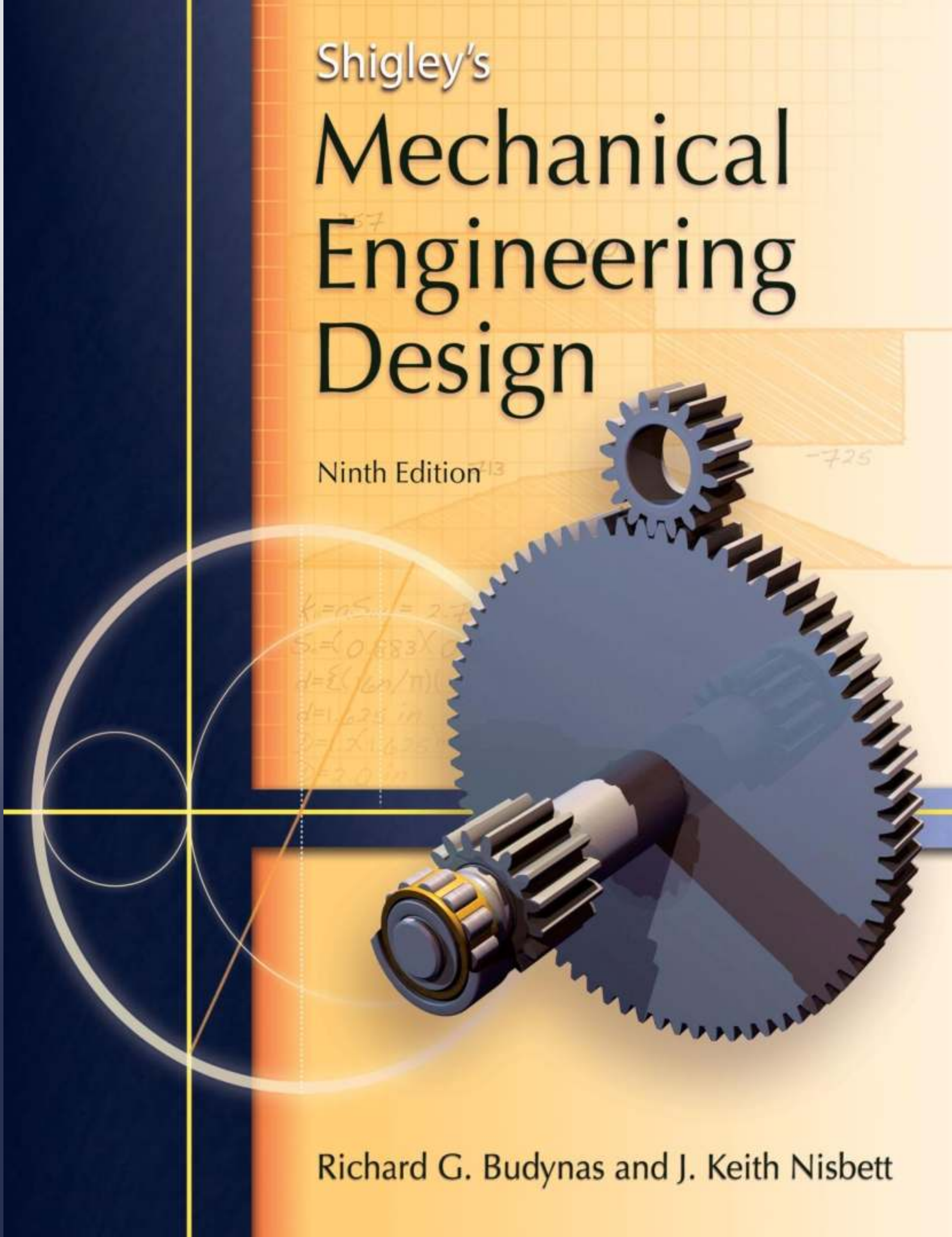
Lecture Slides

Chapter 3

Load and Stress Analysis



Equilibrium and Free-Body Diagram

System: used to denote any *isolated* part or portion of a machine or structure. A system may consist of

- a particle, or several particles,
- a part of rigid body, an entire rigid body, or several rigid bodies.

Equilibrium: A system is said to be in *equilibrium* if it is motionless or has a constant velocity, i.e.; zero acceleration. The phrase *static equilibrium* is also used to imply that the system is at rest. For equilibrium, the forces and moments acting on the system balance such that

$$\sum \vec{\mathbf{F}} = \sum F_x \vec{\mathbf{i}} + \sum F_y \vec{\mathbf{j}} + \sum F_z \vec{\mathbf{k}} = \vec{0} \quad (\text{Vectorial Representation})$$

or

$$\begin{cases} \sum F_x = 0 \\ \sum F_y = 0 \\ \sum F_z = 0 \end{cases} \quad (\text{Scalar Representation})$$

and

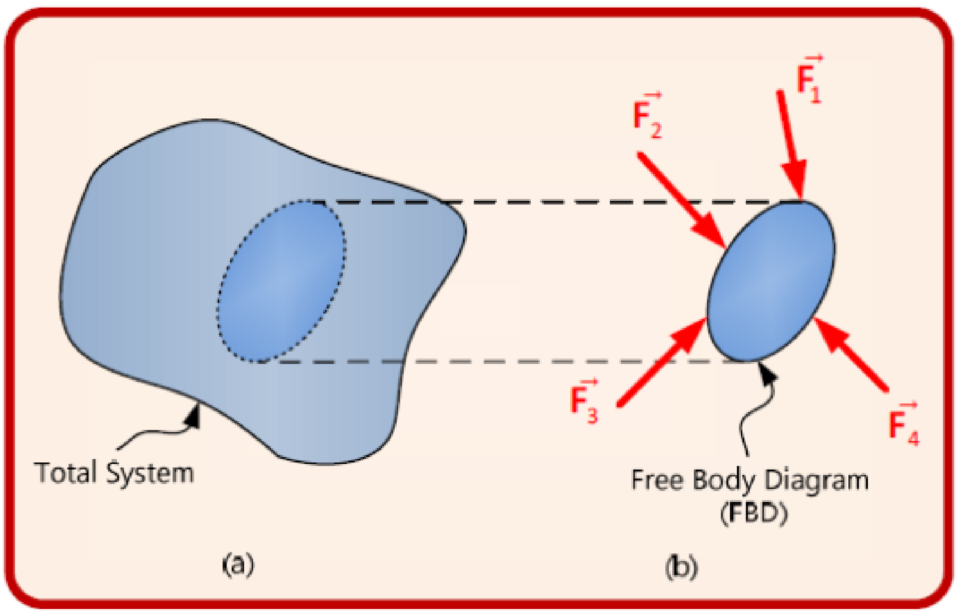
$$\sum \vec{\mathbf{M}} = \sum M_x \vec{\mathbf{i}} + \sum M_y \vec{\mathbf{j}} + \sum M_z \vec{\mathbf{k}} = \vec{0} \quad (\text{Vectorial Representation})$$

or

$$\begin{cases} \sum M_x = 0 \\ \sum M_y = 0 \\ \sum M_z = 0 \end{cases} \quad (\text{Scalar Representation})$$

Free-Body Diagram (FBD)

Free Body Diagram (FBD) is a sketch that shows a *body of interest* isolated from all interacting bodies. Once the *body of interest* is selected, the forces and moments exerted by *all* other bodies *on* the one being considered must be determined and shown in the diagram, see Figure

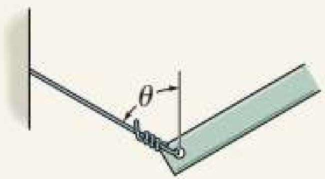
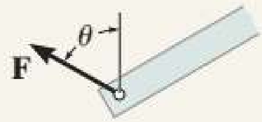
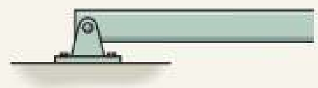
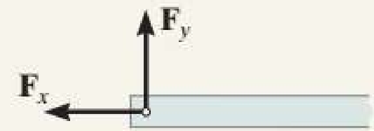
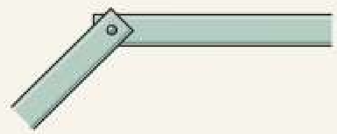
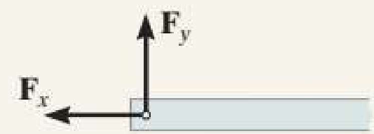
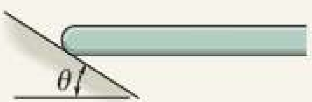
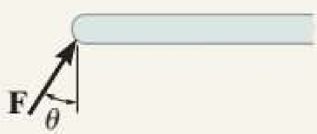
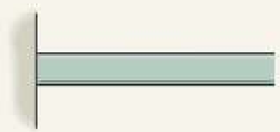
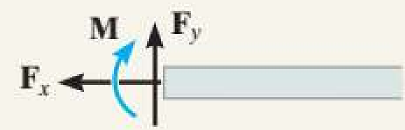
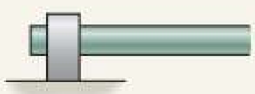
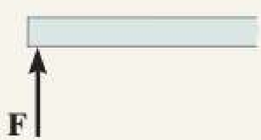
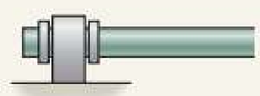
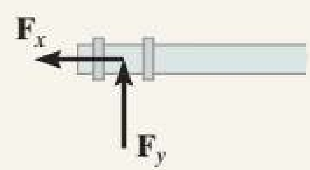


Free-Body Diagram

- The diagram establishes the directions of reference axes, provides a place to record the dimensions of the subsystem and the magnitudes and directions of the known forces, and helps in assuming the directions of unknown forces.
- The diagram simplifies your thinking because it provides a place to store one thought while proceeding to the next.
- The diagram provides a means of communicating your thoughts clearly and unambiguously to other people.
- Careful and complete construction of the diagram clarifies fuzzy thinking by bringing out various points that are not always apparent in the statement or in the geometry of the total problem. Thus, the diagram aids in understanding all facets of the problem.
- The diagram helps in the planning of a logical attack on the problem and in setting up the mathematical relations.
- The diagram helps in recording progress in the solution and in illustrating the methods used.
- The diagram allows others to follow your reasoning, showing *all* forces.

Reactions At Connection

TABLE 1–1

Type of connection	Reaction	Type of connection	Reaction
 Cable	 One unknown: F	 External pin	 Two unknowns: F_x, F_y
 Roller	 One unknown: F	 Internal pin	 Two unknowns: F_x, F_y
 Smooth support	 One unknown: F	 Fixed support	 Three unknowns: F_x, F_y, M
 Journal bearing	 One unknown: F	 Thrust bearing	 Two unknowns: F_x, F_y

Example 3.1

Draw a free body diagram for the beam shown in Fig. 4-2a.

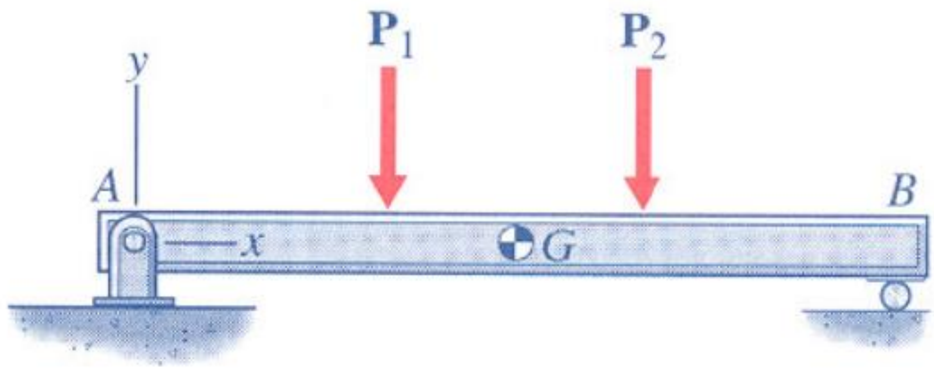


Figure (a)

Solution

Referring to Figure 4-2 (b):

- Two concentrated forces P_1 and P_2 are applied to the beam.
- The weight of the beam is represented by the force W, which has a line of action that passes through the center of gravity G of the beam.

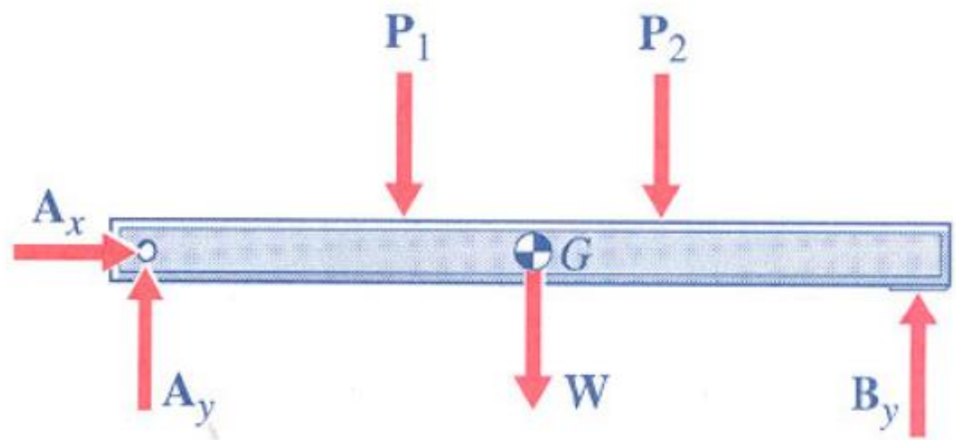


Figure (b)

- The beam is supported at the left end with a smooth pin and bracket and at the right end with a roller.
- The reaction of the left support is represented by the forces A_x and A_y.
- The reaction of the roller is represented by the force B_y, which acts normal to the surface of the beam.

