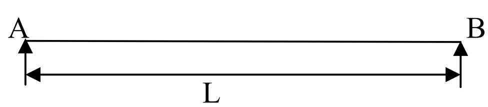
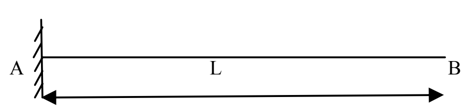
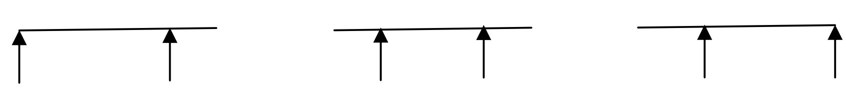
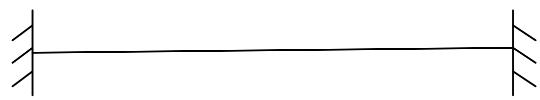
MAHARASHTRA STATE BOARD OF TECHNICAL EDUCATION

(Autonomous)
(ISO/IEC-27001-2005 Certified)
SUMMER- 13 EXAMINATION

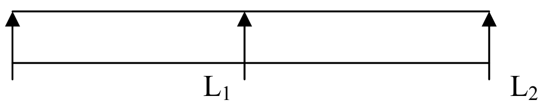
Subject Code: 12043 **Model Answer**

Q. No.	Answer	Marks
1 a)	**Elasticity**:-it is the property of a material by virtue of which it regains its original size & shape after deformation, when the loads causing deformation are removed.	1 M
	Plasticity:- the plasticity of a material is the ability to change its shape without destruction under the action of external loads & to regain the shape given to it when the forces are removed	1 M
b)	**Data** :- L= 500mm, d=22mm, δL=1.2mm, P=105KN =105 x 10^3N.	
	To find:- i) σ ii) e iii) E	
	Solution:- i) Stress $\sigma = P/A = (105X10^3)/((\pi/4)x22^2) = 276.22 N/mm^2$	1/2 M
	ii) Strain :- $e = \delta L/L = 1.2/500 = 0.0024$	1/2 M
	iii) Modulus of elasticity , $E = \sigma/e = 276.22/0.0024 = 115091..67\ N/mm^2$	1 M
	Answer:- i) $\sigma = 276.22 N/mm^2$ ii) e = 0.0024 iii) E = 115091..67 N/mm^2	
c)	1. **Simply supported beam**: a beam which is freely supported on the wall or column its both the ends is called as a simply supported beam.  	1/2 M Each (any four)
	2. **Cantilever Beam**: a beam fixed at one end & free at the other is called as cantilever beam  	
	3. **Overhanging Beam**: if the end portion of the beam extend beyond the support called as an overhanging beam. a beam may be overhanging on one side or both side.  i) overhanging at right side ii) overhanging on both side iii) overhanging on left side	

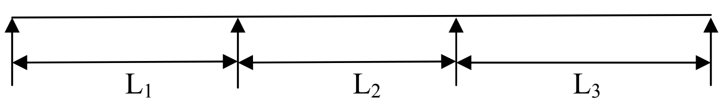
4. **Fixed beam**: A beam whose both the ends are rigidly fixed in wall is called fixed beam, constrained beam, built-in beam or an encastre beam.



5. **Continuous beam**: - a beam which is supported on more than two supports (i.e. a three support) is called continuous beam. The end support of a beam may be simply supported or fixed.



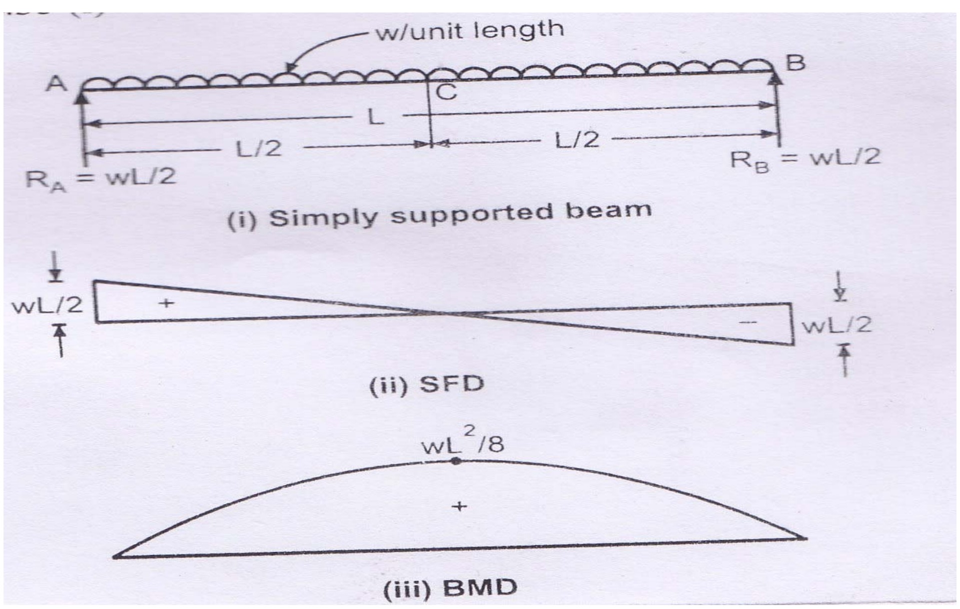
i) two span continuous beam



ii) Three span continuous beam

D

A simply supported beam of span L carrying a U.D.L w/unit length over the entire span as shown in fig.



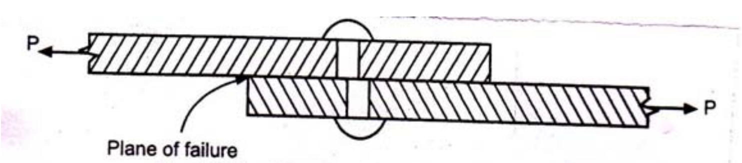
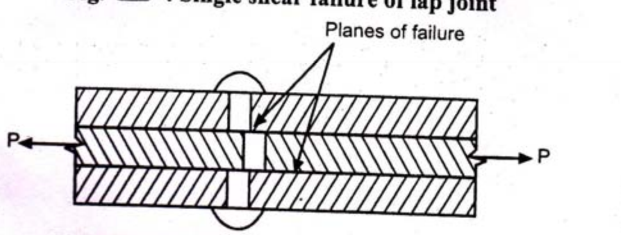
1 M

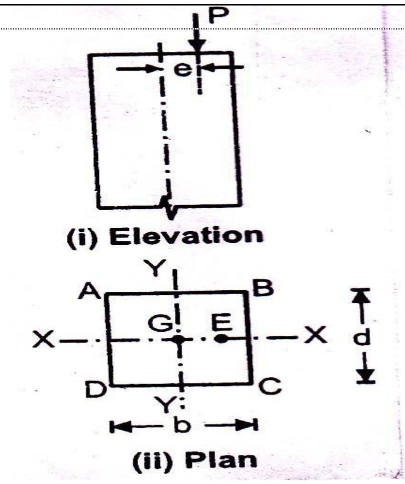
1 M

e

Parallel Axis Theorem. 2 M

It states that "The moment of inertia of a plane section about any axis parallel to the centroidal axis is equal to the moment of inertia of the section about the centroidal axis plus

	the product of the area of the section and the square of the distance between the two axes.”	
f	**Polar moment of inertia** “The moment of inertia of a plane area about an axis perpendicular to the plane of figure is called as Polar moment of inertia with respect to point, where the axis intersects the plane.” $I_p = I_{xx} + I_{yy} = \frac{\pi}{64}D^4 + \frac{\pi}{64}D^4 = 2 \times \frac{\pi}{64}D^4 = \frac{\pi}{32}D^4$	1 M
	For a hollow circular shaft of external diameter ‘D’ and internal diameter ‘d’, $I_p = I_{xx} + I_{yy} = \frac{\pi}{64}(D^4-d^4) + \frac{\pi}{64}(D^4-d^4) = 2x\frac{\pi}{64}(D^4-d^4) = \frac{\pi}{32}(D^4-d^4)$	1 M
g	**Section Modulus**: It is the ratio of mi of the section about the neutral axis & the distance of the most extreme layer from the neutral axis. The flexural formula $\frac{M}{I} = \frac{\sigma}{y}$, Can be written as $m = \sigma \times \frac{I}{y} = \sigma \times Z$ Where $Z = \frac{I}{y}$ is called as modulus of section or section modulus. Z_{XX}= Section modulus about X-X axis = I_{xx}/ y_{max} Z_{YY}= Section modulus about Y-Y axis = I_{yy}/ y_{max}	1 M
	Neutral Axis: Neutral layer or neutral surface. **The intersection of neutral layer with any normal cross section of a beam is called neutral axis (N.A.)** All the layers above the neutral axis are under compression while those below the neutral axis are under tension. Hence the compressive stresses are developed in the layers above the N.A. there is no stress of any kind i.e. **the bending stress at the N.A. is Zero**	1 M
h	Single shear stress = shear load/area subjected to shear = $P/(\pi/4 \times d^2)$	½ M
	Double shear stress = shear load/area subjected to shear = $P/2(\pi/4 \times d^2)$	½ M
	 Fig. [redacted] : Single shear failure of lap joint	1 /2 M
	 Fig. [redacted] : Double shear failure of butt joint	1 /2
i	**Ecentric Loading**: A load whose line of action does not coincide with the axis of a member is called an eccentric load. The distance between the geometric axis of the body and the point of loading is called an eccentric limit or limit of eccentricity. It is denoted by ‘e’.	1 M



1 M

For no tension condition, direct stress should be greater than or equal to bending stress. 1 M

J Mathematically, $\sigma_o \geq \sigma_b$

$$\frac{P}{A} \geq \frac{M}{Z}$$

$$\frac{P}{A} \geq \frac{P.e}{Z}$$

$$\frac{1}{A} \geq \frac{e}{Z}$$

$$\frac{e}{Z} \leq \frac{1}{A}$$

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

1 M

Hence for no tension condition, eccentricity should be less than $\frac{Z}{A}$, or maximum it should be equal to $\frac{Z}{A}$

k **Torque**: 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 a torque which is equal to the product of the force and the radius 1 M

S. I. Unit of Torque:

Since Torque = Force x Radius.

