SUMMER – 14 EXAMINATIONS

Subject Code: 17304 **Model Answer** **Total Pages: 35**

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.

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.

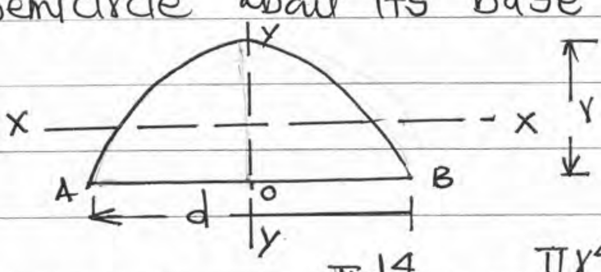
7) For programming language papers, credit may be given to any other programme based on equivalent concept.

Important Notes:

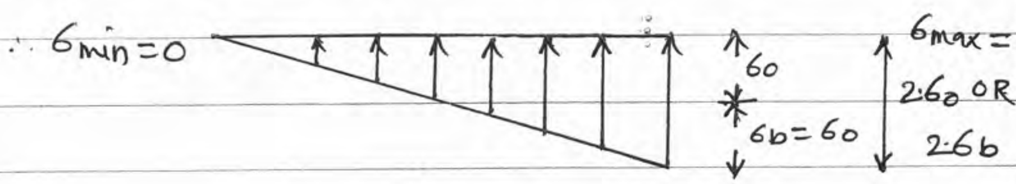
1. **In Q 3(d) in this problem point of contra flexure is not possible, therefore examiner should give proportionate marks.**
2. **In Q 4 (a) this problem is solved by using parallel axis theorem.**

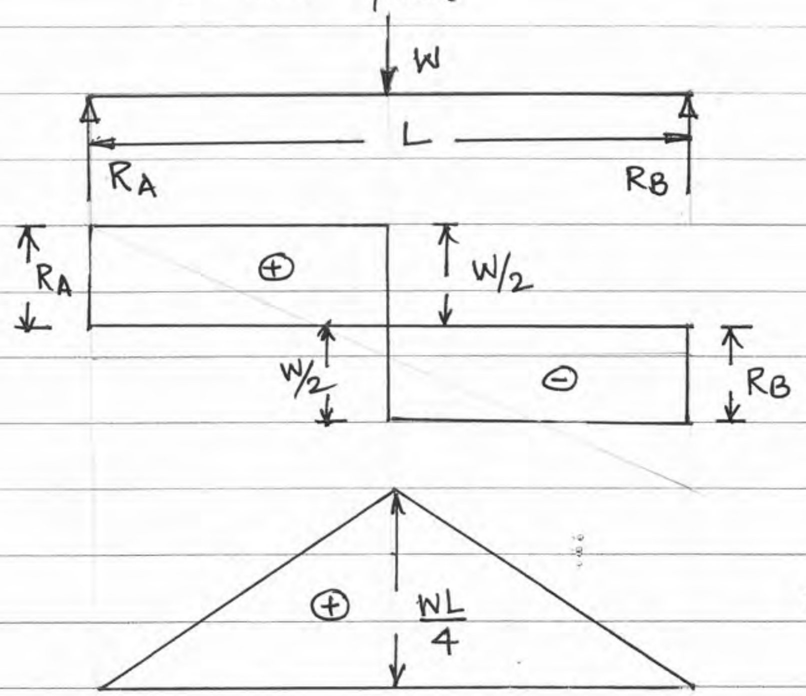
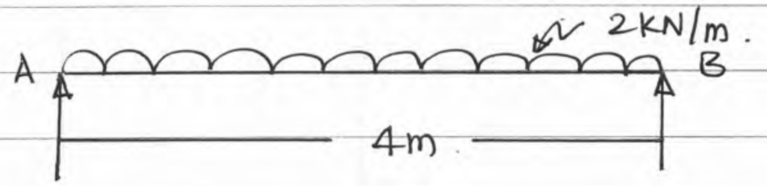
OR

Problem is solved by using standard formula, examiner should give proportionate marks.

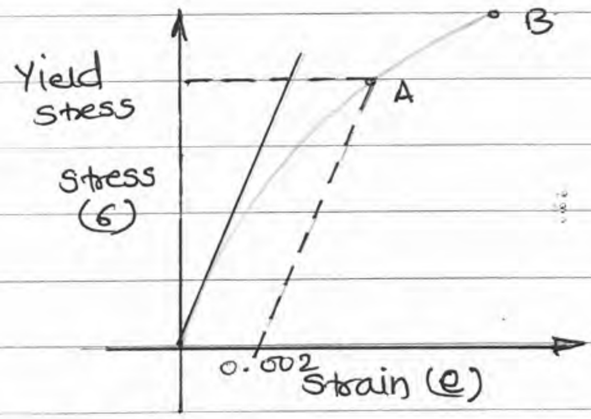
Q. NO	SOLUTION	MARKS
1 A) (A)	Poisson's Ratio :- when material is loaded within its elastic limit, the ratio of Lateral strain to the Longitudinal strain (linear strain) is constant and is called as poisson's Ratio.	1
	Relation between E and K.	
	$\therefore E = 3K(1-2\mu)$	1
(B)	Principal plane :- A plane which carry only Direct stress or normal stress and no shear stress on it, is called Principal plane	1
	Principal stress :- The magnitude of Direct stress or normal stress acting on Principal plane is called Principal stress.	1
(C)	M.I. of Semicircle about its base	
		
	$\therefore I_{base} = I_{AB} = \frac{\pi d^4}{128} = \frac{\pi r^4}{8}$	2
(D)	Direct load :- A load whose line of action coincides with the axis of the member is called an axial load or direct load.	

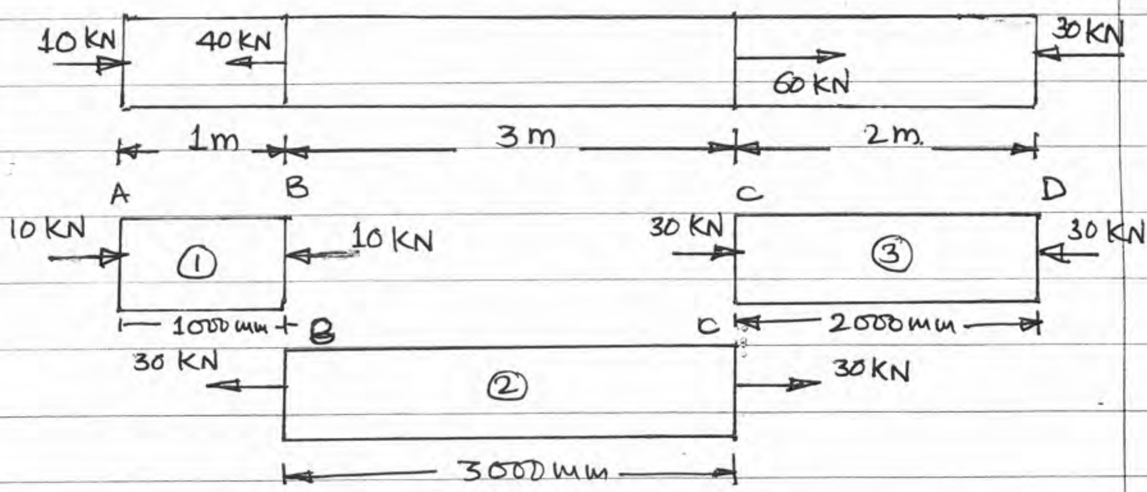
Q.NO	SOLUTION	MARKS
1 (A) (D) cont.	Eccentric Load :- A Load whose line of action does not coincides with the axis of a member is called eccentric load.	
(E)	Power transmitted by shaft. (P)	
	$\therefore P = \frac{2\pi N T}{60}$	1
	$\therefore$ P = Power in watts	
	T = Averge (mean) Torque in N-m	1
	N = Number of revolutions of shaft in rpm.	
(F)	Ductility :- It is the property of a material to undergo large deformation under tension before failure or rupture.	1
	OR.	OR
	It is the ability of material which enables it to draw into thin wires under tension without failure or rupture.	1
	Malleability :- It is the property of a material by virtue of which it gets permananently deformed by compression without failure or rupture.	1
	OR.	OR
	It is the ability of material which enables it to Roll into thin sheets under compression without failure or rupture	1

