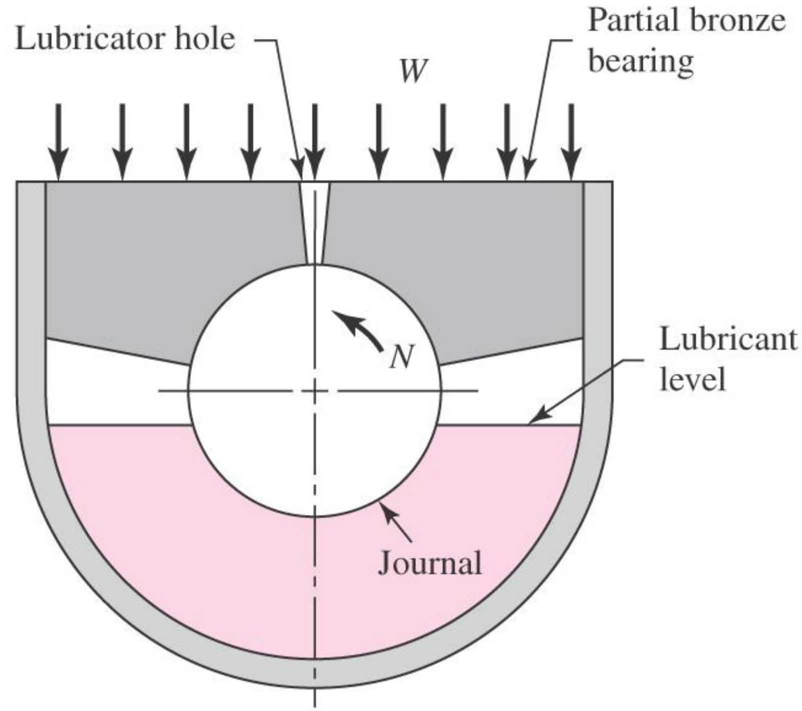
Journal Bearing Nomenclature

- 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 β<360
- Full bearing has β = 360
- Fitted bearing has equal radii of bushing and journal



