IPE 303
Product Design I

Chapter 5: Failures Resulting from Static Loading

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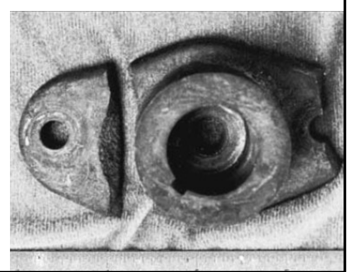

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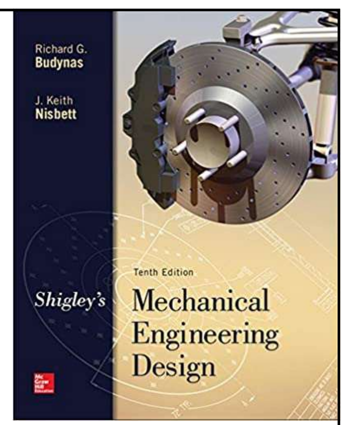

Reference

Shigley's Mechanical Engineering Design, 10th Edition

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Introduction to Stress

- **Static Load:** A static load is a **stationary force**, or couple applied to a member. To be stationary, the force or couple must be **unchanging in magnitude, point or points of application, and direction**.

- **Stress:** Force per unit area. It is a **vector** quantity.

Difference between **stress and pressure?**

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Introduction to Stress › Stress Components

Cartesian Stress Components:

σ_x : $\sigma = Normal\ Stress$
$subscript\ x = direction\ of\ the\ surface\ normal$

τ_{xy}: $\tau = Shear\ Stress$
$subscript\ x = direction\ of\ the\ surface\ normal$
$subscript\ y = direction\ of\ the\ shear\ stress$

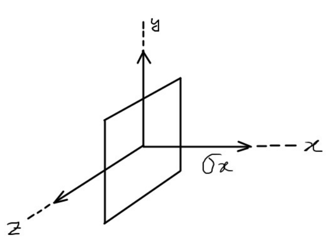

Fig: Stress Components on surface normal to x-axis

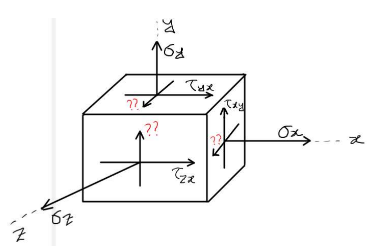

Fig: General Three-Dimensional stress state

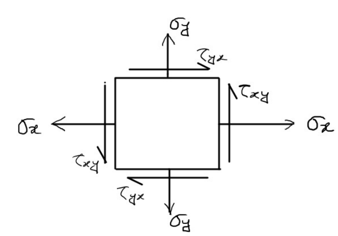

Fig: Plane stress in Equilibrium or cross shear equal (looking from z-axis)

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Introduction to Stress ❯ Stress Components

- Total Stress Components, Nine (9)-

  $\sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yx}, \tau_{yz}, \tau_{zy}, \tau_{zx}, \tau_{xz}$

- In most cases, the stresses are considered in **Equilibrium**, cross-shears are equal, i.e.,

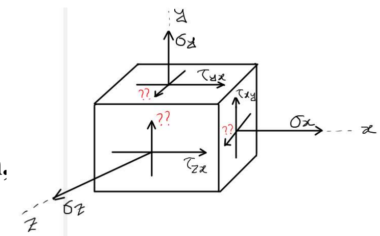
  $\tau_{xy} = \tau_{yx}, \quad \tau_{yz} = \tau_{zy}, \quad \tau_{zx} = \tau_{xz}$

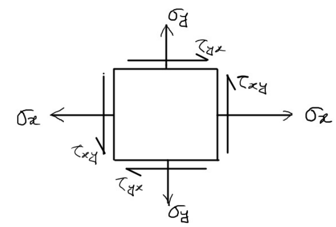
  Stress components reduced to six (6) - $\sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yz}, \tau_{zx}$

- **Plane stress:** stresses on one surface are equal to zero, i.e.,

  $\sigma_z = \tau_{xz} = \tau_{yz} = 0$

  In equilibrium, plane Stress components reduced to three (3) - $\sigma_x, \sigma_y, \tau_{xy}$





