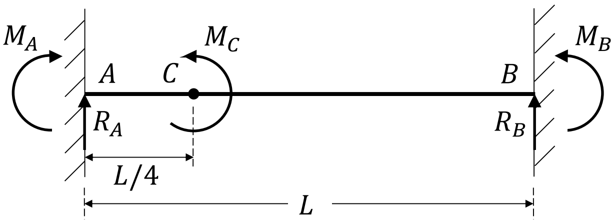
2015-2016 MM2MS3 Exam Solutions

1.

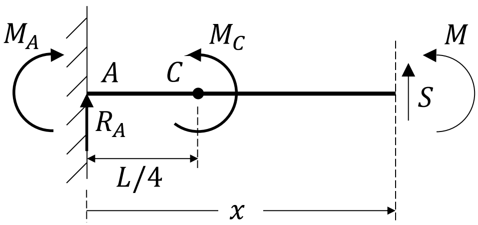
(a)

Drawing a free body diagram of the beam:



[2 marks]

Take origin from right hand side and sectioning the beam after the last discontinuity:



[1 mark]

Taking moments about the section position (remembering to use Macaulay brackets where needed) gives:

$$M + M_C \langle x - \frac{L}{4} \rangle^0 = R_A x + M_A$$

$$\therefore M = R_A x + M_A - M_C \langle x - \frac{L}{4} \rangle^0$$

[1 mark]

Substituting this in the main deflection of beams equation gives:

$$EI \frac{d^2y}{dx^2} = M = R_A x + M_A - M_C \langle x - \frac{L}{4} \rangle^0$$

[1 mark]

Integrating gives:

$$EI \frac{dy}{dx} = \frac{R_A x^2}{2} + M_A x - M_C \langle x - \frac{L}{4} \rangle + A \qquad (1)$$

[1 mark]

Integrating again gives:

$$EIy = \frac{R_A x^3}{6} + \frac{M_A x^2}{2} - \frac{M_C \langle x - \frac{L}{4} \rangle^2}{2} + Ax + B \qquad (2)$$

[1 mark]

Boundary conditions:

(BC1) At $x = 0$, $\frac{dy}{dx} = 0$, therefore from (1):

$$A = 0$$

[1 mark]

(BC2) At $x = 0$, y = 0, therefore from (2):

$$B = 0$$

[1 mark]

(BC3) At $x = L$, $\frac{dy}{dx} = 0$, therefore from (1):

$$0 = \frac{R_A L^2}{2} + M_A L - M_C \langle L - \frac{L}{4} \rangle \qquad (3)$$

[1 mark]

(BC4) At $x = L$, y = 0, therefore from (2):

$$0 = \frac{R_A L^3}{6} + \frac{M_A L^2}{2} - \frac{M_C \langle L - \frac{L}{4} \rangle^2}{2} \qquad (4)$$

[1 mark]

Using simultaneous equations (3) and (4) to solve for R_A and M_A gives:

$$\boldsymbol{R_A = \frac{9M_C}{8L}} \qquad (5)$$

and:

$$\boldsymbol{M_A = \frac{3M_C}{16}} \qquad (6)$$

[2 marks]

From vertical equilibrium:

$$R_A + R_B = 0$$

Therefore, substituting (5) into this gives:

$$\boldsymbol{R_B = -\frac{9M_C}{8L}}$$

[1 mark]

Taking moments about position B gives:

$$M_A + R_A L = M_B + M_C$$

Substituting (5) and (6) into this gives:

$$\boldsymbol{M_B = \frac{5M_C}{16}}$$

[1 mark]

(b)

From (2), at $x = \frac{L}{4}$ (point C):

$$y = \frac{1}{EI}\left(\frac{R_A\left(\frac{L}{4}\right)^3}{6} + \frac{M_A\left(\frac{L}{4}\right)^2}{2}\right)$$

[1 mark]

Substituting (5) and (6) into this gives:

$$y = \frac{1}{EI}\left(\frac{9M_C L^2}{3072} + \frac{3M_C L^2}{512}\right) = \frac{1}{EI}\left(\frac{27M_C L^2}{3072}\right) \qquad (7)$$

where,

$$M_C = 2\left(F \times \frac{d}{2}\right) = Fd$$

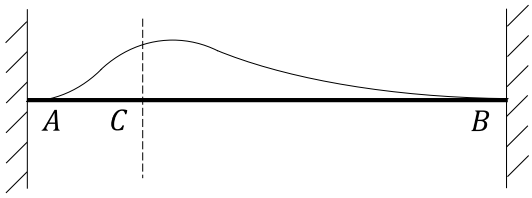
[2 marks]

Substituting this into (7) gives:

$$\boldsymbol{y = \frac{27FdL^2}{3072EI}}$$

[2 marks]

(c)

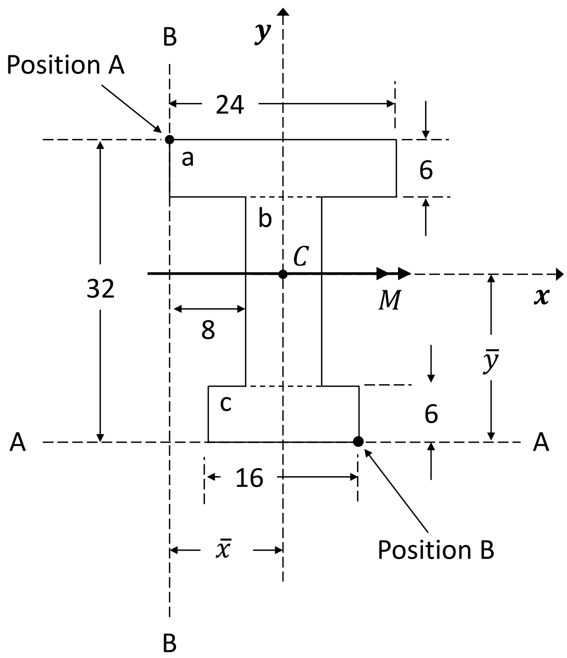


[5 marks]

2.

