MECHANICAL ENGINEERING

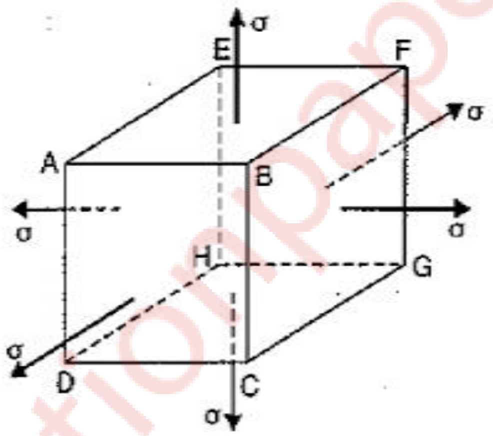
STRENGTH OF MATERIALS

CBCGS (MAY- 2018) Q.P.Code: 27637

1.(a) Define bulk modulus. Derive an expression for Youngs's modulus, in terms of bulk modulus and Poisson's Ratio. (5)

When a body is subjected to three mutually perpendicular like stresses of same intensity then the ratio of direct stress and the corresponding volumetric strain of the body is constant and is known as bulk modulus. It is denoted by K and unit is N/mm^2.

Let us consider a cube ABCDEFGH as displayed in following figure, let us assume that cube is subjected with three mutually perpendicular tensile stress σ of similar intensity.



Let us assume that we have following details as mentioned here

Length of cube = L Change in length of the cube = dL

Young's modulus of elasticity = E Bulk modulus of elasticity = K

Tensile stress acting over cube face = σ Poisson ratio = ν

Longitudinal strain per unit stress = α Lateral strain per unit stress = β

Poisson ratio, (ν) = β / α

E = 1/ [Longitudinal strain/ Longitudinal stress]

E = 1/ α

Initial volume of the cube,

V = Length x width x height = L^3

ΔV = 3 σ. L^3 (α – 2β)

∴**Volumetric strain = ΔV/V**

∴ε_V = 3 σ (α – 2β)

Now, we will find here Bulk modulus of elasticity (K)

K = σ / [3 σ (α – 2β)]

K = 1/ [3 (α – 2β)]

$3\,K\,(\alpha - 2\beta) = 1$

$3K\,(1\text{-}2\,\beta/\alpha) = 1/\,\alpha$

Now,

Young's modulus of elasticity, $E = 1/\,\alpha$

Poisson ratio, $\nu = (\beta/\alpha)$

After replacing the value of $1/\,\alpha$ and (β/α) in above concluded equation,

$$3K\,(1\text{-}2\,\nu) = E$$

$$\mathbf{E = 3K\,(1\text{-}2\,\nu)}$$

1.(b) A short column of external diameter 400 mm and internal diameter 200 mm carries an eccentric load of 80 kN. Find the greatest eccentricity, which the load can have without producing tension on the cross section. (5)

$D = 400mm$, $d=200mm$, $P= 80kN = 80 \times 10^3$ N

For no tension condition

$\sigma_o - \sigma_b = 0$

$\therefore \sigma_o = \sigma_b$

$\therefore \frac{W}{A} = \frac{M}{Z} = \frac{W.e}{Z}$

$\therefore e \leq \frac{Z}{A}$

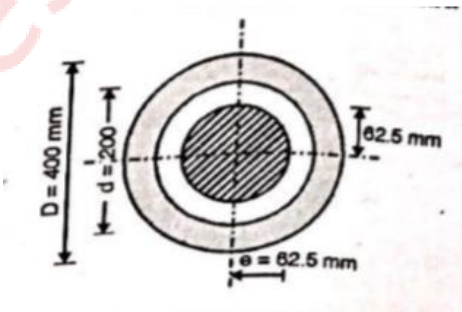
For circular section

$Z = \frac{\pi(D^4 - d^4)}{32D}$ $\quad\quad$ $A = \frac{\pi(D^2 - d^2)}{4}$

$\therefore e \leq \frac{\frac{\pi(D^4-d^4)}{32D}}{\frac{\pi(D^2-d^2)}{4}} \leq \frac{D^2+d^2}{8D}$

$\therefore e \leq \frac{400^2+200^2}{8 \times 400}$

$\therefore \mathbf{e \leq 62.5\ mm}$



1.(c) State the assumptions in the theory of pure bending and derive the formula, $\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$ (5)

1.The material of the beam is homogeneous and isotropic.

2.The value of Young's Modulus of Elasticity is same in tension and compression.

3.The transverse sections which were plane before bending, remain plane after bending also.

4.The beam is initially straight and all longitudinal filaments bend into circular arcs with a common centre of curvature.

5.The radius of curvature is large as compared to the dimensions of the cross-section.

6.Each layer of the beam is free to expand or contract, independently of the layer, above or below it.

Let,

M = bending moment acting on beam

θ = Angle subtended at centre by the arc.

R = Radius of curvature of neutral layer M' N'.

At any distance 'y' from neutral layer MN, consider layer EF.

As shown in the figure the beam because of sagging bending moment. After bending, A'B', C'D', M'N' and E'F' represent final positions of AB, CD, MN and EF in that order.

When produced, A' B' and C' D' intersect each other at the O subtending an angle θ radian at point O, which is centre of curvature.

As L is quite small, arcs A' C', M' N', E' F' and B' D' can be taken as circular.