Shear Force and Bending Moments in Beams

Figure shows a beam supported by **reactions** R_1 and R_2 and loaded by the **concentrated forces** F_1 , F_2, and F_3. If the beam is cut at some section located at $x = x_1$ and the left hand portion is removed as a free body, an internal shear force V and bending moment M must act on the cut surface to ensure equilibrium.

- The shear force is obtained by summing the forces on the isolated sections.
- The bending moment is the sum of the moments of the forces to the left of the section taken about an axis through the isolated section.

The sign conventions used for bending moment and shear force are shown in Figure .

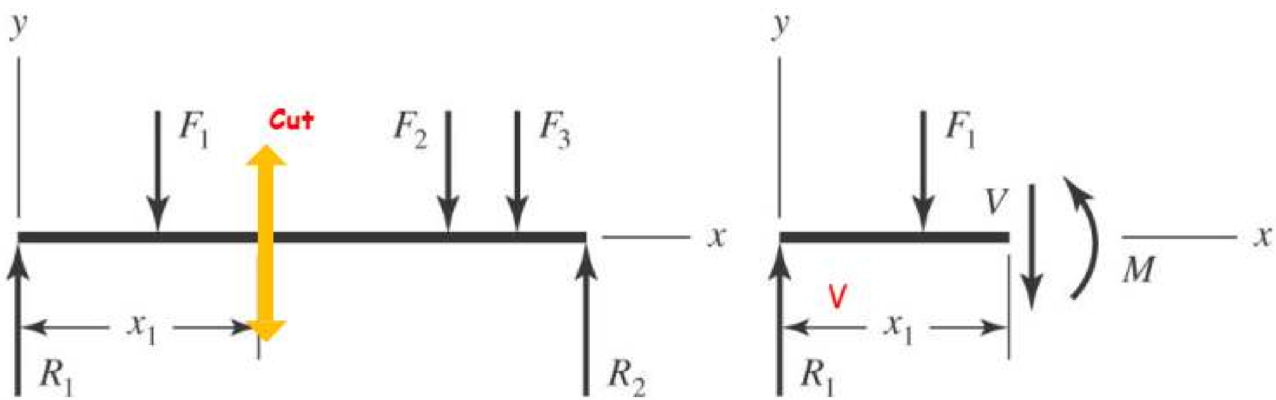
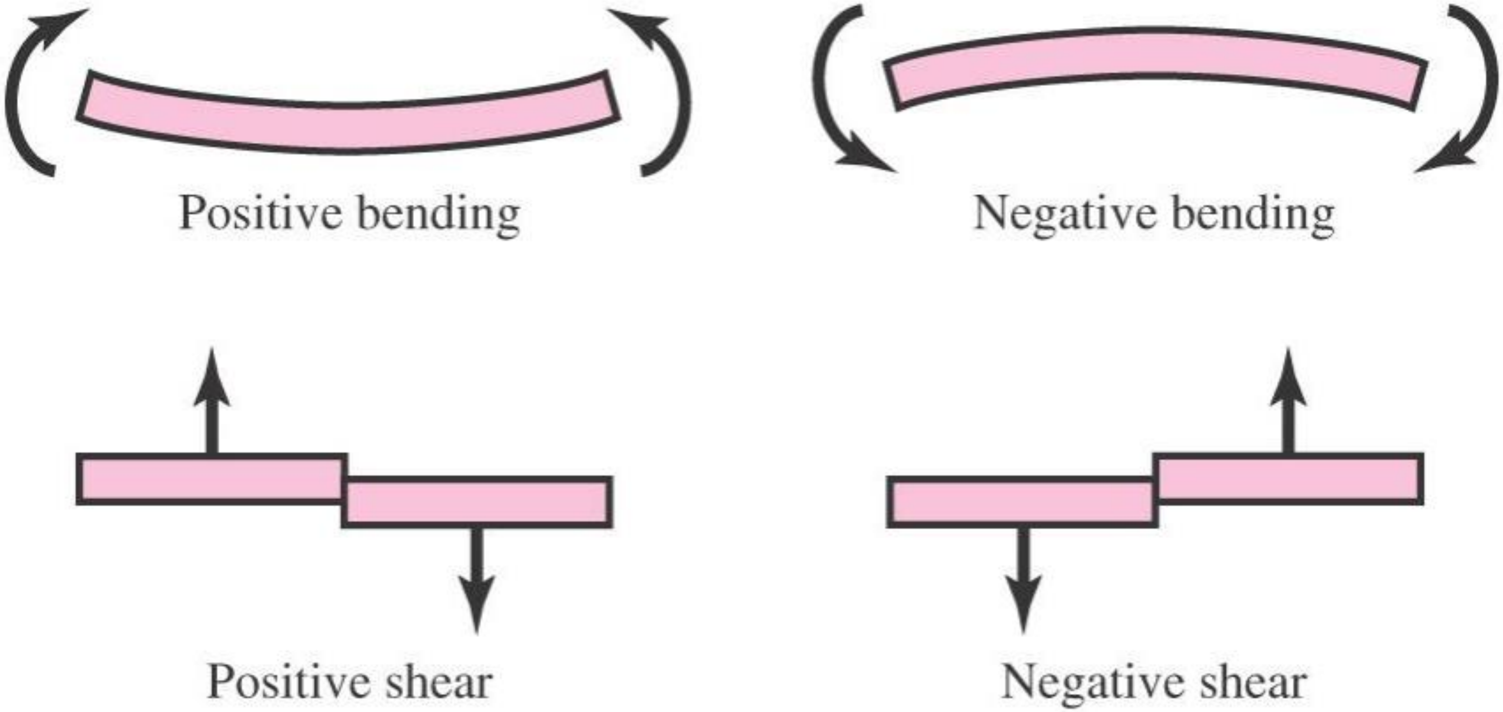


Figure FBD of simply supported beam with V and M shown in positive direction

Sign Conventions for Bending and Shear



Sign conventions for bending and shear.

Shear force and bending moment are related by

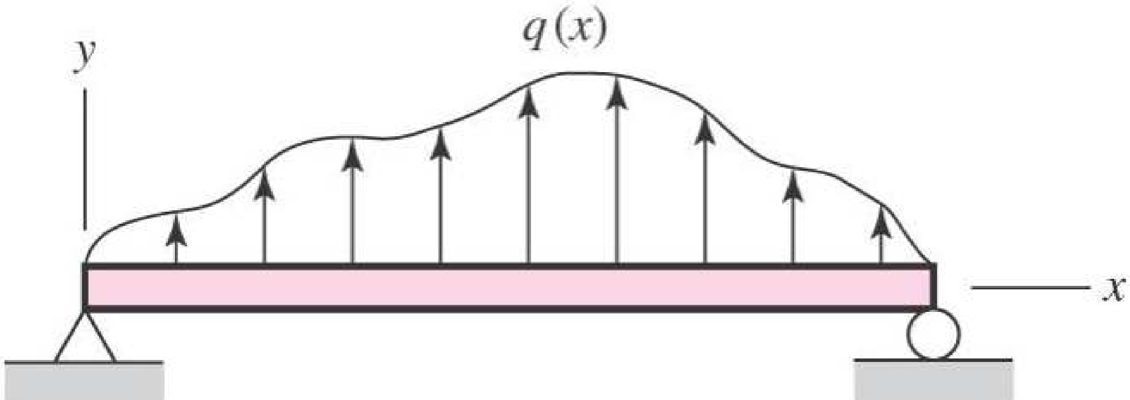
Shear Force

Bending moment

$$V = \frac{dM}{dx} \qquad (3\text{–}3)$$

Distributed Load on Beam

- Sometimes the bending is caused by a distributed load ***q*(*x*)**
- Distributed load ***q*(*x*)** called *load intensity*
- Units of force per unit length and is positive in the positive y direction



Distributed load on beam.

$$\frac{dV}{dx} = \frac{d^2M}{dx^2} = q \qquad (3\text{–}4)$$

Shear-Moment Diagrams

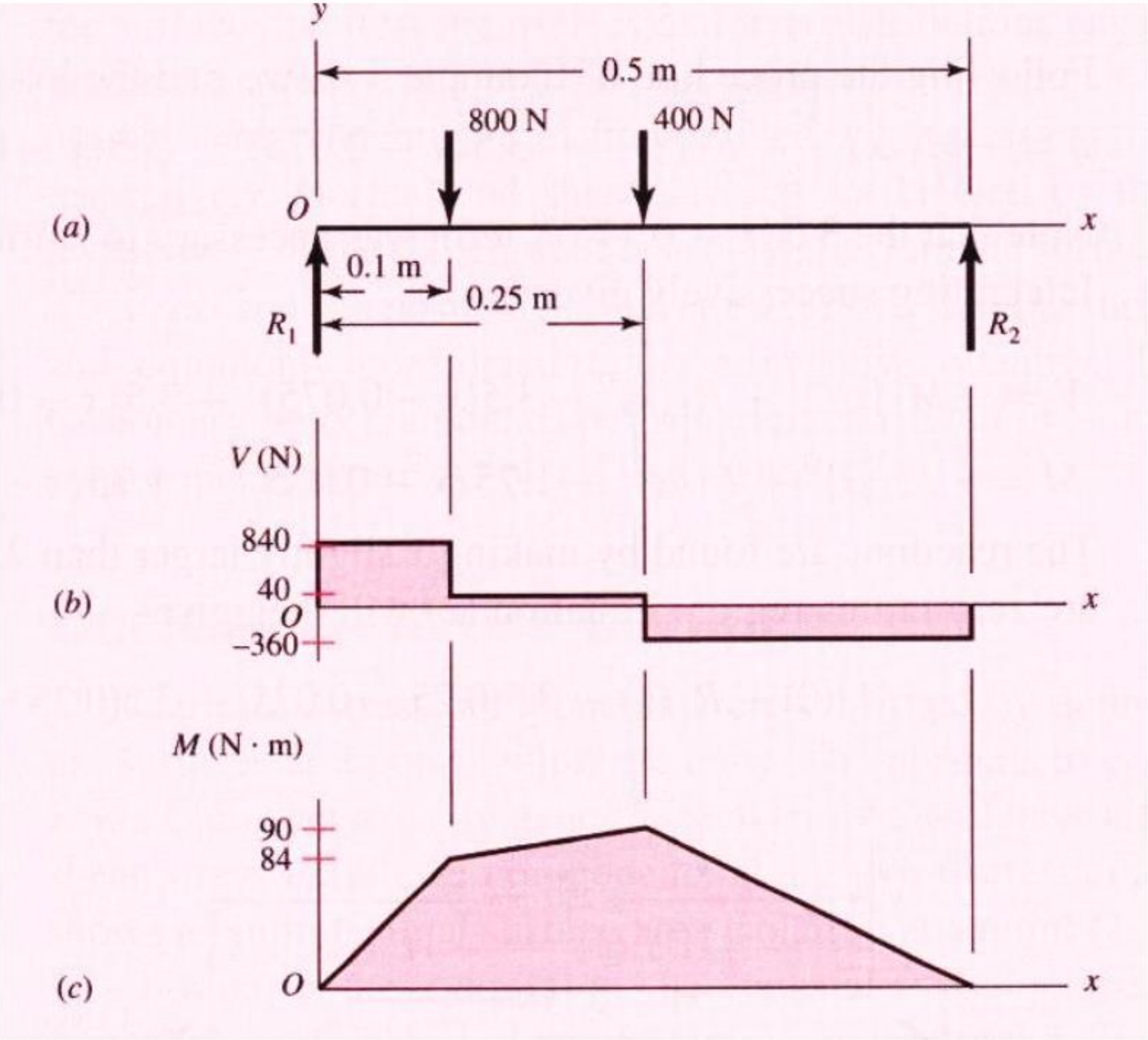


Fig. 3−5

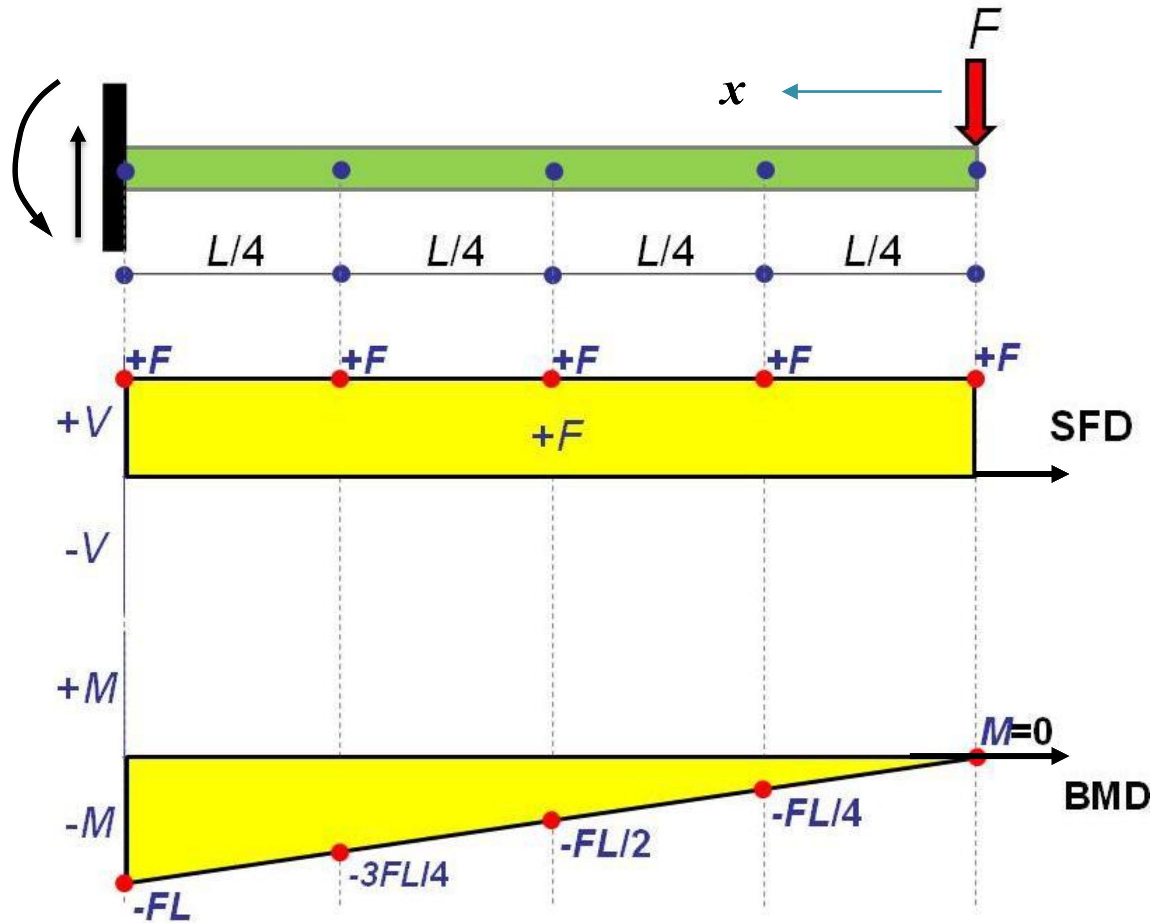
Shear force and bending moment diagrams (SFD and BMD)

