Important Instruction to Examiners:-

1) The answers should be examined by key words & not as word to word as given in the model answers scheme.

2) The model answers & answers written by the candidate may vary but the examiner may try to access the understanding level of the candidate.

3) The language errors such as grammatical, spelling errors should not be given more importance.

4) While assessing figures, examiners, may give credit for principle components indicated in the figure.

5) The figures drawn by candidate & model answer may vary. The examiner may give credit for any equivalent figure drawn.

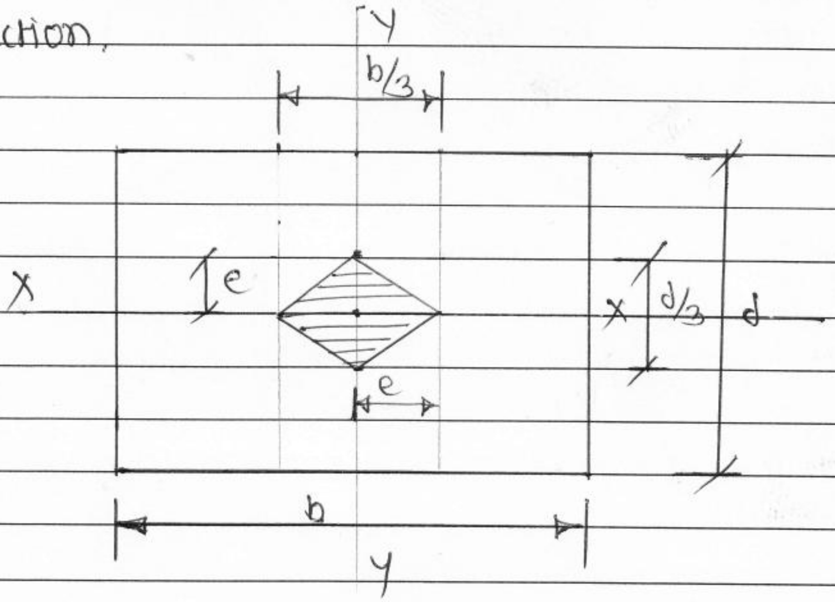
5) Credit may be given step wise for numerical problems. In some cases, the assumed contact values may vary and there may be some difference in the candidate's answers and model answer.

6) In case of some questions credit may be given by judgment on part of examiner of relevant answer based on candidates understanding.

7) For programming language papers, credit may be given to any other programme based on equivalent concept.

Important notes to examiner

Q-1	A> Attempt any six of the following	
	a)i> Define elasticity & plasticity	
	the property by virtue of which a material regains it's shape & size on removal of external load is called as elasticity all metals possesses the property of elasticity.	1 M
	ii) plasticity :- it is the property of a material due to which material undergo permanent deformation without failure or rupture on application of load in short, lack of elasticity is called plasticity.	1 M
	b> i> principal planes :- A planes across which only the normal stresses acts with no shear stress is called as principal planes	1 M
	ii) principal stresses :- The normal stresses or normal stresses acting on the principal plane is called as principal stresses	1 M
	c> Find polar moment of inertia of circle of 50mm diameter.	
→	$d = 50mm$, I_p = Polar M.I	
	$I_p = \frac{\pi}{32} d^4 = \frac{\pi}{32} \times 50^4$	1 M formula
	$I_p = 613.592 \times 10^3 \ mm^4$	1 M calculation

d)	State the expression for power transmitted by a shaft giving meaning of each term used.	
	power = $\left(\frac{2\pi N T}{60}\right)$ watts	01M
	where P = power in watts T = Avg. torque or mean torque in N-m N = Number of revolution of shaft per minute	01M
e)	Draw a neat sketch to show core of a rectangular section.	
		02M
f)	State HOOK's law it state that when a elastic material is loaded within it's elastic limit, then stress produced in directly proportional to strain	01M
	mathematically $\sigma \propto e$ σ = stress e = strain or linear strain	01M

g) write the equation of circumferential stress in thin cylinder and explain each term

$$\sigma_c = \frac{p \cdot d}{2t}$$ where 01M

p = intensity of internal pressure

d = Diameter of the shell 01M

t = thickness of the shell

h) State the condition for no tension at the base of a column.

Direct stress (σ_o) = max. bending stress (σ_b)

$\sigma_o = \sigma_b$ ----- (No tension condition) 02M

Q) Attempt any two of the following

a) A mild steel flat 150mm wide by 20mm thick, 6m long, carries an axial pull of 300 KN. if the modulus of elasticity of steel is 200 KN/mm² & poisson's ratio = 0.25 calculate the change in length width, thickness volume of the flat.

i) To find change in length (δL)

$$\sigma = \frac{P}{A} = \frac{P}{b \times d} = \frac{300}{150 \times 20} = 0.1 \text{ KN/mm}^2$$ ½M

$$e = \frac{\sigma}{E} = \frac{0.1}{200} = 5 \times 10^{-4}$$ ½M

But $e = \frac{\delta L}{L}$ $\therefore \delta L = e \times L$ $\therefore \delta L = 5 \times 10^{-4} \times 6000$

$\boxed{\delta L = 3\text{mm (increase)}}$ 01M

To find change in width (δb) & thickness (δt)

lateral strain (e_L) = $\mu \cdot e = 0.25 \times 5 \times 10^{-4} = 1.25 \times 10^{-4}$

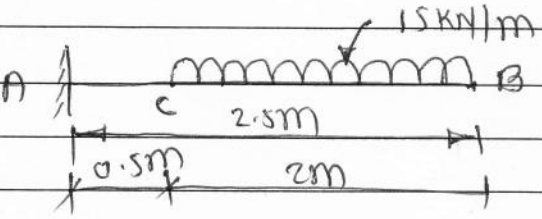
But lateral strain = $\frac{\delta b}{b} = \frac{\delta t}{t} = 1.25 \times 10^{-4}$

$\therefore \delta b = 1.25 \times 10^{-4} \times 150 = 0.01875$ (decrease) ½M
$\delta t = 1.25 \times 10^{-4} \times 20 = 2.5 \times 10^{-3}$ (decrease) ½M

To find change in volume (δv)
we know $\frac{\delta v}{v} = e(1-2\mu)$ ½M

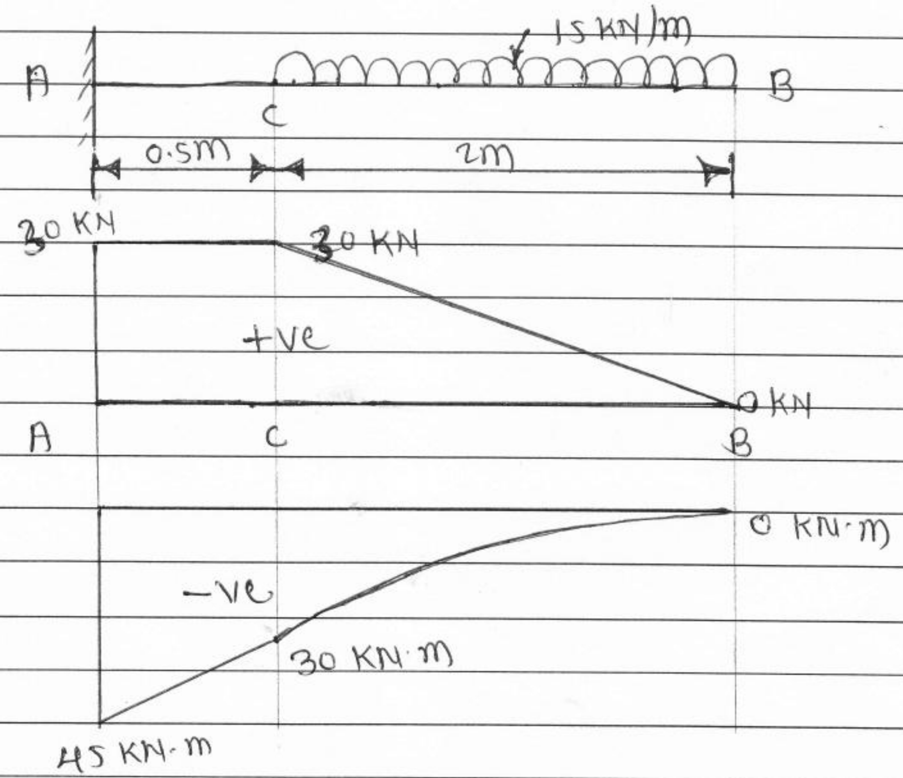
$\therefore \delta v = e(1-2\mu)v$
$= 5 \times 10^{-4}(1-2 \times 0.25)(6000 \times 150 \times 20)$
$\therefore \boxed{\delta v = 4500\ mm^2}$ ½M

b) A cantilever beam 2.5m long carries a udl of 15 KN/m over a length of 2m from the free end. Draw shear force and Bending moment diagram.