(a)

Position of Centroid, C



Total area,

$$A = (24 \times 6)_a + (8 \times 20)_b + (16 \times 6)_c = 400 \text{ mm}^4$$

[1 mark]

Taking moments about AA:

$$\bar{y} = \frac{(24 \times 6 \times 29)_a + (8 \times 20 \times 16)_b + (16 \times 6 \times 3)_c}{400} = \mathbf{17.56\ mm}$$

[2 marks]

Similarly, taking moments about BB:

$$\bar{x} = \frac{(6 \times 24 \times 12)_a + (20 \times 8 \times 12)_b + (6 \times 16 \times 12)_c}{400} = \mathbf{12\ mm}$$

[2 marks]

(b)

2nd Moments of Area and Product Moment of Area about the $x - y$ axes through C

Therefore, using the Parallel Axis Theorem,

$$I_{x'} = (I_x + Ab^2)_a + (I_x + Ab^2)_b + (I_x + Ab^2)_c$$

$$= \left(\frac{24 \times 6^3}{12} + 24 \times 6 \times (29 - 17.56)^2\right) + \left(\frac{8 \times 20^3}{12} + 8 \times 20 \times (16 - 17.56)^2\right)$$

$$+ \left(\frac{16 \times 6^3}{12} + 16 \times 6 \times (3 - 17.56)^2\right)$$

$$= \mathbf{45,639.9\ mm^4}$$

[2 marks]

and,

$$I_{y'} = \left(I_y + Aa^2\right)_a + \left(I_y + Aa^2\right)_b + \left(I_y + Aa^2\right)_c$$

$$= \left(\frac{6 \times 24^3}{12} + 6 \times 24 \times (12 - 12)^2\right) + \left(\frac{20 \times 8^3}{12} + 20 \times 8 \times (12 - 12)^2\right)$$

$$+ \left(\frac{6 \times 16^3}{12} + 6 \times 16 \times (12 - 12)^2\right)$$

$$= 6912 + 853.33 + 2048$$

$$= \mathbf{9,813.33\ mm^4}$$

[2 marks]

Also,

$$I_{x'y'} = \left(I_{xy} + Aab\right)_a + \left(I_{xy} + Aab\right)_b + \left(I_{xy} + Aab\right)_c$$

$$= \left(0 + 24 \times 6 \times (12 - 12) \times (29 - 17.65)\right) + \left(0 + 8 \times 20 \times (12 - 12) \times (16 - 17.56)\right)$$

$$+\left(0 + 16 \times 6 \times (12 - 12) \times (3 - 17.56)\right)$$

$$= \mathbf{0\ mm^4}$$

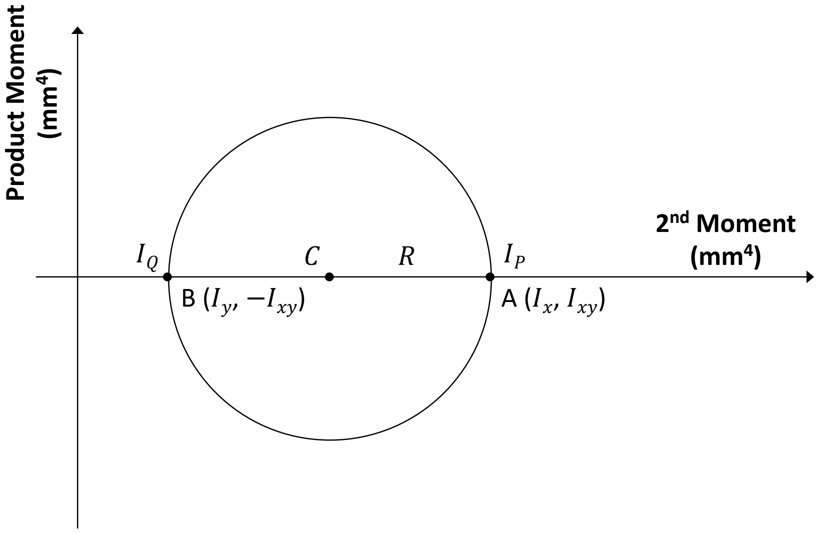
[1 mark]

(c)

Principal Second Moments of Area, direction of the Principal Axes and direction of the Neutral Axis

(By calculation)

Mohr's Circle



$$Centre, C = \frac{I_{x'} + I_{y'}}{2} = \frac{45{,}639.9 + 9{,}813.33}{2} = 27{,}726.17 \text{ mm}^4$$

$$Radius, R = \sqrt{\left(\frac{I_{x'} - I_{y'}}{2}\right)^2 + I_{x'y'}{}^2} = \sqrt{\left(\frac{45{,}639.9 - 9{,}813.33}{2}\right)^2 + 0^2} = 17{,}913.29 \text{ mm}^4$$

