Design: It is the process of finalisation of shape, material and dimensions.

Finalisation of materials & Dimensions :

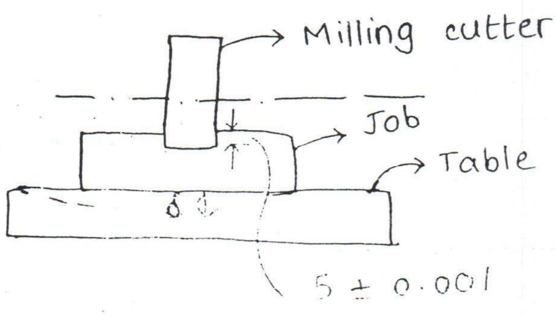
Dimensions of a component are fixed in such a way that critical parameter called "Design Parameter", induced due to applied loads is less than the allowable value of a material.

Design Parameters :

- ✓ Material should have sufficient strength. It can be determined by calculating its "stress." If design is done using strength, it is called as "Strength based Design."
- ✓ Rigidity : Deformation / Deflection per unit length. If the dimensions are fixed according to rigidity, it is called as "Rigidity Based Design." The deflection per unit length value should not exceed the allowable value.

(E.M & S.O.M)

Ex: Machine Tool Beds.



The deflection should be less than the tolerance introduced on the dimensions.

$$\delta = \frac{PL^3}{48EI} \leq 0.001$$

✓ Wear rate : Wear can be defined as the material removed per unit time.

Ex : Gears, Brakes, Clutches, Bearings, etc.

If this is the criteria for design, it is called as "Wear Based Design."

Tribology → science related to friction, lubrication & wear

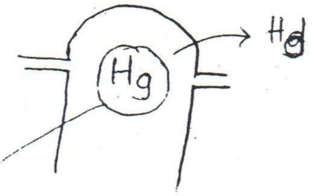
✓ Life : It is the number of cycles completed before failure of the component.

Ex : Bearings, Aircraft applications.

✓ Fatigue, fracture and Creep

✓ Heat Transfer Rate : Heat transferred per unit time across the component.

Ex : Condensers, Heat pipes, Fins of an IC engine.



$$H_d = C A \Delta T$$

If $H_d < H_g$, acumulation of heat takes place, which rises the temperature, which also decreases yield stress of the component

H_g = Heat Generated

H_d = Heat Dissipated.

✓ It is related to Thermodynamics, Fluid Mechanics and Heat Transfer

* Allowable Value : Found using Lab tests, which are conducted on :

(i) standard shape and size.

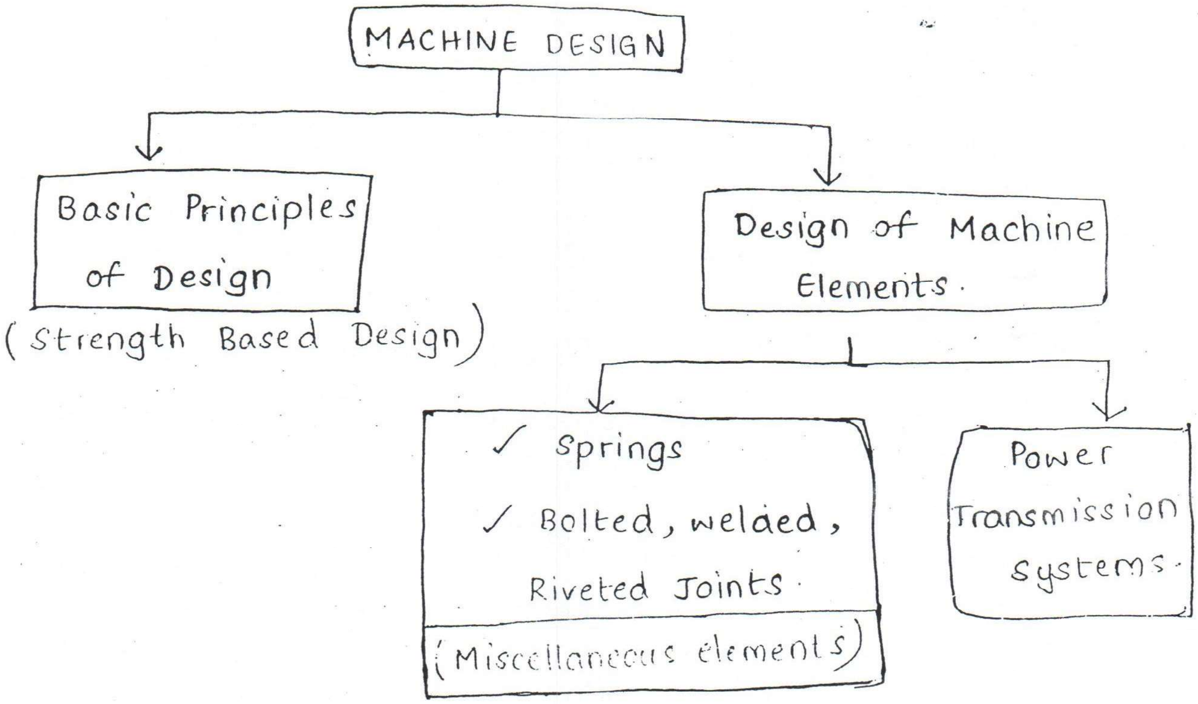
(ii) standard Loads → single loads
→ Gradually applied

Tension Test, Torsion Test, Hardness Test, Impact test, Fatigue Test,

→ constant with time.

(iii) shape is uniform and continuous.

(iv) Room Temperatures are used.



→ In power Transmission systems, power is transmitted, modified and controlled. The elements of these systems are

✓ Shafts.

✓ Bearings

✓ Gears / Belts / chains

✓ Brakes

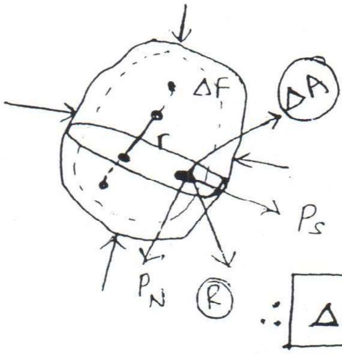
✓ clutches.

BASIC PRINCIPLES OF DESIGN :

- STRENGTH BASED DESIGN

tress : Internal Resistance per unit area.

ubstance : collection of molecules which are held by the intra-molecular forces.



$F \propto \frac{1}{r^2}$

$F' \propto \frac{1}{r'^2}$ $\boxed{F' > F}$

$\therefore \boxed{\Delta F = F' - F}$

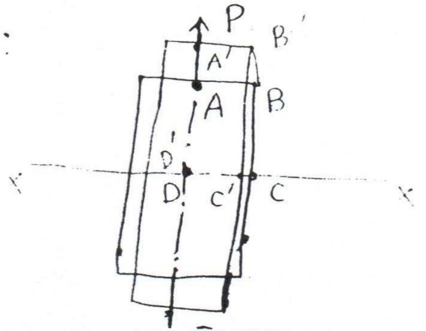
When load is applied, internal particles will relocate to new positions, which change the intra-molecular forces. This intra-molecular force develops the internal resistance.

External force is opposed by change in the intra-molecular force, due to re-location of molecules.

$\therefore$ A force causes "deformation."

The change in intra-molecular force $\propto \frac{1}{r^2}$ along the distance between two molecules.

ension Test :



The applied load may cause molecules to relocate in diff. directions but within the same body..

→ Stress is always calculated over a point where the internal resistance is maximum.

→ Change in internal resistance varies from point to point in same object, subjected to same load.

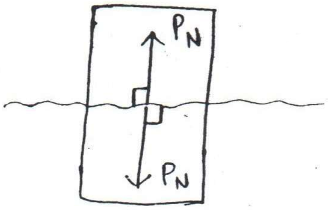
→ The internal resistance is maximum for a molecule which moves greater distances among all molecules in an object.

∴ Search for a point where resistance is maximum. That point is called critical point = Heavily stressed point.

Modes of Failure of a Material:

→ Modes of material separation.

① Brittle mode - Tearing mode - Tensile Mode:



∴ Normal Load = P_N

⇓

perpendicular to Resisting cross-section.

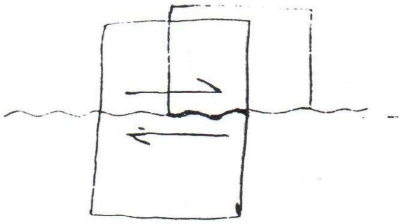
→ surfaces move perpendicular to each other.

→ Failure is sudden.

② Ductile mode - sliding mode - shear mode:

→ surfaces move parallel to each other.

→ shear load is parallel to surface



→ Failure is gradual (or) gives an indication before failure.

ormal stress : σ (sigma)

$$\boxed{\sigma = \lim_{\Delta A \to 0} \frac{P_N}{\Delta A} = \frac{P_N}{A}}$$ if Load is distributed uniformly.

hear stress : τ (Tau)

$$\boxed{\tau = \lim_{\Delta A \to 0} \frac{P_s}{\Delta A} = \frac{P_s}{A}}$$ if Load is distributed uniformly.

Review of Basics of S.O.M :-

Types of Loads :

I) Linear Loads : (loads acting along a Line)

- ✓ Normal loads.
- ✓ Shear loads.
- ✓ Eccentric load

) Angular Loads : (Rotation about an axis)

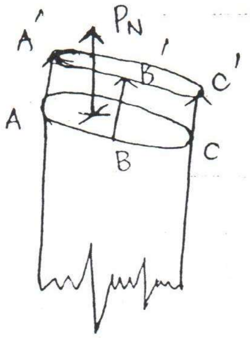
- ✓ Bending Moment
- ✓ Twisting Moment.

Critical Resisting Cross-section :

- ✓ Has minimum area.
- ✓ subjected to maximum Loads

Effect of Normal Load :

- ✓ Load is perpendicular to Critical Resisting Cross-section
- ✓ Passes through C.G. of Critical Resisting Cross-section (CRCS)



Every point on the cross-section moves by the same distance in the direction of applied normal load.

✓ Induced stress is the normal stress perpendicular to the cross-section.

✓ Stress is same at every point on the cross-section.

✓ Normal stress = $\boxed{\sigma = \frac{P_N}{A}}$

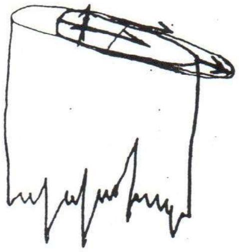
✓ Direction is same as applied Load.

✓ Tensile stress – Length increases

compressive stress – Length decreases.

② Effect of Shear Load :

✓ Load is parallel to critical resisting cross section. and passes through C.G.



✓ Induced stress is shear stress parallel to area in the direction of applied Load.

✓ Stress at every point is same.

✓ shear stress $\boxed{\tau = \frac{P_s}{A}}$

③ Effect of Bending Moment :

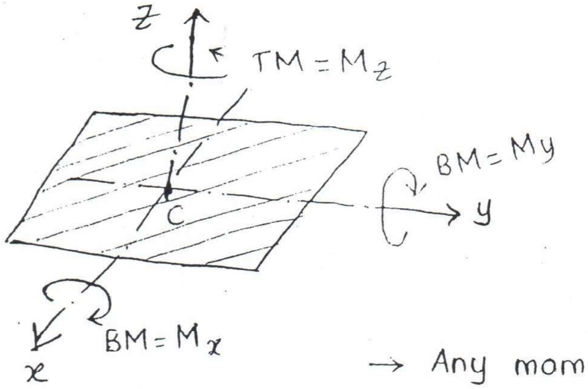
✓ Bending moment is rotation ⊥r to resisting cross-section

(or) Moment axis in the plane of Resisting Cross-section

→ Moment axis is found by Right Hand Thumb Rule.

