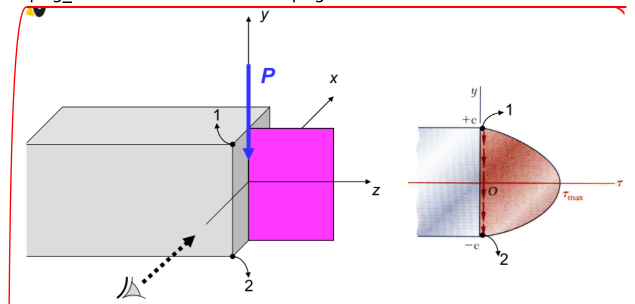
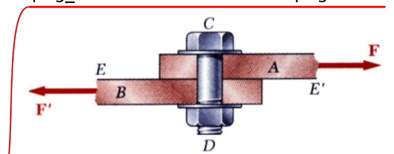
More stresses

Shear Stress

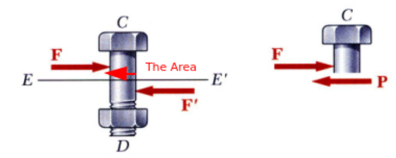
- png_6099228165673189583.png



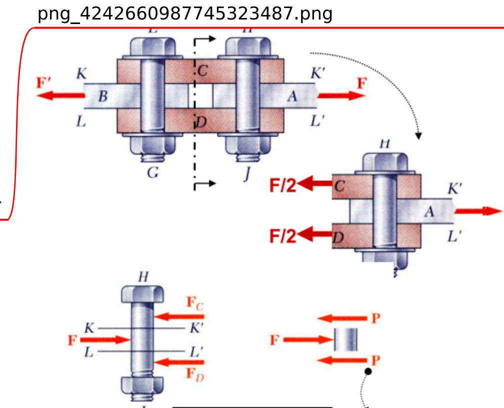
- two forces equal in **mag** opposite in **dn** ->
- $\tau_{ave} = \frac{P}{A}$
- The shear stress distribution ***cannot*** be assumed uniform
- Shear stress distribution varies from 0 to maximum values
  - max values can be much larger than the average value
- Single Shear
  - png_1637618999903357528.png



  - png_996880894476836453.png



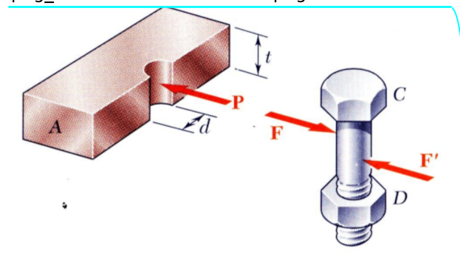
    - $\tau_{ave} = \frac{P}{A} = \frac{F}{A}$
- Double Shear
  - png_4242660987745323487.png



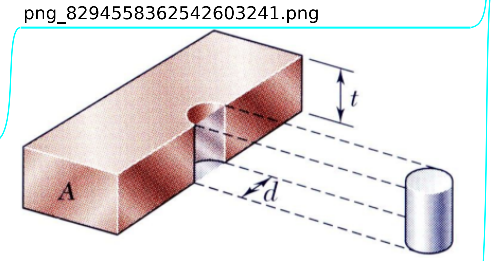
    - $\tau_{ave} = \frac{P}{A} = \frac{F}{2A}$
    - Two shear planes as you see
      - **KK'**
      - **LL'**

Bearing Stress

- png_5076542976437231457.png



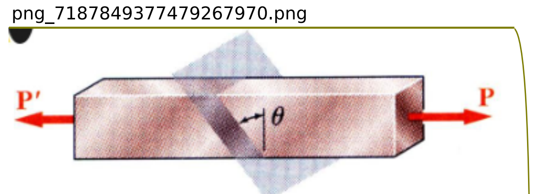
- a bolt/rivet/pin creates stress on the point of contact or the bearing surface of the members they connect
- resultant of force distribution is equal to the force exerted on the pin and opposite in dn ->
- the **bearing area** is defined as the **projected area** of the curved bearing surface
- png_8294558362542603241.png
  - Load / hole diameter * cylinder height



$$\sigma_b = \frac{P}{A} = \frac{P}{td}$$

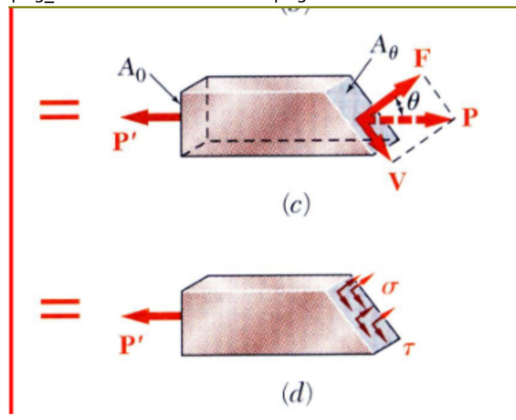
Stresses on an oblique plane under axial loading