At any distance x from V_0 and M_0 at x_0

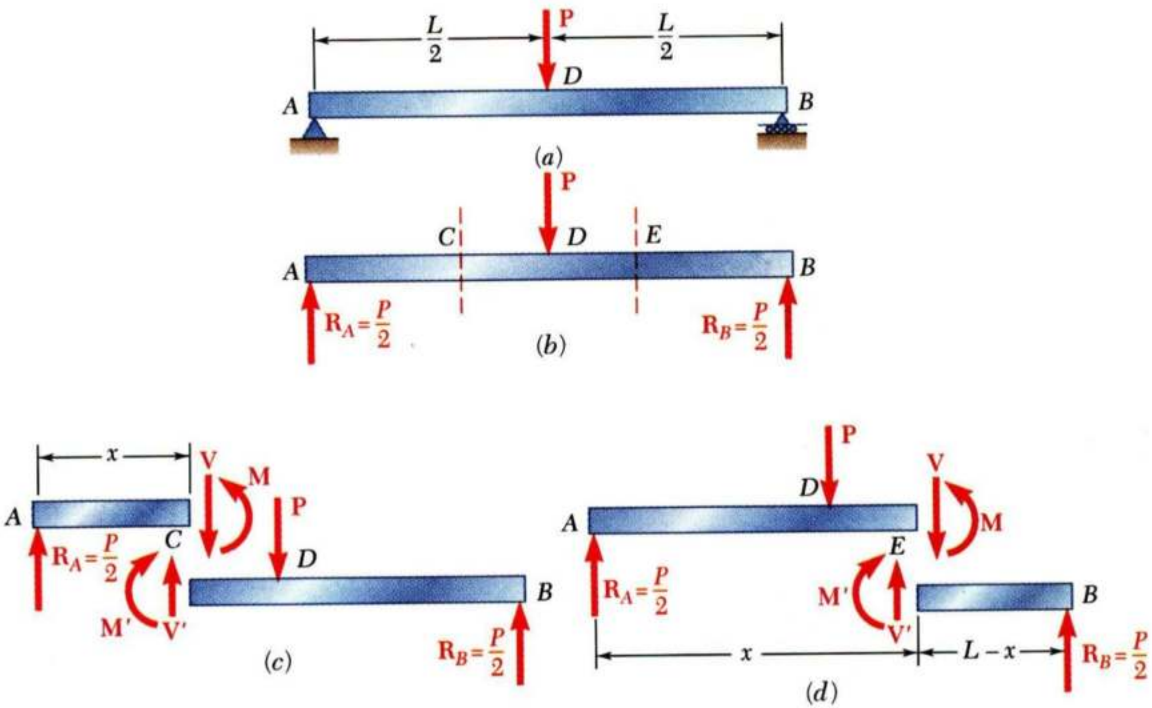
$V = V_0$ + (negative of area under the loading curve from x_0 to x)

$M = M_0$ + (area under the shear diagram from x_0 to x)

If there is no externally applied moment M_0 at $x_0 = 0$



SFD and BMD: Example



- Draw the SFD and BMD.

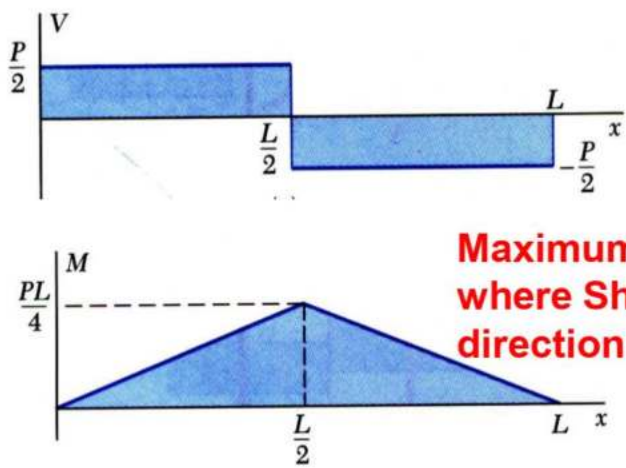
- Determine reactions at supports.

- Cut beam at *C* and consider member *AC*,

$$V = +P/2 \quad M = +Px/2$$

- Cut beam at *E* and consider member *EB*,

$$V = -P/2 \quad M = +P(L-x)/2$$



Maximum BM occurs where Shear changes the direction

- For a **beam** subjected to <u>**concentrated loads**</u>, **shear** is **constant** between **loading points** and **moment varies linearly**

Stress

- *Normal stress* is normal to a surface, designated by σ
- *Tangential shear stress* is tangent to a surface, designated by τ
- Normal stress acting outward on surface is *tensile stress*
- Normal stress acting inward on surface is *compressive stress*
- U.S. Customary units of stress are pounds per square inch (psi)
- SI units of stress are newtons per square meter (N/m^2)
- 1 N/m^2 = 1 pascal (Pa)

Uniformly Distributed Stresses

- Uniformly distributed stress distribution is often assumed for pure tension, pure compression, or pure shear.
- For tension and compression,

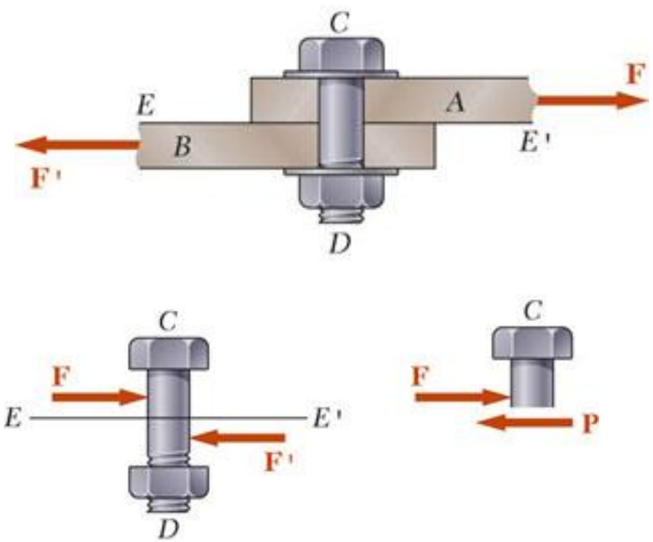
$$\sigma = \frac{F}{A} \tag{3–22}$$

- For direct shear (no bending present),

$$\tau = \frac{V}{A} \tag{3–23}$$

Shearing Stress Examples

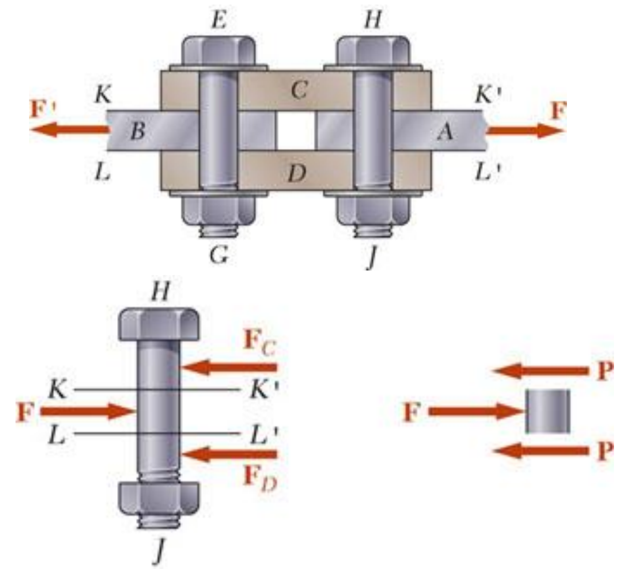
Single Shear



(a) (b)

$$\tau_{\text{ave}} = \frac{P}{A} = \frac{F}{A}$$

Double Shear



(a) (b)

$$\tau_{\text{ave}} = \frac{P}{A} = \frac{F}{2A}$$

Normal Stresses for Beams in Bending

- Straight beam in positive bending
- *x* axis is *neutral axis*
- *xz* plane is *neutral plane*
- *Neutral axis* is coincident with the *centroidal axis* of the cross section

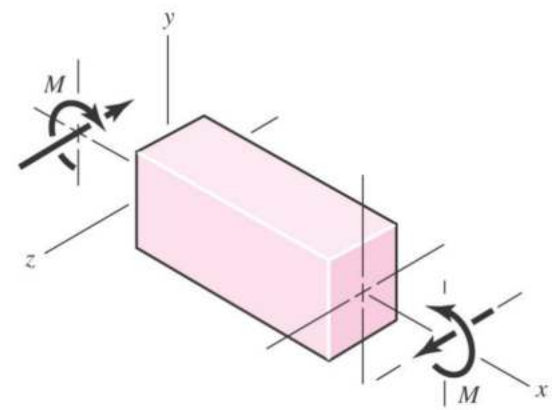


Fig. 3−13

Normal Stresses for Beams in Bending

- Bending stress varies linearly with distance from neutral axis, y

$$\sigma_x = -\frac{My}{I} \tag{3–24}$$

$$I = \int y^2 dA \tag{3–25}$$

- I is the *second-area moment* about the z axis

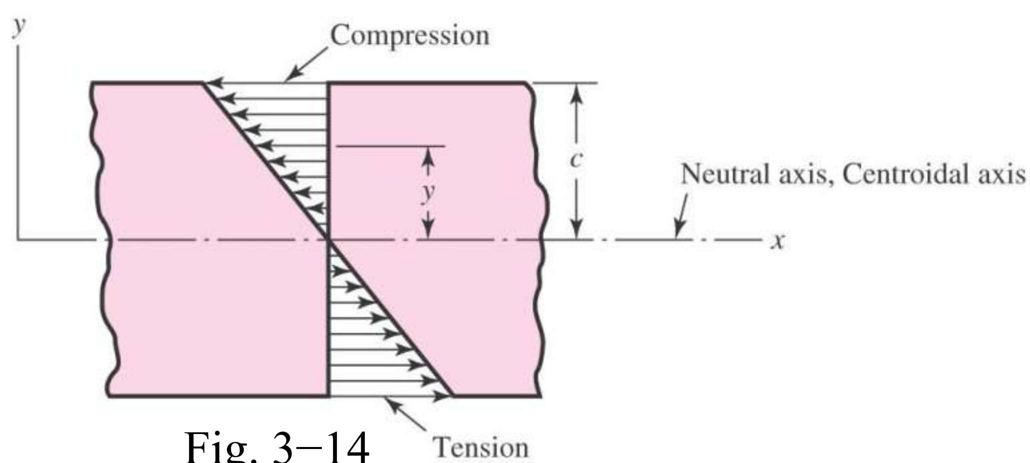


Fig. 3−14

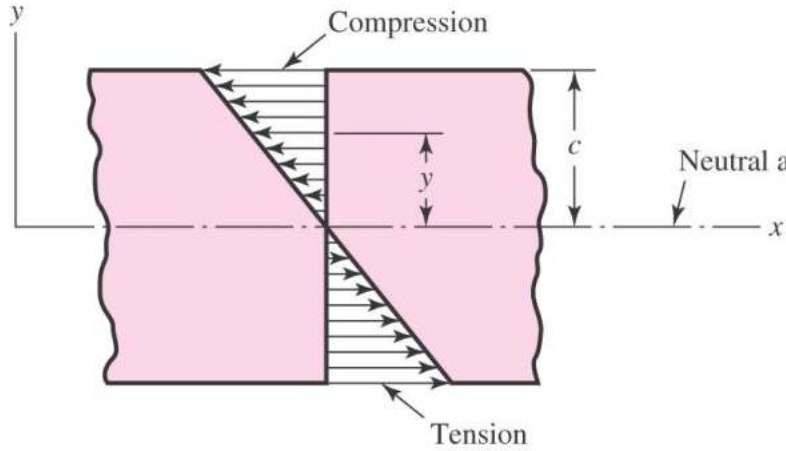
Normal Stresses for Beams in Bending

- Maximum bending stress is where y is greatest.

