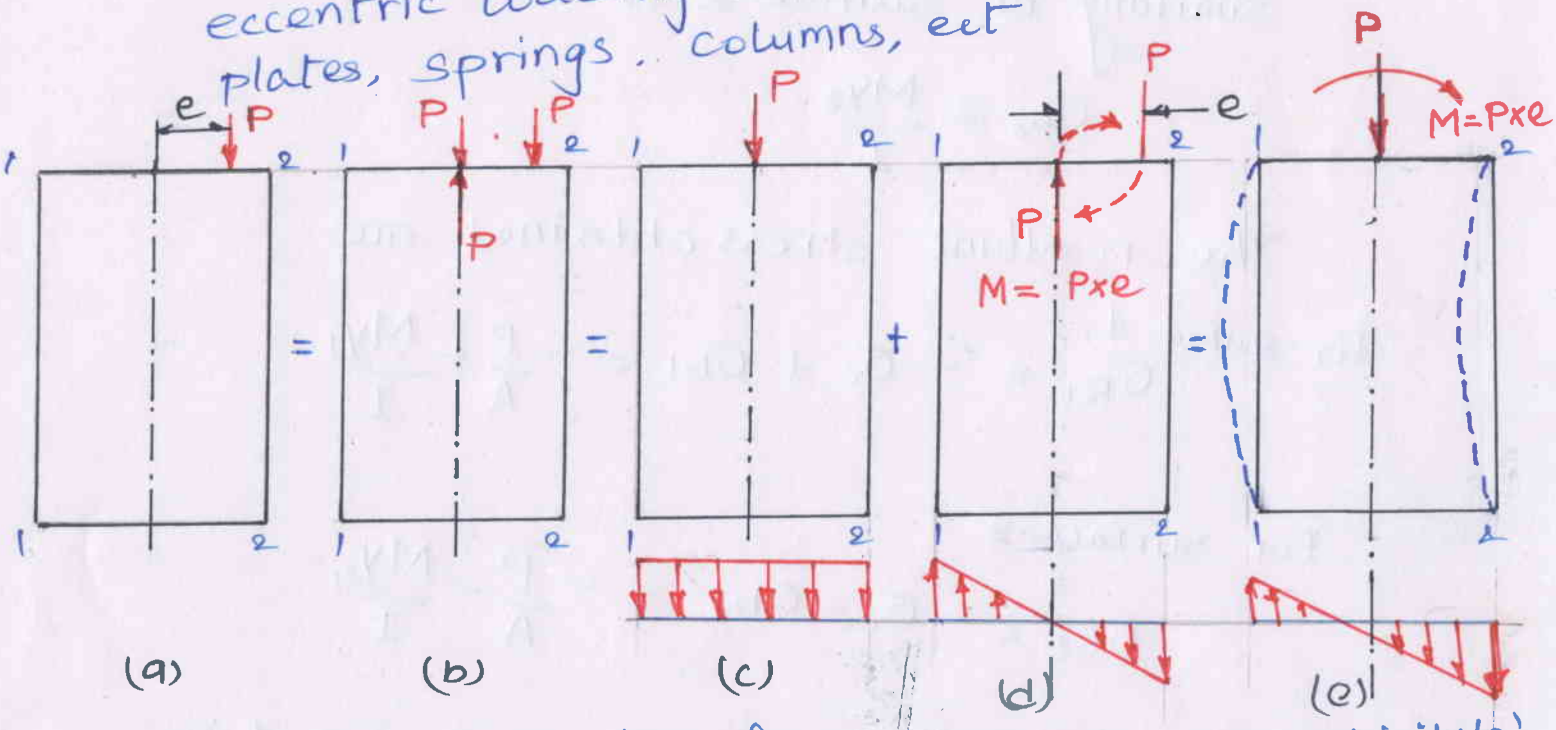
Eccentric Loading

→ In some applications due to defective manufacturing, the Load axis does not coincide with the geometrical or centroidal axis of component.