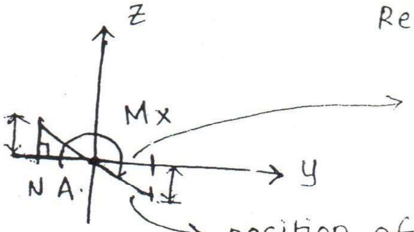
∴ Folded Fingers = Rotation.

Thumb = Moment Axis.



Neutral Axis: Moment Axis passing through CG is called Neutral Axis.

→ Any moment causes rotation of Resisting Cross-section about Neutral axis.



Plane before M_x is applied.

position of plane after M_x is applied.

Flexural Equation is $\boxed{\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}}$

✓ M and I must be about same axis.

✓ y = Distance from Neutral Axis.

✓ Induced stress is "Normal Stress", also called as "Bending Stress."

✓ Stress at a point ∝ "y"

$$\therefore \sigma = \frac{M}{I} \times y \Rightarrow \sigma \propto y.$$

✓ Maximum stress occurs at points which are far away from Neutral Axis.

→ Max. stress = $\sigma_{max} = \frac{M}{I} \times y_{max} = \frac{M}{I/y_{max}} = \frac{M}{Z}$ ⑥

Z = section modulus.

R.C.S (Resisting Cross Section)	σ_{max}
① Solid circular	$\frac{32 M}{\pi d^3}$
② Hollow circular $K = d/D$	$\frac{32 M}{\pi D^3 (1-K^4)}$
③ Rectangle	$\frac{6M}{bd^2}$
④ Square	$\frac{6M}{a^3}$

✓ Stress is same on a line parallel to the Neutral Axis.

④ Effect of Twisting Moment :

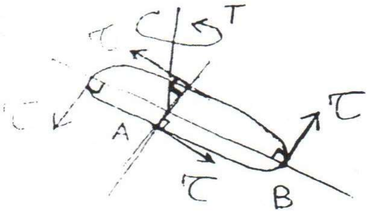
✓ Moment axis is $\perp^r$ to R.C.S. (or) rotation is parallel to Resisting Cross-section.

✓ Flexural Equation is $\boxed{\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L}}$

✓ Induced stress is "Shear stress"

✓ Stress at any point is $\propto$ "r"

r = distance from C.G. of Resisting Cross Section



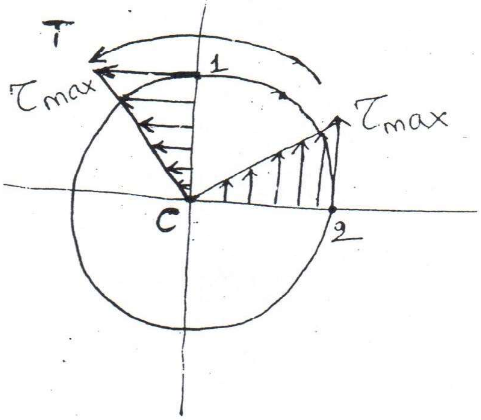
✓ Direction of shear stress is, perpendicular to radius. In same sense as applied Torque.

$$\tau_{max} = \frac{T}{J} \times r_{max} = \frac{T}{J/r_{max}} = \frac{T}{Z_p}$$

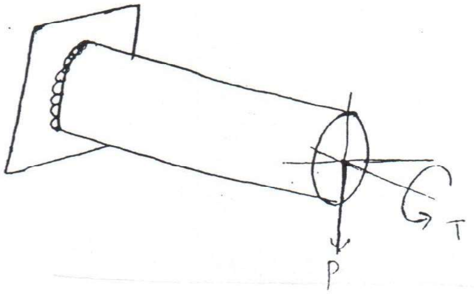
Z_p = Polar Section Modulus.

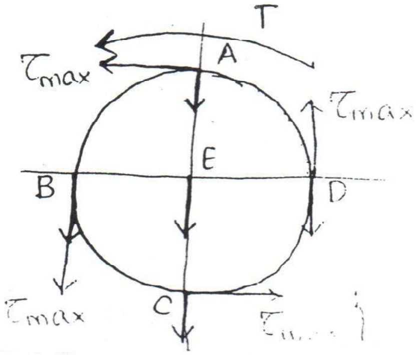
R.C.S	τ_{max}
① Solid Circular	$\frac{16T}{\pi d^3}$
② Hollow circular $k = d/D$	$\frac{16T}{\pi D^3(1-k^4)}$

There is no point where the stress is same.



At point 1 and 2, the magnitude of shear stress is same but, the direction is different.





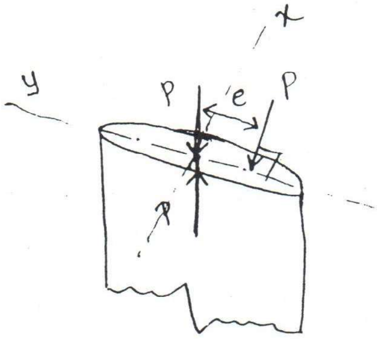
B → Heavily stressed point

D → Least stressed point.

⑤ Eccentric Loads :

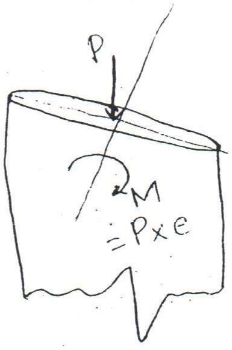
→ Normal / shear load passing away from C.G. of Area.

Normal Eccentric :



Hint: Add two equal and opposite forces at C.G, same as applied force.

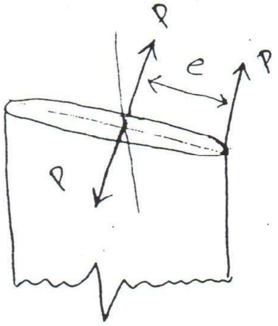
$e = \perp^r$ distance between CG and load

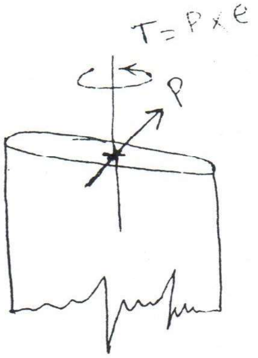


$$\boxed{\text{Eccentric load} = \text{Axial Load} + \text{Moment}}$$

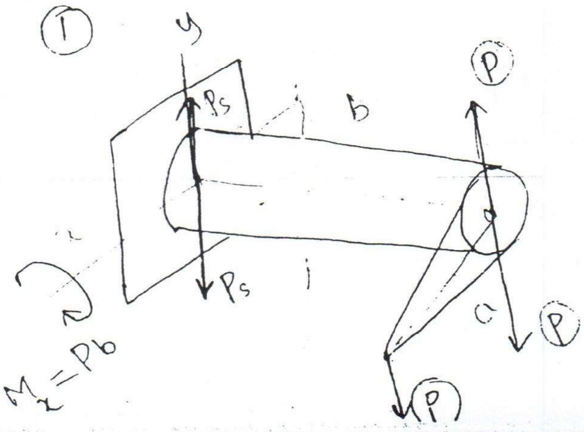
= combined Loads.

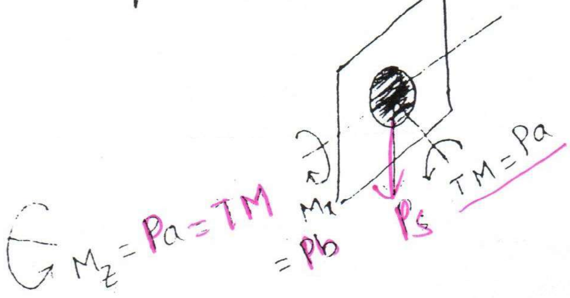
Shear Eccentric :



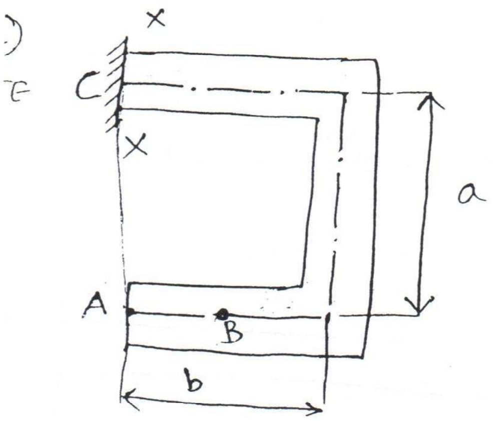


①





Hint : In a cantilever beam, critical Resisting Cross section is Fixed section.



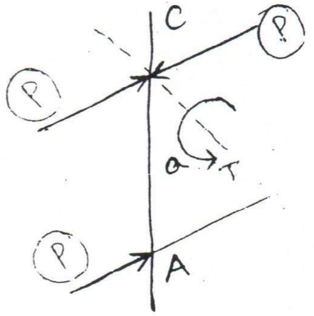
A' Bracket as shown in fig. is subjected to an axial force perpendicular to the plane of board at point 'A'. The loads induced in the section X X is ______

① Direct shear ② Direct axial

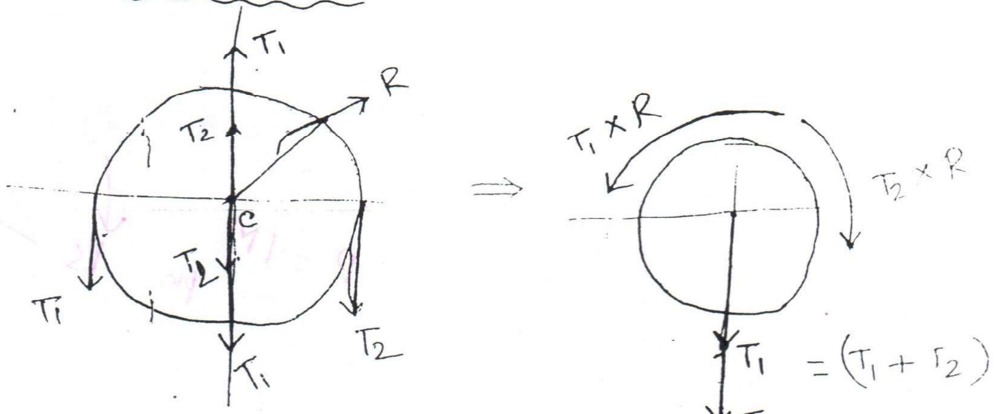
③ Bending ④ Twisting

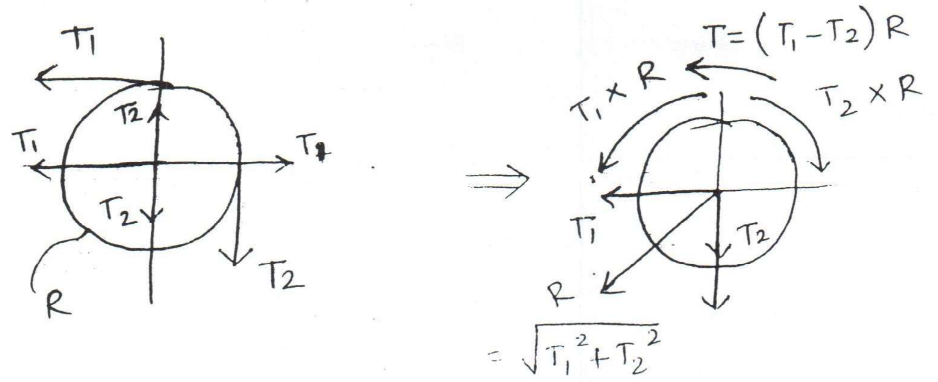
(a) 1 & 3 (b) 2 & 3 (c) 1 & 4 (d) 1,2,3,4

