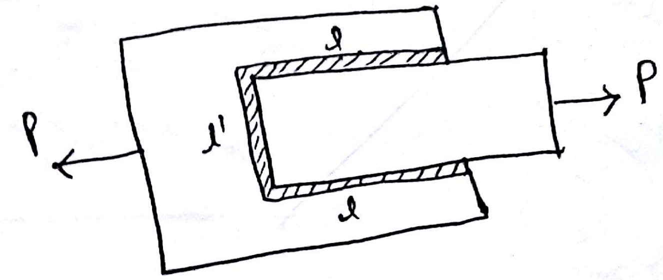
M.D.

CHAPTER-1 [JOINTS]

- welded joints

• steps to design fillet weld

1) find strength of plate



$$P = \sigma_t (w.t)$$

w = width of plate

t = thickness of plate

2) find parallel fillet joint strength

$$P_1 = 2 \times \tau \times (0.707 \times t \times l)$$

3) find transverse fillet joint strength

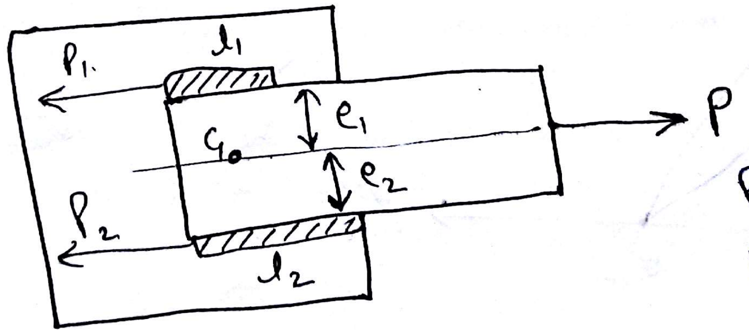
$$P_2 = \sigma_{t_1} \times (0.707 \times t \times l')$$

↳ τ is used in Gate Questions

σ_{t_1} = weld material strength.

4) $P = P_1 + P_2$

⇒



$$P = P_1 + P_2$$

$$P_1 e_1 = P_2 e_2$$

$$P_1 = .707 \times t \times l_1 \times \tau$$

$$P_2 = .707 \times t \times l_2 \times \tau$$

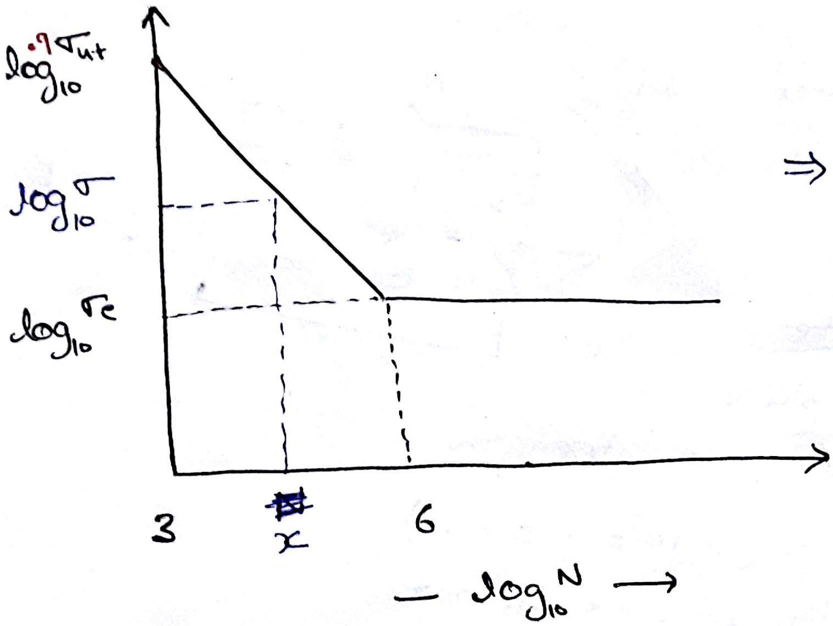
⇒ efficiency of riveted joint

$$\eta = \frac{\text{mini}\{P_{tearing}, P_{crushing}, P_{shearing}\}}{\text{Strength of Plate without rivets}}$$