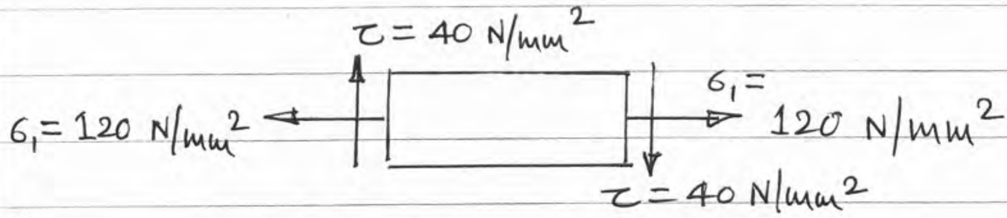
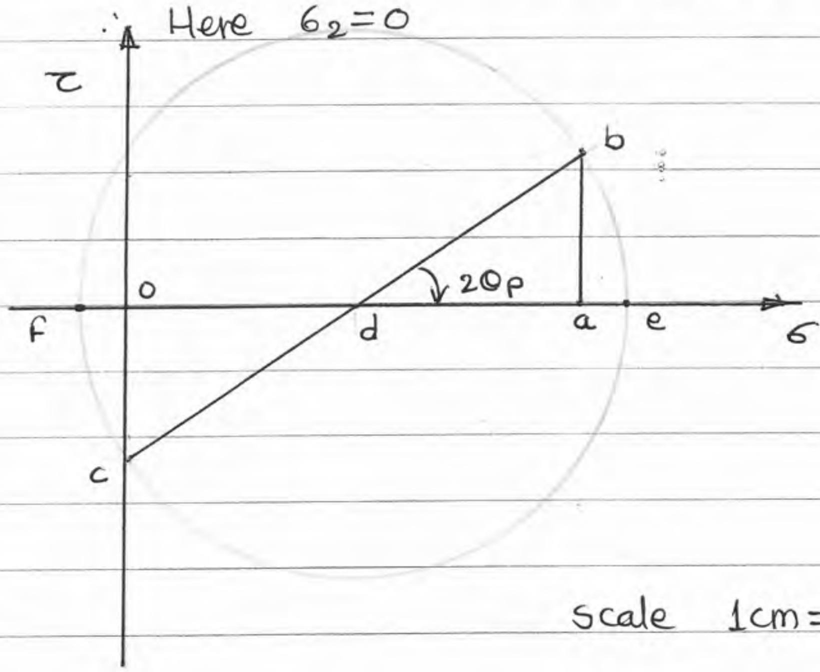
Q.NO	SOLUTION	MARKS
1 (A)(G)	Hoop stress :- The stress which act in the tangential direction to the perimeter (circumference) of the cylinder is called as Hoop stress or Circumferential stress.	1
	$\sigma_c = \frac{pd}{2t}$	1
	p = internal pressure of fluid d = internal dia. of thin cylinder t = thickness of thin cylinder	
(H)	Resultant stress distribution at the base when $\therefore \sigma_0 = \sigma_b$	
		1 M for sketch + 1 M for labels.
B) (a)	Given data. $\therefore l = 2m = 2000\,mm$ $\therefore t_1 = 10^\circ C$ $\therefore t_2 = 80^\circ C$ $\therefore E = 1 \times 10^5\ N/mm^2$ $\therefore \alpha = 12 \times 10^{-6}/^\circ C$	
	Expansion of the rod (δl) $\therefore \delta l = l \alpha t$ -----------------	1 M
	$\therefore \delta l = 2000 \times 12 \times 10^{-6} \times (80 - 10)$	
	$\therefore \delta l = 1.68\ mm.$ -----------------	1 M

Q.NO	SOLUTION	MARKS
1 B) (a) cont--	stress $(\sigma) = \alpha t E$	1
	$= 12 \times 10^{-6} \times (80-10) \times 1 \times 10^{5}$	
	$= 84\ N/mm^2$	1
b)		
	SFD	2
	BMD	2
c)		
	Given $\therefore \sigma_b = 165\ N/mm^2$	
	soln	
	① maximum Bending moment :	
	$M = \frac{\omega l^2}{8} = \frac{2 \times 4^2}{8} = 4\ KN\text{-}m = 4 \times 10^{6}\ N\text{-}mm$	1
	② section modulus (Z) :	

Q.NO	SOLUTION	MARKS
1 B) C) cont--	$M = \sigma \cdot Z$	$\frac{1}{2}$
	$\therefore Z = \frac{M}{\sigma}$	
	$\therefore Z = \frac{4 \times 10^6}{185}$	
	$\therefore Z = 21621.621\ mm^3$	$\frac{1}{2}$
	③ Diameter of the circular beam (d)	
	$\therefore Z = \frac{\pi}{32} d^3$	1
	$\therefore 21621.621 = \frac{\pi}{32} \cdot d^3$	
	$\therefore d = 60.389\ mm$	1
Q2 a)	i) Principal of Superposition :- "If number of forces act on a body, the total effect of all the forces is the summation of the effects of the individual forces."	2 M
	OR	
	"If body is subjected to different direct forces at different sections along the length of the body, then the total deformation of the body will be equal to the Algebraic sum of deformations of the individual sections."	2M
	OR	
	"when number of forces or loads are acting	

Q.NO	SOLUTION	MARKS
Q 2(a)	on a body, the resulting strain will be the algebraic sum of strains caused by individual forces or Loads."	2M
(ii)	effective length when both ends are hinged. $\therefore L_{eff} = l$ effective length when both ends are fixed $\therefore L_{eff} = \frac{l}{2}$	
(b)		2M
	(1) Brittle material do not show clear yield point (2) for finding yield stress a line is drawn parallel to tangent from 0.2% strain (i.e 0.002) to get 'A' pt. (3) Point A is yield stress and Point B is Breaking stress	2M
(c)	Assumptions made in Euler's theory. (i) The column is initially straight and is axially loaded. (ii) The section of the column is uniform. (iii) The column material is perfectly elastic, homogeneous and isotropic and obeys Hooke's law.	

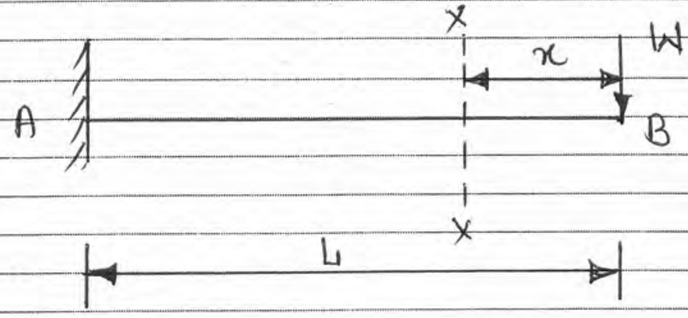
Q.NO	SOLUTION	MARKS
Q 2 (C) cont.---	(iv) The column is long and will fail due to buckling alone	4 (for any four)
	(v) shortening of the column due to direct compression is negligible.	
	(vi) The self weight of the column is negligible.	
(d)		1 M
	FBD	
	Total change in length (δl)	
	$\therefore \delta l = \delta l_{AB} + \delta l_{BC} + \delta l_{CD}$	
	$\therefore \delta l = \frac{P_1 l_1}{AE} + \frac{P_2 l_2}{AE} + \frac{P_3 l_3}{AE}$	
	$\therefore \delta l = \frac{1}{AE}\left(P_1 l_1 + P_2 l_2 + P_3 l_3\right)$	1 M
	$\therefore$ Here $P_1 = 10\,KN = 10000\,N$ (compressive).	
	$P_2 = 30000\,N$ (Tensile)	
	$P_3 = 30000\,N$ (compressive)	

Q.NO	SOLUTION	MARKS
Q 2 (d)	considering Tensile force as +ve comp. force as -ve $\therefore \delta l = \frac{1}{1000 \times 1.05 \times 10^5}\left[-10000 \times 1000 + 30000 \times 3000 - 30000 \times 2000\right]$ $\therefore \delta l = 0.1905$ mm.	2 M
(e)	 $\therefore$ Here $\sigma_2 = 0$  scale 1cm = 20 N/mm^2	2 M
	Principal stresses $\therefore$ Major principal stress $\sigma_{n_1} = l(oe) \times$ scale $= 6.6 \times 20 = 132$ N/mm^2 (Tensile)	$\frac{1}{2}$ M
	$\therefore$ Minor principal stress $\sigma_{n_2} = -\ l(of) \times$ scale $= -0.6 \times 20 = -12$ N/mm^2 (comp.)	$\frac{1}{2}$ M