(In case load is applied at B)

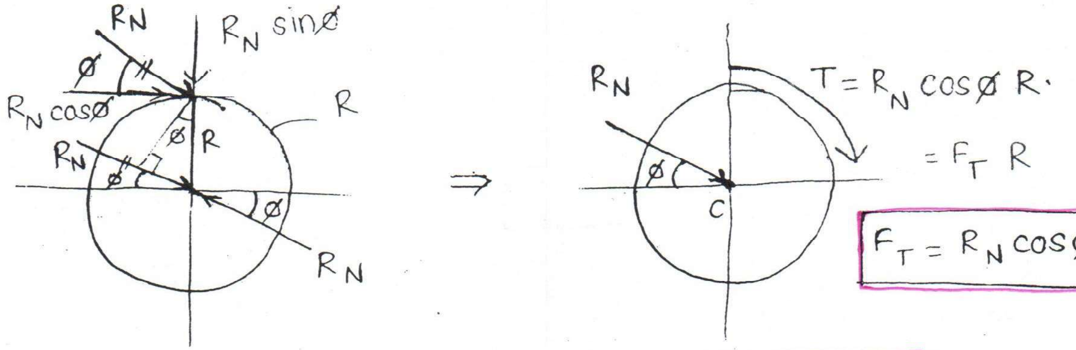


) Belt Drive / Pulley :





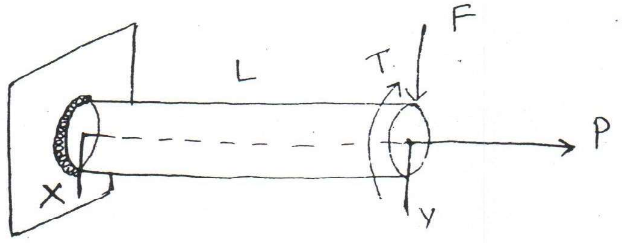
④ Gear Drive :

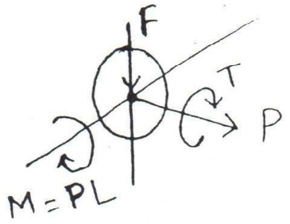


$$\boxed{F_T = R_N \cos\phi}$$

Radial Load on the Gear is $\boxed{R_N = \dfrac{F_T}{\cos\phi}}$

GATE-16
⑨
SET-2





A machine element fixed at end X is subjected to axial load P, transverse load F, and a Twisting moment T at the free end Y. The most critical point from the strength point of view

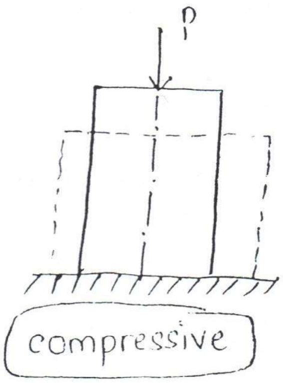
(a) A point on circumference at location Y

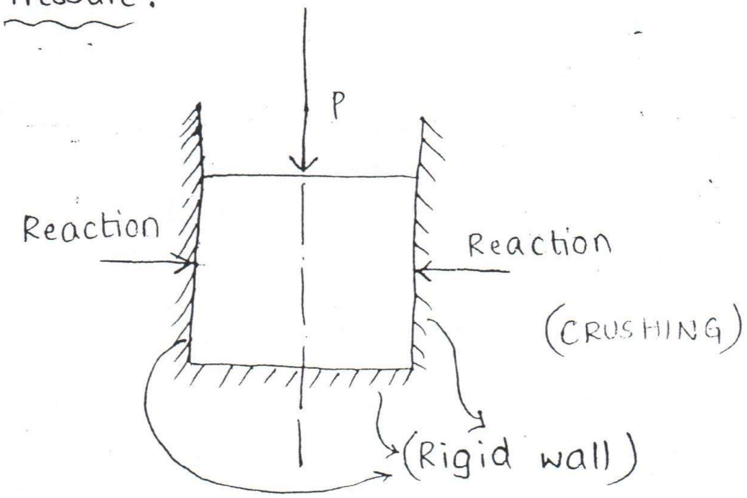
(b) a point on the centre at location Y

(c) a point on circumference at location X

(d) a point at the centre at location X.

ushing (or) Bearing Pressure : ②





Crushing → compressive stress from all directions.

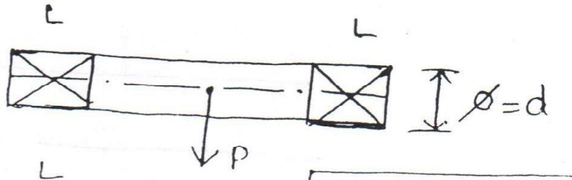
Due to crushing ; a material disintegrates into powder.

$$\boxed{\left.\begin{array}{l}\text{Crushing stress (or)}\\ \text{Bearing Pressure}\end{array}\right\} = \frac{\text{Load}}{\text{Projected Area}}}$$

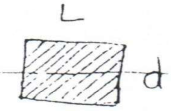
Projected area : area of contact projected view, in ③ the direction of load.

occurs generally in case of Bearings, pins (crank and Gudgeon pins) of joints.

adial Bearings: Load is perpendicular to the axis.



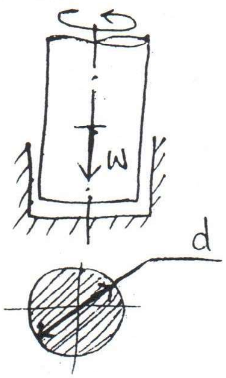
④



$$\boxed{P_b = \sigma_{cr} = \frac{P}{Ld}}$$

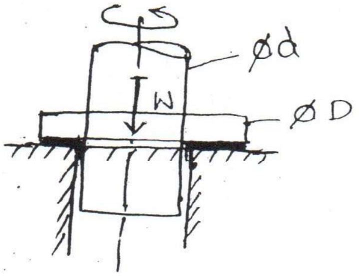
② Thrust Bearings : Load is parallel to axis.

(a) Pivot Bearings:



$$P_b = \frac{W}{\left(\frac{\pi}{4} \times d^2\right)}$$

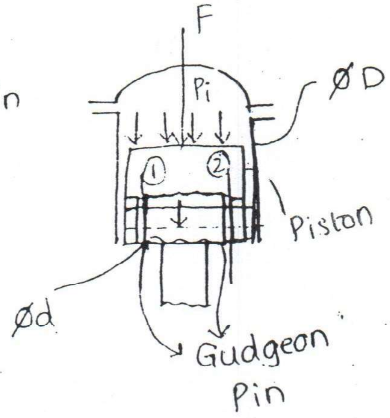
(b) Collar Bearings:



$$P_b = \frac{W}{\frac{\pi}{4}(D^2 - d^2)}$$

③ Pins of Joints :

- ✓ Gudgeon Pin
- ✓ Crank Pin
- ✓ Hinge Joint

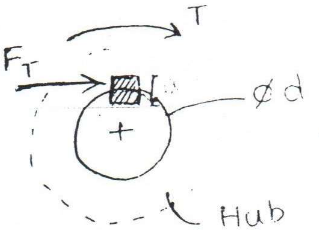


$$F = P_i \times \frac{\pi}{4} D^2$$

$$\tau = \frac{F}{2 \times \frac{\pi}{4} \times d^2}$$

$$P_b = \frac{F}{Ld}$$

④ Keys & Couplings:



$$P_h = \frac{F_T}{a \times L}$$

sign for Uni-axial Loads:

constant, gradually applied.

uniform cross-section.

ctual conditions match with Lab Test conditions.

$$\text{Induced Stress} = S = \frac{S_y \text{ (or) } S_u \text{ (or) } S_p}{F.S.}$$

S_y : Yield stress

S_u : ultimate stress

S_p : stress at proportional limit

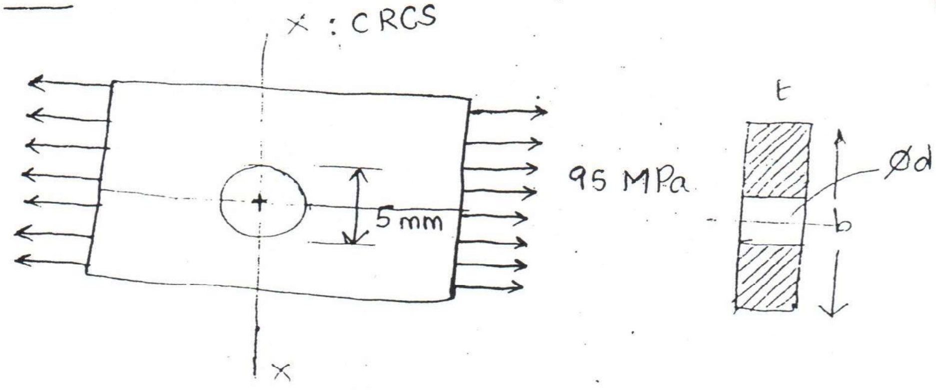
F.S. : Factor of safety.

lowable/working stress.

ctor of safety: It is used to account for ertainities in material properties, loads, defects, etc.

$$\boxed{\therefore \text{Factor of Safety (F.S.)} \geq 1.}$$

- 5 : ⑤



$\sigma = \frac{F}{A}$

$F = \sigma \times A$

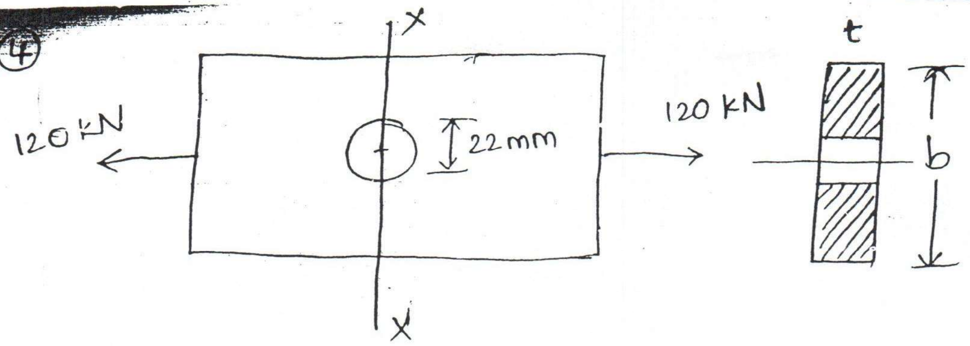
$F = \frac{P}{A}$

$P \neq F \times A$

$$\sigma_{max} = \frac{P_{or\,F}}{(b-d)t} = \frac{P_i \times bt}{(b-d) \times t}$$

$$= \frac{95 \times 100 \times t}{(100-5) \times t} = \underline{100 \text{ MPa}}$$

④



$$\sigma_{max} = \frac{P}{(b-d)t} = \frac{\sigma_y}{(F.S.)}$$

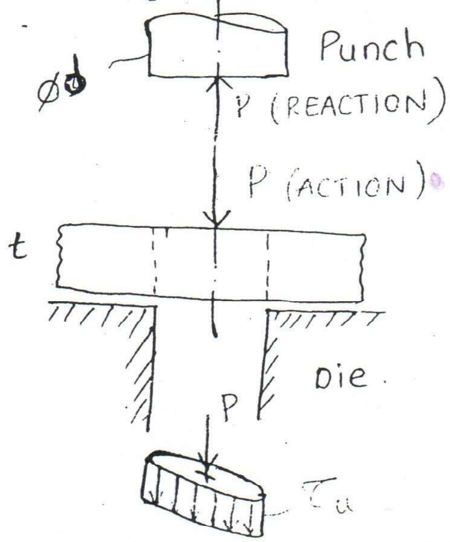
$$\Rightarrow 75 = \frac{120 \times 10^3}{(b-22) \times 13}$$

$$\Rightarrow 75b - (75 \times 22 \times 13) = 120 \times 10^3$$

$$\Rightarrow b = \frac{120 \times 10^3}{75 \times 13} + 22 = 145 \text{ mm}$$

---

⑤ Punching operation:



Punch Force $= P = \tau_u \times \pi d t$

$$\sigma = \frac{P}{\frac{\pi}{4} \times d^2} = \frac{\tau_u \times \cancel{\pi} \cancel{d} t}{\frac{\cancel{\pi}}{4} \times d^{\cancel{2}}}$$

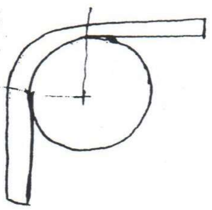
If $\sigma > \sigma_{allow}$, punch fails.