→ The external Load whose line of action does not coincide with the centroidal axis of the componet is called as eccentric load

→ The distance between the axis of external Load and centroidal axis of a component is called as eccentricity (e)

→ The components which are subjected to eccentric Loading are welded plates, bolted plates, springs. columns, ect



- Let 'P' is compressive force acting at an eccentricity 'e'

→ In fig (c) the force P produce direct compressive stress over the cross-section of a column

$$\sigma_c = P/A$$

- In fig (d) the two forces form a couple of magnitude Pxe bending stress $\sigma_b = \frac{M \cdot y}{I}$

σ_b = Bending stress at a distance y from the neutral axis

M = Bending Moment

I = Moment of inertia of cross-section about the neutral axis

The bending stress at surface 1 is

$$\sigma_{b1} = \frac{My_1}{I}$$

similarly at surface 2 is

$$\sigma_{b2} = \frac{My_2}{I}$$

The resultant stress obtained are

For surface 1

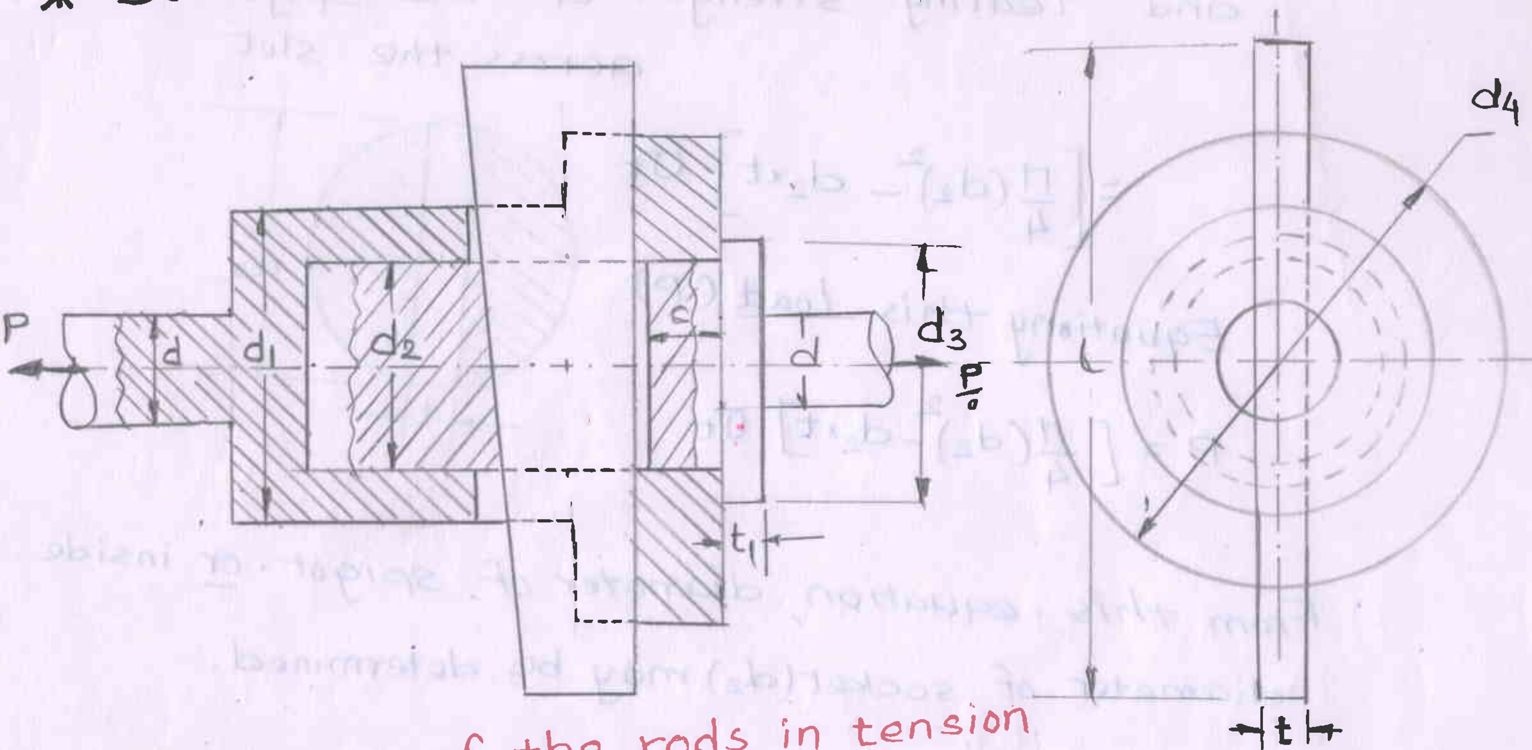
$$\sigma_{R1} = -\sigma_c + \sigma_{b1} = -\frac{P}{A} + \frac{My_1}{I}$$