$$\sigma_{max} = \frac{Mc}{I} \qquad (3\text{–}26a)$$

$$\sigma_{max} = \frac{M}{Z} \qquad (3\text{–}26b)$$

- c is the magnitude of the greatest y
- $Z = I/c$ is the *section modulus*



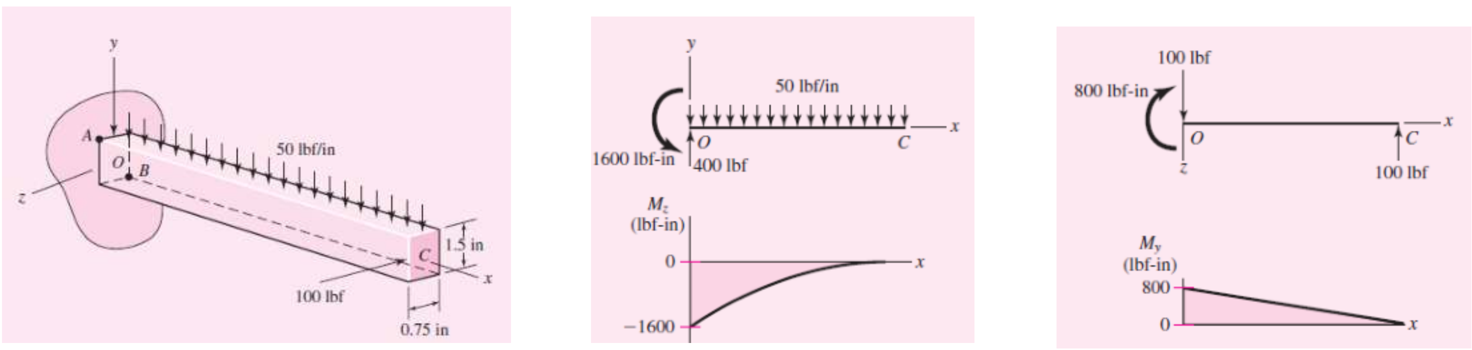
Assumptions for Normal Bending Stress

- Pure bending (though effects of axial, torsional, and shear loads are often assumed to have minimal effect on bending stress)
- Material is isotropic and homogeneous
- Material obeys Hooke's law
- Beam is initially straight with constant cross section
- Beam has axis of symmetry in the plane of bending
- Proportions are such that failure is by bending rather than crushing, wrinkling, or sidewise buckling
- Plane cross sections remain plane during bending

Two-Plane Bending

- Consider bending in both *xy* and *xz* planes
- Cross sections with one or two planes of symmetry only

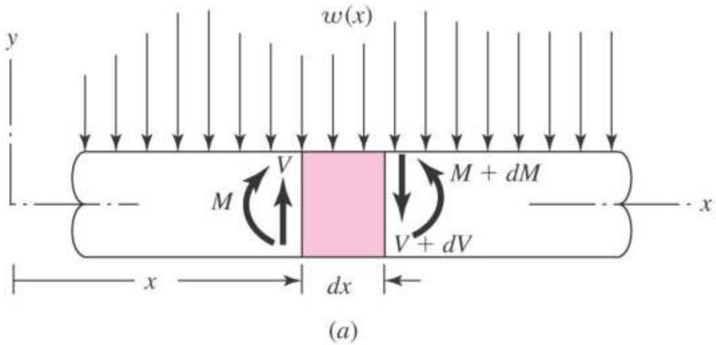
$$\sigma_x = -\frac{M_z y}{I_z} + \frac{M_y z}{I_y} \qquad (3\text{–}27)$$



- For solid circular cross section, the maximum bending stress is

$$\sigma_m = \frac{Mc}{I} = \frac{(M_y^2 + M_z^2)^{1/2}(d/2)}{\pi d^4/64} = \frac{32}{\pi d^3}(M_y^2 + M_z^2)^{1/2} \qquad (3\text{–}28)$$

Shear Stresses for Beams in Bending



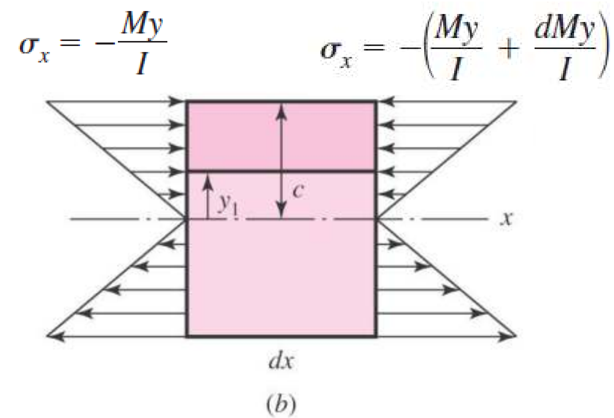
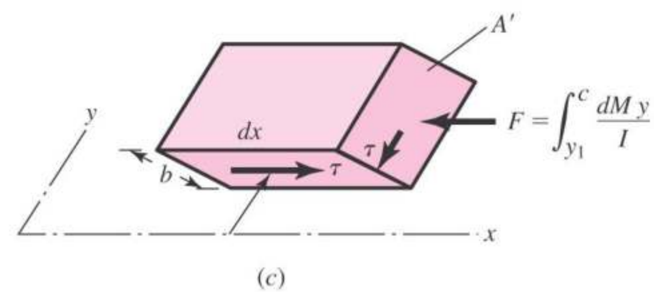


Fig. 3−17



$$\tau b\,dx = \int_{y_1}^{c} \frac{(dM)y}{I} dA$$

$$\tau = \frac{V}{Ib} \int_{y_1}^{c} y\,dA \qquad (3\text{–}29)$$

V = *dM*/*dx*

The integral is the first moment of the area *A′* with respect to the neutral axis

Transverse Shear Stress

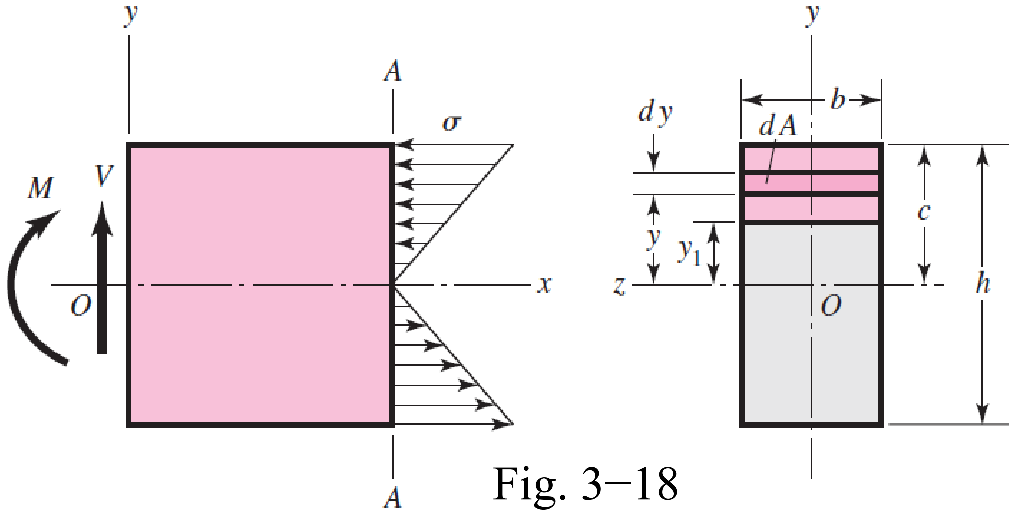


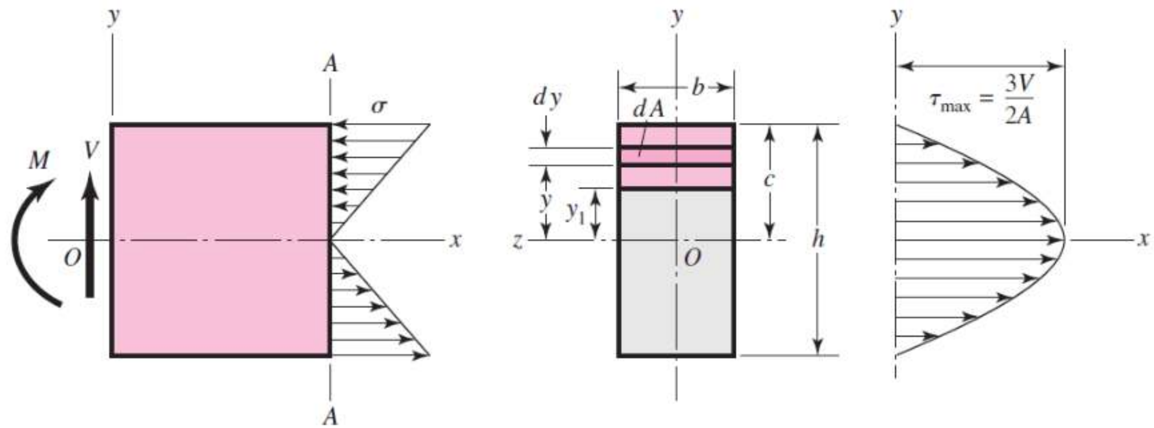
Fig. 3−18

$$Q = \int_{y_1}^{c} y\,dA = \bar{y}' A' \qquad (3\text{–}30)$$

$$\tau = \frac{VQ}{Ib} \qquad (3\text{–}31)$$

- Transverse shear stress is always accompanied with bending stress.

Transverse Shear Stress in a Rectangular Beam



(a) (b) (c)

$$Q = \int_{y_1}^{c} y\,dA = b\int_{y_1}^{c} y\,dy = \left.\frac{by^2}{2}\right|_{y_1}^{c} = \frac{b}{2}\left(c^2 - y_1^2\right)$$

$$\tau = \frac{VQ}{Ib} = \frac{V}{2I}\left(c^2 - y_1^2\right) \qquad I = \frac{Ac^2}{3}$$

$$\tau = \frac{3V}{2A}\left(1 - \frac{y_1^2}{c^2}\right) \qquad (3\text{–}33)$$

Maximum Values of Transverse Shear Stress

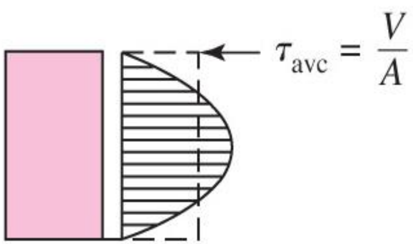
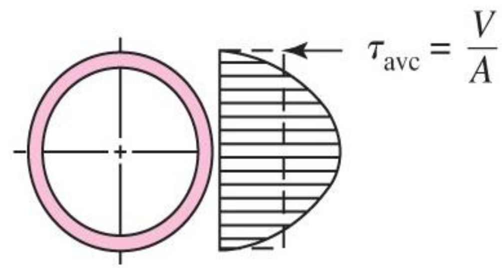
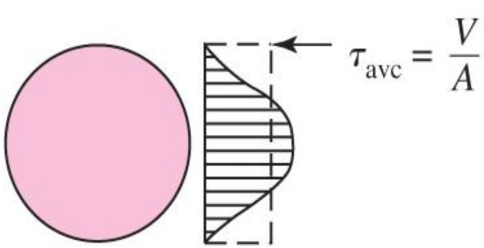
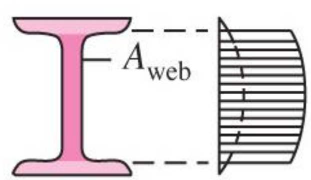
Beam Shape	Formula	Beam Shape	Formula
 Rectangular	$\tau_{max} = \frac{3V}{2A}$	 Hollow, thin-walled round	$\tau_{max} = \frac{2V}{A}$
 Circular	$\tau_{max} = \frac{4V}{3A}$	 Structural I beam (thin-walled)	$\tau_{max} \doteq \frac{V}{A_{web}}$