Now, strain in layer EF because of bending can be given by e = (E F - EF)/EF = (E F - MN)/MN

As MN is the neutral layer, MN = M' N'

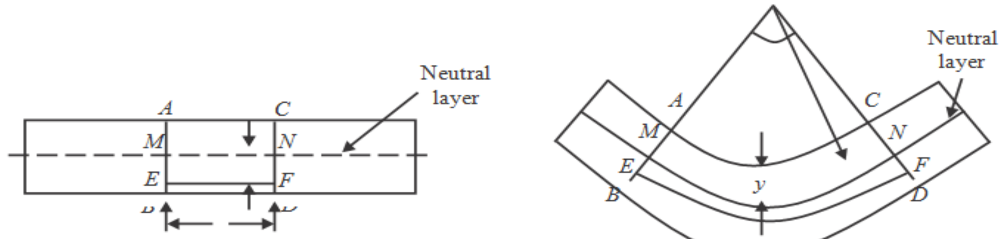
$e = \frac{E'F' - M'N'}{M'N'} = \frac{(R+y)\theta - R\theta}{R\theta} = \frac{y\theta}{R\theta} = \frac{y}{R}$ …….(i)

Let σ = stress set up in layer EF because of bending and E = Young's modulus of material of beam.

$E = \frac{\sigma}{e}$ or $e = \frac{\sigma}{E}$ …….(ii)

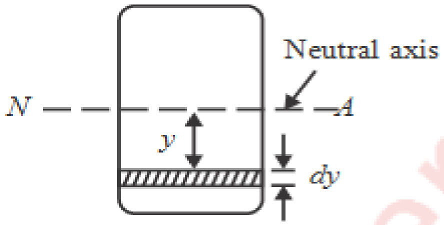
Equate the equation (i) and (ii);

$\frac{y}{R} = \frac{\sigma}{E}$ ….(iii)



Before bending (a) After bending (b)

$\sigma/y = \boldsymbol{E/R}$



At distance 'y', let us consider an elementary strip of quite small thickness dy. We have already assumed that 'σ ' is bending stress in this strip.

Let dA = area of the elementary strip. Then, force developed in this strip = $\sigma.dA$.

Then the, elementary moment of resistance because of this elementary force can be
given by $dM = f.dA.y$

Total moment of resistance because of all such elementary forces can be given by

$$\int dM = \int \sigma \times dA \times y$$

$$M = \int \sigma \times dA \times y \qquad \text{….(iv)}$$

From the Equation (iii),

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

By putting this value of σ in Equation (iv), we get

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

But

$$\int dA \cdot y^2 = 1$$

where I = Moment of inertia of whole area about neutral axis N-A.

$M = (E/R) \cdot I$

$M/I = E/R$

$M/I = \sigma/y = E/R$

Where;

M = Bending moment

I = Moment of Inertia about axis of bending that is I_{xx}

y = Distance of the layer at which the bending stress is consider

E = Modulus of elasticity of beam material.

R = Radius of curvature

1.(d) Find the maximum shear stress induced in a solid circular shaft of diameter 150 mm, when it transmits 150 kW power at 180 rpm. (5)

d=150mm P=150kW N=180rpm

Power transmitted by shaft:

$P = \frac{2\pi NT}{60} \quad \therefore 150 \times 10^3 = \frac{2\pi \times 180T}{60}$

$\therefore T = 7.958 \times 10^6 N\ mm$

Using torsional formula,

$\boldsymbol{\tau = \frac{T}{Z_p} = 12.0084\ N/mm^2}$

1.(e) A steel bar of 50 mm x 50 mm in section and 3 m length is subjected to an axial pull of 140 N. Calculate the strain energy stored in the bar. Find also the extension of the bar. Take E = 200GPa. (5)

b= 50 mm d=50mm L = 3m = 3000mm

P= 140 N E= 200 GPa = 200 x $10^3 N/mm^2$

Volume = 50^2 x 3000 = 7.5 x 10^4 mm^4

Strain energy stored in bar,

$U = \frac{\sigma^2 V}{2E}$

For gradually applied load,

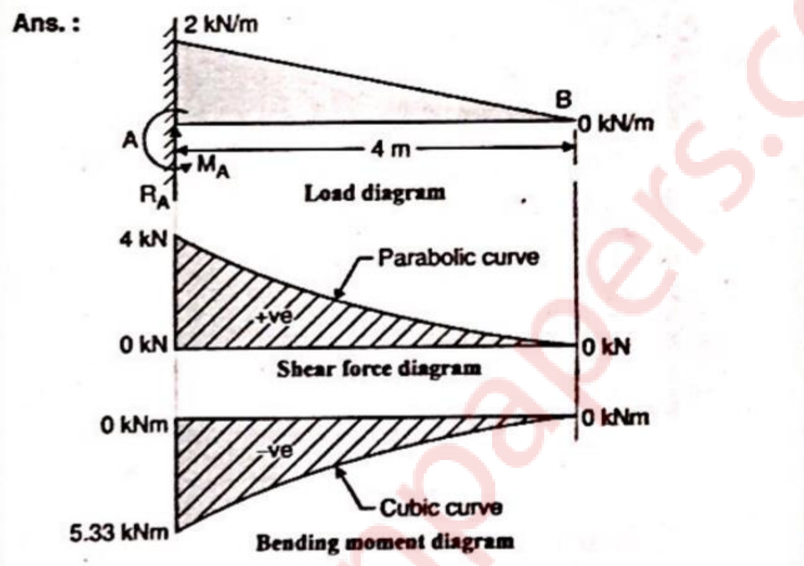
$\sigma = \frac{P}{A} \frac{140 \times 10^3}{50^3} = 56\ MPa$