For surface 2

$$\sigma_{R2} = -\sigma_c - \sigma_{b2} = -\frac{P}{A} - \frac{My_2}{I}$$

It means, resultant stress on surface 1 is smaller as compared to surface 2

* SOCKET AND SPIGOT COTTER JOINT

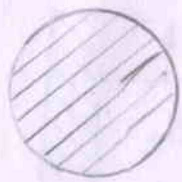


1) Failure of the rods in tension

The rods may fail in tension due to the tensile load P

Area resisting tearing

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



$\therefore$ Tearing strength of the rod

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

Equating this to Load (P)

From this equation, diameter of the rods may be determined

2) Failure of spigot in tension across the weakest section

Since the weakest section of the spigot is that section which has a slot in it for cotter

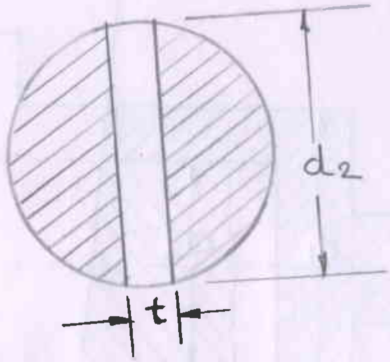
Area resisting tearing of the spigot across the slot

$= \frac{\pi}{4} (d_2)^2 - d_2 \times t$

[where $t = 0.3d$ & $l = 4d$]

and tearing strength of the spigot across the slot

$$= \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right] \sigma_t$$

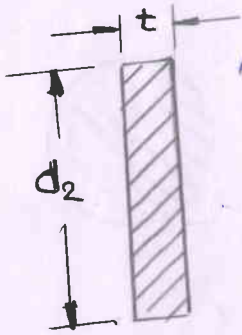


Equating this load (P)

$$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right] \sigma_t$$

From this equation, diameter of spigot or inside diameter of socket (d_2) may be determined.

In addition to tension or compressive stress the crushing stress is also induced in cotter and a spigot at the contact area between



Area that resists crushing of a rod or cotter

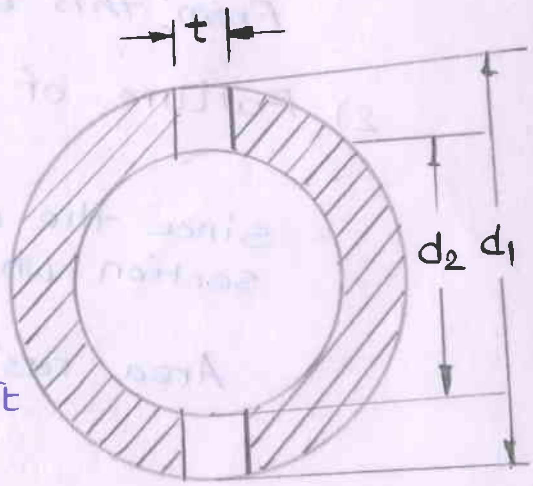
$$= d_2 \times t$$

$\therefore$ Crushing strength $= d_2 \times t \times \sigma_c$

Equating this to load

$$P = d_2 \times t \times \sigma_c$$

3) Failure of the socket in tension across the slot



Area resisting

$$= \frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)t$$

$\therefore$ Tearing strength of the socket across slot

$$= \left\{\frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)t\right\} \sigma_t$$

Equating this to load

$$P = \left\{\frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)t\right\} \sigma_t$$

From this eq^n outside dia. of socket may be determined

4) Distance from the end of slot to end of spigot.

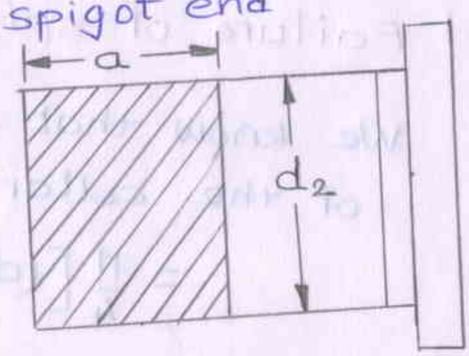
Practically, spigot is subjected to a double shear along the planes containing the surfaces of cotter

since the rod end is in double shear, therefore the area resisting shear of the rod end