1 M

$\boxed{\text{S.I.Unit} = \text{N.m}}$

l

1 M

Power of the shaft $P = \frac{2\pi NT}{60}$ watts

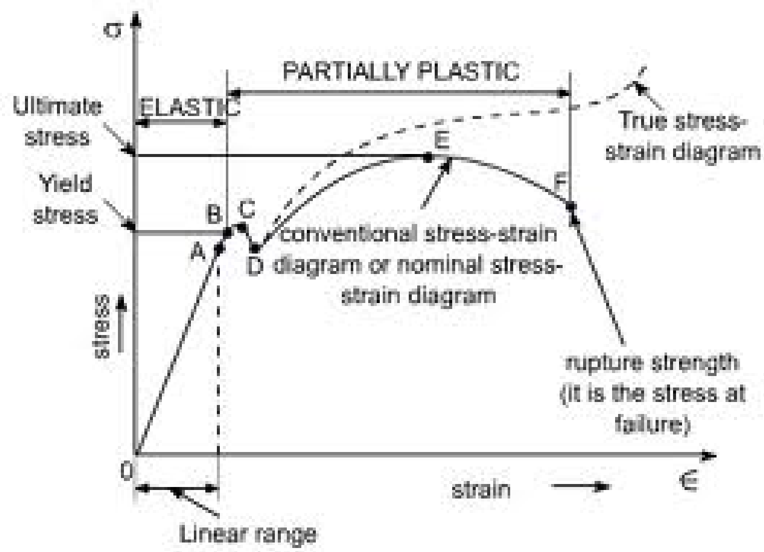
where P=power in watts

T = Average torque in N.m

N = Number of revolution of shaft per minute (r.p.m.)

H.P.Transmitted by a shaft 1 M

$P = \frac{2\pi NT}{4500}$ H.P. Where P = Horse Power, T = Average torque in kg.m

<table>
<tr><td>Q-2</td><td></td><td></td></tr>
<tr><td>A</td><td>


Limit of proportionality- In the range of OA the strain is proportional to the stress and the graph is straight line. Point n A is called as the limit of proportionality. It is the value of the stress up to which stress and strain has the constant ratio and the Hook's law is obeyed.
Elastic limit- at the point A, the curve deviates from the straight line and the stress –strain graph from A to B in nonlinear. If the load is increased beyond the A up to the point B, the material behaves in the elastic manner that is on the removal of the load, the whole deformation will vanish. The value of stress corresponding to point B up to which the material behaves in an elastic manner is called the elastic limit.
Upper Yield point: Point C is called upper yield point.At this point there is an increase in the strain even though there is no increase in stress (load)
A formation of creep makes specimen plastic and the material begins to flow.the value of stress corresponding to point C is called yeild stress or yield strength.The yield stress is defined as that unit stress which will cause an increase in length without an increase in load.
Lower yield point: A load may rise and fall while yielding occures. This is indicated by wavy apperance of the stress-strain graph between C and D.Point D corresponding to lower yield point.after yielding has ceased at D, further stresses and strain can be obtained by increasing the load.
Ultimate Load Point-: after increasing the load beyond the yield point, the stress-strain curve rises till the point E is reached which is called ultimate load; the stress corresponding to this point is called ultimate stress or ultimate tensile strength.
Breaking load point: up to E, the cross-sectional area of the specimen goes on uniformly decreasing forming a neck or waist and the load required to cause further extension is also reduced. As the elongation continues, cross-sectional area becomes smaller and smaller and ultimately the specimen is broken at F into two pieces giving cup cone type of ductile fracture. Point F is called as breaking load point and the stress corresponding to this point is called breaking stress & rupture stress.
</td><td>2 Marks

2 Marks</td></tr>
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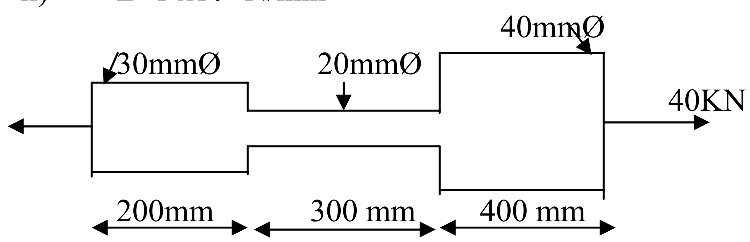
b)

Given

i) P=40KN

ii) $E=1 \times 10^5\ N/mm^2$

1 Mark



L1=200mm   L2=300mm   L3=400mm

$A1=(\pi/4)x30^2=706.86\ mm^2$

$A2=(\pi/4)x20^2=314.16\ mm^2$

$A3=(\pi/4)x40^2=1256.64\ mm^2$

1 Mark

$DL=DL_1+ DL_2+ DL_3$

1 Mark

$= (PL/AE)_1+(PL/AE)_2+(PL/AE)_3$

$= (40\ x10^3/1x10^5)\ X\ ((200/706.86)\ X\ (300/314.15)\ X\ (400/1256.64))$

DL=0.62mm

Max.stress= $P/A_2 = (40\ x10^3)/(314.15)$

1 Mark

$\mathbf{6_{max}=127.32\ N/mm^2}$

C)

Given-:

i) E=170Mpa

ii) μ=0.32

$E=2G(1+\mu)$ **1 Mark**

170=2G(1+0.32)

170= 2.64 G

G=64.39Mpa **1 Mark**

$E=3K(1-2\mu)$ **1 Mark**

170=3K(1-2 x 0.32)

K**=157.40 Mpa** **1 Mark**

d)

Given-:

i) E=200Gpa
ii) μ =0.3
iii) P=20KN
iv) b=50mm
v) d=50mm
vi) L=3000mm

DL=PL/AE

=$(20 \text{x} 10^3 \text{X} 3000)/(50\text{X}50\text{X}200\text{X}10^3)$

DL=0.12mm

1 Mark

To find the change in thickness and width

Linear strain(e) =DL/L

e=0.12/3000

e= $4\text{X}10^{-5}$

1 Mark

Lateral strain = dt/t or db/b

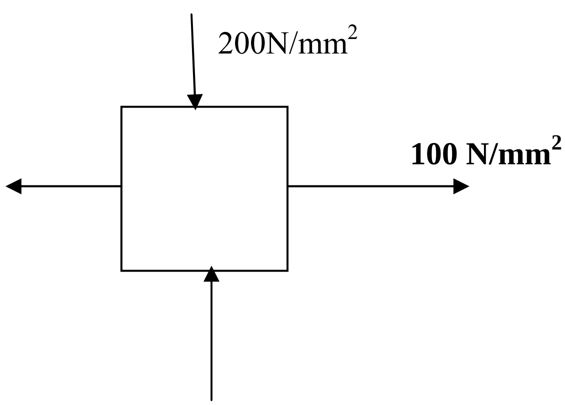
1 Mark

$-1.2\text{X}10^5$ =dt/50

dt=$-6 \text{ x}10^{-4}$ mm (decrease)

1 Mark

e)