Q.NO	SOLUTION	MARKS
Q 2(e) cont.---	position of Principal plane	
	Position of Major Principal plane (θ_1)	
	$\therefore \theta_1 = \angle bda = \frac{2\theta_P}{2} = \frac{33}{2}$	
	$\therefore \theta_1 = 16.5^\circ$	$\frac{1}{2}$M
	Position of Minor Principal plane (θ_2)	
	$\therefore \theta_2 = \theta_1 + 90$	
	$\therefore \theta_2 = 16.5 + 90$	
	$\therefore \theta_2 = 106.5^\circ$	$\frac{1}{2}$M.
(f)	Given Data. Int. fluid pr. (p) = $3\ N/mm^2$; Int. diameter (d) = 500 mm.; permissible tensile stress = $80\ N/mm^2$.	
	$\therefore$ Taking σ_c = permissible tensile stress = $80\ N/mm^2$.	1 M
	circumferential stress (σ_c)	
	$\therefore \sigma_c = \frac{pd}{2t}$	1 M
	$\therefore t = \frac{pd}{2 \times \sigma_c}$	
	$\therefore t = \frac{3 \times 500}{2 \times 80}$	
	$\therefore t = 9.375$ mm	2 M

Q .NO	SOLUTION	MARKS

Q-3 (a)



Consider a cantilever of span 'L' with a point load at it's free end and also consider a section x-x

S.F. calculation :

i) F_B = shear force at B = +W

ii) F_x = shear force at x = +W — ½

iii) F_A = shear force at A = +W

$\boxed{F_{max} = +W}$ max. S.F — ½

B.M. calculation :

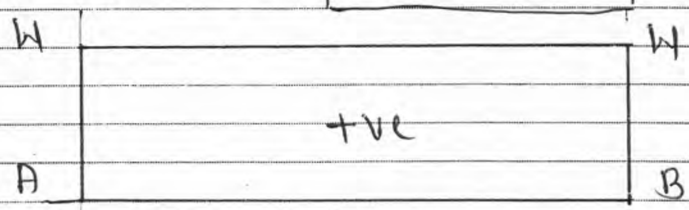
i) M_B = Bending moment at B = 0

ii) M_x = Bending moment at any section x-x — ½

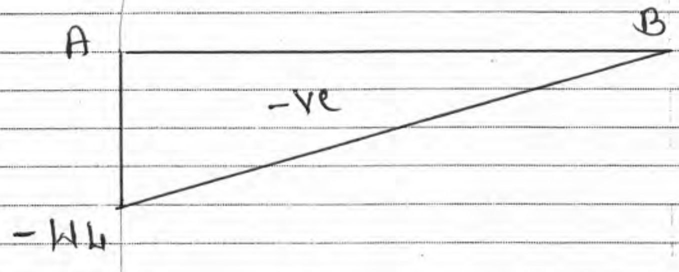
$M_x = -W \cdot x$

iii) M_A = Bending moment 'A' = $-W \cdot L$

$\therefore \boxed{M_{max} = -W \cdot L}$ max. B.M — ½



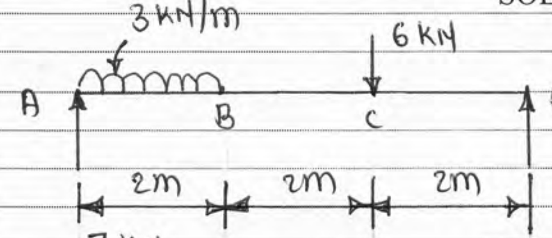
S.F.D — 1



B.M.D — 1

Q .NO	SOLUTION	MARKS

Q-3 (b)



i) Support Reaction

$\Sigma f_y = 0$

$R_A - 3\times2 - 6 + R_D = 0$

$\therefore R_A - 6 - 6 + R_D = 0$

$R_A + R_D = 12 \quad —\ i]$

$\Sigma M_A = 0$

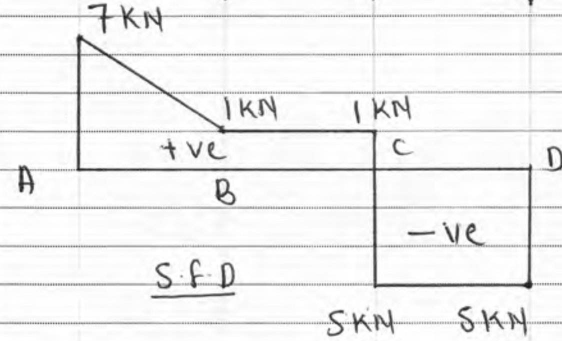
$-R_D \times 6 + 6\times4 + 3\times2\times\frac{2}{2} = 0$

$-6R_D + 24 + 6 = 0$

$-6R_D + 30 = 0$

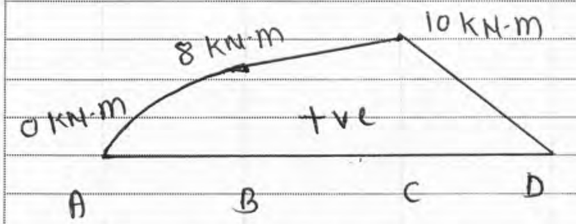
$R_D = \frac{30}{6}$

$\boxed{R_D = 5\ \text{KN}}$



S.F.D

for S.F.D = 1M



B.M.D

for B.M.D = 1M

Put R_D in eq^n i]

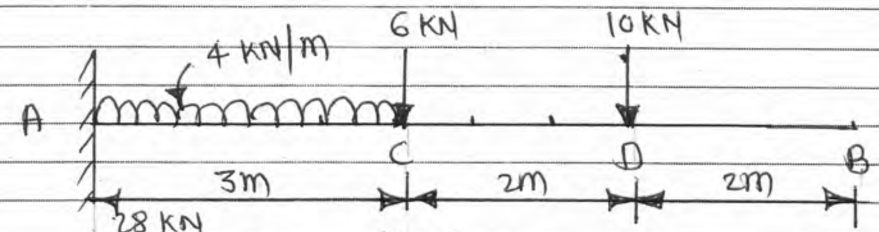
$R_A + R_D = 12 \quad \therefore \boxed{R_A = 7\ \text{KN}}$

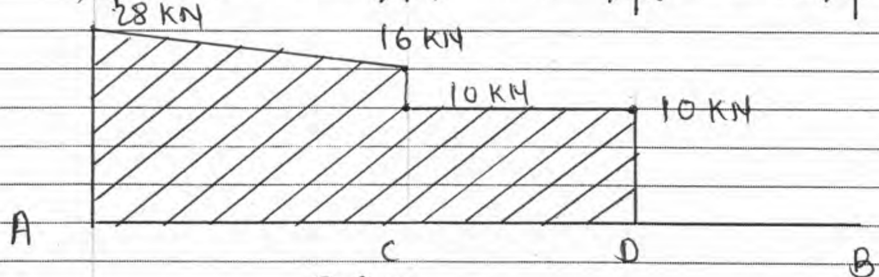
S.F.D. calculation	B.M. calculation.
i) $F_A = 7$ KN	i) $M_A = 0$ KN-m
ii) $F_B = 7 - 3\times2 = 1$ KN	ii) $M_B = 7\times2 - 3\times2\times\frac{2}{2} = 8$ KN-m
iii) $F_{CL} = 1$ KN	iii) $M_C = 7\times4 - 3\times2\left(\frac{2}{2}+2\right)$
iv) $F_{CR} = 1 - 6 = -5$ KN	$= 28 - 3\times2\times3$
v) $F_D = -5$ KN	$M_C = 10$ KN-m
	iv) $M_D = 0$ KN-m.

for S.F.D calculation = 1M

for B.M.D calculation = 1M

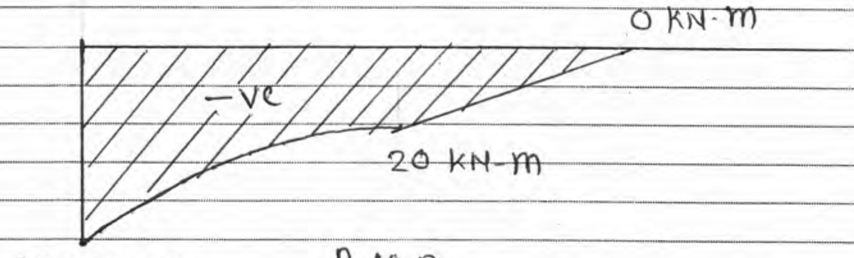
Q .NO	SOLUTION	MARKS
Q-3 (c)		





