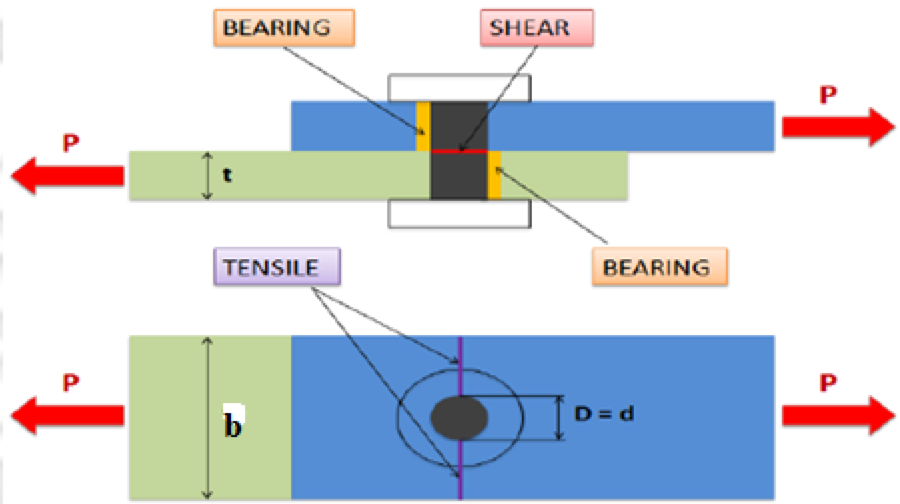
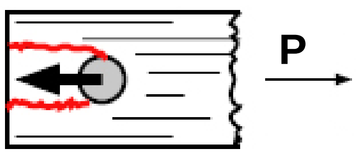
Riveted (Bolted) Joints

There are four types of stresses occur at riveted joints. Therefore, the failure is possible in four locations as follows:

1- Shearing stress failure in rivets
2- Tension stress failure in plate.
3- Bearing stress failure between plate and rivet.
4- Shearing stress failure in plate.





Shear out

Assumption:

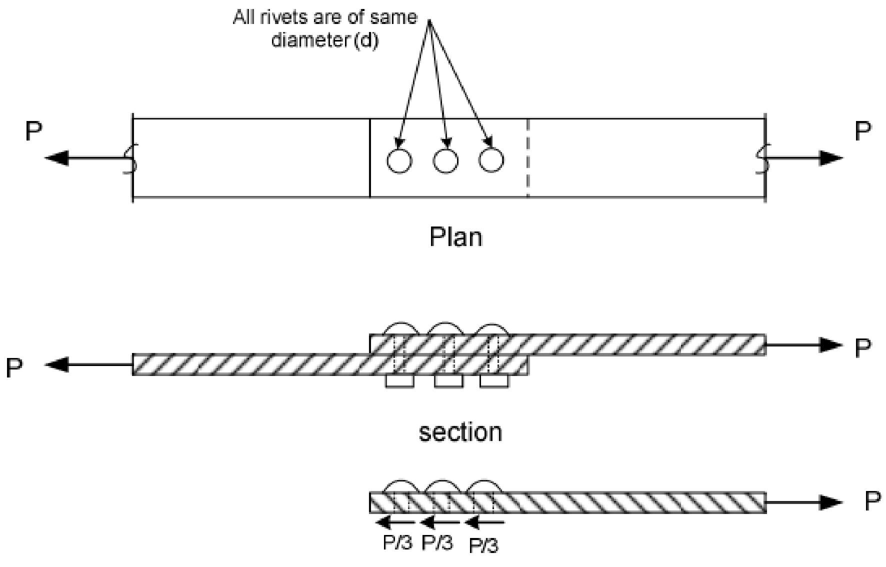
Shearing stress in rivets is equal and uniform. This assumption is approximately true because the shearing stresses are actually distributed in a non-uniform way across the area of the cut.

