TAM 251 Example Multiple Choice Final Exam

- There are XX multiple choice questions, each one worth 1 point.
- There are X "written" questions, combined they are worth XX points.
- This is a 3 hour exam.
- You must not communicate with other students during this test.
- You can use a calculator.
- Close-book, close-note exam.
- Do not turn this page until instructed to do so.
- There are several different versions of this exam.

1. Fill in your information:

Full Name: ______________________________

Discussion Section: ______________________________

UIN (Student Number): ______________________________

NetID: ______________________________

2. Circle your discussion section:

	Tuesday	**Wednesday**	**Thursday**	**Friday**
12–1	ADC: Ouli and Shikhar ADD: Arif and Kallol			
1–2				ADF: Shikhar and Kallol
4–5	ADE: Ouli and Chad		ADB: Arif and Chad	
5–6			ADG: Arif and Chad	

3. Fill in the following answers on the Scantron form:

Equation sheet

Stress: $\sigma = \frac{dF}{dA}$ $\quad \sigma_{ave} = \frac{F}{A}$ $\quad \tau = \frac{dV}{dA}$ $\quad \tau_{ave} = \frac{V}{A}$

Strain: $\epsilon = \frac{du}{dx}$ $\quad \epsilon = \frac{\delta}{L_0}$ $\quad \epsilon = \ln(\frac{L_f}{L_0})$ $\quad \gamma = \frac{du}{dy}$ $\quad \gamma = \frac{\Delta X}{Y}$

Mechanical Properties: $\sigma = E\,\epsilon$ $\quad \tau = G\,\gamma$ $\quad \nu = -\epsilon_{lat}/\,\epsilon_{long}$ $\quad G = \frac{E}{2(1+\nu)}$ $\quad \epsilon = \alpha\Delta T$

Geometric properties of area elements: $J = \int_A \rho^2 dA$ $\quad \bar{y} = \frac{\int_A y dA}{A_{\text{total}}} = \frac{\sum y_i A_i}{A_{\text{total}}}$ $\quad I_x = \int_A y^2 dA$ $\quad Q = A'\,\bar{y}'$

Circle: $J = \frac{1}{2}\pi R^4$ $\quad I_x = I_y = \frac{1}{4}\pi R^4 = \frac{1}{64}\pi D^4$

Semi-circle: $\bar{y} = \frac{4R}{3\pi}$ $\quad I_x = I_y = \frac{1}{8}\pi r^4$

Rectangle: $I_x = \frac{1}{12}bh^3$ $\quad I_y = \frac{1}{12}b^3 h$

Parallel axis-theorem: $I_c = I_{c'} + A\,d_{cc'}^2$

Axial loading: $\delta = \int_{x=0}^{x=L} du$ $\quad \delta = \int_{x=0}^{x=L} \frac{F(x)dx}{E(x)A(x)}$ $\quad \delta = \frac{F\,L}{E\,A}$ $\quad \delta = \alpha\,L\,\Delta T$ $\quad k = \frac{EA}{L} = \frac{1}{f}$

Torsion: $\phi = \int_{x=0}^{x=L} d\phi$ $\quad \phi = \int_{x=0}^{x=L} \frac{T(x)dx}{G(x)J(x)}$ $\quad \phi = \frac{T\,L}{G\,J}$ $\quad \tau = \frac{T\,r}{J}$ $\quad \gamma = \frac{d\phi}{dx}r$ $\quad \gamma = \frac{\phi r}{L}$

Bending: $\frac{dV}{dx} = -w$ $\quad \frac{dM}{dx} = V$ $\quad \sigma_x = -\frac{M_z\,y}{I_z} + \frac{M_y\,z}{I_y}$ $\quad \tau = \frac{V\,Q}{I\,t}$ $\quad q = \tau t = \frac{V\,Q}{I}$

Pressure vessels, thin walls: (cylinder) $\sigma_\theta = p\frac{r}{t}$ $\quad \sigma_z = \frac{1}{2}p\frac{r}{t}$ $\quad$ (sphere) $\sigma_1 = \sigma_2 = \frac{1}{2}p\frac{r}{t}$