$$= 2a \times d_2$$

and shear strength of the spigot end

$$= 2a \times d_2 \times \tau$$



Equating this to load P

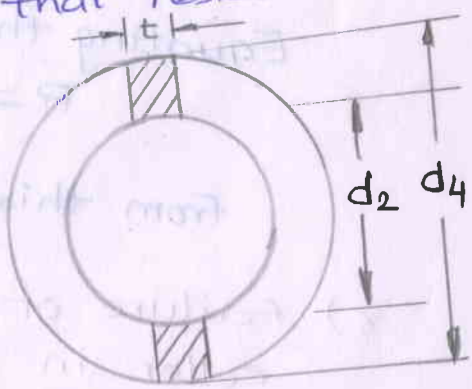
$$P = 2a \times d_2 \times \tau$$

From this eq^n. the distance from the end of slot to end of spigot may be obtained

5) Failure of socket collar in crushing

Considering the failure of socket collar in crushing, we know that area that resists crushing of socket coller

$$= (d_4 - d_2) \times t$$



crushing strength

$$= (d_4 - d_2) \times t \times \sigma_c$$

$$\therefore P = (d_4 - d_2) \times t \times \sigma_c$$

From this eq^n. the diameter of socket collar may be determined

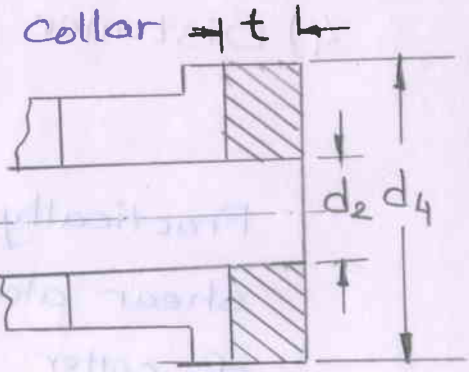
6) Failure of socket end in shearing

The socket end is in double shear area that resists shearing of socket collar

$$= 2(d_4 - d_2) \times c$$

shearing strength of socket collar

$$= 2(d_4 - d_2) \times c \times \tau$$



Equating this to load

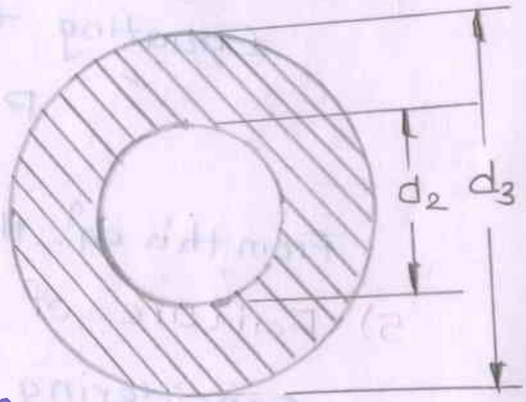
$$P = 2(d_4 - d_2) \times c \times \tau$$

From this eq^n, the thickness of socket collar may be obtained

7) Failure of spigot collar in crushing

We know that area that resists crushing of the collar

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]$$



and crushing strength of the collar

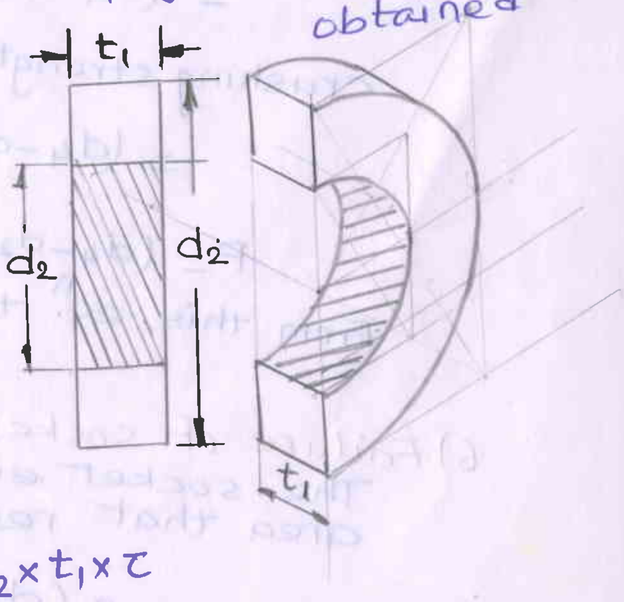
$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \times \sigma_c$$