A) S.F calculation
i) $F_B = 0$
ii) $F_{CR} = 15 \times 2 = 30$ KN
iii) $F_{CL} = 30$ KN 1M

B) B.M. calculation
i) $M_B = 0$ KN-m
ii) $M_C = -15 \times 2 \times 1 = -30$ KN-m 1M
iii) $M_A = -15 \times 2 \times (\frac{2}{2} + 0.5) = -45$ KN-m



1M

1M

c) A simply supported beam of 5m span, carrier a udl of 3 KN/m over the entire span. if the bending stress is not exceed 165 N/mm², find the value of section modulus for the beam and diameter when beam is circular.

i) Bending moment for simply supported Beam $= \frac{wL^2}{8} = \frac{3 \times 5^2}{8} = 9.375$ KN·m — 1M

ii) σ_b (permissible) = 165 N/m ... (given)

iii) $Z = \frac{M}{\sigma_b} = \frac{9.375 \times 10^3}{165} = 56.82\ m^3$ — 1M

iv) $Z = \frac{\pi d^3}{32}$ $\therefore 56.82 = \frac{\pi \times d^3}{32}$ — 1M

$\therefore \frac{56.82 \times 32}{\pi} = d^3$

$\therefore 578.75 = d^3$ — 1M

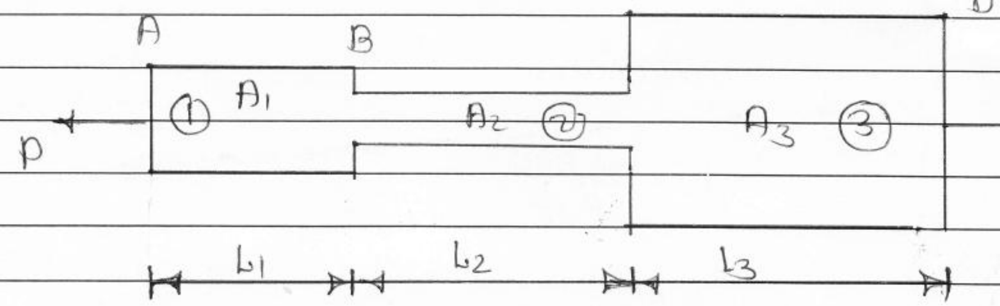
$\therefore \boxed{d = 8.33\ m}$

Q-2 (a) i) Draw the sketch of stepped section showing axial load and state expression for change in length of it

→

a metal bar of varying cross-section A_1, A_2 & A_3 and it is subjected to an axial pull P — ½M

δL = change of length, L_1 L_2 L_3 be the length of ①, ② & ③ section



½M

$$\delta L = \frac{P}{E}\left[\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3}\right] \text{ --- } \{E = \text{constant}\}$$ 01M

or — OR

$$\delta L = P\left[\frac{L_1}{A_1 E_1} + \frac{L_2}{A_2 E_2} + \frac{L_3}{A_3 E_3}\right] \text{ --- } \{E = \text{Different}\}$$ 01M

ii) state the effective length for both ends hinged column and for one end fixed and other end free

a) both ends hinged.

effective length (L_e) = L — 01M

b) one end fixed and other end free

$L_e = 2L$ — 01M

b) A steel rod 4m long 30mm diameter is used as column with one end fixed and other hinged. Determine the crippling load by Euler's formula $E = 2\times10^5$ Mpa

→ $L_{eff} = \frac{L}{\sqrt{2}} = \frac{4}{\sqrt{2}} = 2.828\,m$ 1M

$I = \frac{\pi}{64}\times30^4 = 39.76\times10^3\ mm^4$ 1M

Euler's crippling load

$$P_E = \frac{\pi^2 EI}{L^2} = \frac{\pi^2\times2\times10^5\times39.76\times10^3}{(2828)^2}$$ 1M (formula)

$$\boxed{P_E = 9813.54\ N}$$ 1M (calculation)

c) A rod 300mm long and 20mm in diameter is heated through 100°c and at the same time pulled by a force P. if the total extension is 0.4mm what is the magnitude of P? $E = 2\times10^5\ N/mm^2$ and $\alpha = 12\times10^{-6}/°c$

→ Total extension of the rod = free expansion + Extension δL due to 'P'

$0.4 = (L\alpha t) + \delta L$ due to force 'P' 1M

$0.4 = (300\times12\times10^{-6}\times100) + \delta L$

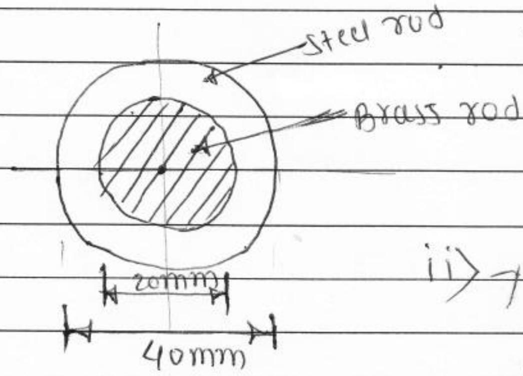
$0.4 = 0.36 + \delta L$ 1M

$\delta L = 0.4 - 0.36 = 0.04$ mm extension due to P

But $\delta L = \frac{PL}{AE}$ $\delta L = \frac{P\times300}{\left(\frac{\pi\times20^2}{4}\right)\times2\times10^5}$ 1M

$$\boxed{P = 8.377\times10^3\ N}$$ 1M

d) A brass rod of 250mm length and 20mm diameter is fixed inside a steel tube of 40mm external diameter and 20mm internal diameter and of same length. the composite bar is subjected to an axial pull of 140 KN. find the stress in each metal Take 200 Gpa & Ebrass = 110 Gpa



i) $A_{steel} = \frac{\pi}{4} \times (40^2 - 20^2)$

$A_{steel} = 942.45 \ mm^2$ — ½M

ii) $A_{brass} = \frac{\pi}{4} \times 20^2$

$A_{brass} = 314.159 \ mm^2$ — ½M

iii) To find stress in each material

σ_s = stress in steel

σ_b = stress in brass

$\frac{\sigma_s}{E_s} - \frac{\sigma_b}{E_b}$ — ½M

$\sigma_s = \frac{E_s}{E_b} \sigma_b$

$\sigma_s = \frac{200}{110} \sigma_b$ $\therefore$ $\boxed{\sigma_s = 1.818 \ \sigma_b}$ — 1M

iv) $P = P_s + P_b$

$P = 140 \ KN$ - - - given

$\therefore P = \sigma_s \cdot A_s + \sigma_b \cdot A_b$ — ½M

$140 \times 10^3 = 1.818 \sigma_b \times 942.45 + \sigma_b \times 314.159$

$140 \times 10^3 = 1713.54 \times \sigma_b + 314.159 \ \sigma_b$

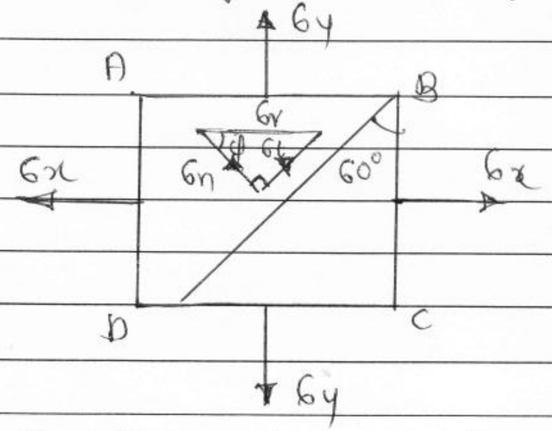
$140 \times 10^3 = 2027.70 \ \sigma_b$

$\boxed{\sigma_b = 69.0435 \ N/mm^2}$ — ½M

$\sigma_s = 1.818 \ \sigma_b$ $\sigma_s = 1.818 \times 69.0435$

$\boxed{\sigma_s = 125.52 \ N/mm^2}$ — ½M

e) The principle stresses at a point in the section of a member are 100 N/mm² an 50 N/mm² both tensile find the normal and tangential stresses across a plane passing through that point inclined at 60° to the plane have 100 N/mm² stress