[3 marks]

Therefore, the Principal 2nd Moments of Area are:

$$I_P = C + R = 27{,}726.17 + 17{,}913.29 = \mathbf{45,639.46\ mm^4}$$

and,

$$I_Q = C - R = 27{,}726.17 - 17{,}913.29 = \mathbf{9,812.88\ mm^4}$$

[1 mark]

Directions of the Principal Axes

From the Mohr's circle above:

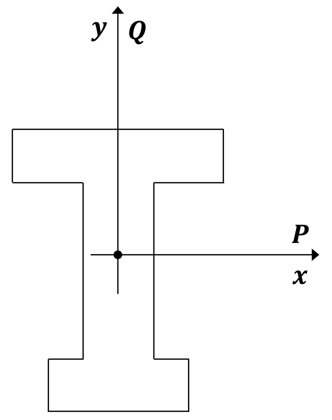
$$sin2\theta = \frac{I_{xy}}{R} = \frac{0}{17{,}913.29}$$

$$\therefore \theta = 0°$$

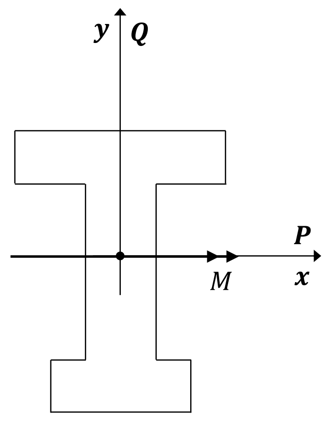
(this can also be seen from observation of the Mohr's Circle, in this case)

[1 mark]

Therefore, the Principal Axes are at 0° from the $x - y$ axes, as shown on the diagram below.



There is an applied Bending Moment, M, of 225Nm about the x-axis as shown below,



Resolve applied bending moment onto Principal Axes

As the applied Bending Moment is along the x-axis, and therefore the P-axis,

$$M_P = M = 325{,}000 \text{ Nmm}$$

and,

$$M_Q = 0 \text{ Nmm}$$

[1 mark]

Calculation of position of Neutral Axis

$$\sigma_b = \frac{M_P Q}{I_P} - \frac{M_Q P}{I_Q}$$

At the Neutral Axis, $\sigma_b = 0$, therefore,

$$\frac{M_P Q}{I_P} - \frac{M_Q P}{I_Q} = 0$$

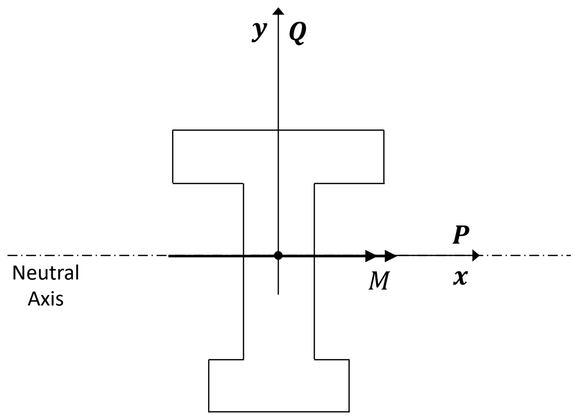
$$\therefore \frac{Q}{P} = \frac{M_Q I_P}{M_P I_Q}$$

Therefore, α, the angle between the neutral axis and the principal axes can be defined as,

$$\alpha = \tan^{-1}\left(\frac{Q}{P}\right) = \tan^{-1}\left(\frac{M_Q I_P}{M_P I_Q}\right) = \tan^{-1}\left(\frac{0 \times 45{,}639.46}{225{,}000 \times 9{,}812.88}\right) = \mathbf{0°}$$

[2 marks]

Therefore, the neutral axis is at 0° from the principal axes as shown below,



The neutral axis is therefore at (0° - 0° =) 0° from the x-axis.

[2 marks]

(By observation)

As this section is symmetrical about the y-axis, it can be seen that,

$$\boldsymbol{I_P = I_{x'} = 45,639.46 \text{ mm}^4}$$

and,

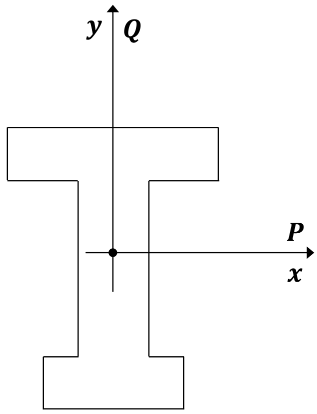
$$\boldsymbol{I_Q = I_{y'} = 9,812.88\ \mathrm{mm}^4}$$

[4 marks]

Also, since $I_{x'y'} = 0$

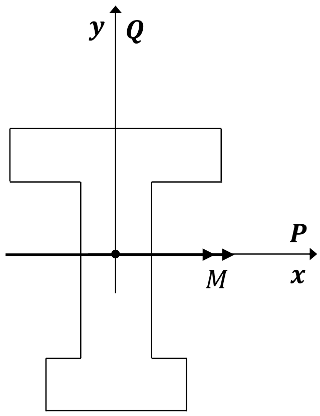
$$\theta = 0°$$

Therefore, the Principal Axes are at 0° from the $x - y$ axes, as shown on the diagram below.



