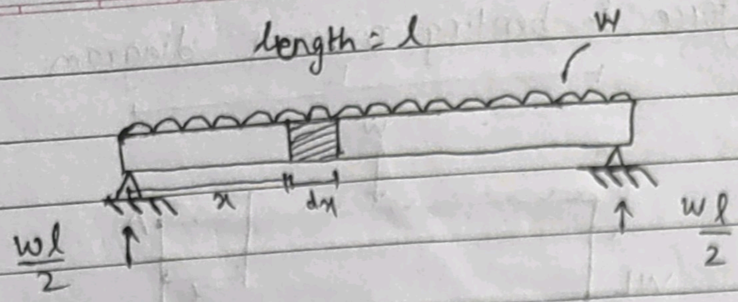


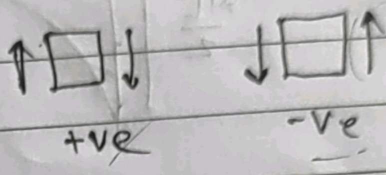

Bending stress/moment (M)

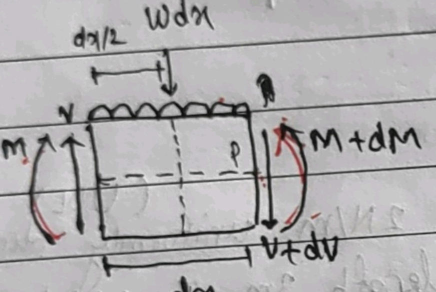
convention -

unit - Nm


Transeverse shear stress (force) (V)

unit - N




$\Sigma F = 0$

$V = wdx + V + dV$

$\therefore w = -dV/dx$

$\Sigma M = 0$

$$\boxed{w = \frac{-dV}{dx}}$$

$\Sigma M = 0$ ..... calculating at point P

$$-M - Vdx + \frac{wdx \cdot dx}{2} + M + dM = 0$$

$$-Vdx + \left(\frac{-dV}{\cancel{dx}}\right)\frac{\cancel{dx}}{2}dx + dM = 0$$

$dx \cdot dx <<< 0$

$$-Vdx - \frac{dV \cdot dx}{2} + dM$$

IMP

$$-Vdx + dM = 0 \qquad \therefore \boxed{V = dM/dx}$$

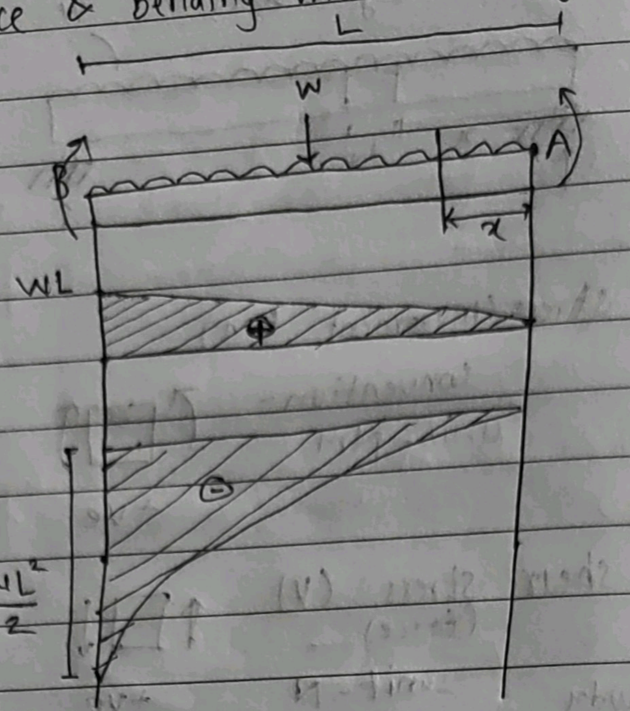
wherever shear force is 0, bending moment has maximum value ∴ or minimum

$$w = -\frac{d}{dx}(V) = -\frac{d^2M}{dx^2}$$

$$\therefore \boxed{-w = \frac{dV}{dx} = \frac{d^2M}{dx^2}}$$

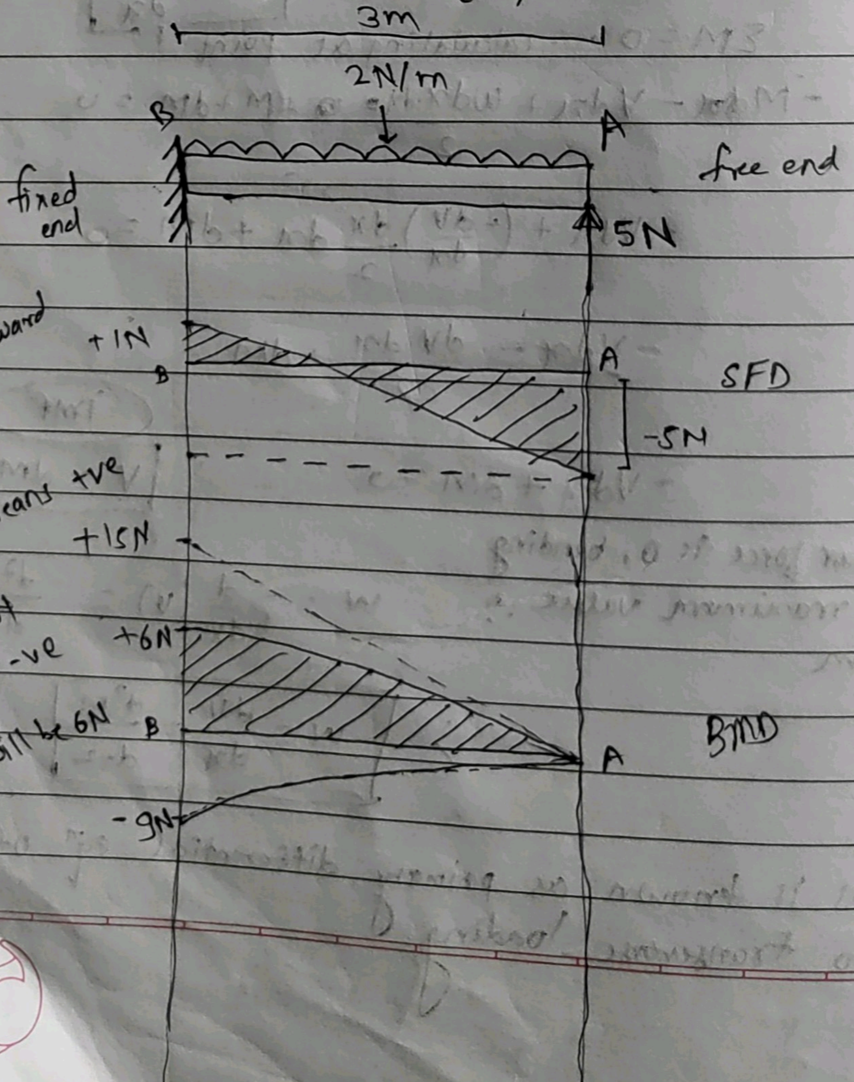
This is known as primary differential eq^n of beam subjected to transeverse loading.

