Important Formulas;

Design of Machine Elements

UNIT-1

Normal stress on oblique plaine;

$$\boxed{\sigma_n = \left(\frac{\sigma_x + \sigma_y}{2}\right) - \left(\frac{\sigma_x - \sigma_y}{2}\right)\cos 2\theta - \tau_{xy}\sin 2\theta}$$

Shear stress on oblique plane;

$$\boxed{\tau = \left(\frac{\sigma_x - \sigma_y}{2}\right)\sin 2\theta - \tau_{xy}\cos 2\theta}$$

Principle plane angle;

$$\boxed{\tan 2\theta_p = \frac{2\tau_{xy}}{\sigma_x - \sigma_y}}$$

Maximum or Major Principle Stress;

$$\boxed{\sigma_{max} = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}}$$

Minimum or Minor Principle stress;

$$\boxed{\sigma_{min} = \left(\frac{\sigma_x + \sigma_y}{2}\right) - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}}$$

Maximum shear stress;

$$\boxed{\tau_{max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}}$$

Factor of Safety;

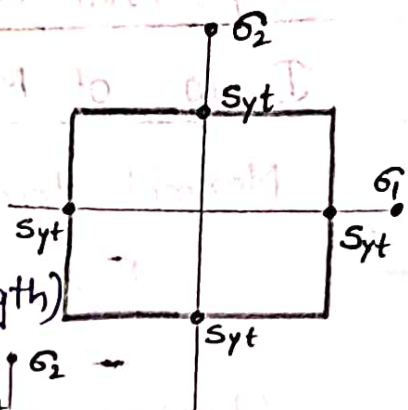
$$\boxed{FOS = \frac{\text{Failure or Max}^m\text{ Stress}}{\text{Working stress}}}$$

Theory of Failure

1. Maximum Principle stress theory,

$$\boxed{\sigma_{max} \leq S_{yt}}$$ (Yield tensile strength)

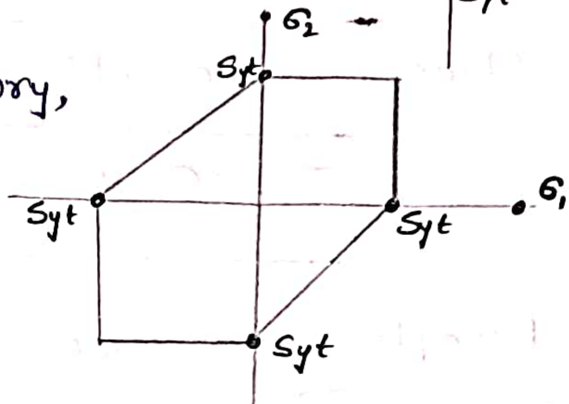
$$\boxed{\sigma_{max} \leq S_{ut}}$$ (Ultimate shear strength)



2. Maximum shear stress theory,

$$\tau_{max} \leq \tau_{permissible}$$

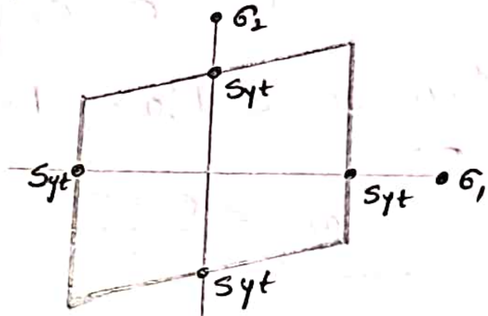
$$\boxed{\sigma_1 - \sigma_2 \leq S_{yt}}$$



3. Maximum principle strain theory (Saint - Venert Theory)

$$\epsilon_{max} \leq \epsilon_{permisible}$$

$$\boxed{\sigma_1 - \frac{\sigma_2}{m} \leq S_{yt}}$$

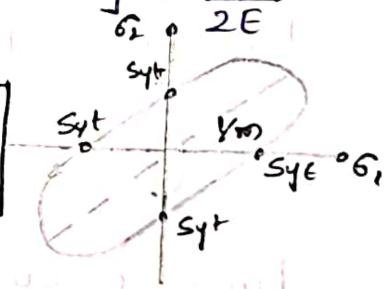


4. Maximum strain energy theory (Haigh theory)

$$U = \frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \frac{2(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)}{m}\right] \leq \frac{S_{yt}^2}{2E}$$

for biaxial,

$$\boxed{\frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 - \frac{2\sigma_1\sigma_2}{m}\right] \leq S_{yt}^2}$$