1- Shearing stress failure in rivets:

Case 1: All rivets in the same diameter

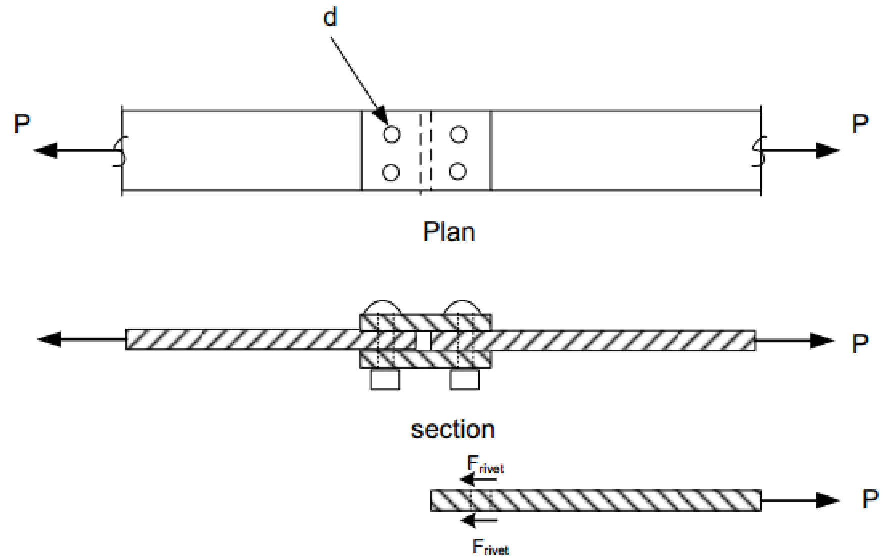
$$F_{rivet} = \frac{P}{3}$$

$$\tau_{rivet} = \frac{P/3}{\pi d^2/4}$$



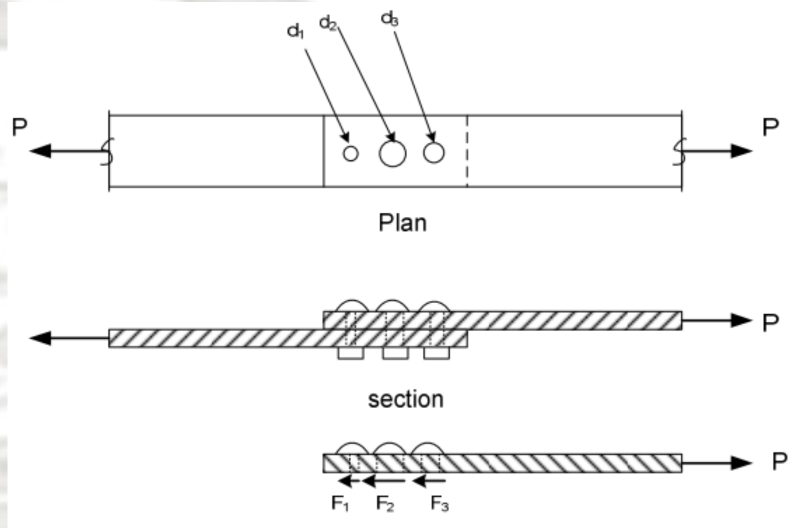
$$F_{rivet} = \frac{P}{4}$$

$$\tau_{rivet} = \frac{P/4}{\pi d^2/4}$$



Case 2: The rivets with different diameters:

According to the assumption, the shear stress must be equal in the three rivets. Therefore, the shear forces must be different.



$$\tau_{rivet} = \frac{P}{\frac{\pi}{4}(d_1^2 + d_2^2 + d_3^2)}$$

$$F_1 = \tau_{rivet} \times \frac{\pi}{4} d_1^2$$

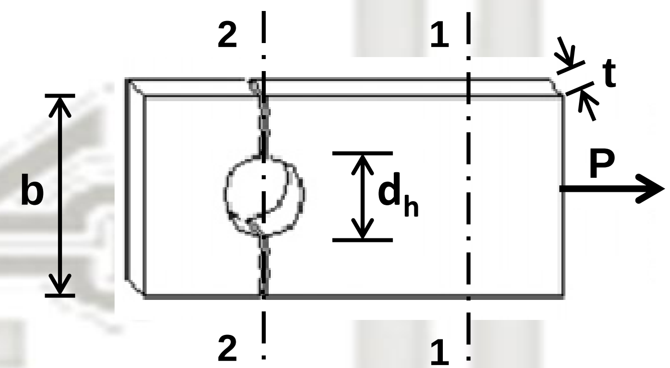
$$F_2 = \tau_{rivet} \times \frac{\pi}{4} d_2^2$$

$$F_3 = \tau_{rivet} \times \frac{\pi}{4} d_3^2$$

2- Tension stress failure in plate.