Transformation of Plane-Stress:

$$\sigma_{x'} = \frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos(2\theta) + \tau_{xy}\sin(2\theta)$$
$$\sigma_{y'} = \frac{\sigma_x + \sigma_y}{2} - \frac{\sigma_x - \sigma_y}{2}\cos(2\theta) - \tau_{xy}\sin(2\theta)$$
$$\tau_{x'y'} = -\frac{\sigma_x - \sigma_y}{2}\sin(2\theta) + \tau_{xy}\cos(2\theta)$$

Mohr's circle: (center): $\sigma_{ave} = \frac{\sigma_x+\sigma_y}{2}$ $\quad$ (radius): $R = \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + (\tau_{xy})^2}$

Principal stresses: $\sigma_1 = \sigma_{ave} + R$ $\quad \sigma_2 = \sigma_{ave} - R$ $\quad \tau = 0$ $\quad \tan(2\theta_p) = \frac{\tau_{xy}}{(\sigma_x-\sigma_y)/2}$

Maximum in-plane shear stress: $\tau_{max} = R$ $\quad \sigma_{avg} = \frac{\sigma_x+\sigma_y}{2}$ $\quad \tan(2\theta_s) = -\frac{(\sigma_x-\sigma_y)/2}{\tau_{xy}}$

Tresca criterion:

$|\sigma_1| = \sigma_Y,\ |\sigma_2| = \sigma_Y$ when $\sigma_1,\ \sigma_2$ have the same sign
$|\sigma_1 - \sigma_2| = \sigma_Y$ when $\sigma_1,\ \sigma_2$ have opposite sign

Von-Mises criterion: $\sigma_1^2 - \sigma_1\,\sigma_2 + \sigma_2^2 = \sigma_Y^2$

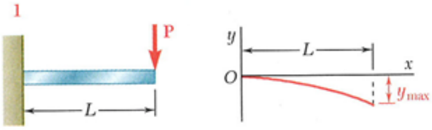
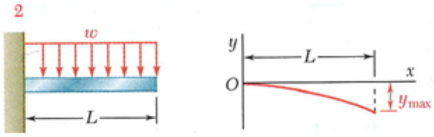
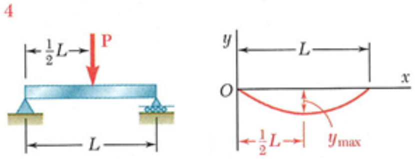
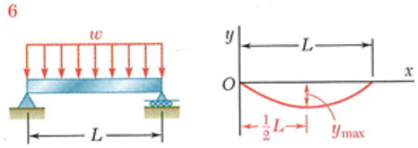
Generalized Hooke's law: $\epsilon_x = \frac{\sigma_x}{E} - \frac{\nu}{E}(\sigma_y + \sigma_z)$ $\quad \epsilon_y = \frac{\sigma_y}{E} - \frac{\nu}{E}(\sigma_x + \sigma_z)$ $\quad \epsilon_z = \frac{\sigma_z}{E} - \frac{\nu}{E}(\sigma_y + \sigma_x)$

Deflection: $y'' = M/(EI)$
Buckling: $P_{cr} = \frac{\pi^2}{K^2}\frac{EI}{L^2}$

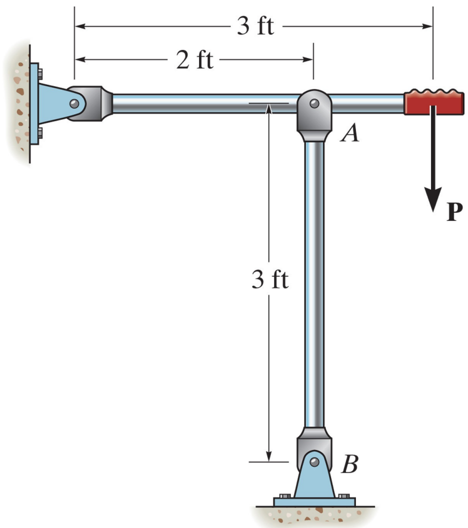
pinned-pinned: $K = 1$; pinned-fixed: $K = 0.7$