## * Shear force & bending moment diagram




Total length = L
Total force = WL

1] A contiliver beam suppose 2 N/m uniformly distributed load (UDL) throughout the span of length 3m. In addition, A concentrated load of 5N acts at free end of upward. Draw S.F. & B.M. diag.




B का right side मे 5N upward लगाया है ∴ force is -ve

B का right side मे 2N downward लगाया है ∴ force is +ve

$5 \times 3 = 15$ N·m anti clockwise means +ve

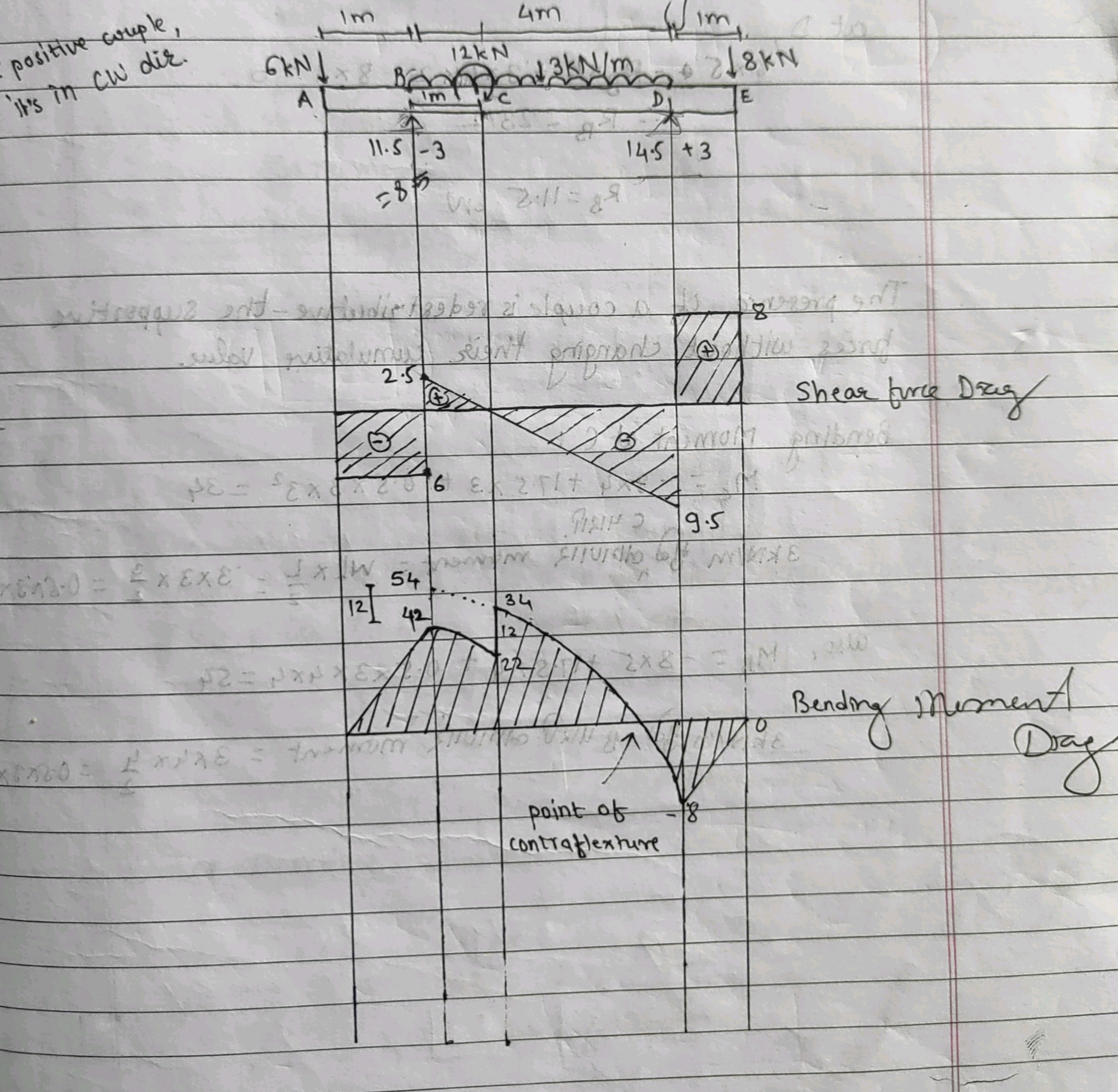
अगर 2N force से होने वाला moment $= WL \times \frac{L}{2} = 9$ N

∵ it is in CW it is -ve

effective of 15 & -9 will be 6N

2] An overhung simple supported beam of length 6m is supported by 1m from each end. The middle 4m is loaded with uniformly distributed load of 3 kN/m. At the left hand end, 6 kN & right hand end, 8 kN is also supported by beam. Additionaley one positive couple 12 kNm is acting at a point 2m from left hand end. Draw BM & SF diag for beam.

∵ positive couple, it's in CW dir.



$R_B$ & $R_D$ are to be calculated (without considering couple)

at B,

$$6\times1-3\times4\times2+R_D\times4-8\times5=0$$

$$\therefore 4R_D=58$$

$$R_D=14.5\ kN$$

at D,

$$6\times5-R_B\times4+3\times4\times2-8\times1=0$$

$$\therefore R_B=23/2$$

$$R_B=11.5\ kN$$

The presence of a couple is redestributive - the supportive forces without changing their cumulative value

Bending Moment at C :

$$M_C=-8\times4+17.5\times3+0.5\times3\times3^2=34$$

3kN/m મુકેલ C પાસેથી લાગનાર moment $= WL\times\frac{L}{2} = 3\times3\times\frac{3}{2} = 0.5\times3\times3^2$

also, $M_B=-8\times5+17.5\times4+0.5\times3\times4\times4=54$

3kN/m મુકેલ B પાસેથી લાગનારું moment $= 3\times4\times\frac{4}{2} = 0.5\times3\times4$

Bending Stress in Beams:

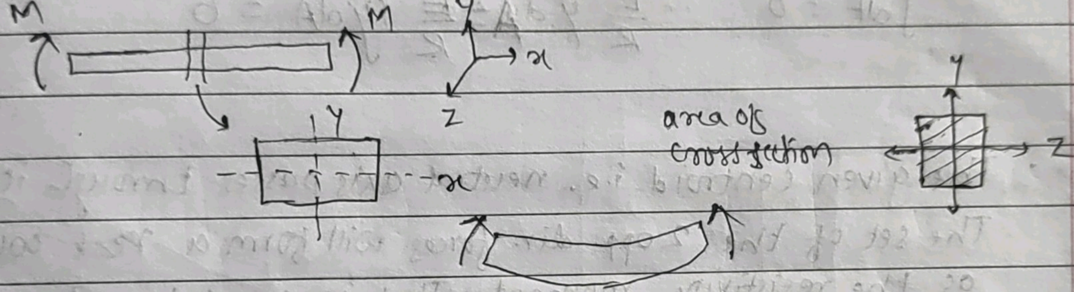
① Pure bending (No shear force is acting)


M = constant

shear force = $\frac{dM}{dx} = 0$

② Modulus of elastisity (Y)

Same in tension & compression

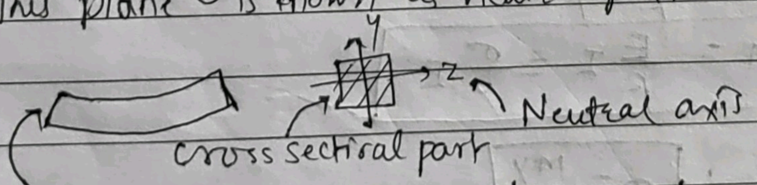



③ The plane section remains plane after bending

④ The sectⁿ under considerⁿ is axisymmetric

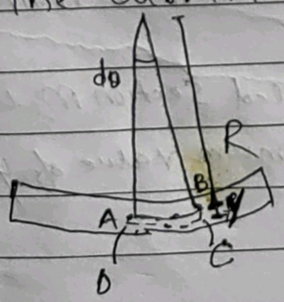
⑤ After bending there is a plane which remains unstrained. (no change in length)

This plane is known as neutral plane.




Cross sectional plane is called neutral plane

⑥ The curvature is circular in nature




CD ~~AB~~ $= (R+y)\,d\theta$

$AB = R\,d\theta$

$\Delta y = y\,d\theta$

As there is no axial force acting on the surface, net summation of the force on the cross section is zero.

$$\text{Strain} = \frac{\Delta y}{R d\theta} = \frac{y d\theta}{R d\theta} = y/R$$

$$\text{stress} = \sigma = \frac{y}{R} \cdot E = \frac{E}{R} \cdot y$$

Force on element

$$dF = \sigma\, dA = \frac{E}{R} y \cdot dA$$

$$\int dF = 0 \quad \therefore \frac{E}{R} y\, dA = \frac{E}{R} \int y\, dA = 0$$

- The given centroid i.e. neutral axis passes through it. The set of the 2 opp. dir. forces will form a ~~rest~~ couple known as the resistivity moment. That is equal to external moment for equilibrium

$$M = \int y \cdot dF = \int y \sigma\, dA = \int y \frac{E}{R} y \cdot dA$$

$$= \frac{E}{R} \int y^2 dA$$

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

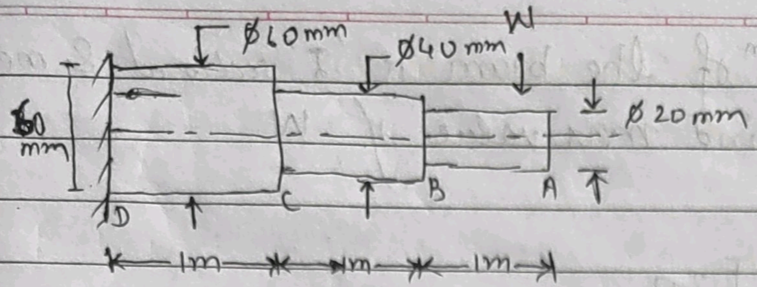
$$\therefore \boxed{\sigma = \frac{My}{I}}$$

I is second moment of inertia i.e. moment of area

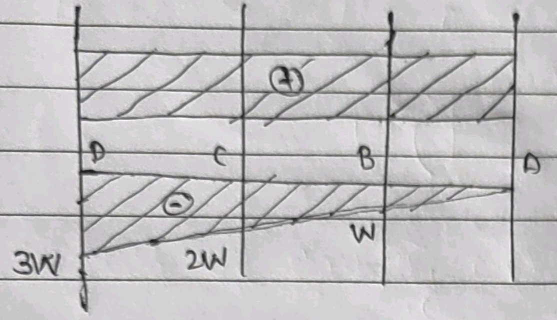
$$\sigma = \frac{M}{I/y}$$ I/y is called Section Modulus at maximum value of y

Modulus of rigidity = $\boxed{G = EI}$

1)




What is the maximum value of W so that stress of 100 MPa is produced




$$I_{AB} = \frac{\pi D^4}{64} = \frac{\pi \times 160000}{64} = \cancel{\pi \times 2500}$$
$$= 7857\ mm^4 = 0.7857 \times 10^4\ mm^4$$

$$I_{BC} = \frac{\pi D^4}{64} = 125663\ mm^4 = 12.56 \times 10^4\ mm^4$$

$$I_{CD} = \frac{\pi D^4}{64} = 63.6172 \times 10^4\ mm^4$$

at D, $\sigma = \frac{My}{I}$

$\therefore 100 \times 10^6 = \cancel{63.6172 \times 10^4 \times 3}\ \frac{M \times 30}{63.6172 \times 10^4}$

$M = 21.20 \times 10^{11}$

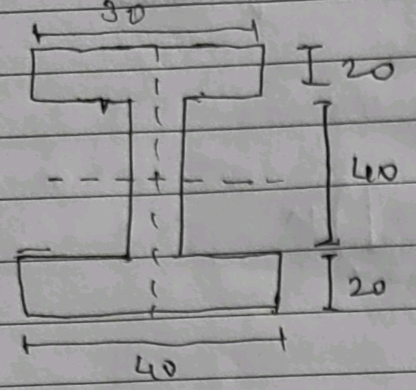
at C, $\sigma = \frac{My}{I}$ $\quad \therefore 100 \times 10^6 = \frac{M \times 20}{12.56 \times 10^4}$

$M = 6.28 \times 10^{11}$

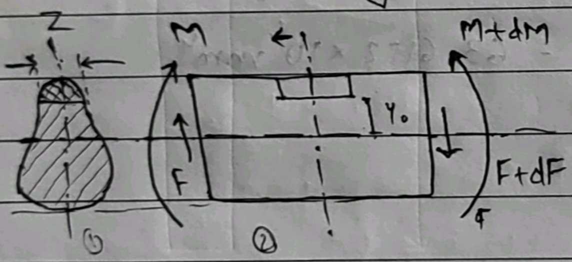
at B $\quad \sigma = \frac{My}{I} \quad 100 \times 10^6 = \frac{M \times 10}{0.7857 \times 10^4}$