$$\sigma_t)_{1-1} = \frac{P}{b \times t}$$

$$\sigma_t)_{2-2} = \frac{P}{(b - d_h) \times t}$$



Where:

t = thickness of plate

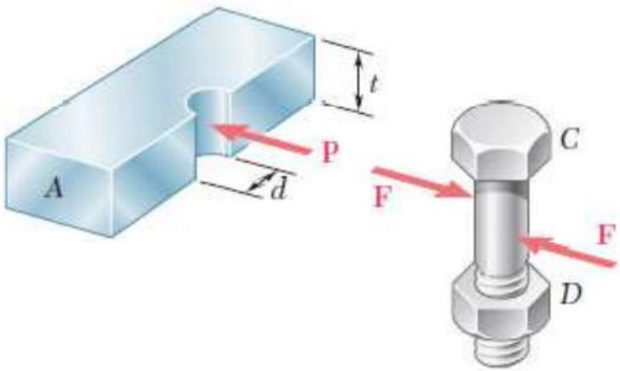
d_h= diameter of hole

d_h= d rivet + 3mm

$\sigma_{2\text{-}2} > \sigma_{1\text{-}1}$

3- Bearing stress between the plate and rivet

$$\sigma_b = \frac{P}{d \times t}$$

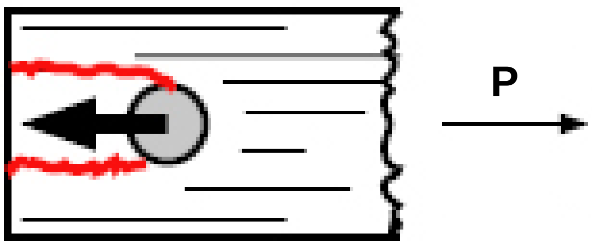


4- Shearing stress failure in plate

$$\sigma_b = \frac{P}{L \times t}$$

Where:

L = Length of the tearing line



Shear out

EXAMPLE 1-11

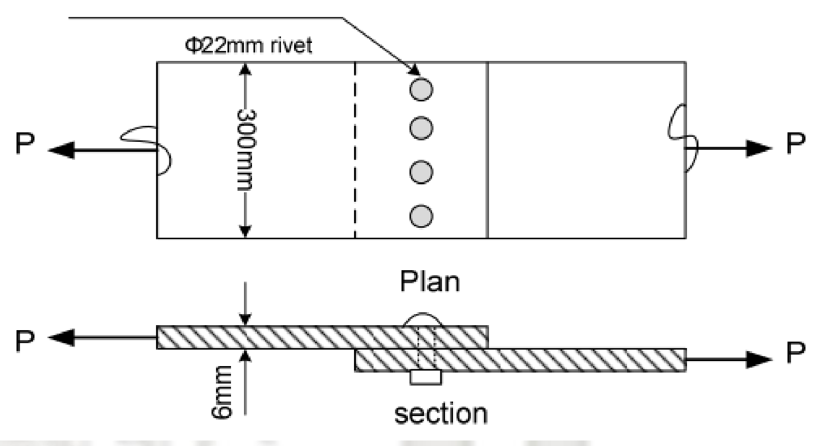
For the lap joint shown in the Figure, calculate the safe axial tensile force (P); if:

$\sigma_{tensile})_{allow.}$=136 MPa

$\tau_{rivet})_{allow.}$=102 MPa

$\sigma_{bearing})_{allow.}$=330 MPa

Assume the diameter of hole =25mm.



Sol.

Shear force in the rivet (F rivet) = P/4

$$\tau_{rivet})_{allow.} = \frac{F_{rivet}}{A_{rivet}} = \frac{P/4}{\pi d^2/4} = 102$$

$$P_{safe} = \pi \times 22^2 \times 102 \times 10^{-3} = 155kN$$

$$\sigma_{tensile})_{allow} = \frac{P}{(b - 4h_{hole}) \times t} = 136$$

$$P_{safe} = (300 - 4 \times 25) \times 6 \times 136 \times 10^{-3} = 163.2kN$$

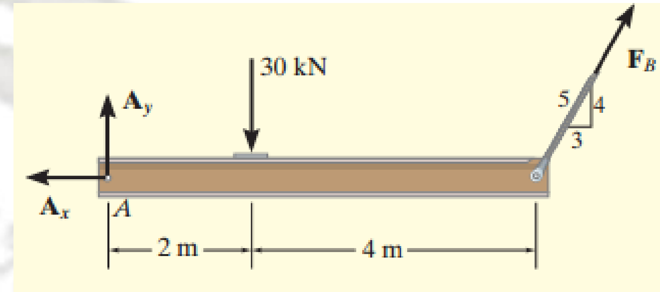
$$\sigma_{bearing})_{allow} = \frac{P/4}{d \times t} = 330$$