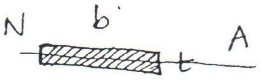
$$\therefore \sigma_{allow} = \frac{\tau_u \times t_{max}}{d/4}$$

$$330 = \frac{140 \times t_{max} \times 4}{17.5}$$

$$\Rightarrow \boxed{t_{max} = 10.3 \text{ mm}}$$

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R} \Rightarrow \sigma = \frac{E}{R} \times y$$





$$y_{max} = \frac{t}{2}$$

$$I = \frac{bt^3}{12}$$

$$\sigma = \frac{100 \times 10^3}{\frac{25}{2}} \times \frac{0.2}{2}$$

$$= 800 \text{ MPa.}$$

---

⑦:

$$\sigma = \frac{E}{R} \times y$$

$$= \frac{210 \times 10^3}{500} \times \frac{1}{2} = 210 \text{ MPa.}$$

$$M = \frac{E}{R} \times I = \frac{210 \times 10^9}{0.500} \times \frac{(0.015) \times (0.001)^3}{12}$$

$$\Rightarrow M = 0.525 \text{ N-m}$$

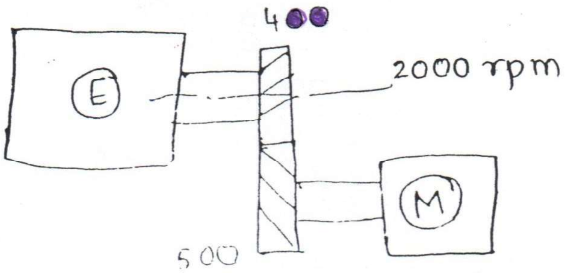
---

⑧ Twisting moment is :- $\frac{T}{J} = \frac{\tau}{r} = \frac{G\theta}{L}$

[illegible]:

$$P = \frac{2\pi NT}{60} \Rightarrow 10 \times 10^3 = \frac{2 \times \pi \times 5000 \times T}{60}$$

$$\Rightarrow \boxed{T = 191 \text{ N-m}}$$



Gear ratio = 4 : 1

Tube ⇒ Hollow circular

d = 30 mm

D = d + 2(t)

D = 30 + 2(3)

D = 36 mm

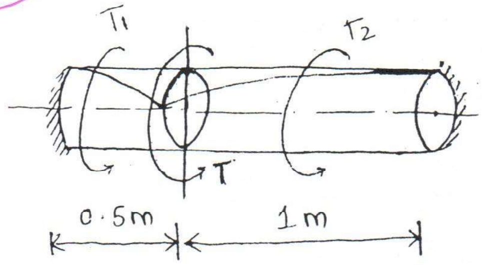
$K = \frac{d}{D}$

$$K = \frac{30}{36} = 0.833$$

$$\tau = \frac{16T}{\pi D^3(1-k^4)} = \frac{16 \times 191}{\pi(0.036)^3(1-(0.833)^4)} \text{ Pa.}$$

$$= \frac{3056}{\pi(0.036)^3(1-0.833^4)} = 40.3 \text{ MPa.}$$

Q2 & ③ :



$= \frac{\theta \times 180°}{\pi}$

$$\theta = \frac{T_1 L_1}{GJ} \pm \frac{T_2 L_2}{GJ} \qquad T = T_1 + F_2$$

$$\tau_{max} = \frac{16 T_{max}}{\pi d^3} \qquad \theta = \frac{T_1 L_1}{GJ} \text{ radians} = \frac{\theta}{\pi} \times 180°$$

$$\frac{T_1 \times 0.5}{\cancel{GJ}} = \frac{T_2 \times 1}{\cancel{GJ}}$$

$$\Rightarrow \boxed{T_1 = 2T_2}$$

$$T = 2T_2 + T_2$$

$$7358 = 3T_2$$

$$\Rightarrow \boxed{T_2 = 2452.6 \text{ N-m}}$$

$$T_1 = 2T_2$$

$$= 2 \times 2452.6$$

$$\boxed{T_1 = 4905.3 \text{ N-m}}$$

$$\therefore \tau_{max} = \frac{16 \times 4905.3}{\pi \times (0.080)^3}$$

$$= \frac{78484.8}{0.0016} = 49 \text{ MPa}$$

$J = \frac{\pi d^4}{32}$

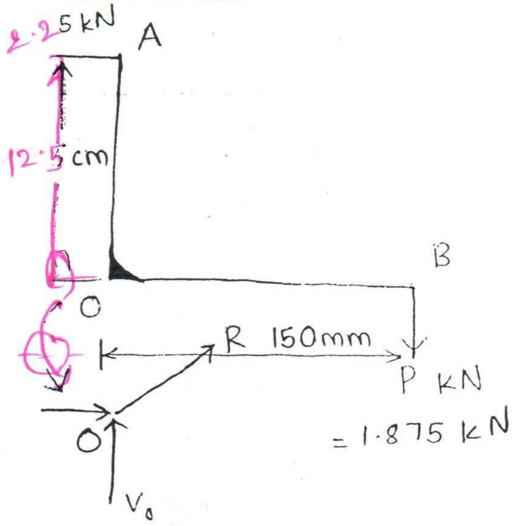
$$\therefore \theta = \frac{\overset{T_1}{4905.3} \times \overset{L_1}{0.5 \times 10^3}}{\underset{G}{0.8 \times 10^5} \times \pi^2 \times ((80)^4} \times \frac{32}{} \times 180 = 0.436°$$

~~$J = \frac{16 M}{\pi d^3}$~~

3.23×10^3

d = 80mm ✓

Q.9 6 & Q10 7:



Bearing Pressure $P_b = \frac{R}{Ld}$

$$\Sigma M_0 = 0$$

$$\Rightarrow (2.25 \times 125) = F \times 150$$

$$\Rightarrow P = \frac{2.25 \times 125}{150}$$

$$\Rightarrow \boxed{P = 1.875 \text{ kN}}$$

$$\Sigma F_H = 0 \Rightarrow 2.25 = H_0 \Rightarrow \boxed{H_0 = 2.25 \text{ kN}}$$

$$\Sigma F_v = 0 \Rightarrow V_0 = P \Rightarrow \boxed{V_0 = 1.875 \text{ kN}}$$

$$\therefore R = \sqrt{H_0^2 + V_0^2}$$

$$= \sqrt{(2.25)^2 + (1.875)^2}$$

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

$$\boxed{R = 2.928 \text{ kN}}$$

$$\therefore P_b = \frac{R}{L \times d}$$

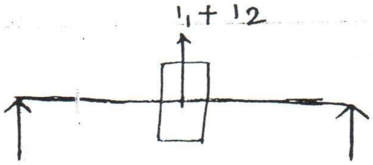
$$\Rightarrow 6.5 = \frac{2.928 \times 10^3}{2 \times d^2}$$

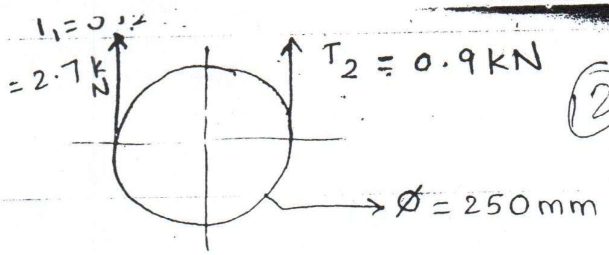
$$\Rightarrow d^2 = \frac{2.928 \times 10^3}{2 \times 6.5}$$

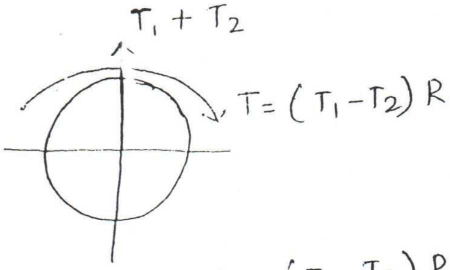
$$\Rightarrow \boxed{d = 15 \text{ mm}}$$

②

(Q11)







$$M = \frac{WL}{4} = \frac{(T_1 + T_2)L}{4}$$

$$= \frac{(2.7 + 0.9) \times 750}{4}$$

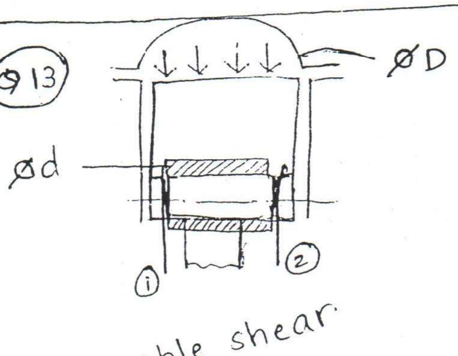
$$\boxed{M = 675 \text{ kN-mm}}$$

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

$$= (2.7 - 0.9)\,\frac{250}{2}$$

$$\boxed{T = 225 \text{ kN-mm}}$$

(Q13)



Double shear.

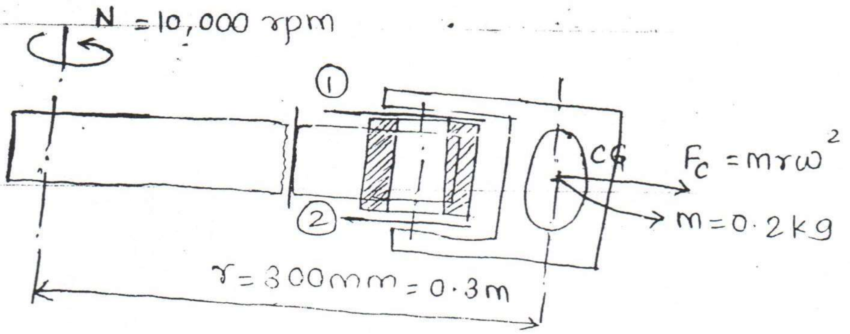
$$\tau = \frac{F}{2 \times \frac{\pi}{4} \times d^2} \qquad P_b = \frac{F}{L \cdot d}$$

$$\Rightarrow 10 = \frac{1.25 \times 10^{-6} \times \frac{\pi}{4} \times (60)^2}{2 \times \frac{\pi}{4} \times d^2}$$

$$\Rightarrow d^2 = \frac{1.25 \times \cancel{10^6} \times 60^2}{2 \times 10}$$

$$\Rightarrow \boxed{d = 15 \text{ mm}}$$

(08)