fixed-fixed: $K = 0.5$; fixed-free: $K = 2$

Figures

Beam and Loading / Elastic Curve	Maximum Deflection	Slope at End	Equation of Elastic Curve
1 	$-\frac{PL^3}{3EI}$	$-\frac{PL^2}{2EI}$	$y = \frac{P}{6EI}(x^3 - 3Lx^2)$
2 	$-\frac{wL^4}{8EI}$	$-\frac{wL^3}{6EI}$	$y = -\frac{w}{24EI}(x^4 - 4Lx^3 + 6L^2x^2)$
4 	$-\frac{PL^3}{48EI}$	$\pm\frac{PL^2}{16EI}$	For $x \le \frac{1}{2}L$: $y = \frac{P}{48EI}(4x^3 - 3L^2x)$
6 	$-\frac{5wL^4}{384EI}$	$\pm\frac{wL^3}{24EI}$	$y = -\frac{w}{24EI}(x^4 - 2Lx^3 + L^3x)$

1. (1 point) Determine the maximum force P that can be applied to the rigid handle so that the steel control rod AB does not buckle. The rod has elastic modulus $E = 29 \times 10^3$ ksi, diameter $d = 1.25$ in and is pinned at both ends.



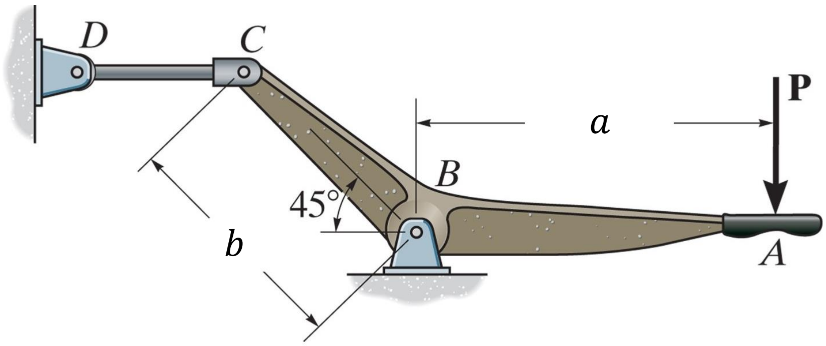
(A) 9.8 kips

(B) 17.6 kips

(C) 12.1 kips

(D) 4.6 kips

2. (1 point) A force $P = 1$ kN is applied to the bell crank below. Determine the average shear stress developed in the 6-mm diameter double sheared pin at B. Assume $a = 450$ mm and $b = 300$ mm.



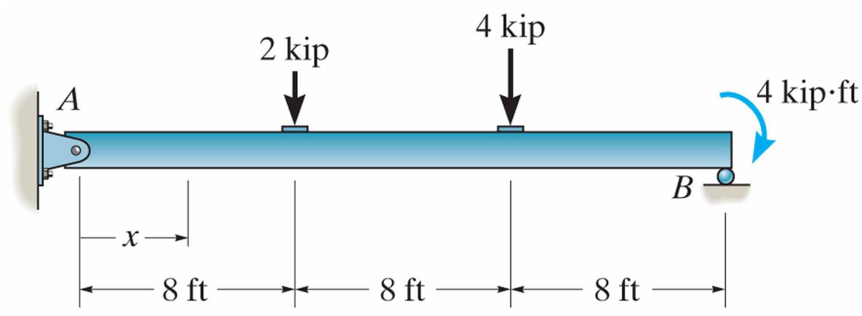
(A) $\tau_{pin} = 41.5$ MPa

(B) $\tau_{pin} = 82.9$ MPa

(C) $\tau_{pin} = 35.4$ MPa

(D) $\tau_{pin} = 75.0$ MPa

3. (1 point) Mark the CORRECT statement:



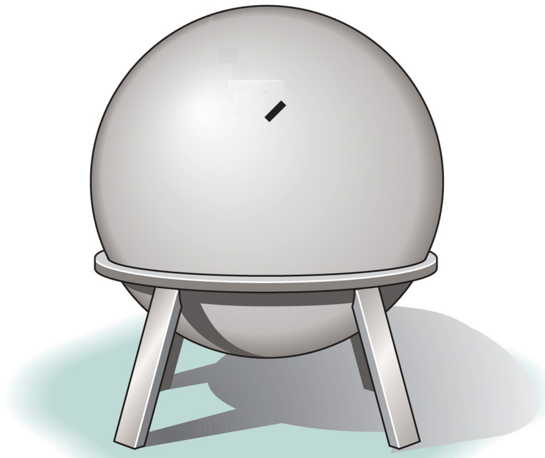
(A) The moment at B is equal to zero

(B) This is a statically indeterminate problem

(C) The slope at B is equal to zero

(D) The moment at A is equal to zero

4. (1 point) The spherical pressure vessel is subjected to an internal gage pressure $p = 90$ psi. If the internal diameter of the tank is 30 in., and the wall thickness is 0.15 in., determine the maximum in-plane shear stress in the spherical vessel.



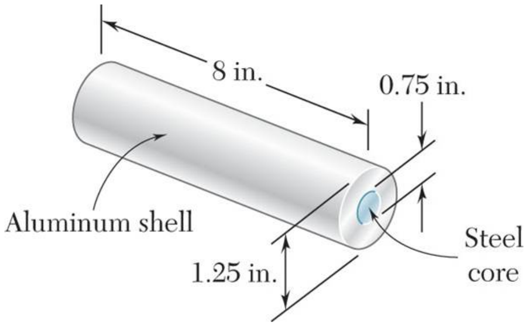
(A) 4500 psi

(B) 9000 psi

(C) 0 psi

(D) 2250 psi

5. (1 point) The assembly shown consists of an aluminum shell ($E_a = 10.6 \times 10^6$ psi and $\alpha_a = 12.9 \times 10^{-6}/^o\text{F}$) fully bonded to a steel core ($E_s = 29 \times 10^6$ psi and $\alpha_a = 6.5 \times 10^{-6}/^o\text{F}$) and is unstressed. Determine the change in length of the assembly when the temperature rises by 70^oF.



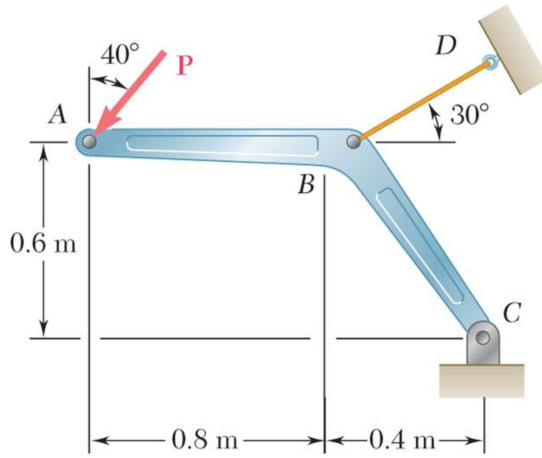
(A) 0.005 in

(B) 0.0025 in

(C) 0.0075 in

(D) 0.003 in

6. (1 point) Member ABC is rigid, pinned at C and supported by cable BD. Cable BD has ultimate stress $\sigma_u = 400$ MPa and was designed to support the load $P = 16$ kN as shown. Determine the minimum cross section area of the cable. Consider a factor of safety of 3 against cable failure.