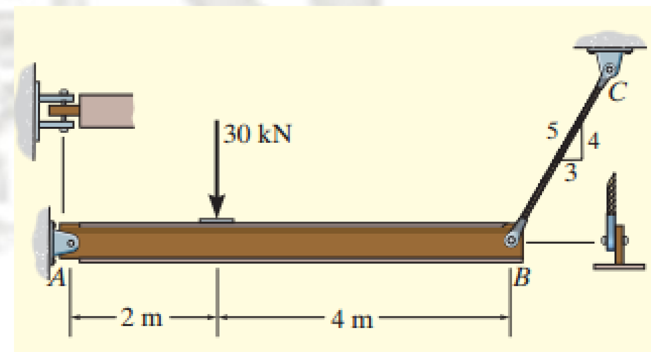
$$P_{safe} = 4 \times 22 \times 6 \times 330 \times 10^{-3} = 174.2kN$$

The safe force which does not cause failure neither in shear nor in tensile nor in bearing is P_{safe} =155kN.

EXAMPLE 1-12

Determine the average shear stress in the 20-mm-diameter pin at A and the 30-mm-diameter pin at B that support the beam in Figure.





$$\sum M_A = 0 \rightarrow F_B \times \frac{4}{5} \times 6 - 30 \times 2 = 0 \rightarrow F_B = 12.5kN$$

$$\sum F_x = 0 \rightarrow 12.5 \times \frac{3}{5} - A_x = 0 \rightarrow A_x = 7.5kN$$

$$\sum F_y = 0 \rightarrow A_y + 12.5 \times \frac{4}{5} - 30 = 0 \rightarrow A_y = 20kN$$

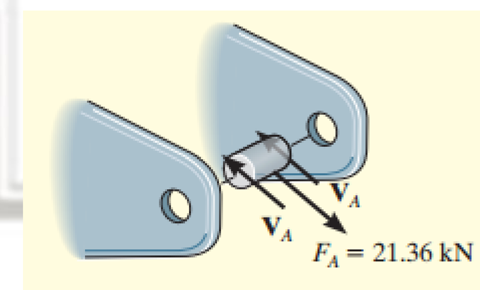
$$F_A = \sqrt{A_x^2 + A_y^2} = \sqrt{7.5^2 + 20^2} = 21.36kN$$

$$V_A = \frac{F_A}{2} = \frac{21.36}{2} = 10.68kN$$

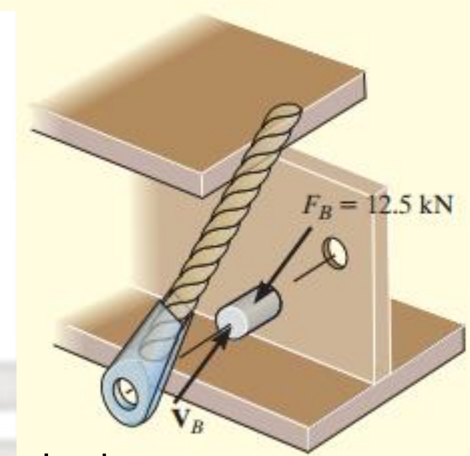
$$V_B = F_B = 12.5kN$$

$$\tau_A = \frac{V_A}{A_A} = \frac{10.68 \times 1000}{\frac{\pi}{4} \times 20^2} = 34MPa$$

$$\tau_B = \frac{V_B}{A_B} = \frac{12.5 \times 1000}{\frac{\pi}{4} \times 30^2} = 17.7MPa$$



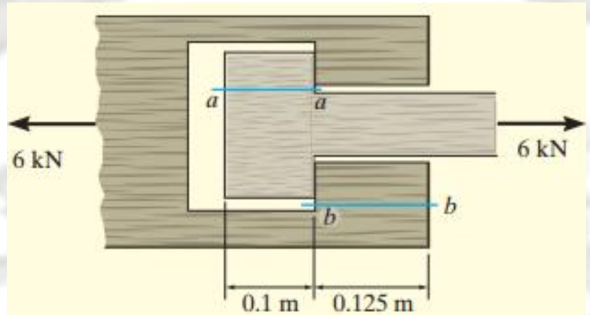
Double shear

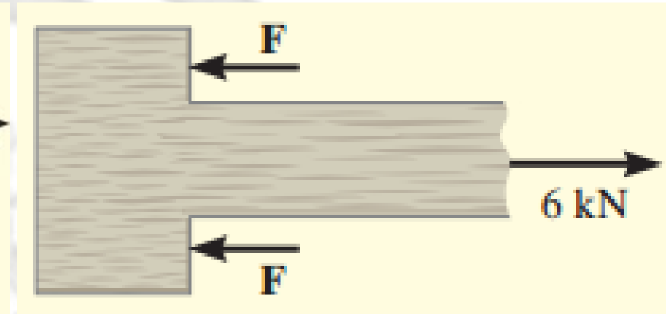


Single shear

EXAMPLE 1-13

If the wood joint in the Figure has a width of 150 mm, determine the average shear stress developed along shear planes a–a and b–b. For each plane, represent the state of stress on an element of the material.