$\mathbf{U} = \frac{56^2 \text{ x } 7.5 \text{ x } 10^6}{2 \text{ x } 200 \text{ x } 10^3}$ = 58800 N-mm = **58.8 J**

Extension, $\delta l = \frac{\sigma L}{E} = \frac{56 \text{ x } 3000}{200 \text{ x } 10^3}$

$\therefore \boldsymbol{\delta l = 0.84 \text{ mm}}$

1.(f) A cantilever of length 4 m carries uniformly varying load of intensities zero at free end and 2 kN/m at fixed end. Draw shears force and bending moment diagrams for the beam. **(5)**



$\sum F_y = 0$

$R_A - (\frac{1}{2} \text{ x } 2 \text{ x } 4\ ) = 0$

$\therefore R_A = 4$ kN

Let M_A is anticlockwise couple at A as shown.

$\sum M_B = 0$

$-M_A + R_A \text{ x } 4 - (\frac{1}{2} \text{ x } 2 \text{ x } 4\ ) \text{ x } \frac{1}{3} \text{ x } 4 = 0$

$\therefore\ M_A = \frac{16}{3} = 5.33$ kNm

Shear force calculations:

Shear force at A_L = 0 kN

Shear force at A_R = 4 kN

Shear force at B = 0 kN

Bending moment calculations:

Bending moment at A_L = 0 kNm

Bending moment at A_R = -5.33 kNm

Bending moment at B = -5.33 + 4 x 4 – ($\frac{1}{2}$ x 2 x 4) x $\frac{2}{3}$ x 4 = 0 kNm

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2.(a) A compound tube consists of a steel tube of 140 mm internal diameter and 160 mm external diameter and an outer brass tube of 160 mm internal diameter and 180 mm external diameter. Both the two tubes are of 1.5 m length. If the compound tube carries an axial compressive load of 900 kN, find its reduction in length. Also, find the stresses and the loads carried by each tube. (10)

$E_s = 2 \times 10^5$ N/mm^2 $\quad$ $E_b = 1 \times 10^5$ N/mm^2

Steel tube, D = 160mm, d = 140mm

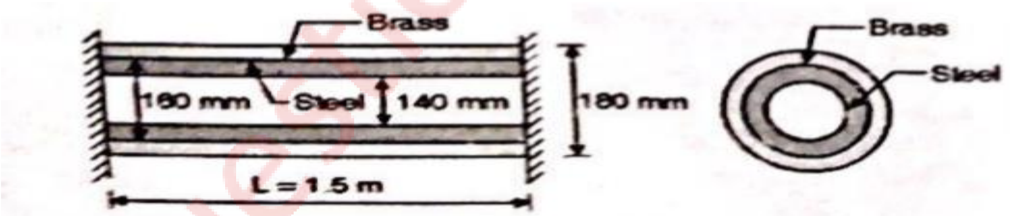
Brass tube, D = 180 mm, d = 160mm

Length L = 1.5 m = 1500 mm

Compressive load P = 900 kN = 900 x 10^3N

E_s= 2 x 10^5N/mm^2 $\quad$ E_b = 1 x 10^5 N/mm^2

Area of brass tube $A_b = \frac{\pi}{4}(D_b{}^2 - d_b{}^2) = \frac{\pi}{4}(180^2 - 160^2)$ = 5340.71 mm^2



Area of steel tube $A_s = \frac{\pi}{4}(D_b{}^2 - d_b{}^2) = \frac{\pi}{4}(160^2 - 140^2)$ = 4712.4 mm^2

For composite member subjected to axial force $e_s = e_b$

$\therefore \frac{\sigma_s}{E_s} = \frac{\sigma_b}{E_b}$ $\qquad$ $\therefore \sigma_s = \frac{2 \times 10^5}{1 \times 10^5}\sigma_b$ $\qquad$ $\therefore \sigma_s = 2\sigma_b$

Total load $P = P_b + P_s \quad = \sigma_b A_b + \sigma_s A_s$

$= \sigma_b \times 5340.71 + 2\sigma_b \times 4712.4$

$\therefore 900 \times 10^3 = 14765.5\sigma_b$

$\therefore$ **$\sigma_b = 60.75$ MPa (Compressive)**

$\therefore \boldsymbol{\sigma_s = 2\sigma_b = 121.91}$ **MPa (Compressive)**

Loads carried by each tube,

Steel, $P_s = \sigma_s A_s = 121.91 \times 4712.4 = 574.5 \times 10^3 N$

Similarly, Copper, $P_c = 325.5$ kN

Reduction in length,

$$\delta l = \frac{\sigma_s A_s}{E_s} = \frac{121.91 \times 1500}{2 \times 10^5}$$

$\therefore$ **δl = 0.914 mm**

2.(b) A point load of 10 kN applied to a simply supported beam at mid span, produces a deflection of 6 mm and a maximum bending stress of 20 N/mm^2. Calculate the maximum value of the momentary stress produced, when a weight of 5 kN is allowed to fall through a height of 18 mm on the beam at the middle of the span. (10)