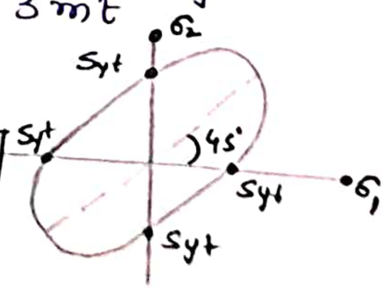


5. Maximum Distortion energy theory (Von-mises theory)

$$\frac{1+m}{6mE}\left[(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2\right] \leq \frac{1+m}{3mE} S_{yt}^2$$

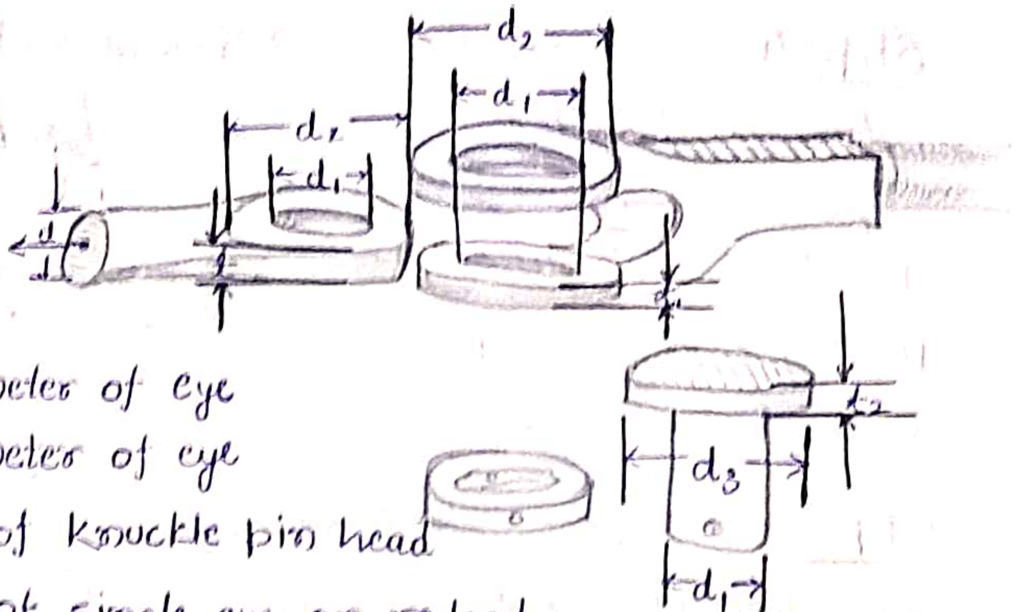
for biaxial,

$$\boxed{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2 \leq S_{yt}^2}$$



UNIT-2

Knuckle Joint :



d - dimension of Circular rod (Diameter)

d_1 - Inner diameter of eye

d_2 - Outer diameter of eye

d_3 - Diameter of knuckle pin head

t - thickness of single eye or rodend

t_1 - thickness of double eye or forkend

t_2 - thickness of knuckle pin head

Step-1 Calculate the diameter of rod, by appling the relation,

$$\sigma_t = \frac{\text{Tensile load}}{\text{Resisting Area}}$$

$$\sigma_t = \frac{P}{\frac{\pi d^2}{4}} , \quad \boxed{d = \sqrt{\frac{4P}{\pi \sigma_t}}}$$

Step-2 Calculate the dimension of knuckle joint assembly parts by these emperical relation;

i). Inner diameter of eye, $\boxed{d_1 = d}$

ii). Outer diameter of eye, $\boxed{d_2 = 2d}$

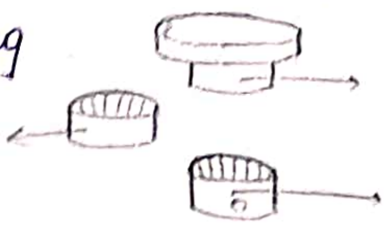
iii). Diameter of knuckle pin head, $\boxed{d_3 = 1.5d}$

