IITians GATE CLASSES
Exclusive GATE Coaching by IIT/IISc Graduates
A division of PhIE Learning Center

QUICK REVISION

FORMULA SHEET

for

GATE -ME MACHINE DESIGN

Table Of Content

OUR COURSES

GATE Online Coaching

Course Features

- Live Interactive Classes
- E-Study Material
- Recordings of Live Classes
- Online Mock Tests

TARGET GATE COURSE

Course Features

- Recorded Videos Lectures
- Online Doubt Support
- E-Study Materials
- Online Test Series

Distance Learning Program

Course Features

- E-Study Material
- Topic Wise Assignments (e-form)
- Online Test Series
- Online Doubt Support
- Previous Year Solved Question Papers

OUR COURSES

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Course Features

- Topic Wise Tests
- Subject Wise Tests
- Module Wise Tests
- Complete Syllabus Tests

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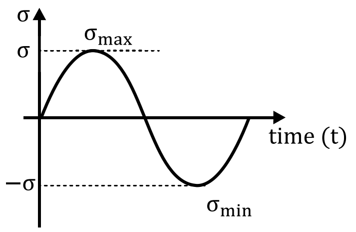
Machine Design

Chapter 1: FATIGUE LOAD

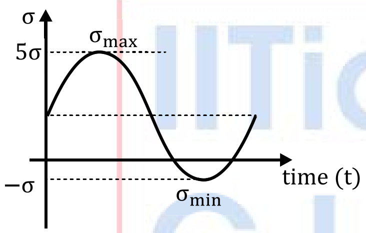
Types of Fatigue (cyclic) stresses

1.



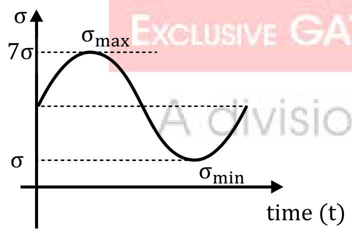
Completely Reversed Stress

2.



Alternating Fatigue Stress

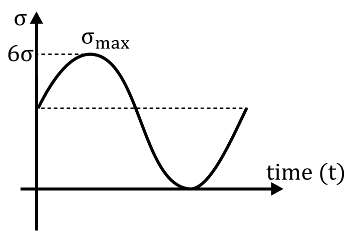
3.



Fluctuating Fatigue Stress

Eg: (Bicycle Spoke)

4.



Repeated Fatigue Stress

Eg: 1. IC Engine; 2. Gear

$$(\sigma_m) = \sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$(\sigma_v) = \sigma_{variable} = \frac{\sigma_{max} - \sigma_{min}}{2}$$

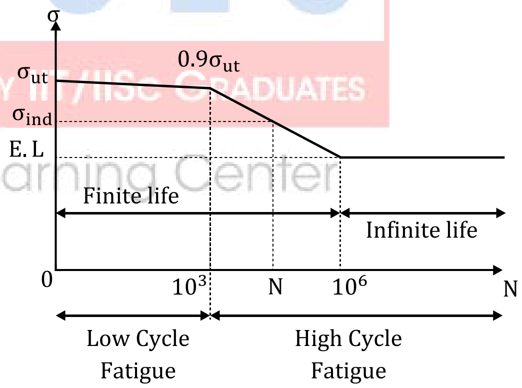
$$\text{Stress Ratio (R)} = \frac{\sigma_{min}}{\sigma_{max}}$$

$$\text{Amplitude ratio (A)} = \frac{\sigma_v}{\sigma_m} = \frac{1-R}{1+R}$$

NOTE: Fatigue test is conducted under completely reversed stress condition.

$$\begin{pmatrix} \sigma_{mean} = 0, \sigma_v = \sigma_{max}, \\ R = -1, A \to \infty \end{pmatrix}$$

S-N Curve for Steel and Ti Specimen:



E.L = Endurance Limit

N = No. of Revolution

If $\sigma_{ind} \le$ E. L → Infinite life ($\ge 10^6$ cycles)

If $\sigma_{ind} >$ E. L → finite life (10^3 to 10^6 cycles)

$$\boxed{\sigma_e = \sigma_e^* K_a K_b K_c K_d K_g}$$

σ_e^* = E. L of a standard specimen

K_a = Size factor

K_b = Surface finish factor

K_c = Reliability factor

$$\boxed{K_d = \frac{1}{K_f}}$$

$$\boxed{K_f = 1 + q(K_t - 1)}$$

$$\boxed{0 \le q \le 1}$$

K_g = load factor

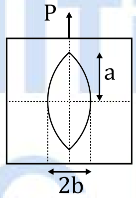
$K_g = 1$ for CRS(Completely reversed Bending load)