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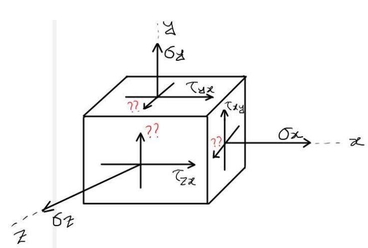
Introduction to Stress ❯ Stress Components

QUIZ – 1

Draw the plane stress in Equilibrium while looking from x-axis and y-axis



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Introduction to Stress › Stress Classification

Stress

- **Normal Stress** (Perpendicular to the surface)
  - Tensile ($\sigma = \frac{P}{A}$)
  - Compressive ($\sigma = \frac{P}{A}$)
  - Flexural/Bending ($\sigma = \frac{Mc}{I}; for\ max\ or\ \sigma = \frac{Mr}{I}; for\ any\ point$)
- **Shear Stress** (Parallel to the surface)
  - Direct shear ($\tau = \frac{P}{A}$)
  - Torsional shear ($\tau = \frac{Tr}{J}$)

where, I = moment of Inertia, and J = Second/polar moment of inertia/area

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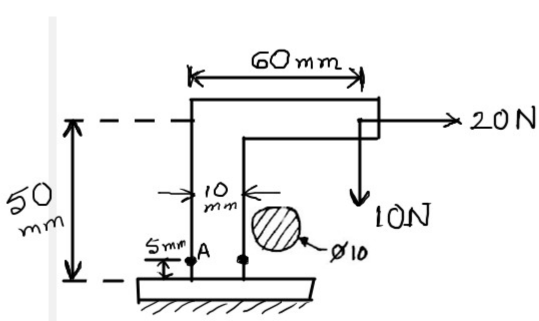
Introduction to Stress › Combined Stress

Find resultant stress at point A.



What will happen if there is a **shear stress** with these stresses?

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Introduction to Stress > Combined Stress

- Since stress is a **vector quantity**, we can not add or subtract **Normal and Shear** stress arithmetically.

- On a machine component, there can be combined stresses such as-
  - Tension/compression + Bending (Arithmetically)
  - Tension/compression + Torsion (Geometrically)
  - Torsion + Bending (Geometrically)
  - Tension/compression + Bending + Torsion (Geometrically)

This Geometrical method is nothing but a **Mohr's Circle!**

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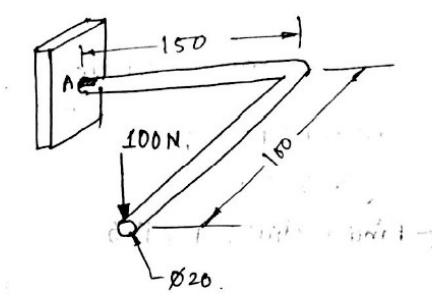
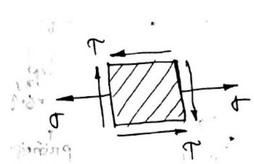
Introduction to Stress > Combined Stress > Mohr's Circle

2D Mohr's Circle

- x and y axis represent normal and shear stresses, respectively.
- Angle in circle is twice the actual angle.

Example -





Consider element A,

$$\sigma_x = \frac{Mc}{I} = \frac{100 \times 150 \times 10 \times 64}{\pi(20)^4} = 19.1\ MPa$$

$$\tau_{xy} = \frac{Tr}{J} = \frac{100 \times 100 \times 10 \times 32}{\pi(20)^4} = 6.4\ MPa$$

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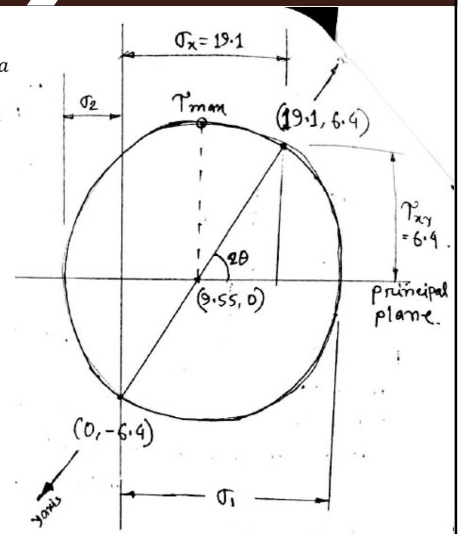
Introduction to Stress | Combined Stress | Mohr's Circle