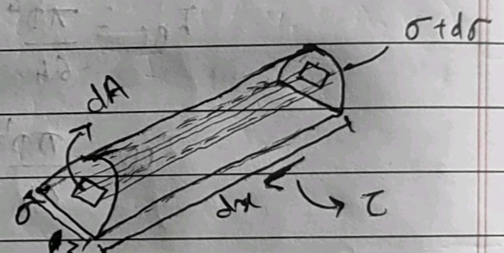
$M = 0.7857 \times 10^{11}$

2] If the cross sect$^n$ of the beam is 'I' sectioned & measurements as given below find max value of W




* Shear Stress in a transversely load beam




① is cross sect$^n$ of ②

The small area dA on the elemental piece is subjected to normal stress due to bending moment M

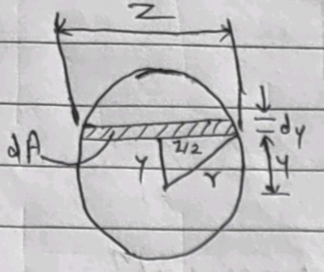
$$\text{Net force acting on element} = d\sigma . dA$$
$$= \frac{dM.\bar{y}}{I} . dA$$

The counter active force is the resistive force restricting the movement & is the shear force.

$$\tau . z . dx = \frac{dM.\bar{y}}{I} . dA$$

$$\therefore \tau = \left(\frac{dM}{dx}\right) . \left(\frac{1}{zI}\right) . (\bar{y} . dA)$$
$$= \frac{Fy}{zI} dA$$

when cross sect$^n$ is circular




$$r^2 = y^2 + \frac{z^2}{4}$$

$$\tau = \frac{F}{zI}\bar{y}A$$

$$y^2 = r^2 - \frac{z^2}{4} \qquad \therefore 2y\,dy = -\frac{1}{4}\cdot 2z\,dz$$

$$\therefore y\,dy = -\frac{z\,dz}{4}$$

$$dA = z\,dy$$

$$\bar{y}\cdot A = \int_y^r y\,dA$$

$$= \int_y^r y\cdot z\,dy =$$

we've to convert it to z limits

as y goes from y to r, z goes from z to 0

$$\bar{y}\cdot A = \int_z^0 -\frac{z^2}{4}\,dz = \frac{1}{4}\int_0^z z^2\,dz = \frac{1}{12}z^3$$

$$\bar{y}A = z^3/12$$

$$\therefore \tau = \frac{F}{zI}\bar{y}A = \frac{F}{zI}\times\frac{z^3}{12} = \frac{Fz^2}{12I}$$

I i.e. moment of inertia of cross sect$^n$ $= I = \pi r^4/4$

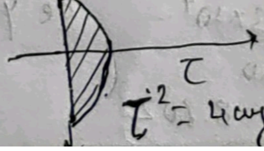
$$\tau = \frac{Fz^2}{12\frac{\pi r^4}{4}} = \frac{Fz^2}{3\pi r^4}$$

$$\boxed{\tau = \frac{Fz^2}{3\pi r^4}}$$ This expression is desirable in terms of y to get variat$^n$ of $\tau$

$$\therefore \boxed{\tau = \frac{Fz^2}{3\pi r^4} = \frac{4F}{3\pi}\left(\frac{r^2-y^2}{r^4}\right)}$$

extremes means $y = r$ neutral axis means $y = 0$

At extremes, $\tau = 0$ at neutral axis $\tau$ is maximum



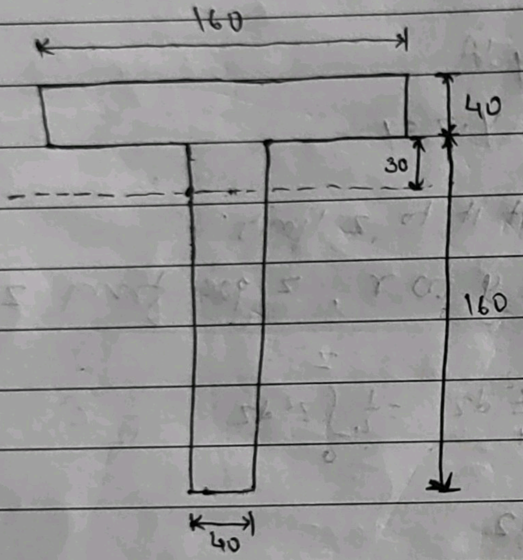

$\tau^2 = 4ay \rightarrow$ parabola

$$\tau_{max} = \frac{4F}{3\pi r^2} \qquad \tau_{avg} = \frac{F_{avg}}{\pi r^2}$$

$$\frac{\tau_{max}}{\tau_{avg}} = \frac{4}{3}$$

The same ratio for rectangular cross sect$^n$,

the ratio $\frac{\tau_{max}}{\tau_{avg}} = 3/2$



$F = 200kN$

$$\tau = \frac{F}{ZI}(A_1\bar{y}_1 + A_2\bar{y}_2)$$

Locat$^n$ of neutral plane is $(y_0)$ from top

$$y_0 = \frac{A_1y_1 + A_2y_2}{A_1 + A_2} = \frac{160\times40\times20 + 160\times40\times120}{2\times160\times40}$$

$= 70$

$\bar{Y}_1$ is locat$^n$ of $Y_1$ from $Y_0$ ie. neutral plane

$\therefore \bar{Y}_1 = 50$

$\bar{Y}_2 = 50$

$A_1\bar{y}_1 = 6400\times50 = 320000\ mm^3$

$A_2\bar{y}_2 = 6400\times50 = 320000\ mm^3$

$I = I_1 + I_2 = 1685\times10^4 + 2965\times10^4 = 4650\times10^4\ mm^4$

$\tau = \frac{F}{IZ}(A\bar{y})$ $\because$ flange is in contact with only upper part $A$ will be $A_1$ $\bar{y}$ will be $\bar{Y}_1$

$$= \frac{200\times10^3 \times 32\times10^4}{4650\times10^4\times160}$$

$= 8.602$

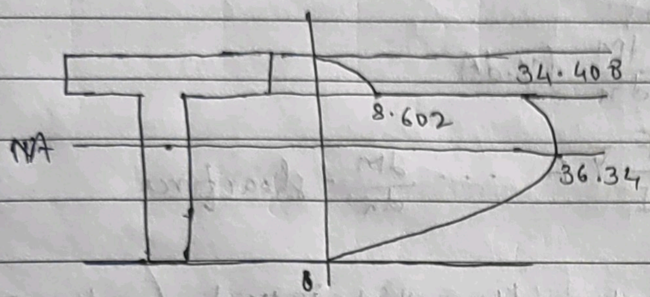
Section just above the web

$$\tau = \frac{160}{40} \times 8.602 = 34.408$$

$\because \tau \propto \frac{1}{Z}$

verticle part ની Z 40 horizontal part ની Z 160 છે.