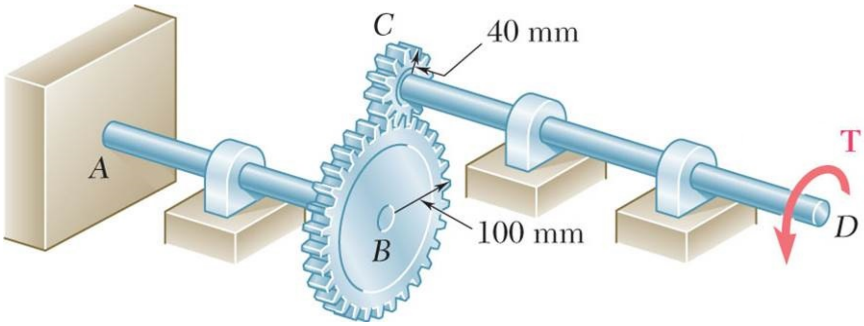
(A) 218 mm^2

(B) 188 mm^2

(C) 109 mm^2

(D) 342 mm^2

7. (1 point) In the system below, determine the reaction torque at the fixed end A when a torque $T = 200$ N.m is applied at D.



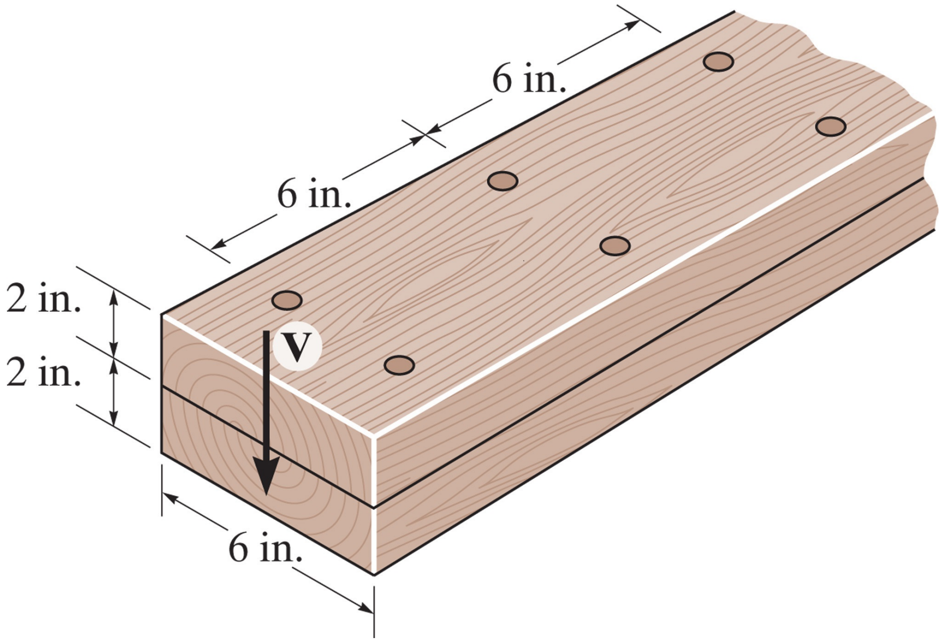
(A) Zero

(B) 80 N.m

(C) 200 N.m

(D) 500 N.m

8. (1 point) The beam is constructed from two boards fastened together at the top and bottom with two rows of nails spaced every 6 in. If an internal shear force of $V = 600$ lb is applied to the boards, determine the shear force resisted by each nail.



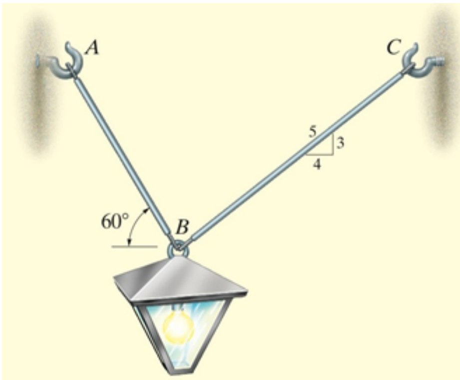
(A) 900 lb

(B) 337.5 lb

(C) 1350 lb

(D) 675 lb

9. (1 point) Cables AB and BC are used to suspend a lamp as illustrated below. Both cables are stress free before hanging the flowerpot and their original length is given by $L_{AB} = 400$ mm and $L_{BC} = 500$ mm. The weight of the lamp is $P = 2$ kN. Both cables have diameter $d = 10$ mm and are made with a material with elastic modulus $E = 70$ GPa. Determine the normal strain in cable BC.