- Draw a Mohr's circle.
- To find-
  - The center.
  - The radius.
  - The principal plane
  - The principal stress
  - (Plane) Max. Shear stress

$$\sigma_x = \frac{Mc}{I} = \frac{100 \times 150 \times 10 \times 64}{\pi(20)^4} = 19.1\,MPa$$

$$\tau_{xy} = \frac{Tr}{J} = \frac{100 \times 100 \times 10 \times 32}{\pi(20)^4} = 6.4\,MPa$$



Solution:

- Principal plane makes an angle 2θ with x-axis in Mohr's circle;
  $tan2\theta = \frac{6.4}{19.1-9.55} \Rightarrow \theta = -16.9°$
- Principal stresses, $\sigma_1 = C + R = 9.55 + 11.5 = 21.05\,MPa;$
  $\sigma_2 = C - R = 9.55 - 11.5 = -1.95\,MPa$
- (Plane) Max. shear stress, $\tau_{max} = +R = 11.5\,MPa$

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Introduction to Stress | Combined Stress | Mohr's Circle

Output of a 2D Mohr's circle:

- **Principal Plane and Principal stress:**
  - A plane where there is only normal stress exists, no shear. The normal stresses acting on the principal plane are Principal stresses.
  - From a 2D Mohr's circle we get two principal stresses - σ_1, σ_2

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

  - No shear stress exists on principal plane.
  - Brittle Materials are very prune to principal stress and fracture of brittle material begins at principal plane.

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Output of a 2D Mohr's circle:

- **Max. Shear plane and Extreme value shear stress:**
  - Maximum shear plane occurs at 45° to the principal plane. From a 2D Mohr's circle we get two extreme value shear stresses- τ_1, τ_2

$$\tau_1, \tau_2 = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

  - Point to be noted that, these extreme value shear stresses might not be actual maximum shear stress. Actual maximum shear stress is encouraged to be derived from 'ordered' principal stresses ($\sigma_1 > \sigma_2 > \sigma_3$), $\tau_{max} = \frac{\sigma_1 - \sigma_3}{2}$.
  - However, if the problem becomes **uniaxial**, then it is guaranteed that **Extreme value shear stress = Max. shear stress**.

3D Mohr's Circle

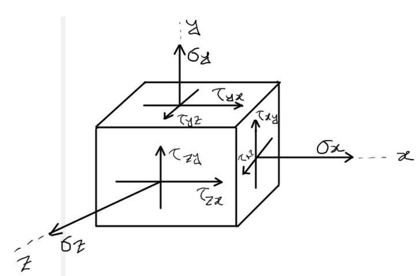


Fig: General Three-Dimensional stress state

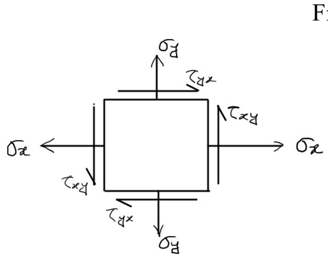


Fig: Plane stress in Equilibrium or cross shear equal (looking from z-axis)

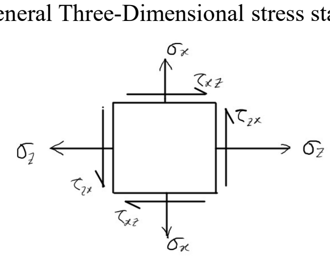


Fig: Plane stress in Equilibrium or cross shear equal (looking from y-axis)

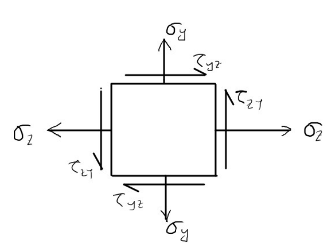


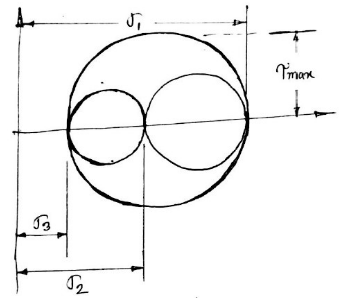
Fig: Plane stress in Equilibrium or cross shear equal (looking from x-axis)