S.F.D

1



B.M.D

1

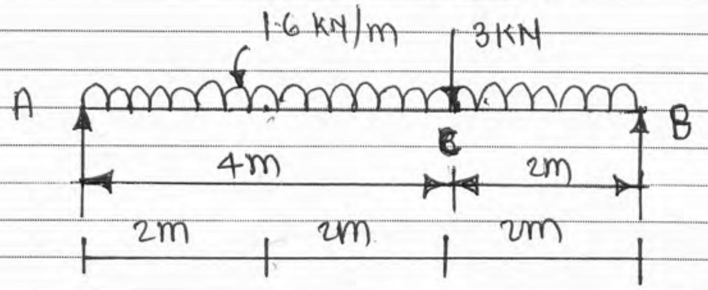
S.F. calculation	B.M. calculation
i) $F_B = 0$ KN	i) $M_B = 0$ KN-m
ii) $F_{DR} = 0$ KN	ii) $M_D = 0$ KN-m
iii) $F_{DL} = 10$ KN	iii) $M_C = -10 \times 2 = -20$ KN-m
iv) $F_{CR} = 10$ KN	iv) $M_A = -10 \times 5 - 6 \times 3 - 4 \times 3 \times 3/2$
v) $F_{CL} = 10 + 6 = 16$ KN	
vi) $F_A = 16 + 4 \times 3$	$M_A = -50 - 18 - 18$
$= 16 + 12$	$M_A = -86$ KN-m
$F_A = 28$ KN	

S.F. Calculation – 1M

B.M. calculation 1M

Q .NO	SOLUTION	MARKS

Q-3 (d)



i) find reactions R_A & R_B

$\Sigma f_y = 0$

$R_A + R_B - 3 - 1.6 \times 6 = 0$

$R_A + R_B = 12.6$ —— i)

$\Sigma M_A = 0$

$-R_B \times 6 + 3 \times 4 + 1.6 \times 6 \times 6/2 = 0$

$-6R_B + 12 + 28.8 = 0$

$-6R_B + 40.8 = 0$

$-6R_B = -40.8$

$\boxed{R_B = 6.8 \text{ kN}}$

Put R_B value in eq^n i)

$\therefore R_A + R_B = 12.6$

$R_A = 12.6 - R_B$

$R_A = 12.6 - 6.8$

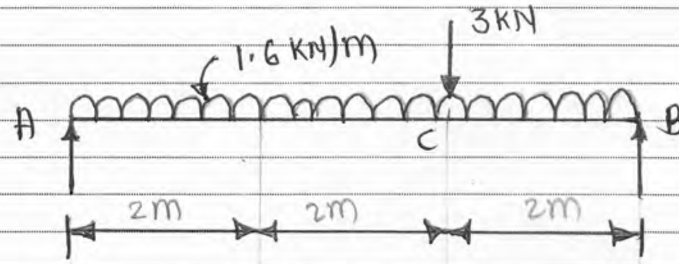
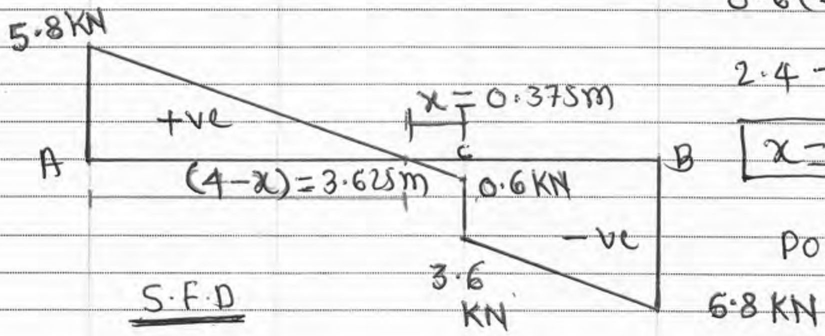
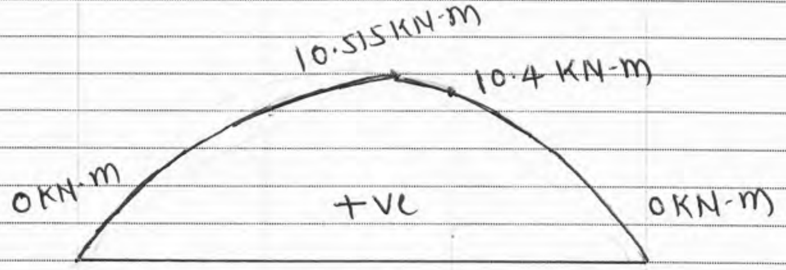
$\boxed{R_A = 5.8 \text{ kN}}$

ii) S.F calculation

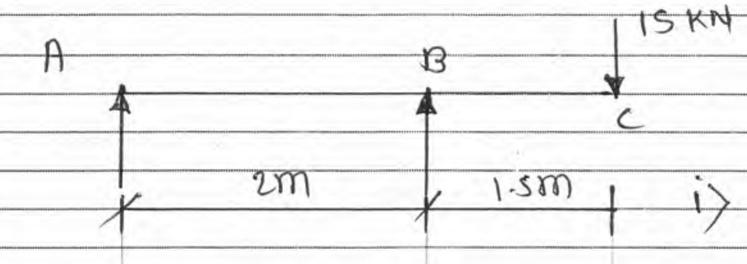
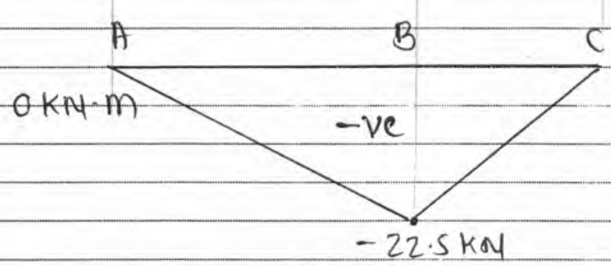
i) $F_B = -6.8$ kN

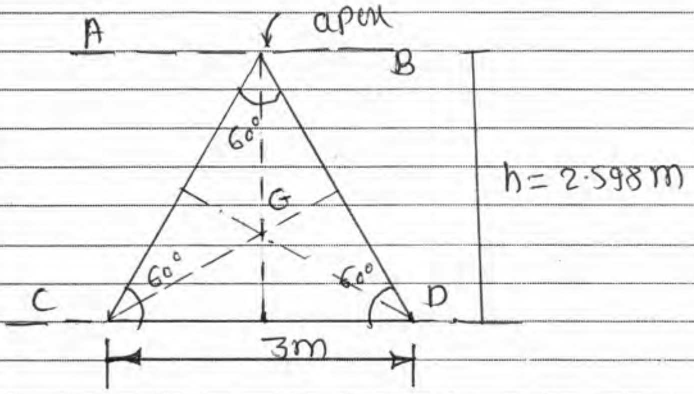
ii) $F_{CR} = -6.8 + (1.6 \times 2)$

$F_{CR} = -3.6$ kN

Q .NO	SOLUTION	MARKS
	$F_{CL} = -3.6 + 3 = -0.6$ KN	
	$F_A = -0.6 + (1.6 \times 4) = 5.8$ KN	01
	iii) B.M. calculation	
	i) $M_B = 0$ KN-m	
	ii) $M_C = 6.8 \times 2 - 1.6 \times 2 \times 2/2$	
	$M_C = 10.4$ KN·m	1/2
	iii) $M_A = 0$ KN-m	
	iv) $M_{CS} = 5.8 \times 3.625 - 1.6 \times 3.625 \times \frac{3.625}{2} = 10.515$ KN-m	1/2
	 Location of point of zero shear $\frac{0.6}{x} = \frac{5.8}{4-x}$ $0.6(4-x) = 5.8x$ $2.4 - 0.6x = 5.8x$ $\boxed{x = 0.375 \text{ m}}$ from Point 'c'	
	 S.F.D	01
	 B.M.D	01

Note: In Q-3(d) in this problem a point of contraflexure is not possiable, therefore examiner should give proportionate marks