Table 3–2

Torsion

- *Torque vector* – a moment vector collinear with axis of a mechanical element
- A bar subjected to a torque vector is said to be in *torsion*

$$\theta = \frac{Tl}{GJ} \qquad J = \frac{\pi d^4}{32} \qquad (3\text{–}35)$$

- *Angle of twist*, in radians, for a solid round bar

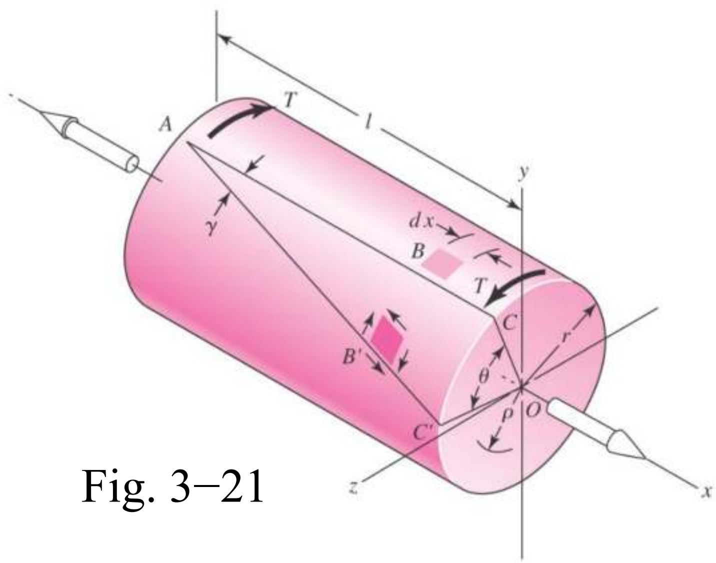


Fig. 3–21

where T = torque
l = length
G = modulus of rigidity
J = polar second moment of area

$$J = \frac{\pi d^4}{32}$$

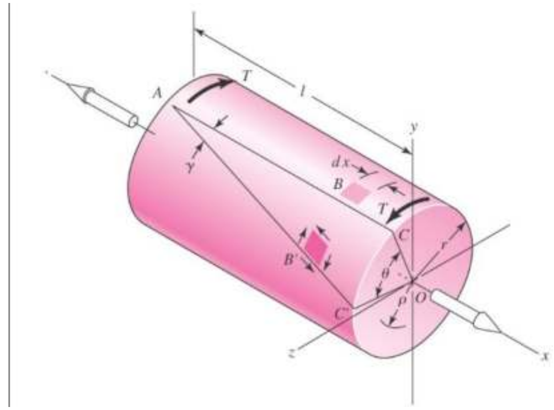
Torsional Shear Stress

- For round bar in torsion, torsional shear stress is proportional to the radius ρ

$$\tau = \frac{T\rho}{J} \qquad (3\text{–}36)$$

$$\tau_{\max} = \frac{Tr}{J} \qquad (3\text{–}37)$$

- Maximum torsional shear stress is at the outer surface

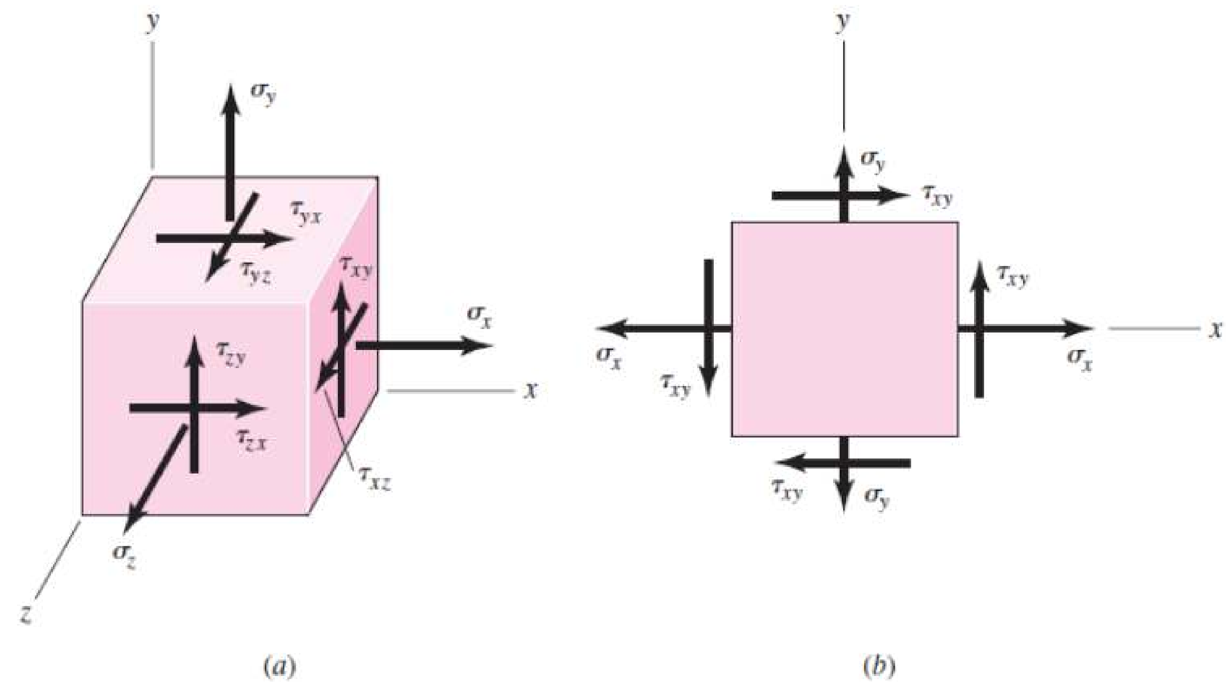


$$J = \frac{\pi d^4}{32}$$

Stress element

Figure 3–8

(*a*) General three-dimensional stress. (*b*) Plane stress with "cross-shears" equal.



- Represents stress *at a point*
- Coordinate directions are arbitrary
- Choosing coordinates which result in zero shear stress will produce principal stresses

Cartesian Stress Components

- Defined by three mutually orthogonal surfaces at a point within a body
- Each surface can have normal and shear stress
- Shear stress is often resolved into perpendicular components
- First subscript indicates direction of surface normal
- Second subscript indicates direction of shear stress

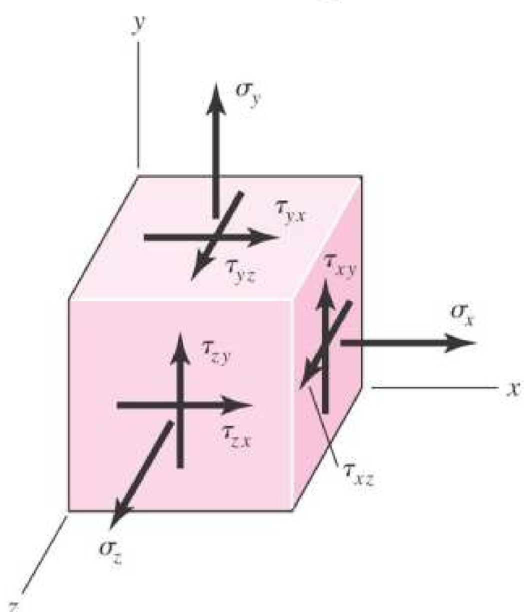


Fig. 3−8 (a)

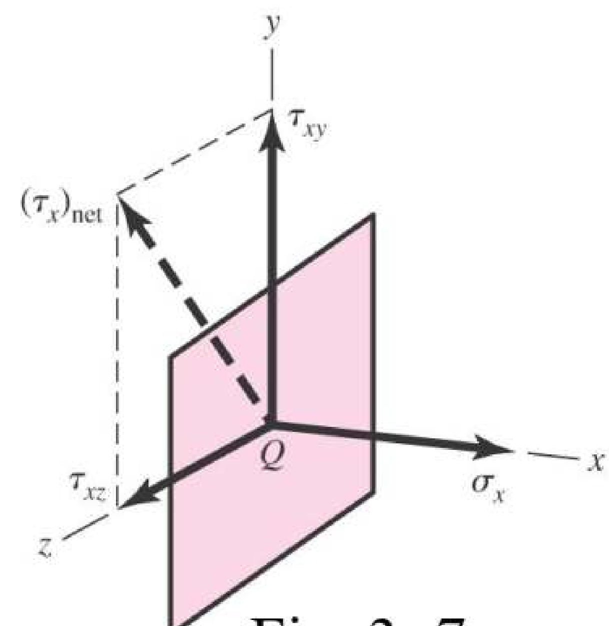


Fig. 3−7

Plane-Stress Transformation Equations

- Cutting plane stress element at an arbitrary angle and balancing stresses gives *plane-stress transformation equations*

$$\sigma = \frac{\sigma_x + \sigma_y}{2} + \frac{\sigma_x - \sigma_y}{2}\cos 2\phi + \tau_{xy}\sin 2\phi \qquad (3\text{–}8)$$

$$\tau = -\frac{\sigma_x - \sigma_y}{2}\sin 2\phi + \tau_{xy}\cos 2\phi \qquad (3\text{–}9)$$

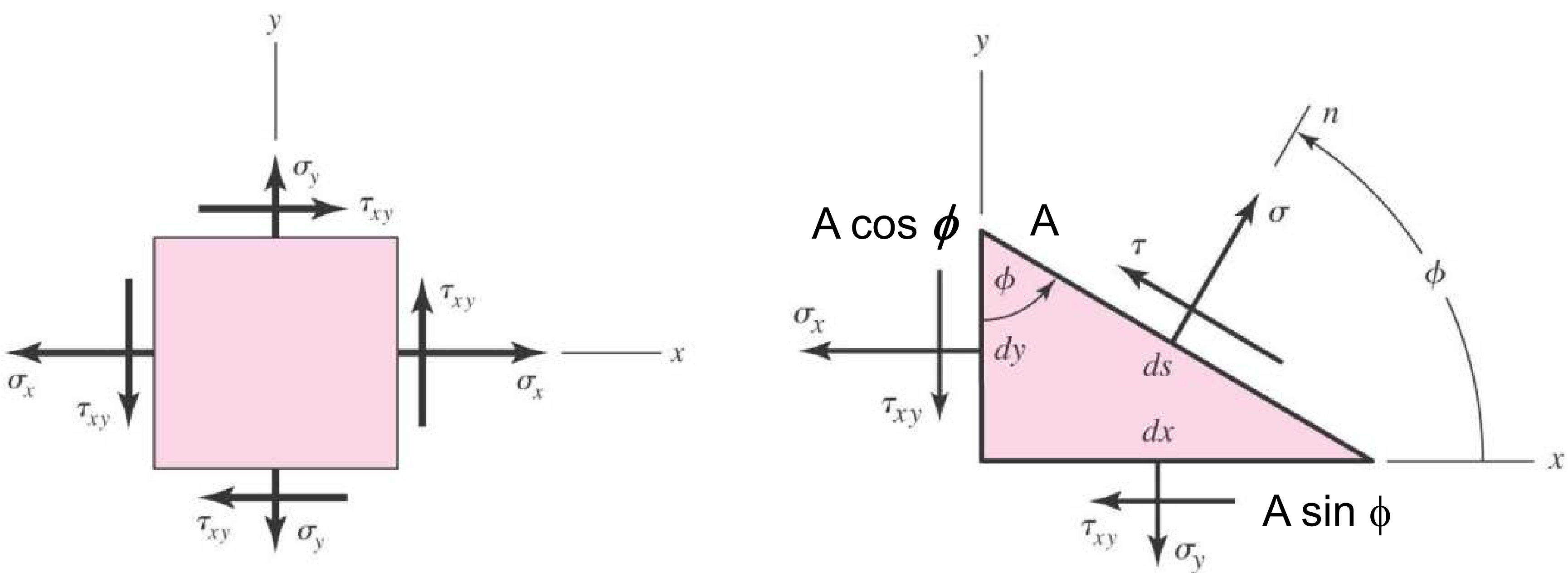


Fig. 3–9

Principal Stresses for Plane Stress

- Differentiating Eq. (3-8) with respect to ϕ and setting equal to zero maximizes σ and gives

$$\tan 2\phi_p = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} \qquad (3\text{–}10)$$

- The two values of $2\phi_p$ are the *principal directions.*
- The stresses in the principal directions are the *principal stresses.*
- The principal direction surfaces have zero shear stresses. $\tau = 0,$
- Substituting Eq. (3-10) into Eq. (3-8) gives expression for the non-zero principal stresses.

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \qquad (3\text{–}13)$$

- Note that there is a third principal stress, equal to zero for plane stress.

Extreme-value Shear Stresses for Plane Stress

- Performing similar procedure with shear stress in Eq. (3-9), the maximum shear stresses are found to be on surfaces that are ±45° from the principal directions.

$$\tan 2\phi_p = \frac{2\tau_{xy}}{\sigma_x - \sigma_y}$$

$$\tan 2\phi_s = -\frac{\sigma_x - \sigma_y}{2\tau_{xy}}$$

- The two extreme-value shear stresses are

$$\tau_1, \tau_2 = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \qquad (3\text{–}14)$$

$$\sigma = \frac{\sigma_x + \sigma_y}{2}$$

At max. shear stress

Equation $\tan 2\phi_s = -\frac{\sigma_x - \sigma_y}{2\tau_{xy}}$ defines the two values of $2\varphi s$ at which the shear stress τ reaches an extreme value. The angle between the two surfaces containing the maximum shear stresses is 90°.