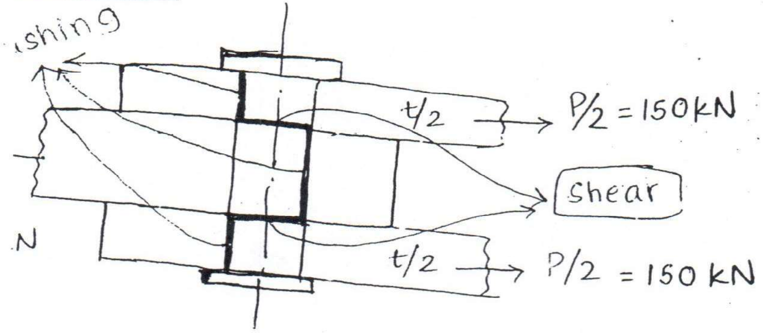
$$\tau = \frac{mr\omega^2}{2\times\frac{\pi}{4}\times d^2} = \frac{0.2\times0.3\times\left(\frac{2\times\pi\times10,000}{60}\right)^2}{2\times\frac{\pi}{4}\times(0.018)^2}$$

$$\boxed{\omega = \frac{2\pi N}{60}}$$

$$\tau = \frac{0.2\times0.3\times1047^2}{2\times\frac{\pi}{4}\times18^2} = 130\text{ MPa}$$

---

5 & 16 :



possible Failures:

shearing of pin

crushing of pin

Tearing of plates

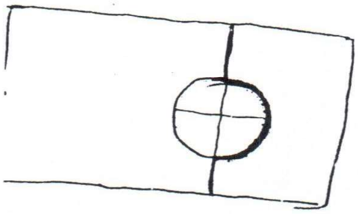
Shearing of Pin = Double shear

$$\tau = \frac{P}{2\times\frac{\pi}{4}\times d^2}$$

$$55 = \frac{300\times10^3\times4}{2\times\pi\times d^2}$$

$$\Rightarrow d^2 = 3472.47$$

$$\Rightarrow \boxed{d = 60\text{mm}}$$



$$\sigma = \frac{P}{(b-d)\times t}$$

$$\sigma = \frac{300\times10^3}{(b-60)\times15}$$

$$(b-60) = \frac{\overset{4}{\cancel{300}}\times10^3}{\cancel{80}\times\cancel{15}} \Rightarrow \boxed{b = 185\text{ mm}} \text{ i.e. } \boxed{b = 125\text{ mm}}$$

(II) Design For Combined Loads :

- ✓ uniform cross-section.
- ✓ Gradually applied and constant with respect to time.

Procedure :

① Find Critical Resisting Cross-Section :

- ✓ Has minimum area.
- ✓ Has maximum loads.

② Transfer all loads onto CG of CRCS.

③ Superimpose the effect of each load on extreme points of CRCS.

④ Find a point where stress is maximum. Such a point is called Heavily Stressed Point.

⑤ At Heavily Stressed Point, Find

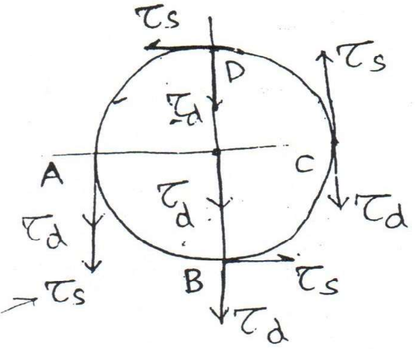
σ = Total Normal Stress $= \sigma_x$

τ = Total shear stress $= \tau_{xy}$

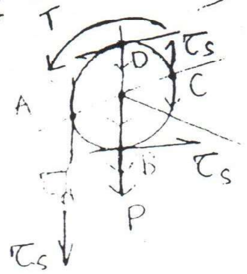
✓ Assumed that $\sigma_y = 0$.

⑥ Find Principle stresses : $\sigma_1, \sigma_2, \sigma_3$

⑦ Use Theories of failure to Design.



Ex : ①



$$\tau_d = \frac{4P}{\pi d^2}$$

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

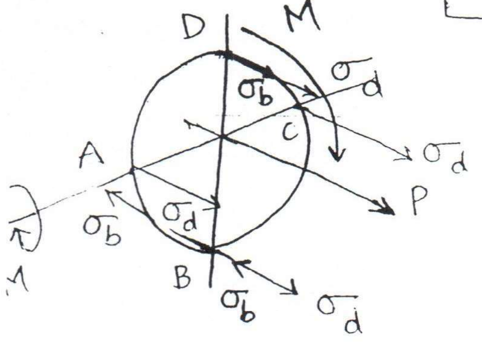
Heavily stress Point = A.

$$\boxed{\begin{array}{c} \tau_{xy} = \tau_d + \tau_s \\ \sigma_x = 0 \quad \sigma_y = 0 \end{array}}$$

$$\boxed{\text{Torsion + Transverse Shear Load}}$$

ex ②:

Axial Load + Bending Moment



$$\sigma_d = \frac{4P}{\pi d^2}$$

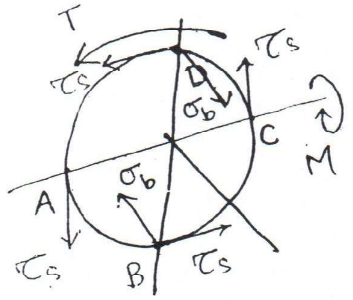
$$\sigma_b = \frac{32M}{\pi d^3}$$

Heavily stressed pt = D.

$$\boxed{\begin{aligned} \sigma_x &= \sigma_d + \sigma_b \\ \sigma_y &= (\\ \tau_{xy} &= 0 \end{aligned}}$$

ex ③:

Bending Moment + Twisting Moment



$$\sigma_b = \frac{32M}{\pi d^3}$$

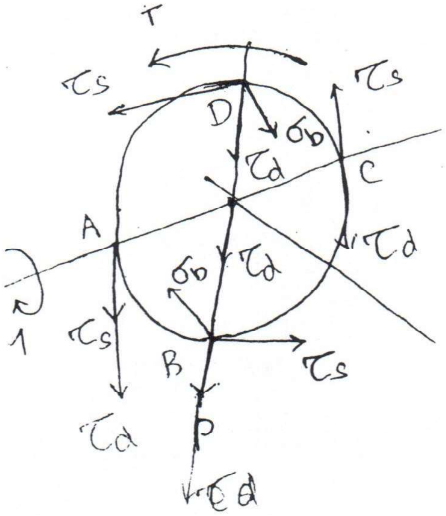
$$\tau_s = \frac{16T}{\pi d^3}$$

Heavily stressed pt = B, D

$$\boxed{\begin{aligned} \sigma_x &= \sigma_b \\ \sigma_y &= 0 \\ \tau_{xy} &= \tau_s \end{aligned}}$$

ex④:

Transverse shear + Torsion + Bending :



Heavily stressed Points : B & D

$$\sigma_b = \frac{32M}{\pi d^3}$$

$$\tau_d = \frac{4P}{\pi d^2}$$

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

$$\boxed{\begin{aligned} \sigma_x &= \sigma_b \\ \sigma_y &= 0 \\ \tau_{xy} &= \sqrt{\tau_s^2 + \tau_d^2} \end{aligned}}$$

For 2D stress / Plane stress / Bi-axial stress :

✓ Stress Tensor $[\sigma] = \begin{bmatrix} \sigma_x & \tau_{xy} \\ \tau_{yx} & \sigma_y \end{bmatrix}$

✓ $[\sigma] = \begin{bmatrix} \sigma_1 & 0 \\ 0 & \sigma_2 \end{bmatrix}$ If shear stresses are zero, σ_x & σ_y become principle stresses.

✓ Always $\sigma_3 = 0$.

✓ $$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\boxed{\sigma_y = 0 \Rightarrow \sigma_{1,2} = \frac{\sigma_x}{2} \pm \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}}$$

✓ $$\tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right| \text{ or } \left|\frac{\sigma_2 - \sigma_3}{2}\right| \text{ or } \left|\frac{\sigma_3 - \sigma_1}{2}\right|$$

$$\boxed{\therefore \tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right| \text{ or } \left|\frac{\sigma_2}{2}\right| \text{ or } \left(\frac{\sigma_1}{2}\right) \quad [\because \text{If } \sigma_3 = 0]}$$

$$\boxed{\tau_{max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \quad \text{if } \sigma_y = 0}$$

* Theories of Failure :-

→ are used to predict the failure condition due to combined stresses / loads.

1. Maximum Principle Stress Theory :

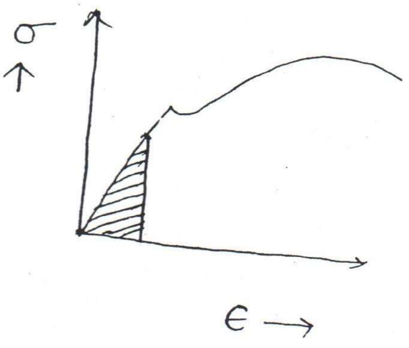
→ also called Rankine Theory.

2. Maximum Principle strain Theory → St. Venant's Theory

3. Maximum Shear stress Theory → Guest's / Tresca's / Coulomb's Theory.

4. Maximum strain Energy Theory → Haigh's Theory

Strain energy → Energy stored in the material due to the deformation.



$$\text{area} = \sigma \times \epsilon$$

$$= \frac{P}{A} \times \frac{\Delta L}{L}$$

$$\boxed{E = \frac{1}{2} \times \sigma \times \epsilon \times \text{volume}}$$

$$\boxed{S.E = \frac{\sigma^2}{2E} \times \text{volume}}$$

5. Maximum Shear strain Energy Theory :

→ Distortion Energy Theory

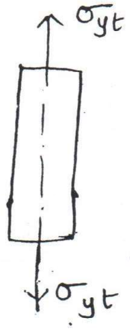
→ von-Mises Theory / Hencky's Theory.

→ Strain energy due to shear stresses alone.

① Maximum Principle Stress Theory :

According to this theory, failure occurs if the "maximum principle stress" due to combined Loads, either equals (or) exceeds the "maximum principle stress at yield point", in the simple tension test.

Let, $\sigma_1 > \sigma_2 > \sigma_3$.



$$\boxed{\begin{array}{c} \sigma_x = \sigma \\ \sigma_y = 0 \\ \tau_{xy} = 0 \end{array}}$$

$\sigma_1 = \sigma$

$\sigma_2 = 0$

$\tau_{max} = \frac{\sigma}{2}$

σ_{yt} = stress at yield point in simple tension test.

∴ As per maximum principle stress Theory,

Failure occurs if, $\boxed{\sigma_1 \geq \sigma_{yt}}$

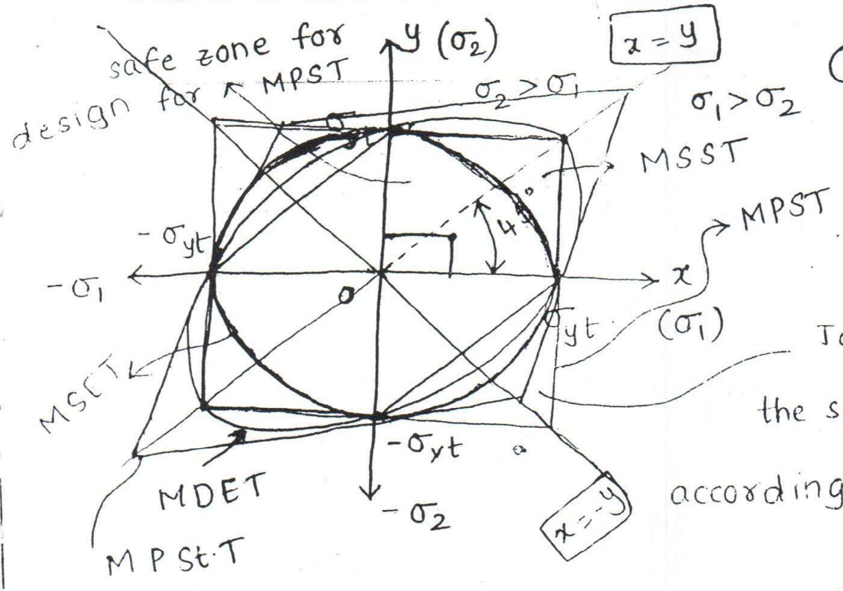
For safe Design, $\boxed{\sigma_1 = \frac{\sigma_{yt}}{F.S.}}$ allowable / working / permissible stress

→ Generally, this theory leads to Brittle Failure.