Introduction to Stress | Combined Stress | Mohr's Circle

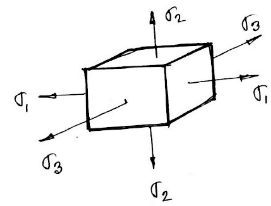
Output of a 3D Mohr's circle:

- **Principal stresses and Absolute maximum shear stress:**
  - From the three plane stress state, three 2D Mohr's circle can be drawn. From each 2D Mohr's circle, two principal stresses can be found using the following formula-

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$



  - From the principal stress values, three unique principal stresses can be found, i.e., $\sigma_1, \sigma_2, \sigma_3$ and we can draw a 3D Mohr's circle.
  - Absolute maximum shear stress, $\boldsymbol{\tau_{max} = \frac{\sigma_1 - \sigma_3}{2}}$, **where** $\boldsymbol{\sigma_1 > \sigma_2 > \sigma_3}$



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Introduction to Stress | Combined Stress | Mohr's Circle

Practice Problem:

Find τ_{max} from the following stresses.

(a) $\sigma_x = 70$ kpsi, $\sigma_y = 70$ kpsi, $\tau_{xy} = 0$ kpsi
(b) $\sigma_x = 60$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = -15$ kpsi
(c) $\sigma_x = 0$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = 45$ kpsi
(d) $\sigma_x = -40$ kpsi, $\sigma_y = -60$ kpsi, $\tau_{xy} = 15$ kpsi
(e) $\sigma_1 = 30$ kpsi, $\sigma_2 = 30$ kpsi, $\sigma_3 = 30$ kpsi

Hints:

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_1, \tau_2 = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2}$, where $\sigma_1 > \sigma_2 > \sigma_3$

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Introduction to Stress › Combined Stress › Von Mises Stress

Von Mises Stress:

For principal stresses,

$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2}$$

For principal stresses in plane,

$$\sigma' = (\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2)^{1/2}$$

For general 3D stress state,

$$\sigma' = \frac{1}{\sqrt{2}}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{1/2}$$

For plane stress in equilibrium,

$$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$$

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Introduction to Stress › Combined Stress › Von Mises Stress

QUIZ – 2

Find von mises stress, σ' from the following stresses.

(*a*) $\sigma_x = 70$ kpsi, $\sigma_y = 70$ kpsi, $\tau_{xy} = 0$ kpsi
(*b*) $\sigma_x = 60$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = -15$ kpsi
(*c*) $\sigma_x = 0$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = 45$ kpsi
(*d*) $\sigma_x = -40$ kpsi, $\sigma_y = -60$ kpsi, $\tau_{xy} = 15$ kpsi
(*e*) $\sigma_1 = 30$ kpsi, $\sigma_2 = 30$ kpsi, $\sigma_3 = 30$ kpsi

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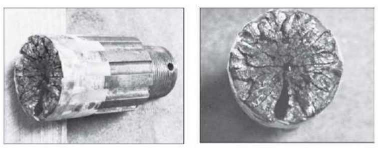
5-3 Failure Theories

Failure in Machine Elements:

- Failure in machine design means that a part becomes permanently distorted, thus its function is compromised.
- Failure occurs when **stress becomes equal or greater than failure strength**.
  - **Failure strength:** The maximum limit of stress when failure occurs. For example- yield, Ultimate strength, elastic limit, proportional limit, etc.



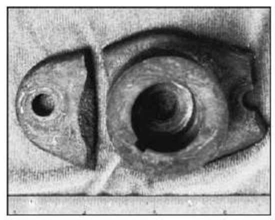
(a) (b)

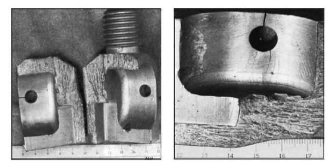
Figure 5–1

(*a*) Failure of a truck drive-shaft spline due to corrosion fatigue. Note that it was necessary to use clear tape to hold the pieces in place. (*b*) Direct end view of failure. *(For permission to reprint Figs. 5–1 through 5–5, the authors are grateful for the personal photographs of Larry D. Mitchell, co-author of Mechanical Engineering Design, 4th ed., McGraw-Hill, New York, 1983.)*

Figure 5–2

Impact failure of a lawn-mower blade driver hub. The blade impacted a surveying pipe marker.