Given –

μ =0.3

E= $2 \text{ x}10^5$ N/mm²

Strain in x-direction

e_x=1/E **x** $(6_x- \mu 6_y)$

1 Mark

e_x =$1/2 \text{ x}10^5$ x ((100)-(-0.25 X200))

e_x **=$7.5 \text{ x } 10^{-4}$ (Tensile)**

1 Mark

	$e_y = 1/E \mathbf{x} (6_y - \mu 6_x)$ $e_y = 1/2 \times 10^5 \times ((-200)-(0.25 \times 100))$ $e_y = \mathbf{1.125 \times 10^{-3}}$ **(compressive)**	1 Mark 1 Mark
f)	Given-: i) P=8KN ii) $A_C=20mm^2$ iii) $A_s=30mm^2$ iv) $E_C= 1 \times 10^5 MPa$ v) $E_s= 20 \times 10^5 Mpa$ $\mathbf{e_s=e_c}$ $(6_s/E_S)= (6_C/E_C)$ $6_s=((20 \times 10^5)/ (20 \times 10^5))\, 6_C$ $\mathbf{6_s=20\ 6_C}$ $P=P_S+ P_C$ $P=6_S A_S+6_C A_C$ $8 \times 10^3=20\ 6_C\ 30+6_C\ 20$ $8 \times 10^3=620 6_C$ $6_C=(8 \times 10^3)/(620)$ $\mathbf{6_C=12.90\ N/mm^2}$ $6_S=20 6_C$ $6_S=20 \times 12.90$ $\mathbf{6_s=258\ N/mm^2}$	1 Mark 1 Mark 1 Mark 1 Mark
Q-3 a)	Given -: i) P=35KN ii) $E=2 \times 10^5 N/mm^2$ iii) $\mu=0.3$ iv) b=20mm v) t=15mm vi) L=2000mm **To calculate change of length** dL=PL/AE $=(35 \times 10^3 \times 2000)/(20 \times 15 \times 2 \times 10^5)$ **dL=1.167mm** **To calculate change of thickness and width** **e = dL/L** = 1.167/2000	1 Mark

b)

e = 5.83X10^{-4} 1 Mark

change of thickness

dt/t= - μ X e

dt/15 = - 0.3 X 5.83X10^{-4}

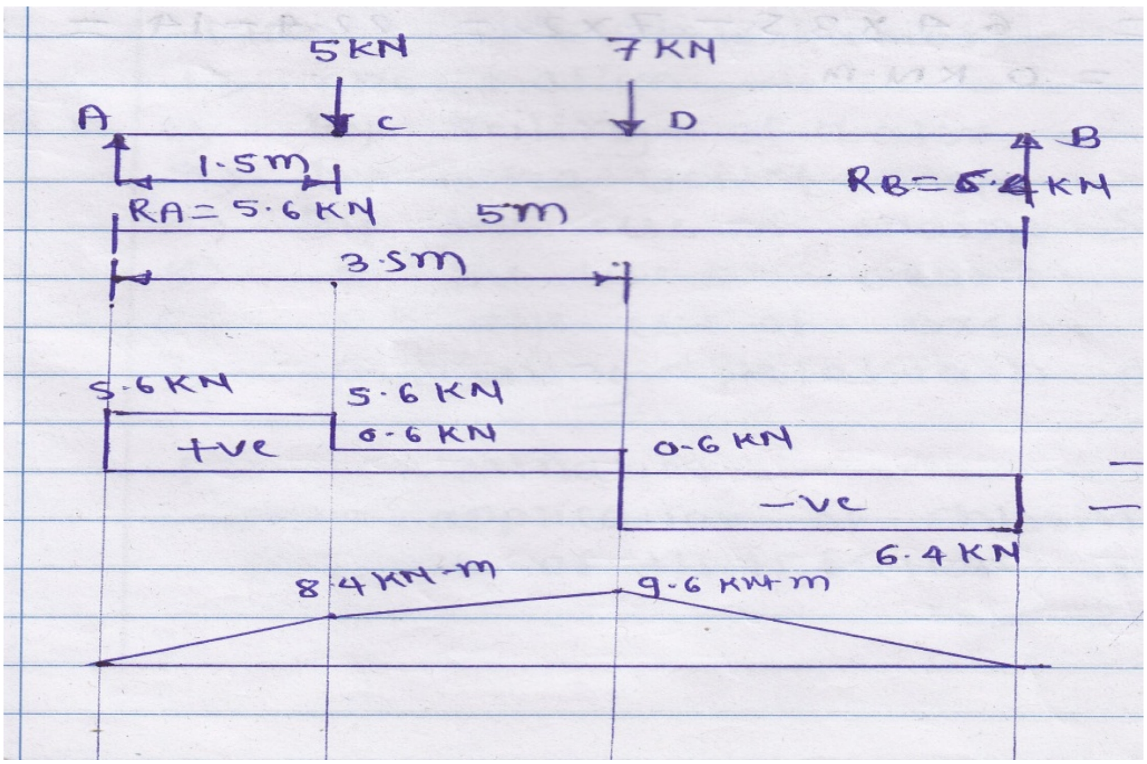
dt= 2.62 X10^{-3} (decrease) 1 Mark

change of width

db/b= - μ X e

db/20= - μ X 5.83X10^{-4}

db= 3.498 X10^{-3} (decrease) 1 Mark



1 Mark

1 Mark

Reaction calculation-

∑ Fy =0

R_A+R_B=12……………………………….**i**

∑ M_A =0

-R_B X 5+ 7 X 3.5 + 5 X 1.5 =0

R_B= 6.4 KN

R_A= 5.6KN

S.F Calculation

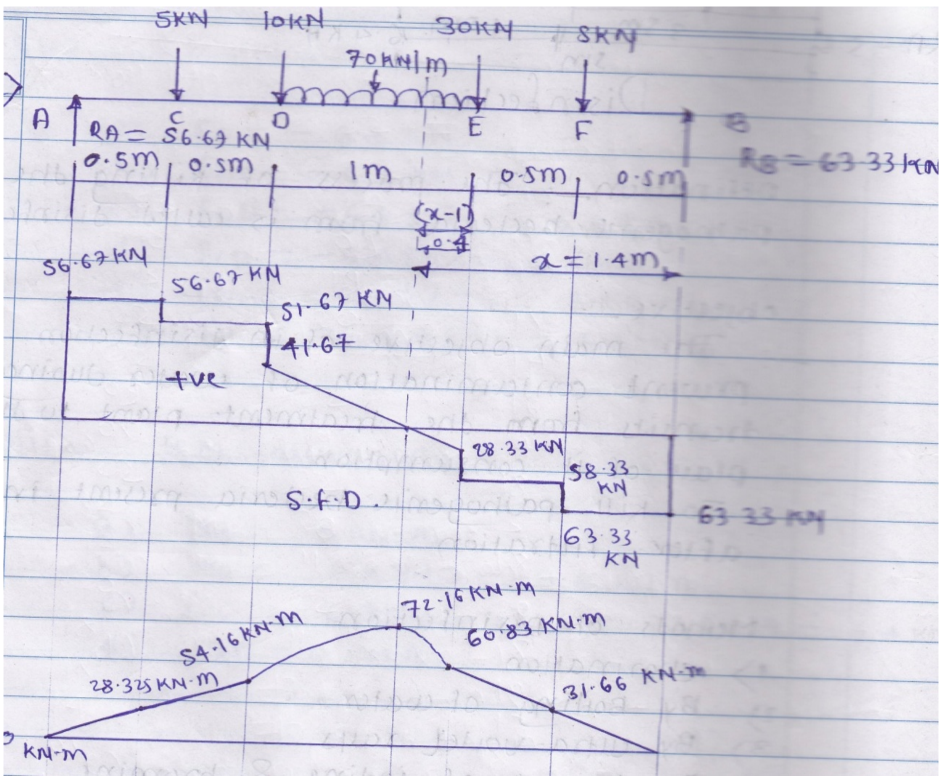
a) F_B = - 6.4 KN

b) F_{DR}= - 6.4 KN

b) F_{DL}= 0.6 KN
b) F_{CR}= 0.6 KN
b) F_{CL}= 5.6 KN
b) F_A= 5.6 KN **1 Mark**

B.M Calculation
a) M_B = 0 KN-m
b) M_D= 9.6 KN-m **1 Mark**
b) M_C= 8.4 KN-m
b) M_A= 0 KN-m

C)



1 Mark

1 Mark

Reaction calculation-

$\sum Fy = 0$
$R_A + R_B = 120$ ………………………………….i
$\sum M_A = 0$
$-R_B \times 3 + 5 \times 2.5 + 30 \times 2 + 70 \times 1(0.5+1) + 10 \times 1 + 5 \times 0.5 = 0$
R_B= 63.33 KN
R_A= 56.67KN

S.F Calculation
a) F_B = - 63.33 KN
b) F_{FR}= - 63.33 KN
c) F_{FL}= - 58.33 KN
d) F_{ER}= - 58.33 KN

e) F_{EL}= - 28.33 KN
f) F_{DR}= 41.67 KN
g) F_{DL} = 51.67 KN
i) F_{CR}= 51.67 KN
b) F_{CL}= 56.67 KN
b) F_A= 56.67 KN **1 Mark**