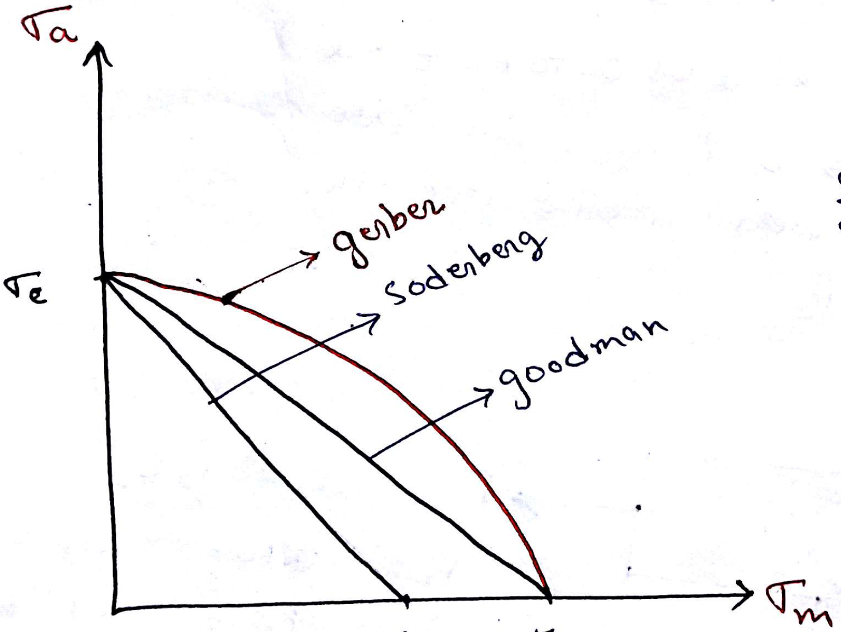
⇒ while designing a Threaded or bolted joint, nominal (outer) dia should be used unless relation b/w core dia and nominal dia is given in Question

CHAPTER-2 [Design against fatigue]

- Stress ratio $= \dfrac{\sigma_{min}}{\sigma_{max}}$
- $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$
- Amplitude ratio $= \dfrac{\sigma_a}{\sigma_m}$
- $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$



$\Rightarrow N = 10^x$



Soderberg's eqn

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_{yt}} = \frac{1}{FOS}$$

Goodman's eqn

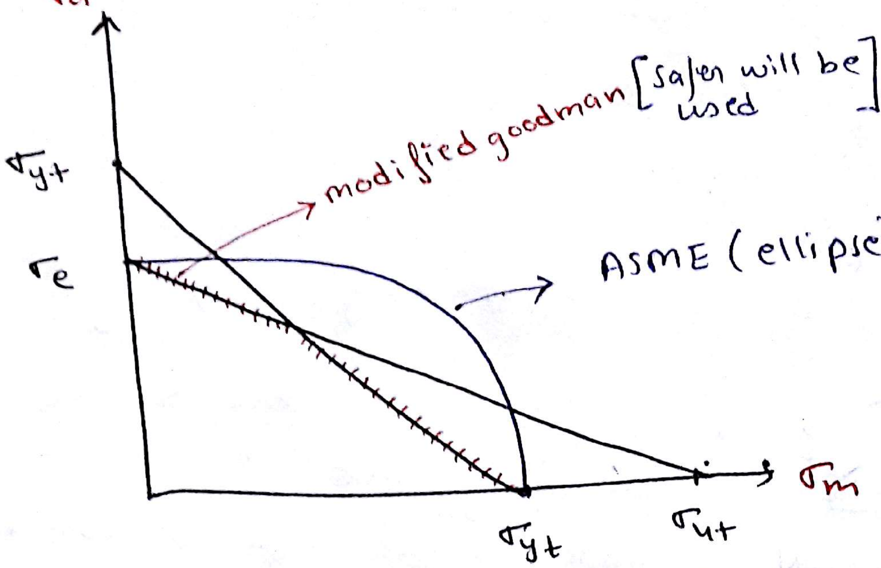
$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_{ut}} = \frac{1}{FOS}$$

Gerber's eqn

$$(FOS)\frac{\sigma_a}{\sigma_e} + \left[(FOS)\frac{\sigma_m}{\sigma_{ut}}\right]^2 = 1$$