(a) (b)

Figure 5–4

Chain test fixture that failed in one cycle. To alleviate complaints of excessive wear, the manufacturer decided to case-harden the material. (*a*) Two halves showing fracture; this is an excellent example of brittle fracture initiated by stress concentration. (*b*) Enlarged view of one portion to show cracks induced by stress concentration at the support-pin holes.

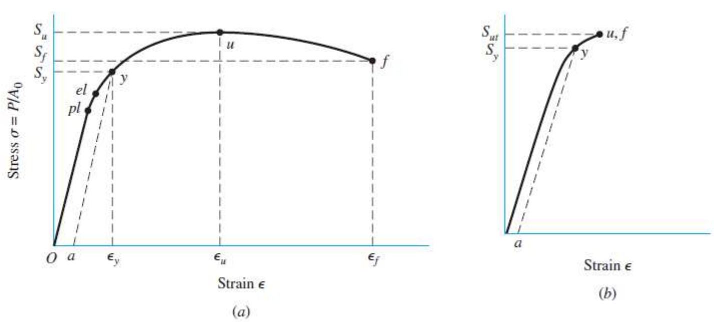
5-3 Failure Theories

Failure in Machine Elements:

- For ductile material, failure strength means yield strength (s_{yt} or s_{yc})
- For brittle material, failure strength means ultimate strength (U_t or U_c)

Figure 2–2

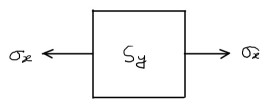
Stress-strain diagram obtained from the standard tensile test (*a*) Ductile material; (*b*) brittle material.
pl marks the proportional limit; *el*, the elastic limit; *y*, the offset-yield strength as defined by offset strain *a*; *u*, the maximum or ultimate strength; and *f*, the fracture strength.



5-3 Failure Theories

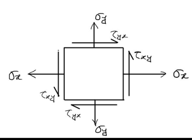
Failure in Machine Elements:

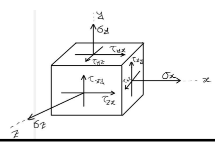
- Most of the **ductile materials** are very prone to **maximum shear stress**. That means ductile material fails because of shear stress, at max. shear plane.
- On the contrary, **brittle materials** are very prone to Maximum normal stresses, specifically **principal stresses**. Failure initiates at principal plane.
- For materials subjected to **uniaxial** stress (without shear), it fails when the stress reaches the failure strength (i.e., yield strength, ultimate strength, percent elongation)



Fails when $\sigma_x \geq s_y$

- To deal with **biaxial and triaxial** stresses, we need to use **failure theories**.





Fails according to **Failure Theories**

5-3 Failure Theories

Some recognized failure theories:

- Failure theories take multi-axial stresses as input and return ‘something’ which can be easily compared to Failure Strengths.
- Because ductile and brittle materials have different failure mechanism, different failure theories are used for each of them.
- **Failure theories for Ductile Material (yield criterion, $\varepsilon_f \geq 0.05$)**
  - Ductile materials fail on the maximum shear plane due to maximum shear stress.
  - Yield strength is considered as the Failure strength of the ductile materials.
  - Some recognized failure theories for ductile materials-
    - **Max. Shear Stress Theory (MSS)**
    - **Distortion Energy Theory (DE)**
    - **Ductile Coulomb-Mohr Theory (DCM)**

5-3 Failure Theories

Some recognized failure theories:

- **Failure theories for Brittle Material (yield criterion, $\varepsilon_f < 0.05$)**
  - Brittle materials fail on the principal plane due to principal stress.
  - Ultimate strength is considered as the Failure strength of the brittle materials.
  - Some recognized failure theories for ductile materials-
    - **Max. Normal Stress Theory (MNS)**
    - **Brittle Coulomb-Mohr Theory (BCM)**
    - **Modified-Mohr Theory (MM)**

5-4 Maximum-Shear-Stress Theory for Ductile Materials

MSS Theory:

- According to MSS, failure occurs when –

$$\tau_{max} \geq S_{sy} \text{ and } S_{sy} = 0.5S_y$$

- Design equation-

$$\tau_{max} = \frac{0.5S_y}{n}$$

- More conservative than other failure theories.