To Locate the point of contashear

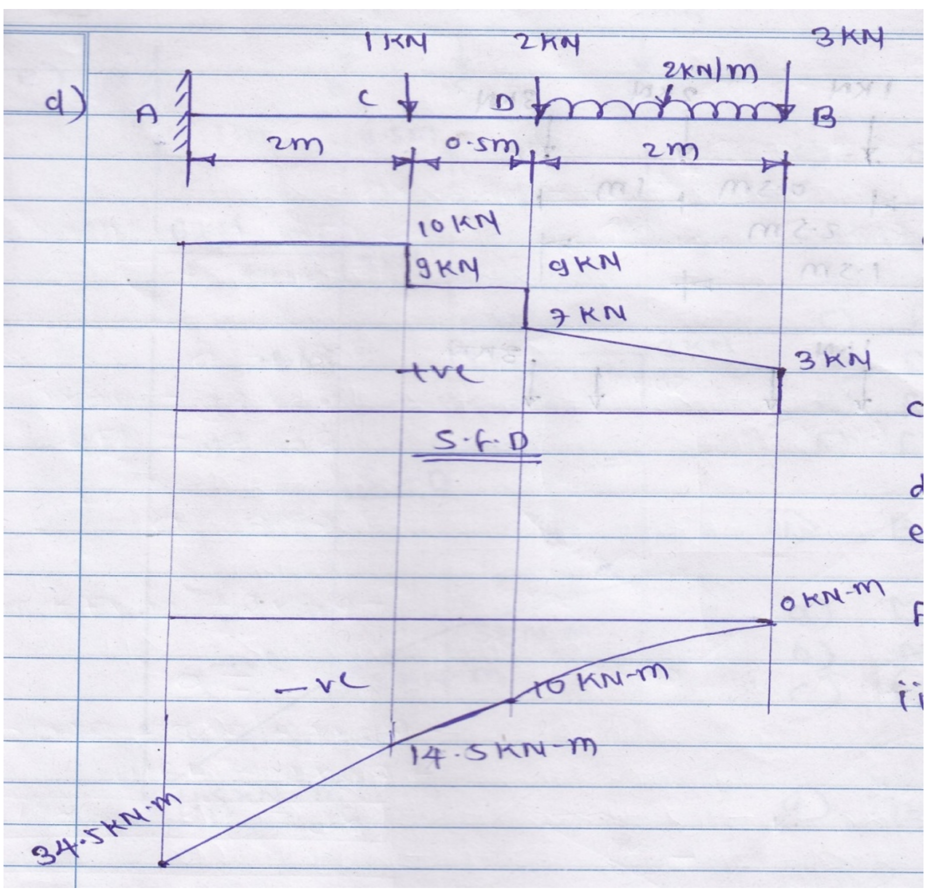
Fpzc=-63.33+5+30+70(x-1)

X= 1.40m

B.M Calculation

a) M_B = 0 KN-m
b) M_F= 31.66 KN-m
c) M_E= 60.83 KN-m
d) M_D= 54.16 KN-m
e) M_C= 28.325 KN-m
f) M_A= 0 KN-m
g) M_{max}= 72.162 KN-m **1 Mark**

d)



1 Mark

1 Mark

e)

S.F Calculation

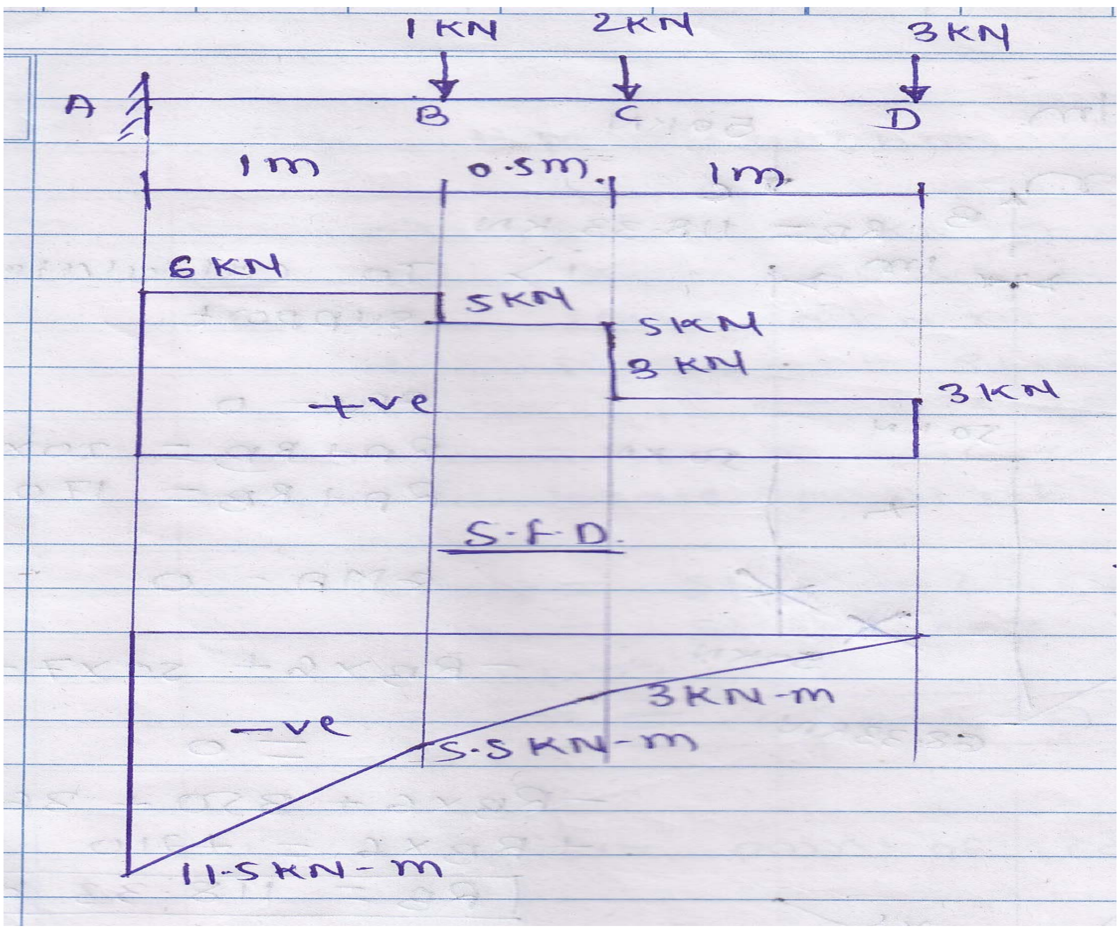
a) $F_B = 3$ KN
b) $F_{DR}= 7$ KN
c) $F_{DL}= 9$ KN
d) $F_{CR}= 9$ KN
e) $F_{CL}=10$ KN
f) $F_A= 10$ KN