$\sigma x = 100 \text{ N/mm}^2$ (T)
$\sigma y = 50 \text{ N/mm}^2$ (T)
$\theta = 60°$
$q = 0$

i) To find normal stress (σn)

$$\sigma n = \left(\frac{\sigma x + \sigma y}{2}\right) + \left(\frac{\sigma x - \sigma y}{2}\right)\cos 2\theta + q \cdot \sin 2\theta$$ 1M

$$\sigma n = \left(\frac{100+50}{2}\right) + \left(\frac{100-50}{2}\right)\cos 2 \times 60 + 0$$

$$\sigma n = 75 + 25 \times \cos 120°$$

$$\sigma n = 62.5 \text{ N/mm}^2 \text{ (T)}$$ 1M

ii) To find tangential stress (σt)

$$\sigma t = \left(\frac{\sigma x - \sigma y}{2}\right)\sin 2\theta - q\cos 2\theta$$ 1M

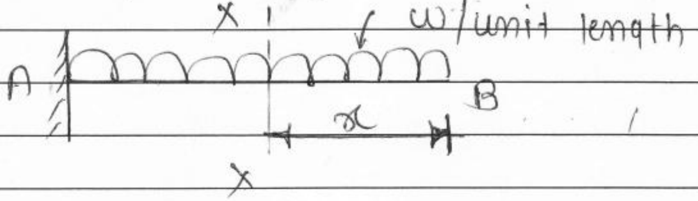
$$\sigma t = \left(\frac{100-50}{2}\right)\sin 2 \times 60 + 0$$

$$\sigma t = 25 \times \sin 120°$$

$$\boxed{\sigma t = 21.65 \text{ N/mm}^2}$$ 1M

f)	find hoop stress and longitudinal stress induced in a cylindrical boiler 1.6m internal diameter subjected to an internal pressure of 2.5 Mpa. Thickness of wall is 30mm	
→	$d = 1.6m$ $p = 2.5$ Mpa $t = 30$ mm	
	i) longitudinal stress $(\sigma_L) = \frac{pd}{4t}$	1M
	$= \frac{2.5 \times 1600}{4 \times 30}$	
	$\boxed{\sigma_L = 33.33 \ N/mm^2}$	1M
	ii) σ_c = hoop stress $= \frac{pd}{2t}$	1M
	$= \frac{2.5 \times 1600}{2 \times 30}$	
	$\boxed{\sigma_c = 66.67 \ N/mm^2}$	1M
	or	OR
	Relation betn $\sigma_c = 2\sigma_L$	1M
	$\sigma_c = 2 \times 33.33$	
	$\boxed{\sigma_c = 66.67 \ N/mm^2}$	1M

Q-3 (a) A cantilever beam of span L is subjected to udl of w/unit length over entire span. Draw S.F & B.M dia



i) S.F calculation

F_B = shear force at B = 0 kN — ½M

let shear force at any section (x-x) at distance x from free end

F_x = Rate of loading × length

$= -w \cdot x$ — ½M

Hence, we find that S.F is zero at B where x = 0 and increase linearly with increase in length x

$F_A = w \cdot L$ — ½M

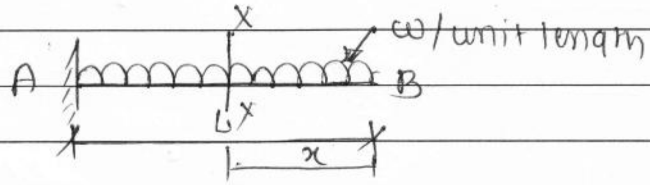
ii) B.M calculation

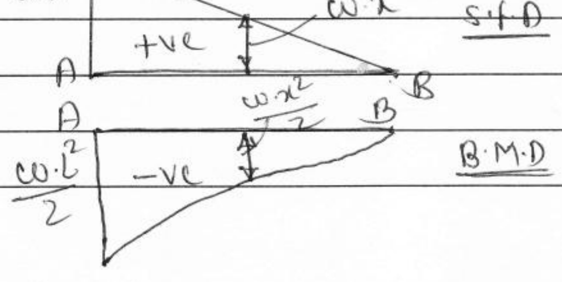
load over length x from free end of cantilever = $w \cdot x$ it acts at $x/2$ from section x-x

$M_x = -w \cdot x \times \frac{x}{2} = -\frac{wx^2}{2}$ — ½M

at $x = 0$ ∴ M_B = B.M at B = 0 — ½M

at $x = L$ ∴ M_A = B.M at A $= -\frac{w \cdot L^2}{2}$ — ½M

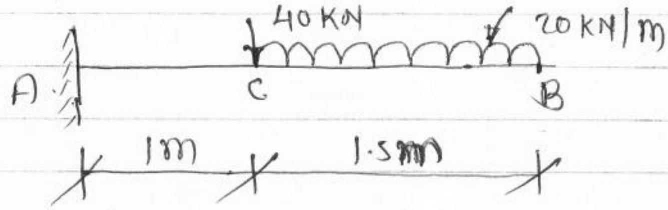




½M

½M

b)



A) S.F. calculation

i) F_B = S.F at B = 0

ii) $F_{CR} = 20 \times 1.5 = 30$ KN

iii) $F_{CL} = 20 \times 1.5 + 40 = 70$ KN

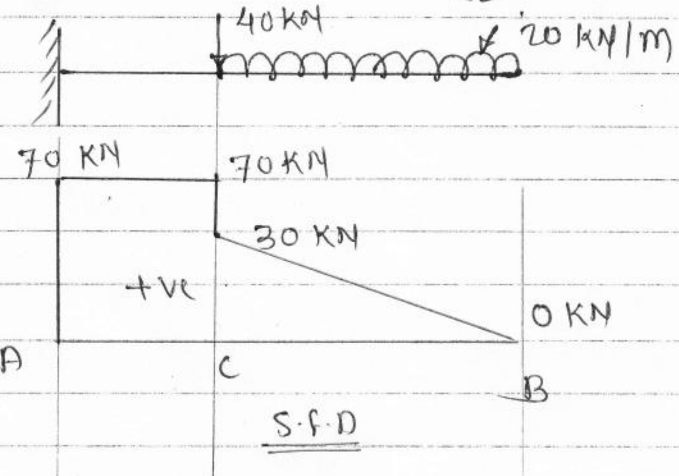
iv $F_A = 70$ KN (1M)

B) B.M. calculation

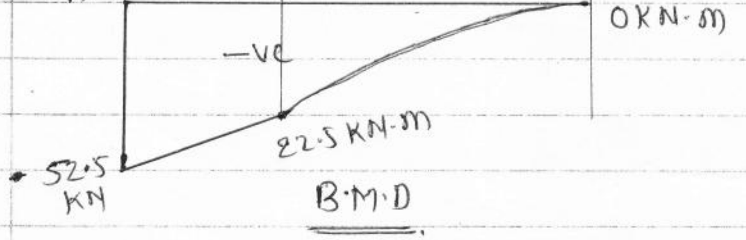
i) $M_B = 0$ KN·m

ii) $M_C = -20 \times 1.5 \times \frac{1.5}{2} = -22.5$ KN·m

iii) $M_A = -20 \times 1.5 \times \left(\frac{1.5}{2} + 1\right) = -52.5$ KN-m (1M)

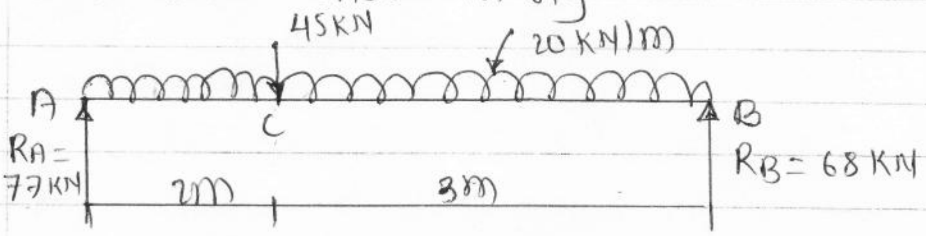


S.F.D (1M)



B.M.D (1M)

Q-3 (c) Draw S.F & B.M. diagram for the simply supported beam shown in fig



i) To find RA & RB

$\Sigma fy = 0$

$R_A + R_B = 45 + 20 \times 5$

$R_A + R_B = 145$ —— (1)