Point load W_s = 10kN

Deflection δ = 6 mm

Max. bending stress σ_{max} = 20 N/mm^2

Falling weight W = 5 kN

Height of fall h = 18 mm = 0.018 m

Let W_e be the static load equivalent to the given impact load, then

$$\frac{W_e}{\delta} = \frac{W_s}{y}$$

$$\frac{10}{6 \times 10^{-3}} = \frac{W_s}{y}$$

$$\therefore y = \frac{6 \times 10^{-3} W_e}{10} = 0.0006 W_e \text{ m}$$

Work done by equivalent static load = Work done by given falling load

$$\frac{1}{2} . W_e . y = W(h+y)$$

$$\frac{1}{2} . W_e \times 0.0006 W_e = 5(0.018 + 0.0006 W_e)$$

$\therefore W_e$ =23 kN

Since a static load 10 kN induced a maximum bending stress 20 N/mm^2, then the maximum bending stress produced in impact case for which equivalent load is 23 kN

$$\frac{\sigma_{static}}{\sigma_{max}} = \frac{W_e}{W}$$

$$\therefore \boldsymbol{\sigma_{static} = \frac{23}{10} \text{ x } 20 = 46 \text{ N/mm}^2}$$

3.(a)Two mutually perpendicular planes of an element of material are subjected to tensile stress of 105 N/mm^2, compressive stress of 35 N/mm^2 and shear stress of 70 N/mm^2. Find graphically or otherwise,

i. Magnitude and direction of principal stresses

ii. Magnitude of the normal and the shear stresses on a plane, on which the shear stress is maximum. (10)

σ_s = 105 N/mm^2

σ_y = -35 N/mm^2

τ_{xy} = 70 N/mm^2

$$\tan 2\theta_{P1} = \frac{2\tau}{\sigma_x - \sigma_y} = \frac{2 \text{ x } 70}{105-(-70)} = \frac{4}{5}$$

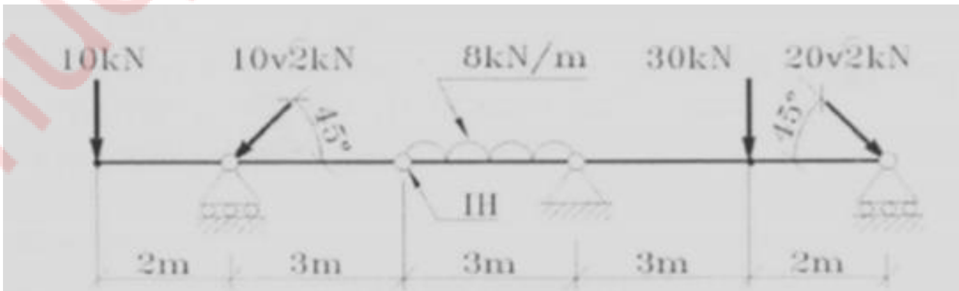
$\therefore$ $2\theta_{P1}$= 38.65 or θ_{P1}= 19.32^0 and θ_{P2} = 199.329^0

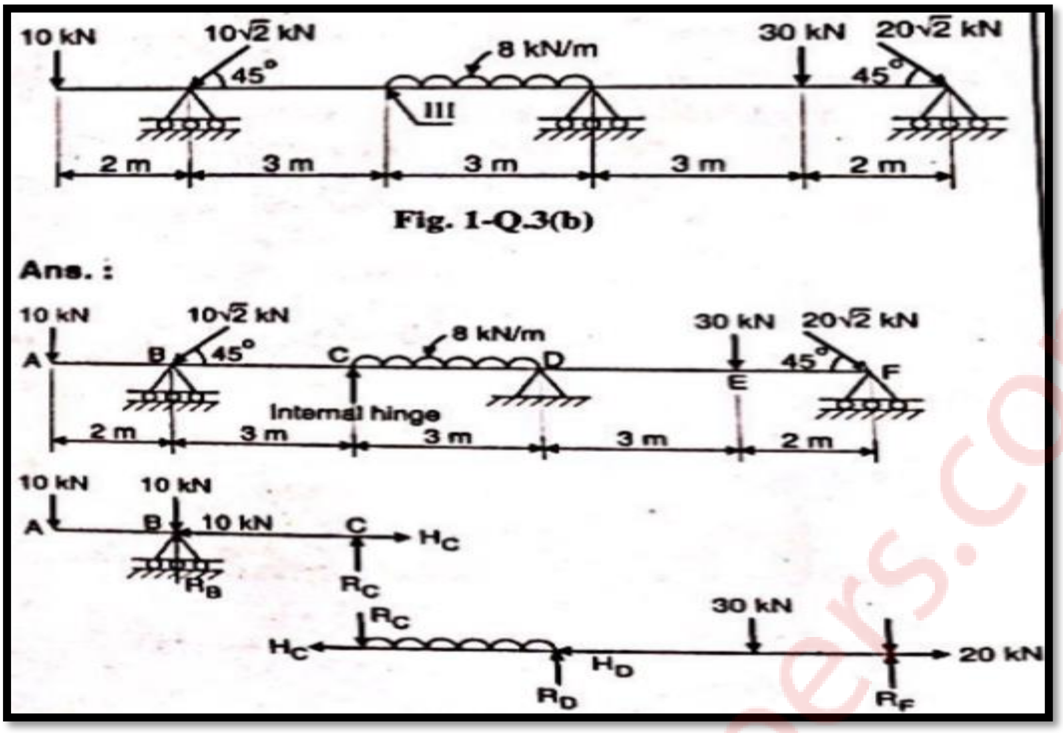
$$\sigma_{s1} = \frac{\sigma_s + \sigma_y}{2} \pm \sqrt{(\frac{\sigma_s - \sigma_y}{2})^2 + \tau_{xy}{}^2}$$

