Notre Dame University
Faculty of Engineering
Mechanical Engineering Department

Mechanics of Materials II (MEN 302)

Exam # 1
Spring 2015
April 18, 2015

Name : .... **Section :** A (MWF 10:00 – 11:00) □

ID : .... B (MWF 15:00 – 16:00) □

NOTES:

1. **ONLY one A4 paper (2 pages)** formula sheet is allowed.
2. This exam consists of 4 problems and 14 pages.
3. Cheating will not be tolerated.
4. The exam duration is 90 minutes.
5. A small formulas sheet is present at the last page.

PROBLEMS	POINTS	GRADE
Problem 1	20	
Problem 2	30	
Problem 3	20	
Problem 4	30	
FINAL GRADE		

Dr. Chady Ghnatios

Problem 1 [20 points]:

We consider a cube at position x and the reference axis $(\vec{\imath},\vec{\jmath},\vec{k})$. The cube is subjected to the following three stress experiments :

a) $\bar{\bar{\tau}}_1 = \begin{bmatrix} 0 & \sigma_0 & 0 \\ \sigma_0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}$

b) $\bar{\bar{\tau}}_2 = \begin{bmatrix} 0 & 0 & \sigma_0 \\ 0 & 0 & 0 \\ \sigma_0 & 0 & 0 \end{bmatrix}$

c) $\bar{\bar{\tau}}_3 = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & \sigma_0 \\ 0 & \sigma_0 & 0 \end{bmatrix}$

1) When performing the 3 experiences simultaneously, write the corresponding stress tensor. This stress tensor will be used for the rest of the problem.
2) Compute, when $\sigma_0 = 10$ MPa, the normal and shearing stresses on a plane normal to the vector $\vec{n} = \vec{\imath} + \vec{\jmath} + \vec{k}$, where $(\vec{\imath},\vec{\jmath},\vec{k})$ are the basis vectors.
3) Compute the principal stresses and deduce the highest principal stress direction without extra calculations.
4) Find the absolute maximum shearing stress in the element.

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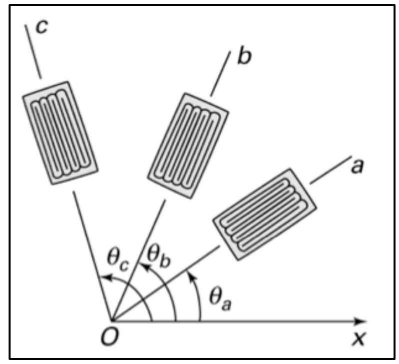
Problem 2 [30 points]:

Strain rosettes are placed on the wing of an airplane giving the following readings:



$$\epsilon_a = 800\mu$$

$$\epsilon_b = 100\mu$$

$$\epsilon_c = 400\mu$$

When the angles between the axis a, b and c and the x axis are $\theta_a = 0°$; $\theta_b = 45°$ and $\theta_c = 90°$.

The stresses in the wing are not to exceed are 75 MPa for normal stress and 35.5 MPa for shear stress. Is the wing working in safe conditions? Assume plane stresses in the beam. The modulus of elasticity is $E_{\text{wing}} = 76$ GPa and the Poisson ratio is $\nu = 0.35$.

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Problem 3 [20 points]:

A test model (Figure 1) is subjected to a uniform compressive load that produces a compressive stress with magnitude σ_0. Design specifications require that the stresses in the member do not exceed a tensile stress of 400 MPa, a compressive stress of 560 MPa, and a shear stress of 160 MPa. The compressive stress σ_0 is increased until one of these values is reached.