[1 mark]

There is an applied Bending Moment, M, of 325 Nm about the x-axis as shown below,



Resolve applied Bending Moment onto Principal Axes

As the applied Bending Moment is along the x-axis, and therefore the P-axis, it can be seen by observation that,

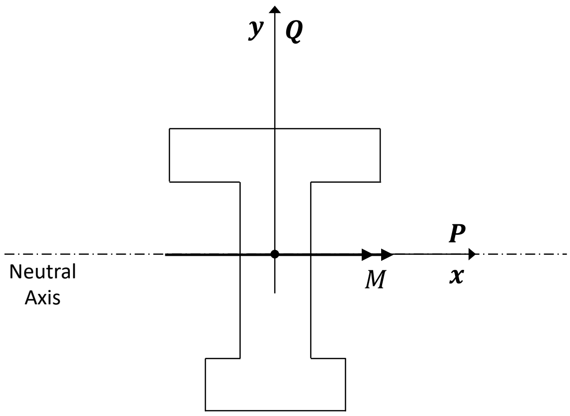
$$M_P = M = 325{,}000\ \mathrm{Nmm}$$

and,

$$M_Q = 0 \text{ Nmm}$$

[1 mark]

Again, as the section is symmetrical about the y-axis, the angle between the $P - Q$ axis (and therefore also the $x - y$ axis) and the Neutral Axis, $\boldsymbol{\alpha}$, **must be at 0°**, as shown below,



The neutral axis is therefore at (0° - 0° =) 0° from the x-axis.

[4 marks]

(d)

Stresses at positions A and B

As above,

$$\sigma_b = \frac{M_P Q}{I_P} - \frac{M_Q P}{I_Q}$$

Therefore, the co-ordinates of point A on the $P - Q$ axes are required. As the $P - Q$ axis lies on the $x - y$ axis, the P co-ordinates are equal to the x co-ordinates and the Q co-ordinates are equal to the y co-ordinates.

[1 mark]

Therefore, for point A, $P = x = -12$ mm and $Q = y = 14.44$ mm. Therefore, these $P - Q$ co-ordinates for position A can now be substituted into the equation for bending stress to give:

$$\sigma_{bA} = \frac{M_P Q}{I_P} - \frac{M_Q P}{I_Q} = \frac{325{,}000 \times 14.44}{45{,}639.46} - \frac{0 \times -12}{9{,}812.88}$$

$$\therefore \boldsymbol{\sigma_{bA} = 102.83 \text{ MPa}}$$

[2 marks]

And for point B, $P = x = 12\ \mathrm{mm}$ and $Q = y = -17.56\ \mathrm{mm}$. Therefore, these $P - Q$ co-ordinates for position A can now be substituted into the equation for bending stress to give:

$$\sigma_{bB} = \frac{M_P Q}{I_P} - \frac{M_Q P}{I_Q} = \frac{325{,}000 \times -17.56}{45{,}639.46} - \frac{0 \times 12}{9{,}812.88}$$

$$\therefore \boldsymbol{\sigma_{bB} = -125.05\ \mathrm{MPa}}$$

[2 marks]

3.

(a)

$$\sigma_z = \frac{R_i^2 p_i - R_{o^2} p_o}{(r_o^2 - r_i^2)}$$

[2 marks]

Substituting in values

$$\sigma_z = \frac{50^2 \times 140 - 100^2 \times 20^2}{(r_o^2 - r_i^2)} = \mathbf{20\ MPa}$$

[3 marks]

(b)

Lame's Equations

$$\sigma_r = A - \frac{B}{r^2}$$

$$\sigma_\theta = A + \frac{B}{r^2}$$

[2 marks]

Using radial equation and applying BCs

At ID:

$$\sigma_r = -140 = A - \frac{B}{50^2}$$

At OD:

$$\sigma_r = -20 = A - \frac{B}{100^2}$$

[2 marks]

Subtract ID from OD

$$-20 + 140 = 120 = -\frac{B}{100^2} + \frac{B}{50^2}$$

$$\therefore 120 = B(\frac{1}{2500} - \frac{1}{10000})$$

$$\therefore B = \frac{120}{0.0003} = 400000$$

[3 marks]

Therefore, from OD:

$$A = -20 + \frac{400000}{10000} = 20$$

[2 marks]

Calculating the hoop stress value at the ID:

$$\sigma_\theta = 20 + \frac{400000}{50^2} = \mathbf{180\ MPa}$$

[3 marks]

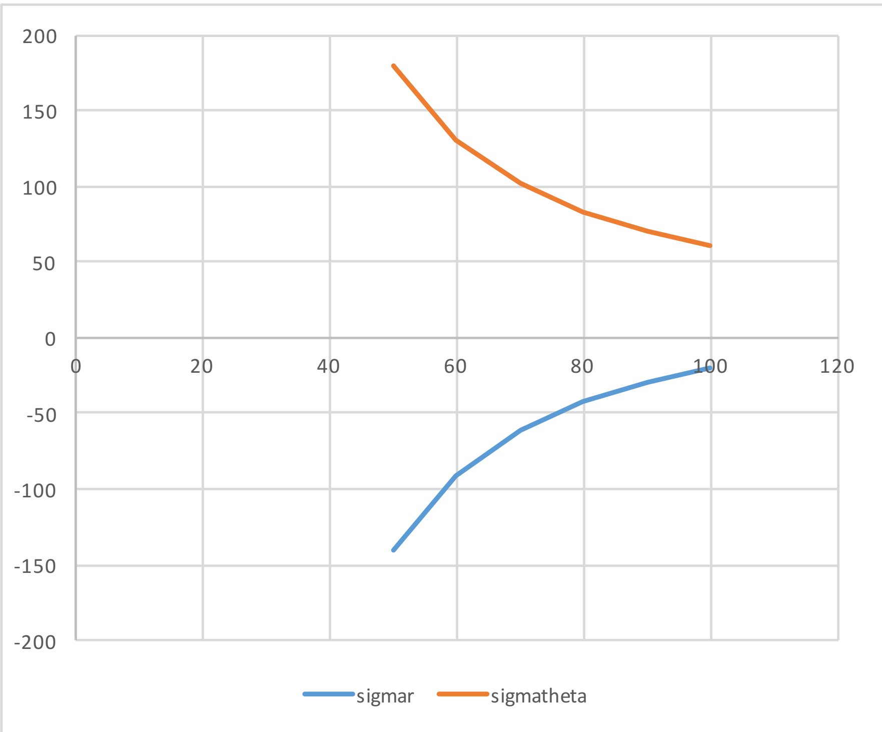
and at the OD:

$$\sigma_\theta = 20 + \frac{400000}{100^2} = \mathbf{60\ MPa}$$

[3 marks]

(c)

Distributions of hoop and radial stress with r



[5 marks]

4.

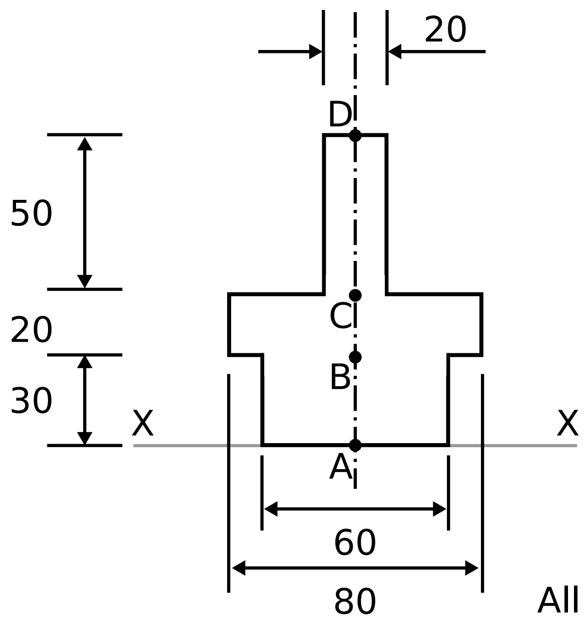
The section shown in Figure Q4 carries a shear force, $S = 45$ kN down the vertical centre line.

(a)

Using first moment of area:

$$\bar{y} = \frac{\sum A\bar{y}_n}{A_T}$$

[1 mark]



	Width	Length	$\bar{y}_n$	A	$A\bar{y}_n$
A	20	50	75	1000	75000
B	80	20	40	1600	64000
C	60	30	15	1800	27000
			Sum:	4400	166000
				$\bar{y}$	**37.7**

[3 marks]

(b)

$$I = \sum\left(\frac{BD^3}{12} + A(\bar{y}_n - \bar{y})^2\right)$$

[2 marks]

$\bar{y}_n - \bar{y}$	$A(\bar{y}_n - \bar{y})^2$	$\frac{BD^3}{12}$	$A(\bar{y}_n - \bar{y})^2 + \frac{BD^3}{12}$
37.3	1389256.198	208333.3333	1597589.532
2.3	8264.46281	53333.33333	61597.79614
-22.7	929752.0661	135000	1064752.066
		I	**2723939.4**

[3 marks]

(c)

$$\tau = \frac{SQ}{Iz}$$

[1 mark]

A & D are free surfaces so:

$$\tau_A = \tau_D = 0$$

[1 mark]

Can use area below at B, 2 values due to section change:

$$\tau_{B1} = \frac{45000 \times (60 \times 30) \times (37.7 - 15)}{2723939.4 \times 60} = \mathbf{11.3\ MPa}$$

[2 marks]

$$\tau_{B2} = \frac{45000 \times (60 \times 30) \times (37.7 - 15)}{2723939.4 \times 80} = \mathbf{8.4\ MPa}$$

[2 marks]

At G (centroid), need to consider the effect of two areas:

In this case $Q = \sum A(y_n - y)$

$$Q = \left((20 \times 50) \times (75 - 37.7)\right) + \left(80 \times (50 - 37.7) \times \left(\frac{50 - 37.7}{2}\right)\right)$$

$$Q = 37272.8 + 6051.6 = 43324.4$$

and therefore:

$$\tau_G = \frac{45000 \times 43324.4}{2723939.4 \times 80} = \mathbf{8.9\ MPa}$$

[2 marks]

Two values of shear stress at C due to section change (we can use area above):

$$\tau_{C1} = \frac{45000 \times (20 \times 50) \times (75 - 37.7)}{2723939.4 \times 80} = \mathbf{7.7\ MPa}$$

[2 marks]

