Sliding (Journal) bearings:

- According to the direction of the applied load, sliding bearings are dived into:

A- Radial bearings(Journal bearings):

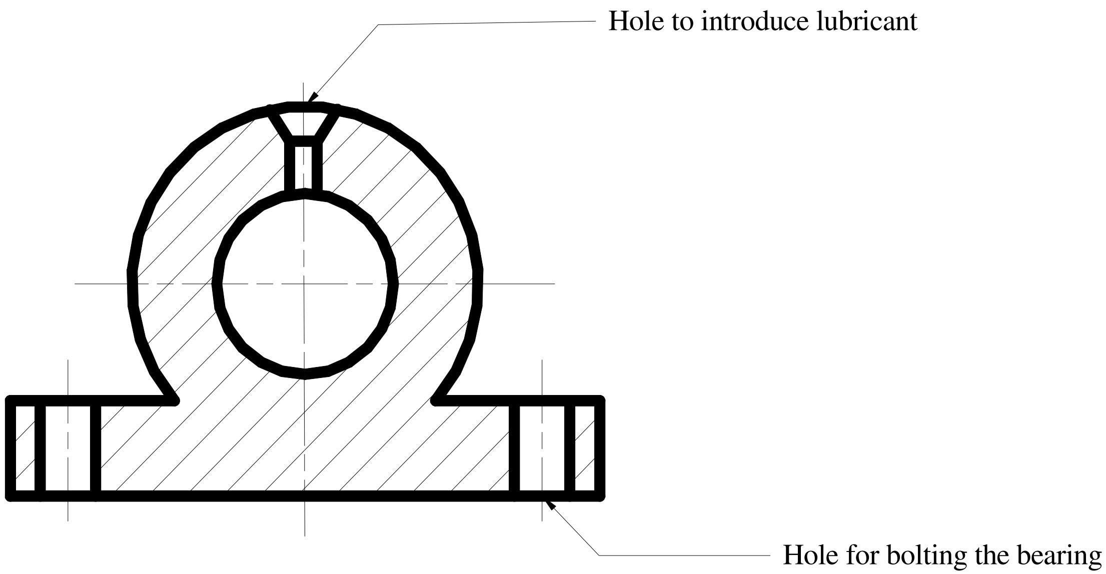
- The journal bearings are used to support only the normal or radial loads (loads acting perpendicular to the shaft axis).
- The journal bearings rotates inside a stationary bush or sleeve. The journal is that part of the shaft which is in contact with the bearing.

B- Thrust bearings:

- Thrust bearings are used where loads acting along shaft axis are to be supported.

Radial bearings (Journal bearings):

1- Solid bearing:



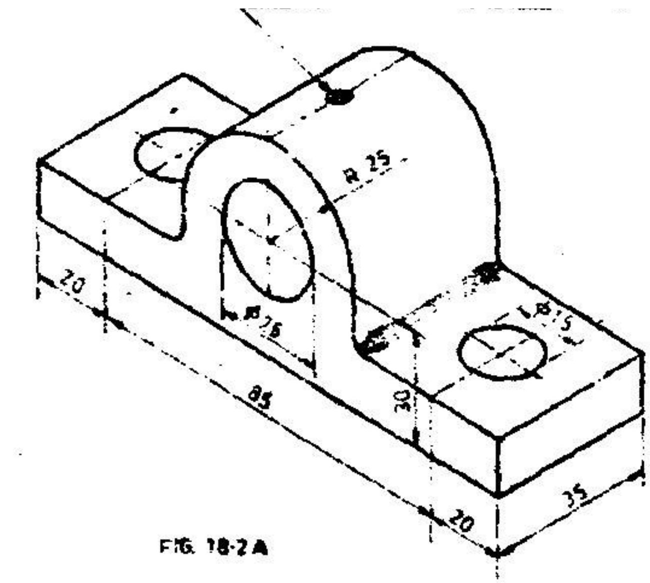
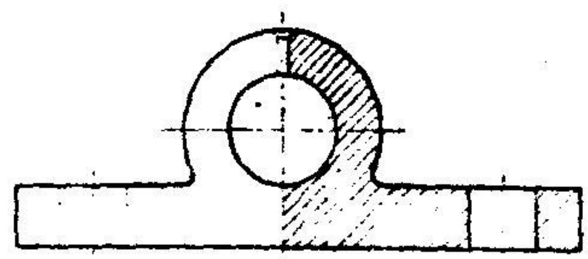


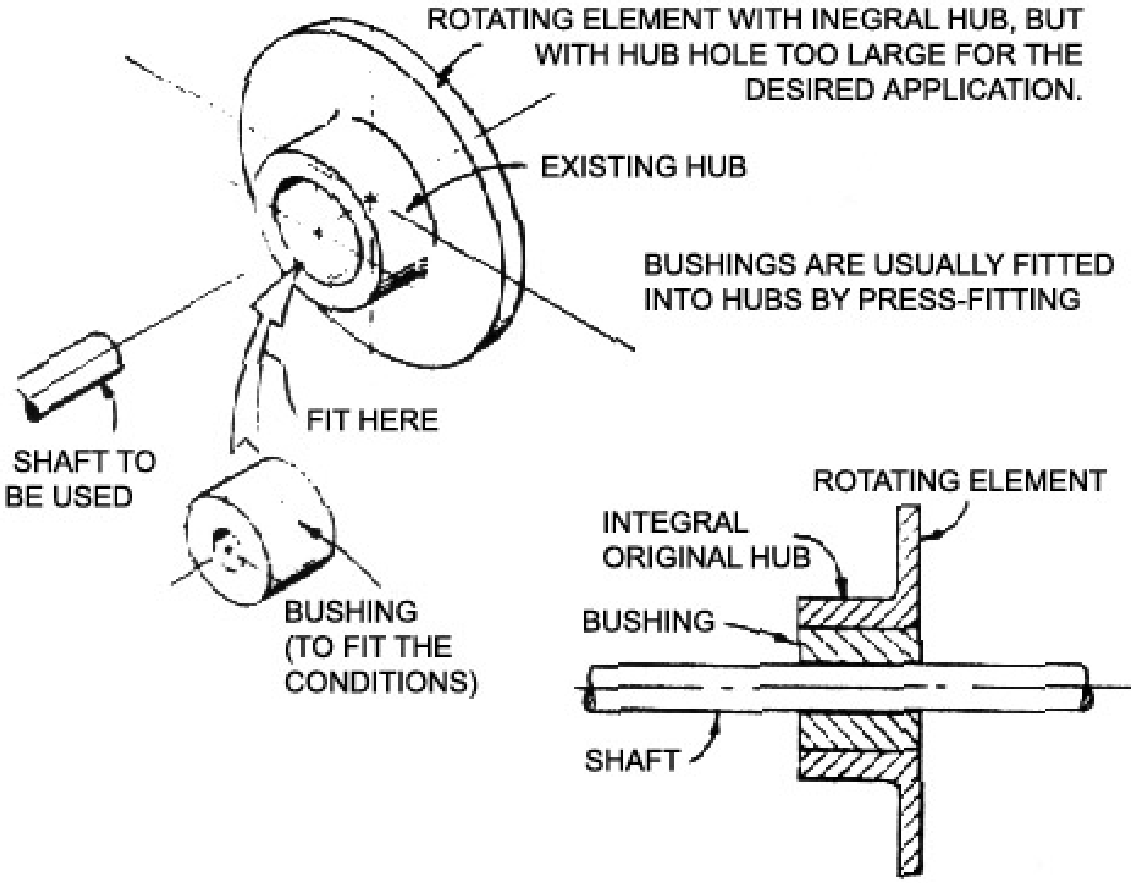
FIG. 18·2A



- Is the simplest form of the journal bearings.
- This is usually made of C.I.
- As the name implies, this is consists of one block in which a hole is bored to receive the journal.

- The rectangular base of the bearings has two holes which are used for bolting down the bearing.
- A hole provided at the top is used to introduce lubricant into the bearing.
- This type of bearing is used for light duty service only. (low speeds, low loads).
- The drawback of this bearing is that it has to be discarded once the inner surface of the bearing gets worn-out as there is no provision for adjustment for wear.

2- Bushed bearing:



Bronze Bearings

- Bushed bearings consist mainly of two parts, the body and the bush.
- The body or the main block is made of cast iron.
- The bush being usually made of soft materials like brass, bronze, undergoes wear and can be periodically replaced.

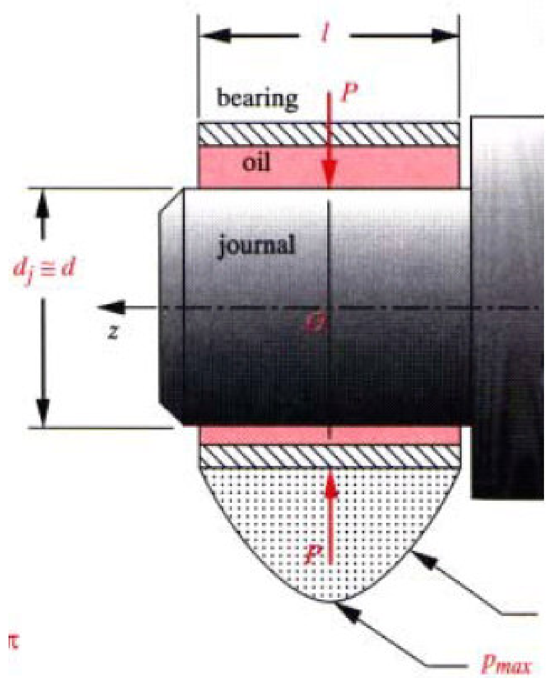
- The bush can be fixed either by:

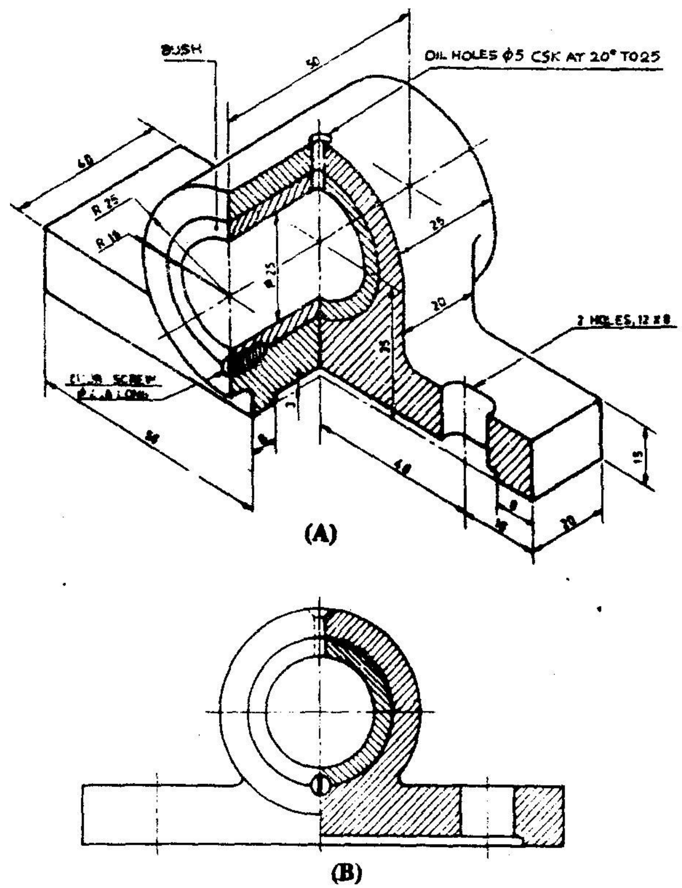
a- Pressing fit into place (H7,H8/n6.p6,r6,s6).

b- Set screw, which is fitted half in the main block and half in the bush and may help to prevent relative motion or axial movement of the bush in the block.

- Bushes re standardized are defined by d X DX L (Bush 25H7 X 32r6 X 50).

- The oil hole provided at the top is used to introduce the required lubricant.





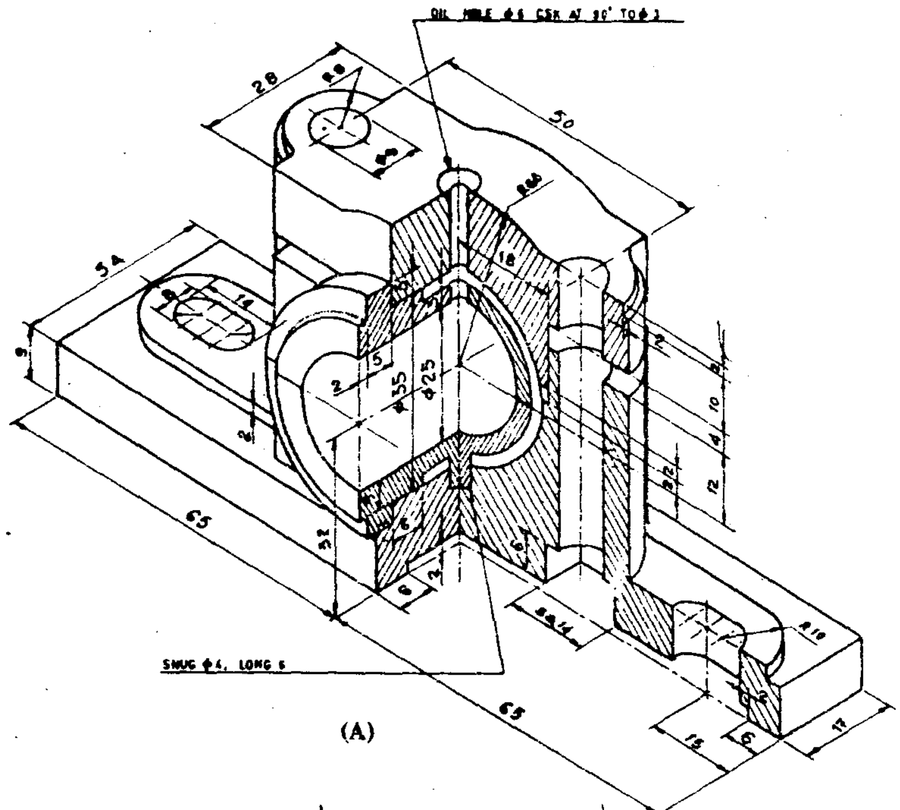
3- Pedestal (Split) Bearing

- For long shafts requiring intermediate supports, pedestal bearings (Plummer blocks) are preferred in place of ordinary bushed bearing.

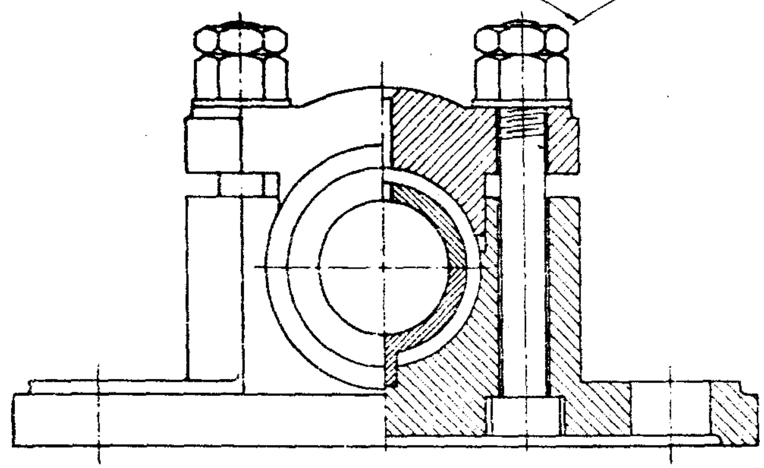
- Pedestal bearings consist mainly of a pedestal, a cap and a bush split into two halves called 'brasses.'

- Easy assembly of the unit and the periodical replacement of the brasses is made by the split parts.

- Flanges are provided to prevent the axial movement.

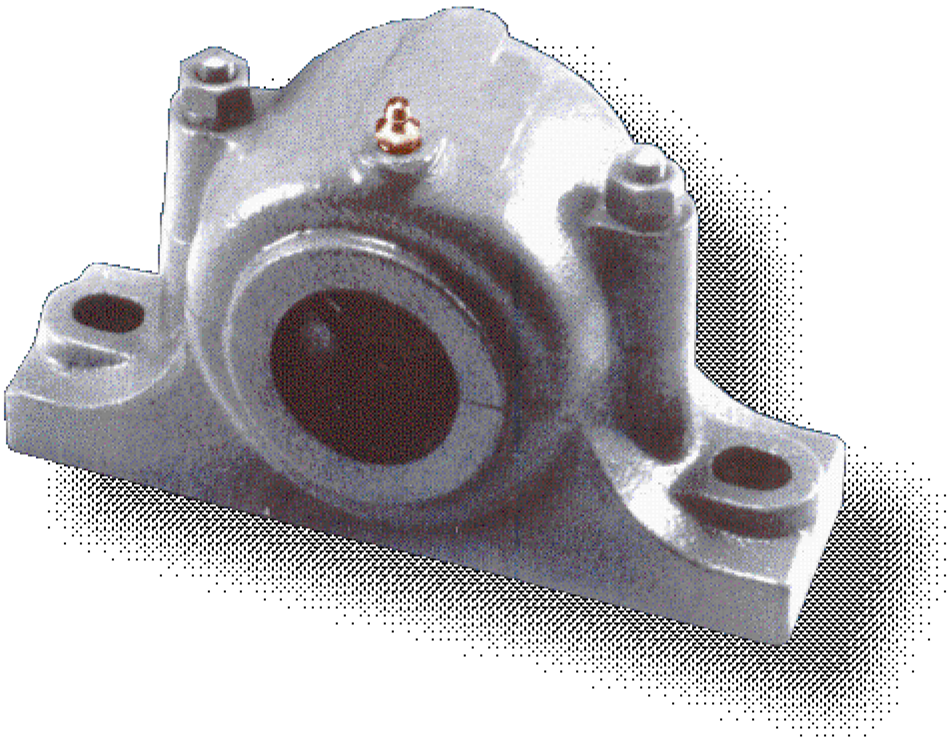


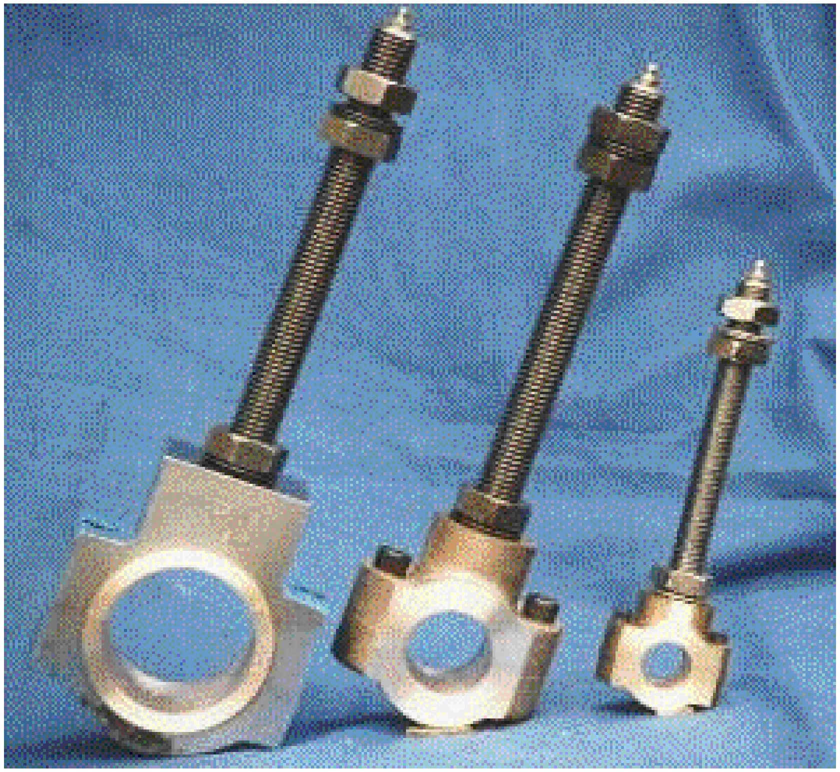
(A)