Mohr's Circle Diagram

- A graphical method for visualizing the stress state at a point
- Represents relation between x-y stresses and principal stresses
- Parametric relationship between σ and τ (with 2ϕ as parameter)
- Relationship is a circle with center at

$$C = (\sigma, \tau) = [(\sigma_x + \sigma_y)/2, 0]$$

and radius of

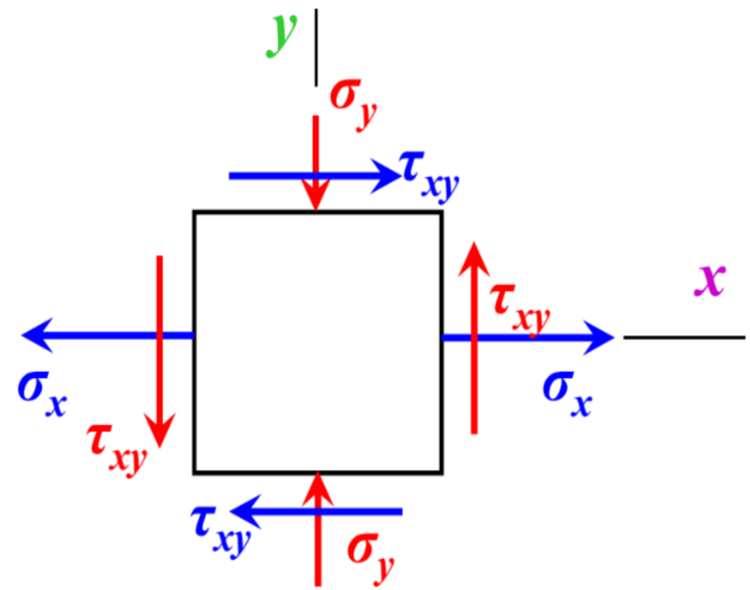
$$R = \sqrt{\left[\frac{\sigma_x - \sigma_y}{2}\right]^2 + \tau_{xy}^2}$$

Drawing a Mohr Circle

Sign Convention

- Normal Stress (σ)
  - Tensile are +ve
  - Compression is -ve

- Shear Stress (τ)
  - Shear stresses tending to rotate the element clockwise (cw) are plotted above the σ axis
  - Shear stresses tending to rotate the element counterclockwise (ccw) are plotted below the σ axis



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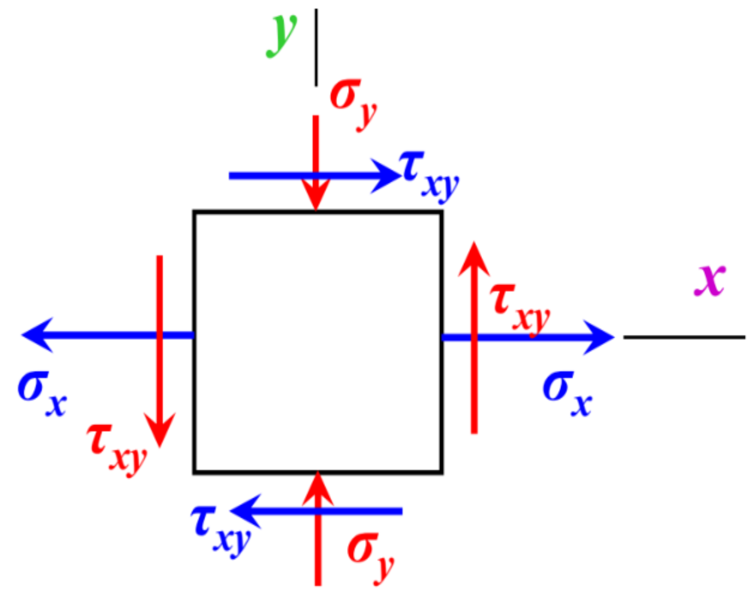
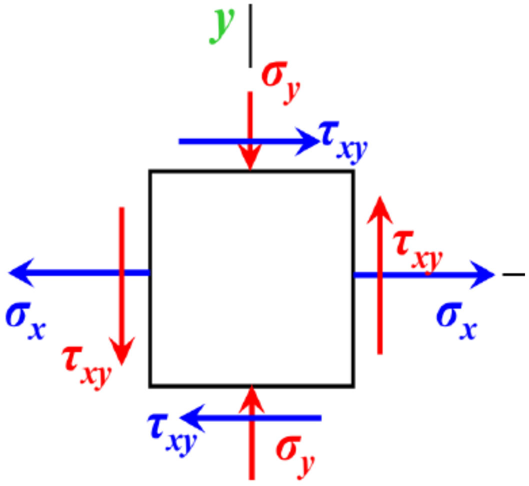



y
σy
τxy
τxy
σx
σx
τxy
τxy
σy

Mohr's Circle Diagram

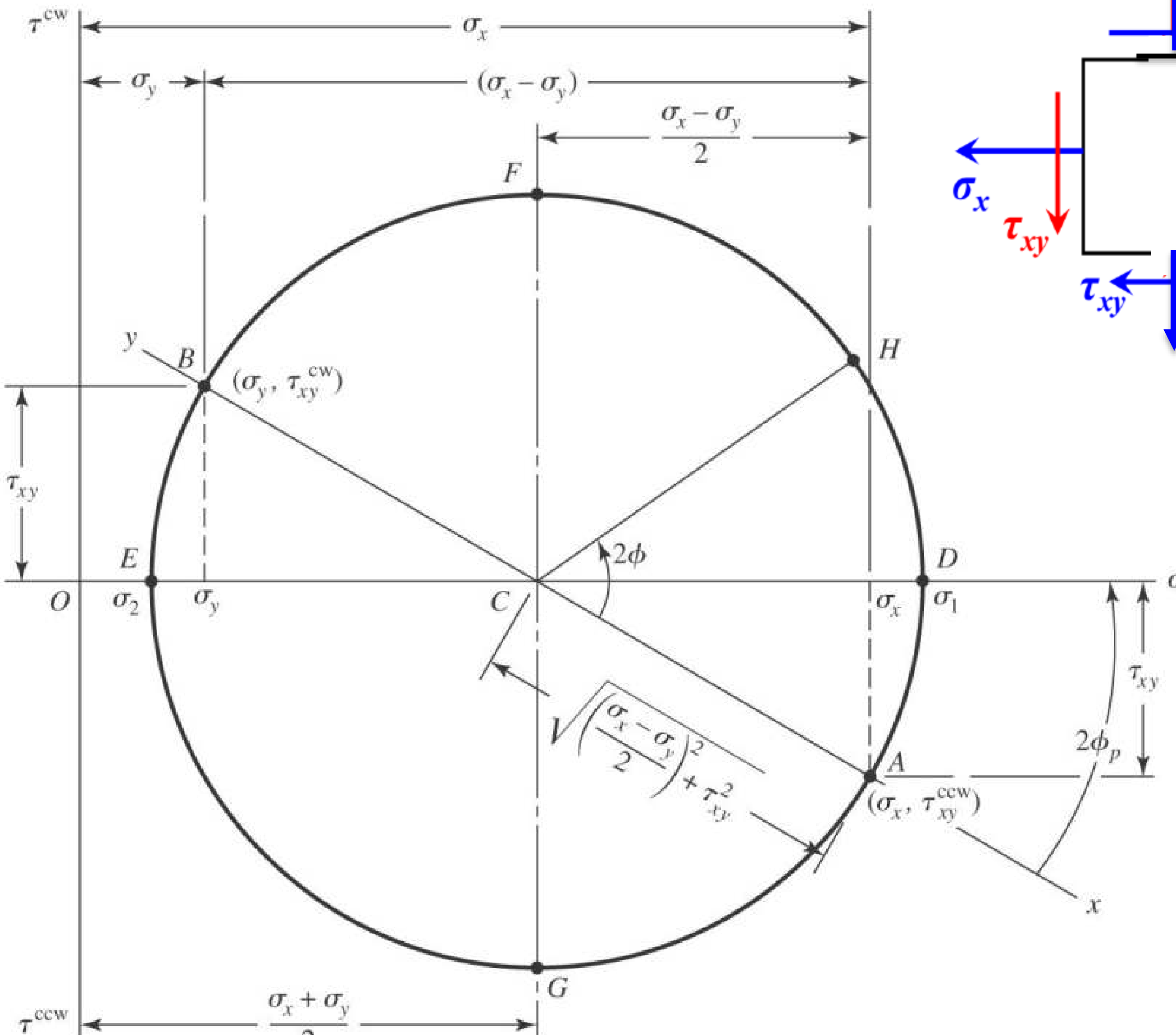
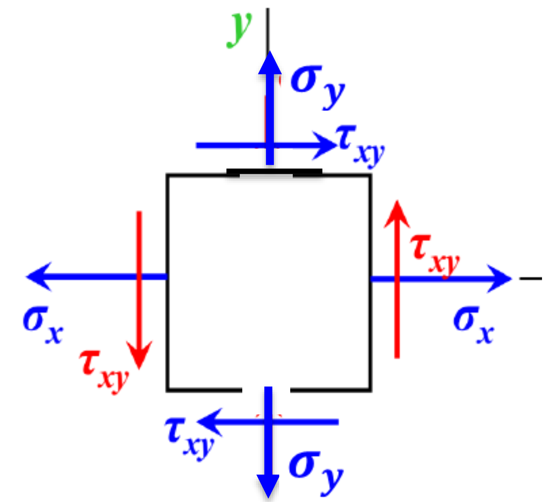


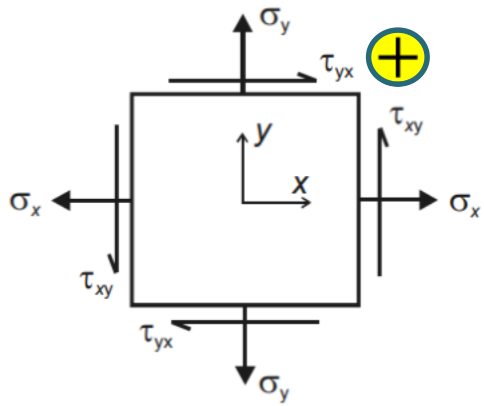
Fig. 3–10

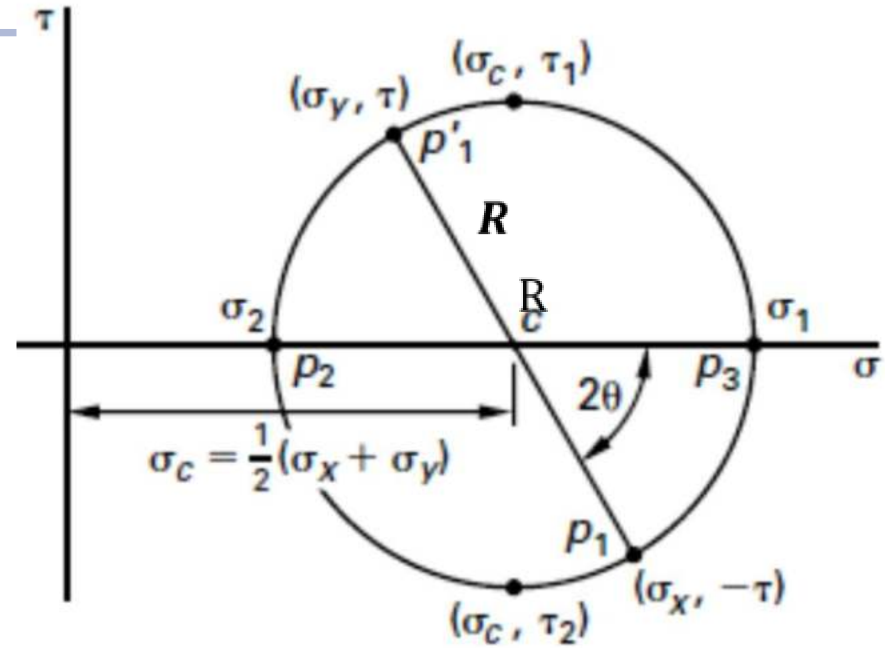


Five steps to construct Mohr’s Circle

Plane stress element in 2D

σ_x, σ_y
$\tau_{xy} = \tau_{yx}$
$\sigma_z = 0$





Five simplified steps to construct Mohr’s circle

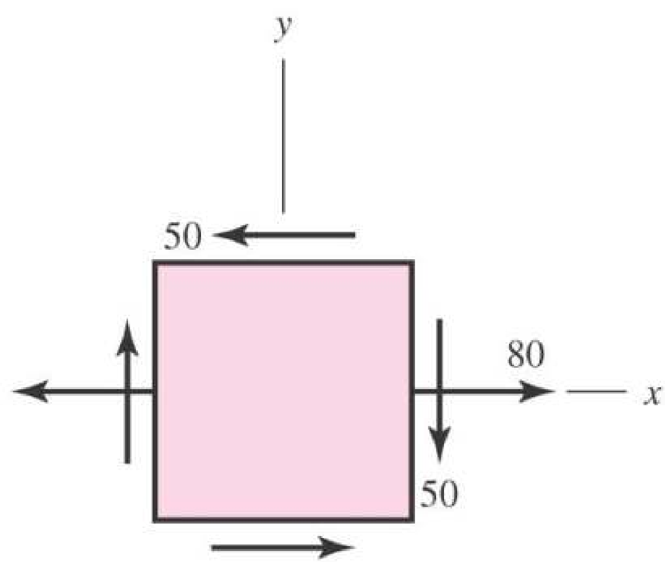
1. Determine the applied stresses **($\boldsymbol{\sigma_x}$, $\boldsymbol{\sigma_y}$, $\boldsymbol{\tau_{xy}}$)**.
2. Draw a set of $\boldsymbol{\sigma} - \boldsymbol{\tau}$ axes. *Locate the points* $\boxed{P_1}$ $(\boldsymbol{\sigma_x}, -\boldsymbol{\tau_{xy}})$ & $\boxed{P_1'}$ $(\boldsymbol{\sigma_x}, \boldsymbol{\tau_{xy}})$
3. Locate the center: $\boldsymbol{\sigma_c = 1/2(\sigma_x + \sigma_y)}$
4. Find the radius, $\boldsymbol{R}$ (or $\boldsymbol{\tau_{max}}$): , $\boldsymbol{\tau_{max}} = \boldsymbol{R} = \sqrt{\left(\frac{\boldsymbol{\sigma_x - \sigma_y}}{\mathbf{2}}\right)^{\mathbf{2}} + \boldsymbol{\tau_{xy}}^{\mathbf{2}}}$
5. Draw Mohr’s circle.

Example 3-4

A stress element has $\sigma_x = 80$ MPa and $\tau_{xy} = 50$ MPa cw, as shown in Fig. 3–11*a*.

(*a*) Using Mohr's circle, find the principal stresses and directions, and show these on a stress element correctly aligned with respect to the xy coordinates. Draw another stress element to show τ_1 and τ_2, find the corresponding normal stresses, and label the drawing completely.

(*b*) Repeat part *a* using the transformation equations only.



(*a*)

Fig. 3−11

Example 3-4

(a) In the semigraphical approach used here, we first make an approximate freehand sketch of Mohr's circle and then use the geometry of the figure to obtain the desired information.