iv). Thickness of single eye, $\boxed{t = 1.25d}$

v). Thickness of double eye, $\boxed{t_1 = 0.75d}$

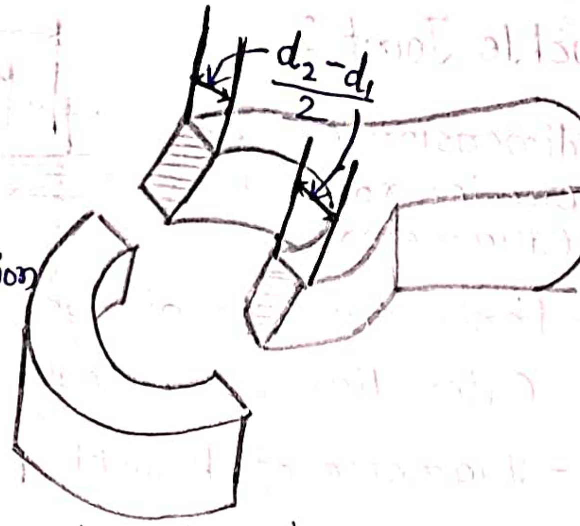
vi). Thickness of knuckle pin head, $\boxed{t_2 = 0.5d}$

Step-3 ~~Calculate the diameter~~ check the dimension of failure of knuckle pin due to shearing



$$\boxed{\tau_{cal} = \frac{2P}{\pi d_1^2}}$$; If $\tau_{cal} \leq \tau_{given}$, then dimension is safe.

Step-4 Check the condition of failure in single eye due to tension.



$$\boxed{\sigma_{t_{cal}} = \frac{P}{t(d_2-d_1)}}$$

If $\sigma_{t_{cal}} \leq \sigma_{t_{given}}$, then dimension is in safe design.

Step-5 Check the condition of failure in single eye due to shearing.

$$\boxed{\tau_{cal} = \frac{P}{t(d_2-d_1)}}$$

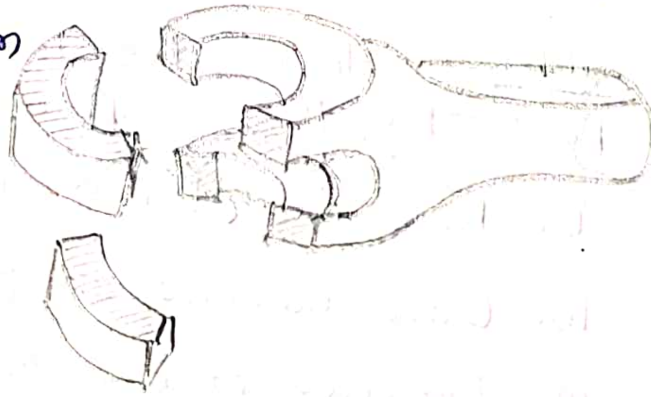
If $\tau_{cal} \leq \tau_{given}$, then dimension is in safe design.

Step-6 Check the condition of failure in single eye due to crushing.

$$\boxed{\sigma_{c_{cal}} = \frac{P}{d_1 \times t}}$$

If $\sigma_{c_{cal}} \leq \sigma_{given}$, then dimension is in safe design.

Step-7 Check the condition of failure due to tension.



$$\boxed{\sigma_{t_{cal}} = \frac{P}{2(d_2-d_1)t_1}}$$

If $\sigma_{t_{cal}} \leq \sigma_{t_{given}}$, then dimension is in safe design.

Step-8 Check the condition of failure due to shearing.

$$\boxed{\tau_{cal} = \frac{P}{2(d_2-d_1)t_1}}$$

If $\tau_{cal} \leq \tau_{given}$, then demesion is in safe design

Step-9 Check the condition of failure due to crushing

$$\boxed{\sigma_{c_{cal}} = \frac{P}{2t_1 d_1}}$$

If $\sigma_{c_{cal}} \leq \sigma_{c\,given}$, then dimension is in safe design.