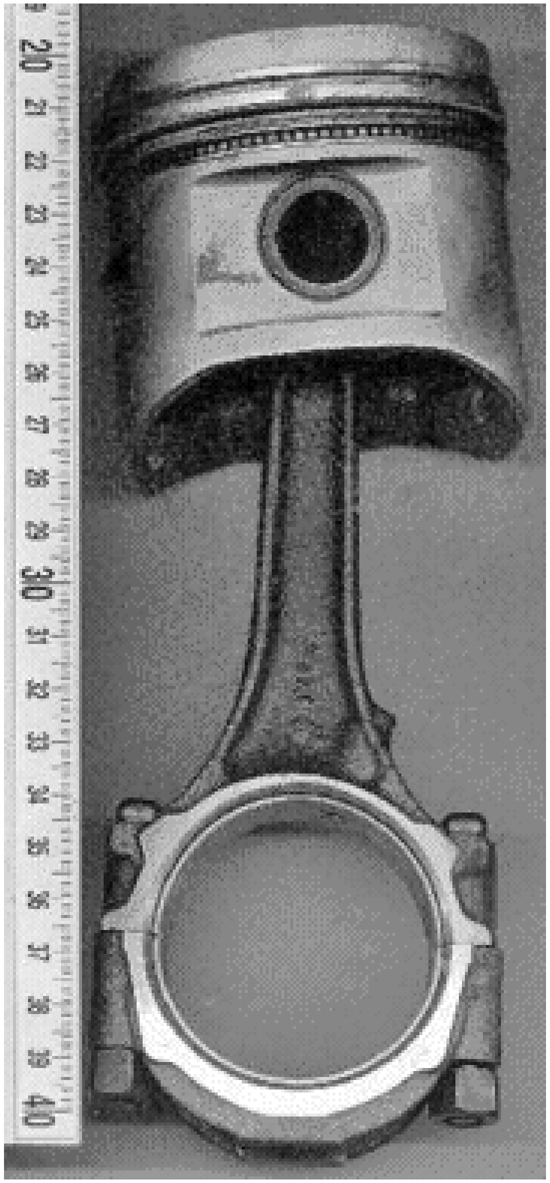


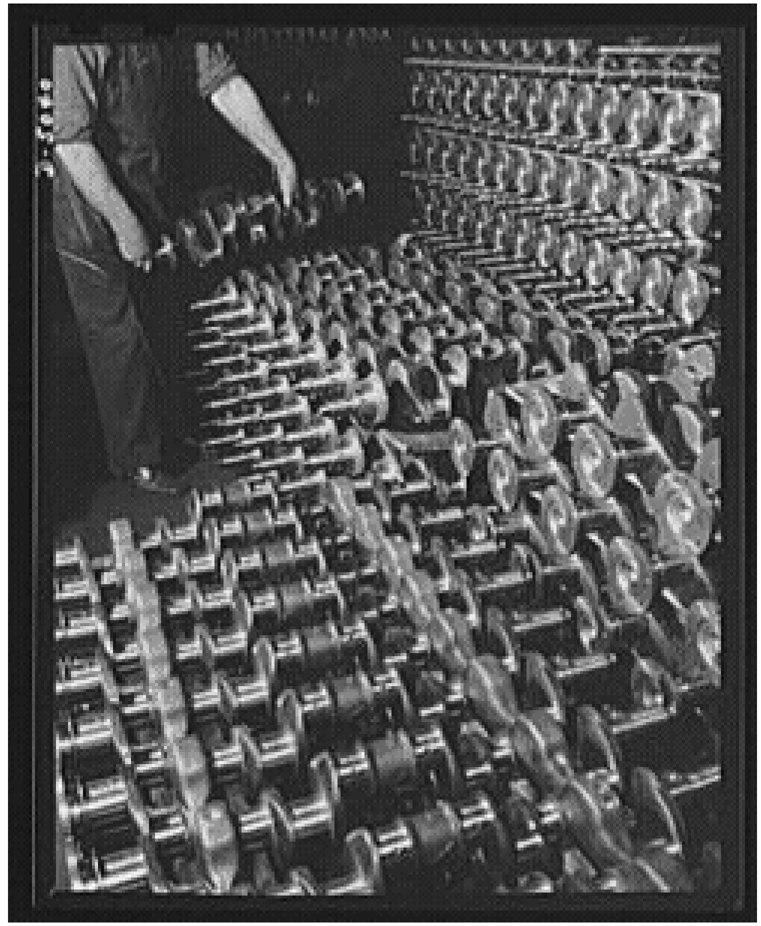
(B)

Examples of pedestal bearings:





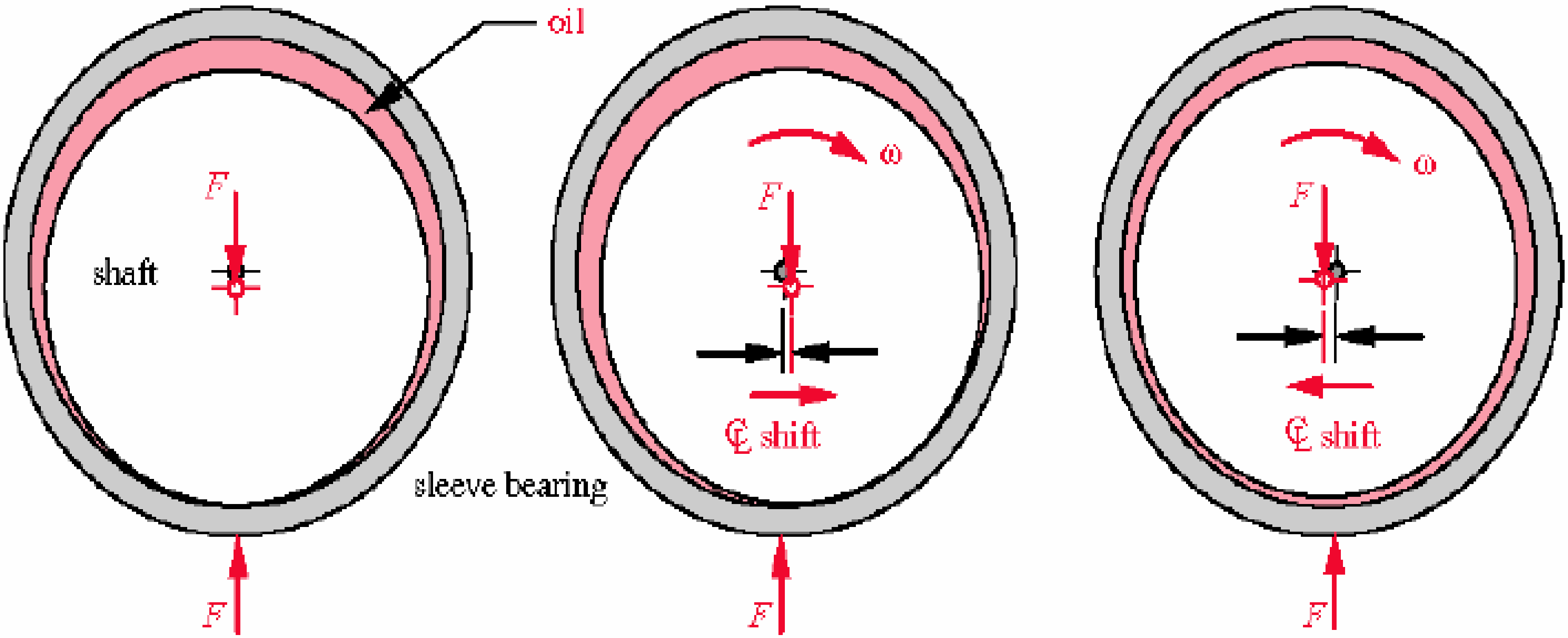




Types of lubrication:

1- Hydrodynamic lubrication:

- The most effective technique in journal bearings.
- The surface of the mating parts are separated by a relatively thick
  film of lubricant.
- The film pressure is created by the moving surfaces itself.
- Surface wear does not occur.
- Film thicknesses 0.008-0.020 mm.
- F = 0.002-0.010.



(a) Shaft stationary - metal contact
forces and centers in line

(b) Shaft rotating slowly
- boundary lubrication
- contact point leads centerline

(c) Shaft rotating rapidly
- hydrodynamic lubrication
- no metal contact
- fluid is pumped by shaft
- shaft lags bearing centerline

GURE 10-3

2- Hydrostatic lubrication:

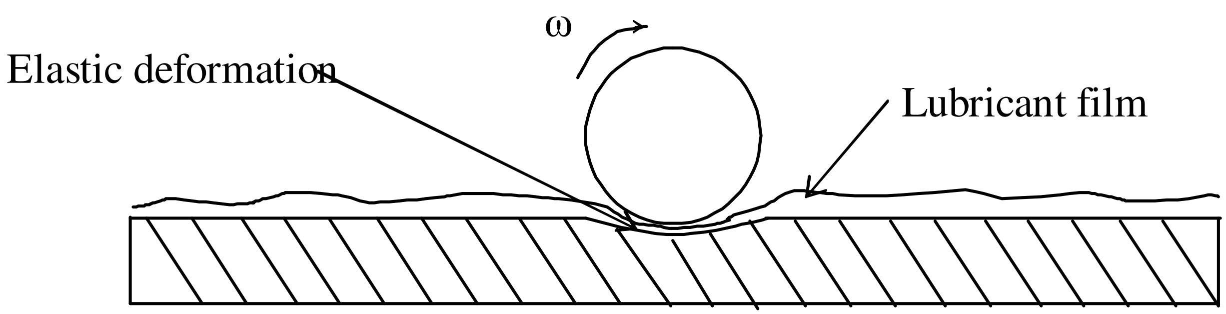
- The lubricant is introduced at a pressure high enough to separate the surfaces with a thick film of lubricant.
- Continuous flow of lubricant to the sliding interface.
- e.g air hockey, hovercraft.
- f=0.002-0.010