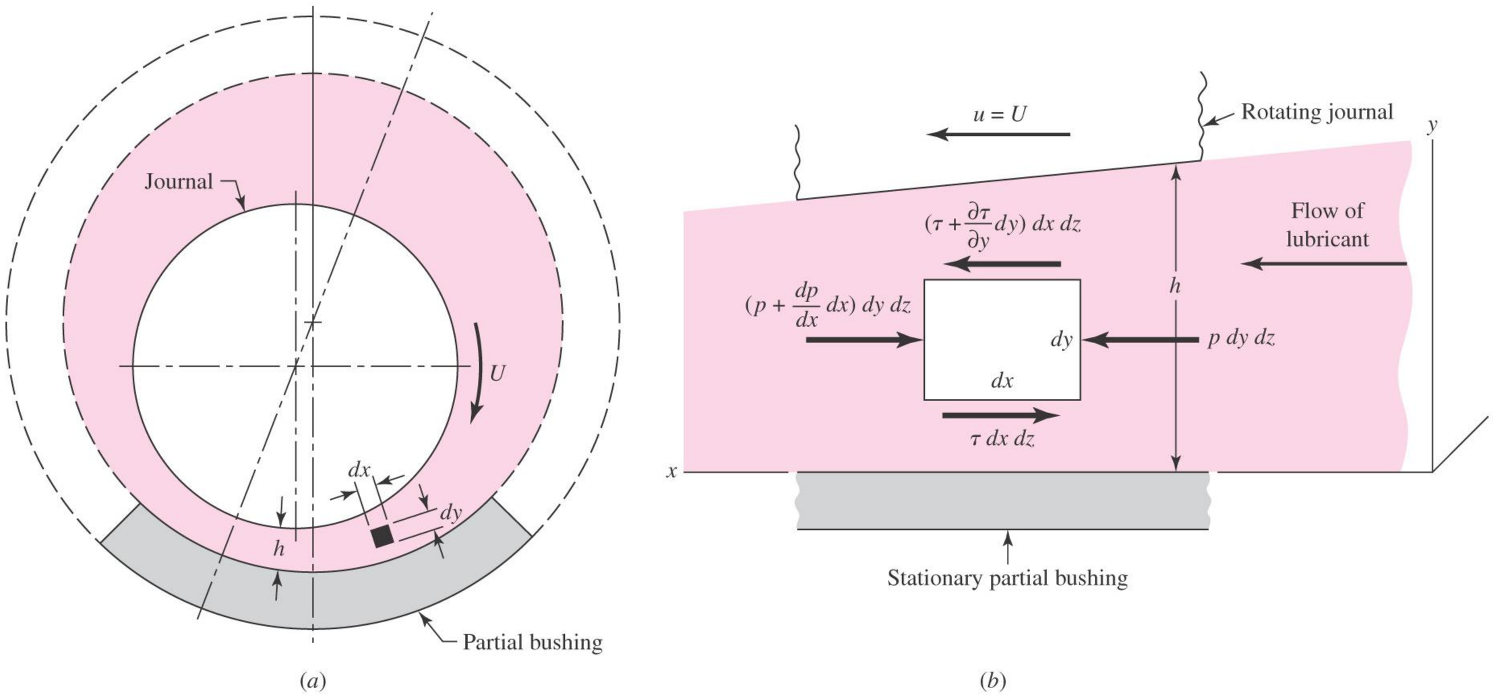
Tower Experiments

- Tower observed very low coefficient for rail road journal bearing in 1880s
- He tried to plug the lubricant hole by cork and wooden plug but they popped out



Reynolds Plane Slider Bearing

- Reynolds realized fluid films were so thin in comparison with bearing radius that curvature could be neglected
- Replaced curved bearing with flat bearing
- Termed plane slider bearing



Reynolds Equation

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

$$\frac{dp}{dx} = \frac{\partial \tau}{\partial y} \quad (b)$$

$$\tau = \mu \frac{\partial u}{\partial y} \quad (c)$$

$$\frac{dp}{dx} = \mu \frac{\partial^2 u}{\partial y^2} \quad (d)$$

$$\frac{\partial u}{\partial y} = \frac{1}{\mu}\frac{dp}{dx}y + C_1$$

$$(e)$$

$$u = \frac{1}{2\mu}\frac{dp}{dx}y^2 + C_1 y + C_2$$

Velocity Distribution

$$\text{At} \quad y = 0, \; u = 0$$

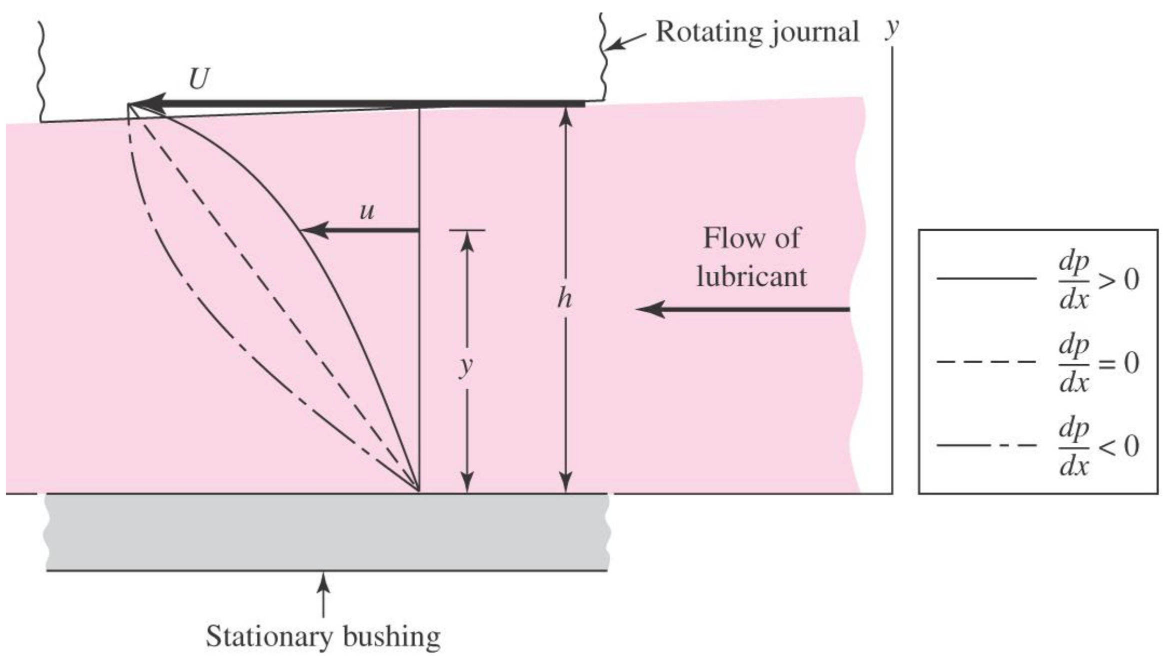
$$\text{At} \quad y = h, \; u = U$$

$$C_1 = \frac{U}{h} - \frac{h}{2\mu}\frac{dp}{dx} \qquad C_2 = 0$$

$$u = \frac{1}{2\mu}\frac{dp}{dx}(y^2 - hy) + \frac{U}{h}y$$

Velocity Profile

- Velocity distribution superposes parabolic distribution onto linear distribution
- When pressure is maximum, dp/dx = 0 and u = (U/h) y



Reynold's Equation

$$Q = \int_0^h u \, dy$$

$$Q = \frac{Uh}{2} - \frac{h^3}{12\mu} \frac{dp}{dx}$$

$$\frac{dQ}{dx} = 0$$

$$\frac{dQ}{dx} = \frac{U}{2} \frac{dh}{dx} - \frac{d}{dx}\left(\frac{h^3}{12\mu} \frac{dp}{dx}\right) = 0$$