$K_g = 0.577$ for torsional loading

$K_g = 0.8$ for axial loading

Theoretical stress concentration factor (K_t)

$$K_t = \frac{\sigma_{max}}{\sigma_{nominal}}$$



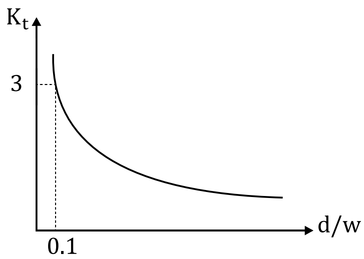
$$K_t = 1 + 2\left(\frac{b}{a}\right)$$

K_f (Fatigue concentration factor)

$$K_f = \frac{\text{E. L without stress concentration}}{\text{E. L. with stress concentration}}$$

q (Notch sensitivity factor)

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



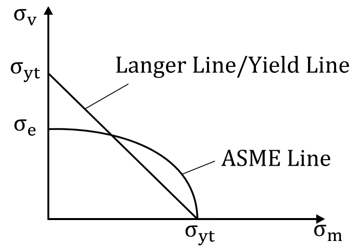
d= diameter of hole

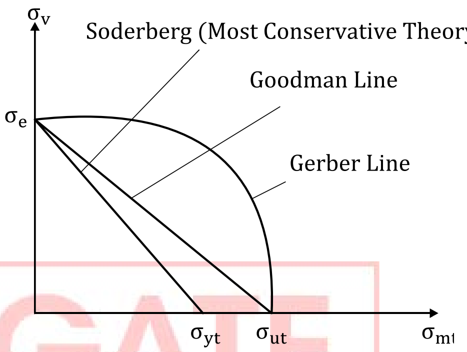
w= width of plate

if $\frac{d}{w} < 0.1$

then $K_t = 3$

If K_t value not given then assume $K_t = 1$





$$\boxed{\frac{\sigma_m}{\sigma_{yt}} + \frac{\sigma_v}{\sigma_e} = \frac{1}{FOS}} \left(\begin{matrix}\text{Soderberg line equation} \\ \text{(For Ductile material)}\end{matrix}\right)$$

$$\boxed{\frac{\sigma_v}{\sigma_e} + \frac{\sigma_m}{\sigma_{ut}} = \frac{1}{FOS}} \left(\begin{matrix}\text{Goodman equation} \\ \text{(For Brittle material)}\end{matrix}\right)$$

$$\left.\begin{matrix}\dfrac{\sigma_v}{\sigma_e} + \dfrac{\sigma_m}{\sigma_{ut}} = \dfrac{1}{FOS} \\ + \\ \dfrac{\sigma_v}{\sigma_{yt}} + \dfrac{\sigma_m}{\sigma_{yt}} = \dfrac{1}{FOS}\end{matrix}\right\} \text{(Modified Goodman)}$$

$$\sigma_e = \frac{\sigma_e^* K_a K_b \ldots}{K_f}$$

$\sigma_m = K_t \times \sigma_m^*$ (for Brittle material)

$$\left(\frac{\sigma_v}{\frac{\sigma_e}{FOS}}\right) + \left(\frac{\sigma_m}{\frac{\sigma_{ut}}{FOS}}\right)^2 = 1 \Rightarrow \text{Gerber line}$$

$$\left(\frac{\sigma_v}{\sigma_e}\right)^2 + \left(\frac{\sigma_m}{\sigma_{yt}}\right)^2 = \frac{1}{(FOS)^2}$$

$\Rightarrow$ ASME ellipsed equation

$$\frac{\sigma_v}{\sigma_{yt}} + \frac{\sigma_m}{\sigma_{yt}} = \frac{1}{FOS} \Rightarrow \text{Langer Line}$$

FOS= Factor of safety

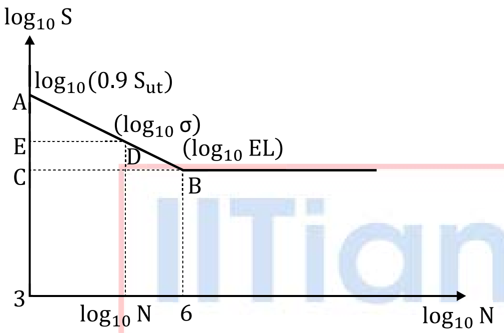
Note:

1. If combined load acting on member $(\tau_{eq} \text{ and } \sigma_{eq})$ then

$$\left.\begin{aligned}(\sigma_{mean})_{eq} &= \sqrt{\sigma_m^2 + 3\tau_m^2} \\ (\sigma_v)_{eq} &= \sqrt{\sigma_v^2 + 3\tau_v^2}\end{aligned}\right\} \text{MDET Theory}$$

2. Surface Roughness $\uparrow \Rightarrow$ Cracks $\uparrow \Rightarrow$ E.L $\downarrow$ $\Rightarrow K_b \downarrow$

$\log S - \log N$ graph $\left(\begin{matrix}\text{Design for} \\ \text{finite life}\end{matrix}\right)$



$\sigma_e = EL =$ Endurance strength/limit

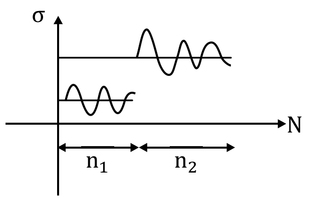
$$\frac{\log_{10}(0.9\ S_{ut}) - \log_{10}\sigma_e}{6-3} = \frac{\log_{10}(\sigma) - \log_{10}\sigma_e}{6 - \log_{10} N}$$

Cumulative damage in fatigue (Miner's Equation):

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \cdots = 1$$

$N_1 =$ No. of stress cycle before fatigue failure when σ_1 acting alone

$N_2 =$ No. of stress cycle before fatigue failure when σ_2 acting alone



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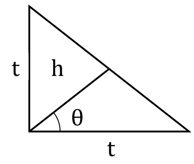
Chapter 2: WELDED JOINT

In GATE Questions

Leg of weld will be given (always) t

$$h = \frac{t}{\cos\theta + \sin\theta}$$



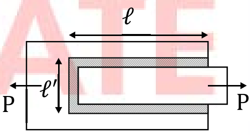
h = throat thickness

Area of fillet weld = h l_e

$l_e = \ell$ (SFWJ)

$l_e = 2\ell$ (DFWJ)

$l_e =$ Effective length of weld



w = Width of plate

t = Thinkness of plate

1. $P = \sigma_t (w \times t) \rightarrow$ Strength of plate
2. $P_1 =$ Parallel fillet joint strength (After welding)

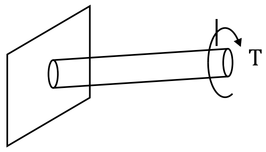
   $P_1 = 2 \times \tau \times (0.707 \times t \times \ell)$
3. $P_2 =$ Transverse fillet weldd joint strength.

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

in place of σ_{t_1}, τ is used in Gate questions

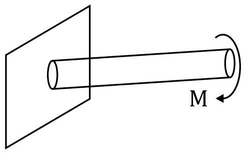
4. $P = P_1 + P_2$

Circular fillet welded joint under torsion.



$$\tau = \frac{T}{2\pi r^2 h} \qquad \boxed{h = \frac{t}{\sqrt{2}}}$$

$$\tau = \frac{2.83\ T}{\pi d^2 t} \quad t = \text{leg of the weld}$$



$$(\sigma_b)_{max} = \frac{5.66\ M}{\pi d^2 t}$$

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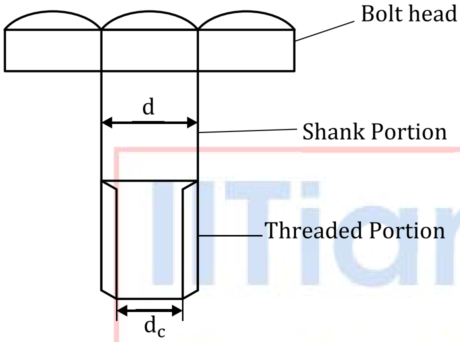
Chapter 3: BOLTED JOINT



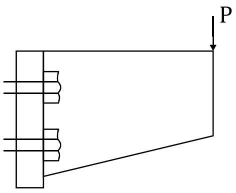
$$d = \frac{d_c}{0.84}$$

d =Major/Nominal diameter

d_c = Core diameter

Eccentric Loaded Bolted Joint:

Case A: d_c used for stress calculation



$$\tau_s = \frac{\frac{P}{n}}{\frac{\pi}{4} {d_c}^2}$$

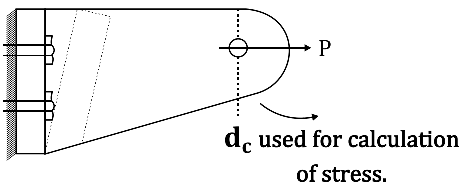
$$(\sigma_t)_{max} = \frac{(P_t)_{max}}{\frac{\pi}{4} {d_c}^2}$$

$$\frac{S_{yt}}{2(FOS)} = \tau_{per} = \frac{1}{2}\sqrt{{(\sigma_t)_{max}}^2 + 4\tau_s^2}$$

From above equation core diameter (d_c) will be known

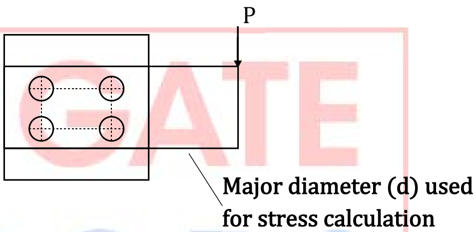
Case B:

primary and secondary tensile stresses induced.



Case C:

Primary and Secondary shear stress induced



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Chapter 4: RIVETED JOINT

Riveted Joint

Case(i) b(width) is given,

1. **Tearing Strength of a Rivet:**

   $P_t = (b - n_R d_h)\ t\ \sigma_t$

   n_R = No. of rivets in a row

   N = Total No. of rivets in a plate

   d_h = Diameter of hole(d+clearance)

   d = Diameter of rivet

   σ_t = permissibile tensile stress

2. **Shear strength of a Rivet:**

   $$P_s = N \times K \times \frac{\pi}{4} d^2 \times \tau_{per}$$

   K = No. of shear

   K = 1 for single strap

   K = 2 for Double strap

τ_{per} = Permissibile shear stress

3. **Crushing Strength of Rivet:**

$P_c = N\,d\,t\,(\sigma_c)_{per}$

t = Thickness of plate

permissible crushing stress

$$\text{Strength of Rivet} = \frac{1}{N}\begin{pmatrix}\text{Minimum of}\\ P_s \text{ and } P_c\end{pmatrix}$$

Stength of solid plate w/o Rivet

$P_{solid} = b \times t \times (\sigma_t)_{per}$

$$\eta_{riveted} = \frac{\text{Minimum of } (P_s, P_c, P_t)}{P_{solid}} \times 100$$

$$\eta_{tearing} = \frac{P_t}{P_{solid}} \times 100$$

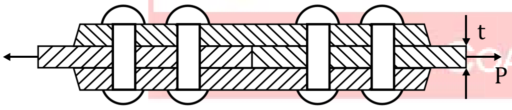
Case(ii) Pitch (p) given

$P_t = (P - d_h)t\,\sigma_t$

$P_c = n \times d \times t \times \sigma_c$

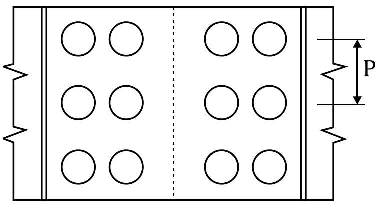
$P_s = n \times k \times \frac{\pi}{4} d^2 \times \tau_{per}$

n = No. of Rivets/Pitch



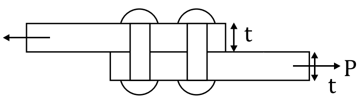
Double riveted double strap butt joint

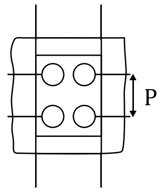
n = 2, k = 2



$\boxed{N = 6}$

Lap Joint





N = 4 n = 2

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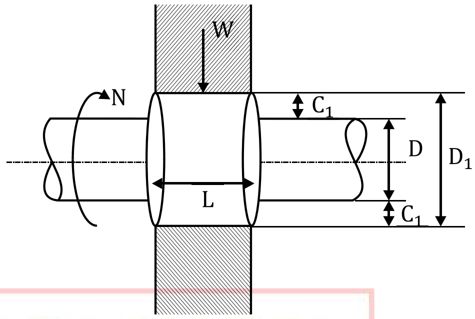
Chapter 5: BEARING