$\Sigma M_A = 0$

$-R_B \times 5 + 20 \times 5 \times \frac{5}{2} + 45 \times 2 = 0$

$-R_B \times 5 + 250 + 90 = 0$

$-R_B \times 5 + 340 = 0$

$-R_B \times 5 = -340$

$\boxed{R_B = 68\ KN}$

$\therefore \boxed{R_A = 77\ KN}$

ii) S.F calculation

$f_B = -68\ KN$

$f_{CR} = -68 + 20 \times 3 = -8\ KN$

$f_{CL} = -8 + 45 = +37\ KN$

$f_A = 37 + 20 \times 2 = +77\ KN$ 1M

iii) B.M. calculation

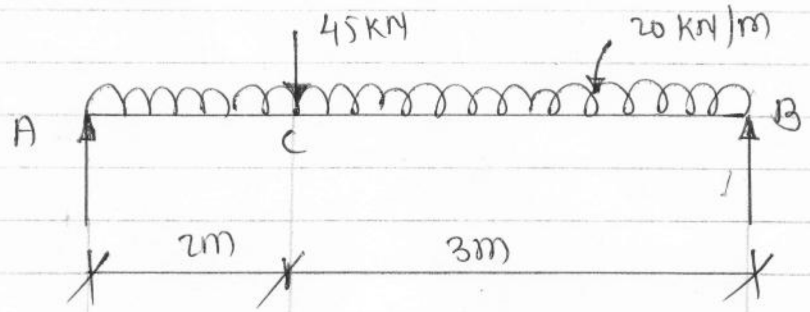
$M_B = 0\ KN\text{-}m$

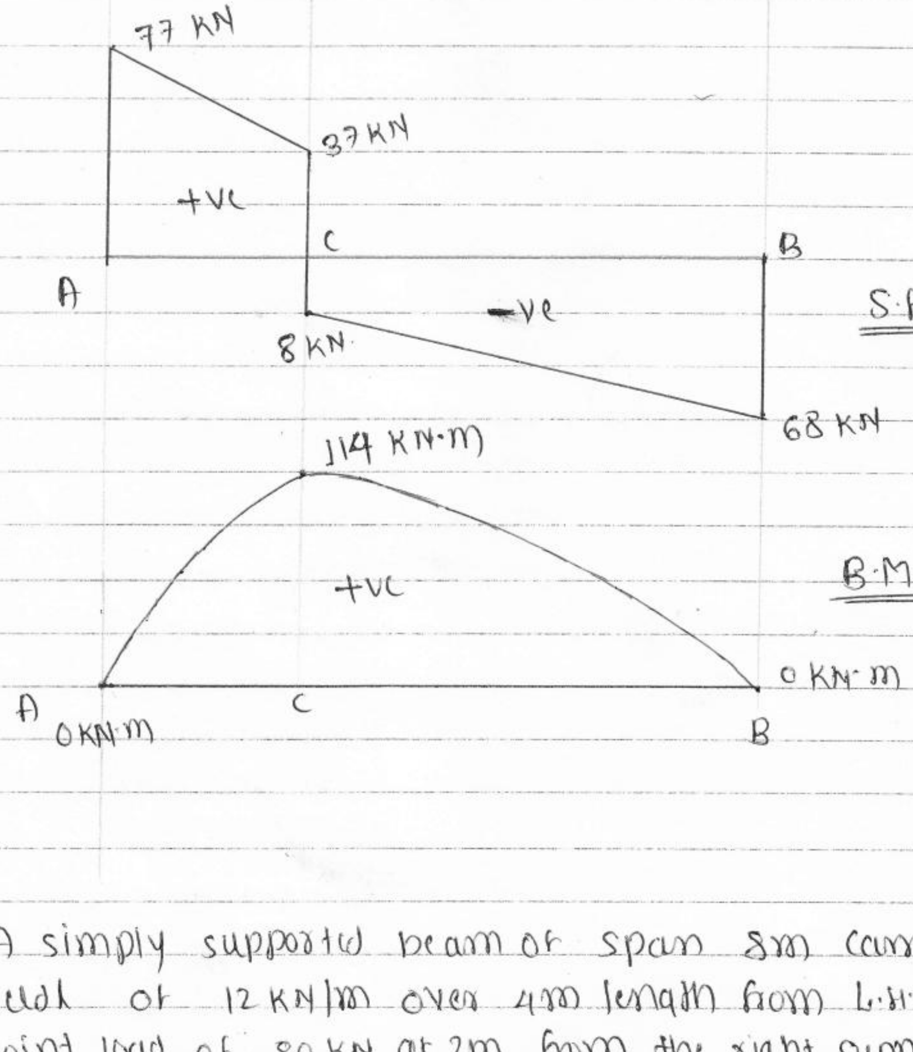
$M_C = R_B \times 3 - 20 \times 3 \times 3/2$

$M_C = 68 \times 3 - 20 \times 3 \times 3/2$

$M_C = +204 - 90$ $\boxed{M_C = 114\ KN.m}$ 1M

$M_A = 0\ KN.m$

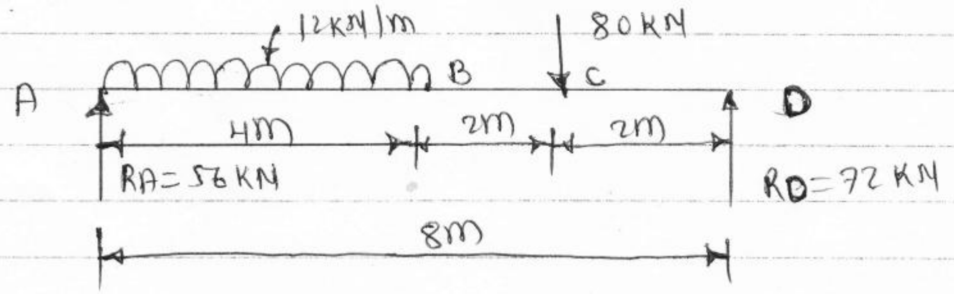




S.F.D 1M

B.M.D 1M

d) A simply supported beam of span 8m carries a udl of 12 KN/m over 4m length from L.H.S and a point load of 80 KN at 2m from the right support. Draw S.F & B.M. diagram



i) Reaction calculation

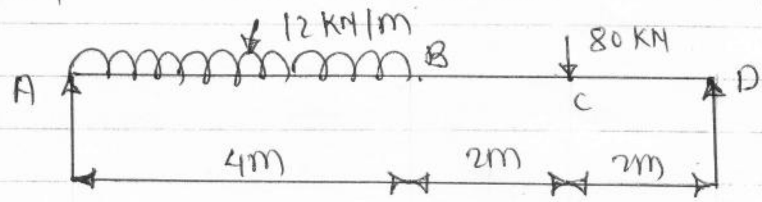
$\Sigma fy = 0$
$R_A + R_B = 12 \times 4 + 80$
$R_A + R_B = 128$ —— (1)

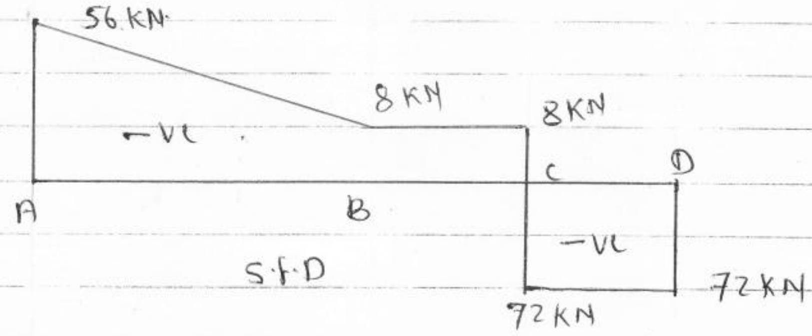
$\Sigma M_A = 0$
$-R_B \times 8 + 80 \times 6 + 12 \times 4 \times 2 = 0$
$-R_B \times 8 + 480 + 96 = 0$
$\boxed{R_D = 72\ KN}$

$\therefore \boxed{R_A = 56\ KN}$

ii) S.F.D calculation
$F_A = 56\ KN$
$F_B = 56 - 12 \times 4$
$F_B = 8\ KN$
$F_{CL} = 8\ KN$
$F_{CR} = 8 - 80 = -72\ KN$
$F_D = -72\ KN$ 1M

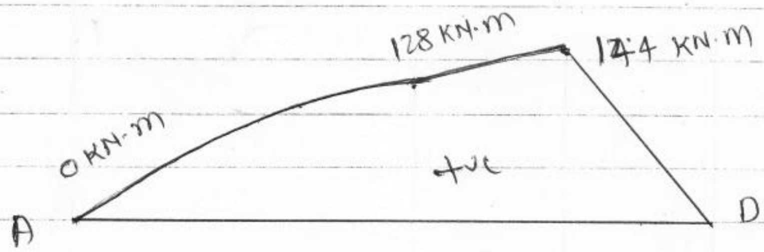
iii) B.M.D. calculation
$M_A = 0\ KN \cdot m$
$M_C = R_D \times 2 = 72 \times 2 = 144\ KN \cdot m$
$M_B = R_D \times 4 - 80 \times 2$
$= 72 \times 4 - 80 \times 2$
$= 288 - 160$
$= 128\ KN \cdot m$
$M_D = 0\ KN \cdot m$ 1M