Where,
$\tau_{max} = Maximum\ shear\ stress$
$S_y = Yield\ strength$
$S_{sy} = Yield\ strength\ in\ shear$
$n = Factor\ of\ safety$

5-5 Distortion-Energy Theory for Ductile Materials

DE Theory:

- According to DE, failure occurs when –

$$\sigma' \geq S_y$$

- Design equation-

$$\sigma' = \frac{S_y}{n}$$

- According to DE, $S_{sy} = 0.577 S_y$

Where,
$\sigma' = Von\ Mises\ stress$
$S_y = Yield\ strength$
$S_{sy} = Yield\ strength\ in\ shear$
$n = Factor\ of\ safety$

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5-5 Distortion-Energy Theory for Ductile Materials

EXAMPLE 5–1 A hot-rolled steel has a yield strength of $S_{yt} = S_{yc} = 100$ kpsi and a true strain at fracture of $\varepsilon_f = 0.55$. Estimate the factor of safety for the following principal stress states:
(*a*) $\sigma_x = 70$ kpsi, $\sigma_y = 70$ kpsi, $\tau_{xy} = 0$ kpsi
(*b*) $\sigma_x = 60$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = -15$ kpsi
(*c*) $\sigma_x = 0$ kpsi, $\sigma_y = 40$ kpsi, $\tau_{xy} = 45$ kpsi
(*d*) $\sigma_x = -40$ kpsi, $\sigma_y = -60$ kpsi, $\tau_{xy} = 15$ kpsi
(*e*) $\sigma_1 = 30$ kpsi, $\sigma_2 = 30$ kpsi, $\sigma_3 = 30$ kpsi

Hints

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_1, \tau_2 = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2}, \text{ where } \sigma_1 > \sigma_2 > \sigma_3$$

$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2}$$

$$\sigma' = (\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2)^{1/2}$$

$$\sigma' = \frac{1}{\sqrt{2}}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{1/2}$$

$$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$$

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5-6 Coulomb-Mohr Theory for Ductile Materials

DCM Theory:

- According to DCM, when $S_{yt} \neq S_{yc}$ for any material, failure condition is–

$$\frac{\sigma_1}{S_{yt}} - \frac{\sigma_3}{S_{yc}} \geq 1$$

- Design equation-

$$\frac{\sigma_1}{S_{yt}} - \frac{\sigma_3}{S_{yc}} = \frac{1}{n}$$

- According to DCM,

$$S_{sy} = \frac{S_{yt} S_{yc}}{S_{yt} + S_{yc}}$$

Where,
σ_1, σ_3 *are Principal stresses*
S_{yt} = *Tensile yield strength*
S_{yc} = *Compression yield strength*
S_{sy} = *Yield strength in shear*
n = *Factor of safety*

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Failure Theory for Ductile Materials

QUIZ – 3

EXAMPLE 5–2

A 40-mm-diameter shaft is statistically torqued to 350 Nm. It is made of cast 195-T6 aluminum with a yield strength in tension of 160 MPa and a yield strength in compression of 170 MPa. Estimate the Factor of safety of the shaft using –

i. DCM Theory
ii. MSS Theory

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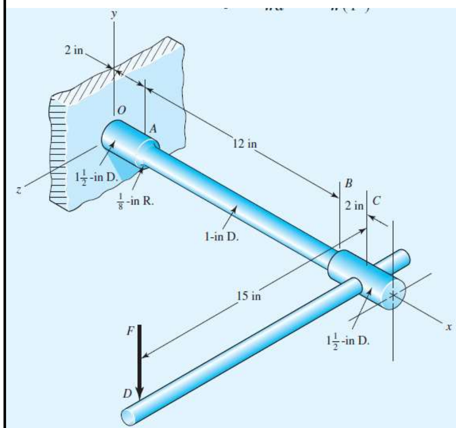
5-5 Distortion-Energy Theory for Ductile Materials