1 Mark

B.M Calculation

a) $M_B = 0$ KN-m
b) $M_D= -10$ KN-m
c) $M_C= -14.5$ KN-m
d) $M_A= -34.5$ KN-m

1 Mark



1 Mark

1 Mark

S.F Calculation

a) $F_B = 3$ KN
b) $F_{CR}= 3$ KN
c) $F_{CL}= 5$ KN
d) $F_{BR}= 5$ KN

1 Mark

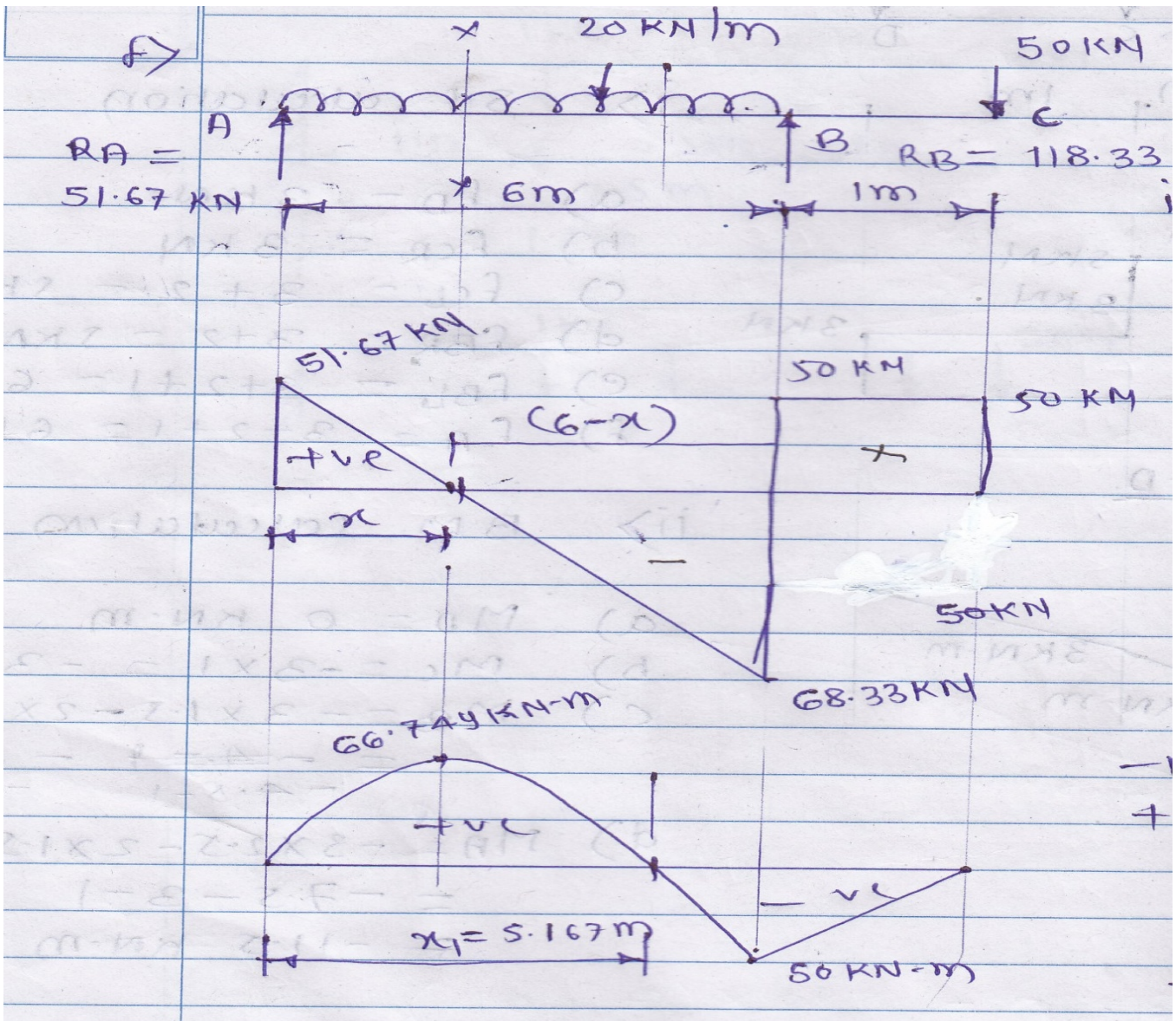
e) F_{BL}=6 KN
f) F_A= 6 KN

B.M Calculation

a) M_D = 0 KN-m
b) M_C= -3 KN-m
c) M_B= -5.5 KN-m
d) M_A= -11.5 KN-m

1 Mark

f)



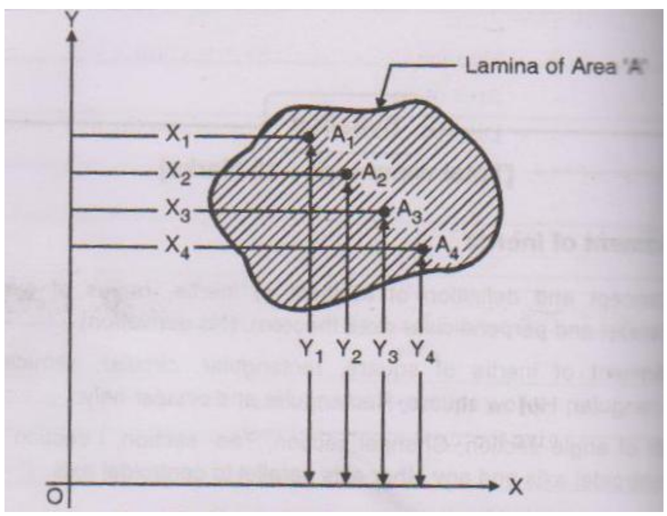
1 Mark

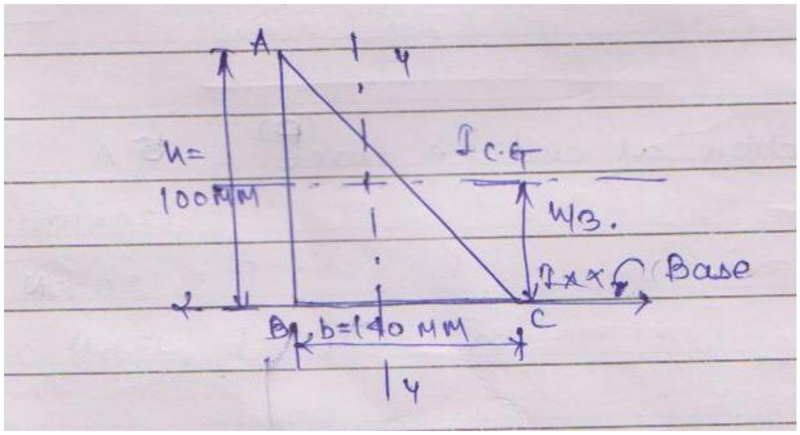
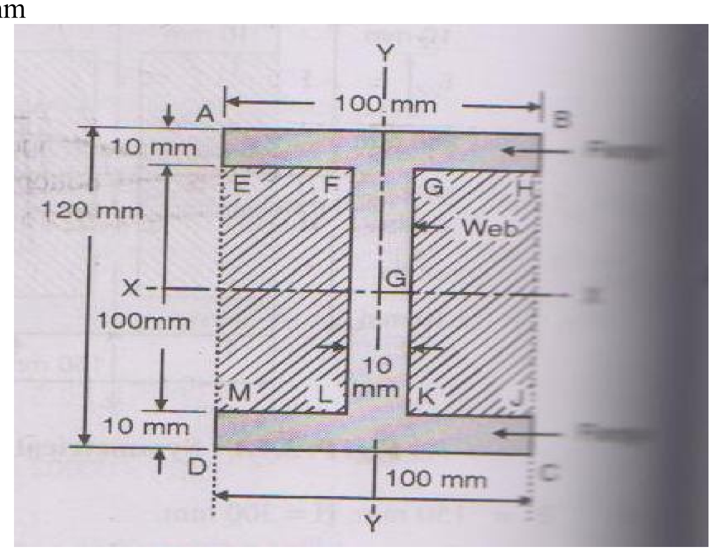
1 Mark

Reaction calculation-

$\sum Fy = 0$
RA+RB=170 ……………………………….i
$\sum M_A = 0$
-RB X 6+ 50 X 7 + 20x 6 X 6/2 =0
RB= 118.33 KN
RA= 51.67KN

	S.F Calculation a) F_c = 3 KN b) F_{BR}= 3 KN c) F_{BL}= 5 KN d) F_A= 5 KN **To locate point of contrashear** Fpc= 51.67 – 20 X **X = 2.5835m** **B.M Calculation** a) M_C = 0 KN-m b) M_B= -50 KN-m c) M_A= 0 KN-m d) M_{max}= 51.67 x 2.5835 – 20 X 2.5835 x(2.5835/2) **M_{max}=66.749 KN-m** **To locate point of contraflexure** Mpc= 51.67 X_1 -20 X_1(X_1/2) 0= 51.67 X_1-10 X_1^2 51.67-10X_1=0 **X_1=5.167m**	**1 Mark** **1 Mark**

Q.4 a]	**Given-** for a rod of square cross section Dimensions- 10mm*10mm so Area=100mm^2 L= length of rod=1000mm, E=Young's Modulus=$2*10^5$MPA=$2*10^5$N/mm^2 α= Coefficient of linear expansion =$12*10^{-6}/^0C$ ΔT= Change in temperature=50^0C **To find-** End reactions due to rise in temperature i.e. force **Solution-** We know that Temperature stress =Eα ΔT = $2*10^5*12*10^{-6}*50$ **= 120N/mm^2** (Compressive in nature) Now, Reaction at the end due to rise in temperature P=σ*A =120*100=**12000N**	 1 marks 1 Marks 2 Marks
Q.4 b] i]	**Moment of Inertia** It is defined as the algebraic sum of the product of area and the square of its distance from the fixed axis. It is also called as the second moment of area. It is denoted by "I" **I=$\sum$Ah^2** Where A=Area of the cross section h=distance of the centroid of area from the axis to be considered Unit- mm^4, m^4. 	 1 Marks 1 Mark
b]-ii]	**Radius of Gyration** It is defined as the distance at which area "A" is supposed to be concentrated to give the same moment of inertia. It is denoted by 'K'. **I=Ak^2** $\boldsymbol{K = \sqrt{\frac{I}{A}}}$ Where I=M.I. about the axis to be considered. A= Area of the section. K= Radius of gyration Unit-mm, cm, m	 1 Mark 1 Mark

Q.4 c]	**Given –** for the right angle triangle Base=b=140mm Height=h=100mm  **To find-** 1) Moment of inertia about the base. 2) Moment of inertia about the axis passing centroid. **Solution-** 1) Moment of inertia about the base $I\ base = \frac{bh3}{12}$ = $(140* 100^3)/12$ **I = $11.67*10^6 mm^4$** 2) Moment of inertia about the axis passing centroid $I = \frac{bh3}{36}$ I= $(140*100^3)/36$ **I = $3.89*10^6\ mm^4$**	1 Marks 1 Marks 1 Marks 1 Marks
Q.4 d]	**Given-** for the symmetrical I section Dimensions- flanges=100mm*10mm, Web= 10mm*100mm  	1 Mark

B= Width of the rectangle ABCD=100mm
H= Height of the rectangle ABCD=120mm
b= width of the shaded rectangle= (100-10) =90mm
h= height of the shaded rectangle= (120-20) = 100mm

To find=Polar moment of inertia of the section.