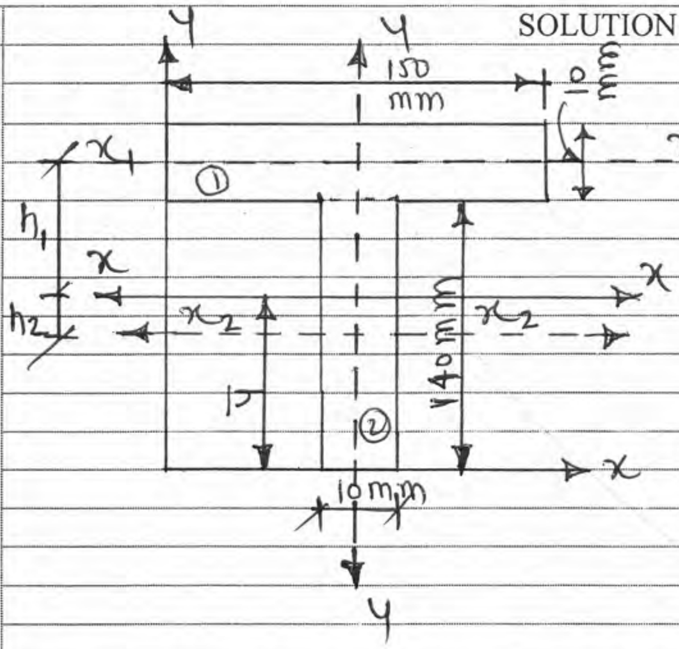
Q .NO	SOLUTION	MARKS
Q-3(e)	B.M.D	for B.M.D = 1M
	i) To find reaction $\Sigma fy = 0$ $R_A + R_B = 15$ ---- i]	
	$\Sigma M_A = 0$ $-R_B \times 2 + 15 \times 3.5 = 0$ $-2R_B + 52.5 = 0$ $\boxed{R_B = 26.25\ KN}$	½M
	put in eq^n i] $R_A + R_B = 15$ $\therefore R_A = -11.25\ KN$ $\therefore \boxed{R_A = 11.25\ KN\ (\downarrow)}$	½M
	ii) B.M. calculation i) M_C = B.M. at free end = 0 KN-m ii) $M_B = -15 \times 1.5 = -22.5$ KN-m iii) $M_A = 15 \times 3.5 - 26.25 \times 2 = 0$ KN.m	02M
Q-3(f)	i) perpendicular axis theorem it states that the moment of inertia of a plane section about an axis perpendicular to the figure and passing through the centroid is equal to the sum of moment of inertia of the plane figure about two mutually perpendicular axis passing through the C.G or centroid 'G'	02

Q .NO	SOLUTION	MARKS
	ii> parallel axis Theorem	
	the moment of inertia of a plane section about any axis is equal to M.I of the plane section about a parallel axis passing through its centroid or centre of gravity plus the product of area and the square of perpendicular distance betn two axes.	02
Q-4 (a)		
	i> $\sin 60° = \frac{h}{3}$	
	$\boxed{h = 2.598\, m}$	½
	ii> M.I at apex	
	By using parallel axis theorem	
	$I_{AB} = \frac{bh^3}{36} = \frac{3 \times 2.598^3}{36} = 1.4613\, m^4$	½
	$A = \frac{1}{2} \times b \times h = \frac{1}{2} \times 3 \times 2.598 = 3.897\, m^2$	½
	$h = \frac{2}{3}\, h$ from apex	½
	$h = \frac{2}{3} \times 2.598 = 1.732\, m$	
	$I_{AB} = I_G + Ah^2 = 1.4613 + 3.897 \times 1.732^2$	
	$\boxed{I_{AB} = 13.152\, m^4}$ from apex	01

Q .NO	SOLUTION	MARKS
	ii) M.I at ~~apex~~ base	
	$I_{CD} = \frac{bh^3}{12} = \frac{3 \times 2.598^3}{12} = 4.38386\ m^4$ from base	½
	$\boxed{I_{CD} = 4.38386\ m^4}$	½
	or	
	this problem can be solvet by using formule	
	i) M.I at apex	
	$I_{AB} = \frac{bh^3}{4} = \frac{3 \times 2.598^3}{4}$	01 M
	$\boxed{I_{AB} = 4.38386\ m^4}$	01 M
	ii) M.I at base	
	$I_{CD} = \frac{bh^3}{12} = \frac{3 \times 2.598^3}{12}$	01 M
	$\boxed{I_{CD} = 4.38386\ m^4}$	01 M

Q.NO	SOLUTION	MARKS

Q-4 (b)



i) $\bar{X} = \frac{150}{2} = 75$ mm — ½M

Due to symmetry

ii) $A_1 = 150 \times 10 = 1500\,mm^2$

$A_2 = 140 \times 10 = 1400\,mm^2$

iii) $Y_1 = 140 + \frac{10}{2} = 145$ mm

$Y_2 = \frac{140}{2} = 70$ mm

$\bar{Y} = \frac{1500 \times 145 + 1400 \times 70}{1500 + 1400}$

$\bar{Y} = 108.79$ mm — ½M

iv) To find Ixx

$I_{xx} = I_{x_1x_1} + I_{x_2x_2}$ By parallel axis theorem

$I_{xx} = [I_{G_1} + A_1h_1^2] + [I_{G_2} + A_2h_2^2]$ — ½M

$I_{xx} = \left[\frac{150 \times 10^3}{12} + 1500(\bar{Y} - Y_1)^2\right] + \left[\frac{10 \times 140^3}{12} + 1400(\bar{Y} - Y_2)^2\right]$

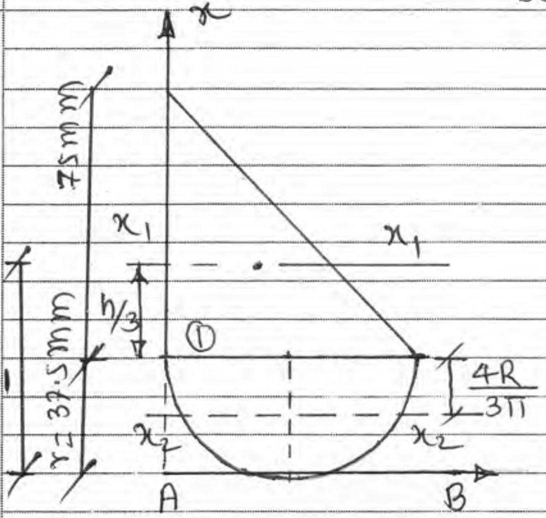
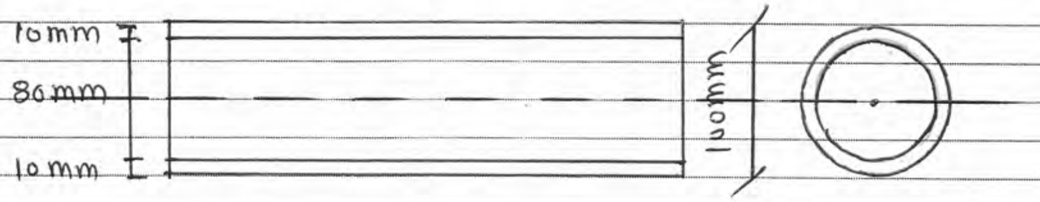
$= \left[\frac{150 \times 10^3}{12} + 1500(108.79 - 145)^2\right] + \left[\frac{10 \times 140^3}{12} + 1400(108.79 - 70)^2\right]$

$= [12500 + (1.967 \times 10^6)] + [2.2866 \times 10^6 + 2.1065 \times 10^6]$

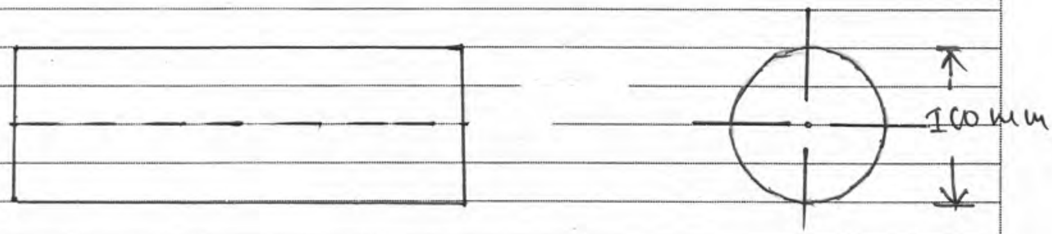
$= 1.979 \times 10^6 + 4.39316 \times 10^6$