$$\sum F_x = 0 \rightarrow 6 - F - F = 0$$

$$F = 3kN$$

$$\sum F_x = 0 \rightarrow V_a - 3 = 0$$

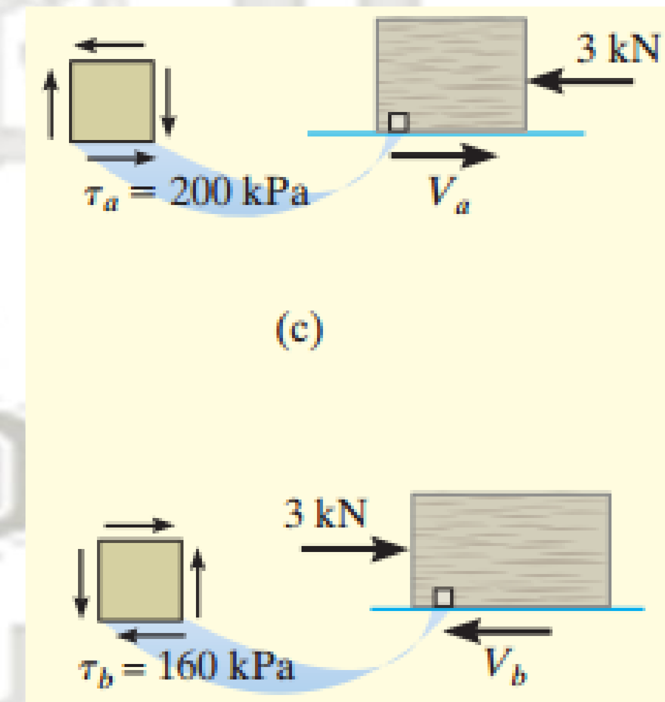
$$V_a = 3kN$$

$$\sum F_x = 0 \rightarrow 3 - V_b = 0$$

$$V_b = 3kN$$

$$\tau_A = \frac{V_a}{A_a} = \frac{3 \times 1000}{0.1 \times 0.15} = 200kPa$$

$$\tau_B = \frac{V_b}{A_b} = \frac{3 \times 1000}{0.125 \times 0.15} = 160kPa$$



EXAMPLE 1-14

Find the smallest diameter bolt that can be used in the connector shown in the Figure, if P = 400 kN. The shearing strength of the bolt is 300 MPa. If the thickness of plate = 30mm, find the bearing stress.