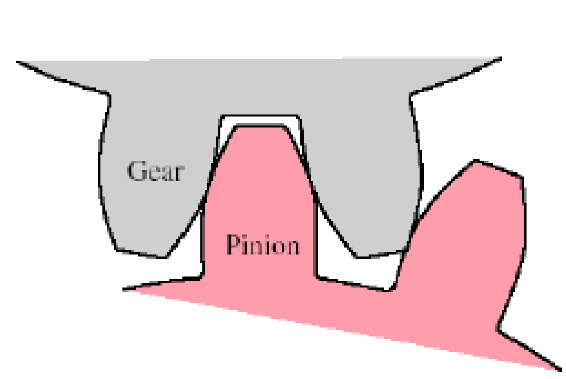
Air hockey

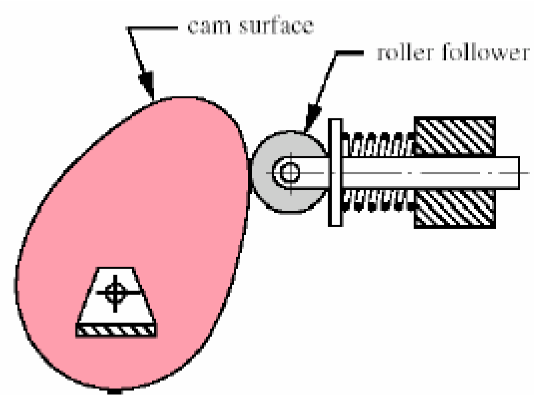
Hovercraft

3- Elastohydrodynamic lubrication:

- When a lubricant is introduced between surfaces which are in rolling contacts, such as mating gears, cams or rolling bearing.
- Elastohydrodynamic: occurs if the contacting surfaces nonconforming as with the gear teeth or cam and follower. Small contact patch allows a full hydrodynamic film to form.
- Depends on elastic deformation of parts.







4- Boundary lubrication:

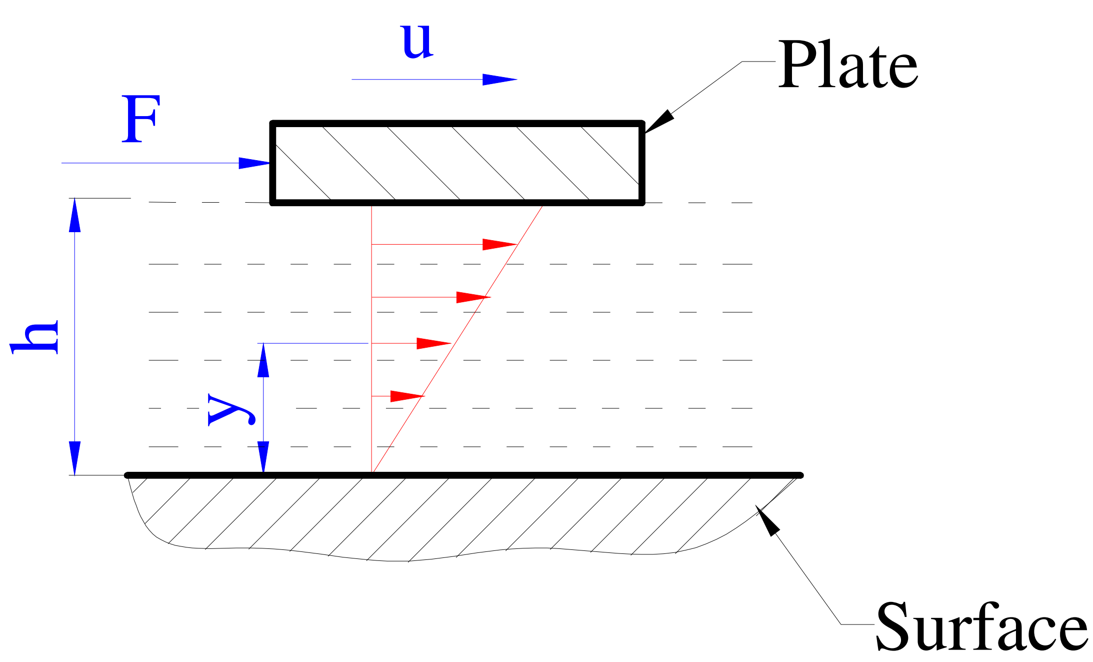
- Insufficient surface area, drop in the velocity of the moving surface, increase load or increase in the lubricant temperature, may prevent the built up of enough film thickness.
- $f=0.05-0.20$.

5- Solid film:

- When bearings must be operated at extreme temperature, a solid film lubricant such as graphite may be used because the ordinary mineral oils are not satisfactory.
- Low coefficient of friction.

Viscosity:

- Viscosity is a measure of fluid's resistance to shear.
- Viscosity, μ, for fluids is analagous to shear modulus, G, for solids.
- Temperature increases, viscosity decreases.
- Pressure increases, viscosity increases.
- To derive the absolute viscosity we consider two parallel surfaces, one moving relative two the other with a fluid trapped between the two surfaces.



$$\tau = \frac{F}{A} = \mu \frac{du}{dy}$$ Newton's law of viscous flow

Where:

μ Is the absolute (dynamic) viscosity.

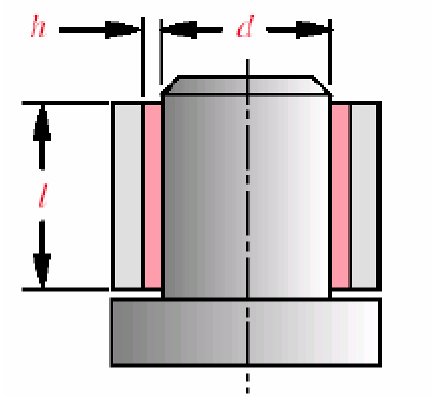
- The shear stress is proportional to the rate of change of velocity w.r.t the y.
- Assume μ = constant

$\therefore$ $$\frac{du}{dy} = \frac{u}{h}$$

$\therefore$ $$\tau = \mu \frac{u}{h}$$

- Units of viscosity:
  a- μ $(N/m^2)/s^{-1}$ = pa.s
  b- The Poise, P = dyn.s/cm^2, dyn = gm.cm/s^2
  Cp = 1/100 P

Petroff's Law:



- r : Shaft radius
- c : Clearance (filled with oil)
- l : length of bearing
- N : rev/s
- If the shaft rotates at N (rev/s), then its surface velocity,

$$u = \omega . r = 2\pi N . r$$

$$\therefore \quad \tau = \mu \frac{u}{h} = \mu \frac{2\pi N r}{c}$$

- The torque:

$$
\begin{aligned}
T &= F \times r \\
&= (\tau \times \mathrm{A}) \times r \\
&= \tau \times (2\pi r l) \times r \\
&= \frac{2\pi \mathrm{Nr}}{c} \mu \times 2\pi r l \times r \qquad (1) \\
&= \frac{4\pi^2 r^3 l \mu N}{C}
\end{aligned}
$$

- If W is the radial force acting on the bearing, then the pressure:

$$P = \frac{W}{2rl}$$

- $\therefore$ The frictional force is fW, where f is the coefficient of friction.

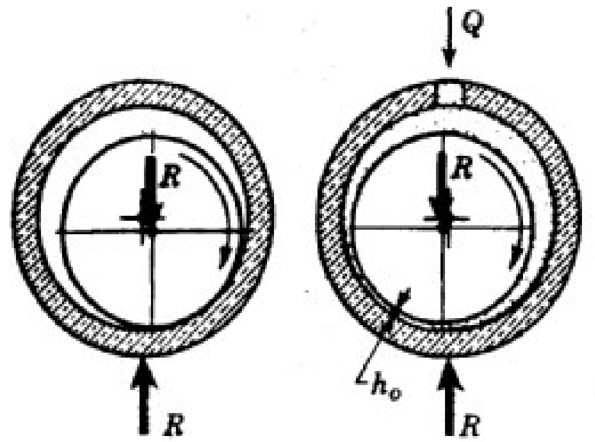
- The frictional torque T:

$$
\begin{aligned}
T &= (f \times W) \times r \\
&= f(2rlP) \times r \qquad (2) \\
&= 2r^2 flP
\end{aligned}
$$

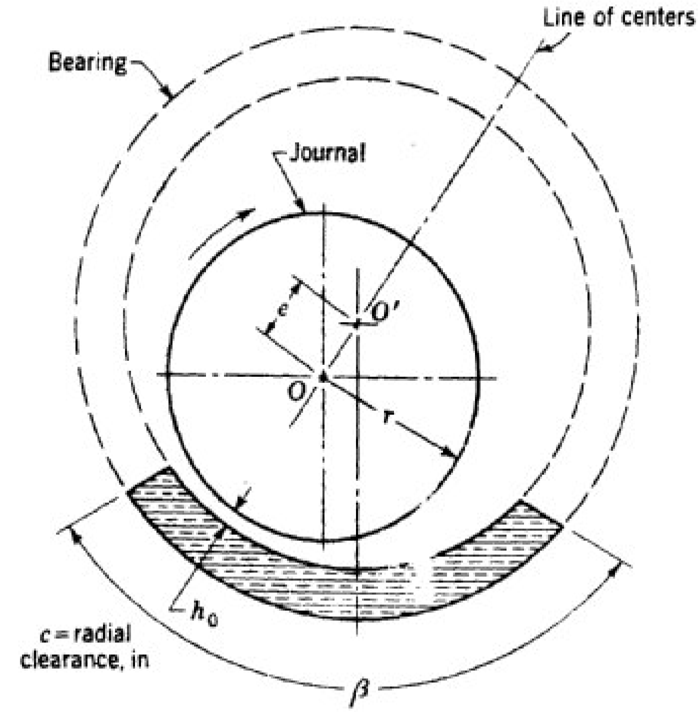
- From equations (1) and (2), the coefficient of friction:

$$f = 2\pi r^2 \frac{\mu N}{P} \frac{r}{c} \qquad \text{Petroff's Law}$$

Sliding bearings nomenclature:



(a) Dry (b) Lubricated



- O' : Center of the bearing
- O : Center of the journal
- c : The radial clearance (difference in the radii of the
- bearing and journal)
- e : Eccentricity
- h : Oil film thickness at any point
- h_o : Minimum oil film thickness and it occurs at the line of centers.
- ε : e/c ……. Eccentricity ratio

Sliding bearings design:

Two groups of variables in the design of journal bearings:

1- Selected (chosen) parameters:

A- The viscosity, μ.