S.F.D

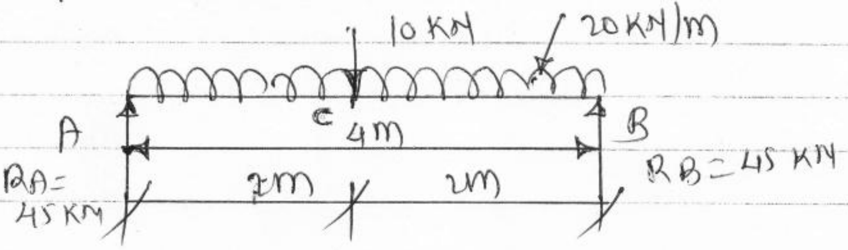
1M



1M

B.M.D

e) A beam ABC 6m long is supported at A & B 4m apart, BC = 2m the beam carries a udl of 20 KN/m over entire length along with a downward load a 10KN at point 'C' plot S.F.D & B.M.D for Beam & locate point of contraflexure



i) Reaction calculation

$\Sigma F_y = 0$

$R_A + R_B = 10 + 20 \times 4$

$R_A + R_B = 10 + 80$

$R_A + R_B = 90$ ——— (1)

$\Sigma M_A = 0$

$-R_B \times 4 + 20 \times 4 \times 2 + 10 \times 2 = 0$

$-R_B \times 4 + 160 + 20 = 0$

$-R_B \times 4 + 180 = 0$

$-R_B \times 4 = -180$

$\boxed{R_B = 45\ KN}$

$\therefore \boxed{R_A = 45\ KN}$

ii) S.F. calculation

i) $F_B = -45\ KN$

ii) $F_{CR} = -45 + 20 \times 2 = -5\ KN.$

iii) $F_{CL} = -45 + 20 \times 2 + 10 = 5\ KN$

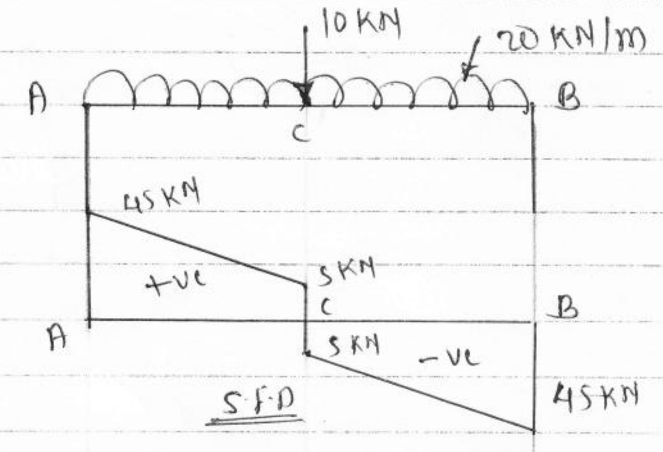
iv) $F_A = -45 + 20 \times 4 + 10 = 45\ KN$ 1M

iii) B.M. calculation

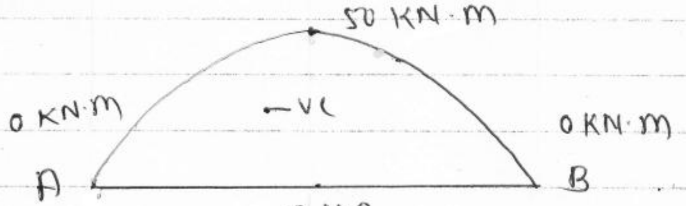
i) $M_B = 0\ KN.m$

ii) $M_C = 45 \times 2 - 20 \times 2 \times 1 = 90 - 40 = 50\ KN.m$

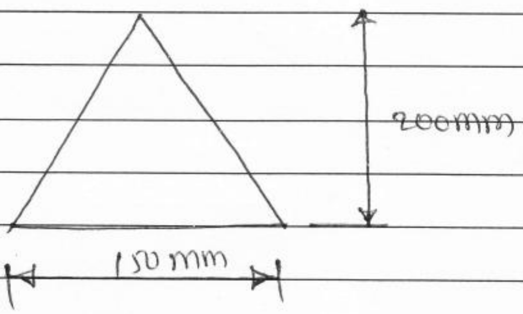
iii) $M_A = 45 \times 4 - 20 \times 4 \times 2 - 10 \times 2 = 0\ KN.m$ 1M



S.F.D 1M

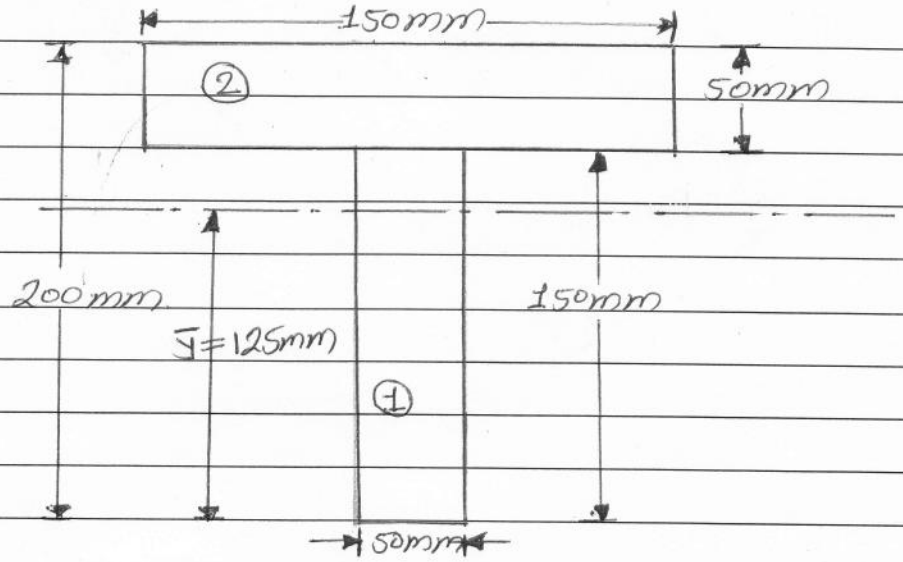


B.M.D 1M

f)	calculate the M.I of triangle having base 150mm & height 200mm about its base and centroidal axis parallel to base	
		
	i) M.I about base	
	$M.I_{base} = \frac{bh^3}{12} = \frac{150 \times 200^3}{12} = 100 \times 10^6 mm^4$	02 M
	ii) M.I about centroidal axis parallel to base	
	$M.I = \frac{bh^3}{36}$	
	$M.I = \frac{150 \times 200^3}{36} = 33.33 \times 10^6 mm^4$	02 M

Q4a)

→



1) Position of Centroidal axis

$a_1 = 150 \times 50 = 7500\,mm^2$	$a_2 = 150 \times 50 = 7500\,mm^2$
$x_1 = 50 + 50/2 = 75\,mm$	$x_2 = 150/2 = 75\,mm$
$y_1 = 150/2 = 75\,mm$	$y_2 = 150 + 50/2 = 175\,mm$

$$\therefore \bar{y} = \frac{a_1 y_1 + a_2 y_2}{a_1 + a_2} = \frac{(7500 \times 75) + (7500 \times 175)}{7500 + 7500}$$

$\bar{y} = 125\,mm$ from bottom. 1M

2) Moment of Inertia about x-x axis

$I_{xx} = I_{xx_1} + I_{xx_2}$

$I_{xx_1} = I_{G_1} + A_1 h_1^2 = \frac{bd^3}{12} + a_1(\bar{y} - y_1)^2$ ½M

$I_{xx_1} = \frac{50 \times 150^3}{12} + 7500(125 - 75)^2 = 32.81 \times 10^6\,mm^4$ ½M

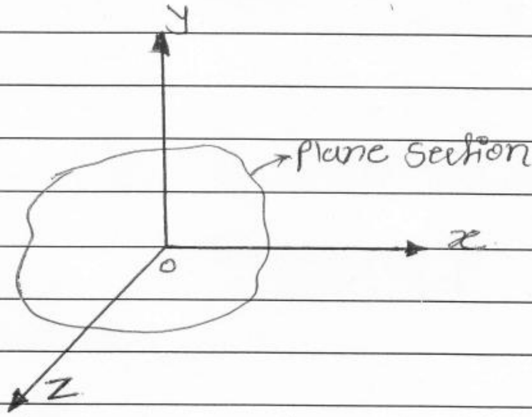
$I_{xx_2} = I_{G_2} + A_2 h_2^2 = \frac{bd^3}{12} + a_2(\bar{y} - y_2)^2$ ½M