Equating this to Load, we have

$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \times \sigma_c$$

From this eq^n the dia. of spigot collar may be obtained

8) Failure of the spigot collar in shear



We know that area that resists shearing of the collar

$$= \pi d_2 \times t_1$$

shearing strength of the collar

$$= \pi d_2 \times t_1 \times \tau$$

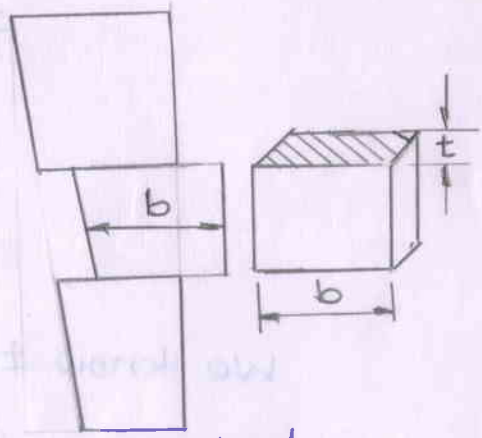
Equating this load $P = \pi d_2 \times t_1 \times \tau$

From this eq^n. the tickness of spigot collar may be obtained

9) Failure of cotter in shear

The cottar is in double shear, therefore shearing area of the cotter

$= 2b \times t$



Shearing strength of the cotter

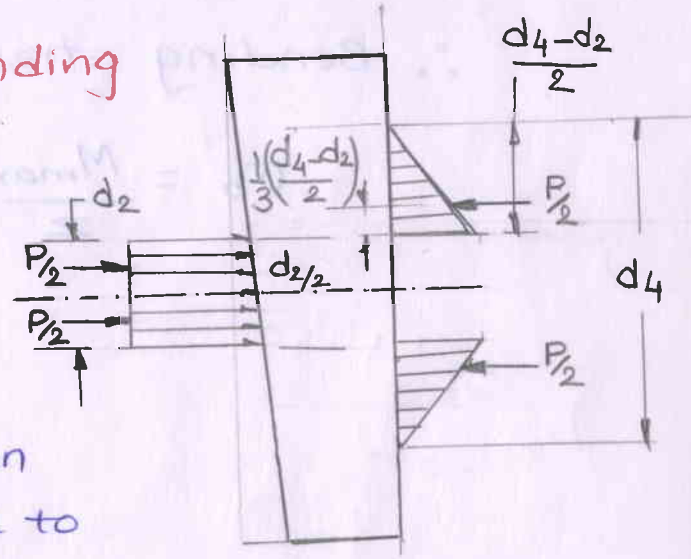
$= 2b \times t \times \tau$

Equating this Load

$P = 2 \times b \times t \times \tau$

From this eq^n. width of cotter may be obtained

10) Failure of cotter in bending



During analysis, it is assumed that the Load is uniformly distributed over the variouse cross-section of the joint. But actual practice, this does not happen and the cotter is subjected to bending. In order to find out the bending stress induced, It is assumed that the Load on the cotter in the spigot end is uniformly distributed while in the socket end it varies from zero at the outer diameter (d_4) and Maximum at inner diameter(d_2)

The maximum bending moment occurs at the Centre of the cotter

$$M_{max} = \frac{P}{2}\left(\frac{1}{3} \times \frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2} \times \frac{d_2}{4}$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right)$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