$\therefore$ $\boldsymbol{\sigma_{s1}}$ **= 133.994 N/mm^2**

And $\boldsymbol{\sigma_{s2}}$ **= -63.99 N/mm^2**

3.(b) Draw axial force, shear force and bending moment diagrams for the beam loaded as shown in figure. Locate all important points. (10)





For AC beam

$\sum F_y = 0$

$-10-10+R_B + R_C = 0$

$R_B + R_C = 20$

$\sum F_x = 0$

$-10 + H_C = 0$

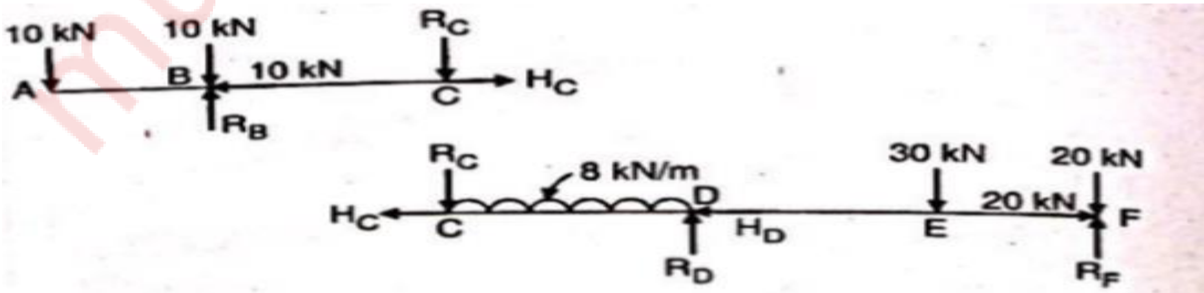
$H_C = 10$ kN

$\sum M_C = 0$

$-10 \times 5 - 10 \times 3 + R_B \times 3 = 0$

$R_B = 26.667$ kN

$R_C = -6.667$ kN



For CF beam

$\sum F_y = 0$

$R_C - (8 \times 3) + R_D - 30 - 20 + R_f = 0$

But $R_C = 6.667$

$6.667 - 24 + R_D - 50 + R_F = 0$

$R_D + R_F = 67.33$

$\sum M_F = 0$

$6.667 \times 8 - (8 \times 3) \times (6.5) + R_D \times 5 - 30 \times 2 = 0$

$\therefore 5R_D = 24 \times 6.5 + 60 - 6.667 \times 8$

$\therefore$ **R_D = 32.53 kN**

R_F = 34.79 kN

$\sum F_H = 0$

Assume H_D is left side.

$-H_C - H_D + 20 = 0$ But $H_C = 10$ kN

$-10 - H_D + 20 = 0$

H_D = 10 kN

Shear force calculations

S.F at $A_L = 0$ kN

$A_R = -10$ kN

$B_L = -10$ kN

$B_R = -10 - 10 + 26.667 = 6.667$ kN

$C_L = 6.667$ kN

$C_R = 6.667$ kN

$D_L = 6.667 - (8 \times 3) = -17.333$ kN

$D_R = -17.333 + 32.53 = 15.197$ kN

$E_L = 15.197$ kN

$E_R = 15.197 - 30 = -14.803$ kN

F_L = -14.803 kN

F_R = -14.8 -20 + 34.79 = 0kN

Bending moment calculation

B.M at A = 0

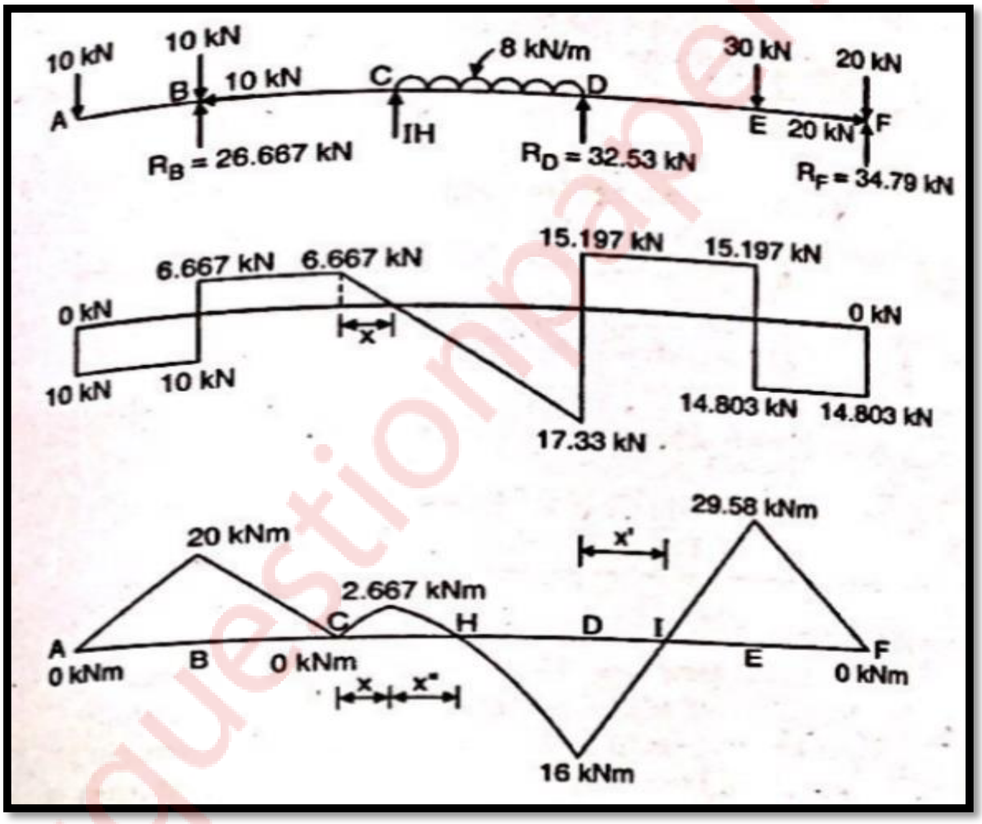
B = 10 x 2 = 20 kNm

C = 10 x 5 + 10 x 3 -26.6667 x 3 = 0

D = 34.79 x 5 – 20 x 5 -30 x 3 = -16 kNm

E = 34.79 x 2 – 20 x 2 = 29.58 kNm

F = 0 kNm



From SFD point G is the zero-shear force need to be locate and find the B.M at C_2