Solution- as per the given dimensions in order to find the polar moment of inertia we have to use the perpendicular axis theorem

$I_{ZZ}= I_{XX}+I_{YY}$

I_{XX}= moment of inertia about the X axis

$$IXX = \left(\frac{BH3}{12}\right) - \left(\frac{bh3}{12}\right)$$

$=[(100*120^3)/12]-[(90*100^3)/12]$ — 1 Mark

$\mathbf{I_{XX}=6.9*10^6\ mm^4}$

Flanges and web are symmetrical about the Y axis so no need to apply the parallel axis theorem

I_{YY}= (2* moment of inertia of the flanges) + M.I. of the web — 1 Mark

$= \{2[(10*100^3)/12)] + [(100*10^3)/12]\}$

$\mathbf{I_{YY}= 1.675*10^6\ mm^4}$

Now, let's find the polar moment of inertia i.e. I_{ZZ}

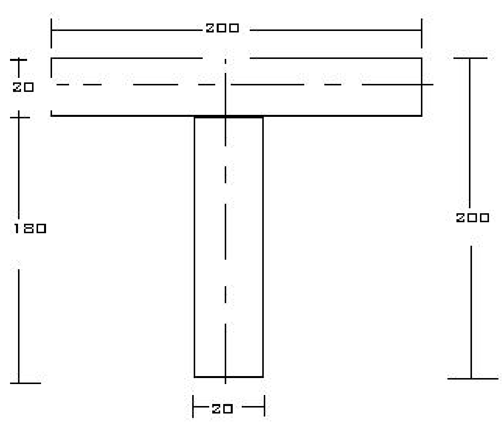
$I_{ZZ}= I_{XX}+I_{YY}$ — 1marks

$= (6.9*10^6) + (1.675*10^6)$

$\mathbf{I_{ZZ} = 8.575*10^6 mm^4}$

Q.4 e]

Given- for the T section
Dimensions-200mm*200mm*20mm



To find- M.I. about the centriodal axis i.e. I_{XX} & I_{YY}

Solution- 1) MI about the X axis

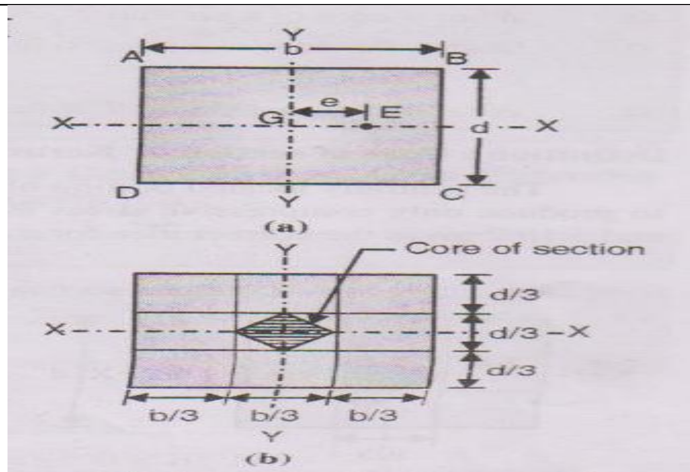
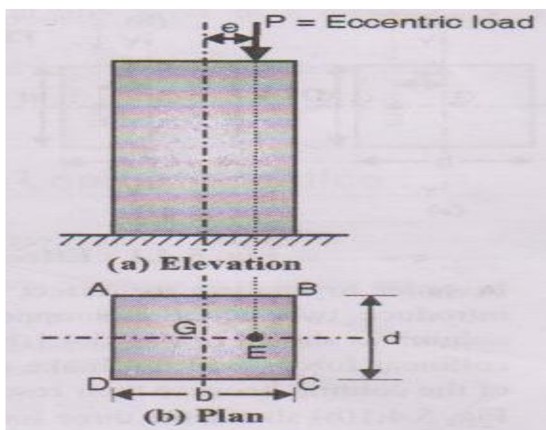
$I_{XX}=I_{XX1}+I_{XX2}$

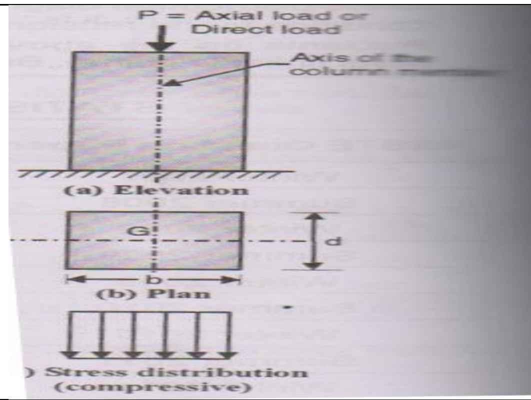
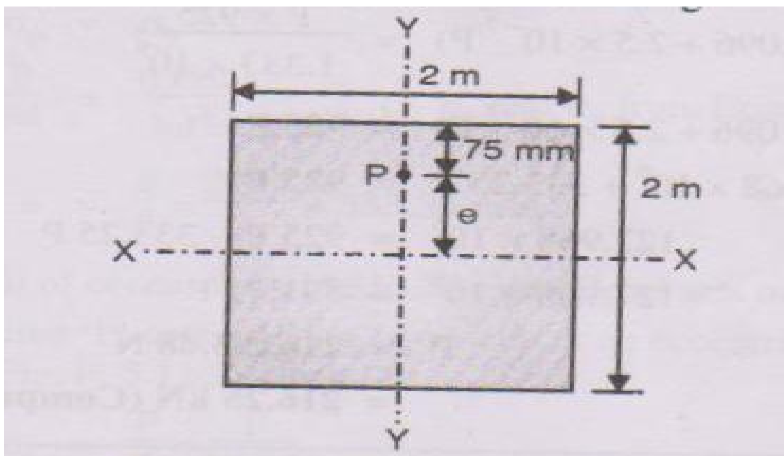
$= \{(I_{G1}+A_1h_1^2) + (I_{G2}+A_2h_2^2)\}$

$I_{G1}= (b_1d_1^3)/12= (200*20^3)/12$

$\mathbf{I_{G1} =133.33*10^3 mm^4}$ — 1 mark

	$I_{G2}= (b_2d_2^3)/12= (20*180^3)/12$ $\mathbf{I_{G2}=9 .72*10^6 mm^4}$ $A_1=200*20=4000mm^2$ $A_2=180*20=3600mm^4$ Let's find $\bar{X}$ & $\bar{Y}$ $\bar{X} =x_1=x_2= (200/2) = 100mm$…due to symmetry $y_1=180+ (20/2) = 190mm$ $y_2= (180/2)= 90mm$ $Y = (A_1y_1+A_2y_2)/(A_1+A_2)$ $= \{[(4000*190) + (3600*90)]/ (4000+3600)\}$ $= \{[(760*10^3) + (324*10^3)]/ (7600)\}$ $\mathbf{= 142.63mm}$ $h_1=190-142.63$ $\mathbf{=47.36mm}$ $h_2=142.43-90$ $\mathbf{=52.63mm}$ $I_{xx}=\{(I_{G1}+A_1h_1^2) + (I_{G2}+A_2h_2^2)\}$ $=\{[(133.33*10^3) + (4000*47.3684^2)] +[(9.72*10^6)+(3600*52.63^2)]\}$ $= \{[(133.33*10^3) + (8.957*10^{6)}] + [(9.72*10^6) + (9.971*10^6)\}$ $= (9.1083*10^6) + (19.691*10^6)$ $\mathbf{I_{XX} = 28.7993*10^6 mm^4}$ **To find the MI about the y axis** $I_{YY}=I_{YY1}+I_{YY2}$ $= \{(I_{G1}+A_1h_1^2) + (I_{G2}+A_2h_2^2)\}$ Due to symmetry about Y axis $h_1=h_2=0$ $I_{YY=} (I_{G1}+I_{G2})$ $=\{[(20*200^3)/12]+[(180*20^3)/12]\}$ $= [(13.33*10^6) +(120*10^3]$ $\mathbf{I_{YY} =13.45*10^6 mm^4}$	1 mark 1 Mark 1 Mark 1 Mark
Q.5 b] i]	**Limit of eccentricity** It is the limit at which the stress is purely compressive in nature. It can be found out by using middle third rule of eccentricity.	1marks 1marks

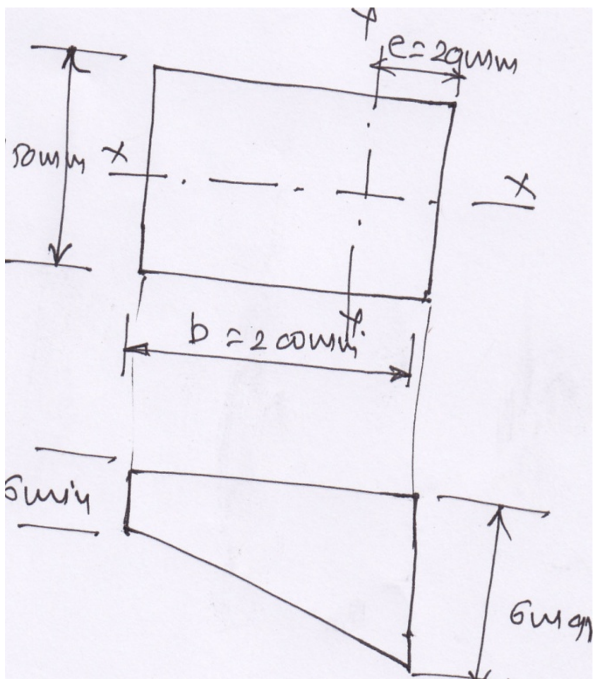
		
Q.5 b] ii]	**Eccentric load** If the load is acting at the eccentric distance that is at a certain distance from the centriodal axis, then it is called as the eccentric load. For the eccentric loading the load axis doesn't coincide with the axis of the member. Due to eccentric load the combined stresses are induced in the member.  **Axial load-** It is defined as the load whose line of action coincides with the axis of the member, it is also called as the direct load, due to axial load only direct stress induced in the member.	1 Mark 1 Mark