Step 10 Calculate the total length of knuckle.

$$\boxed{l = 4d}$$

Lever:

Step 1 Calculate the diameter (d) of rod or shaft by appling the relation.

$$\boxed{T = P \times L}$$

where $P \cong 800\,N$ or $400\,N$
L - Effective length of lever.

and from torsional equation,

$$\boxed{T = \frac{\pi}{16} d^3 \tau}$$

Step 2 Calculate the diameter of boss (d_2) and thickness (t_2) and length (l_2) appling relation;

$$\boxed{d_2 = 1.6d}$$

$$\boxed{t_2 = 0.3d}$$

$$\boxed{l_2 = 1.25d}$$

Step 3 Calculate the diameter of shaft at the centre of bearing (d_1)

Calculate the equivalent torque

$$\boxed{T_e = \sqrt{M^2 + T^2}}$$

$$\boxed{T_e = \frac{\pi}{16} d_1^2 \times \tau_{given}}$$

for bending moment
$M = P \times l$, $l = 2l_2$
twisting moment
$T = P \times L$ (given)

Step-4 Design of key ;

- Width of key, $\boxed{w = \frac{d}{4}}$
- thickness of key, $\boxed{t = \frac{d}{6}}$
- length of key, $\boxed{l_1 = l_2}$

or $\boxed{T = \tau w l_1 \times \frac{d}{2}}$ or $\boxed{T = \sigma_c \times t \times l \times \frac{d}{4}}$

Life of Antifriction Bearing ;

$\boxed{\text{Life} = \text{Total number of revolution}}$

or $\boxed{L_{90/10} = \frac{60 N L_{10hr}}{10^6}}$

average life, $\boxed{L_{50} = 5 L_{90}}$

Load Life Relation ;

$\boxed{L_{90} \text{ or } L_{10} = \left[\frac{C}{P}\right]^n}$

where,
L_{90} or L_{10} ~ rated life in Mrev.

$L_{90} \text{ or } L_{10} = \frac{60 N L_{10hr}}{10^6}$

C - dynamic load carring capacity

$n = k = p = 3$ (for ball bearing)
$= 10/3$ (for roller bearing)

Load carrying capacity ;

$\boxed{C = P [L_{10}]^{1/n}}$

P = equivalent load

$\boxed{P = V \times F_r + Y F_a}$; F_r & F_a is given

F_r - Radial load, F_a - Axial load

V - Races rotation factor
= 1, inner races is rotating
= 1.2, outer races is rotating

UNIT-3

Design Procedure of shaft :

Step 1 Calculate the torque transmitted by shaft.

$$\boxed{T = \frac{60P}{2\pi N}}$$

P - Power
N - speed in rpm
T - Torque transmitted

Step-2 Calculate the diameter from torsional equation,

$$\boxed{T = \frac{\pi}{16} d^3 \tau}$$ for solid shaft

$$\boxed{T = \frac{\pi (D^4 - d^4) \tau}{16 D}}$$ for hollow shaft

Step-3 Calculate the diameter of shaft on rigidity critaria,

$$\boxed{T = \frac{G\theta}{l} \times \frac{\pi d^4}{32}}$$ for solid shaft

$$\boxed{T = \frac{G\theta}{l} \times \frac{\pi (D^4 - d^4)}{32}}$$ for hollow shaft

Design Procedure of ractangular sunk key ;

Step-1 Calculate torque transmitted by shaft

$$\boxed{T = \frac{60P}{2\pi N}}$$

Step 2 Calculate diameter of shaft from torsional eqn

$$\boxed{T = \frac{\pi d^3 \tau}{16}}$$

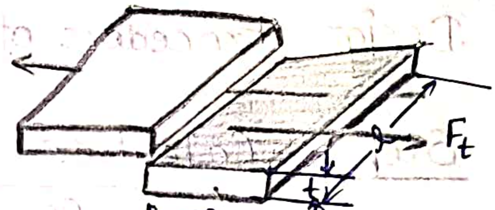
Step 3 Calculate the width and thickness of key

$$\boxed{t = \frac{d}{6}}$$

$$\boxed{w = \frac{d}{4}}$$

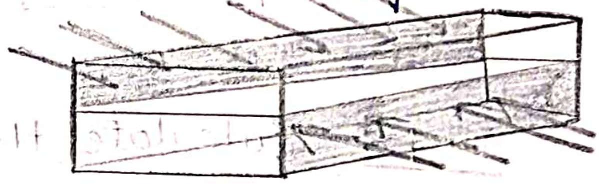
Step 4 Calculate the length of key in shearing,

$$\boxed{T = \tau_{key} \times w \times l \times \frac{d}{2}}$$



Step 5 Calculate the length of key in crushing,

$$\boxed{T = \sigma_c \times t \times l \times \frac{d}{4}}$$



Shaft strength factor (e), "Ratio of strength of shaft with keyway to strength of shaf without keyway."