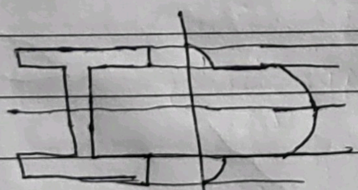
$\therefore$ verticle part નું $\tau$ will be 4 times that of hor. part $\tau$

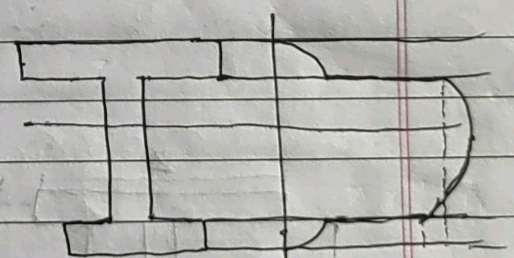



$\tau$ Through neutral axis, will be calculated as $\tau$ above NA $-$ $\tau$ below NA or simply $\tau = \frac{F}{ZI}(A_1\bar{Y}_1 + A_2\bar{Y}_2)$

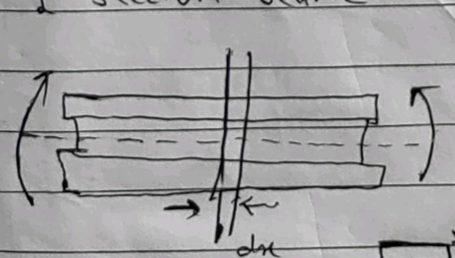
however $A_2\bar{Y}_2$ will be in -ve sense here because o/s point being below NA

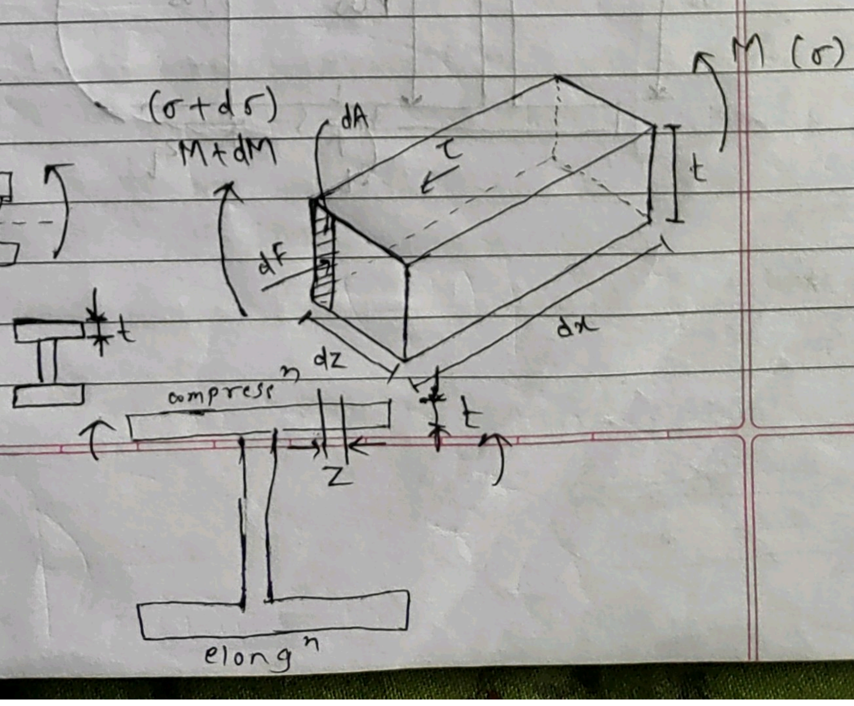
& thus $\tau$ will go on decreasing until it becomes 0





* I section beam



$\therefore$ Net force acting on ~~double~~ shaded area $= d\sigma . dA$

$$dF = \frac{dM . y}{I} . dA$$

$$F = \int \frac{dM y . dA}{I}$$

$\because$ syst is in equilibrium, restoring force will be shear force

$$\tau . (t dx) = \int \frac{dM y . dA}{I}$$

$$\therefore q = \tau t = \frac{dM}{I dx} \int y . dA$$

$$= \frac{F}{I} \bar{Y} . A \quad \ldots \frac{dM}{dx} = \text{shear force}$$

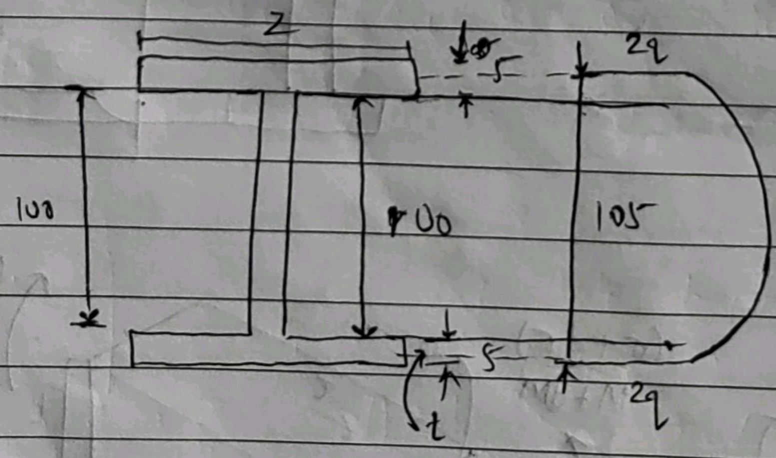
$\tau t$ is known as shear flow which is used for thin sectn beam analysis;