→ Brittle materials must be always loaded in the same directions i.e. one load is positive and other being negative is not valid.

	$(\sigma_1 > \sigma_2 > \sigma_3)$	Safe Design for 2D $\sigma_1 > \sigma_2 > (\sigma_3 = 0)$	Remarks.
1. Max. Pr. Stress Theory (Rankine Theory)	$\sigma_1 \geq \sigma_{yt}$	$\sigma_1 = \frac{\sigma_{yt}}{F.S}$	$\rightarrow$ $\boxed{\text{III choice.}}$ For Brittle materials.
2. Max. Pr. strain Theory (St. Venant's Theory)	$(\sigma_1 - \mu\sigma_2 - \mu\sigma_3) \geq \sigma_{yt}$	$(\sigma_1 - \mu\sigma_2) = \frac{\sigma_{yt}}{F.S.}$	$\rightarrow$ NOT USED (overpredicts strength of material) $x = \left(\frac{\sigma_{yt}}{1-\mu}\right)$
3. Max. sh stress Theory (Guest/Tresca Theory)	$\tau_{max} \geq \frac{\sigma_{yt}}{2}$	$\tau_{max} = \frac{\sigma_{yt}}{2 \times F.S.}$	$\rightarrow$ $\boxed{\text{I choice (Default Theory)}}$ Ductile m/l without Testing.
4. Max. str. Ener. Theory (Haigh's Theory)	$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1) \geq \sigma_{yt}^2$	$(\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2) = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$	$\rightarrow$ NOT USED (underpredicts the strength of material)
5. Max. Dist. Ener. Theory (von-mises Theory)	$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \geq 2 \times \sigma_{yt}^2$	$(\sigma_1 - \sigma_2)^2 = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$ i.e. $\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 = \frac{\sigma_{yt}}{F.S.}$ $\boxed{\sigma_x^2 + 3\tau_{xy}^2 = \left(\frac{\sigma_{yt}}{F.S}\right)^2}$ if $\sigma_y = 0$	$\rightarrow$ $\boxed{\text{II choice}}$ For Ductile materials with Testing

* safe zones for different Theories of Failure :



① MPST : $\sigma_1 > \sigma_2$

$\sigma_1 \geq \sigma_{yt} \rightarrow$ Fails

Total square indicates the safe zone for design according to MPST.

② MSST : $\tau_{max} = \frac{\sigma_{yt}}{2 \times FS}$

$$\tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right| \text{ or } \left|\frac{\sigma_1}{2}\right| \text{ or } \left|\frac{\sigma_2}{2}\right|$$

✓ Both are of same sign, $\tau_{max} = \frac{\sigma_1}{2} \geq \frac{\sigma_{yt}}{2 \times \textcircled{1}}$ → F.S. = 1.

(I & III Quadrants)

✓ Predictions according to MPST & MSST are same, if σ_1 & σ_2 are of same sign.

✓ If σ_1 & σ_2 are of opposite sign,

$$\tau_{max} = \frac{\sigma_1 + \sigma_2}{2} \geq \frac{\sigma_{yt}}{2}$$

✓ MSST is most conservative & rigid theory when compared to MPST. For entering into safe zone, stresses have to be reduced ⇒ dimensions are to be increased.

MSST is more conservative (Rigid)

⇒ Leads to Bigger Dimensions.

MPST is Liberal ⇒ Leads to smaller dimensions.

MSST is also called as "Hexa-Hedron Theory".

Max. DET:

$$\sigma_1^2+\sigma_2^2-\sigma_1\sigma_2 \geq \sigma_{yt}^2$$

$$\left.\begin{array}{l} t\ \sigma_1=x \\ \sigma_2=y \end{array}\right\} \Rightarrow x^2+y^2-xy \geq a \rightarrow \text{Eilipse equation.}$$

If $x=0$, $y=\pm a$.

If $y=0$, $x=\pm a$.

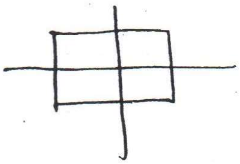
If $x=y$; $x=y=\pm a$

If $x=-y$; $x=-y=\pm \frac{1}{\sqrt{3}} a$.

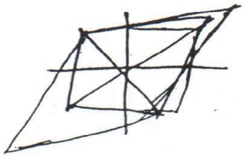
$\therefore x = 0.577\, \sigma_{yt}$

MDET is Liberal compared to MSST

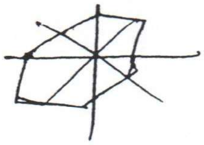
① MPST



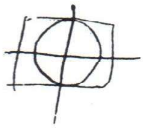
② MP st.T



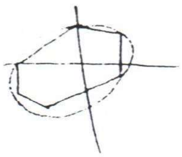
③ MSST:



④ MSET:



⑤ MDET:



Choice of Theories of Failure:

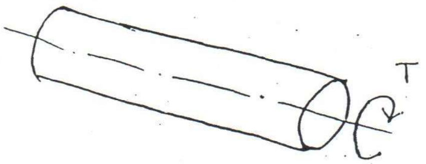
① MSST : $\tau_{max} = \frac{\sigma_{yt}}{2 \times F.S.}$

② MDET : $\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$

$$= \sigma_x^2 + 3\tau_{xy}^2$$

③ MPST : $\sigma_1 = \frac{\sigma_{yt}}{F.S}$

⑨ Relation between Yield stress in shear and Tension with Max. Distortion Energy Theory is ______



$\sigma_x = 0$

$\sigma_y = 0$

$\tau_{xy} = \frac{16T}{\pi d^3}$

∴ Failure due to max Distortion Energy Theory is

$$\sigma_x^2 + 3\tau_{xy}^2 = \sigma_{yt}^2$$

$$0 + 3\tau_{xy}^2 = \sigma_{yt}^2$$

$$\Rightarrow \boxed{\tau_{xy} = \frac{1}{\sqrt{3}}\sigma_{yt} = 0.577\,\sigma_{yt}}$$

with MSST, $\tau_{max} = \frac{\sigma_{yt}}{2}$

$$\tau_{max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ if } \sigma_y = 0 \Rightarrow \tau_{max} = \frac{\sigma_{yt}}{2}$$

$$\Rightarrow \boxed{\tau_{max} = 0.5\,\sigma_{yt}}$$

age-14:

(Q1)
$$\sigma = \begin{bmatrix} \overset{\sigma_1}{40} & 0 \\ 0 & \underset{\sigma_2}{-30} \end{bmatrix} \qquad \sigma_{yt} = 350 \text{ MPa.}$$

$F.S. = ?$ MSST.

$\sigma_1 = 40$ MPa

$\sigma_2 = -30$ MPa

$$\tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right|$$

$$= \left|\frac{40-(-30)}{2}\right| = \underline{\underline{35 \text{ MPa}}}$$

$$\text{MSST} \Rightarrow \tau_{max} = \frac{\sigma_{yt}}{2 \times F.S.}$$

$$\Rightarrow 35 = \frac{350}{2 \times F.S.} \Rightarrow F.S. = \frac{350}{2 \times 35} = \underline{\underline{5}}$$

---

6)

$\sigma_1 = 200$ MPa ; $\sigma_2 = -100$ MPa ; $\sigma_{yt} = 500$ MPa

Tresca → MSST.

$$\tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right| = \left|\frac{200+100}{2}\right| = 150 \text{ MPa}$$

$$\text{MSST} \Rightarrow \tau_{max} = \frac{\sigma_{yt}}{2 \times F.S.}$$

$$\Rightarrow F.S. = \frac{\overset{10}{\cancel{500}}}{2 \times \underset{3}{\cancel{150}}} = \frac{\overset{5}{\cancel{10}}}{\underset{3}{\cancel{6}}} = 1.67$$

$$\Rightarrow \boxed{F.S. = 1.67}$$

(Q10) $\sigma_1 = 100\,MPa$; $\sigma_2 = 60\,MPa$; MDET.

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$$

$$\Rightarrow 100^2 + 60^2 - (100 \times 60) = \sigma_w^2$$

$$\Rightarrow \boxed{\sigma_w = 87.1\ MPa}$$

(Q3) $\sigma_x = 60\,MPa$ and $\tau_{xy} = 40\,MPa$; $\sigma_{yt} = 330\,MPa$.

MPST

$$\sigma_{1,2} = \frac{\sigma_x}{2} \pm \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{60}{2} \pm \sqrt{\left(\frac{60}{2}\right)^2 + (40)^2}$$

$$= \frac{60}{2} \pm 50 \Rightarrow \sigma_1 = 80\,MPa$$

$$\sigma_2 = -20\,MPa$$

$$MPST \Rightarrow \sigma_1 = \frac{\sigma_{yt}}{F.S.} \Rightarrow F.S. = \frac{\sigma_{yt}}{\sigma_1} = [illegible] = \frac{6330}{80}$$

$$\Rightarrow \boxed{\cancel{F.S. = 6}} \quad \boxed{F.S. = 4.125}$$

(Q7) MSST $\Rightarrow \tau_{max} = \dfrac{\sigma_{yt}}{2 \times F.S}$ $\qquad \sigma_{yt} = \sigma_p = 284\,MPa$

$\sigma_x = 55\,MPa$; $\tau_{xy} = 31.5\,MPa$;

$$\tau_{max}\ \cancel{\sigma_1} = \cancel{\frac{\sigma_x}{2}} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

$$= \cancel{\frac{55}{2}} + \sqrt{\left(\frac{55}{2}\right)^2 + (31.5)^2}$$

$$= \cancel{27.5} + 41.81 \Rightarrow \overset{41.81}{\cancel{\sigma_1 = 69.31}} = \frac{284}{2 \times F.S.} \Rightarrow \boxed{F.S = 3.4}$$

5) $\sigma = \begin{bmatrix} 100 & 40 \\ 40 & 40 \end{bmatrix}$ MPa ; $\sigma_{yt} = 360$ MPa

MDET ;

$\sigma_x = 100$ MPa ; $\tau_{xy} = 40$ MPa ; $\sigma_y = 40$ MPa ;

$$\sigma_{1,2} = \frac{100+40}{2} \pm \sqrt{\left(\frac{100-40}{2}\right)^2 + (40)^2}$$

$$= 70 \pm \sqrt{30^2 + 40^2}$$

$$= 70 \pm 50 \Rightarrow \sigma_{1,2} = 120 \text{ MPa } \& \ 20 \text{ MPa}$$

$\boxed{\sigma_1 = 120 \text{ MPa}}$ $\boxed{\sigma_2 = 20 \text{ MPa}}$

$$\text{MDET} \Rightarrow (\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2) = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$$

$$\Rightarrow (120)^2 + (20)^2 - (120 \times 20) = \left(\frac{360}{F.S}\right)^2$$

$$12400 = \left(\frac{360}{F.S.}\right)^2$$

$$\Rightarrow F.S.^2 = \frac{36\cancel{0} \times 36\cancel{0}}{124\cancel{00}}$$

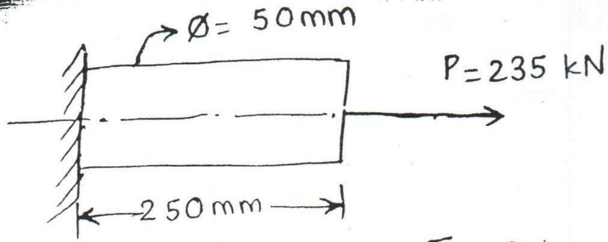
$$\Rightarrow \boxed{F.S. = 3.23}$$

$$\text{Max. shear stress} = \tau_{max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \left|\frac{\sigma_1 - \sigma_2}{2}\right|$$

$$= \left|\frac{120-20}{2}\right| \text{ or } \left|\frac{20-0}{2}\right| \text{ or } \left|\frac{0-120}{2}\right|$$

$$\Rightarrow \boxed{\tau_{max} = 60 \text{ MPa}}$$

(Q2)