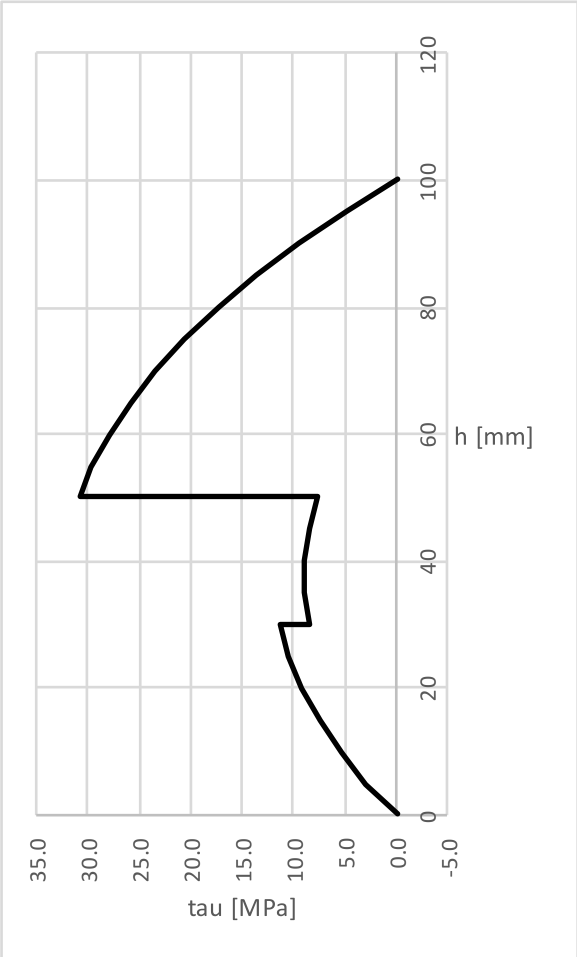
$$\tau_{C2} = \frac{45000 \times (20 \times 50) \times (75 - 37.7)}{2723939.4 \times 20} = \mathbf{30.8\ MPa}$$

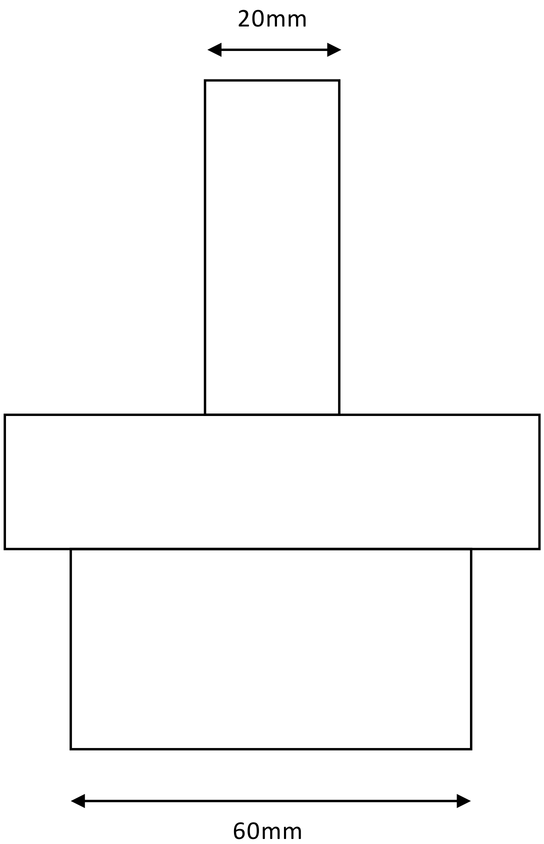
[2 marks]

(d)

h	τ	
0	**0.0**	**A**
5	2.9	
10	5.4	
15	7.5	
20	9.2	
25	10.4	
30	**11.3**	**B1**
30	**8.4**	**B1**
35	8.9	
37.7	**8.9**	**G**
40	8.9	
45	8.5	
50	**7.7**	**C1**
50	**30.8**	**C2**
55	29.6	
60	27.9	
65	25.9	
70	23.4	
75	20.6	
80	17.3	
85	13.6	
90	9.5	
95	4.9	
100	**0.0**	**D**

[2 marks]


120
100
80
60
40
20
0
h [mm]
35.0
30.0
25.0
20.0
15.0
10.0
5.0
0.0
-5.0
tau [MPa]

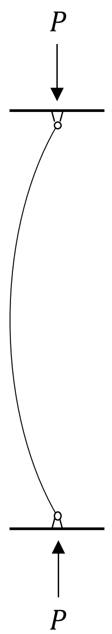

20mm
60mm

[2 marks]

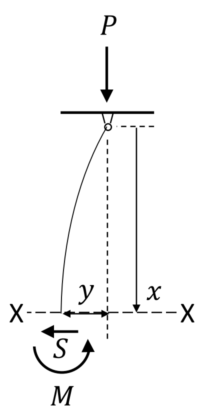
5.

(a)

Below is a diagrammatic representation of a pinned-pinned strut.



Sectioning this beam in order to determine the bending moment:



[1 mark]

Taking moments about the section position, X-X:

$$M = Py \qquad (1)$$

2nd order differential equation for a beam under bending:

$$EI\frac{d^2y}{dx^2} = M$$

Substituting (1) into this:

$$EI\frac{d^2y}{dx^2} = Py$$

[1 mark]

Let $y = A_O e^{\alpha x}$:

$$\therefore EI\alpha^2 + P = 0$$

and:

$$\alpha = \pm\sqrt{\frac{P}{EI}}i$$

We get:

$$y = Asin\left(\sqrt{\frac{P}{EI}}x\right) + Bcos\left(\sqrt{\frac{P}{EI}}x\right) \qquad (2)$$

where A_O, A and B are constants.