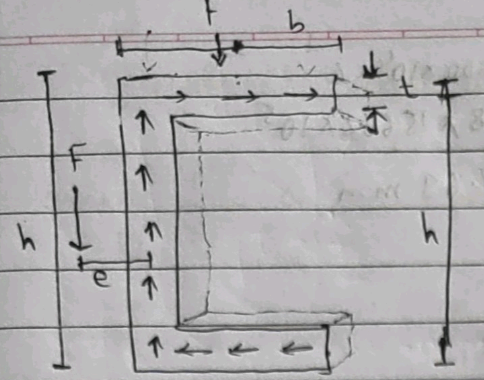
unit of shear flow is $N/m$

now, $A = zt$

$$q = \tau t = \frac{F \bar{Y} . zt}{I}$$

$$\therefore \tau = \frac{F \bar{Y} . z}{I}$$






back part is grouted
here front part will twist like this

$h >> t$

induced shear force $F = \frac{Ftb^2h}{4I}$

couple $= \frac{Ftb^2h}{4I} \cdot \frac{h}{2}$

let F be the restoring force applied at a horizontal distance e as shown

$\therefore$ Restoring couple $= Fe$

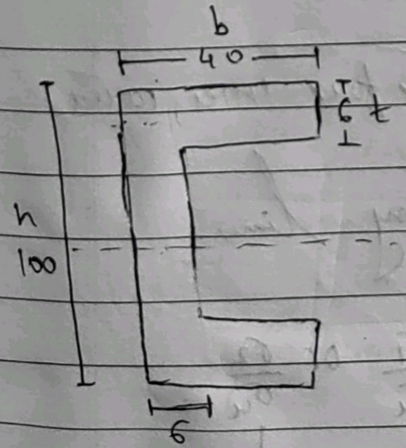
for no twisting of the beam,

$$Fe = \frac{Ftb^2h}{4I} \cdot \frac{h}{2}$$

Vimp

$$\therefore e = \frac{b^2h^2t}{8I}$$

This e is known as shear centre of cross section. If the (verticle) transeverse force is applied through shear centre of the sect$^n$, no twisting will occur. The shear centre may be within the body or outside




Moment of Area of top & bottom flange $= I_1 = I_3$

$= \frac{bd^3}{12} = \frac{40 \times 6^3}{12} = 720$

M.O.A through neutral axis $= MOA + A\bar{y}^2$

$= 720 + 40 \times 6 \times 47^2$

$= 5.301 \times 10^5$

MOA of verticle part $= \frac{bd^3}{12} = \frac{6 \times 100^3}{12} = 5 \times 10^5$

$\therefore I_{net} = I_1 + I_2 + I_3 = (5.301 + 5 + 5.301) 10^5$

$= 15.602 \times 10^5$

$$e = \frac{b^2h^2t}{8I} = \frac{1600 \times 10^4 \times 6}{8 \times 15.602 \times 10^5}$$

$$e = 7.69 \text{ mm}$$

* Theories of failure

failure point : ductile point $\sigma_y$ (yield stress)
brittle point $\sigma_u$ (ultimate stress)

They can be estimated for any material by uniaxial tension test by UTM (universal testing machine)

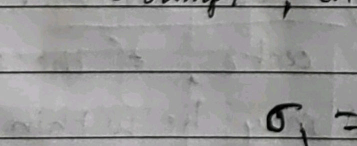
$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

* Rankine's theory (maximum normal stress theory)

It states that the brittle material fails when any of the principle stress reaches the ultimate point ($\sigma_u$) with an assumptn, the strain in tension & in compression are same

$$\sigma_1 = \pm\sigma_u \text{ or } \sigma_2 = \pm\sigma_u$$

+ve for tension -ve for compression




any state beyond the four lines causes failure
The lines are called boundary lines.

The normalized form is expressed as $\frac{\sigma_1}{\sigma_u}$ or $\frac{\sigma_2}{\sigma_u}$
↓ x axis ↓ y axis

Reciprocal of above terms is known as factor of safety

among $\frac{\sigma_u}{\sigma_1}$ & $\frac{\sigma_u}{\sigma_2}$ whichever gives minimum absolute value is factor of safety

The brittle material in general, exhibits stronger behaviour in compression

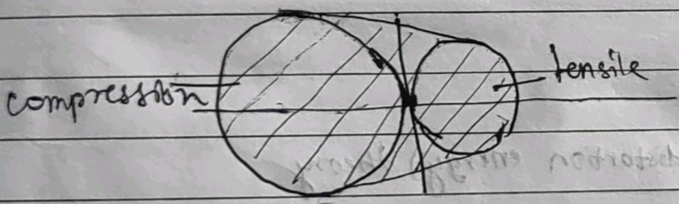
Imp

* Mohr's theory

C - compressive t - tensile

It states that the material fails under tensile & compressive load according to ( $\sigma_{uc}$ or $\sigma_{ut}$) depending on type of loading when any one of the principle stress crosses the limit

shaded is safety region




* Ductile material ($\sigma_y$)

Material fails due to shear when induced shear stress reaches the yield point value of the shear stress obtained by simple tension test

(1) Maximum shear stress theory (Tresca's criterion)

At principle axis, shear stress = 0 & in its perpendicular the $\tau_{yp} = \pm\left(\frac{\sigma_1 - \sigma_2}{2}\right)$

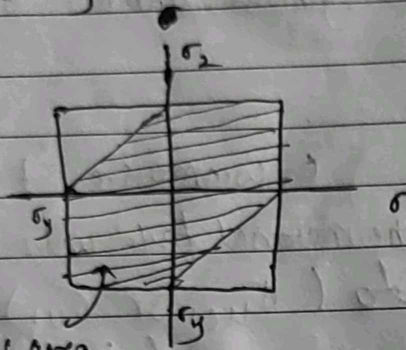
for uniaxial tensile test, $\sigma_2 = 0$ & $\tau_y = \sigma_{yp}/2$

~~ductile mat~~

The ductile material fails when the maximum shear stress induced exceeds the yield shear stress obtained by simple tension test.

In simple tension test, $\tau = \frac{\sigma}{2}$

$\sigma_y = \pm(\sigma_1 - \sigma_2)$



$\sigma_y = \sigma_1 - \sigma_2$
$\sigma_y = \sigma_2 - \sigma_1$ } two lines

Tresca's area

This is the safe area where failure is not happening

② Maximum Strain energy (distortion energy) theory

Distorsion means the change of shape & usually it is due to induced shear stress. The element absorb energy for this distortion. The specific distort$^n$ energy is defined by the distort$^n$ energy absorbed per unit vol. In triaxial stress con$^n$, it is given as

$$U = \frac{1}{12G}\left[(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2\right]$$