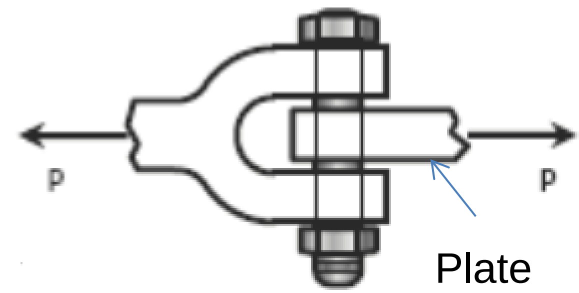


Plate 30mm

The bolt is subjected to double shear

$$V = \frac{P}{2}$$

$$\tau = \frac{V}{A} \rightarrow A = \frac{V}{\tau}$$

$$\frac{\pi}{4} d^2 = \frac{400000}{2 \times 300}$$

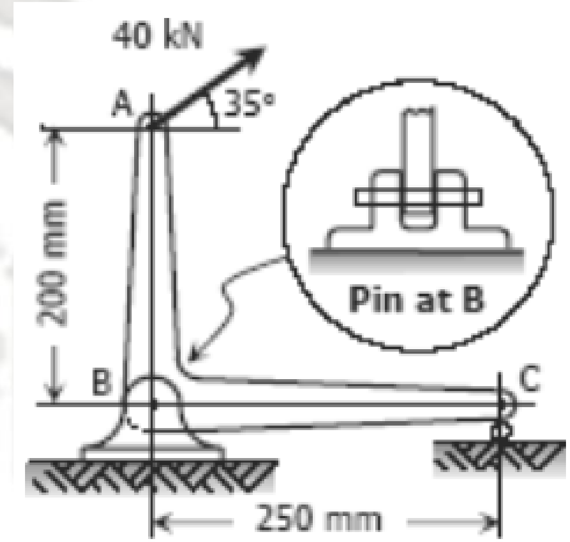
$$d^2 = 848.826$$

$$d = 29.13mm$$

$$\sigma_b = \frac{P}{dt} = \frac{400000}{29.13 \times 30} = 457.72MPa$$

EXAMPLE 1-15

Compute the shearing stress in the pin at B for the member supported as shown in the Figure. The pin diameter is 20 mm.



$$\sum M_c = 0 \rightarrow 0.25B_y = 0.25\times 40\sin 35^\circ + 0.2\times 40\cos 35^\circ$$

$$B_y = 49.156kN$$

$$\sum F_x = 0 \rightarrow B_x = 40\cos 35^\circ = 32.766kN$$

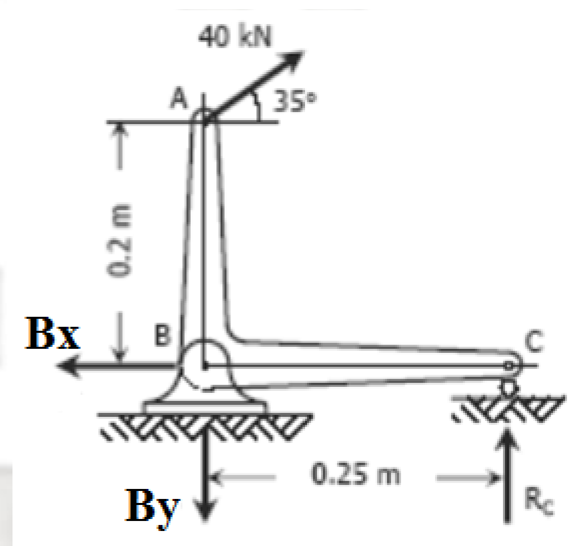
$$R_B = \sqrt{B_x^2 + B_y^2} = \sqrt{32.766^2 + 49.156^2} = 59.076kN$$

$$V_B = \frac{R_B}{2} = \frac{59.076}{2} = 29.538kN$$

$$\tau_B = \frac{29.538}{A}$$

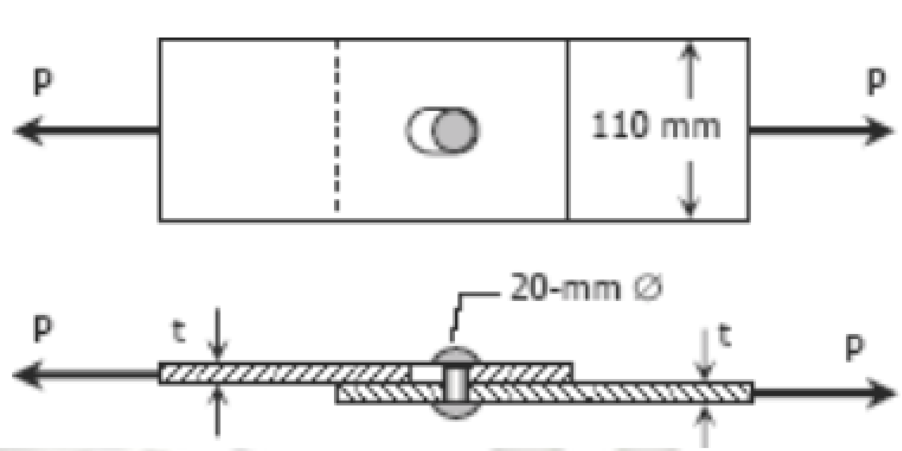
$$\tau_B = \frac{29.538\times 1000}{\frac{\pi}{4}\times 20^2}$$

$$\tau_B = 94.02MPa$$



EXAMPLE 1-16

In the Figure shown, assume that a 20-mm-diameter rivet joins the plates that are each 110 mm wide. The allowable stresses are 120 MPa for bearing in the plate material and 60 MPa for shearing of rivet. Determine (a) the minimum thickness of each plate; and (b) the largest average tensile stress in the plates. (Assume d_h=25mm).