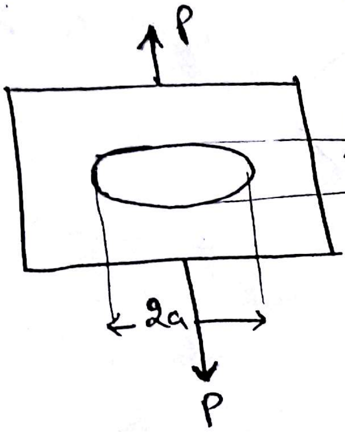
- If various stress concentration factors are given then, replace,

$$\sigma_e \Rightarrow \frac{K_s . K_L . \cdots}{K_f}\sigma_e \quad \& \quad \sigma_m \Rightarrow K_t \sigma_m$$



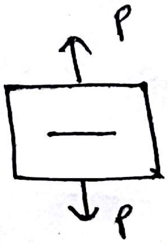
ASME (ellipse) $\Rightarrow \left(\frac{\sigma_a}{\sigma_e}\right)^2 + \left(\frac{\sigma_m}{\sigma_{yt}}\right)^2 = \frac{1}{(FOS)^2}$

$\Rightarrow$ theoretical stress concentration factor (K_t)



$$\boxed{K_t = \left(1 + \frac{2a}{b}\right)}$$

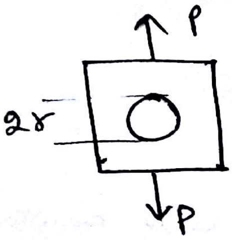
(i)



$K_t = \left(1 + \frac{2a}{0}\right)$

$= \infty$

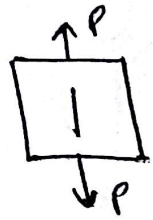
(iii)



$K_t = 1 + \frac{2r}{r}$

$K_t = 3$

(ii)



$K_t = \left(1 + \frac{2 \times 0}{b}\right)$

$= 0$

$\Rightarrow$ failure stress concentration factor (K_f)

$K_f = \dfrac{(\sigma_e)\text{ without notch}}{[(\sigma_e)\text{ with notch}]_{\text{practically}}}$ $\qquad K_t = \dfrac{(\sigma_e)\text{ without notch}}{[(\sigma_e)\text{ with notch}]_{\text{theore.}}}$

$K_t > K_f$ always

$\Rightarrow$ notch sensitivity (q) $= \dfrac{(\sigma_{max})_{actual} - \sigma_o}{(\sigma_{max})_{th.} - \sigma_o} = \dfrac{K_f\sigma_o - \sigma_o}{K_t\sigma_o - \sigma_o}$

$$q = \frac{K_f - 1}{K_t - 1}$$

- $q = 0 \Rightarrow K_f = 1 \Rightarrow$ not sensitive to notch
- $q = 1 \Rightarrow K_f = K_t \Rightarrow$ Purely sensitive to notch

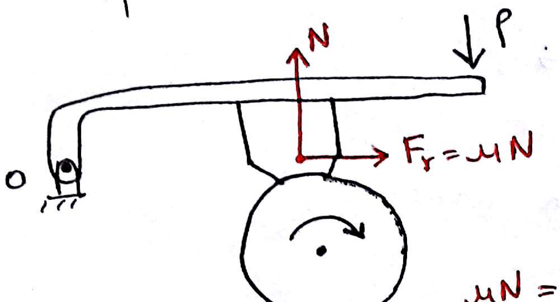
- $K_{\text{surface finish}} = 1$ for Cast iron

CHAPTER-3 [BRAKES]