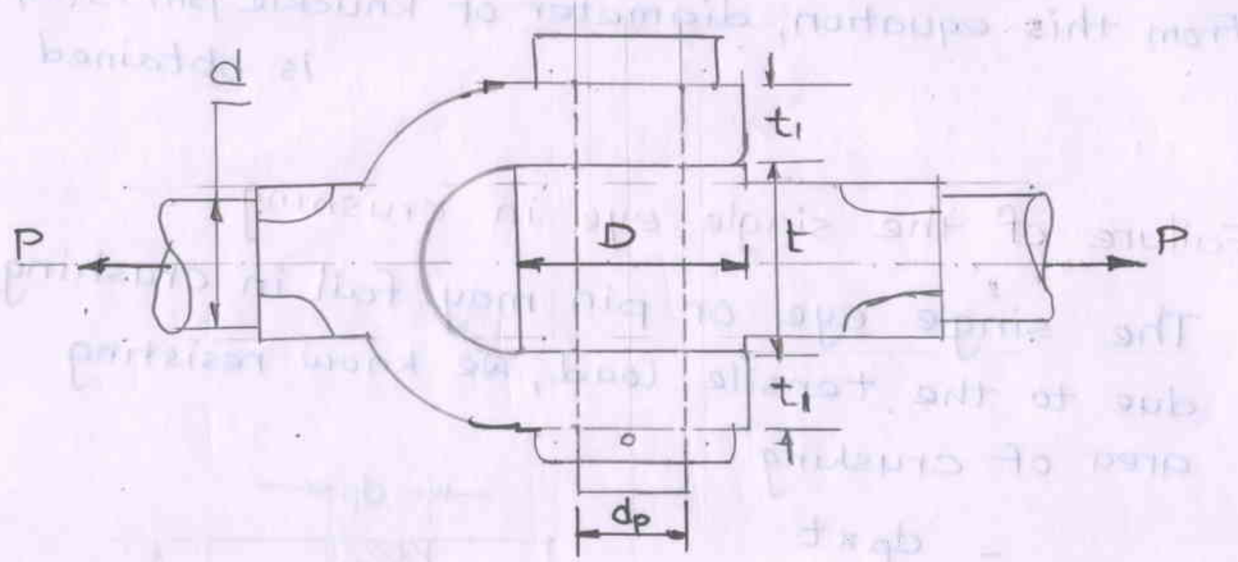
We know that section modulas of the cotter

$$Z = \frac{t \times b^2}{6} \quad \left[\because I = \frac{t \times b^3}{12} \;\&\; y_{max} = \frac{b}{2},\; Z = \frac{I}{y_{max}}\right]$$

$\therefore$ Bending stress induced in cotter

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{\frac{t \times b^2}{6}}$$

* DESIGN OF KNUCKLE JOINT

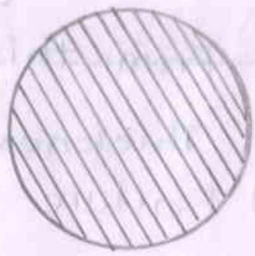


1) Failure of the solid rod in tension

The rods are subjected to direct tension

Tensile strength of the rod

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



Equating this to the load

$$\boxed{P = \frac{\pi}{4} \times d^2 \times \sigma_t}$$

From this eq^n, diameter of the rod (d) may be obtained

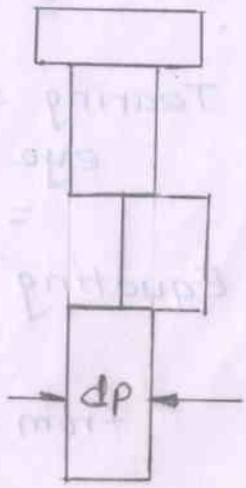
2) Failure of the knuckel pin in shear

During the working, the knuckel pin is subjected to double shear

$$= 2 \times \frac{\pi}{4} \times (d_p)^2$$

shear strength of pin

$$\boxed{P = 2 \times \frac{\pi}{4} \times (d_p)^2 \times \tau}$$



Equating this to load acting on the rod

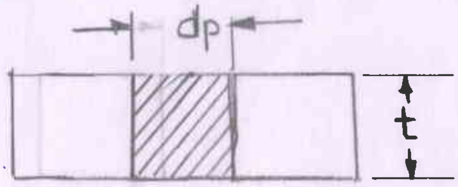
$$P = 2 \times \frac{\pi}{4} (d_p)^2 \times \tau$$

From this equation, diameter of knuckle pin is (d_p) is obtained

3) Failure of the single eye in crushing

The single eye or pin may fail in crushing due to the tensile load, we know resisting area of crushing