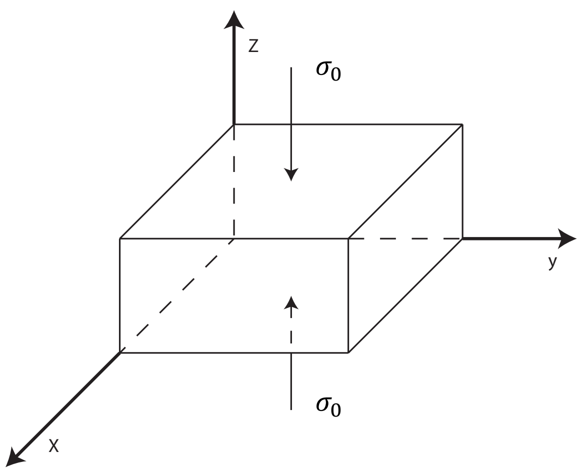

Figure 1

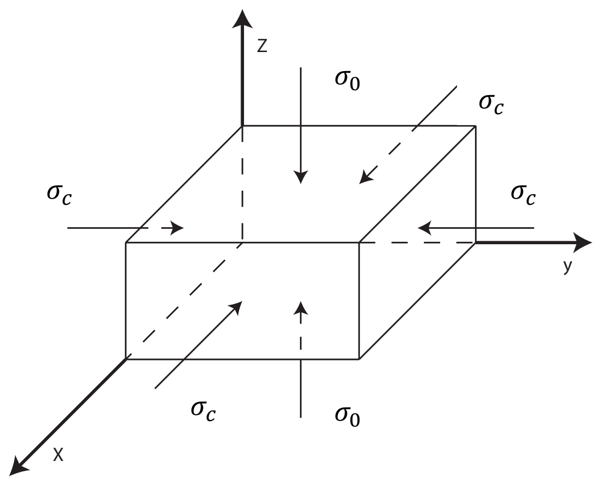

Figure 2

a) Which one of these values is the limiting one (will be attained first) and what is the corresponding value of σ_0.
b) Assuming σ_0 is less than 560 MPa. We wish to add a uniform lateral compression σ_c (as shown in Figure 2) so that σ_0 may reach the value of 560 MPa before exceeding any of the other design requirements. Compute the value of the lateral normal stress σ_c added to the system.

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Problem 4 [30 points]:

Consider the following thin walled pressure vessel illustrated in Figure 1. The pressure vessel has a radius of 2 m and a thickness of 10 mm. A torsion moment T of 10 kNm and an axial load F of 50 kN are applied to the pressure vessel.

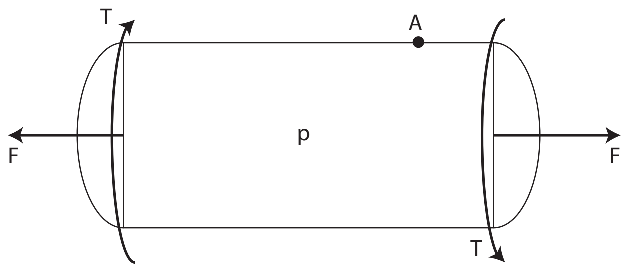

Figure 1

a) What would be the maximum allowable internal pressure p of the pressure vessel so that the maximum normal stress does not exceed 500 kPa, and the maximum shear stress should not exceed 25 kPa, on a point A located at the outer surface of the pressure vessel? What is the corresponding maximum shear stress?
b) Consider p=1.7 kPa. What are the corresponding strains at point the same point A?

Use E=200 GPa and ν=0.3

For thin tubes : $A_c = 2\pi Rt$ and $J = 2\pi R^3 t$

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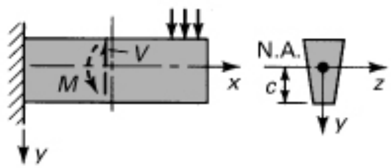
TABLE 1.1. *Commonly Used Elementary Formulas for Stress*[a]

1. *Prismatic Bars of Linearly Elastic Material*

Axial loading: $\sigma_x = \dfrac{P}{A}$ (a)

Torsion: $\tau = \dfrac{T\rho}{J}, \qquad \tau_{\max} = \dfrac{Tr}{J}$ (b)



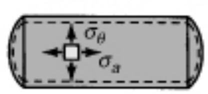
Bending: $\sigma_x = -\dfrac{My}{I}, \qquad \sigma_{\max} = \dfrac{Mc}{I}$ (c)

Shear: $\tau_{xy} = \dfrac{VQ}{Ib}$ (d)

where

σ_x = normal axial stress
τ = shearing stress due to torque
τ_{xy} = shearing stress due to vertical shear force
P = axial force
T = torque
V = vertical shear force
M = bending moment about z axis
A = cross-sectional area
y, z = centroidal principal axes of the area
I = moment of inertia about neutral axis (N.A.)
J = polar moment of inertia of circular cross section
b = width of bar at which τ_{xy} is calculated
r = radius
Q = first moment about N.A. of the area beyond the point at which τ_{xy} is calculated

2. *Thin-Walled Pressure Vessels*



Cylinder: $\sigma_\theta = \dfrac{pr}{t}, \qquad \sigma_a = \dfrac{pr}{2t}$ (e)

Sphere: $\sigma = \dfrac{pr}{2t}$ (f)

where

σ_θ = tangential stress in cylinder wall
σ_a = axial stress in cylinder wall
σ = membrane stress in sphere wall
p = internal pressure
t = wall thickness
r = mean radius

[a]Detailed derivations and limitations of the use of these formulas are discussed in Sections 1.6, 5.7, 6.2, and 13.13.