EXAMPLE 5–3

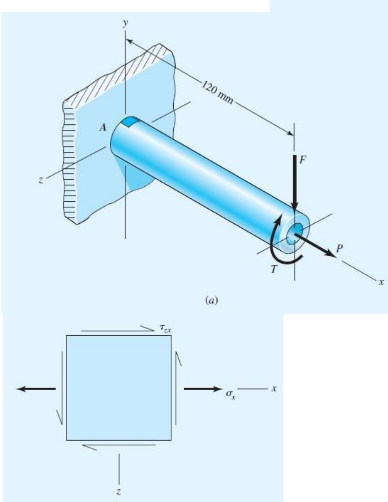
A certain force F applied at D near the end of the 15-in lever shown in Fig. 5–16, which is quite similar to a socket wrench, results in certain stresses in the cantilevered bar $OABC$. This bar ($OABC$) is of AISI 1035 steel, forged and heat-treated so that it has a minimum (ASTM) yield strength of 81 kpsi. We presume that this component would be of no value after yielding. Thus the force F required to initiate yielding can be regarded as the strength of the component part. Find this force.



5-5 Distortion-Energy Theory for Ductile Materials

EXAMPLE 5–4

The cantilevered tube shown in Fig. 5–17 is to be made of 2014 aluminum alloy treated to obtain a specified minimum yield strength of 276 MPa. We wish to select a stock-size tube from Table A–8 using a design factor $n_d = 4$. The bending load is $F = 1.75$ kN, the axial tension is $P = 9.0$ kN, and the torsion is $T = 72$ N · m. What is the realized factor of safety?



Practice Yourself!

5-8 Maximum-Normal-Stress Theory for Brittle Materials

MNS Theory:

- According to MNS, failure occurs whenever one of the three principal stresses becomes equal or exceeds the Ultimate tensile or compressive strength.

$$\sigma_1 \geq S_{ut} \qquad \text{or} \qquad \sigma_3 \leq -S_{uc}$$

- Design equation-

$$\sigma_1 = \frac{S_{ut}}{n}$$

$$\sigma_3 = \frac{-S_{uc}}{n}$$

Where,
σ_1, σ_3 *are Principal stresses*
S_{ut} = *Ultimate tensile strength*
S_{uc} = *Ultimate compressive strength*
n = *Factor of safety*

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5-9 Modifications of the Mohr Theory for Brittle Materials

Brittle-Coulomb-Mohr (BCM) Theory:

- According to BCM, failure condition is–

$$\frac{\sigma_1}{S_{ut}} - \frac{\sigma_3}{S_{uc}} \geq 1$$

- Design equation-

$$\frac{\sigma_1}{S_{ut}} - \frac{\sigma_3}{S_{uc}} = \frac{1}{n}$$

Where,
σ_1, σ_3 *are Principal stresses*
S_{ut} = *Ultimate tensile strength*
S_{uc} = *Ultimate compressive strength*
n = *Factor of safety*

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5-9 Modifications of the Mohr Theory for Brittle Materials

Modified Mohr (MM) Theory:

Where,
σ_A, σ_B *are two non* − *zero principal stresses for plane stress*
S_{ut} = *Ultimate tensile strength*
S_{uc} = *Ultimate compressive strength*
n = *Factor of safety*

- According to MM theory, design equations are–

$$\sigma_A = \frac{S_{ut}}{n} \quad \text{when,} \begin{cases} \sigma_A \geq \sigma_B \geq 0 \\ \sigma_A \geq 0 \geq \sigma_B \quad \text{and} \quad \left|\dfrac{\sigma_B}{\sigma_A}\right| \leq 1 \end{cases}$$

$$\frac{(S_{uc} - S_{ut})\sigma_A}{S_{uc}S_{ut}} - \frac{\sigma_B}{S_{uc}} = \frac{1}{n} \quad \text{when,} \left\{ \sigma_A \geq 0 \geq \sigma_B \quad \text{and} \quad \left|\frac{\sigma_B}{\sigma_A}\right| > 1 \right.$$

$$\sigma_B = -\frac{S_{uc}}{n} \quad \text{when,} \left\{ 0 \geq \sigma_A \geq \sigma_B \right.$$

5-9 Modifications of the Mohr Theory for Brittle Materials

EXAMPLE 5–5 Consider the wrench in Ex. 5–3, Fig. 5–16, as made of cast iron, machined to dimension. The force F required to fracture this part can be regarded as the strength of the component part. If the material is ASTM grade 30 cast iron, find the force F with
(*a*) Coulomb-Mohr failure model.
(*b*) Modified Mohr failure model.