$\boxed{I_{xx} = 6.3724 \times 10^6\ mm^4}$ — 01M

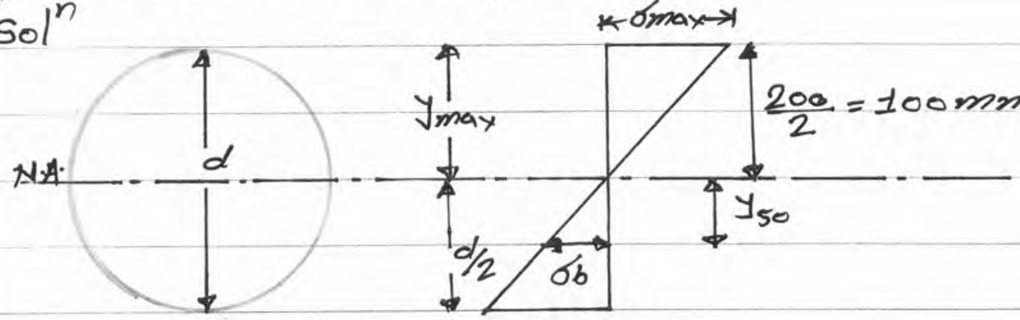
Q .NO	SOLUTION	MARKS
	v) To find I_{yy}	
	$I_{yy} = I_{G_1} + I_{G_2}$	
	$I_{yy} = \frac{d_1 b_1^3}{12} + \frac{d_2 b_2^3}{12}$	1/2 M
	$I_{yy} = \frac{10 \times 150^3}{12} + \frac{140 \times 10^3}{12}$	
	$I_{yy} = 2.8125 \times 10^6 + 11.67 \times 10^3$	
	$\boxed{I_{yy} = 2.824 \times 10^6 \ mm^4}$	01 M

Q .NO	SOLUTION	MARKS
Q-4 (c)		
	i) $A_1 = \frac{1}{2} \times 75 \times 75 = 2812.5\ mm^2$	
	ii) $A_2 = \frac{\pi R^2}{2} = \frac{\pi \times 37.5^2}{2}$	
	$A_2 = 2.2089 \times 10^3\ mm^2$	½ M
	iii) $I_{AB} = I_{x_1x_1} + I_{x_2x_2}$	
	$I_{AB} = \left[I_{G_1} + A_1h_1^2\right] + \left[I_{G_2} + A_2h_2^2\right]$ —— ①	½ M
	$I_{G_1} = \frac{bh^3}{36} = \frac{75 \times 75^3}{36} = 878.9 \times 10^3\ mm^4$	½ M
	$I_{G_2} = 0.11 R^4 = 0.11 \times 37.5^4 = 217.529 \times 10^3\ mm^4$	½ M
	$h_1 = h/3 + R = \frac{75}{3} + 37.5 = 62.5\ mm$	½ M
	$h_2 = R - \frac{4R}{3\pi} = 37.5 - \frac{4 \times 37.5}{3 \times \pi} = 21.58\ mm$	½ M
	$I_{AB} = \left[878.9 \times 10^3 + 2812.5 \times 62.5^2\right] + \left[217.529 \times 10^3 + 2.2089 \times 10^3 \times 21.58^2\right]$	½ M
	$I_{AB} = 11.8649 \times 10^6 + 1.246 \times 10^6$	
	$\boxed{I_{AB} = 13.1109 \times 10^6\ mm^4}$	½ M
Q-4 (d)		

Q .NO	SOLUTION	MARKS
	i) To find moment of inertia (I_{dia})	
	$I_{dia} = \frac{\pi}{64}(D^4 - d^4)$ | $A = \frac{\pi}{4}(D^2 - d^2)$	for I_{dia} = 1M
	$I_{dia} = \frac{\pi}{64}(100^4 - 80^4)$ | $A = \frac{\pi}{4}(100^2 - 80^2)$	
	$\boxed{I_{dia} = 2.898 \times 10^6\ mm^4}$ | $\boxed{A = 2.827433 \times 10^3\ mm^2}$	for A = 1M
	ii) To find radius of gyration (K)	for formulae 1M
	$K = \sqrt{\frac{I_{dia}}{A}} = \sqrt{\frac{2.8981 \times 10^6}{2.8274 \times 10^3}} = \boxed{K = 32.015\ mm}$	1M
Q-4 (e)	Assumption in Theory of simple Bending	
	i) Transverse section of beam which is plane before bending will remain plane after the bending	
	ii) The beam is stressed well up to proportional limit such that it must obey's Hook's law	
	iii) The value of young's modulus (E) is same in tension and compression.	
	iv) the elastic limit is not exceeded.	
	v) The beam is initially straight & unstressed.	1 M for each (write any four)
	vi) Each longitudinal fibre is free to expand or contract independently from every other layer	
	vii) The resultant force across transverse section of the beam is zero.	
	viii) the deformation of the section due to shear force is neglected.	
	ix) The material of the beam is homogeneous and isotropic	

Q .NO	SOLUTION	MARKS
Q-4 (F)	Given shear force = 25 KN	
		
	$d = 100\ mm$ $F = 25 \times 10^3\ N$	
	To find the maximum shear stress (q_{max})	
	$A = \frac{\pi}{4} \times d^2$ $A = \frac{\pi}{4} \times 100^2$ $A = 7853.98\ mm^2$	01
	$q_{av} = \frac{F}{A} = \frac{25 \times 10^3}{7853.98}$ $q_{av} = 3.18\ N/mm^2$	01
	for circular section $q_{max} = \frac{4}{3}\ q_{av}$	01
	$= \frac{4}{3} \times 3.18$ $\boxed{q_{max} = 4.244\ N/mm^2}$	01

Q.NO	SOLUTION	MARKS
Q5 a)	Given, $d = 200\,mm$ $\sigma_{max} = 100\,N/mm^2$	
	find σ_b at 50 mm from N.A	
→	Sol^n	



1) Using the flexural formula

$$\frac{M}{I} = \frac{\sigma_{max}}{y_{max}}$$

$I = \frac{\pi}{64}(d)^4 = \frac{\pi}{64}(200)^4 = 78.539 \times 10^6\,mm^4$ — 01

$y_{max} = d/2 = \frac{200}{2} = 100\,mm$

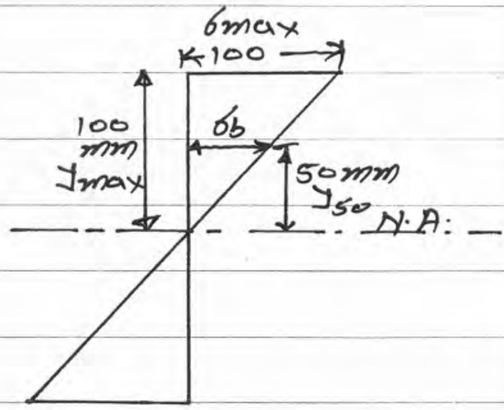
$\sigma_{max} = 100\,N/mm^2$ ....(Given)

$\therefore M = \frac{\sigma_{max}}{y_{max}} \times I = \frac{100}{100} \times 78.539 \times 10^6$

$\therefore M = 78.539 \times 10^6\,N \cdot mm$ — 01

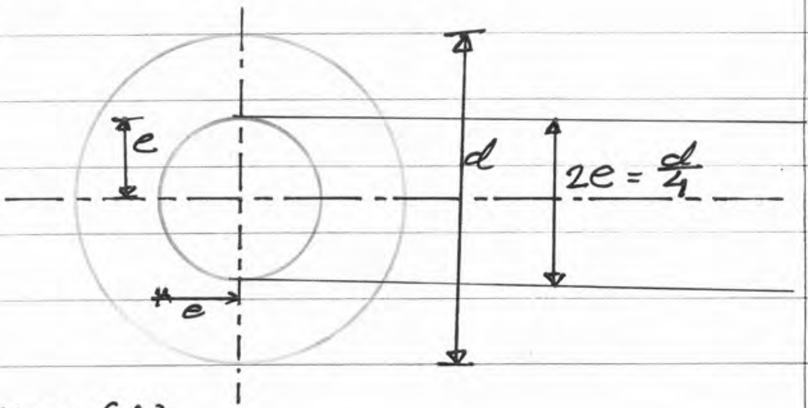
2) Bending stress at 50 mm from N.A
Using the flexural formula.