$$\boxed{e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)}$$

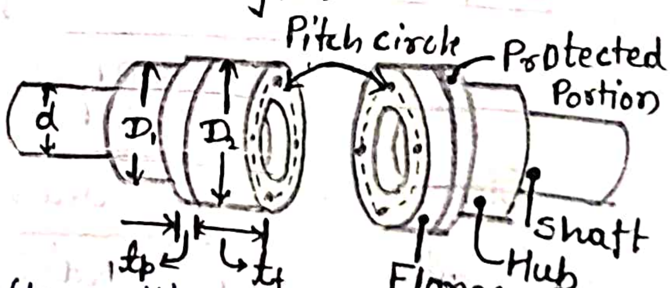
Equivalent Torque,

$$\boxed{T_e = \sqrt{M^2 + T^2}}$$

Equivalent moment

$$\boxed{M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]}$$

Design of Flange Coupling (Protected Type)



Let,

d = diameter of shaft

D_1 = Diameter of hub

D_2 = Outer diameter of flange ~~top~~ coupling

t_f = thickness of flange

t_p = thickness of protected portion.

A. Design of shaft and hub,

Step 1 Calculate torque transmitted by shaft,

$$\boxed{P = \frac{2\pi N T}{60}} \rightarrow ?$$

Step 2 Calculate the diameter of shaft,

i). from torsional strength,

$$\boxed{T = \frac{\pi}{16} d^3 \tau}$$

ii). from rigidity critaria,

$$\boxed{T = \frac{G\theta}{l} \times \frac{\pi}{32} d^4}$$

Step 3 Calculate the inner diameter of hub,

$$\boxed{d_i = d}$$

Step 4 Calculate the outside diameter of hub,

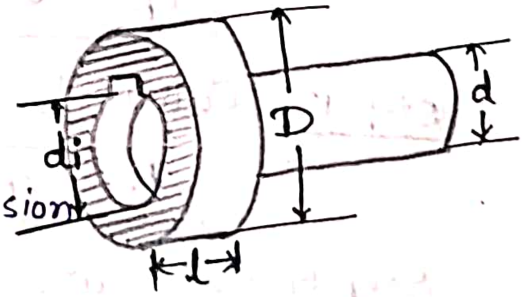
$$\boxed{D = 2d}$$

Step 5 Calculate the length of hub,

$$\boxed{l = 1.5d}$$

Step 6 Check the dimension of hub, assuming fail in shearing,

$$T = \frac{\pi}{16}\left(\frac{D^4 - d_1^4}{D}\right) \tau_{Hub}$$



If $\tau_{Hub} < \tau_{Hub}$ (given) then dimension is in safe design.

B. Design of Key.

Step 7 Calculate width and thikness of key,

$$\boxed{w = \frac{d}{4} \ \& \ t = \frac{d}{6}}$$ for ractangular key

$$\boxed{w = t = \frac{d}{4}}$$ for square key

Step - 8 Calculate the length of key

$$\boxed{l_k = l_H = 1.5\,d}$$

Step-9 Check shear stress induced in key,

$$\boxed{T = \tau_{key} \times w \times l \times \frac{d}{2}}$$

If $\tau_{key} < \tau_{key}$(given) then dimension is in safe design.

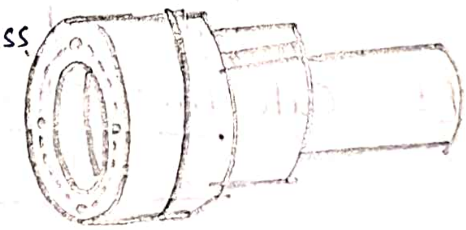
Step 10 Check the condition of failure in key due to crushing,

$$\boxed{T = \sigma_c \times t \times l \times \frac{d}{2}}$$

If $\sigma_c < \sigma_c$(given) then dimension is in safe design.