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

Reynold's Solution

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

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

- With side leakage included,

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

Reynold's Eqn

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

$$\frac{r}{c}f = \emptyset\left[\underbrace{\left(\frac{r}{c}\right)^2 \frac{\mu N}{P}}_{\textit{Sommerfeld number}}\right]$$

$$\emptyset := functional\, relationship$$

Design Considerations

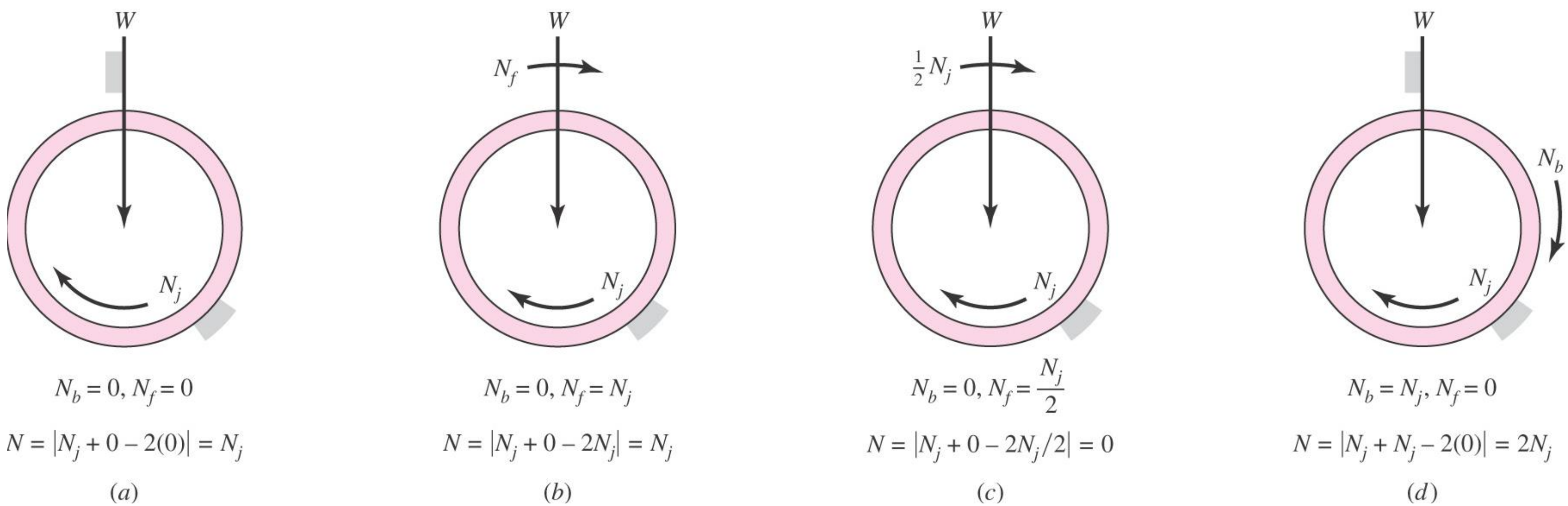
Variables either given or under control of designer

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

Significant Angular Speed



$$\boxed{N = \left|N_j + N_b - 2N_f\right|}$$

where :

N_j: journal angular speed *(rev/s)*

N_b: bearing (bushing) angular speed *(rev/s)*

N_f: load vector angular speed *(rev/s)*

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

$$\boxed{h_0 \geq 0.00508 + 0.00004d} \quad mm$$

where d is the journal diameter in "*mm*".

- Maximum temperature to prevent vaporization of lighter lubricant components

$$\boxed{T_{max} \leq 121} \ °C$$

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

s:

$$\boxed{\frac{W_{st}}{ld} \leq 2068} \quad kPa$$

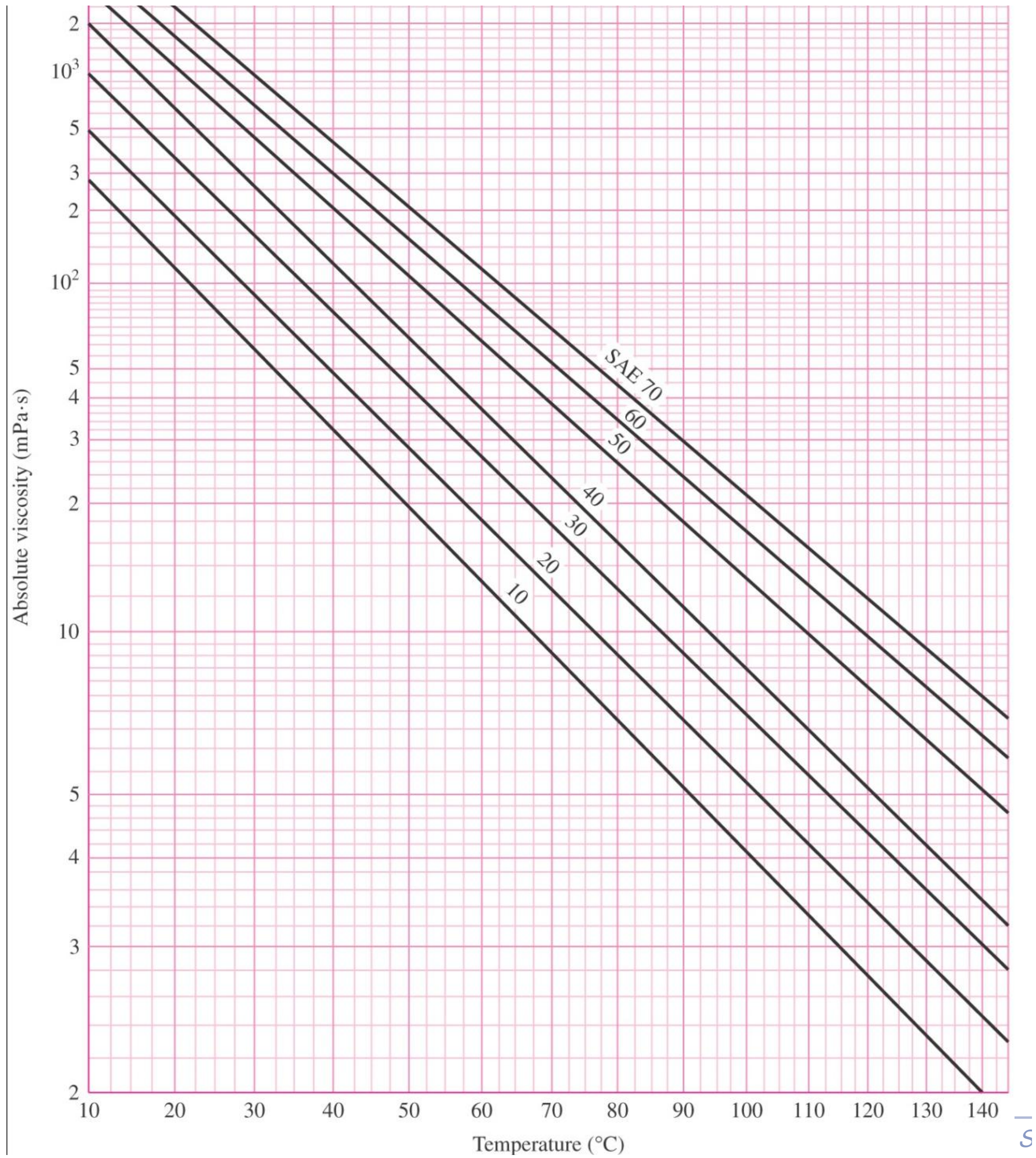
Note that starting load is usually <u>smaller</u> than running load

Viscosity

- Viscosity is clearly a function of temperature
- 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