$I_{xx_2} = \frac{150 \times 50^3}{12} + 7500(125 - 175)^2 = 20.31 \times 10^6\,mm^4$ ½M

$\therefore I_{xx} = 32.81 \times 10^6 + 20.31 \times 10^6 = 53.12 \times 10^6\,mm^4$ 1M

Q4b

→ i) Perpendicular axis theorem:-

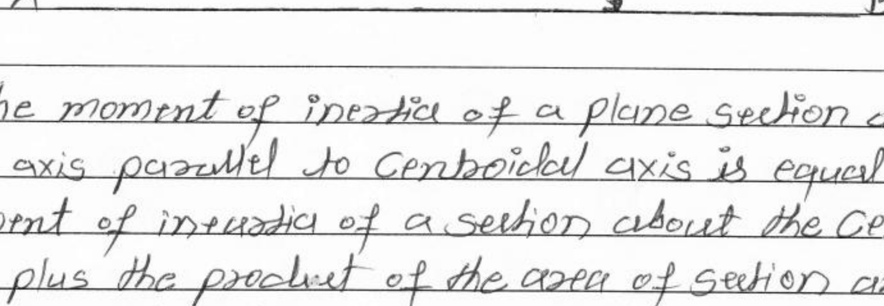


½ m

If I_{xx} & I_{yy} are the moment of Inertia of a plane section about the two mutually perpendicular axes meeting at 'o', then the moment of inertia I_{zz} about the third axis Z-Z perpendicular to the plane and passing through the intersection of x-x and Y-Y is given by, — 1 m

$$\boxed{I_{zz} = I_{xx} + I_{yy}}$$ ½ m

ii) Parallel axis theorem:-

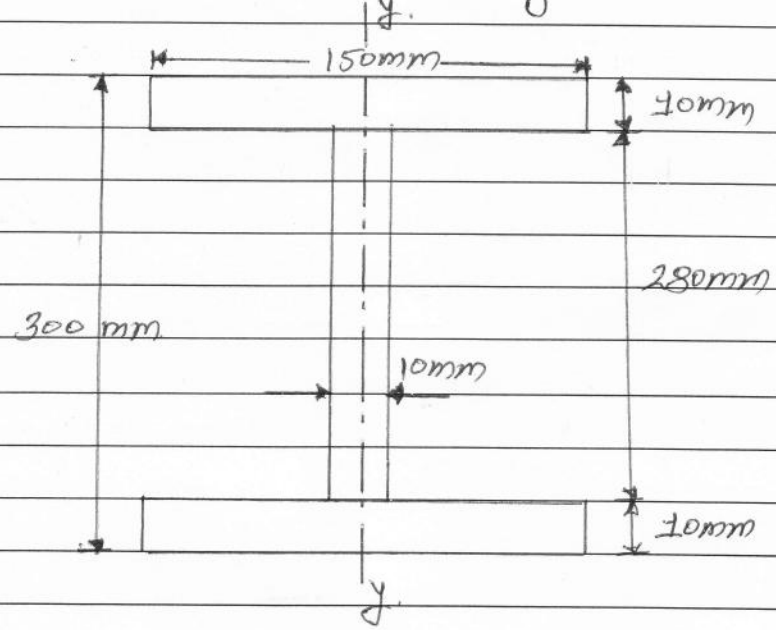


½ m

The moment of inertia of a plane section about any axis parallel to centroidal axis is equal to the moment of inertia of a section about the centroidal axis plus the product of the area of section and the square of the distance between the two axes. — 1 m

$$\boxed{I_{AB} = I_G + Ah^2}$$ ½ m

Q4c) →



1) M.I about axis perpendicular to flanges (Y-Y axis)

Flanges and web are symmetrical about Y-Y axis, hence no need to apply parallel axis theorem.

$I_{yy} = 2 \times M.I \text{ of flange} + M.I \text{ of web}$ — 1m

$= 2 \times \left[\frac{db^3}{12}\right]_f + \left[\frac{db^3}{12}\right]_w$ — 1m

$= 2 \times \frac{10 \times 150^3}{12} + \frac{280 \times 10^3}{12}$

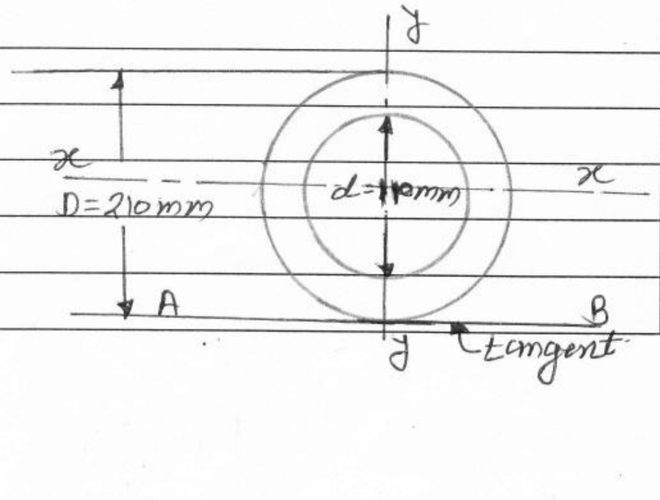
$= 5.625 \times 10^6 + 23.33 \times 10^3$ — 1m

$I_{yy} = 5.648 \times 10^6 \ mm^4$ — 1m.

Q4d) →

$D = 210mm$

$d = 110mm$



Q4d cont...	1) M.I. of hollow circular section	
	$I_{xx} = I_{yy} = \frac{\pi}{64}(D^4 - d^4)$	½ m
	$= \frac{\pi}{64}(210^4 - 110^4)$	
	$I_{xx} = I_{yy} = 88.278 \times 10^6 \ mm^4$	½ m
	2) Polar moment of Inertia (I_{zz})	
	$I_P = I_{zz} = I_{xx} + I_{yy}$	½ m
	$= 88.278 \times 10^6 + 88.278 \times 10^6$	
	$I_P = I_{zz} = 176.556 \times 10^6 \ mm^4$	½ m
	3) Moment of Inertia about tangent AB	
	By parallel axis theorem.	
	$I_{AB} = I_G + Ah^2$	½ m
	$A = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(210^2 - 110^2) = 25.13 \times 10^3 \ mm^2$	½ m
	$h = \frac{D}{2} = \frac{210}{2} = 105 \ mm$	
	$\therefore I_{AB} = 88.278 \times 10^6 + 25.13 \times 10^3 (105)^2$	
	$I_{AB} = 365.33 \times 10^6 \ mm^4$	1 m
Q4e)		
→	Assumptions of theory of simple bending.	
	1) Transverse section of beam which is plane before bending will remain plane after the bending.	
	2) The material of the beam is homogeneous & isotropic	1 m each for any Four
	3) The value of youngs modulus (E) is same in tension and compression.	
	4) the elastic limit is not exceeded.	
	5) The beam is initially straight and unstressed.	
	6) The deformation of the section due to shear force is neglected.	

7) The beam is stressed well up to proportional limit such that it must obeys Hooks law.

8) Each longitudinal fiber is free to expand or Contract independently from every othe layer

9) The resultant force across transverse section of the beam is zero.