- png_7187849377479267970.png



- png_1961980206277222522.png

- png_2717733390904087357.png



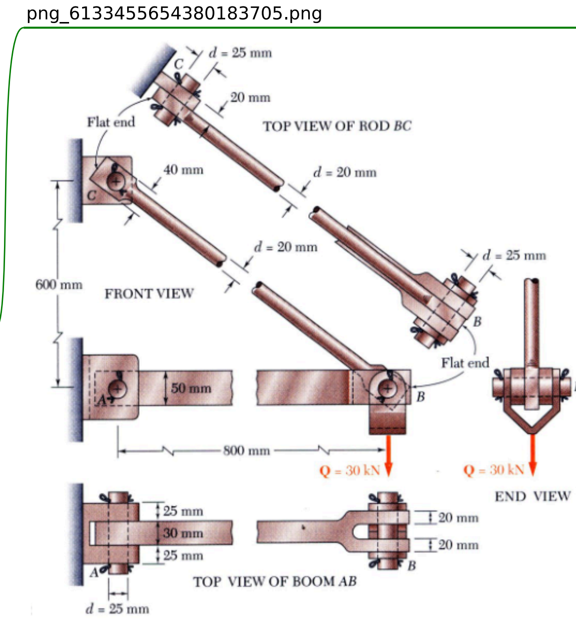
- $\sigma = \frac{F}{A_\theta} = \frac{P\cos(\theta)}{\frac{A_0}{\cos(\theta)}} = \frac{P}{A_0} * \cos^2(\theta)$
- $\tau = \frac{V}{A_\theta} = \frac{P\sin(\theta)}{\frac{A_0}{\cos(\theta)}} = \frac{P}{A_0} * \sin(\theta) * \cos(\theta)$
- you can analyze the previous results at angles 0 and 45
  - at 0 max Normal
  - at 45 max Shear
- at angle 90 there is no stress

Notes

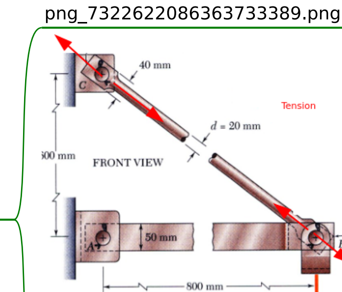
- in a problem if you can get the reqiured using two ways (stresses) use the smallest one in order to be in the allowed region

Problems

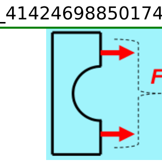
- png_6133455654380183705.png



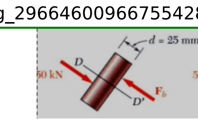
  - Normal Stresses
    - ON member AB, we know Fab =40kN (Compression)
      - there exist a stress on the rectangular area
        - $\sigma_{AB} = \frac{P}{A} = \frac{-40*10^3}{30*50*10^{-6}} = -26.7MPa$
      - since the force is compressive -> the boom pushes the pins and hence there is not a stress on the ends
    - On member BC, Fbc = 50kN (Tension)
      - **the axial 50kN force causes a stress on the cylindrical area of the rod**
        - $\sigma_{BC} = \frac{P}{A} = \frac{50*10^3}{\pi*\frac{1}{4}*(20*10^{-3})^2} = +159MPa$
      - **since the internal force is tension, the rod pulls the pins, so there exist a stress on the flat ends of the rod**
        - png_7322622086363733389.png



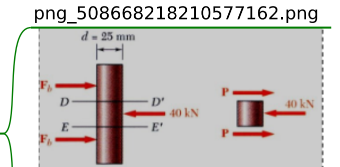
          - png_4142469885017492115.png



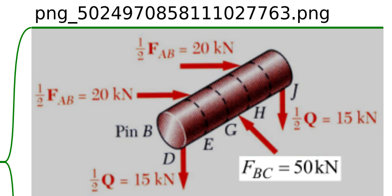
        - $\sigma_{BC,end} = \frac{P}{A} = \frac{50*10^3}{20*(40-25)} = +167MPa$
      - this is a rule
  - Shear Stress
    - Pin C
      - there is only one shear plane so
        - png_2966460096675542858.png



        - $\tau_{C,ave} = \frac{P}{A} = \frac{50*10^3}{\pi*0..25*(25)^2} = 102MPa$
    - Pin A
      - two shear planes = double shear
        - png_5086682182105771162.png



        - $\tau_{A,ave} = \frac{P}{A} = \frac{F}{2A} = \frac{40*10^3}{2*(\pi*0..25*(25)^2)} = 40MPa$
    - Pin B
      - png_5024970858111027763.png



      - from the symmetry we conclude that the maximum shear force is P_{G} = 25kN
        - $\tau_{B,ave} = \frac{P_G}{A} = \frac{F}{2A} = \frac{25*10^3}{2*(\pi*0..25*(25)^2)} = 50.9MPa$
  - Bearing Stress
    - at A
      - the stud creates a bearing stress on the boom AB
        - $\sigma_b = \frac{P}{td} = 40*10^3 30*25 = 53.3MPa$
      - it also creates another bearing stress on the bracket
        - $\sigma_b = \frac{P}{td} = 40*10^3(2*25)*25 = 32MPa$