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

Viscosity Charts



Notation of Raimondi and Boyd

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

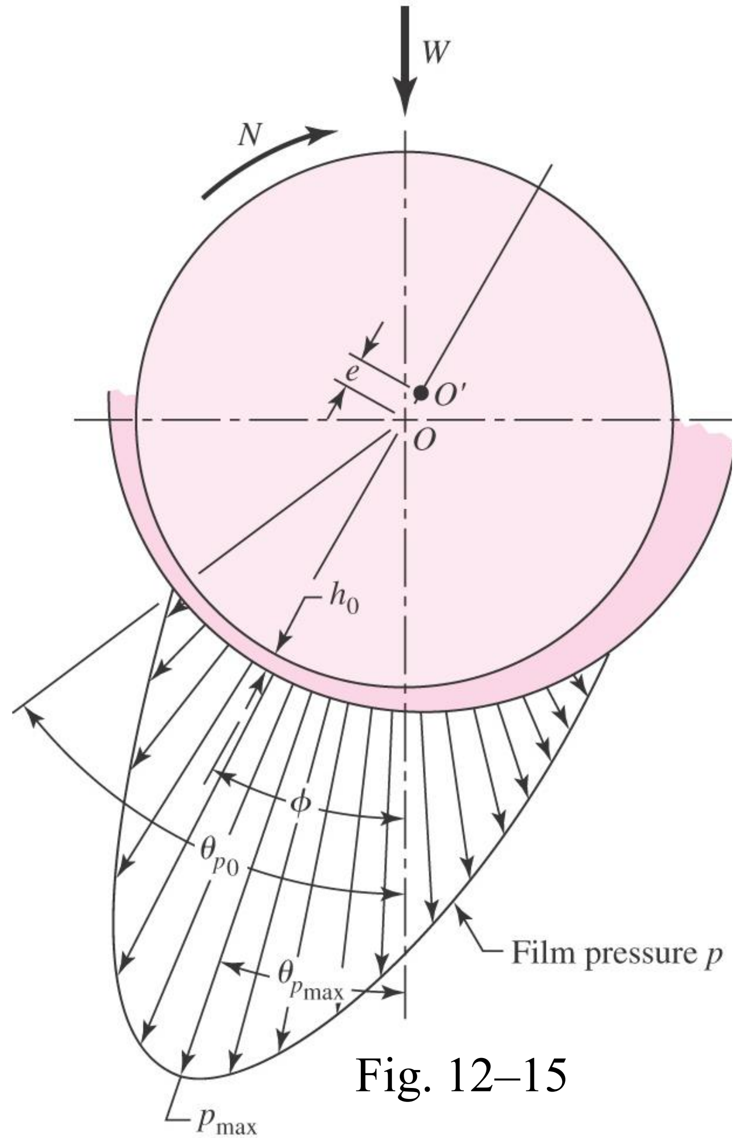
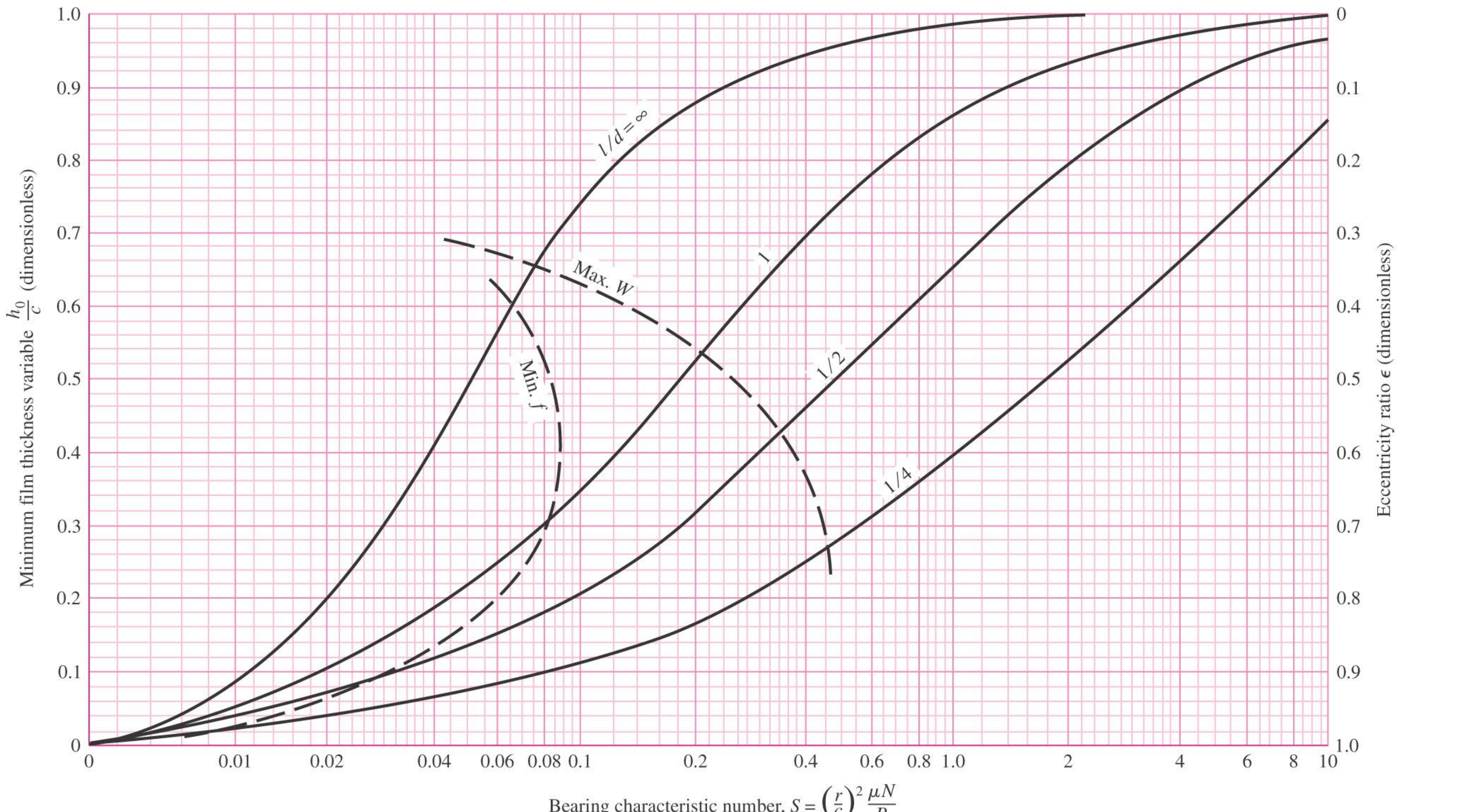
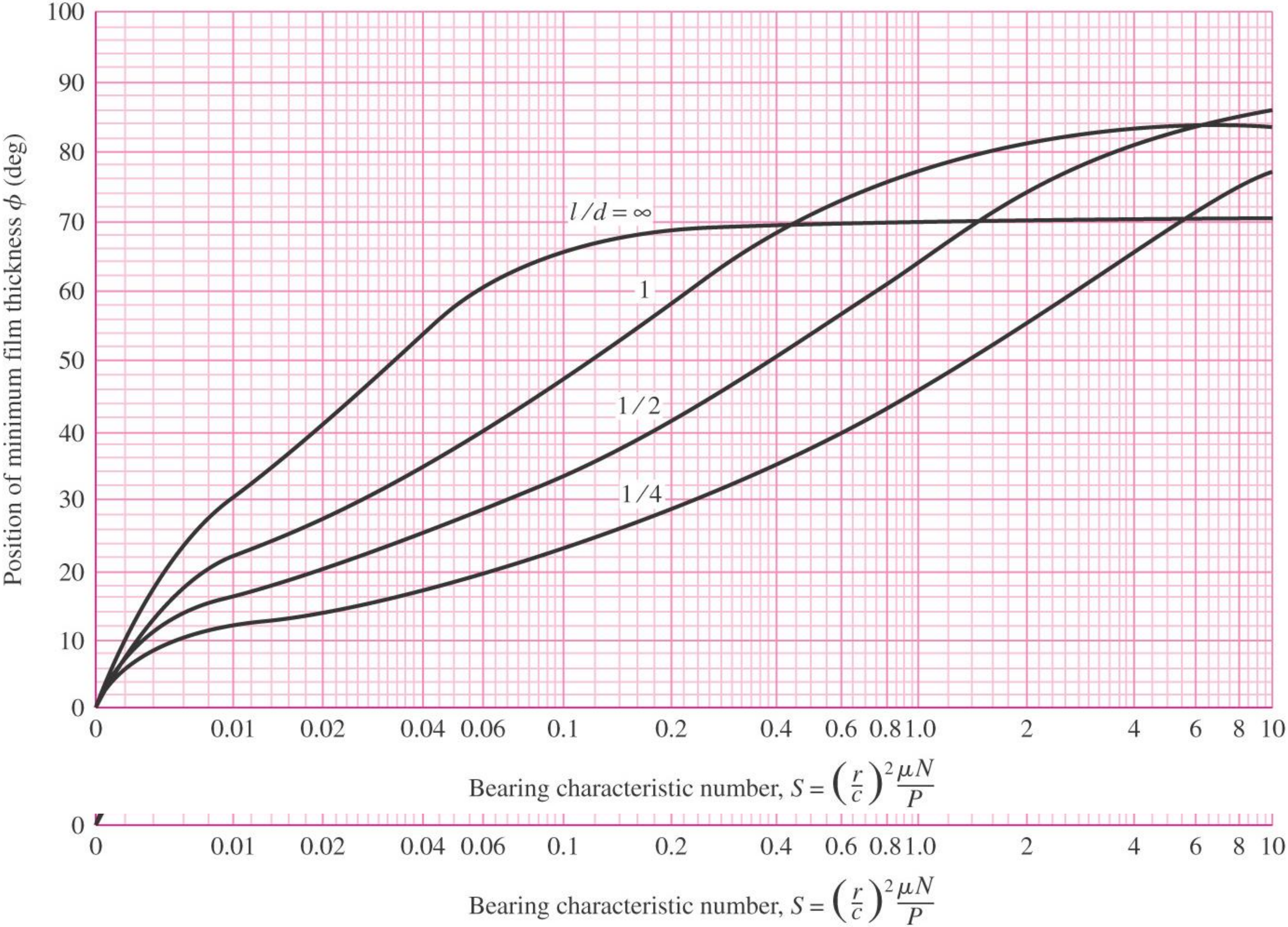