a) From shearing of rivet

$$P = \tau \times A_{rivet} = 60 \times \frac{\pi}{4} \times 20^2 = 18849.5N$$

From bearing of plate

$$P = \sigma_b \times A_b$$

$$18849.5 = 120 \times 20 \times t \rightarrow t = 7.85mm$$

b) Largest average tensile stress in the plate

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

$$\sigma = \frac{18849.5}{7.85(110-25)}$$

$$\sigma = 28.25MPa$$

Allowable stresses; Factor of Safety

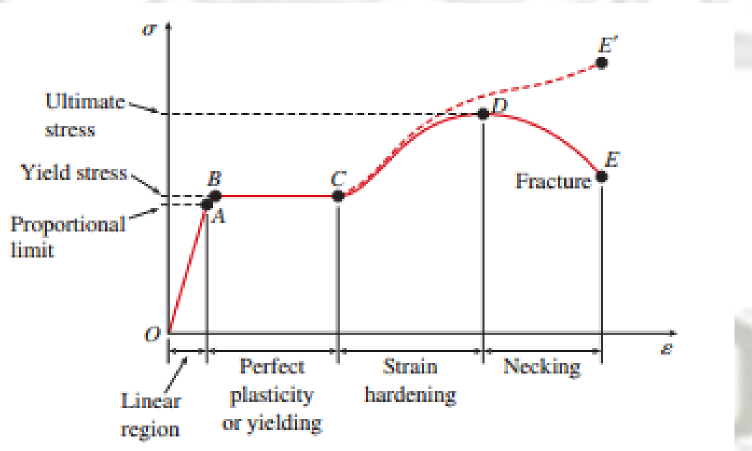
To properly design a structural member or mechanical element it is necessary to restrict the stress in the material to a level that will be safe. To ensure this safety, it is therefore necessary to choose an allowable stress that restricts the applied load to one that is less than the load the member can fully support. One method of specifying the allowable load for a member is to use a number called the **factor of safety**. The factor of safety (F.S.) is a ratio of the **ultimate load** to the **allowable load**. Here the ultimate load is found from experimental testing of the material. The factor of safety must be greater than 1 in order to avoid the potential for failure. the F.S. used in the design of aircraft or space vehicle components may be close to 1 in order to reduce the weight of the vehicle. Or, in the case of a nuclear power plant, the factor of safety for some of its components may be as high as 3 due to uncertainties in loading or material behavior. In design, the area of the member or element is the unknown, while the force is known. However, information on the parameter, stress, must be provided. In fact, information on the stress of material can be gathered from tests.

$$F.S = \frac{F_{failure}}{F_{allowable}}$$

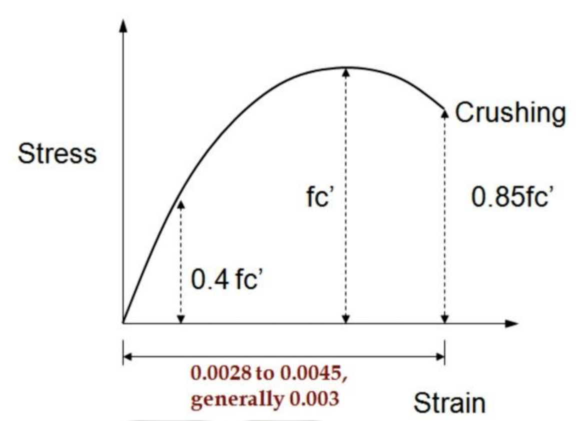
Or

$$F.S = \frac{F_{ultimate}}{F_{allowable}}$$

$$F.S = \frac{Stress_{ultimate}}{Stress_{allowable}}$$



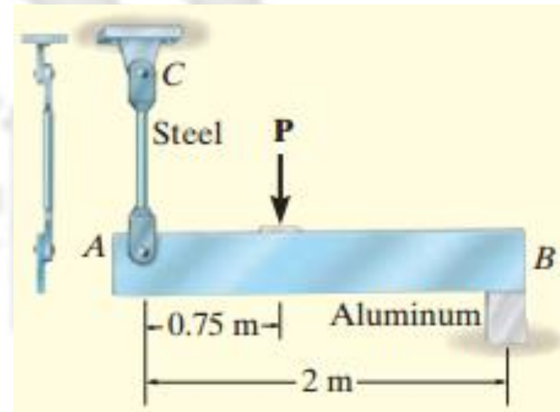
Stress Strain Curve of Concrete



Practically, the stress reaches its maximum value and the corresponding stress (at point D for an example of structural steel in tension) is called **ultimate stress**. However, the stress value used in design is set significantly lower than the ultimate stress and known as **allowable stress** by use of **factors of safety**.