L = length of Journal

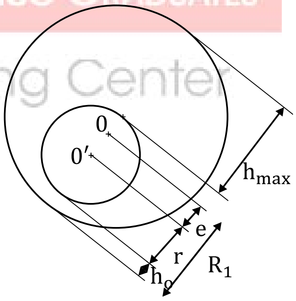
D = Diameter of journal or shaft

D_1 = Diameter of bearing

C_1 = Radial clearance = $R_1 - R$

$C_d = C$ = Diametral clearance $C = 2C_1$

$$p = \text{Bearing pressure} = \frac{W}{L \times D}$$



$R_1 = r + h_o + e$

$R_1 - r = h_o + e$

$C_1 = R_1 - R = h_o + e$

$e = C_1 - h_o$

e = eccentricity

ε(Eccentricity ratio @ Attitude of bearing)

$$= \frac{e}{C_1} = 1 - \frac{h_o}{C_1}$$

μ(Coefficient of friction)

$$= 0.326 \left(\frac{ZN}{p}\right)\left(\frac{D}{C_d}\right) + k$$

N(in rpm); $p(\text{in } N/m^2)$

$Z =$ Viscosity in Pa sec

$k = 0.002$ if $0.75 \le \frac{L}{D} \le 2.8$

$k = 0.003$ if $\frac{L}{D} > 2.8$

Sommerfield No: (no unit)

$$= \frac{Z\,N}{p}\left(\frac{D}{C_d}\right)^2$$

$\mu = 2\pi^2 \left(\frac{Z\,N_s}{p}\right)\left(\frac{r}{C}\right)$ (Pettroff equation)

this is used when $e = 0$

Power loss due to friction $(Q_g) = \mu WV = \mu Wr\omega$

Or

Heat generated due to firction (Q_g)

$\omega =$ angular velocity

r =radius of journal

W = weight on Journal

$Q_d =$ Heat dissipated

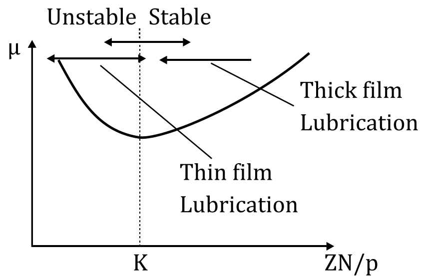
$= C_d \times L \times D_1(t_b - t_a)$

$t_b =$ Bearing temperature

$t_o =$ atmosphere temperature

If $Q_g \le Q_d \rightarrow$ Natural cooling

$Q_g > Q_d \rightarrow$ Artificial cooling



Note:

$\frac{ZN}{p} > K$ (always)

$K =$ Bearing modulus

$3K \le \frac{ZN}{p} \le 5K$ for static load

$13K \le \frac{ZN}{p} \le 15K$ for fatigue/Impact load

$\frac{ZN}{p} =$ Bearing characterstics number

Rolling Contact Bearing:

SKF 6 3 10

First number: Types of Bearing

$3 \rightarrow$ tappered roller

$4 \rightarrow$ needle roller

$5 \rightarrow$ Cylindrical roller

$6 \rightarrow$ Deep groove

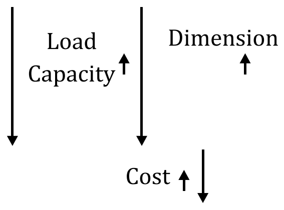
SKF 3: Duty Series

$1 \rightarrow$ extra light

$2 \rightarrow$ light series

$3 \rightarrow$ Medium series

$4 \rightarrow$ Heavy series



Stribeck's Equation:

Static load Capacity/Rating

$$(C_o) = \frac{kzd^2}{5}\begin{pmatrix}\text{Ball}\\ \text{bearing}\end{pmatrix} \text{ or } \frac{k(zL)d}{5}\begin{pmatrix}\text{Roller}\\ \text{Bearing}\end{pmatrix}$$

$z =$ No. of balls

$d =$ diameter of rolling element

$k =$ constant depends on modulus of elasticity of material and radius of curvature of contacting surface

Dynamic Load Rolling (C):

Load on bearing which gives 1 million (10^6) revolution.