$$= d_p \times t$$



crushing strength $= d_p \times t \times \sigma_c$

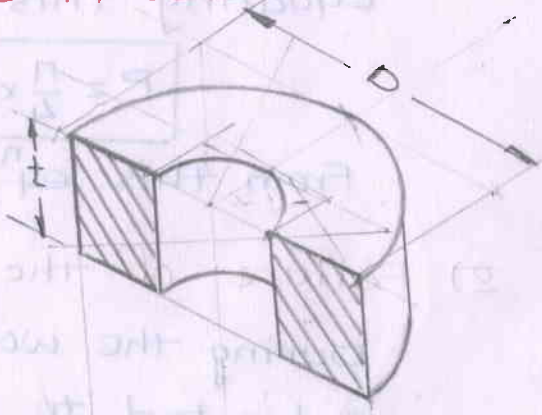
Equating this to the load

$$\boxed{P = d_p \times t \times \sigma_c}$$

From this equation, thickness of single eye is (t) obtained

Thickness of single eye is also given by $t = 1.25d$

4) Failure of the single eye in tension

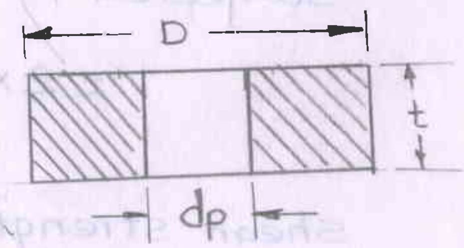


The single eye may fail or tear off due to the tensile load, we know that ~~as~~ area resisting tearing

$$= (D - d_p)\,t$$

Tearing strength of single eye

$$= (D - d_p) \times t \times \sigma_t$$



Equating this to load

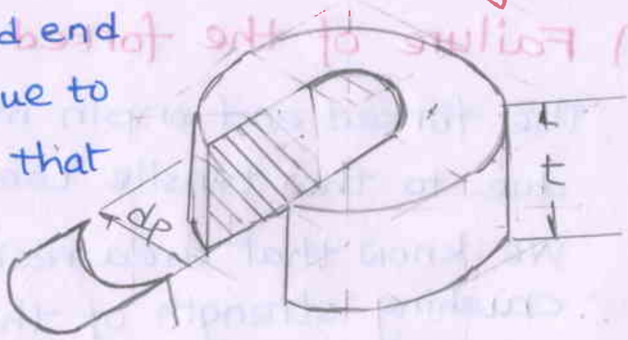
$$\boxed{P = (D - d_p) \times t \times \sigma_t}$$

From this eq^n, outer diameter of single eye (D) is obtained

5) Failure of the single eye in shearing

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing

$= (D - d_p)t$



$\therefore$ shearing strength of single eye

$P = (D - d_p) \cdot t \times \tau$

From this eq^n. the induced shear stress (τ) for the single may be cheeked

→ The mininum thickness of the single eye is taken as

$t = 1.25d$

6) Failure of the forked end in tension

The forked end or double eye may fail in tension due to the tensile load.

We know that area resisting tearing

$= (D - d_p) \times t_1 \times 2$

$\therefore$ Tearing strength of forked end

$P = (D - d_p) \times 2t_1 \times \sigma_t$

From this eq^n. thickness of forked end (t_1) is obtained

$t_1 = 0.75d$

7) Failure of the forked end in crushing

The forked end or pin may fail in crushing due to the tensile Load.

We know that area resisting crushing $= d_p \times 2t_1$

$\therefore$ crushing strength of the forked end $= d_p \times 2t_1 \times \sigma_c$

Equating this to the Load (P) $= d_p \times 2t_1 \times \sigma_c$

From this eq^n^. the induced crushing stress for the forked end may be checked or thickness of forked end (t_1) is obtained

8) Failure of the forked end in shear

The forked end may fail in shearing due to the tensile load

We know that area resisting shearing

$$= (D - d_p) \times 2t_1$$

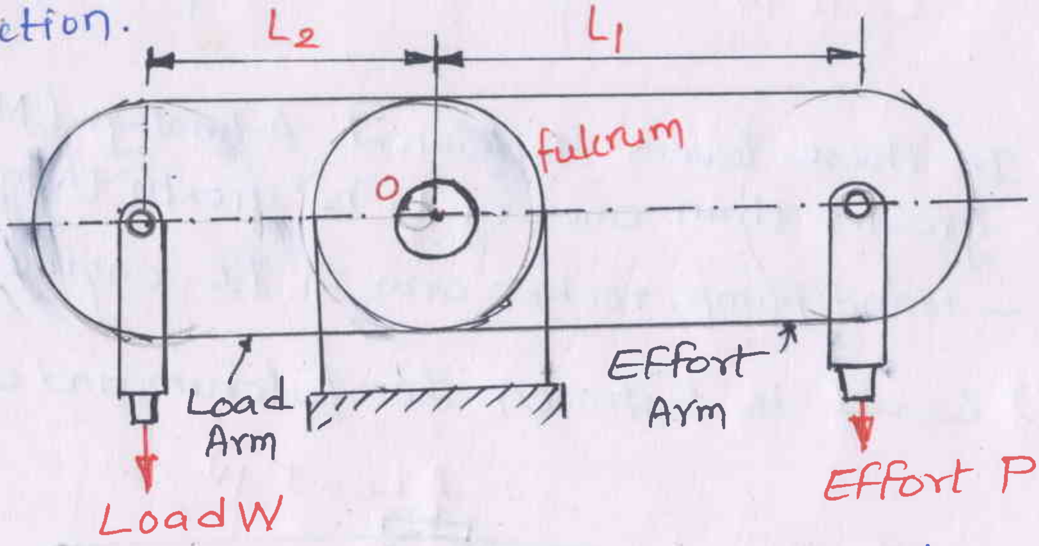
$\therefore$ shearing strength of the forked end