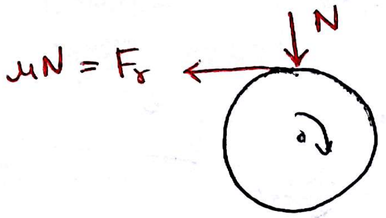
- Brake factor $= \frac{F_f}{P}$

if F_f supports $P \Rightarrow$ called as self energising brake

1. Simple shoe brake

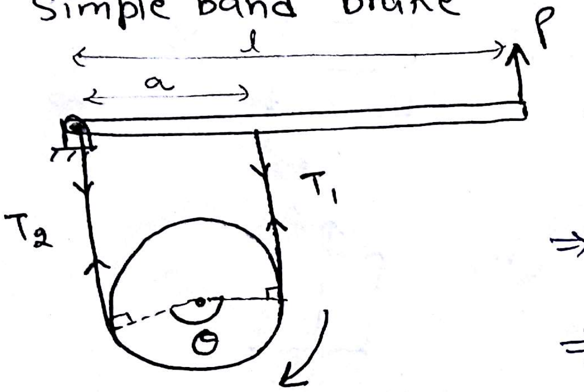


- Simply balance the torque on hing Point
- Apply torque equ^n on wheel/disc



$T_f = F_r \cdot R$

2. Simple band brake

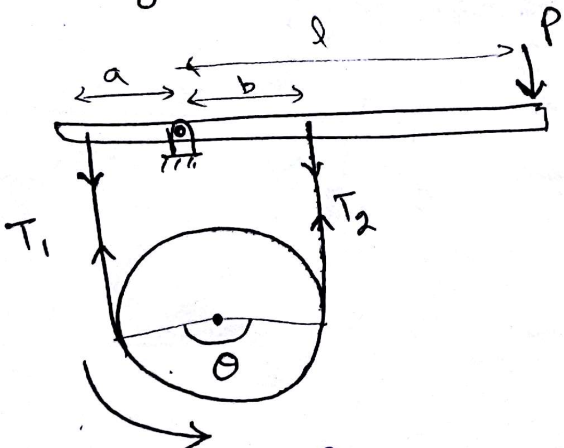


$\Rightarrow \frac{T_1}{T_2} = e^{\mu\theta}$

$\Rightarrow T_f = (T_1 - T_2) R$

$\Rightarrow T_1 a = P l$

3. Differential Band Brake

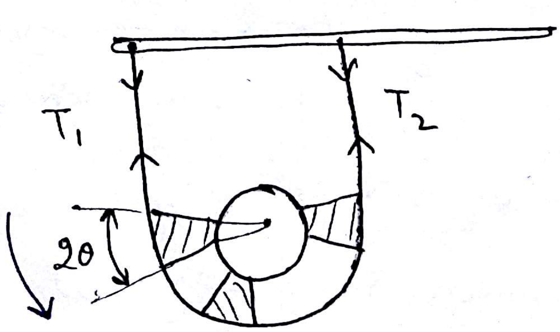


$\frac{T_1}{T_2} = e^{\mu\theta}$

$T_1 a = T_2 b + P l$

$(T_1 - T_2) R = T_f$

4. Band & Block Brake



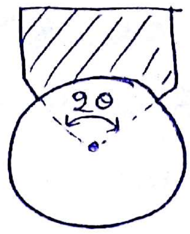
$\frac{T_1}{T_2} = \left(\frac{1 + \mu \tan\theta}{1 - \mu \tan\theta}\right)^n$

$T_f = (T_1 - T_2) R$

$n =$ no. of block

- effective coefficient of friction μ'

$$\mu' = \frac{4\mu \sin\theta}{2\theta + \sin(2\theta)}$$



CHAPTER-4 [Clutches / DISC BRAKES]

New Brakes (Uniform Pressure theory)

$\Rightarrow P = \dfrac{F}{\pi(r_2^2 - r_1^2)}$

$\Rightarrow T_f = \mu . F . R_m$

$\Rightarrow R_m = \dfrac{2}{3}\dfrac{(r_2^3 - r_1^3)}{(r_2^2 - r_1^2)}$

Old Brakes (Uniform wear theory)

$\Rightarrow P = \dfrac{F}{2\pi(r_2 - r_1)r}$

$\Rightarrow T_f = \mu . F . R_m$

$\Rightarrow R_m = \dfrac{r_1 + r_2}{2}$

- Above formulas are for single pairing surface for 'n' pairs, multiply T_f by 'n' to get total torque transmitted.
- F remain same for all pairs of clutches
- F is distributed for thrust bearings

$$T_{loss} = \mu . n \times (F_{collar}) \times R = \mu \cancel{n} \times \left(\frac{F}{\cancel{n}}\right) . R = \mu F R$$

↳ but still independent of no. of collar

$\Rightarrow$ Cone clutch

$$T_f = \mu\left(\frac{F}{\sin\alpha}\right) . R_m$$

$R_m \to \dfrac{2}{3}\left(\dfrac{r_2^3 - r_1^3}{r_2^2 - r_1^2}\right) \Rightarrow$ UPT

$R_m \to \dfrac{r_1 + r_2}{2} \Rightarrow$ U.W.T.