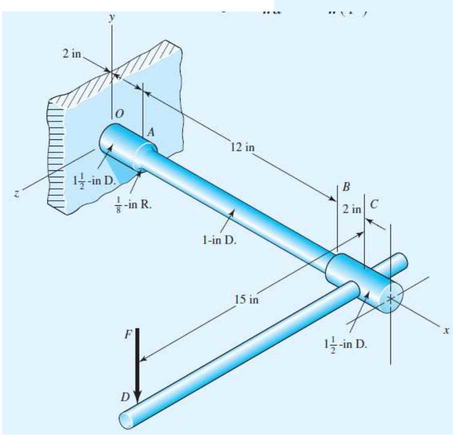


Table A–24

Mechanical Properties of Three Non-Steel Metals
(a) Typical Properties of Gray Cast Iron
[The American Society for Testing and Materials (ASTM) numbering system for gray cast iron is such that the numbers correspond to the *minimum tensile strength* in kpsi. Thus an ASTM No. 20 cast iron has a minimum tensile strength of 20 kpsi. Note particularly that the tabulations are *typical* of several heats.]

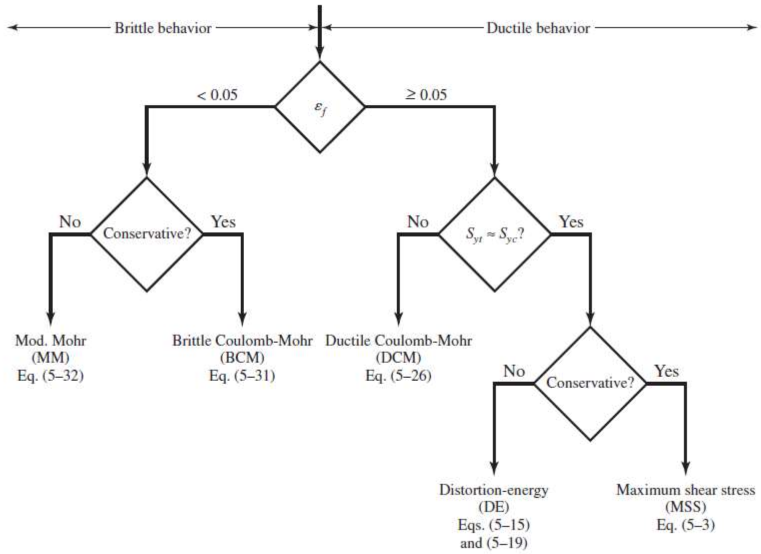
ASTM Number	Tensile Strength S_{ut}, kpsi	Compressive Strength S_{uc}, kpsi	Shear Modulus of Rupture S_{su}, kpsi	Modulus of Elasticity, Mpsi: Tension†	Modulus of Elasticity, Mpsi: Torsion	Endurance Limit* S_e, kpsi	Brinell Hardness H_B	Fatigue Stress-Concentration Factor K_f
20	22	83	26	9.6–14	3.9–5.6	10	156	1.00
25	26	97	32	11.5–14.8	4.6–6.0	11.5	174	1.05
30	31	109	40	13–16.4	5.2–6.6	14	201	1.10
35	36.5	124	48.5	14.5–17.2	5.8–6.9	16	212	1.15
40	42.5	140	57	16–20	6.4–7.8	18.5	235	1.25
50	52.5	164	73	18.8–22.8	7.2–8.0	21.5	262	1.35
60	62.5	187.5	88.5	20.4–23.5	7.8–8.5	24.5	302	1.50

*Polished or machined specimens.
†The modulus of elasticity of cast iron in compression corresponds closely to the upper value in the range given for tension and is a more constant value than that for tension.

5-11 Selection of Failure Criteria

Figure 5–21

Failure theory selection flowchart.



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