B- The load per unit of projected bearing area, P.

C- The speed N (rev/s)

D- The bearing dimensions (r, c, β, l)

2- Dependent variables:

A- The coefficient of friction, f.

B- The temperature rise, ΔT.

C- The oil flow, Q.

D- The minimum oil film thickness, h_o.

E- Angle for maximum oil film pressure.

Bearing characteristic number (Sommerfeld number):

This quantity is defined by the equation:

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

Where:

S: bearing characteristic number

r: journal radius

c: radial clearance

μ: absolute viscosity

N: speed (rev/s)

P: load per unit of projected bearing area.

Design steps:

1- Temperature rise:

- The average working temperature of the oil:

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

Where:

T_i : is the inlet temperature

ΔT : is the temperature rise

- The dimensionless temperature rise variable is:

$$T_{\text{var}} = \frac{\gamma C_H \Delta T}{P}$$

Where:

γ : density of oil (861 kg/m^3)

C_H : specific heat of the lubricant (1760 J/kg C^o)

Procedure for determining the temperature rise:

a- Estimate the average temperature of the oil.

$$\Delta T = 15^{O}C - 20^{O}C$$

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

b- Find μ for the chosen oil at T_{av}. (from figure 12-10& 12-11)

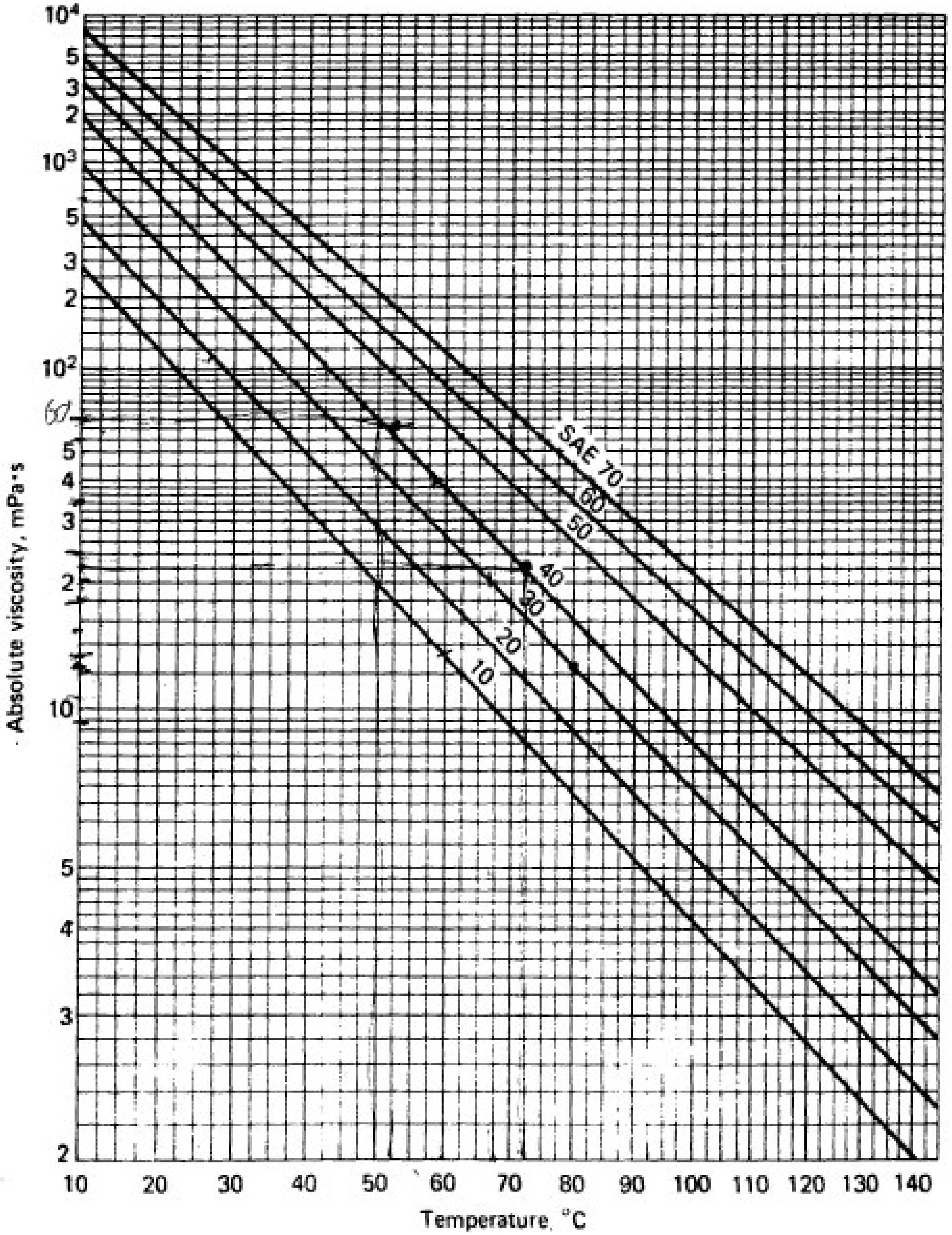
c- Calculate the bearing characteristic number.

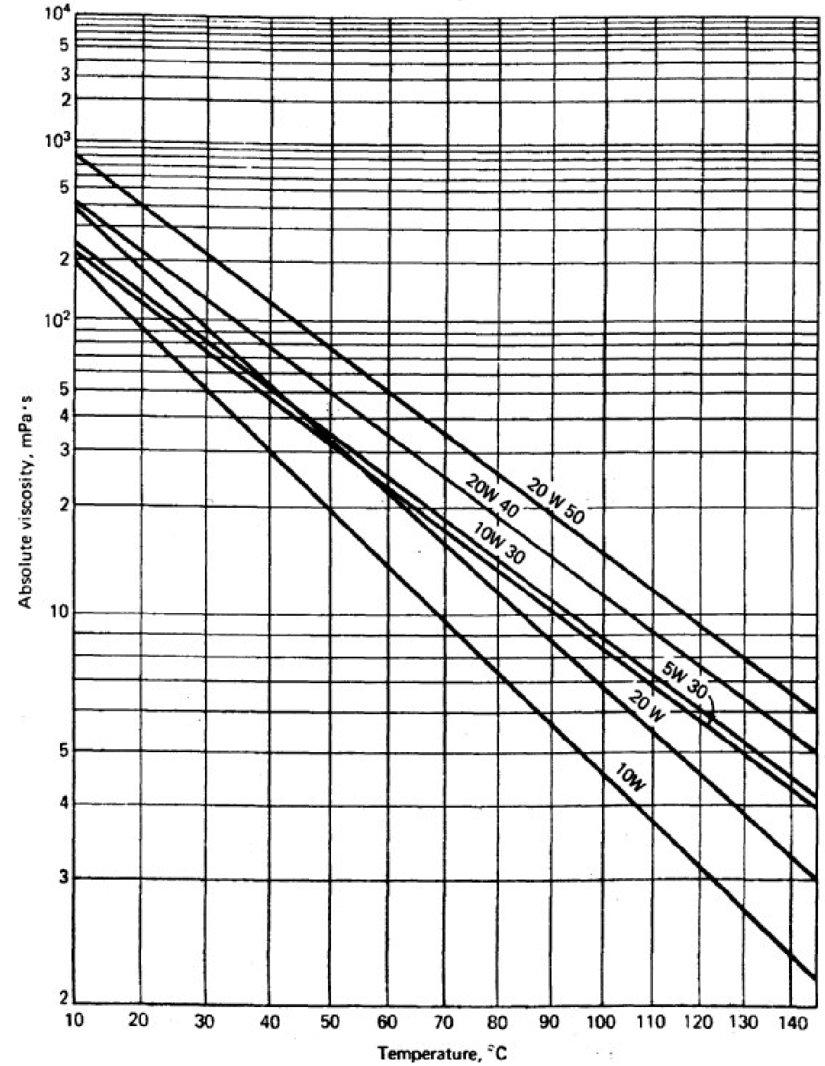
$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

d- Find the temperature rise variable T_{var}. (from figure 12-12)

e- Determine the temperature rise ΔT from the relation:

$$T_{\mathrm{var}} = \frac{\gamma C_H \Delta T}{P}$$





Viscosity temperature chart

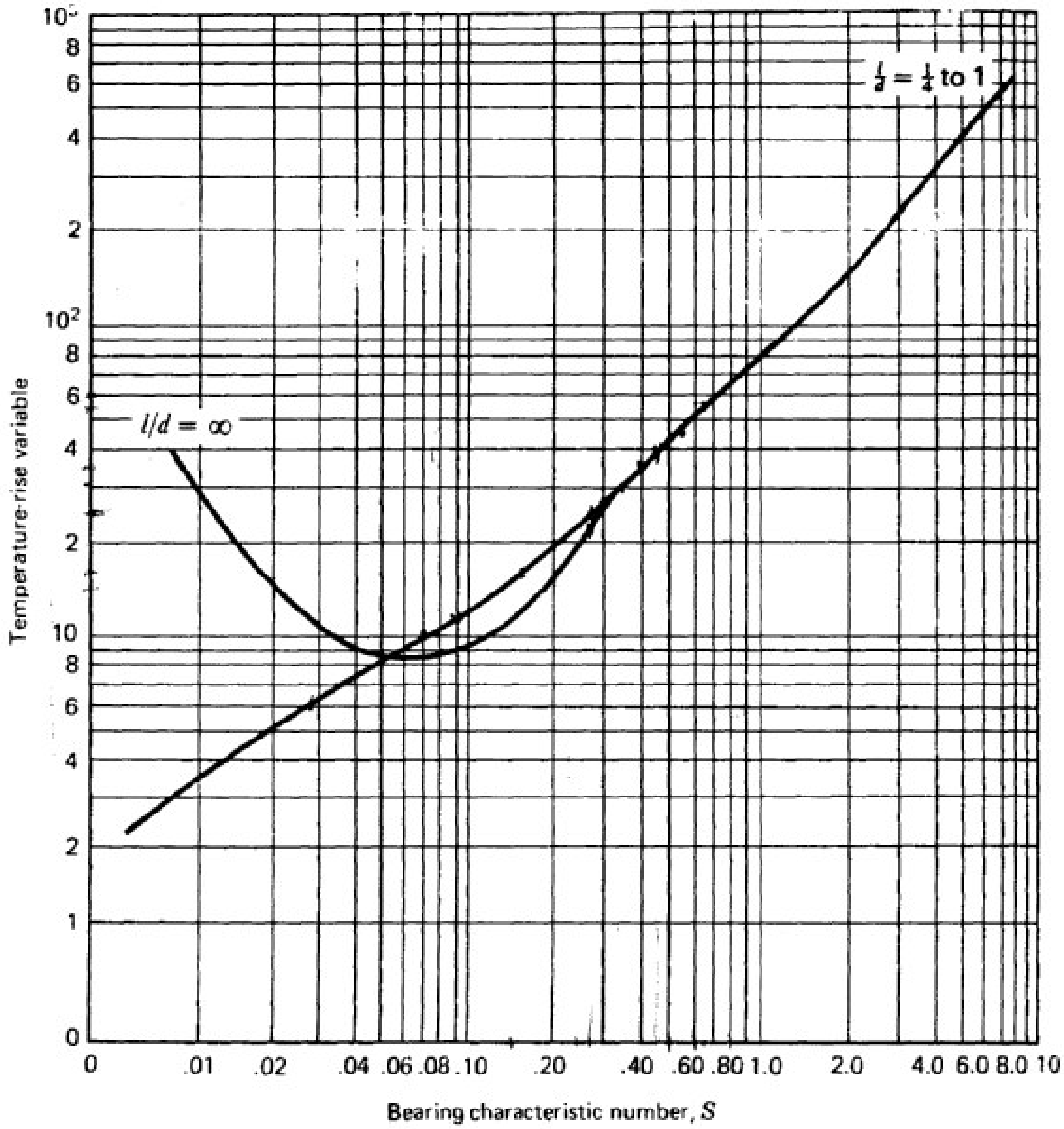


Chart for temperature variable

f- Find T_{av} from the relation:

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

g- Repeat again from step (b) until to get two successive T_{av} very close.

$$\left(T_{av}^{new} - T_{av}^{old} \approx 2^{oC}\right)$$

h- According to the last T_{av}, we have to get μ and S.

2- From figure and for certain l/d and S→ Find h_o/c and ε.

(The minimum oil film thickness variable and eccentricity ratio)

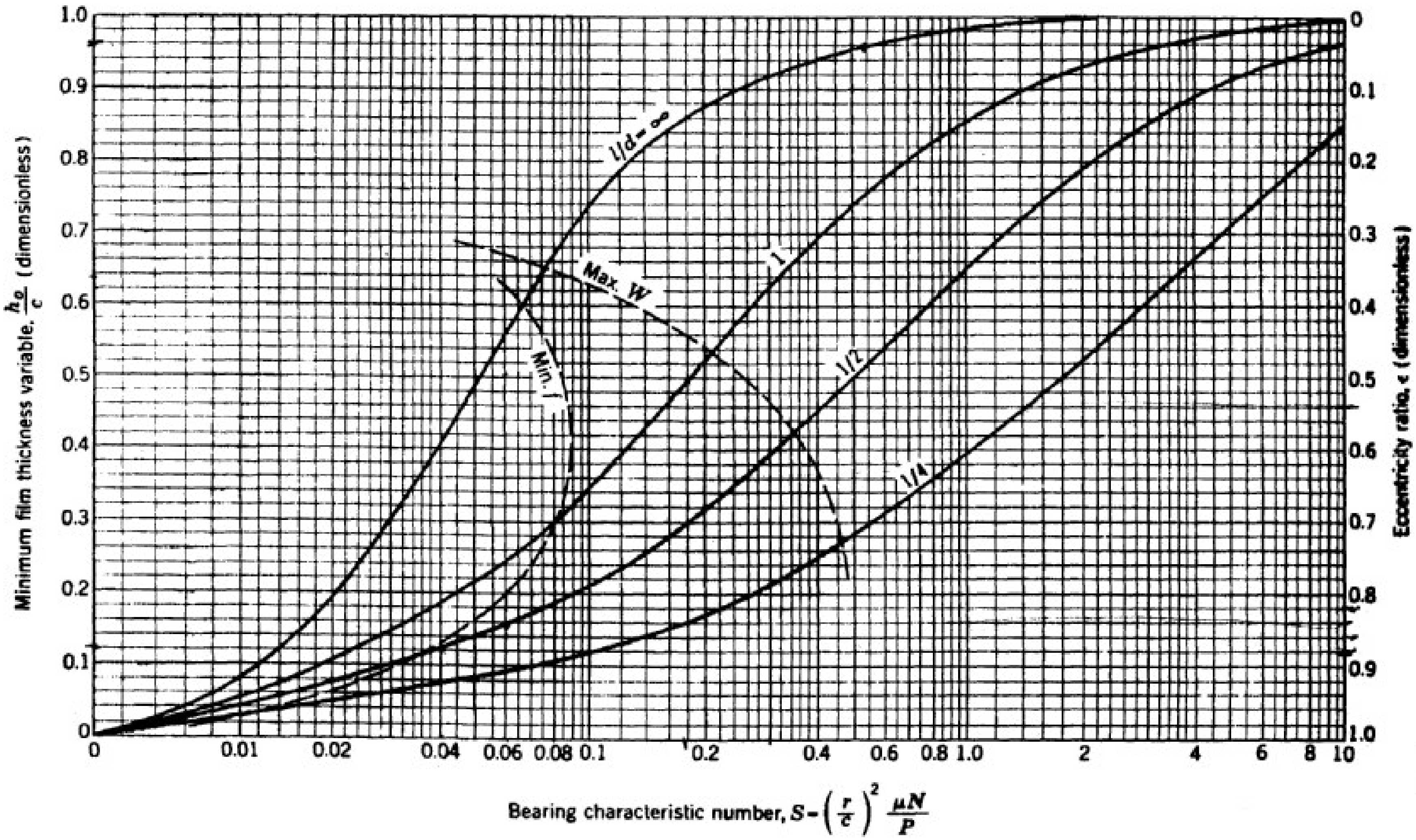


Chart for temperature variable

3- From figure and for certain l/d and S→ Find position of minimum of minimum film thickness Φ°.

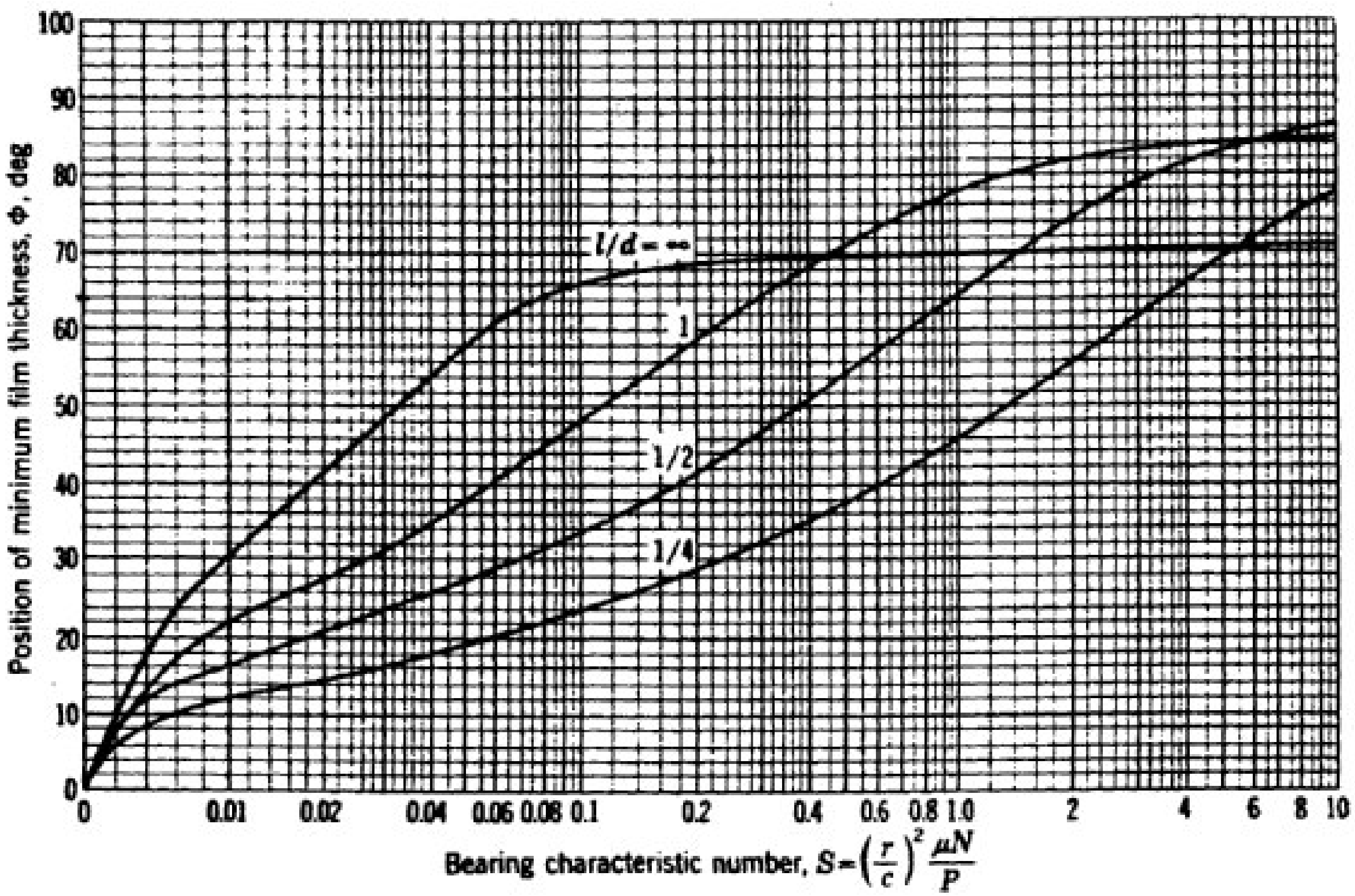


Chart for determining the position of the minimum oil film thickness

4- From figure and for certain l/d and S→ Find the coefficient of friction variable (r/c)f.