In uniaxial stress con$^n$, as $\sigma_2 = \sigma_3 = 0$,

$$U = \frac{\sigma_1^2}{6G}$$

for biaxial con$^n$, $\sigma_3 = 0$

$$U = \left[(\sigma_1-\sigma_2)^2 + \sigma_1^2 + \sigma_2^2\right]\frac{1}{12G}$$

$$= \frac{1}{12G}\left(2\sigma_1^2 - 2\sigma_1\sigma_2 + 2\sigma_2^2\right)$$

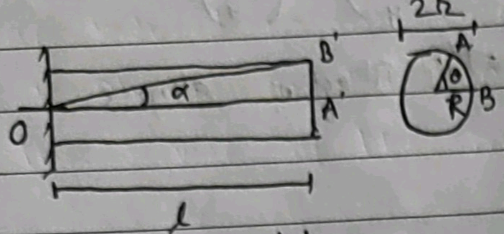
$$U = \frac{1}{6G}\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2\right)$$

$$\sigma_y^2 = \sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2$$

$$1 = \left(\frac{\sigma_1}{\sigma_y}\right)^2 - \left(\frac{\sigma_1\sigma_2}{\sigma_y^2}\right) + \left(\frac{\sigma_2}{\sigma_y}\right)^2$$

This is eqn of ellipse

# Torsion



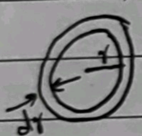

$\tan\alpha = \alpha = A'B' / OA$

$AB = R\theta$

$$\alpha = \frac{R\theta}{l} \qquad \sigma = E\varepsilon$$

$$\tau_R = \frac{GR\theta}{l}$$

If $\tau$ is stress developed on elemental area,




$$dF = \tau(2\pi r\,dr)$$

$$dT = \tau(2\pi r\,dr)r \qquad T \text{ is torque}$$

$$\text{at } r,\ \tau = \frac{Gr\theta}{l}$$

$$\therefore dT = \frac{Gr^2\theta}{l}(2\pi r\,dr)$$

$$T = \frac{G\theta}{l}\int (2\pi r\,dr)r^2$$

$$= \frac{G\theta}{l} I_z$$

J or $I_z$ is moment of area about z axis

$I_z = J = I_x + I_y$

$$\boxed{T = \frac{G\theta J}{l}}$$

$$T = \frac{G\theta r J}{r l} \qquad T = \frac{J\tau_r}{r}$$

$$\therefore \boxed{\tau_r = \frac{Tr}{J}} \qquad \boxed{\tau_{max} = \frac{TR}{J}}$$

IMP

for a circle of R, priorly we used to take $I = \pi d^4/64$

here, $J = I_z = I_x + I_y = \pi d^4/64 + \pi d^4/64$

$= \pi d^4/32$

Page

Date

* Assumptⁿ

homogeneous

Isotropic

Twisting is within elastic limit & small

Compression & Elongⁿ has sam effect

* Solid circular shaft transmitting power

Power = $P = T\omega$

$\sigma_m = \frac{M_{eq} y}{I}$ $y = r = d/2$ $I = \pi d^4/64$

$\sigma_m = \frac{M_{eq} d}{2 \times \frac{\pi d^4}{64}}$ $\therefore$ $\boxed{\sigma_m = \frac{32 M_{eq}}{\pi d^3}}$ d is shaft diameter

$\tau_m = \frac{T_{eq} r}{J} = \frac{T_{eq} d}{2J} = \frac{T_{eq} d}{2 \frac{\pi d^4}{32}}$ $\therefore$ $\boxed{\tau_m = \frac{16 T_{eq}}{\pi d^3}}$

yield stress $\approx 0.8 \times$ ultimate stress

$\sigma_{eq} = \frac{\sigma}{2} + \sqrt{(\sigma/2)^2 + \tau^2}$

$\tau_{eq} = \sqrt{(\sigma/2)^2 + \tau^2}$

IMP

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

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

$\sigma_y \approx 0.8\, \sigma_u$ $\tau_y = \frac{1}{2}\sigma_y$

$\sigma_y$ & $\sigma_u$ are available by simple tension test (for steel)

* Factor of safety (F.S.):

It is a factor by which the magnitude of $\sigma_y$ or $\sigma_u$ is divided (as the case may be), to limit the stress value induced in an element for the sake of its safety. It is always greater than 1 & chosen judiciously depending on the service of the machine element.

$$[\sigma] = \sigma_a = \frac{\sigma_y}{F.S.}$$

$\sigma_a \rightarrow$ alloy

$$[\tau] = \tau_a = \frac{\tau_y}{F.S.}$$

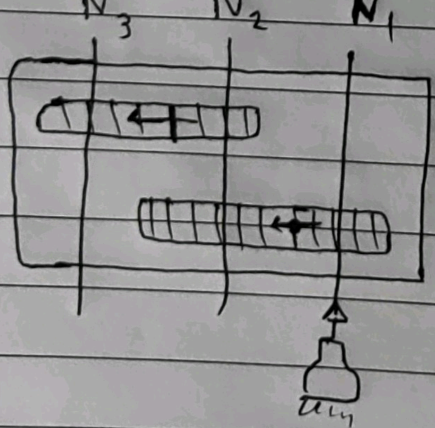
ex. F.S. = 2 $\quad \sigma_u = 300\,MPa$

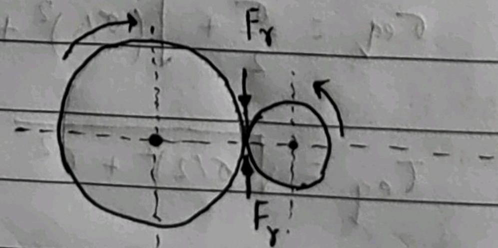
$\sigma_y = 0.8\,\sigma_u = 240\,MPa$

$\tau_y = \frac{1}{2}\sigma_y = 120\,MPa$

$\sigma_a = \frac{240}{2} = 120\,MPa$

$\tau_a = \frac{\tau_y}{FS} = 60\,MPa$



1] Input power 6 kW, input rpm = 1500, diameter of input gear = 50 mm dia of 1st intermidiate gear = 100 mm, 2nd = 50 mm, dia of output gear = 100 mm. Distance b/w the bearing support for all the shaft is 225 mm. All gears are equally spaced. find safe diameter of all these shafts with $\sigma_u = 300$, F.S. = 2.5

* The rigidity of shaft is given by angle of twist ($\theta$)

$$\theta = \frac{TL}{GJ}$$

T - Torque
L - length b/w transmission point
G - Modulus of rigidity
J - polar moment of area

for steel $E = 210$ GPa

Imp $G = 80$ GPa

1] Compare the shear stress induced & angle of twist b/w to shafts of same material & same area of cross section, one being solid & other being hollow with inner to outer diameter ratio 1:2

G & L will be same for both cases

area of solid cross sect$^n$ = area of hollow cross sect$^n$

$$\frac{\pi d^2}{4} = \frac{\pi (d_2^2 - d_1^2)}{4}$$

$d_2 = 2d_1$ $\quad d^2 = 3d_1^2 \quad d = \sqrt{3}\, d_1$

$$\frac{\tau_{m\,(solid)}}{\tau_{m\,(hollow)}} = \frac{16 T_{eq}/\pi d^3}{16 T_{eq}/\pi(d_2^3 - d_1^3)} = \frac{8d_1^3 - d_1^3}{3\sqrt{3}\, d_1^3} = \frac{7}{3\sqrt{3}} = \frac{7\sqrt{3}}{9}$$