$$\frac{M}{I} = \frac{\sigma_b}{y_{50}}$$ — 01

Q.NO	SOLUTION	MARKS
Q5a) Cont....	$\frac{78.539\times10^6}{78.539\times10^6} = \frac{\sigma_b}{50}$	
	$\therefore \sigma_b = 50\ N/mm^2$	01
	OR	OR
		
	From similar triangles	
	$\frac{\sigma_{max}}{y_{max}} = \frac{\sigma_b}{y_{50}}$	
	$\frac{100}{100} = \frac{\sigma_b}{50}$	
	$\therefore \sigma_b = 50\ N/mm^2$	01
Q5b	Given, Column size - 600×600 mm, P=6000 kN	
	e = ?, No tension Condition	
→	Sol^n	
	1) For No Tension Condition	
	Direct stress (σ_0) = Bending stress (σ_b)	
	2) Direct stress $\sigma_0 = \frac{P}{A} = \frac{6000\times10^3}{600\times600}$	
	$\therefore \sigma_0 = 16.67\ N/mm^2$	01
	3) Bending stress $\sigma_b = \frac{M}{Z} = \frac{P.e}{Z}$	

Q.NO	SOLUTION	MARKS
Q5b Cont....	$Z = \frac{I}{y_{max}} = \frac{\frac{bd^3}{12}}{d/2} = \frac{bd^2}{6} = \frac{600 \times 600^2}{6}$	
	$Z = 36 \times 10^6\ mm^3$	01
	$\therefore \sigma_b = \frac{6000 \times 10^3 \times e}{36 \times 10^6} = 0.1667\,e\ \ N/mm^2$	01
	Now equate $\sigma_o = \sigma_b$	
	$16.67 = 0.1667\,e$	
	$\therefore e = 100\ mm$	01
	A point load of 6000 KN be placed at an eccentricity of 100mm on the centroidal axis so as to produced no tension condition.	
Q5 c)	$P = 85\ KN,\ b = 3.5t,\ \sigma = 60\ MPa.$	
→	solution, As the load is axial.	
	stress $\sigma = \frac{load}{c/s\ area} = \frac{P}{b \times t}$	01
	$= \frac{P}{3.5t \times t} = \frac{P}{3.5t^2}$	01
	$60 = \frac{85 \times 10^3}{3.5t^2}$	
	$\therefore t = 20.11\ mm$	01
	$\therefore b = 3.5 \times 20.11 = 70.38\ mm.$	01

Q.NO	SOLUTION	MARKS
Q5d	Given, diameter = d, load = P	
	eccentricity = e	



1> Area of section (A)

$$A = \frac{\pi}{4}(d)^2$$

2> Section Modulus (Z)

$$Z = \frac{I}{y_{max}}$$ 01

$$I = \frac{\pi}{64}(d)^4 \qquad y_{max} = \frac{d}{2}$$

$$\therefore Z = \frac{\frac{\pi}{64}(d)^4}{d/2} = \frac{\pi}{32}(d)^3$$ 01

3> For No Tension Condition

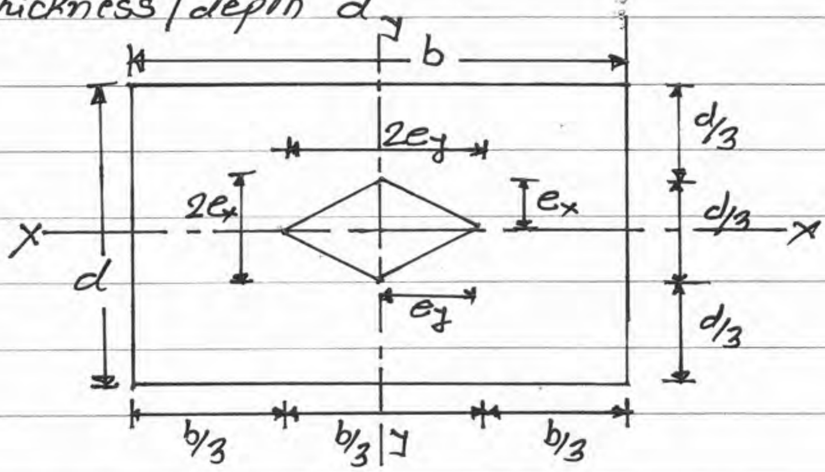
$$e \leq \frac{Z}{A}$$ 01

$$e \leq \frac{\frac{\pi}{32}(d)^3}{\frac{\pi}{4}(d)^2}$$

$$e \leq \frac{d}{8}$$ 01

$$\therefore e_{max} = \frac{d}{8}$$

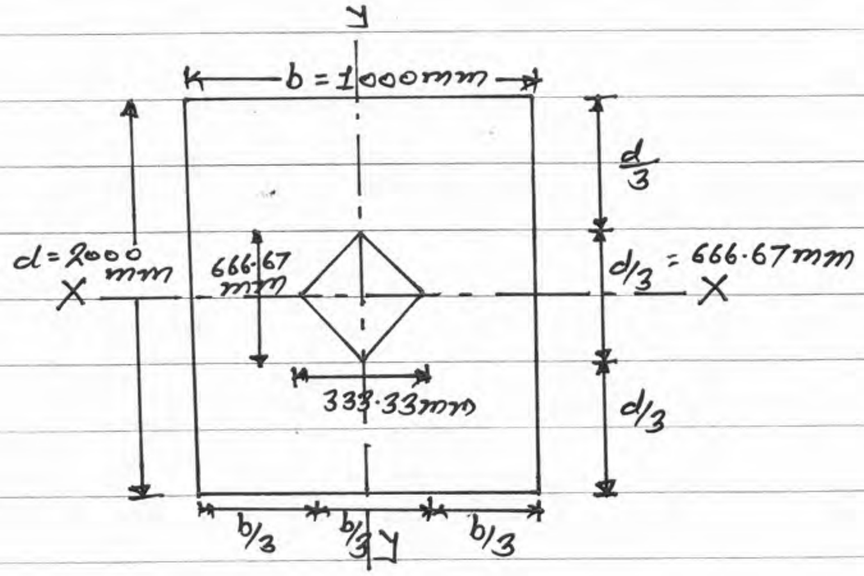
Q.NO	SOLUTION	MARKS
Q5d Cont...	$\therefore 2e_{max} = 2 \times \frac{d}{8} = d/4$	
	For no tension condition the load must lie within a circle of diameter 2e i.e d/4.	
Q5e		
→	Core of a section :- The Centrally located portion of a section within which the load must act so as to produce only Compressive stress is called a Core or Kernel of a Section.	01
	Let us Consider a rectangular section of width b & thickness/depth d.	



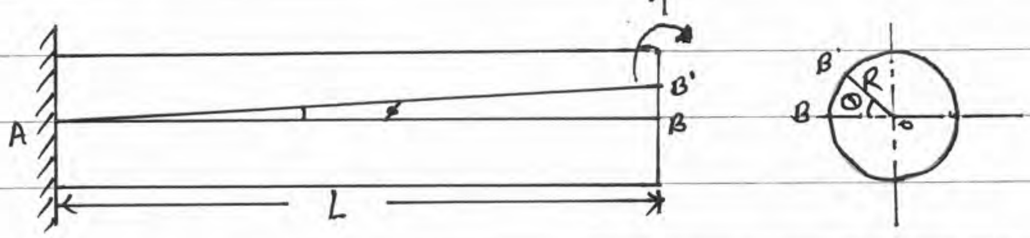
1) Area of section $A = b \times d$

2) Section modulus

$$Z_{xx} = \frac{I_{xx}}{y_{max}} \qquad Z_{yy} = \frac{I_{yy}}{y_{max}}$$

Q.NO	SOLUTION	MARKS
Q5e Cont...	$Z_{xx} = \dfrac{\frac{bd^3}{12}}{d/2}$ $\qquad$ $Z_{yy} = \dfrac{\frac{db^3}{12}}{b/2}$	
	$Z_{xx} = \dfrac{bd^2}{6}$ $\qquad$ $Z_{yy} = \dfrac{db^2}{6}$	01
	For No Tension Condition	
	$e_x \le \dfrac{Z_{xx}}{A}$ $\qquad$ $e_y \le \dfrac{Z_{yy}}{A}$	01
	$\le \dfrac{\frac{bd^2}{6}}{b \times d}$ $\qquad$ $\le \dfrac{\frac{db^2}{6}}{bd}$	
	$\therefore \quad e_x = d/6$ $\qquad$ $e_y = \dfrac{b}{6}$	
	$\therefore \quad 2e_x = d/3$ $\qquad$ $2e_y = b/3$	01
	The load must lie within the middle third area of eccentricity 2e as shown in fig. It is called as core or kernel of section.	
Q5f →	Sol^n 	01