α = half cone angle

$\Rightarrow$ Centrifugal clutch

$$T_f = n . \mu N R$$

n = no. of shoes

$N = mr(\omega^2 - \omega_0^2)$

ω_0 = speed at which shoe just tuoch the rim.

R = rim radius

r = radius of shoe center

CHAPTER-5 [GEAR Design [spur]]

⇒ According to beam strength (S_b)

1) if material is same, design for pinion
   if material is different then design for weaker
   $\left((\sigma_b)_{per} \cdot Y\right)_g$ & $\left((\sigma_b)_{per} Y\right)_p$ smaller will be considered as weaker.

2) find beam strength.

$$S_b = (\sigma_b)_{per} \cdot b \cdot m \cdot Y$$

$(\sigma_b)_{per}$ = Permissible bending stress at tooth root. due to P_t

b = face width

m = module

Y = lewis form factor or tooth geometry factor

$$Y = \pi\left(0.154 - \frac{0.912}{z}\right) \Rightarrow \text{for } 20^\circ = \phi$$

z = no. of teeth

3) calculate P_{eff}.

$$P_{eff} = (FOS) \cdot \frac{C_s}{C_v} \cdot P_t$$

P_t = tangential load on tooth

C_s = Service factor = $\dfrac{\text{Starting torque}}{\text{rated torque}}$

C_v = Velocity factor

$$= \frac{3}{3+V} \qquad 0 < V < 10 \text{ m/s}$$

$$= \frac{6}{6+V} \qquad 10 \le V \le 20 \text{ m/s}$$

$$= \frac{5.6}{5.6+\sqrt{V}} \qquad 20 < V \text{ m/s}$$

4) compare S_b & P_{eff}.

if $P_{eff} \le S_b \rightarrow$ safe design

$\Rightarrow$ According to wear strength of tooth (S_w)

$S_w = d_p \cdot Q \cdot b \cdot K$

d_p = dia of Pinion

Q = Ratio factor $= \frac{2T}{T+t}$

K = load stress factor

$= 0.16 \times \left(\frac{BHN}{100}\right)^2$ for $\phi = 20°$

$= \frac{(\sigma_e) \sin^2\phi}{1.4} \times \left(\frac{1}{E_p} + \frac{1}{E_g}\right)$

$P_{eff} \le S_w \Rightarrow$ safe design

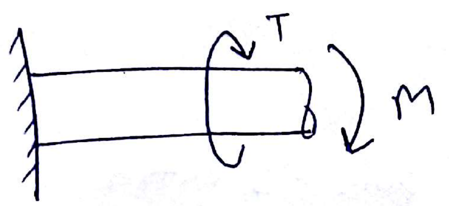
Practically

$P_{eff} \le S_b \le S_w$

CHAPTER-6 [Shafts]

Bending + Torsion $\Rightarrow$ use MSST

Pure Torsion $\Rightarrow$ use MDET



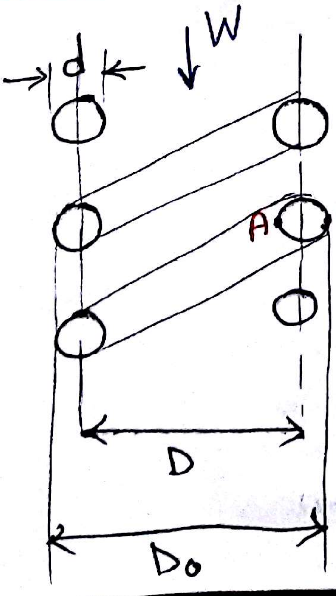
$\tau_{max} = \frac{16}{\pi d^3}\left[\sqrt{(K_b M)^2 + (K_T \cdot T)^2}\right]$