To locate point G

Let CG = x m

$\therefore \frac{6.667}{x} = \frac{17.33}{3-x}$

$\therefore x = 0.835$ m

Bending moments at G = -8 x 0.835 x $\frac{0.835}{2}$ + 6.667 x 0.833 = 2.77 kNm

Again there are two important point H and I where B.M. is zero.

$\therefore \frac{16}{x'} = \frac{29.58}{3-x'}$

48 – 16 x$'$ = 29.58 x$'$

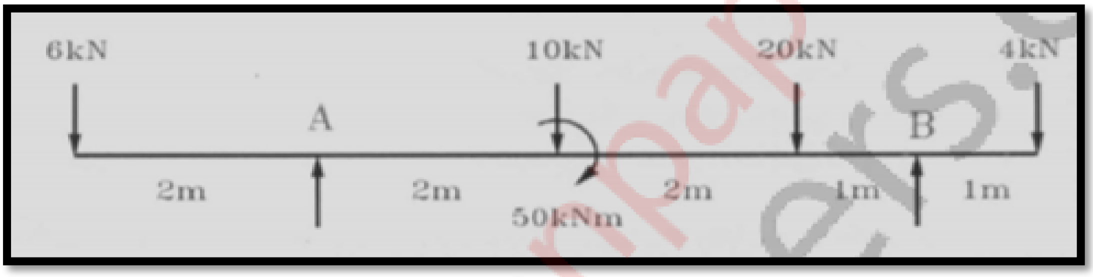
$\therefore$ x$'$ = 1.053 m

To locate point H

B.M at H = 0 = 6.667 x x'' – 8 x x x $\frac{x''}{2}$

$\therefore$ x$''$ = 1.6667 m

4.(a) Determine the position and the amount of maximum deflection for the beam shown in figure. Take EI = 1.8 x 10^4 kNm^2 **(10)**



Step 1: Support Reactions

$\sum M = 0$

-6 x 2 + 10 x 2 + 50 +20 x 4– R_B x 5 + 4 x 6= 0

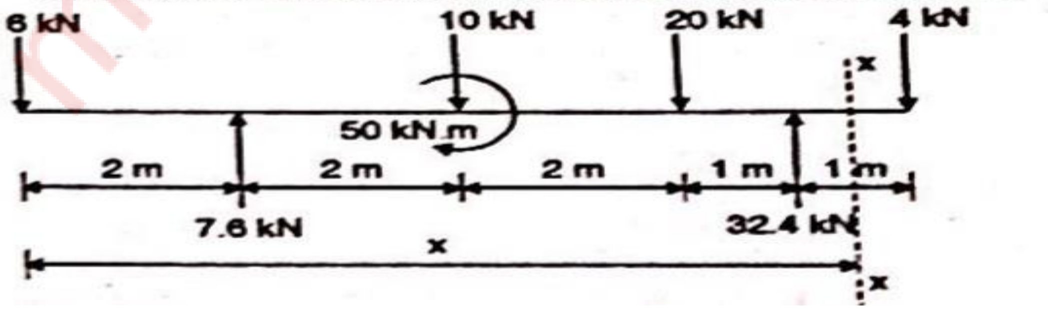
$\therefore$ R_B = 32.4 kN

$\sum F = 0$

$\therefore$ R_A = 115.5 kN

Step 2 : Deflection using Macaulay's method

Consider a section x at a distance x-x from left end of a beam.



$EI\frac{d^2y}{dx^2}$ = -6x| + 7.6(x-2) |- $-10(x-4)$| + 50(x-4)^0| - $20(x-6)$

Slope Equation

$EI\frac{dy}{dx} = C_1 - \frac{6x^2}{2} | + \frac{7.6\,(x-2)^2}{2} | - 5(x-4)^2| + 50(x-4)| - 10(x-6)^2| + 16.2(x-7)^2$

Deflection equation

$EI.y = C_1x + C_2 - x^2| + 1.267(x-2)^3| - 1.857(x-4)^2| + 25(x-4)^2| - 3.33(x-6)^3| + 5.4(x-7)^3$

At x=2, y=0 $\therefore$ $2C_1 + C_2 = 8$ ….(i)

At x=7, y=0 $\therefore$ $7C_1 + C_2 = 7.968$ …..(ii)

$C_1 = -6.4 \times 10^{-9}$ and $C_2 = 8.0128$

Step 3 : Location of maximum deflection

Putting values of C_1 and C_2 in slope equation,

$18x^2 - 79.2x + 359.2 = 0$

$\therefore$ **x = 5.314 m**

Step 4: Maximum deflection

Putting values of C_1 and C_2 in deflction equation,

y_{max} = 3.25mm

4.(b) A weight of 200 kN is supported by three adjacent short pillars in a row, each 500 mm^2 in section. The central pillar is made of steel and the outer ones are of copper. The pillars are adjusted such that at 15^{0}C each carries equal load. The temperature is then raised to 115^{0}C. Estimate the stresses in each pillar at 15^{0}C and 115^{0}C. Take $E_S = 0.8 \times 10^5$ N/mm^2, α_s = 1.2 x 10^{-5}/^{0}C, α_c = 1.85 x 10^{-5}/^{0}C (10)