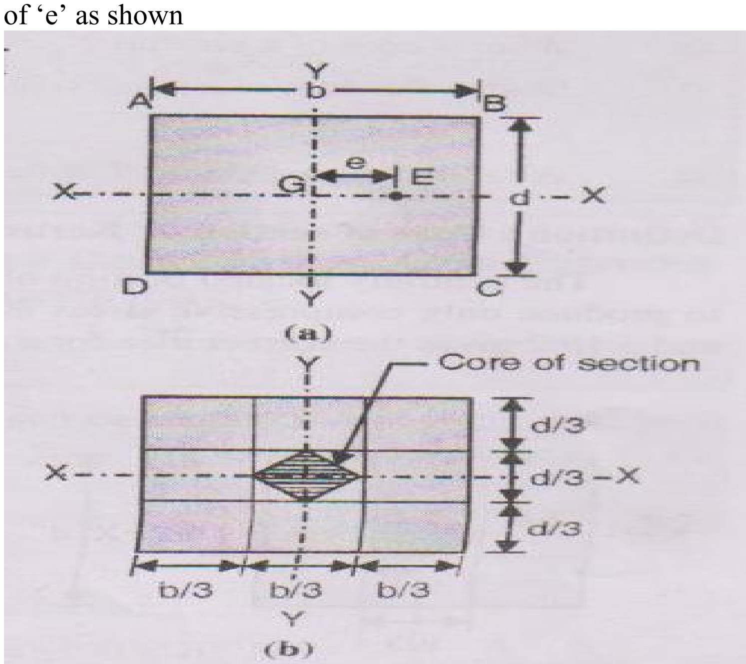
<table>
<tr><td></td><td>
</td><td></td></tr>
<tr><td>Q.5
c]</td><td>Given- for the rectangular column


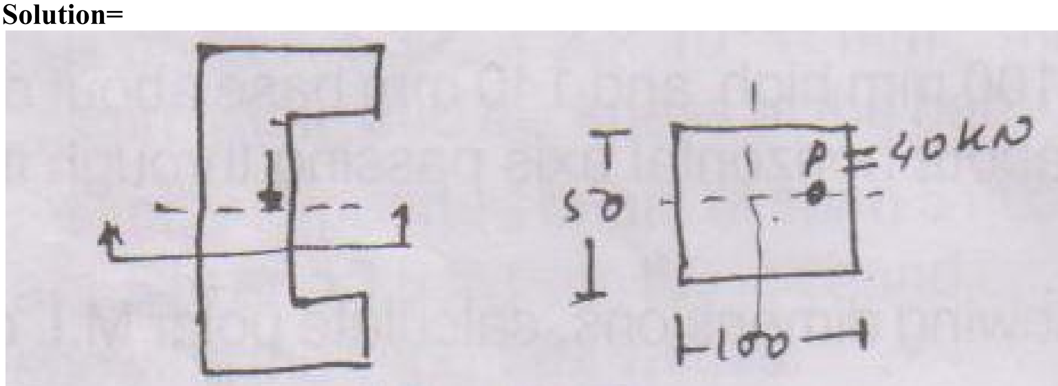
b= width=2000mm
d=depth=2000mm
e=eccentricity
=1000-75=925mm
W=weight= 385kN

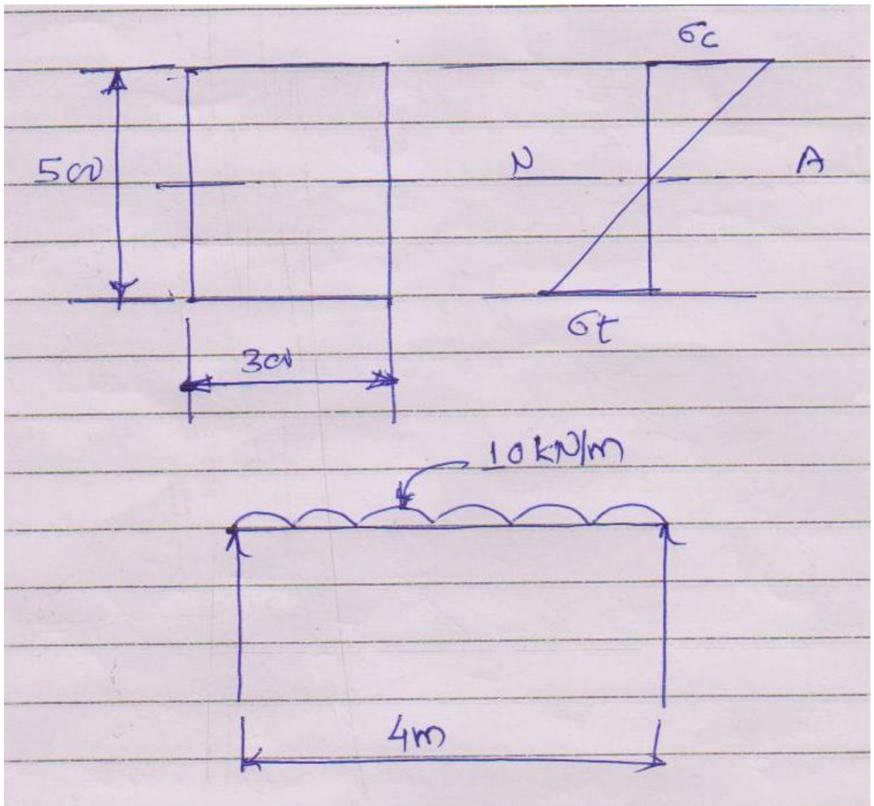
To find- the value of eccentric load for the no tension condition
Solution- for the no tension condition
$\sigma_0=\sigma_b$
σ_0= total direct stress
=direct stress due to own weight+ direct stress due to compression
= (W/A)+(P/A)
A= Area of the section
= (2000*2000)
$=4*10^6 mm^2$
σ_0= (W/A)+(P/A)
$=\{[(385*10^5)/(4*10^6)]+[(P)/(4*10^6)$
$=0.096+2.5*10^{-7}P$
σ_b= Bending stress
$=M/Z_{XX}$
M= P*e=925P</td><td>1 Mark