$\sigma_{ut} = 480 \text{ MPa}$

$$\sigma_x = \frac{235 \times 10^3 \times 4}{\pi \times 50^2}$$

$\sigma_y = 0$;

MSST: $\tau_{xy} = 0$; $\sigma_x = 119.68 \text{ MPa} = \sigma_1$

$$\tau_{max} = \frac{\sigma_{ut}}{2 \times F.S.} \Rightarrow \tau_{max} = \left|\frac{\sigma_1}{2}\right| = \left|\frac{119.68}{2}\right| = 59.84 \text{ MPa}$$

$$\Rightarrow \tau_{max} = \frac{\sigma_{ut}}{2 \times F.S} \Rightarrow F.S = \frac{480}{2 \times 59.84}$$

$$\Rightarrow \boxed{F.S. = 4}$$

(Q4) $P_1 = 48 \text{ kN}$; $A_{RCS} = 600 \text{ mm}^2$; $\sigma_x = \frac{48 \times 10^3}{600} = 80 \text{ MPa}$.

$P_s = 18 \text{ kN}$;

MSST: $\tau_{xy} = \frac{18 \times 10^3}{600} = 30 \text{ MPa}$;

$\sigma_y = 0$;

$$\sigma_{1,2} = \frac{\sigma_x}{2} \pm \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

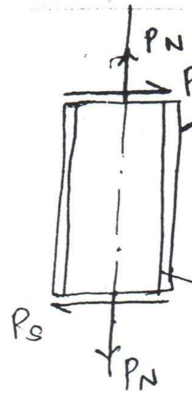
$$= \frac{80}{2} \pm \sqrt{\left(\frac{80}{2}\right)^2 + (30)^2}$$

$$= 40 \pm 50 \quad \sigma_1 = 90 \text{ MPa} ; \quad \sigma_2 = -10 \text{ MPa} ;$$

$$\tau_{max} = \left|\frac{\sigma_1 - \sigma_2}{2}\right| = \left|\frac{90 + 10}{2}\right| = 50 \text{ MPa}$$

$$\text{MSST} \Rightarrow \tau_{max} = \frac{\sigma_{yt}}{2 \times F.S} \Rightarrow F.S = \frac{200}{2 \times 50}$$

$$\Rightarrow \boxed{F.S. = 2}$$



d = Nominal size (diametre.)
= Bolt size

BOLTS

d_c = core dia.

$$\boxed{d_c = (0.8 \text{ to } 0.9) \times d}$$

---

(18) $P_N = 20$ kN ; $\sigma_{yt} = 360$ MPa ;

$P_S = 15$ KN ; F.S. = 3 ; MDET

$$\text{MDET} \Rightarrow \sigma_x^2 + 3\tau_{xy}^2 = \left(\frac{\sigma_{yt}}{F.S.}\right)^2$$

$$\Rightarrow \left(\frac{4 \times 20 \times 10^3}{\pi \times d^2}\right)^2 + 3\left(\frac{4 \times 15 \times 10^3}{\pi \times d^2}\right)^2 = \left(\frac{360}{3}\right)^2$$

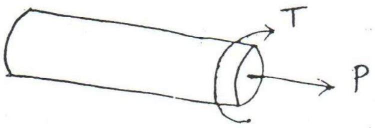
$$\Rightarrow \frac{1}{d^4}\left[\left(\frac{4 \times 20 \times 10^3}{\pi}\right)^2 + 3\left(\frac{4 \times 15 \times 10^3}{\pi}\right)^2\right] = 14,400$$

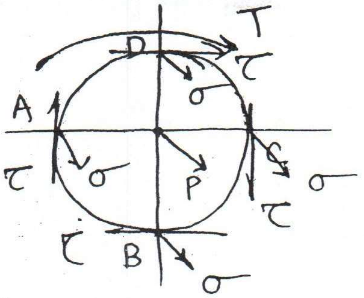
$$\Rightarrow \frac{1}{d^4}\left[(6,48,45,55,75,310) + (1094268783)\right] = 14400$$

$$\Rightarrow d^4 = 4507628.06$$

$$\Rightarrow \boxed{d = 18.65 \text{ mm}} \Rightarrow \text{value should be selected more than } \underline{18}.$$

Q9 **MDET :**





H.S.P = any poi

on circumference

$$\sigma_x = \frac{4P}{\pi d^2}$$

$$= \frac{4 \times 40 \times 10^3}{\pi \times 20^2}$$

$$= \boxed{127.32 \text{ MPa}}$$

$$\tau_{xy} = \frac{16T}{\pi d^3}$$

MDET : $\sigma_x^2 + 3\tau_{xy}^2 = \left(\frac{\sigma_{yt}}{FS}\right)^2$

$$\Rightarrow (127.32)^2 + \left(\frac{3 \times 16T}{\pi \times 20^3}\right)^2 = \left(\frac{310}{2}\right)^2$$

$$\Rightarrow \tau_{xy}^2 = \frac{24025 - (16210.3)}{3}$$

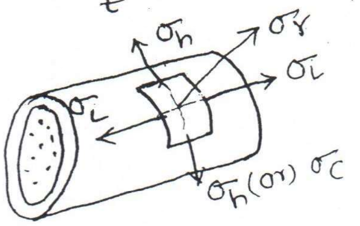
$$\Rightarrow \tau_{xy} = 51.03 = \frac{16T}{\pi d^3}$$

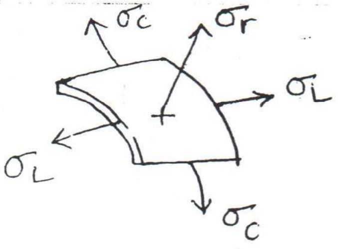
$$\Rightarrow T = \frac{51.03 \times \pi \times 20^3}{16}$$

$$\boxed{T = 80 \text{ N-m}}$$

⑮ $d = 4.6\text{m}$; $t = 16\text{mm}$; Pressure = 210 KPa ;

If $\frac{D}{t} > 20 \Rightarrow$ Thin cylinder





$$\sigma_c = \frac{Pd}{2t} \; ; \; \sigma_L = \frac{Pd}{4t} \; ; \; \sigma_r = 0 \; ;$$

$$\sigma_x = \sigma_c = \frac{Pd}{2t} = \frac{210 \times 4600}{2 \times 16} \Rightarrow \sigma_1 = 30.187 \text{ MPa}$$

$$\sigma_y = \sigma_L = \frac{Pd}{4t} = \frac{210 \times 4600}{4 \times 16} \Rightarrow \sigma_2 = 15.093 \text{ MPa}$$

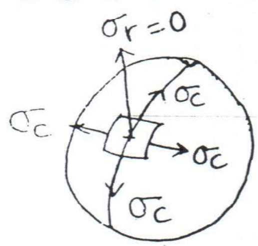
$$\sigma_r = \sigma_3 = 0 \quad \tau_{max} = \left|\frac{\sigma_1}{2}\right| = \left|\frac{30.187}{2}\right| = 15.093 \text{ MPa}$$

$$\text{MSST} \Rightarrow \tau_{max} = \frac{\sigma_{yt}}{2 \times F.S}$$

$$\Rightarrow F.S. = \frac{260}{2 \times 15.093}$$

$$\Rightarrow \boxed{F.S = 8.62}$$

Spherical shells:



$$\sigma_c = \frac{PD}{4t} = \sigma_1 = \sigma_2 \qquad \sigma_r = \sigma_3 = 0$$

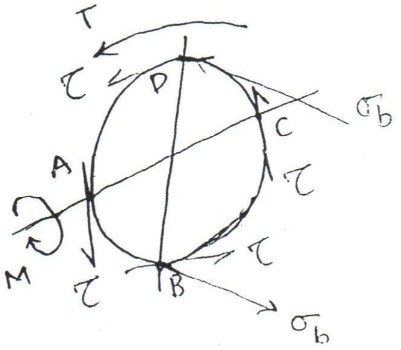
$$\boxed{\tau_{max} = \frac{PD}{8t}}$$

Shafts subjected to Bending & Twisting Moments :

Let M = Maximum Bending Moment on Shaft.

T = Maximum Twisting Moment on shaft

$$\sigma_b = \frac{32M}{\pi d^3} \qquad \tau = \frac{16T}{\pi d^3}$$



Heavily Stressed Points are = B & D

At Heavily stressed Points,

$$\boxed{\begin{aligned} \sigma_x = \sigma_b &= \frac{32M}{\pi d^3} \\ \tau_{xy} = \tau &= \frac{16T}{\pi d^3} \; ; \; \sigma_y = 0 \end{aligned}}$$

$$\sigma_{1,2} = \frac{\sigma_x}{2} \pm \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \; ; \text{ if } \sigma_y = 0 \; ;$$

$$\sigma_{1,2} = \frac{16M}{\pi d^3} \pm \sqrt{\left(\frac{16M}{\pi d^3}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^3}$$

$$\boxed{\sigma_{1,2} = \frac{16}{\pi d^3}\left[M \pm \sqrt{M^2 + T^2}\right]}$$

$$\therefore \boxed{\tau_{max} = \frac{16}{\pi d^3}\left[\sqrt{M^2 + T^2}\right]}$$

Equivalent Twisting Moment = $T_{equ.}$ = It is the pure twisting moment, when applied will induce same maximum shear stress as that of combined Bending & Twisting moments.

$$\tau_{max} = \frac{16}{\pi d^3}\left[\sqrt{M^2+T^2}\right] = \frac{16\, T_{equ.}}{\pi d^3}$$

$$\therefore \boxed{T_{equ.} = \sqrt{M^2+T^2}}$$

As per maximum shear stress Theory,

$$\boxed{\tau_{max} = \frac{16\, T_{equ.}}{\pi d^3} = \frac{\sigma_{yt}}{2 \times F.S.}}$$

Equivalent Bending moment $= M_{equ} =$ It is the pure bending moment when applied, will induce same maximum principle stress, as that of combined bending and twisting moments.

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

As per maximum Principle Stress Theory,

$$\boxed{\sigma_1 = \frac{32 M_{equ.}}{\pi . d^3} = \frac{\sigma_{yt}}{F.S.}}$$

(?12) $d = 7.5\,cm = 75\,mm$; $= 0.075\,m$; $M = 250$ N-m ; $T = 420$ N-m ; $\sigma_{yt} = 3800 \frac{N}{cm^2}$; F.S. = ?