Draw the σ and τ axes first (Fig. 3–11b) and from the x face locate $\sigma_x = 80$ MPa along the σ axis. On the x face of the element, we see that the shear stress is 50 MPa in the cw direction. Thus, for the x face, this establishes point A $(80, 50^{\text{cw}})$ MPa. Corresponding to the y face, the stress is $\sigma = 0$ and $\tau = 50$ MPa in the ccw direction. This locates point B $(0, 50^{\text{ccw}})$ MPa. The line AB forms the diameter of the required circle, which can now be drawn. The intersection of the circle with the σ axis defines σ_1 and σ_2 as shown. Now, noting the triangle ACD, indicate on the sketch the length of the legs AD and CD as 50 and 40 MPa, respectively. The length of the hypotenuse AC is

$$\tau_1 = \sqrt{(50)^2 + (40)^2} = 64.0 \text{ MPa}$$

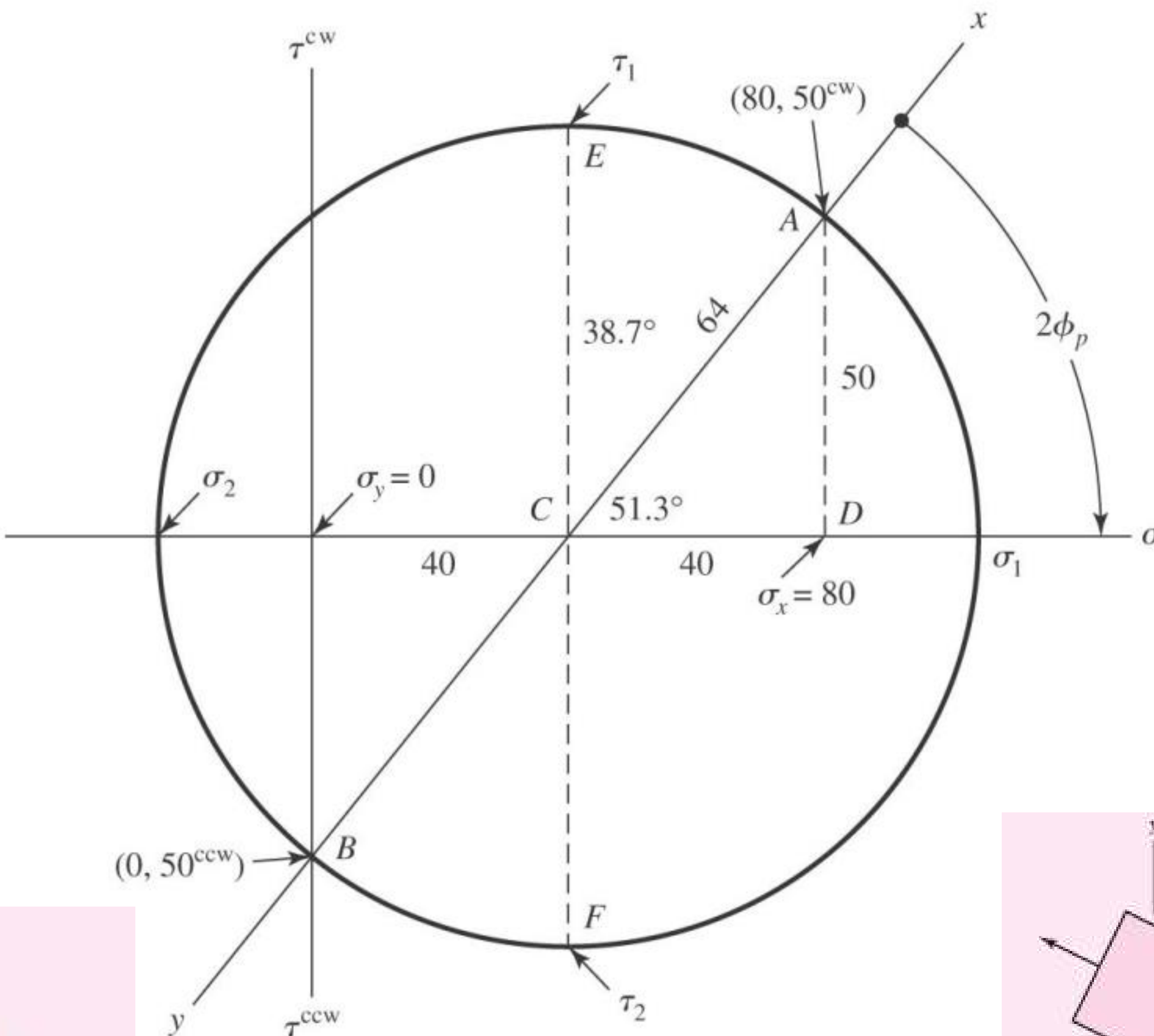
and this should be labeled on the sketch too. Since intersection C is 40 MPa from the origin, the principal stresses are now found to be

$$\sigma_1 = 40 + 64 = 104 \text{ MPa} \qquad \text{and} \qquad \sigma_2 = 40 - 64 = -24 \text{ MPa}$$

The angle 2ϕ from the x axis cw to σ_1 is

$$2\phi_p = \tan^{-1} \tfrac{50}{40} = 51.3^\circ$$

Example 3-4



(b)

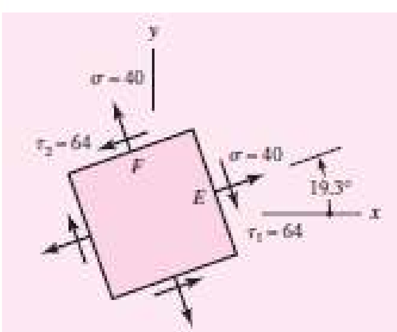
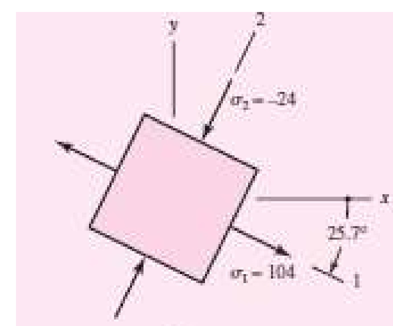


Fig. 3−11



Problem 1

For each of the plane stress states listed below, draw a Mohr's circle diagram properly labeled, find the principal normal and shear stresses, and determine the angle from the x axis to σ_1. Draw stress elements as in Fig. 3–11*c* and *d* and label all details.

(*a*) $\sigma_x = 20\,\text{kpsi}$, $\sigma_y = -10\,\text{kpsi}$, $\tau_{xy} = 8\,\text{kpsi}$ cw

(*b*) $\sigma_x = 16\,\text{kpsi}$, $\sigma_y = 9\,\text{kpsi}$, $\tau_{xy} = 5\,\text{kpsi}$ ccw

(*c*) $\sigma_x = 10\,\text{kpsi}$, $\sigma_y = 24\,\text{kpsi}$, $\tau_{xy} = 6\,\text{kpsi}$ ccw

(*d*) $\sigma_x = -12\,\text{kpsi}$, $\sigma_y = 22\,\text{kpsi}$, $\tau_{xy} = 12\,\text{kpsi}$ cw

(a)

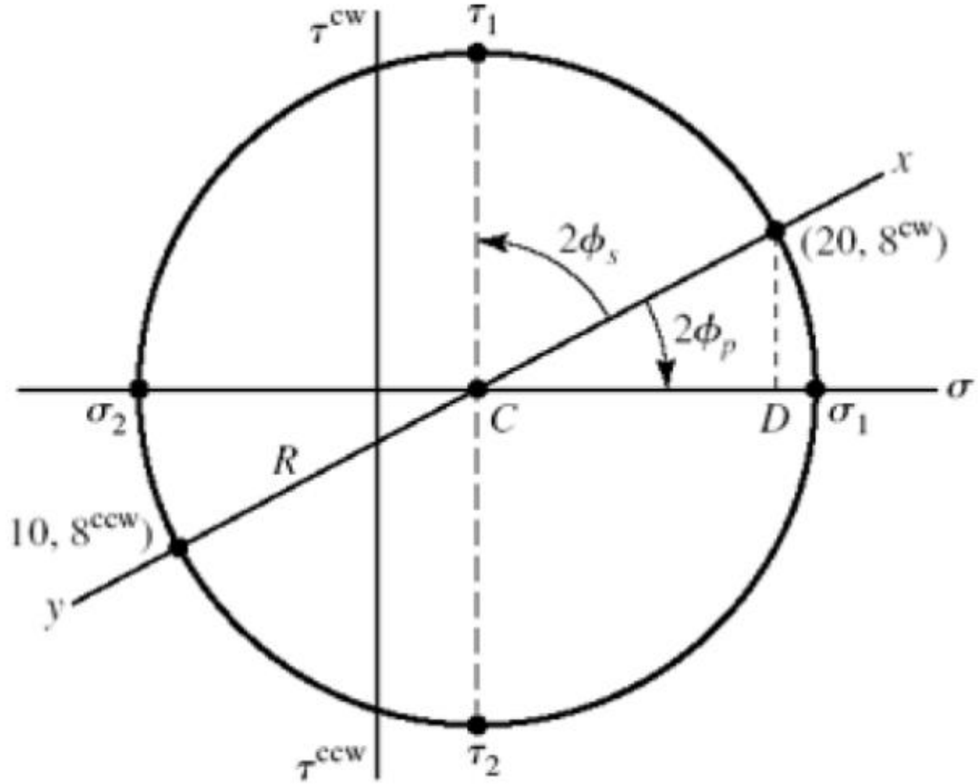
$$C=\frac{20-10}{2}=5\text{ kpsi}$$

$$CD=\frac{20+10}{2}=15\text{ kpsi}$$

$$R=\sqrt{15^2+8^2}=17\text{ kpsi}$$

$$\sigma_1=5+17=22\text{ kpsi}$$

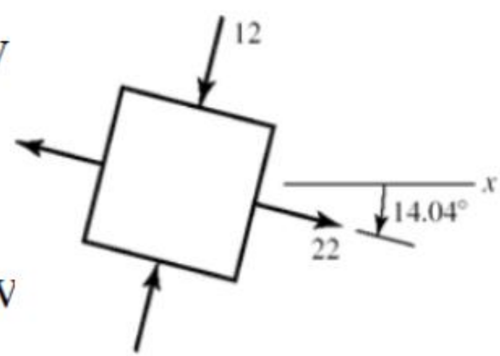
$$\sigma_2=5-17=-12\text{ kpsi}$$

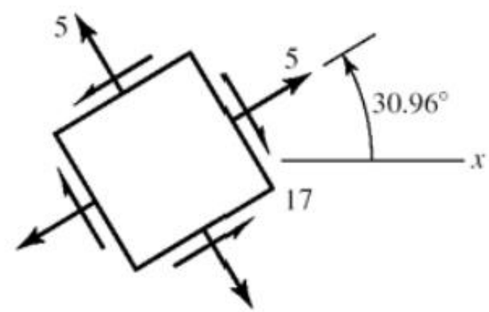


$$\phi_p=\frac{1}{2}\tan^{-1}\left(\frac{8}{15}\right)=14.04^\circ\text{ cw}$$

$$\tau_1=R=17\text{ kpsi}$$

$$\phi_s=45^\circ-14.04^\circ=30.96^\circ\text{ ccw}$$





(b)

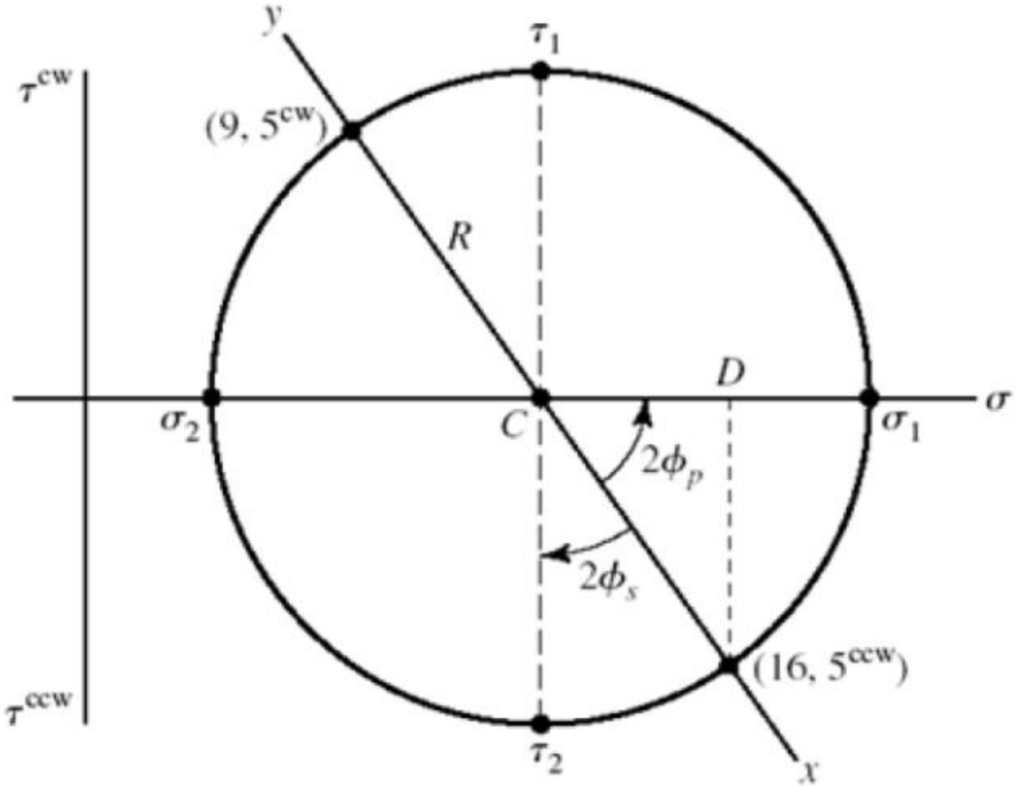
$$C = \frac{9+16}{2} = 12.5 \text{ kpsi}$$

$$CD = \frac{16-9}{2} = 3.5 \text{ kpsi}$$

$$R = \sqrt{5^2 + 3.5^2} = 6.10 \text{ kpsi}$$

$$\sigma_1 = 12.5 + 6.1 = 18.6 \text{ kpsi}$$

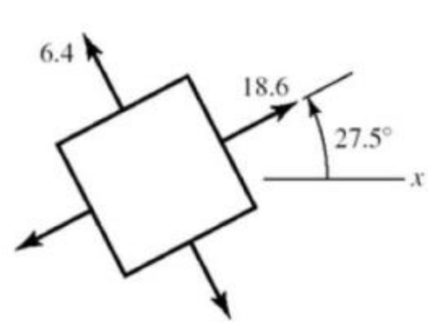
$$\sigma_2 = 12.5 - 6.1 = 6.4 \text{ kpsi}$$