[1 mark]

Boundary conditions:

(BC1) At $x = 0$, $y = 0$, therefore from (2):

$$B = 0$$

[1 mark]

(BC2) At $x = L$, $y = 0$, therefore from (2):

$$Asin\left(\sqrt{\frac{P}{EI}}L\right) = 0$$

Since $A \neq 0$ for non-trivial solution:

$$\sqrt{\frac{P}{EI}}L = n\pi$$

[1 mark]

$$\therefore \boldsymbol{P} = \frac{\boldsymbol{n^2 \pi^2 EI}}{\boldsymbol{L^2}} \qquad (3)$$

where $n = 1, 2, \ldots$

[2 marks]

(b)

$$\sigma = \frac{P}{A}$$

[2 marks]

Substituting the buckling load (from (3)) into this equation:

$$\sigma = \frac{n^2 \pi^2 EI}{L^2 A} = \frac{n^2 \pi^2 EAk^2}{L^2 A} = \frac{\pi^2 E}{\left(\frac{L}{k}\right)^2}$$

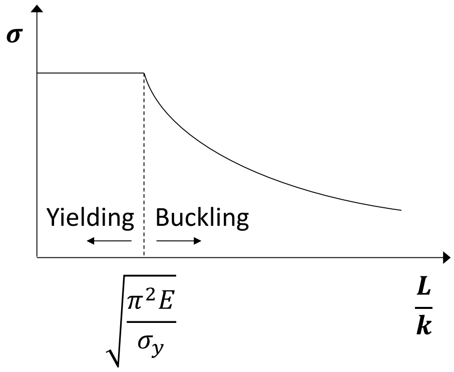
[2 marks]

Therefore when $\sigma = \sigma_y$:

$$\boldsymbol{\frac{L}{k} = \sqrt{\frac{\pi^2 E}{\sigma_y}}}$$

[2 marks]

The figure below therefore displays the transition between yielding and buckling in terms of slenderness ratio.



[4 marks]

(c)

$$\frac{L}{k} = \sqrt{\frac{\pi^2 E}{\sigma_y}} = \sqrt{\frac{\pi^2 \times 70{,}000}{230}} = 54.81 \qquad (4)$$

[2 marks]

where k is calculated by:

$$I = \frac{bd^3}{12} = \frac{t^4}{12} = t^2k^2$$

[2 marks]

where $b = d = t$ and $I = Ak^2 = bdk = t^2k^2$.

$$\therefore k = 8.66 \text{ mm}$$

[2 marks]

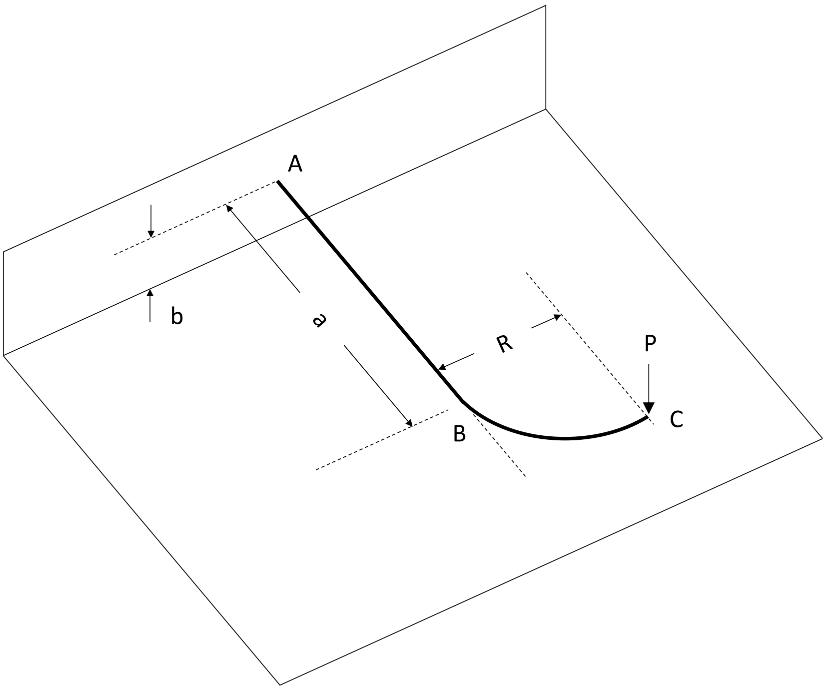
Therefore from (4):

$$\boldsymbol{L = 474.66\ \mathrm{mm}}$$

[2 marks]

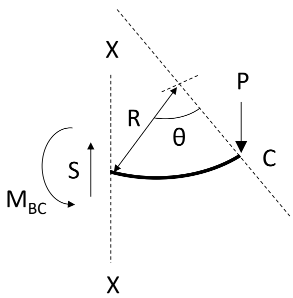
6.