EXAMPLE 1-17

The rigid bar AB shown in the Figure is supported by a steel rod AC having a diameter of 20 mm and an aluminum block having a cross sectional area of 1800 mm^2. The 18-mm-diameter pins at A and C are subjected to single shear. If the failure stress for the steel and aluminum is 680 MPa and 70 MPa respectively, and the failure shear stress for each pin is 900 MPa, determine the largest load P that can be applied to the bar. Apply a factor of safety of F.S=2.



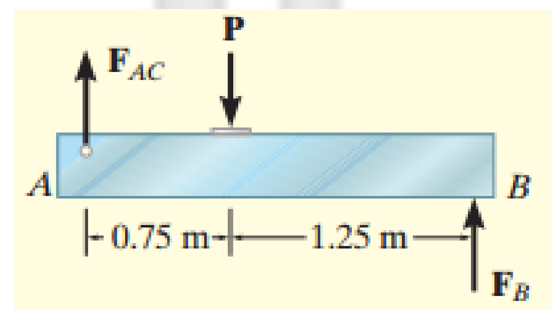
$$(\sigma_{st})_{allow} = \frac{(\sigma_{st})_{failure}}{F.S} = \frac{680}{2} = 340 MPa$$

$$(\sigma_{al})_{allow} = \frac{(\sigma_{al})_{failure}}{F.S} = \frac{70}{2} = 35 MPa$$

$$\tau_{allow} = \frac{\tau_{failure}}{F.S} = \frac{900}{2} = 450 MPa$$

$$\sum M_B = 0 \rightarrow P \times 1.25 = F_{AC} \times 2$$

$$\sum M_A = 0 \rightarrow P \times 0.75 = F_B \times 2$$



Rod AC: $F_{AC} = (\sigma_{st})_{allow} \times A_{AC} = 340 \times 10^{-3} \times \pi \times 10^2 = 106.8 kN$

$$P = \frac{106.8 \times 2}{1.25} = 171 kN$$

Block B: $F_B = (\sigma_{al})_{allow} \times A_B = 35 \times 10^{-3} \times 1800 = 63 kN$

$$P = \frac{63 \times 2}{0.75} = 168 kN$$

Pin A or C, due to single shear:

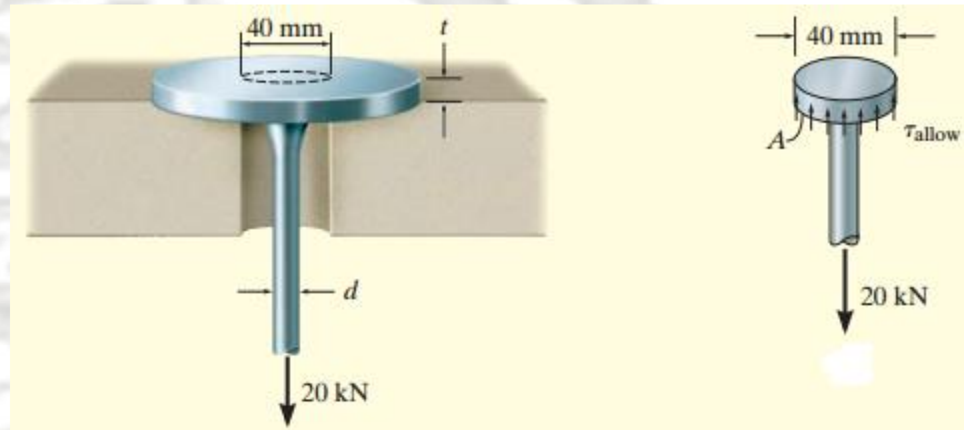
$$F_{AC} = V = \tau_{allow} \times A = 450 \times 10^{-3} \times \pi \times 9^2 = 114.5 kN$$

$$P = \frac{114.5 \times 2}{1.25} = 183 kN$$

By comparison, as P reaches its smallest value (168 kN), the allowable normal stress will first be developed in the aluminum block. Hence, P = 168 kN.

H.W.

The suspender rod is supported at its end by a fixed-connected circular disk as shown in Figure. If the rod passes through a 40-mm-diameter hole, determine the minimum required diameter of the rod and the minimum thickness of the disk needed to support the 20-kN load. The allowable normal stress for the rod is 60 MPa and the allowable shear stress for the disk is 35 MPa.



Class Work

The two solid cylindrical steel rods AB and BC are welded together at B and loaded as shown in the figure. Knowing that the normal stress must not exceed 175MPa in rod AB and 150MPa in rod BC determine the smallest allowable values of d1 and d2.