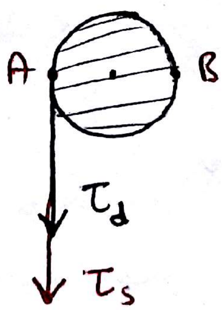
$\sigma_{max} = \frac{16}{\pi d^3}\left[(K_b \cdot M) + \sqrt{(K_b \cdot M)^2 + (K_T \cdot T)^2}\right]$

K_b = combined shock & fatigue factor for bending

K_T = combined shock & fatigue factor for torsion

CHAPTER-7 [Helical springs]





$\tau_{max} = \tau_s + \tau_d$

$\tau_{max} = K_w \cdot \frac{16 T_s}{\pi d^3}$

$T_s = \frac{WD}{2}$

$K_w = \frac{4C-1}{4C-4} + \frac{.615}{C}$

C = Spring index $= \frac{D}{d}$

K_w = wahl's correction factor $= K_c \cdot K_s$

K_c = curvature factor

K_s = shear stress factor

⇒ Stiffness of spring $= s = \frac{GD}{8nC^4}$

n = no. of coils

⇒ Axial loading ⟶ $\tau_s + \tau_d$

twisting of spring ⟶ bending stress created

Bending of spring ⟶ twisting stress created

CHAPTER – 8 [Design of flywheel] [only ESE]

- For rim type flywheel

$$V_{max} = \sqrt{\frac{\sigma_n}{\rho}}$$

- for disc type flywheel

$$V_{max} = \sqrt{\frac{8\,\sigma_n}{\rho\,(\mu+3)}}$$

μ = poison's ratio

CHAPTER – 9 [Bearings]

A) Rolling contact Bearings

SKF 6 3 10

SKF → manufacturer

6 → Type of Bearing

3 → Duty series

10 → $\frac{1}{5} \times$ Core dia

Type of Bearing

3 – tapered roller

4 – needle roller

5 – cylindrical roller

6 – Deep groove

Duty series

1 – Extra light

2 – light series

3 – medium series

4 – Heavy series

- Static load rating $= C_o = \dfrac{z K d^2}{5}$ (ball bearing) or $\dfrac{z K d l}{5}$ (roller bearing)

z = no. of rolling element
d = dia of rolling element
K = constant depends on modulus of elasticity of material & radius of curvature of contacting surface

- Dynamic load rating (C) = load on bearing which gives (10^6) i.e. 1 million revolution [inner race as rotating]

- <u>life of a bearing</u>

$$L_{90} = \left(\frac{C}{P_e}\right)^n \text{ million revolution}$$

C = Dynamic load rating (given by manufacturer)
P_e = equivalent Dynamic load (applied load)

$n = 3 \rightarrow$ for ball bearing
$= \frac{10}{3} \rightarrow$ roller bearing

$$L_H \times N \times 60 = \left(\frac{C}{P_e}\right)^n \times 10^6$$

$L_H \rightarrow$ life in hrs
$N \rightarrow$ rpm

- Relation b/w life & reliability

$$\frac{L}{a} = \left(\log_{10}\frac{1}{R}\right)^{\frac{1}{b}}$$

$a = 6.84$
$b = 1.17$

<u>example</u> :-

$$\frac{L_{90}}{L_{99}} = \left[\frac{\log_{10}\left(\frac{1}{.9}\right)}{\log_{10}\left(\frac{1}{.99}\right)}\right]^{1/1.17}$$

using above formula life of Bearing at any % reliability can be calculated

• for Bearing under cyclic loading

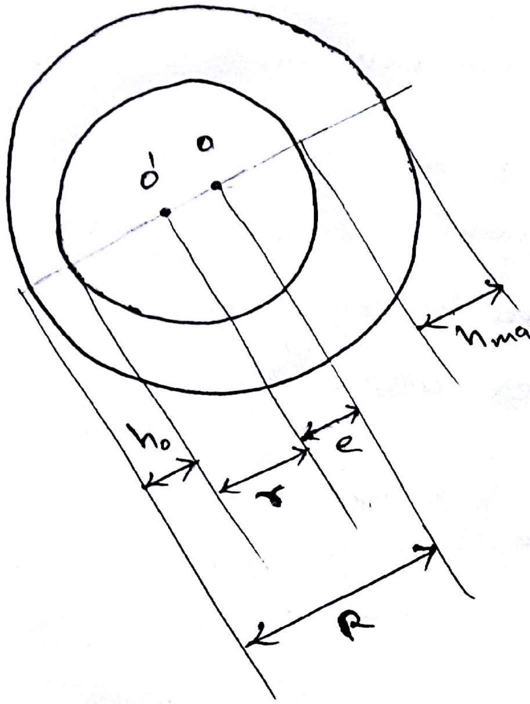
$$P_e = \left(\frac{P_1^n x_1 + P_2^n x_2 - - -}{x_1 + x_2 + - - -} \right)^{1/n}$$