Q4 f)

→ $b = 100 mm$

$S = 50 KN$

$\tau_{max} = 4 N/mm^2$ or $q_{max} = 4 N/mm^2$

i) $q_{max} = 1.5 \cdot q_{avg}$ —(i) 1m

$q_{avg} = \frac{\text{Shear force}}{\text{c/s area}} = \frac{50 \times 10^3}{100 \times d}$ 1m

$q_{avg} = \frac{500}{d}$ ..... substitute in eqn (i)

$q_{max} = 1.5 \times \frac{500}{d}$

$4 = 1.5 \times \frac{500}{d}$ 1m

$\therefore d = \frac{1.5 \times 500}{4}$

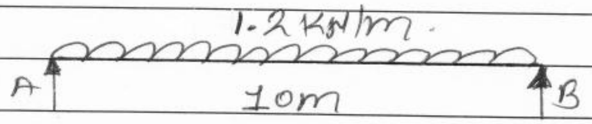
$d = 187.5$ mm. 1m

Q5 a)

→ $D = 120mm$

$L = 10m$

$w = 1200 \, N/m = 1.2 \, KN/m.$



1) Maximum Bending moment

$$\text{Max. B.M} = \frac{wl^2}{8} = \frac{1.2 \times 10^2}{8} = 15 \, KN.m.$$ 1m

2) Using flexural formula.

$$\frac{M}{I} = \frac{\sigma_b}{y}$$ ½m

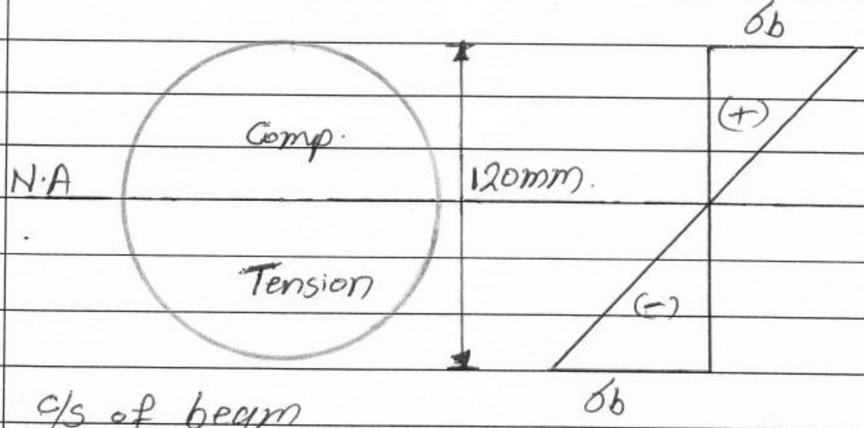
$M = 15 \times 10^6 \, N.mm$

$$I = \frac{\pi}{64} D^4 = \frac{\pi}{64}(120)^4 = 10.178 \times 10^6 \, mm^4$$ ½m

$y = D/2 = 120/2 = 60mm.$

$$\frac{15 \times 10^6}{10.178 \times 10^6} = \frac{\sigma_b}{60}$$

$\therefore \sigma_b = 88.42 \, N/mm^2$ 1m



1m

c/s of beam

Bending stress distribution diagram.

Q5b)

→ $b = d = 210 mm$ – square column.
$P = 105 KN$
$e = 65 mm$.

1) Geometrical properties of section.
i) c/s area $A = b \times d = 210 \times 210 = 44100 mm^2$ — 1M

ii) Section modulus $Z = \frac{bd^2}{6} = \frac{210 \times 210^2}{6} = 1.5435 \times 10^6 mm^3$

2) Direct stress $\sigma_0 = \frac{P}{A} = \frac{105 \times 10^3}{44100}$

$\sigma_0 = 2.38 N/mm^2$ — 1M

3) Bending stress $\sigma_b = \frac{M}{Z} = \frac{P \times e}{Z} = \frac{105 \times 10^3 \times 65}{1.5435 \times 10^6}$

$\sigma_b = 4.42 N/mm^2$ — 1M

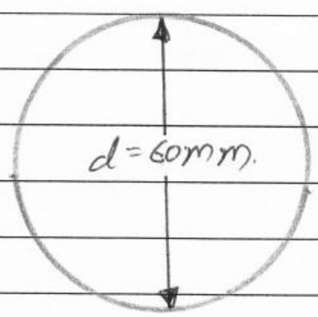
4) Resultant stresses.

$\sigma_{max} = \sigma_0 + \sigma_b = 2.38 + 4.42 = 6.80 N/mm^2$

$\sigma_{min} = \sigma_0 - \sigma_b = 2.38 - 4.42 = -2.04 N/mm^2$ … (Tensile) — 1M

Q5c)

→



1) Geometrical properties
i) c/s area $A = \frac{\pi}{4} D^2 = \frac{\pi}{4} \times 60^2$

$A = 2827.43 mm^2$

ii) Section modulus $Z = \frac{\pi}{32} D^3$

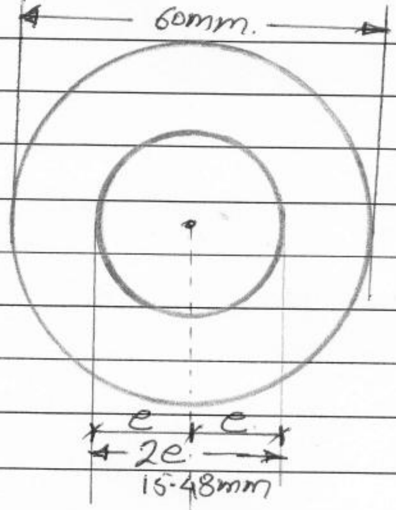
$Z = \frac{\pi}{32} \times 60^3 = 21.889 \times 10^3 mm^3$ — 1M

2) limit of eccentricity $e \le \frac{Z}{A}$ 1m

$e \le \frac{21.889 \times 10^3}{2827.43}$

$e = 7.74\ mm.$ ½m

$\therefore\ 2e = 15.48\ mm.$ ½m



1m

Q5d
→ $D = 250mm$
$d = 200mm$
$P = 20KN$
$e = 400mm$

1) Geometrical properties of section

i) c/s area $A = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4} \times (250^2 - 200^2)$

$A = 17.67 \times 10^3\ mm^2$

ii) section modulus $Z = \frac{\pi}{32}\left(\frac{D^4 - d^4}{D}\right)$

$Z = \frac{\pi}{32}\left(\frac{250^4 - 200^4}{250}\right)$ 1m

$Z = 905.66 \times 10^3\ mm^3$

2) Direct stress $\sigma_0 = \frac{P}{A} = \frac{20\times10^3}{17.67\times10^3}$

$\sigma_b = 1.13 \ N/mm^2$ — 1M

3) Bending stress $\sigma_b = \frac{M}{Z} = \frac{P\times e}{Z}$

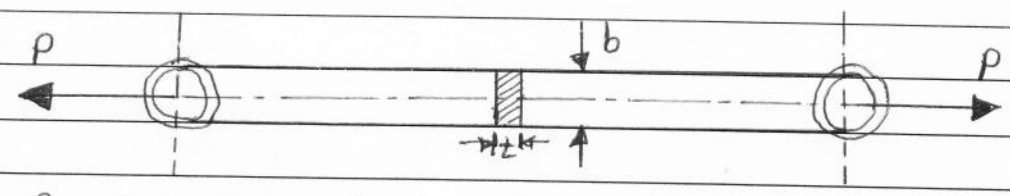
$\sigma_b = \frac{20\times10^3\times400}{905.66\times10^3} = 8.83 \ N/mm^2$ — 1M

4) Resultant stresses

$\sigma max = \sigma_0 + \sigma_b = 1.13 + 8.83 = 9.96 \ N/mm^2$ — ½M

$\sigma min = \sigma_0 - \sigma_b = 1.13 - 8.83 = -7.70 \ N/mm^2$ ... (Tensile) — ½M

Q5c) →



$P = 90 KN$ $\quad \sigma = 65 \ MPa$

$b = 3t$

i) Tensile stress σ

$\sigma = \frac{P}{A}$ — 1m

$\sigma = \frac{P}{b\times t}$

$65 = \frac{90\times10^3}{3t\times t}$ .... $b = 3t$ — 1m

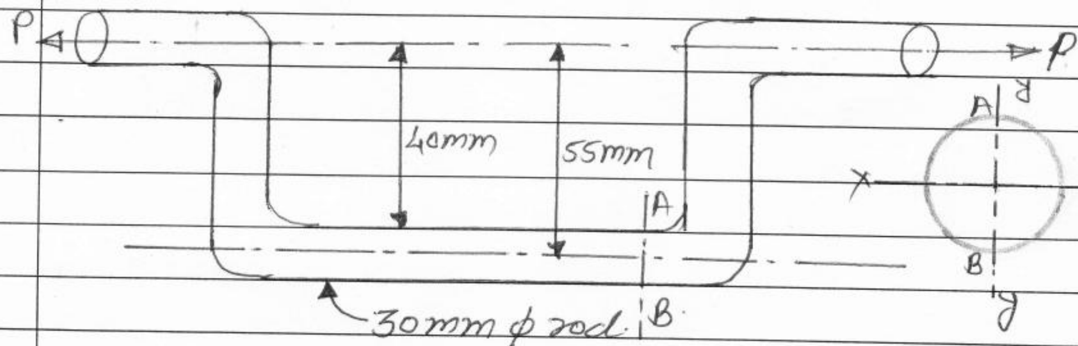
$3t^2 = \frac{90\times10^3}{65}$

$3t^2 = 1384.61$

$\therefore t = 21.48 \ mm$ — 1m

$\therefore b = 64.44 \ mm$ — 1m

Q5 f)		
→	D = 30mm Let P be the axial force in N.	
	$\sigma_{max} = 80\ N/mm^2$ $e = 40 + \frac{30}{2} = 55mm$.	