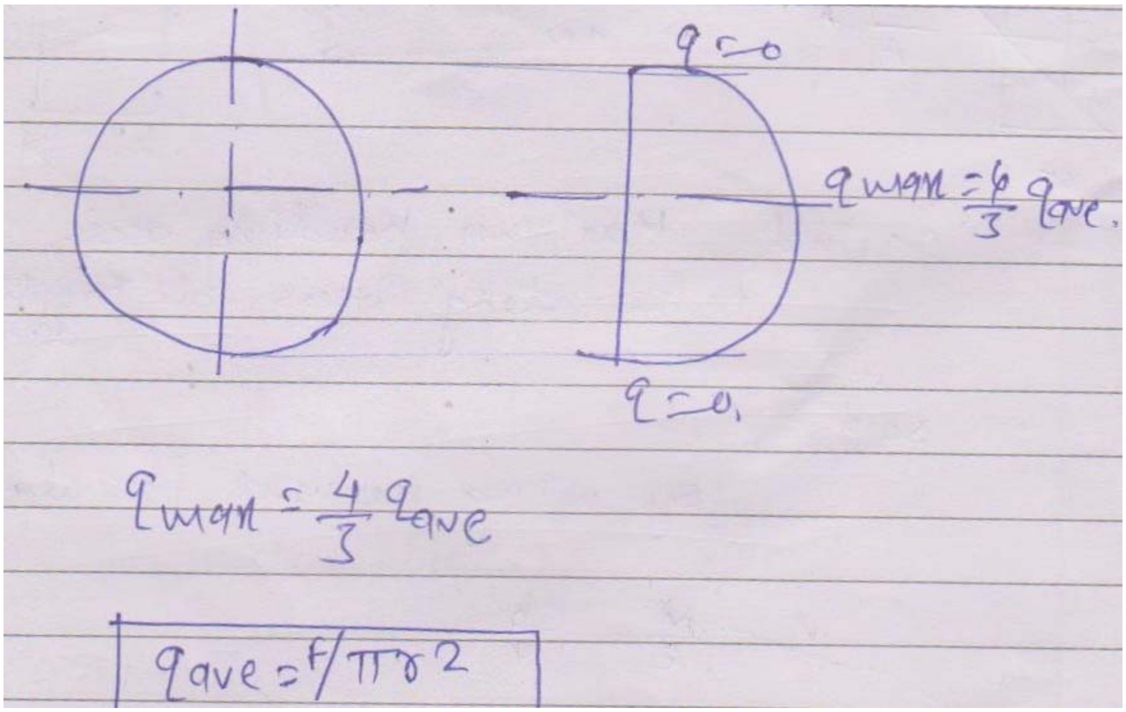
1 Mark</td></tr>
</table>

	Zxx= section modulus about the X axis $=(bd^2)/6$ $=(2000*2000^2)/6$ $Z_{xx} = 1.333*10^9\ mm^3$ $\sigma_b= (925*P)/(1.333*10^9)$ $\boldsymbol{\sigma_0=\sigma_b}$ $(0.096+2.5*10^{-7}P) =(925*P)/(1.333*10^9)$ $(1.333*10^9)*(0.096+2.5*10^{-7}P=925*P$ **P=216.25 kN compressive in nature**	] 1 Marks 1 Marks
Q.5 d]	**Given-** for a column section w=width=200mm d=t=thickness=150mm $P=200kN=200*10^3N$,e=eccentricity=20mm To find= maximum and minimum stress. **Solution=**  $\boldsymbol{\sigma_{MAX}=\sigma_0+\sigma_B}$ σ_0= Direct stress =P/A $=(200*10^3)/(200*150)$ $\mathbf{= 6.66N/mm^2}$ M= Bending moment about the Y axis $=P*e=200*10^3*20$ $\mathbf{= 4000*10^3N\text{-}mm}$ Z= Section modulus about the Y axis $=db^2/6$ $=(150*200^2)/(6)$ $\mathbf{=1*10^6\ mm^3}$	1 Mark 2 Marks

	$\sigma_{B=}$ bending stress =M/Z =$(4000*10^3 N)/(1*10^6)$ **$\sigma_B = 4N/mm^2$** $\sigma_{MAX=}\sigma_0+\sigma_B$ **σ_{MAX} =6.66+4=10.66N/mm^2** $\sigma_{Mini=}\sigma_0-\sigma_B$ **σ_{Min}= 6.66-4=2.66N/mm^2**	 1 Mark
Q.5 e]	**Core of the section** The centrally located portion of a section within the load line falls so as to produce only compressive stress; **it is called as the core of the section**. It is also known as the Kernel of the section. By using the middle third rule of eccentricity, we can find out the core of the rectangular section, we can find out the limit of eccentricity so as to only produce the compressive stresses. Let us consider the rectangular section with the width 'b' and the thickness 'd' at the eccentric distance of 'e' as shown  If the load P is acting on the X axis such that it bisect the X axis i.e. bisecting the thickness at the eccentricity of 'e'. σ_0= direct stress=P/A = P/(bd) σ_B= bending stress =$M/Z_{yy}=(P*e)/(db^2/6)$ =$(6P*e)/(db^2)$	1 Mark 1 Mark 1 Mark

	For the no tension condition $\sigma_0=\sigma_B$ $P/(bd)=(6P*e)/(db^2)$ e= b/6 Similarly if the load is acting on the Y axis bisecting the width then we can say that e=d/6 Thus "e" will be at the distance of b/6 to the left as well as the right of y axis and at a distance of d/6 upward as well as downward with respect to the X axis. The formed with the (b/3) and (d/3) is diagonals as which is situated at the middle third portion of the section of the rectangle giving the core of the section.	1 Mark
Q.5 f]	**Given =**for the rectangular rod bent into the C section with the dimensions as b=width=100mm, d=thickness=50mm P=load acting=40kN=$40*10^3$N Acing on the y axis. e= eccentricity=40mm **To find-** resultant stresses developed at X-X section. **Solution=**  σ_0= direct stress=P/A $= (40*10^3)/(50*100)$ **= 8 N/mm²** σ_B= bending stress=(My)/I M= Moment=P*e= $40*10^3*40$ **= $1.6*10^6$ N-mm** $I=(db^2)/6=(50*100^2)/(6)$ **= $4.17*10^6$N/mm²** y = 100/2 **=50mm** $\sigma_B=(My)/I=(1.6*10^6*50)/(4.17*10^6)$ **= 19.18 N/mm²** As the rectangular rod is bent into the 'C' section, the tensile stress is considered as positive while the compressive stress is considered as negative. Resultant stress developed is $\sigma_{max}=\sigma_0+\sigma_B$ = 8+19.18=**27.18N/mm²** $\sigma_{mini}=\sigma_0-\sigma_B$ = 8-19.18=**-11.18N/mm²**	1 Mark 2 Mark 1 Mark

Q.6 a]	**Given=** for a simply supported beam of rectangular section b=width of the section=300mm d=depth =500mm ,y=500/2=250mm L=Span of the beam=4m=4000mm w=UDL acting over the entire span=10kN/m  	1 mark
	To find=1) maximum bending stress **To draw-** 1) Bending stress distribution diagram **Solution=** by using the flexural formulae $(M/I)=(\sigma_b/y)$ $M_{max}= (wL^2)/8$ ....udl on simply supported beam $=10*(4)^2/(8)$ $\mathbf{M_{max} =20*10^6 N.mm}$	1 mark
	I= MI of the section=$bd^3/12$ $=(300*500^3)/12$ $\mathbf{I= 3.125*10^9\ mm^4}$	1 mark
	$(M/I)=(\sigma_b/y)$ $(20*10^6)/(3.125*10^9)=(\sigma_b/250)$ $\mathbf{\sigma_b= 1.6 N/mm^2}$	1Mark

Q.6 b] i]	**Shear stress equation is given by** $q = (FA\bar{y})/(Ib)$ where , q= shear stress at any section in N/mm^2 A= area of the portion above the N-A in mm^2 $\bar{y}$ = distance of the centroid of the area considered above the N-A in mm. I= MI about the N-A in mm^4 b=width of the section above the N-A in mm. $A\bar{y}$= First moment of area.	1 marks 1 Marks
Q.6 b] ii]	Shear stress distribution diagram for the circular section  	1 Marks 1 Mark
Q.6 c]	**Assumptions made in the theory of torsion** 1. The shaft should be perfectly straight and uniform in cross section. 2. Material of the shaft is homogeneous and isotropic. 3. Circular shaft remains circular after twisting. 4. Plane section of the shaft remains plane before and after twisting. 5. Twist is uniform along the length of the shaft. 6. Maximum shear stress induced in the shaft does not exceed elastic limit. 7. Torque is applied on the shaft in the plane perpendicular to the axis of the shaft. 8. Shaft is acted upon by pure torsion. 9. Shear stress is proportional to the shear strain.	½ marks each.

Q.6 d]	**Given-** d= diameter of the shaft=40mm N= Speed in RPM=200 RPM τ= Shear stress= 85N/mm^2 **To find=**power to be transmitted **Solution=** The power transmitted by the shaft is given by P= (2ΠNT)/60 in Watts The strength of the shaft is given by T= (Π/16)*τ*d^3 = (Π/16)*85*40^3 T= 1.068*10^6 N-mm **T = 1.068*10^3 N-m** P= (2Π*200*1.068*10^3)/60 P = 22.36*10^3 Watt **P =22.36 kW.**	1 Marks 1 Marks 2 Marks
Q.6 e]	**Given=** Power (P)=200HP =200*735.751 Watts P =147.15kW = 147.15*10^3 Watts N=180 RPM τ=80 N/mm^2 Ø= 1^{0}C= (1*Π/180) = 0.01745 radians C= Modulus of rigidity=0.8* 10^{5}N/mm^2 L= length of the shaft=3m=3000mm **To select=** suitable diameter of the shaft **Solution=** 1) Diameter of the shaft on the basis of strength By Torsional formula (T/J)= (τ/R) T can be found out from the power given P= (2ΠNT)/60 147 .15*10^3= (2Π*180*T)/ (60) T= (7.8065*10^3) N-m **T=7.8065*10^6 N-mm** (7.8065*10^6)/(Π/32*D^4) = (80)/ (D/2) (7.8065*10^6)/ (Π/32*D^4) = (160)/ (D) (7.8065*10^6)/ (Π/32*D^3) = (160) **D=79.20mm** 2) Diameter of the shaft on the basis of angle of twist **(T/J)= (GØ/L)** (7.8065*10^6)/ (Π/32*D^4) = (0.82*10^5*0.01745)/ (3000) D^4=166.71*10^6 **D= 113.62mm** **Selecting the larger diameter of two, So the suitable diameter of shaft is113.62mm**	1 mark 1 mark 1Marks 1Marks

Q.6 f]	**Torsional formula** $\left(\frac{T}{J}\right) = \left(\frac{G\text{Ø}}{L}\right) = \left(\frac{\tau}{R}\right)$ Where, T= twisting moment in N-mm J= polar MI=I_{XX}+I_{YY} in mm^4 G= Modulus of rigidity in N/mm^2.('C' also be used) Ø= angle of twist in radian, τ= maximum Shear stress, R=radius of the shaft in mm L= length of the shaft in mm	2 mark 2Marks