$P = 200kN = 200 \times 10^3$ N

Now, $\delta l_s = \delta l_c$

$\therefore \frac{\sigma_S\,.l}{2\,x\,10^5} = \frac{\sigma_C\,.l}{0.8\,x\,10^5}$

$\therefore \sigma_s = 2.5\sigma_c$

Also,

$200 \times 10^3 = \sigma_s A_s + \sigma_c A_c$

Putting the values of σ_s and A_s, A_c

We get,

$\sigma_c = 114.28$ N/mm^2

$\therefore \sigma_s = 285.7$ N/mm^2

Stresses before the rise in temp = 285.7 N/mm^2

When temperature is increased,

Force in steel = Force in copper

$P_s = P_c$

$\therefore \sigma_s A_s = \sigma_c A_c$

Now, $A_s = A_c$

$\therefore \sigma_s = \sigma_c$

Now,

$$\frac{\sigma_s}{E_s} + \frac{\sigma_c}{E_c} = t\,(\alpha_c - \alpha_s)$$

Putting the respective values, we get

$\sigma_c(1.75 \times 10^{-5}) = 6.5 \times 10^{-4}$

$\therefore \sigma_c = 37.14 = \sigma_s$

5.(a) A hollow shaft, having an internal diameter 40% of its external diameter, transmits 562.5kW power at 100 rpm. Determine external diameter of the shaft, if shear stress is not to exceed 60 N/mm^2, and the twist in a length of 2.5 m should not exceed 1.3^0. Assume torque is 1.25 times the mean torque and G= 9 x 10^4 N/mm^2. (10)

Length L = 2.5m = 2500mm

Internal diameter = 40% external diameter = 0.4D

Power P = 562.5 kW = 562.5×10^3 W

No of revolutions = 100 rpm.

Shear stress $\tau = 60$ N/mm^2

Angle of twist $\theta = 1.3^0 = 1.3 \times \frac{\pi}{180} = 0.0227$ radian

Maximum torque = 1.25 x mean torque

Shear modulus G = 9 x 10^4 N/mm^2

To find: External diameter D

Power transmitted P = $\frac{2\pi NT}{60}$

$$562.5 \times 10^3 = \frac{2\pi \times 100 \times T_{mean}}{60}$$

$$\therefore T_{mean} = 53.715 \times 10^3 \text{ Nm} = 53.715 \times 10^6 \text{ Nmm}$$

Maximum torque T_{max} = 1.25 T_{mean} = 1.25 x 53.715 x 10^6 N-mm

Condition (i): Diameter based on shear stress:

$$\frac{T}{J} = \frac{\tau}{R}$$

$\therefore \frac{67.14 \times 10^6}{\frac{\pi}{32}(D^4 - d^4)} = \frac{60}{D/2}$

Putting d=0.4D, we get

D =180mm

Condition (ii): Diameter based on angle of twist:

$$\frac{T}{J} = \frac{G\theta}{L}$$

$$\frac{67.14 \times 10^6}{\frac{\pi}{32}(D^4 - d^4)} = \frac{9 \times 10^4 \times 0.0227}{2500}$$

$\therefore$ D = 171.2 mm

Safe diameter is the greater value from condition (i) and(ii)

$\therefore$ D = 180mm

5.(b) A closed cylindrical vessel made of steel plates 4 mm thick with plane ends carries fluid under a pressure of 3 N/mm^2. The diameter of the cylinder is 250 mm and length are 750 mm. Calculate the longitudinal and hoop stresses in the cylinder wall and determine the changes in diameter, length and volume of the cylinder. (10)

E = 2.1 x 10^5 MPa $\frac{1}{m}$ = 0.236

L = 750mm D=250mm

$p = 3\ N/mm^2$ $t = 4\ mm$

$E = 2.1 \times 10^5$ MPa $\frac{1}{m} = 0.236$

Hoop stress $\sigma_H = \frac{pd}{2t}$

$$\therefore \boldsymbol{\sigma_H = 93.75\ MPa}$$

Longitudinal stress $\sigma_L = \frac{pd}{4t}$

$$\therefore \boldsymbol{\sigma_L = 46.875\ MPa}$$

Hoop Strain $e_H = \frac{1}{E}(\sigma_H - \mu\sigma_L)$

$$\therefore \boldsymbol{e_H = 3.826 \times 10^{-6}}$$

Change in diameter $e_H = \frac{\delta D}{D}$

$\therefore$ **δD** $= e_H \times D = 3.826 \times 10^{-4} \times 250 =$ **0.096 mm**

6.(a) A hollow cast iron column of 200 mm external diameter 150 mm internal diameter and a 8 m long has both ends fixed. It is subjected to axial compressive load. Taking factor of safety as 6, σ_c = 560 N/mm², $\alpha = \frac{1}{1600}$, determine the safe Rankine load. **(10)**