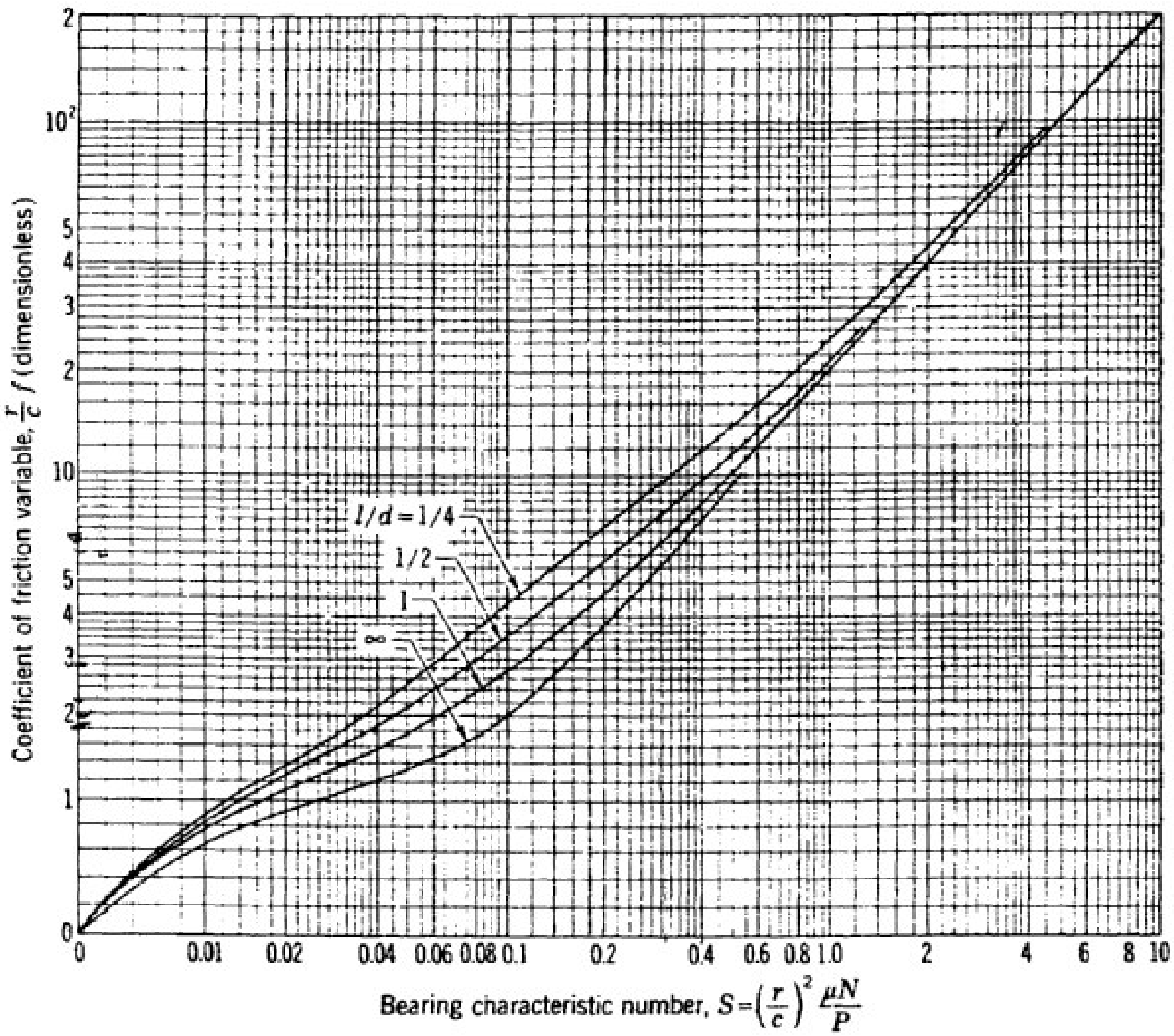


Chart for coefficient of friction variable

5- The torque required to overcome friction:

$$T = f \times W \times r$$

Where:

f : Coefficient of friction

W : Radial load on bearing

r : Radius of bearing

6- The power loss (due to friction):

$$\begin{aligned} H &= T \times \omega \\ &= T \times (2\pi N) \end{aligned}$$

7- From figure (12-17) and for certain l/d and S→ Find the flow variable Q/rcNl

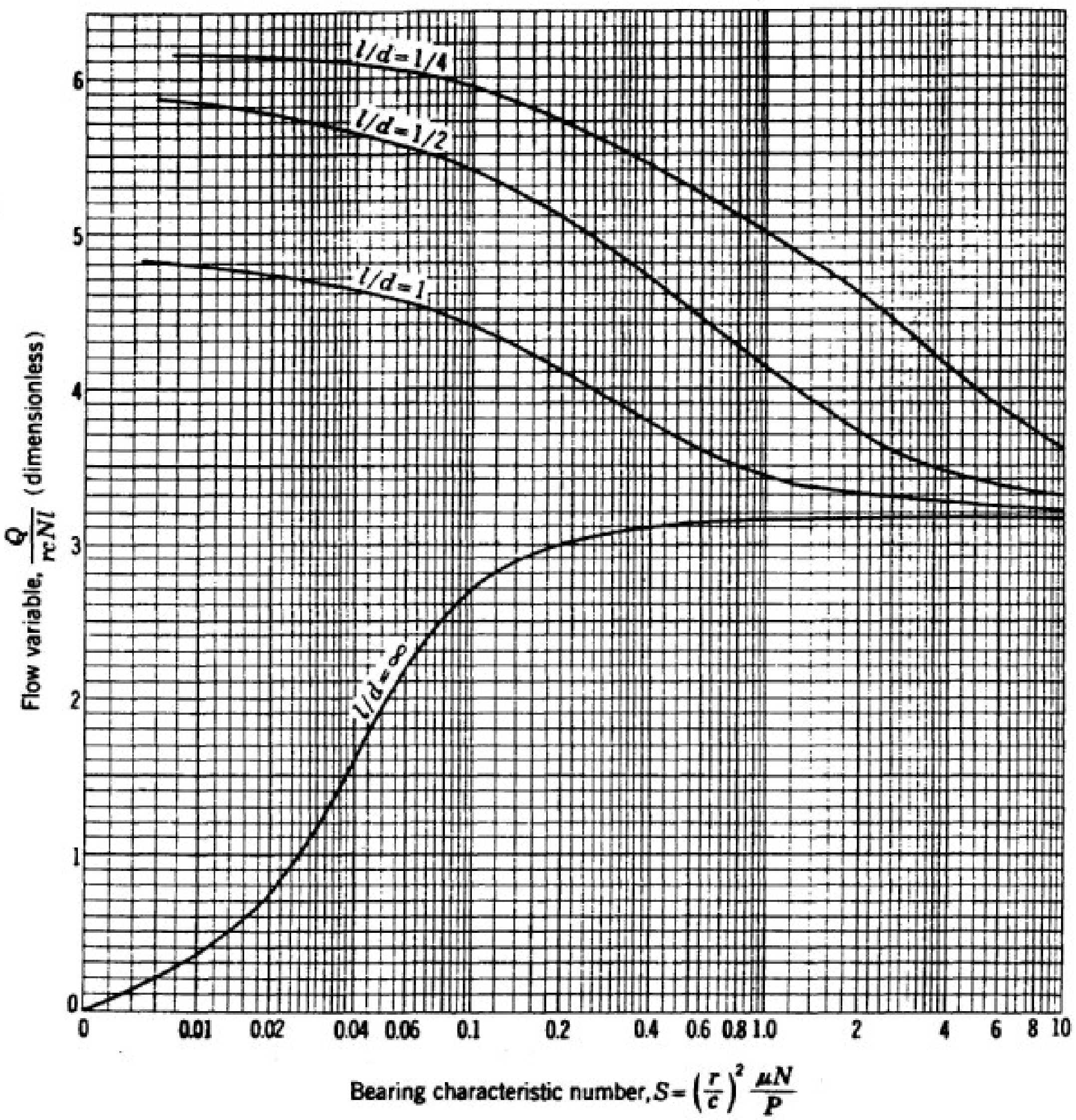


Chart for flow variable

- NOTE:

The amount of oil supplied to the bearing must be > Q

8- From figure and for certain l/d and S→ Find the side leakage variable (Q_s/Q).