1) Geometrical Properties of section.

i) C/s area $A = \frac{\pi}{4} D^2 = \frac{\pi}{4} \times (30)^2 = 706.85\ mm^2$

ii) Section modulus $Z = \frac{\pi}{32} D^3 = \frac{\pi}{32} \times (30)^3 = 2650.71\ mm^3$ } 1m

2) Direct stress $\sigma_0 = \frac{P}{A} = \frac{P}{706.85} = 1.41 \times 10^{-3} P$ ½ m

3) Bending stress $\sigma_b = \frac{M}{Z} = \frac{P \times e}{Z} = \frac{P \times 55}{2650.71} = 20.74 \times 10^{-3} P$ ½ m

4) To find maximum value of P.

$\sigma_{max} = \sigma_0 + \sigma_b$ 1m

$80 = (1.41 \times 10^{-3} + 20.74 \times 10^{-3}) P$

$\therefore P = 3609.47\ N$

OR 1m

$P = 3.609\ KN$

Q6a)

→ Assumptions in theory of pure torsion.

1) Plane sections before twisting remain plane after twisting and do not twist or warp.

2) The shaft material is homogenious and isotropic.

3) The shaft is straight having uniform circular cross-section.

4) Circular sections remain circular even after twisting.

5) Twist along the shaft is uniform.

6) Stresses do not exceed the proportional limit.

7) Shaft is loaded by twisting couples in the planes that are perpendicular to the axis of the shaft.

1m each for Any Four

Q6b)

→ $D = 200\,mm$

$\theta = 2^\circ = 2 \times \pi/180 = 0.0349$ rad.

$L = 8m$.

$C = 0.8 \times 10^5$ MPa.

1) Using Torsional Formula.

$$\frac{T}{I_p} = \frac{C\theta}{L}$$ 1m

$I_p = \frac{\pi}{32} D^4 = \frac{\pi}{32} \times 200^4 = 157.07 \times 10^6\,mm^4$. 1m

$$\frac{T}{157.07 \times 10^6} = \frac{0.8 \times 10^5 \times 0.0349}{8000}$$ 1m

	$\therefore T = 54.82 \times 10^6$ N·mm	
	OR	1m
	$T = 54.82 \times 10^3$ N·m.	
Q6c)		
→	$D = 120$ mm	
	$d = 90$ mm	
	$T = 20$ KN·m	
	i) Using Torsional formula.	
	$\frac{T}{I_p} = \frac{\tau}{R}$	1m
	$T = 20$ KN·m $= 20 \times 10^6$ N·mm.	
	$I_p = \frac{\pi}{32}(D^4 - d^4) = \frac{\pi}{32}(120^4 - 90^4) = 13.91 \times 10^6$ mm^4	1m
	$R = D/2 = \frac{120}{2} = 60$ mm.	1/2m
	$\therefore \tau = \frac{T}{I_p} \times R$	
	$= \frac{20 \times 10^6}{13.91 \times 10^6} \times 60.$	1/2m
	$\tau = 86.26$ N/mm^2	1m.
Q6d)	Let, d = Diameter of solid shaft.	
→	$d_2 = \frac{2}{3} d_1$ d_1 = External dia. of hollow shaft.	
	d_2 = Internal dia. of hollow shaft.	
	For same strength Polar modulus Z_p of both the shaft must be equal.	
	Z_p of solid shaft = Z_p of hollow shaft.	1/2m

Q6 d) Contd.

$$\frac{\pi}{16} d^3 = \frac{\pi}{16}\left(\frac{d_1^4 - d_2^4}{d_1}\right)$$

$$d^3 = \frac{d_1^4 - \left(\frac{2}{3}d_1\right)^4}{d_1}$$

$$d^3 = \frac{d_1^4}{d_1}\left[1 - \left(\frac{2}{3}\right)^4\right]$$

$$d^3 = 0.802 d_1^3$$

$$\therefore \boxed{d = 0.929\, d_1}$$ 1m

2) % saving in material

$$\%\ \text{saving} = \frac{\text{Area of solid shaft} - \text{Area of hollow shaft}}{\text{Area of solid shaft}} \times 100$$ ½m

$$= \left[\frac{\frac{\pi}{4}d^2 - \frac{\pi}{4}(d_1^2 - d_2^2)}{\frac{\pi}{4}d^2}\right] \times 100$$

$$= \left[\frac{d^2 - (d_1^2 - d_2^2)}{d^2}\right] \times 100$$ 1m

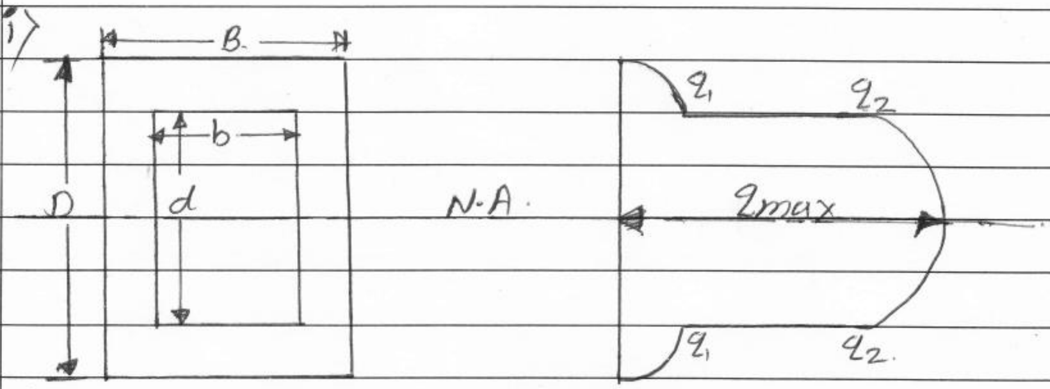
$$= \left[1 - \left[\frac{d_1^2 - \left(\frac{2}{3}d_1\right)^2}{d^2}\right]\right] \times 100$$

$$= \left[1 - \frac{d_1^2}{d^2}\left[1 - \left(\frac{2}{3}\right)^2\right]\right] \times 100$$

$$= \left[1 - \frac{0.555 d_1^2}{(0.929)^2}\right] \times 100 \quad \cdots d = 0.802 d_1$$

% saving in material $= 35.70\,\%$ 1m

Q6 e)		
→	Let, D = dia of solid shaft = 80mm	
	d_1 = External dia. of hollow shaft	
	d_2 = Internal dia. of hollow shaft	
	$\therefore d_1 = 1.5 d_2$	
	In order that the solid and hollow shaft Should have same strength to resist the torque, the Polar modulus of the shaft must be equal.	
	Z_p of solid shaft = Z_p of hollow shaft	
	$\frac{\pi}{16} D^3 = \frac{\pi}{16}\left[\frac{d_1^4 - d_2^4}{d_1}\right]$	1m
	$D^3 = \frac{(1.5d_2)^4 - d_2^4}{1.5 d_2}$	
	$80^3 = \frac{d_2^4}{d_2}\left[\frac{(1.5)^4 - 1}{1.5}\right]$	
	$512 \times 10^3 = d_2^3 \, 2.708$	
	$d_2^3 = \frac{512 \times 10^3}{2.708}$	
	$d_2^3 = 189.06 \times 10^3$	1m
	$\therefore d_2 = (189.06 \times 10^3)^{1/3}$	
	$\therefore d_2 = 57.39$ mm	1m
	$\therefore d_1 = 1.5 d_2 = 1.5 \times 57.39 = 86.08$ mm.	1m
	$\therefore$ External dia $d_1 = 86.08$ mm	
	Internal dia $d_2 = 57.39$ mm.	

Q6 f)		
→	i) 	1m
	hollow rectangular section — shear stress distribution diagram.	
	$q_{max} = q_2 + q_{additional}$	1m
	where,	
	q_2 = Shear stress at bottom of flange by taking width as $B-b$	
	$q_{additional}$ = Additional shear stress due to web area above (or below) N.A.	
	ii) Definition:- When a tangential force is applied to a shaft at the circumference, in the plane of its transverse cross-section, the shaft is said to be subjected to a twisting moment or torque which is equal to the product of the force and the radius.	1m
	S.I. Unit of Torque,	
	Torque = Force × Radius	
	Torque N × m	
	∴ S.I. Unit of torque = N·m.	1m