Fig. 12–15

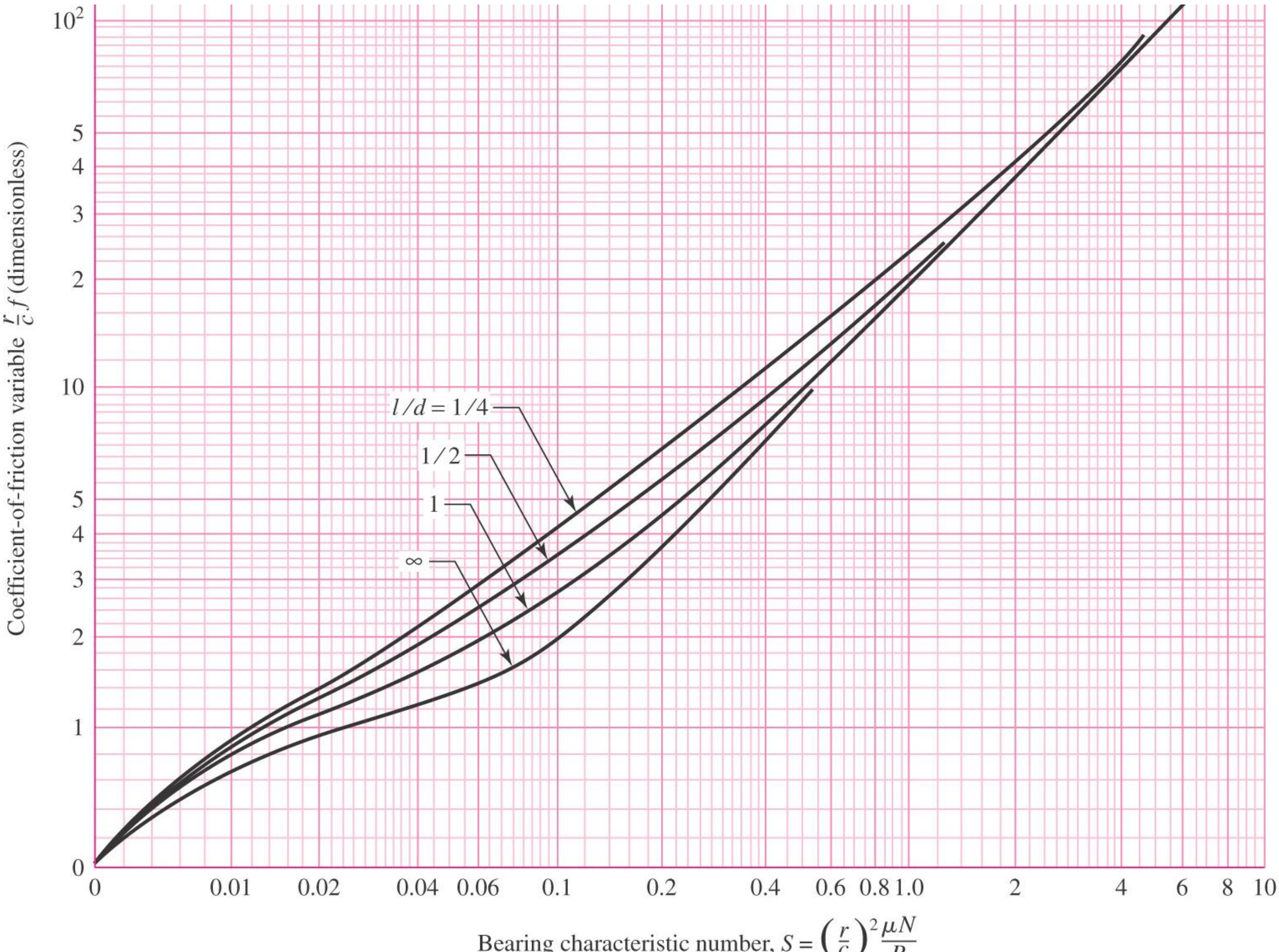
Minimum Film Thickness and Eccentricity Ratio