Q.NO	SOLUTION	MARKS
Q5 f. Cont...	For No Tension Condition	
	$e_x \le \frac{Z_{xx}}{A}$ $\quad$ $e_y \le \frac{Z_{yy}}{A}$	01
	$e_x \le \frac{\frac{bd^2}{6}}{b \cdot d}$ $\quad$ $e_y \le \frac{\frac{db^2}{6}}{b \times d}$	
	$e_x = d/6$ $\quad$ $e_y = b/6$	01
	$\therefore\ 2e_x = d/3$ $\quad$ $2e_y = b/3$	
	$= \frac{2000}{3}$	
	$2e_x = 666.67mm$ $\quad$ $2e_y = \frac{1000}{3}$	
	$2e_y = 333.33mm$	01
	The Core of section is shown in fig. above of size $2e_x \times 2e_y$.	

Q.NO	SOLUTION	MARKS
Q6 a		
→	Pure Torsion	
	If the shaft is subjected to two equal and opposite torques at its two ends, it is said to be under pure torsion. A shaft is subjected to pure torsion if it is subjected to only twisting moment and no other bending moment or thrust acts on the shaft.	01
	* Assumption's in the theory of torsion	
	1) The shaft is homogeneous & isotropic.	
	2) The shaft is straight having uniform circular cross section	02
	3) Twisting along the shaft is uniform	
	4) Plain section before twisting remain plain after twisting.	
	5) Stresses do not exceed the proportional limit.	
		01
	Longitudinal section cross-section	

Q.NO	SOLUTION	MARKS
Q6 b	Given, $P = 800\,kW$, $N = 200\,rpm$, $T_{max} = 1.3\,T_{mean}$	
	$\tau_{max} = 80\,N/mm^2$	
→	Sol^n	
	1) Power transmitted by shaft	
	$P = \frac{2\pi N T_{mean}}{60}$; $800\times10^3 = \frac{2\pi\times200\times T_{mean}}{60}$	01
	$\therefore T_{mean} = 38197.18\ N\cdot m$	
	As $T_{max} = 1.30\,T_{mean}$	
	$T_{max} = 49.656\times10^6\ N\cdot mm$	01
	2) Using Torsion formula.	
	$\frac{T_{max}}{I_P} = \frac{\tau}{R}$	
	$I_P = I_{xx} + I_{yy} = 2\times\frac{\pi}{64}(d)^4 = \frac{\pi}{32}\times d^4$	
	$R = d/2$	
	$\therefore T_{max} = I_P\times\frac{\tau}{R} = \frac{\frac{\pi}{32}\times(d)^4\times 80}{d/2}$	
	$T_{max} = \frac{\pi}{16}d^3\times 80$	01
	$49.656\times10^6 = \frac{\pi}{16}\times d^3\times 80$	
	$\therefore d = 146.76\ mm$	01

Q .NO	SOLUTION	MARKS
Q6 C	Given, $P = 200\ HP$, $N = 180\ rpm$, $\tau = 90\ N/mm^2$	
	$\theta = 1^\circ = \frac{\pi}{180}\ rad$, $L = 5m$ $C/G = 0.82\times10^5\ N/mm^2$	
$\longrightarrow$	Sol^n	
1>	Power transmitted by shaft in HP	
	$P = \frac{2\pi N T}{4500}$ .... T is in kg.m	$\frac{1}{2}$
	$200 = \frac{2\times\pi\times180\times T}{4500}$	
	$T = 795.774\ kg.m$	
	$= 795.774\times9.81$	
	$T = 7806.549\ N.m$	
	$T = 7.806\times10^6\ N.mm$	1
2>	Diameter of shaft based on strength	
	Using the relation, $\frac{T}{I_p} = \frac{\tau}{R}$	$\frac{1}{2}$
	$\frac{7.806\times10^6}{\frac{\pi}{32}\times d^4} = \frac{90}{d/2}$	
	$d^3 = 441.760\times10^3$	
	$\therefore d = 76.16\ mm.$	$\frac{1}{2}$
3>	Diameter based on stiffness.	
	Using the relation, $\frac{T}{I_p} = \frac{G\theta}{L}$	$\frac{1}{2}$

Q.NO	SOLUTION	MARKS
Q6 c Cont...	$\dfrac{7.806\times10^6}{\frac{\pi}{32}\times d^4} = \dfrac{0.82\times10^5\times(1\times\frac{\pi}{180})}{5000}$	
	$\therefore d^4 = 279.70\times10^6$	
	$\therefore d = 129.32\ mm$	01
	$\therefore$ The suitable diameter is greater of above two values $\therefore d = 129.32\ mm$.	
Q6 d)	Given, $D = 400mm$, $d = 200\ mm$. $\theta = 1.5^\circ$	
	$L = 10m$, $C/G = 0.85\times10^5\ N/mm^2$	
→	Solution	
1)	Torque transmitted by shaf.	
	Using the relation	
	$\dfrac{T}{I_p} = \dfrac{G\theta}{L}$	01
	$\dfrac{T}{\frac{\pi}{32}\times(D^4-d^4)} = \dfrac{0.85\times10^5\times(1.5\times\frac{\pi}{180})}{10000}$	01
	$\therefore T = \frac{\pi}{32}(400^4-200^4)\times\dfrac{0.85\times10^5\times(1.5\times\frac{\pi}{180})}{10000}$	
	$T = 524.32\times10^6\ N.mm$	
	$T = 524.32\ KN.m$	02

Q.NO	SOLUTION	MARKS
Q6 e	Given, $T = 36\ KN\cdot m$, $d = 0.6D$, $\tau = 83\ N/mm^2$	
→	Solution, Using the Relation.	
	$\dfrac{T}{I_p} = \dfrac{\tau}{R}$	
	$I_p = I_{xx} + I_{yy} = 2 \times \dfrac{\pi}{64}(D^4 - d^4) = \dfrac{\pi}{32}(D^4 - d^4)$	01
	$R = D/2$	
	$\therefore T = I_p \times \dfrac{\tau}{R}$	
	$= \dfrac{\dfrac{\pi}{32}(D^4 - d^4) \times \tau}{\dfrac{D}{2}}$	
	$T = \dfrac{\pi}{16}\dfrac{(D^4 - d^4)}{D} \times \tau$	01
	But $d = 0.6D$ put in above eq^n.	
	$T = \dfrac{\pi}{16}\left[\dfrac{D^4 - (0.40)^4}{D}\right] \times 83$	
	$36 \times 10^6 = \dfrac{\pi}{16}\left[\dfrac{0.9744 D^4}{D}\right] \times 83$	
	$36 \times 10^6 = 15.879\ D^3$	
	$\therefore D = 131.36\ mm$	01
	$\therefore d = 0.6 \times 131.36$	
	$d = 78.82\ mm$	01

Q.NO	SOLUTION	MARKS
Q6 f		
→	Section Modulus (Z)	
	It is the ratio of moment of Inertia of the section about the neutral axis and the distance of the most extreme layer from the neutral axis.	01
	Section Modulus about x-x axis $Z_{xx} = \frac{I_{xx}}{y_{max}}$	
	Section Modulus about y-y axis $Z_{yy} = \frac{I_{yy}}{y_{max}}$	
	S.I. Unit of section modulus is = mm^3, cm^3, m^3 etc.	01
2)	Strength of hollow shaft.	
	$\frac{T}{I_p} = \frac{\tau}{R}$	
	$T = \frac{I_p \times \tau}{R}$ .... $I_p = \frac{\pi}{32}(D^4 - d^4)$, $R = D/2$	
	$= \frac{\frac{\pi}{32}(D^4 - d^4)\tau}{D/2}$	
	$\boxed{T = \frac{\pi}{16}\left[\frac{D^4 - d^4}{D}\right] \times \tau}$	01
	where, T = Torque transmitted by shaft (N·mm) D = External dia. of shaft (mm) d = Internal dia. of shaft (mm) τ = Permissible shear stress of shaft material (N/mm²)	01