Label bracket dimensions:



Section BC (*bending only*)

Free Body Diagram:



[2 marks]

Taking moments about X-X:

$$M_{AB} = PRsin\theta$$

[2 marks]

Substituting this into the equation for Strain Energy in a beam under bending gives,

$$U_{BC} = \int_0^x \frac{M_{AB}{}^2}{2EI} dx = \int_0^{\pi/2} \frac{(PRsin\theta)^2}{2EI} Rd\theta$$

where,

$$dx = Rd\theta$$

$$\therefore U_{BC} = \frac{P^2R^3}{2EI}\int_0^{\pi/2} sin^2\theta d\theta \qquad (1)$$

[2 marks]

Trigonometric Identities:

$$sin^2\theta + cos^2\theta = 1 \qquad (2)$$

and,

$$cos2\theta = cos^2\theta - sin^2\theta \qquad (3)$$

Rearranging equation (2) gives,

$$cos^2\theta = 1 - sin^2\theta$$

Substituting this into equation (3) gives,

$$cos2\theta = 1 - 2sin^2\theta$$

$$\therefore sin^2\theta = \frac{1}{2} - \frac{cos2\theta}{2}$$

[2 marks]

Substituting this into equation (1) gives,

$$U_{BC} = \frac{P^2R^3}{2EI}\int_0^{\pi/2}\left(\frac{1}{2} - \frac{cos2\theta}{2}\right)d\theta$$

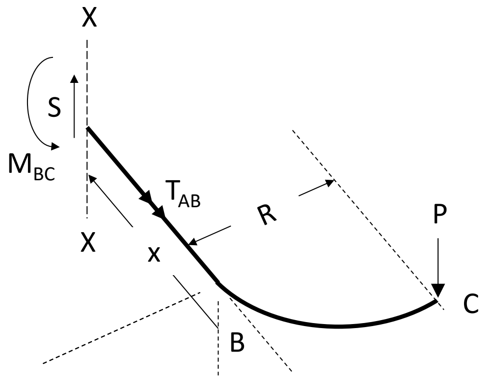
$$= \frac{P^2R^3}{2EI}\left[\frac{\theta}{2} - \frac{sin2\theta}{4}\right]_0^{\pi/2} = \frac{P^2R^3}{2EI}\left(\left(\frac{\pi}{4} - \frac{sin\pi}{4}\right) - \left(\frac{0}{2} - \frac{sin(0)}{4}\right)\right)$$

$$\therefore U_{BC} = \frac{P^2R^3\pi}{8EI}$$

[2 marks]

Section AB

Free Body Diagram:



[2 marks]

Bending

Taking moments about X-X:

$$M_{AB} = P(R + x)$$

[2 marks]

Substituting this into the equation for Strain Energy in a beam under bending gives,

$$U_{AB}^{bending} = \int_0^a \frac{{M_{AB}}^2}{2EI} dx = \int_0^a \frac{\big(P(R + x)\big)^2}{2EI} dx = \frac{P^2}{2EI} \int_0^a (R + x)^2 dx = \frac{P^2}{2EI} \int_0^a (R^2 + 2Rx + x^2) dx$$

$$= \frac{P^2}{2EI} \left[R^2 x + R x^2 + \frac{x^3}{3} \right]_0^a$$

$$\therefore U_{AB}^{bending} = \frac{P^2}{2EI} \left(R^2 a + R a^2 + \frac{a^3}{3} \right)$$

[2 marks]

Torsion

Taking moments about X-X:

$$T_{AB} = PR$$

[2 marks]

Substituting this into the equation for Strain Energy in a beam under bending gives,

$$U_{AB}^{torsion} = \int_0^a \frac{{T_{AB}}^2}{2GJ} dx = \int_0^a \frac{(PR)^2}{2GJ} dx = \frac{P^2 R^2}{2GJ} \int_0^a 1 dx = \frac{P^2 R^2}{2GJ} [x]_0^a = \frac{P^2 R^2}{2GJ} (a - 0)$$

$$\therefore U_{AB}^{torsion} = \frac{P^2R^2a}{2GJ}$$

[2 marks]

$$U_{AB} = U_{AB}^{bending} + U_{AB}^{torsion} = \frac{P^2}{2EI}\left(R^2a + Ra^2 + \frac{a^3}{3}\right) + \frac{P^2R^2a}{2GJ}$$

[1 mark]

Total Strain Energy

$$U = U_{AB} + U_{BC} = \frac{P^2}{2EI}\left(R^2a + Ra^2 + \frac{a^3}{3}\right) + \frac{P^2R^2a}{2GJ} + \frac{P^2R^3\pi}{8EI}$$

$$\therefore U = \frac{P^2}{2}\left(\frac{R^2a}{EI} + \frac{Ra^2}{EI} + \frac{a^3}{3EI} + \frac{R^2a}{GJ} + \frac{R^3\pi}{4EI}\right)$$

[1 mark]

Deflection Calculation using Castigliano's Theorem

$$u_{vA} = \frac{\delta U}{\delta P} = P\left(\frac{R^2a}{EI} + \frac{Ra^2}{EI} + \frac{a^3}{3EI} + \frac{R^2a}{GJ} + \frac{R^3\pi}{4EI}\right)$$

[1 mark]

where,

$$I = \frac{\pi d^4}{64} = \frac{\pi \times 28^4}{64} = 30171.86\ \mathrm{mm}^4$$

and,

$$J = \frac{\pi d^4}{32}(= 2I) = 60343.71\ \mathrm{mm}^4$$

$$\therefore u_{vA} = 500\left(\frac{500^2 \times 1500}{210000 \times 30171.86} + \frac{500 \times 1500^2}{210000 \times 30171.86} + \frac{1500^3}{3 \times 210000 \times 3.02 \times 10^{-8}} + \frac{500^2 \times 1500}{77000 \times 6.03 \times 10^{-8}} + \frac{500^3 \times \pi}{4 \times 210000 \times 3.02 \times 10^{-8}}\right)$$

$$= 257.5\ \mathrm{mm}$$

Therefore, as the calculated deflection at point C is more than 250 mm, **point C will make contact with the ground**.

[2 marks]