$= 1.3471$

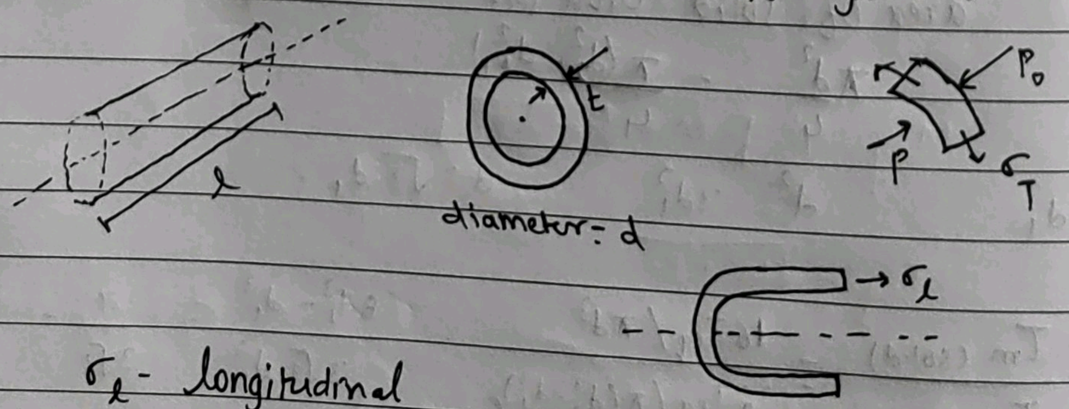
$1\ N/mm^2 = 9.869$ atm
$\approx 10$ atm

# Pressure Vessel

* The pressure vessels are closed vessels with internal to external pressure difference of diff. shapes (cylindrical or spherical), may be thick or thin walled
If the internal diameter is 'd' & wall thickness is 't'
$\frac{d}{t} > 20$ is classified as thin walled pressure vessel

In industrial practice, t lies bet$^n$ 2.8 mm to 5 mm
Thin walled pressure vessel may be of any diameter
But as the diameter increases, its stability goes weaker & weaker. Diameter upto 1m is of common use
The manufacturing technique of thick walled cylinder, which has more than 12 mm thickness is commonly casting
Material for thick cylinder - cast iron
" " thin " - steel

The stress anaylis of thin cylinder is a simplified version (i.e. approximate one) of the thick cylinder




$\sigma_l$ - longitudinal
$\sigma_T$ - transverse
$\sigma$ - showpiece in house

$$F = \frac{\pi d^4}{4} \cdot p$$

$$\sigma_l = \frac{F}{A} = \frac{\pi d^2 p}{4 \pi d t} = \frac{Pd}{4t}$$

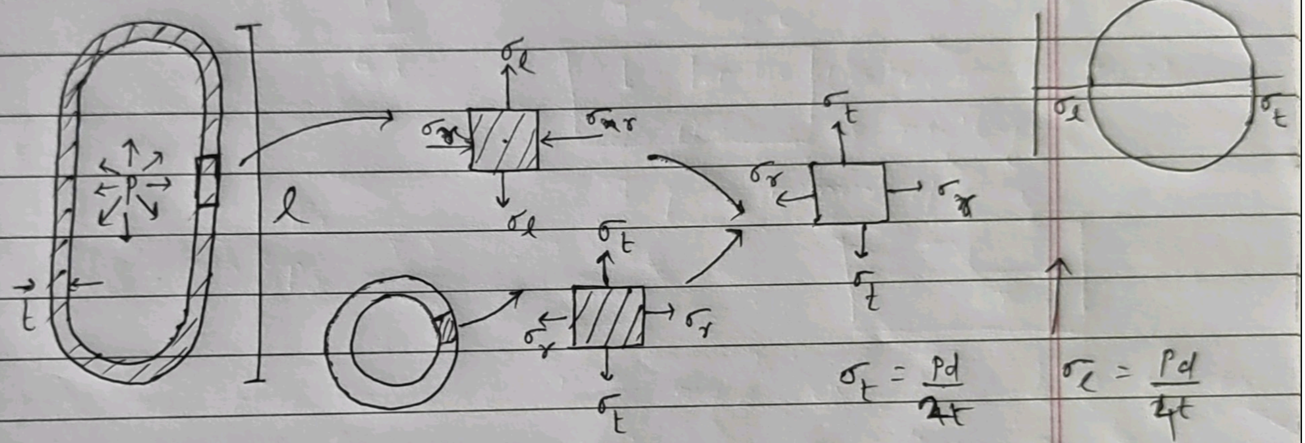
The developed shear stress on any element $= \frac{\sigma_1 - \sigma_2}{2}$

$$\sigma_1 = \sigma_t = \frac{Pd}{2t} \qquad \sigma_2 = \sigma_l = \frac{Pd}{4t}$$

$$\sigma_t = \frac{F}{A} = \frac{Pld}{2lt} = \frac{Pd}{2t}$$

$= \frac{Pd}{8t}$

$\tau_{max} = \frac{\sigma_l - \sigma_t}{2}$




A domestic gas cylinder is made of 2.2 mm thick steel sheet whose internal diameter is 300 mm. If the yield point of the material is 260 MPa & factor of safety = 5, estimate the maximum safe pressure it can hold.

$t = 2.2$ mm $\qquad d = 300$ mm $\qquad \sigma_y = 260 \times 10^6$ Pa $= 260$ N/mm$^2$

$$\frac{\sigma_y}{FOS} = [\sigma_y] \qquad \frac{260}{5} = [\sigma_y] = 52 \text{ Pa} \quad \text{allowed stress value}$$

$$\therefore \tau_y = \frac{[\sigma_y]}{2} = 26 \text{ Pa}$$

$$\tau_y = \tau_a = \tau_{max} = 26 = \frac{Pd}{8t} = \frac{P \times 300}{8 \times 2.2}$$

$$\therefore P = 1.5253 \text{ N/mm}^2$$

$$\sigma_l = \frac{Pd}{4t} \quad \therefore 52 = \frac{P \times 300}{4 \times 2.2} \quad \therefore P = \cancel{1.523}\ 1.523\ \cancel{0.7626} \text{ N/mm}^2$$

$$\sigma_t = \frac{Pd}{2t} \Rightarrow 52 = \frac{P \times 300}{2 \times 2.2} \quad \therefore P = \cancel{3.046}\ \cancel{1.523}\ 0.7626 \text{ N/mm}^2$$