Life of bearing (Load – Life Equation)

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

$n = 3$ for BB

$n = \frac{10}{3}$ for RB

Or

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

$C =$ Dynamic load rating (Given by manufacturer)

$P_e = C_s(XVF_r + YF_a)$

$C_s =$ Service factor

$F_r =$ radial load

$F_a =$ Axial load

$V = 1$ for inner race rotation

$V = 1.2$ outer race rotation

Relation between Life and Reliability:

$$\frac{L}{L_{90}} = \left(\frac{\ln\left(\frac{1}{R}\right)}{\ln\left(\frac{1}{R_{90}}\right)}\right)^{1/1.17}$$

BB → Ball bearing

RB → Roller bearing

For Bearing Under Cyclic Loading:

$$P_e = \left(\frac{P_{e_1}^n \cdot n_1 + P_{e_2}^n\, n_2 + \cdots}{n_1 + n_2 + \cdots}\right)^{1/n}$$

Note: Load on bearing not on shaft

$n_1 = N \times \text{time}$

$n_1 =$ No. of revolutions

$n = 3$ for BB

$n = \frac{10}{3}$ for RB

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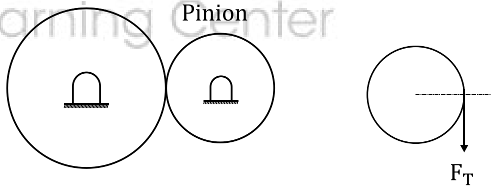
Chapter 6: GEARS (SPUR GEAR)

Gears (Spur Gear):

Design of gear tooth is based on bending moment.



Gear Design (Steps):

According to Beam strength (F_b)

1. **If material is same, then design for pinion.**

   If material is different, then design for weaker

   $\left((\sigma_b)_{per} \cdot Y\right)_{gear}$ and $\left((\sigma_b)_{per}\, Y\right)_{pinion}$

   Smaller of these will be considered for design.

2. **Find Beam Strength**

$F_{t_{max}} = F_b = (\sigma_b)_{per} b\, m\, Y$

$(\sigma_b)_{per}$ = Permissible bending stress at tooth root due to F_t (tangential load)

b = Face width

m = Module

Y = Lewis form factor or tooth geometry factor

$$Y = \pi\left(0.154 - \frac{0.912}{Z}\right)$$

for $\phi = 20°$

ϕ = pressure angle

Z = No. of teeth

3. **$F_{dynamic}$ load Calculation:**

$F_{dynamic} = F_{actual}$

$= F_{static} \times C_V \times S \times FOS$

$$\text{Power} = \frac{2\pi NT}{60}$$

For safety $\boxed{F_{dynamic} \le (F_t)_{max}}$

$$F_{static} = \frac{2T}{D}$$

$F_{static} \times C_V \times S \times FOS \le b\, m\, Y\, (\sigma_b)_{per}$

C_V = Velocity factor

S = service factor

above equation is used when $C_V > 1$

$$C_V = \frac{3+V}{3} \text{ for } V \le 10 \text{ m/sec}$$

$$= \frac{6+V}{V}, \text{ for } V > 10 \text{ m/sec}$$

$$= \frac{(5.6+\sqrt{V})}{5.6} \text{ for } V > 20 \text{ m/sec}$$

$$S = C_s = \frac{\text{Starting torque}}{\text{rated torque}}$$

4. **According to wear strength of tooth (F_w)**

$F_w = D_p Q\, K\, b$

D_p = Diameter of pinion

$$Q = \text{Ratio factor} = \frac{2G}{G \pm 1}$$

+ For external gear

− for internal gear

K = Material combination factor

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

E_p, E_g = Young's modulus for gear & pinion

or

$$K = 0.16\left(\frac{BHN}{100}\right)^2$$

Used when both gears are made of steel and $\phi = 20°$

For safety $F_{act} \le F_{wear}(F_w)$

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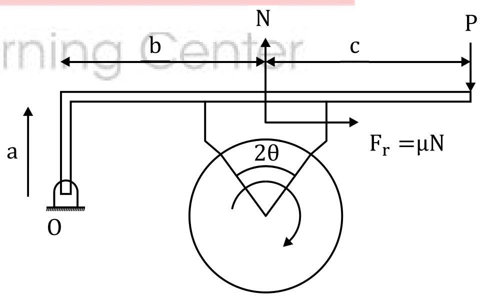
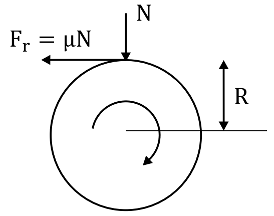
Chapter 7: BRAKES