C. Design of flange portion;

Step 11 Calculate the thickness of flange (t_f)



$$\boxed{t_f = 0.5d}$$

Step 12 Calculate the thickness of protected portion,

$$\boxed{t_p = 0.25\,d}$$

Step 13 Calculate the outside diameter of flange,

$$\boxed{D_2 = 4\,d}$$

Step 14 Calculate the pitch circle diameter,

$$\boxed{D = 3d}$$

Step 15 Check the shear stress induced in flange,

$$\boxed{T = \tau_{flange} \times \pi D \times t_f \times D}$$

If $\tau_{flange} < \tau_{flange}$(given) then the dimension are in safe design.

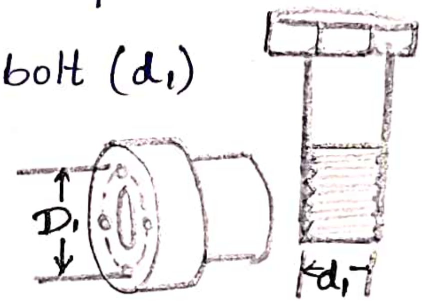
D. Design of nut and Bolt,

Step 16 Calculate the number of bolts according to international standard (I.S. 1990-1996)

Dia. of shaf	upto 40mm	upto 100 mm	upto 180mm
no. of bolts	3	4	5

Step 17 Calculate the diameter of bolt (d_1)

$$\boxed{T = n \times \frac{\pi}{4} d_1^2 \times \tau \times \frac{D_1}{2}}$$



Gear Terminology

- Module $= \dfrac{\text{Pitch circle diameter}}{\text{No. of Teeth}} = \dfrac{D}{T} = \dfrac{2R}{T}$

- Diametral Pitch $= \dfrac{\text{No. of Teeth}}{\text{Pitch circle dia}} = \dfrac{T}{D}$; $P_d = \dfrac{1}{m}$

- Circular Pitch, $P_c = \dfrac{\pi D}{T} = \pi m$; $P_c \times P_d = \pi$

Lewi's equation,

Tangential force on tooth,

$$\boxed{F_t = \pi \sigma_b \times b \times m \times y}$$

where y - Lewi's form factor

$$\boxed{y = \frac{t^2}{6h P_d}}$$

UNIT-4

Spring Terminology:

1. Solid length,

$$\boxed{L_s = nd}$$

2. Free length,

$$\boxed{L_f = L_s + \delta_{max} + 0.15\,\delta_{max}}$$

3. Spring Index,

$$\boxed{C = \frac{D}{d}}$$

4. Spring rate/Spring constant/stiffness (k),

$$\boxed{k = \frac{W}{\delta}}$$

5. Pitch,

$$\boxed{p = \frac{\text{Free length}}{n-1} = \frac{L_f - L_d}{n} + d}$$

Stress developed in helical spring:

Torsional shear stress induced in wire,

$$\boxed{\tau_1 = \frac{8WD}{\pi d^3}} \quad \text{or} \quad \boxed{\tau_1 = \frac{8WC}{\pi d^2}}$$

Direct shear stress developed in helical spring,

$$\boxed{\tau_2 = \frac{4W}{\pi d^2}}$$

Resultant shear stress induced in wire,

$$\boxed{\tau = \tau_1 \pm \tau_2}$$

Maximum shear stress induced in wire,

$$\boxed{\tau_{max} = \frac{8WD}{\pi d^3}\left[1 + \frac{d}{2D}\right] = \frac{8WD}{\pi d^3}\left[1 + \frac{1}{2C}\right] = \frac{8WD}{\pi d^3} k_s}$$

where $\boxed{k_s = 1 + \frac{1}{2C}}$ k_s - Shear stress factor

Including curvature stress,

$$\boxed{\tau_{max} = k\frac{8WD}{\pi d^3}}$$

where k - Wahl's factor

$$\boxed{k = \frac{4c-1}{4c-4} + \frac{0.615}{c}}$$

Deflection in spring:

1. Total active length of wire,

$$\boxed{l = \pi D n}$$

2. Axial deflection of spring,

$$\boxed{\delta = \theta \times \frac{D}{2}}$$

$$\boxed{\delta = \frac{16WD^3 n}{Gd^4}}$$

Deflection per unit active coil,

$$\boxed{\frac{\delta}{n} = \frac{8WD^3}{Gd^4}}$$

3. Spring constant

$$\boxed{k = \frac{Gd^4}{8D^3 n}} \quad \text{or} \quad \boxed{k = \frac{Gd}{8c^3 n}} \quad \boxed{c = \frac{D}{d}}$$