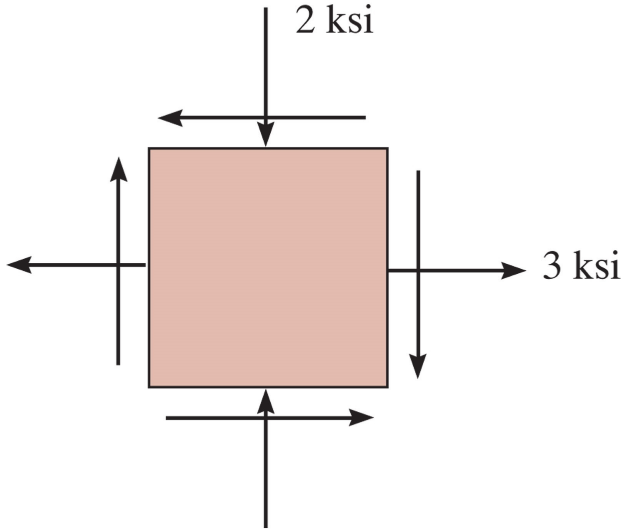
(A) $\epsilon = 0.18 \times 10^{-3}$

(B) $\epsilon = 0.14 \times 10^{-3}$

(C) $\epsilon = 0.09 \times 10^{-3}$

(D) $\epsilon = 0.29 \times 10^{-3}$

10. (1 point) For the element shown, determine the magnitude of the shear stress that gives a maximum in-plane shear stress equal to 3 ksi.



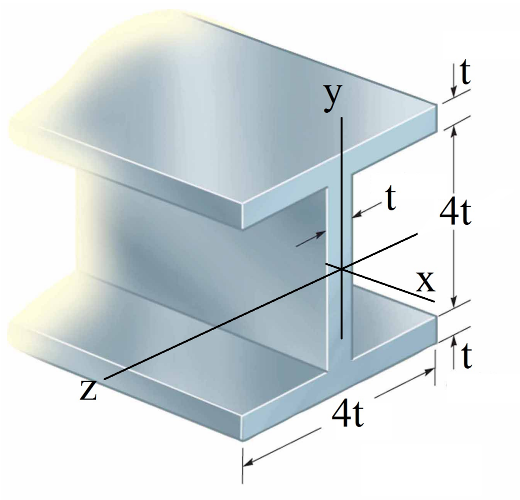
(A) 9 ksi

(B) 2.91 ksi

(C) 1 ksi

(D) 1.66 ksi

11. (1 point) Determine the moment of inertia I_z of the cross-section below with respect to the centroidal axis z. Use $t = 20$ mm.



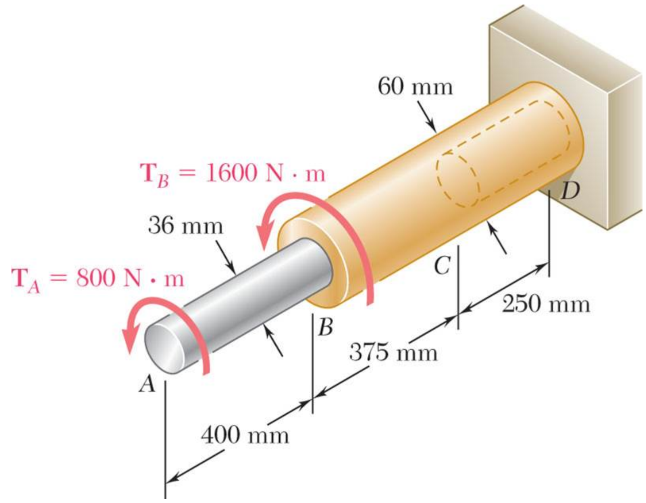
(A) $I_z = 12.5 \times 10^6$ mm^4

(B) $I_z = 4.91 \times 10^6$ mm^4

(C) $I_z = 8.96 \times 10^6$ mm^4

(D) $I_z = 6.08 \times 10^6$ mm^4

12. (1 point) The aluminum rod AB ($G_1 = 27$ GPa) is bonded to the brass rod BD ($G_2 = 39$ GPa). Knowing that portion CD of the brass rod is hollow and has inner diameter of 40 mm, determine the angle of twist at A.