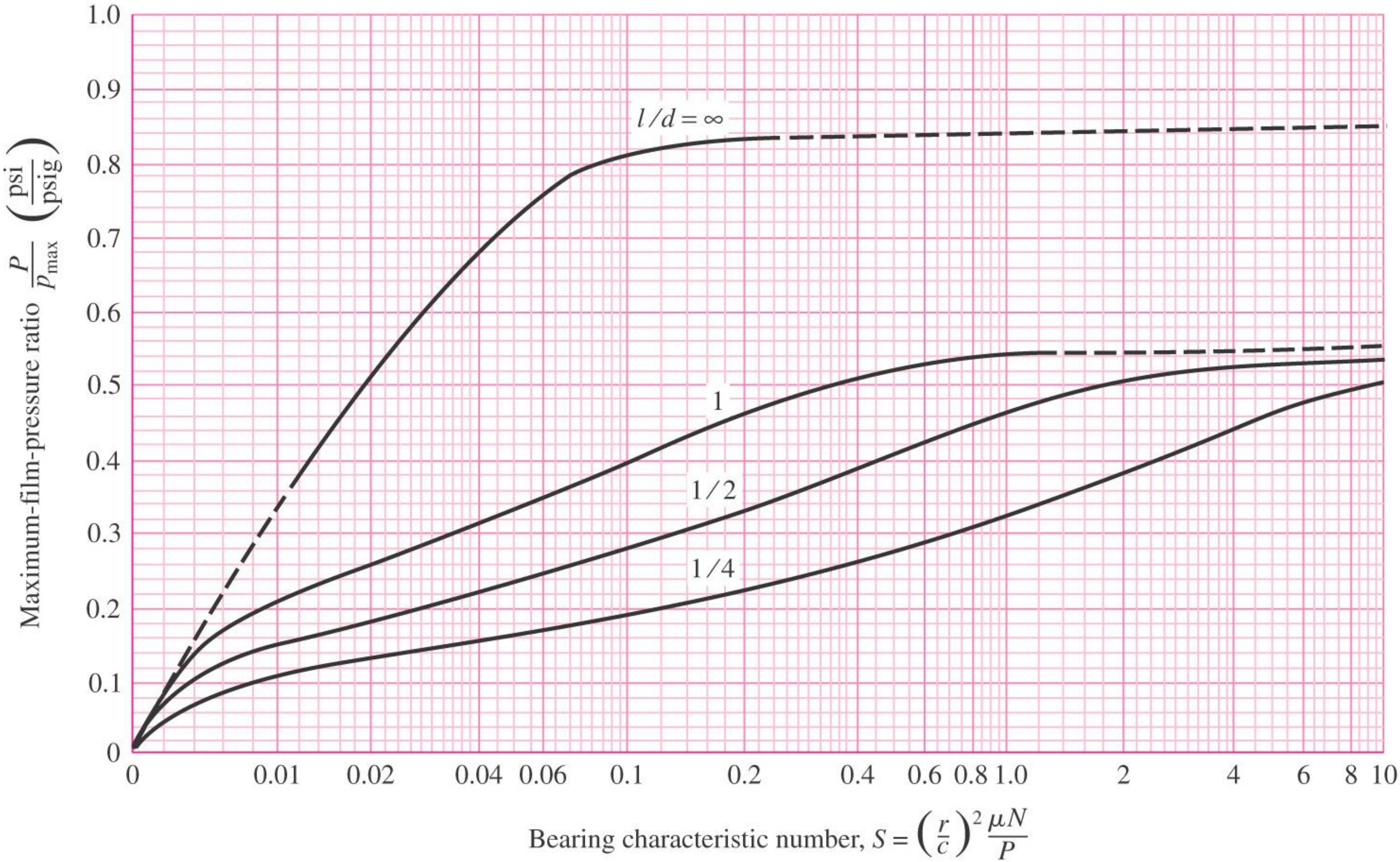


Position of minimum film thickness ϕ (deg)
100
90
80
70
60
50
40
30
20
10
0
$l/d = \infty$
1
1/2
1/4
0
0.01
0.02
0.04
0.06
0.1
0.2
0.4
0.6
0.8
1.0
2
4
6
8
10
Bearing characteristic number, $S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$

Coefficient of Friction



Maximum Film Pressure



Example Problem#1

Determine h_0 and e using the following given parameters: $\mu = 4$ μreyn, $N = 30$ rev/s, $W = 500$ lbf (bearing load), $r = 0.75$ in, $c = 0.0015$ in, and $l = 1.5$ in.

Solution

The nominal bearing pressure (in projected area of the journal) is

$$P = \frac{W}{2rl} = \frac{500}{2(0.75)1.5} = 222 \text{ psi}$$

The Sommerfeld number is, from Eq. (12–7), where $N = N_j = 30$ rev/s,

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

Example Problem#1

Also, $l/d = 1.50/[2(0.75)] = 1$. Entering Fig. 12–16 with $S = 0.135$ and $l/d = 1$ gives $h_0/c = 0.42$ and $\epsilon = 0.58$. The quantity h_0/c is called the *minimum film thickness variable*. Since $c = 0.0015$ in, the minimum film thickness h_0 is

$$h_0 = 0.42(0.0015) = 0.000\,63 \text{ in}$$

We can find the angular location ϕ of the minimum film thickness from the chart of Fig. 12–17. Entering with $S = 0.135$ and $l/d = 1$ gives $\phi = 53°$.

The eccentricity ratio is $\epsilon = e/c = 0.58$. This means the eccentricity e is

$$e = 0.58(0.0015) = 0.000\,87 \text{ in}$$

Example #2

Using the parameters given in Ex. 12–1, determine the coefficient of friction, the torque to overcome friction, and the power loss to friction.

We enter Fig. 12–18 with $S = 0.135$ and $l/d = 1$ and find $(r/c)f = 3.50$. The coefficient of friction f is

$$f = 3.50\, c/r = 3.50(0.0015/0.75) = 0.0070$$

The friction torque on the journal is

$$T = fWr = 0.007(500)0.75 = 2.62 \text{ lbf} \cdot \text{in}$$

The power loss in horsepower is

$$(hp)_{\text{loss}} = \frac{TN}{1050} = \frac{2.62(30)}{1050} = 0.075 \text{ hp}$$

or, expressed in Btu/s,

Example # 3

Using the parameters given in Ex. 12–1, determine the maximum film pressure and the locations of the maximum and terminating pressures.

Solution

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

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

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