Energy stored in helical spring of circular wire:

$$\boxed{U = \frac{\tau^2}{4k^2G} \times \left(\pi DN \times \frac{\pi d^2}{4}\right)}$$

[Area × length = Volume

$\frac{\pi d^2}{4} \times \pi DN$ = Volume of spring]

$$\boxed{U = \frac{\tau^2}{4k^2G} \times \text{Volume of spring}}$$

Series connection of spring,

(i) $W = W_1 = W_2$

(ii) $\delta = \delta_1 + \delta_2$

$\therefore \frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2}$

$$\boxed{K = \frac{K_1 K_2}{K_1 + K_2}}$$

Parallel connection of spring,

(i) $W = W_1 + W_2$

(ii) $\delta = \delta_1 = \delta_2$

$\therefore \boxed{K = K_1 + K_2}$

Terminology of Power Screw

1. Nominal Diameter

$$\boxed{D_o = D + P}$$

2. Core diameter

$$\boxed{D_c = D - P}$$

3. Mean diameter

$$\boxed{D = \frac{D_o + D_c}{2}}$$

4. Pitch, P = Central Distance between two adjecent thread

5. Lead,

$$\boxed{l = nP}$$

n- no. of thread

P- Pitch

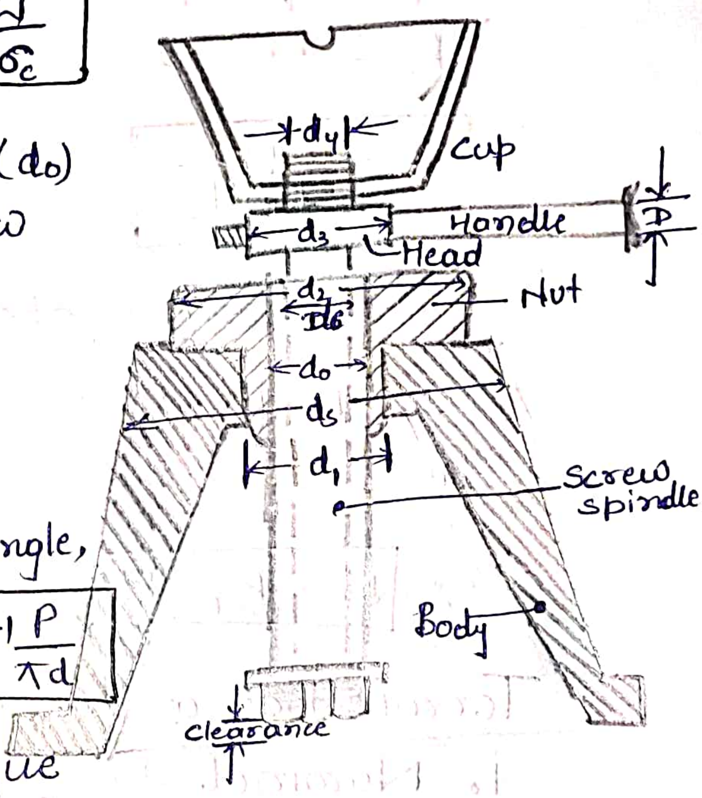
6. Helix angle (α),

$$\boxed{\tan\alpha = \frac{l}{\pi d}}$$

Design procedure of screw jack:

Step-1 Calculate the core diameter (d_c) of screw spindle

$$\sigma_c = \frac{W}{\frac{\pi}{4} d_c^2} \; ; \; \boxed{d_c = \sqrt{\frac{4W}{\pi \sigma_c}}}$$



Step-2 Calculate outer dia (d_o) and mean dia (d) of screw spindle

Outer dia., $\boxed{d_o = d_c + P}$

Mean dia., $\boxed{d = \frac{d_o + d_c}{2}}$

Step-3 Calculate the helix angle,

$$\tan\alpha = \frac{P}{\pi d} \; ; \; \boxed{\alpha = \tan^{-1}\frac{P}{\pi d}}$$