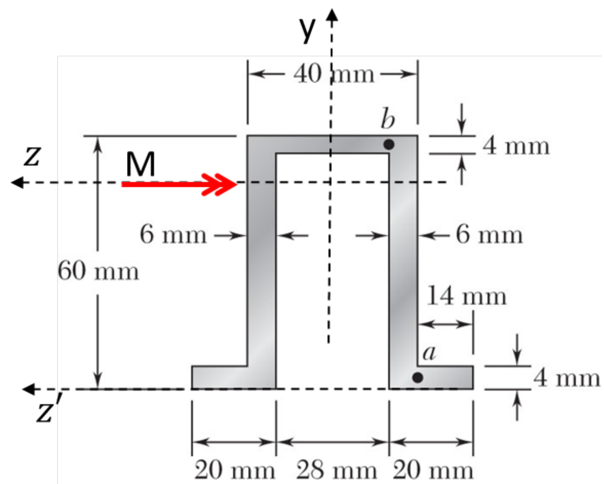
(A) 0.0824 rad

(B) 0.054 rad

(C) 0.079 rad

(D) 0.105 rad

13. (1 point) The centroid of the cross section below is located at $\bar{y} = 30$ mm measured from the z'-axis. The moment of inertia with respect to the centroidal axis z is given by $I = 0.39 \times 10^6$ mm^4. Determine the maximum tensile stress when the indicated bending moment is applied. The magnitude of the moment is 400 N.m.



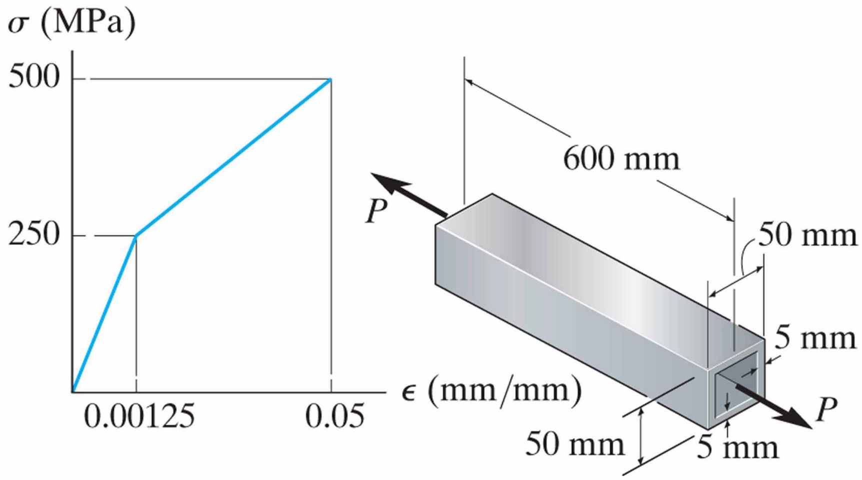
(A) 30.7 MPa

(B) 10.2 MPa

(C) 28.7 MPa

(D) 15.35 MPa

14. (1 point) The material for the hollow bar has the stress-strain diagram shown below. Consider the applied load $P = 180$ kN. Mark the correct statement:



(A) The deformation in the rod is plastic, the yielding strength is 250 MPa and the rupture strength is 500 MPa.

(B) The deformation in the rod is elastic, the yielding strength is 250 MPa and the rupture strength is 500 MPa.

(C) The deformation in the rod is elastic, the yielding strength is 250 MPa and the elastic modulus is 500 MPa.

(D) The deformation in the rod is plastic, the yielding strength is 250 MPa and the elastic modulus is 500 MPa.

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