MSST $\Rightarrow \frac{16\, T_{equ}}{\pi d^3} = \frac{\sigma_{yt}}{2 \times F.S.}$

$$T_{equ} = \sqrt{M^2 + T^2}$$

$$= \sqrt{(25000)^2 + (42000)^2}$$

$$= 48,877.39 \text{ N-cm}$$

$$\Rightarrow \frac{16 \times 48,877.39}{\pi \times (7.5)^3} = \frac{3800}{2 \times F.S.}$$

$$590 = \frac{1900}{F.S.} \Rightarrow \boxed{F.S = 3.22.}$$

(Q16) T = 10 kN-m ; T = 5 kN-m

M = 10 kN-m ; M = 6 kN-m **MSST**

F.S. = 1.5 F.S. = ?

$$(T_{equ.})_2 = \sqrt{M^2 + T^2} = \sqrt{6^2 + 5^2} = 7.810 \text{ kN-m.}$$

$$(T_{equ.})_1 = \sqrt{10^2 + 10^2} = 14.14 \text{ kN-m}$$

$$\text{MSST} \Rightarrow \tau_{max} = \frac{16T}{\pi d^3} \nless \frac{\sigma_{yt}}{2 \times F.S}$$

$$\Rightarrow T \propto \frac{1}{F.S}$$

$$\Rightarrow \frac{T_1}{T_2} = \frac{(F.S)_2}{(F.S)_1}$$

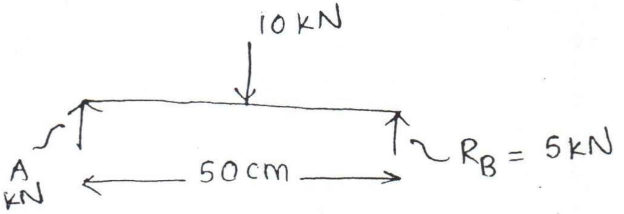
$$\Rightarrow \frac{14.14}{7.810} = \frac{(F.S)_2}{1.5} \Rightarrow \boxed{(F.S.)_2 = 2.71}$$

Q13 $P = 30\,kW$; $N = 710\,rpm$;

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

$$\Rightarrow T = \frac{P \times 60}{2\pi N} = \frac{30 \times 10^3 \times 60}{2 \times \pi \times 710} \Rightarrow \boxed{T = 403.49\ N\text{-}m}$$

25 cm



$$B.M. = \frac{WL}{4} = \frac{10 \times 10^3 \times 50 \times 10^{-2}}{4}$$

$$\boxed{M = 1250\ N\text{-}m}$$

$$T_{equ.} = \sqrt{M^2 + T^2}$$

$$= \sqrt{(1250)^2 + (403.49)^2}$$

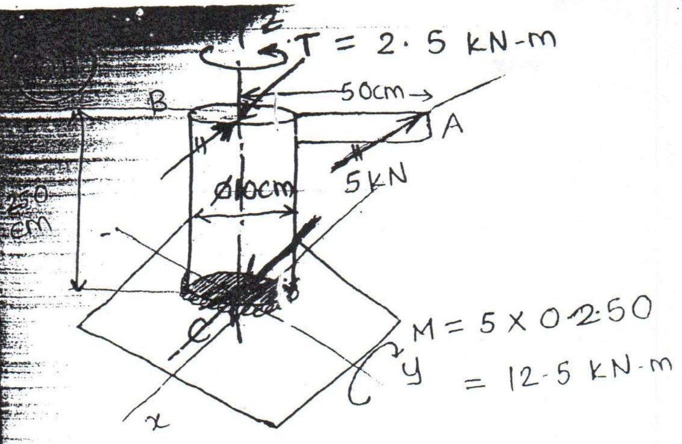
$$\boxed{T_{equ.} = 1313.508\ N\text{-}m}$$

$$MSST \Rightarrow \tau_{max} = \frac{16\,T_{equ}}{\pi d^3} = \frac{\sigma_{yt}}{2 \times F.S}$$

$$\Rightarrow \frac{16 \times 1313.508}{\pi \times (40 \times 10^{-3})^3} = \frac{420 \times 10^6}{2 \times F.S}$$

$$519.86 = \frac{210000000}{F.S}$$

$$\boxed{F.S. = 3}$$



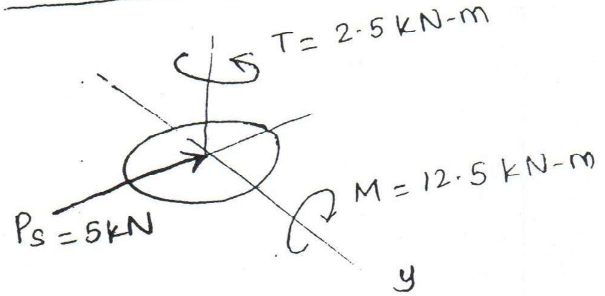
T = 2·5 kN-m ;

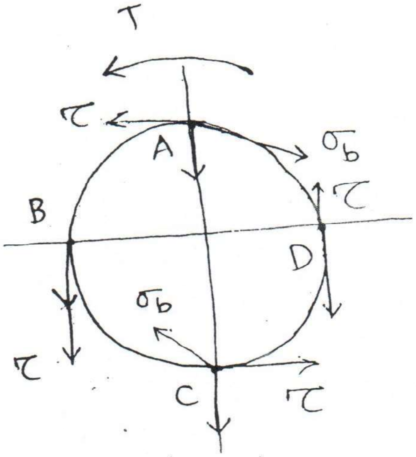
M = 12·5 kN-m ;

P_s = 5 kN ; ø = 10 cm

= 0·1m

Loads at CRCS :





Heavily Stressed Point = A.

at Heavily stressed point,

$$\boxed{\sigma_x = \sigma_b \ ; \qquad \sigma_y = 0 \\ \tau_{xy} = \sqrt{\left(\frac{16T}{\pi d^3}\right)^2 + \cancel{\left(\frac{4P}{\pi d^2}\right)^2}} \ ;}$$

$\cong 0$

(Neglected initially in design as it leads to complex calculations).

→ In initial design of effects of direct shear and Normal will be neglected for simplification.

$$T_{equ.} = \sqrt{M^2 + T^2}$$

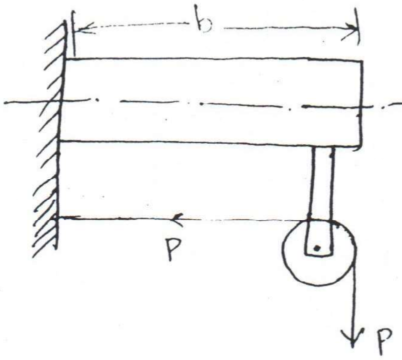
$$= \sqrt{(12 \cdot 5)^2 + (2 \cdot 5)^2} = \underline{12 \cdot 747} \text{ kN-m}$$

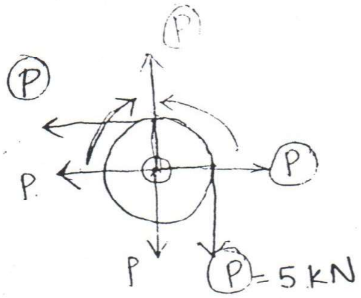
$$\therefore \text{M.S.S.T} \Rightarrow \frac{16\,T_{equ.}}{\pi d^3} = \frac{\sigma_{yt}}{2 \times F.S}$$

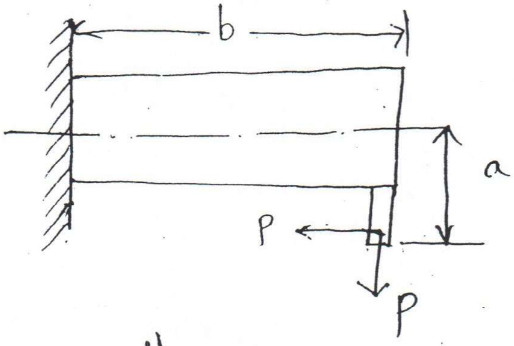
$$\Rightarrow \frac{16 \times 12.747 \times 10^3}{\pi \times (0.1)^3} = \frac{425 \times 10^6}{2 \times F.S}$$

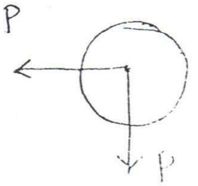
$$\Rightarrow \boxed{F.S. = 1.27}$$

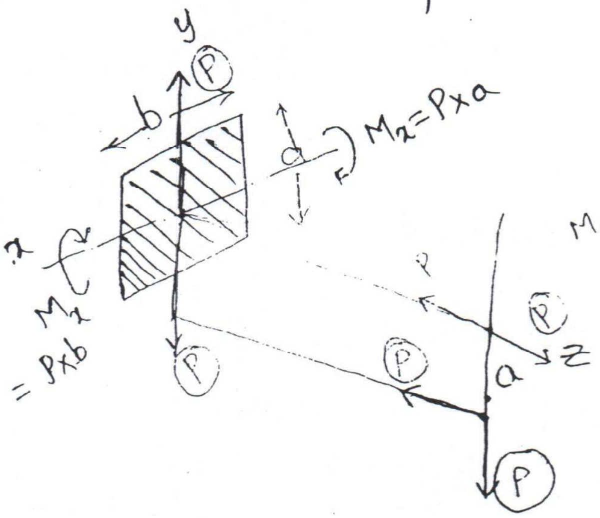
(Q14)

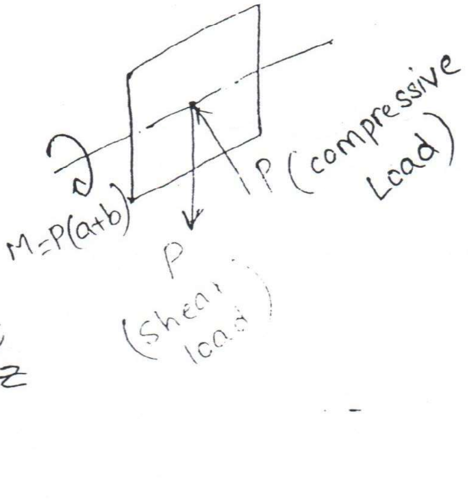












$\therefore$ Neglecting direct shear and normal loads,

$$\therefore M = P(a+b)$$

$$= 5(500+1500)$$

$$\boxed{M = 10 \text{ kN-m}}$$

MPST: $\sigma_1 = \dfrac{6M}{bd^2} = \dfrac{\sigma_{yt}}{F.S}$

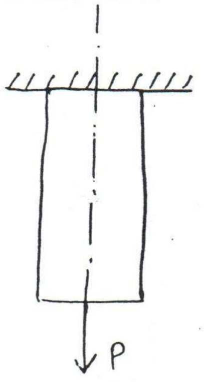
$$\Rightarrow \frac{6\times10\times10^3\times10^3}{b\times4b^2} = \frac{200}{2.5}$$

$$\Rightarrow \frac{6\times10^7\times2.5}{200\times4} = b^3$$

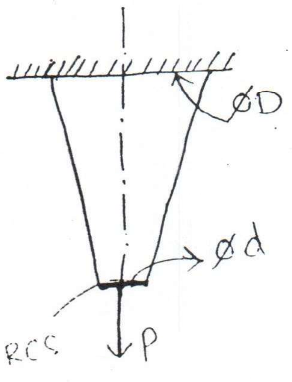
$$\Rightarrow b = 57.23 \text{ mm}$$

$$\Rightarrow \boxed{b = 57.4\text{mm}}$$

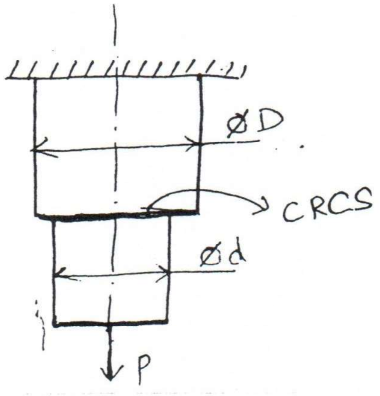
Ⅲ Design for Varying Cross-sections:



uniform c/s



uniformly varying c/s



suddenly varying c/s