$n = 3 \rightarrow$ ball bearing

$= \frac{10}{3} \rightarrow$ roller bearing

$x =$ no. of ~~relov~~ revolution $=$ N x time

B) Sliding Contact Bearing / Journal Bearing



$R = r + h_o + e$

$R - r = h_o + e$

$c = R - r$ = radial clearance

$2c = 2(R-r) = D - d$

= diametural clearance

$c = h_o + e$

$1 = \frac{h_o}{c} + \frac{e}{c}$

$1 = \frac{h_o}{c} + \varepsilon \quad [\because e = c\varepsilon]$

$\Rightarrow h_o = (1-\varepsilon)c$

$h_{max} = (1+\varepsilon)c$

$\Rightarrow \frac{h_o}{c}$ = minimum oil thickness variable

$$f = \frac{33}{10^8} \times \left(\frac{\mu N}{P}\right) \times \left(\frac{r}{c}\right) + K$$

μ = dynamic viscosity

N = rpm

P = Pressure on bearing $= \frac{W}{Dl}$

K = leakage factor

Reynold's eqn

1) $\frac{r}{c} \cdot f = \psi \cdot \left[\left(\frac{r}{c}\right)^2 \cdot \left(\frac{\mu n_j}{P}\right) \right]$

2) $\frac{r}{c} \cdot f =$ coefficient of friction variable = CFV

$n_j = rps$

3) $\left(\frac{\mu n_j}{P}\right) \cdot \left(\frac{r}{c}\right)^2 =$ Sommerfield's no.

4) $\frac{Q}{r \cdot c \cdot n_j \cdot l} = FV =$ coefficient of flow variable

5) $8.3 \times P \times \left(\frac{CFV}{FV}\right) =$ Tempr rise (°C) of lubricant

⇒ Power loss due to friction = $f(W) \cdot r \cdot \omega$

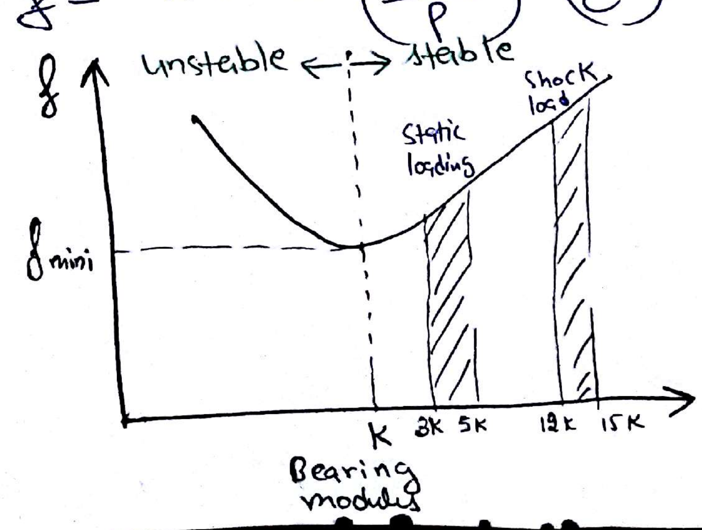
ω = angular velocity
r = radius of journal
W = weight on journal

⇒ Power loss due to viscosity = $\mu \frac{(r\omega)}{h_o} \times (\pi D l) \times r \times \omega$

⇒ $\tan \phi = \mu$; ϕ = friction angle
x = friction radius = $r \sin \phi$

⇒ Petroff's eqn

$f = 2\pi^2 \times \left(\frac{\mu n_j}{P}\right) \times \left(\frac{r}{c}\right)$ ⟶ when $e = 0$

⇒ 

$\frac{\mu n_j}{P}$ = Bearing characteristic number