A. Simple Shoe Brake



P = applied force



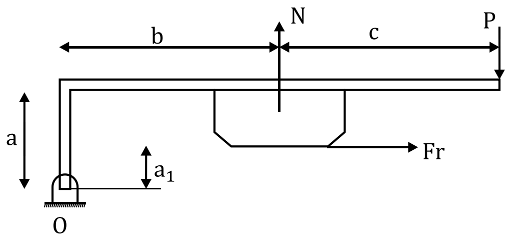
T_f = Frictional torque

$T_f = F_r \times R$

R = Radius of wheel

$\sum M_o = 0$

F_r = Friction force



$$P(b + c) + Fr\,(a_1) = N\,(b)$$

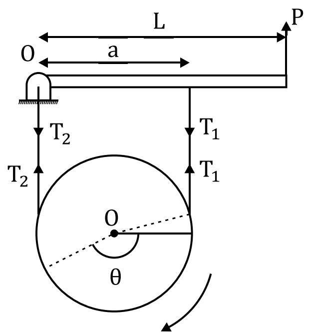
$$P = \frac{N(b) - \mu N(a_1)}{b + c}$$

Note:

1. When applied force moment is acting in the same direction of friction force moment then Brake ⇒ self-energizing brake.
2. When P = 0 Self locked brake
   When P < 0 uncontrollable braking
   When P > 0 controllable braking
3. When $2\theta > 45°$

   Then $\mu' = \dfrac{4\mu \sin\theta}{2\theta + \sin 2\theta}$

B. Simple Band Brake:



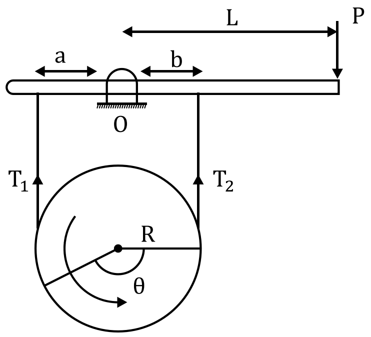
$$\frac{T_1}{T_2} = e^{\mu\theta}$$

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

$$\sum M_o = 0$$

$$T_1(a) = P(L)$$

C. Differential Band Brake:



$\sum M_o = 0$

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

$$\sum M_o = 0$$

$$T_1(a) = T_2(b) + P(\,L\,)$$

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

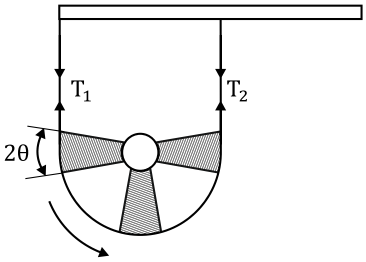
$$\text{Pressure} = \frac{T}{Rw}$$

$$P_{max} = \frac{T_1}{Rw}$$

$$P_{min} = \frac{T_2}{Rw}$$

w = width of band

D. Band and 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

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Chapter 8: FRICTION CLUTCHES /DISC BRAKE

Friction Clutches/Disc Brakes

- **New Brake (Uniform pressure theory):**

$$p = \frac{W}{\pi(r_2^2 - r_1^2)}$$

$$T_f = \mu W R_m$$

$$R_m = \frac{2}{3}\left(\frac{r_2^3 - r_1^3}{r_2^2 - r_1^2}\right)$$

- **Old Brake:(uniform wear theory)**

$$p = \frac{W}{2\pi(r_2 - r_1)r}$$

$$T_f = \mu W R_m$$

$$R_m = \frac{r_1 + r_2}{2}$$

Where p = pressure; T_f = Friction torque

W = axial thrust with which the friction surfaces are held together

above formulas are for single pairing surface , for n pairs multiply T_f by 'n'

Cone Clutch:

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

$$R_m = \frac{2}{3}\left(\frac{r_2^3 - r_1^3}{r_2^2 - r_1^2}\right) \quad \text{(UPT)}$$

$$R_m = \frac{r_1 + r_2}{2} \quad \text{(UWT)}$$

α =Half of cone angle

$$n = n_1 + n_2 - 1$$

No. of effective pairs of surfaces

Centrifugal Clutch:

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

Where n = No. of shoes

$$N = m r_g(\omega_2^2 - \omega_1^2)$$

ω_1 = Speed at which shoes just touches the rim

R = Rim radius

r_g = radius of shoe centre

$N = P_a(l\,b)$

p_a = pressure

l, b = Dimension of friction lining

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