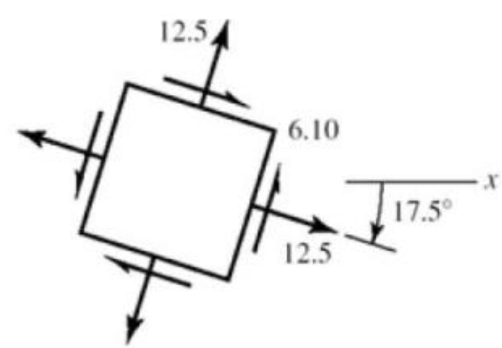


$$\phi_p = \frac{1}{2}\tan^{-1}\left(\frac{5}{3.5}\right) = 27.5^\circ \text{ ccw}$$

$$\tau_1 = R = 6.10 \text{ kpsi}$$

$$\phi_s = 45^\circ - 27.5^\circ = 17.5^\circ \text{ cw}$$





(c)

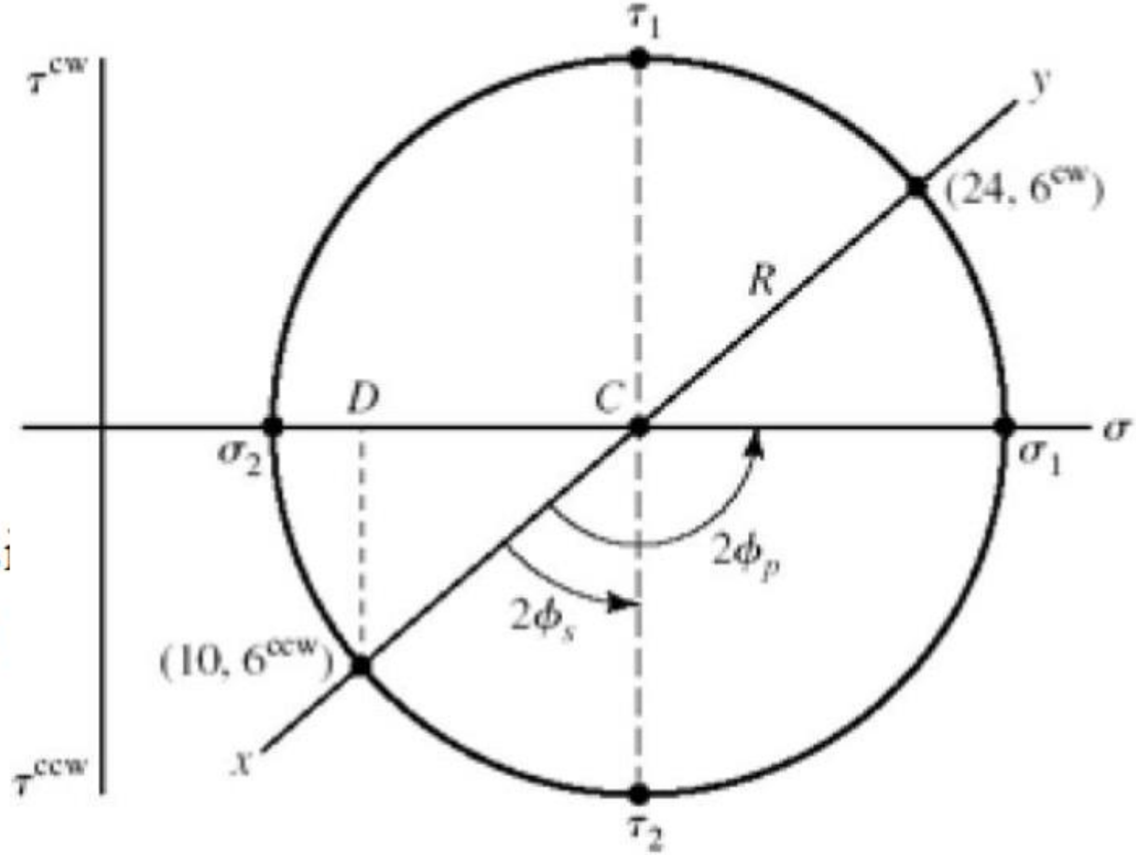
$$C = \frac{24+10}{2} = 17 \text{ kpsi}$$

$$CD = \frac{24-10}{2} = 7 \text{ kpsi}$$

$$R = \sqrt{7^2 + 6^2} = 9.22 \text{ kpsi}$$

$$\sigma_1 = 17 + 9.22 = 26.22 \text{ kpsi}$$

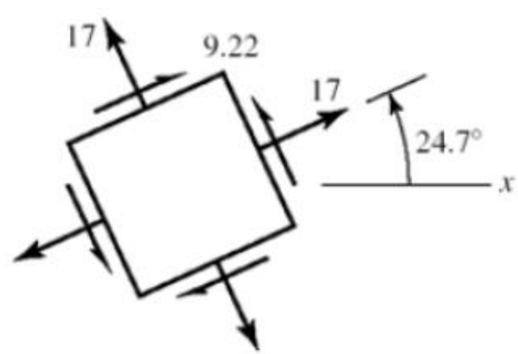
$$\sigma_2 = 17 - 9.22 = 7.78 \text{ kpsi}$$

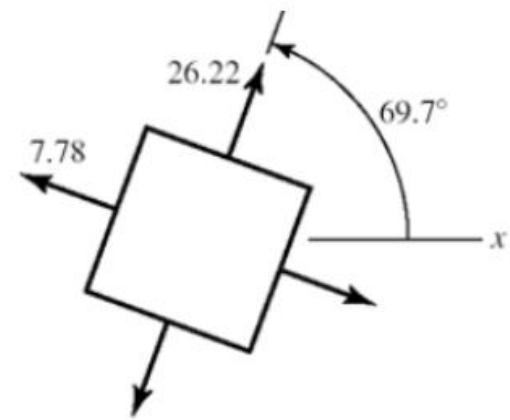


$$\phi_p = \frac{1}{2}\left[90^\circ + \tan^{-1}\left(\frac{7}{6}\right)\right] = 69.7^\circ \text{ ccw}$$

$$\tau_1 = R = 9.22 \text{ kpsi}$$

$$\phi_s = 69.7^\circ - 45^\circ = 24.7^\circ \text{ ccw}$$





(d)

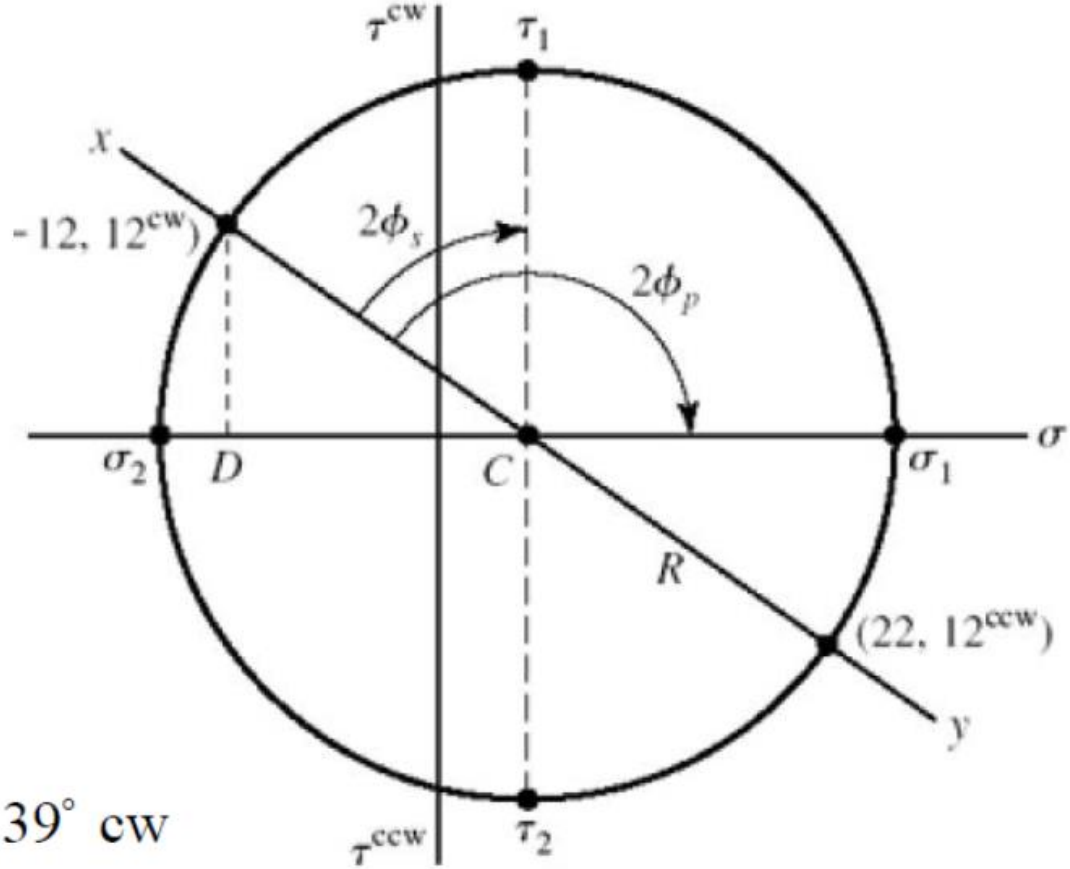
$$C = \frac{-12+22}{2} = 5 \text{ kpsi}$$

$$CD = \frac{12+22}{2} = 17 \text{ kpsi}$$

$$R = \sqrt{17^2 + 12^2} = 20.81 \text{ kpsi}$$

$$\sigma_1 = 5 + 20.81 = 25.81 \text{ kpsi}$$

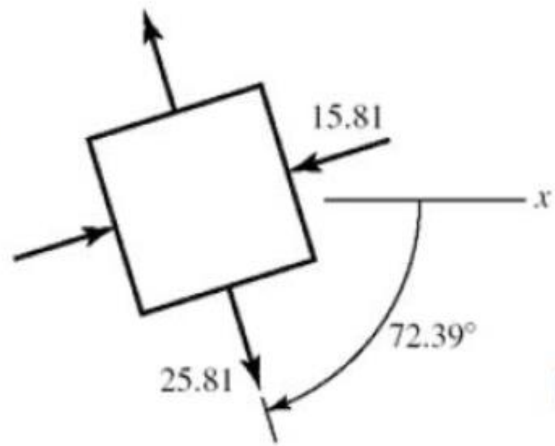
$$\sigma_2 = 5 - 20.81 = -15.81 \text{ kpsi}$$



$$\phi_p = \frac{1}{2}\left[90^\circ + \tan^{-1}\left(\frac{17}{12}\right)\right] = 72.39^\circ \text{ cw}$$

$$\tau_1 = R = 20.81 \text{ kpsi}$$

$$\phi_s = 72.39 - 45 = 27.39^\circ$$



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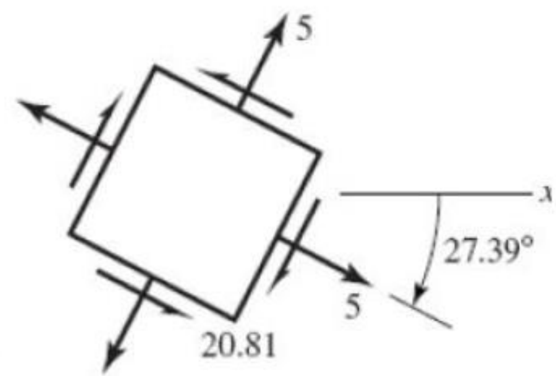

Homework

For each of the plane stress states listed below, draw a Mohr's circle diagram properly labeled, find the principal normal and shear stresses, and determine the angle from the x axis to σ_1. Draw stress elements as in Fig. 3–11c and d and label all details.

(a) $\sigma_x = -8$ MPa, $\sigma_y = 7$ MPa, $\tau_{xy} = 6$ MPa cw
(b) $\sigma_x = 9$ MPa, $\sigma_y = -6$ MPa, $\tau_{xy} = 3$ MPa cw
(c) $\sigma_x = -4$ MPa, $\sigma_y = 12$ MPa, $\tau_{xy} = 7$ MPa ccw
(d) $\sigma_x = 6$ MPa, $\sigma_y = -5$ MPa, $\tau_{xy} = 8$ MPa ccw