Step-4 Calculate the torque

$$T_1 = W \tan(\alpha + \theta) \times \frac{d}{2}$$

Step-5 Calculate the compressive stress

$$\boxed{\sigma_c = \frac{4W}{\pi d_c^2}}$$

Step-6 Calculate the shear stress

$$\boxed{\tau = \frac{16 T_1}{\pi d_c^3}}$$

Calculate the principle stress

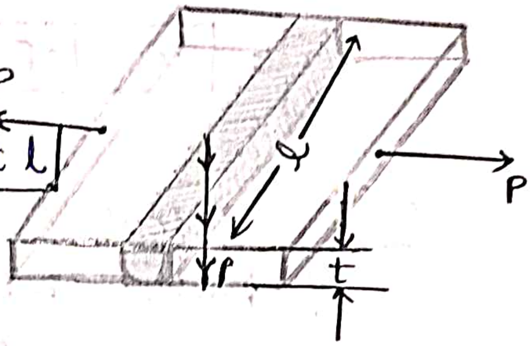
$$\boxed{\sigma_{max} = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}}$$

UNIT-5

Strength of butt joint ;

When tensile or compressive force/stress is given,

$$\sigma_t \text{ or } \sigma_c = \frac{P}{t \times l}$$



$$\boxed{P = \sigma_t (t \times l)} \quad \text{or} \quad \boxed{P = \sigma_c \, t \, l}$$

When shear strength is given,

$$\tau_{max} = \frac{P}{t \times l}$$

$$\boxed{P = \tau_{max} (t \times l)}$$

1. Transverse fillet joint ;

If shear stress is given,

for maximum load (single transverse fillet),

$$\boxed{P = 0.828 \, h \times l \times \tau_{max}}$$

for maximum load in double transverse fillet,

$$\boxed{P = 2 \times 0.828 \, h \times l \times \tau_{max}}$$

If tensile stress is given,

for maximum strength,

$$\boxed{P = 2 \times 0.707 \times h \times l \times \sigma_t}$$

2. Parallel or longitudnal fillet joint ;

Max^m load acting on single parallel fillet,

$$\boxed{P = 0.707 \, h \times l \times \tau_{max}}$$

Max^m load acting on single parallel fillet,

$$\boxed{P = 2 \times 0.707 \, h \times l \times \tau_{max}}$$

$$\boxed{P_{tranverse} = 1.17\, P_{parallel}}$$

Design Procedure of Rivet Joint;

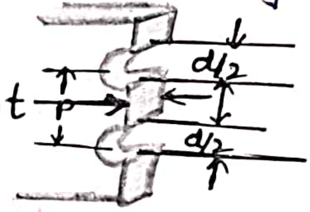
Step-1 Calculate the margin by applying the relation,

$$\boxed{m = 1.5d}$$

d- dia of rivet

$$\boxed{d = 6\sqrt{t}}$$

Step-2 Calculate the tearing strength (tensile) strength of plate by applying the relation,



$$\sigma_t = \frac{P_t}{(P-d)t}$$

P- Pitch

$$\boxed{P_t = \sigma_t\, t\, (P-d)}$$

If $P_t > P$(given)[Applied Load] then dimension are in safe condition.

Step 3 Calculate the shearing resistance of the rivet,

$$\boxed{P_s = n \times \frac{\pi}{2} d^2\, \tau_{given}}$$ for double shear

$$\boxed{P_s = n \times 1.875 \times \frac{\pi}{4} d^2\, \tau_{given}}$$

Step 4 Calculate the crushing resistance of rivet joint,

$$\boxed{P_c = n\, d\, t\, \sigma_c}$$

Step 5 Calculate efficiency of rivet joint

$$\boxed{\eta = \frac{\text{least of } P_t \text{ or } P_s \text{ or } P_c}{\text{Strength of solid plate without hole (P)}}}$$

$P_t = \sigma_t\, t\, (P-d)$
$P_s = n \frac{\pi}{2} d^2\, \tau_{given}$
$P_c = n\, d\, t\, \sigma_c$
$P = p \times t \times \sigma_t$

for butt joint,

Step 6 Calculate the thickness of cover plate by the relation,

$\boxed{t_1 = 1.125\, t}$ for single cover

$\boxed{t_1 = 0.625\, t}$ for double cover