D= 200 mm d= 150mm

L= 8m = 800mm

Column is fixed at both ends

$L_e = \frac{L}{2} = 4000$ mm

$\sigma_c = 560\ N/mm^2$ FOS=6

M.I. $= \frac{\pi}{64}(D^4 - d^4) = 53.68 \times 10^6\ mm^4$

Area $= \frac{\pi}{4}(D^2 - d^2) = 13.74 \times 10^3\ mm^2$

$K = \sqrt{\frac{I}{A}} = \sqrt{\frac{53.68 \times 10^6}{13.74 \times 10^3}} = 62.50$ mm

$\therefore$ Rankine crippling load $= \frac{\sigma_c . A}{1+\alpha(\frac{L_e}{k})^2} = 360.21$ kN

∴ Safe Rankine Load = 360.21 kN

6.(b) A simply supported beam carries a UDL of intensity 2.5 kN/m over a span of 5m. The cross section is T section having flange 125 mm x 125 mm and web 175 mm x 25 mm. Calculate maximum bending stress and shear stress for the section of the beam. Also, draw the shear stress distribution diagram for maximum shear force. (10)

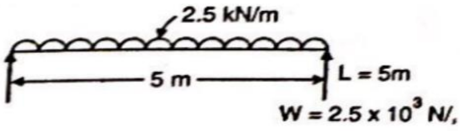
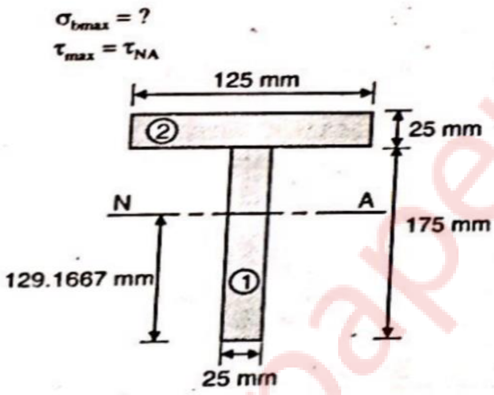


Fig. 1-Q.6(b)



Let us find I_{xx} for the given c/s of beam

$X_1 = \frac{175}{2} = 87.5$mm $\qquad$ $A_1 = 175 \times 25 = 4375$ mm^2

$X_1 = 175 + \frac{25}{2} = 187.5$mm $\qquad$ $A_2 = 125 \times 25 = 3125$ mm^2

$\therefore \bar{X} = \frac{A_1X_1 + A_2X_2}{A_1+A_2} = 129.1667$ mm

$I_{xx} = (I_{x1} + A_1h_1^2) + (I_{x2} + A_2h_2^2)$

$= 11.664 \times 10^6 + 10.861 \times 10^6$

$\mathbf{I_{xx} = 22.525 \times 10^6}$ **mm^4**

Bending stress equation

$$\frac{M}{I_{xx}} = \frac{\sigma_b}{y}$$

$\therefore \sigma_b = \frac{M}{I_{xx}} . y$

As the beam is simply supported on the upper fibre the bending stress is compressive and lower fiber is in tensile.

$\therefore y_c = 70.833$ mm $\qquad y_t = 129.16667$ mm

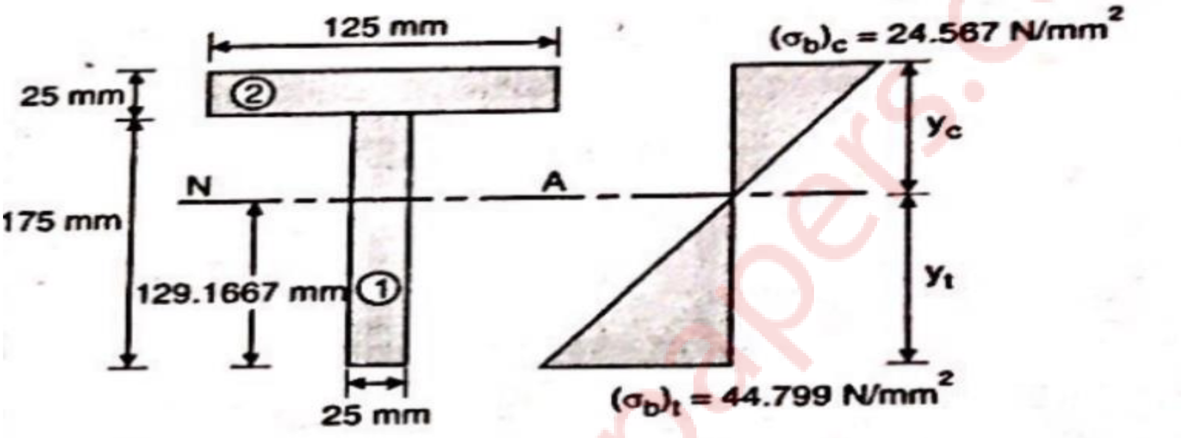
Let us find maximum bending moment for simply supported beam with UDL

$M = \frac{WL^2}{8} = 7.8125 \times 10^3$ Nm

Now,

$\sigma_{bc} = \frac{M}{I_{xx}} . y_c = 24.567$ N/mm²

$\sigma_{bt} = \frac{M}{I_{xx}} . y_t = 44.799$ N/mm2



Shear stress equation

$\tau = \frac{SAy}{\tau b}$ $\qquad$ let $S = \frac{Wl}{2} = 625 \times 10^3$N

$\tau_1 = 0 = \tau_4$ as area above and below section is zero

$\tau_2 = \frac{SAy}{\tau b_2} = 2.0232$ N/mm²

$\tau_3 = \frac{\tau_2 \ x \ b_2}{b_3} = 0.289$ N/mm²

$\tau_{NA} = 2.314$ N/mm²