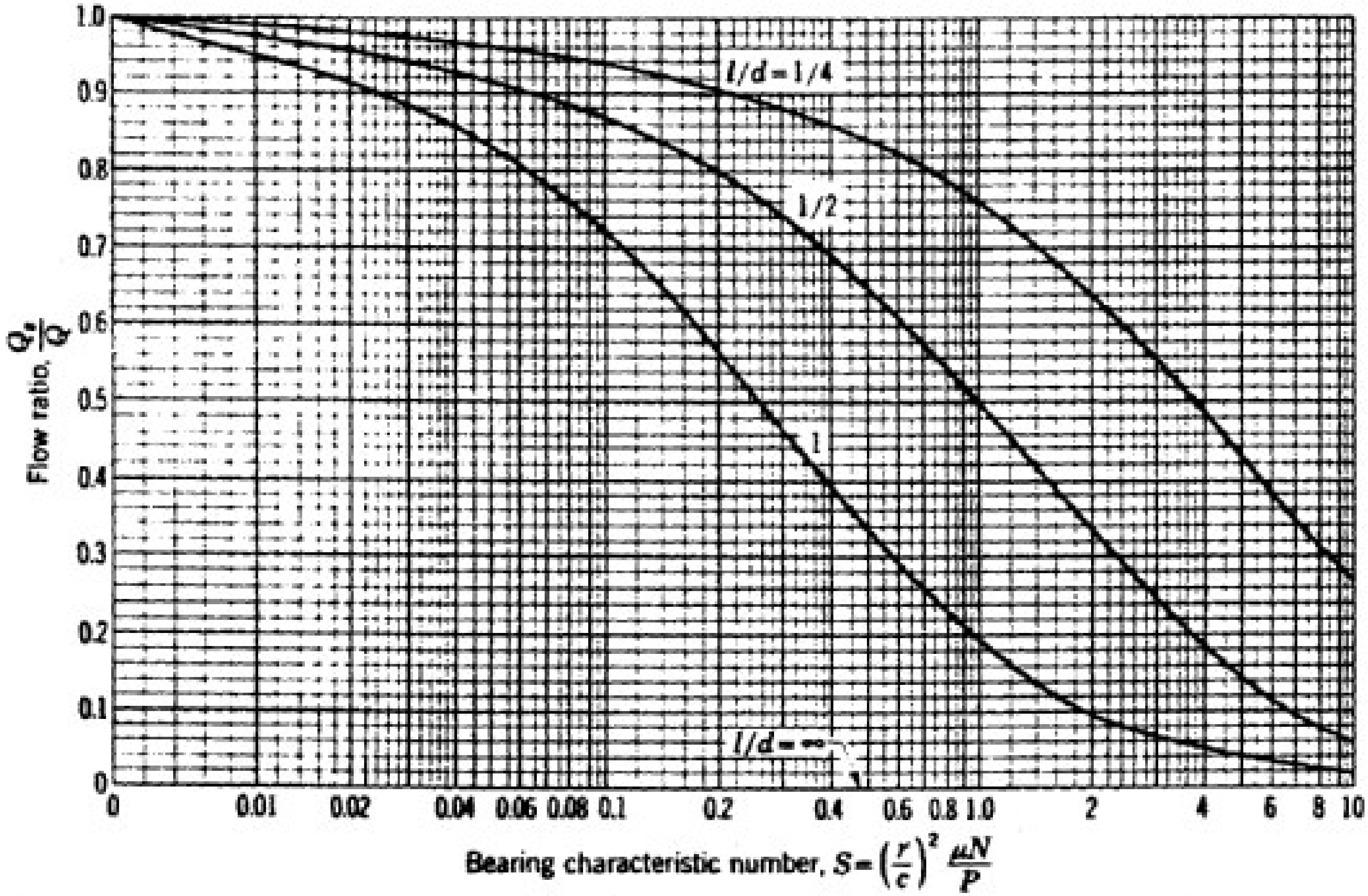


Chart for side leakage variable

9- From figures and for certain l/d and S→ Find the maximum film pressure variable (P/P_{max}) → figure & its angular location (θ_{Pmax}) and the terminating position of the oil film (θ_{Po})

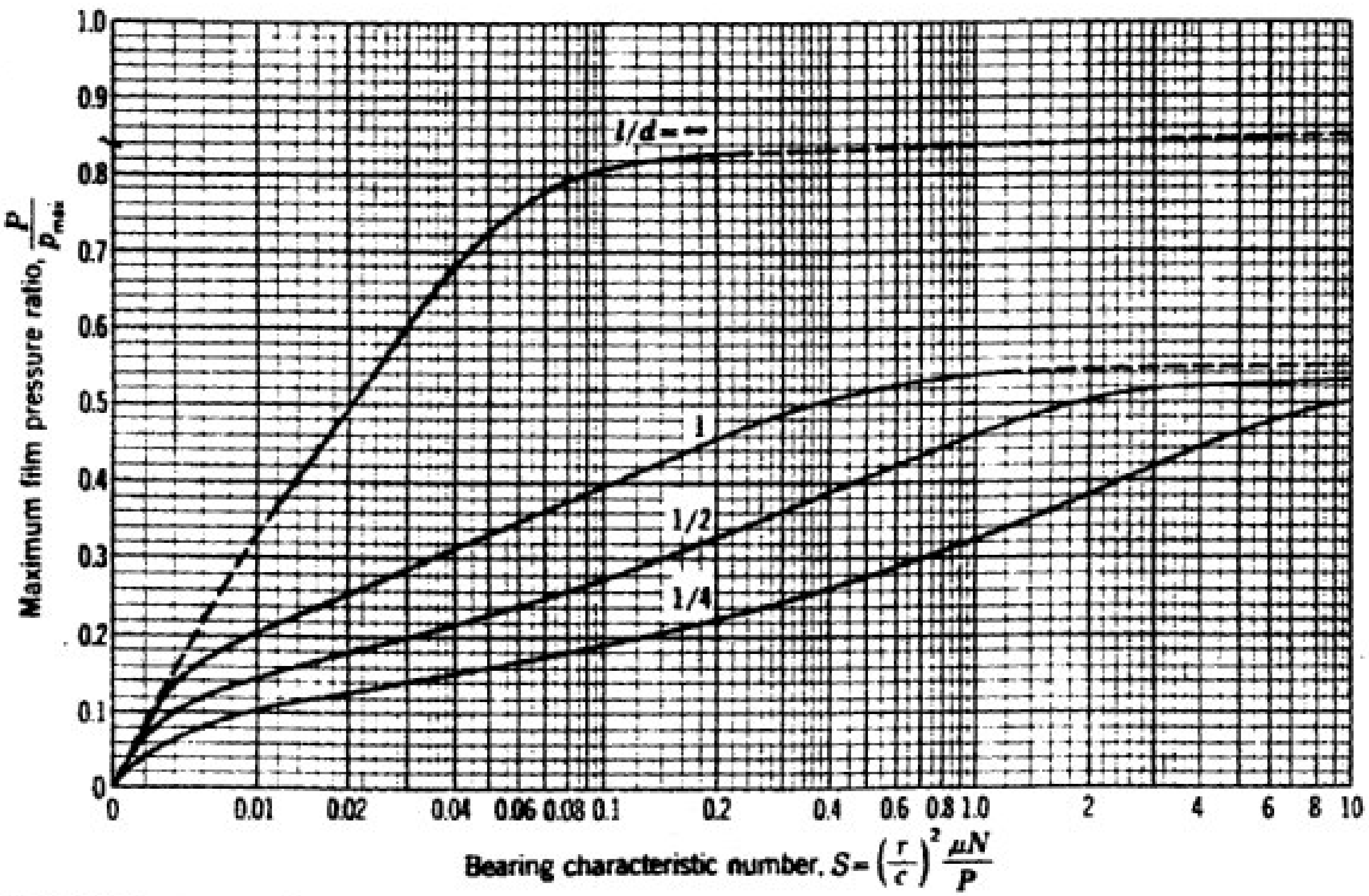


Chart for maximum film pressure variable

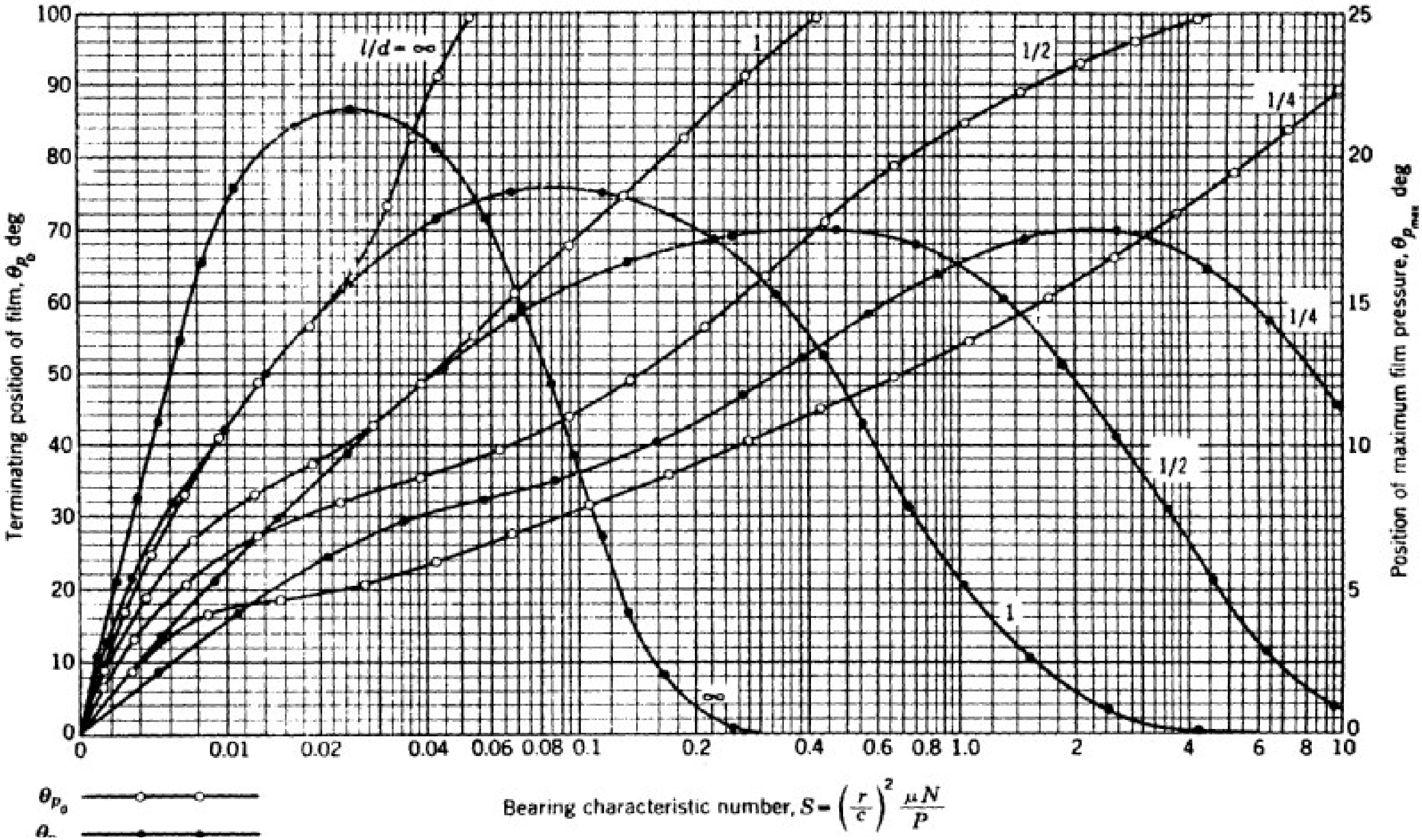


Chart for finding the terminating position of the lubricant film and the position of maximum film pressure