$$= (D - d_p) \times 2t_1 \times \tau$$

Equating this to Load $P = (D - d_p) \times 2t_1 \times \tau$

Levers

→ A Lever is a mechanical device in the form of a rigid rod or bar pivoted about a point known as Fulcrum and used to overcome a Load by the application of a small effort or to transfer the force in disired direction.



W = Load to be produced in N

P = Required effort in N

L_1 = Length of effort arm in mm

L_2 = Length of Load arm in mm

considering moment about the ~~fucrum~~ fulcrum 'O'

$$\sum M_0 = 0 = P \times L_1 - W \times L_2$$

$$P \times L_1 = W \times L_2$$

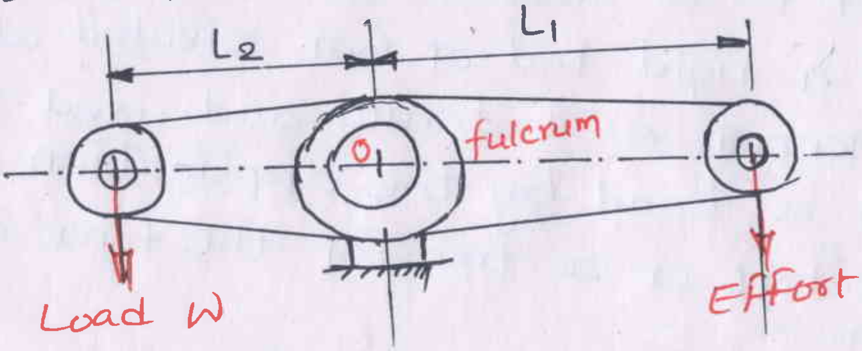
$$\frac{W}{P} = \frac{L_1}{L_2}$$

The ratio of Load to effort (W/P) is called as Mechanical Advantage of the lever and the ratio of effort arm to Load arm (L_1/L_2) is called as Leverage.

Mechanical advantage = Leverage

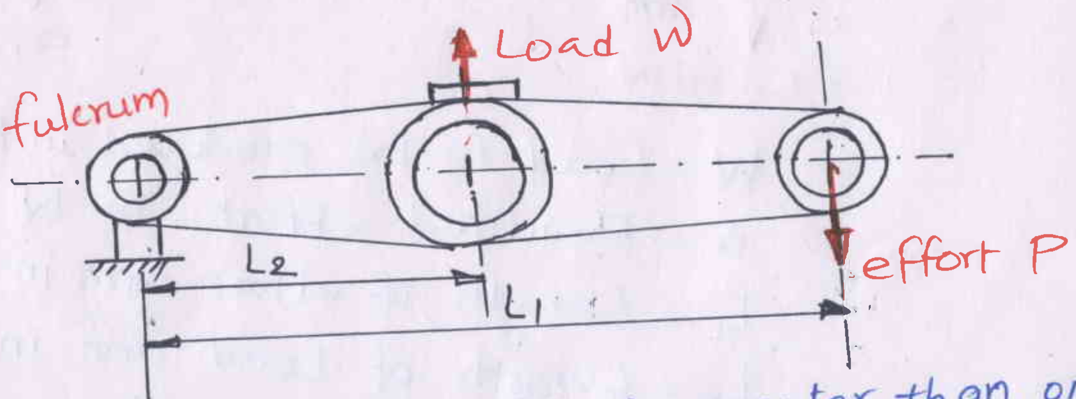
Types of Levers

i) Fulcrum is between the Load and effort.



- In these levers Mechanical Advantage (M.A.) is greater than one or L_1 is greater then L_2
- Hand pump, rocker arm in I.C. engine

(ii) Load is between the fulcrum and effort

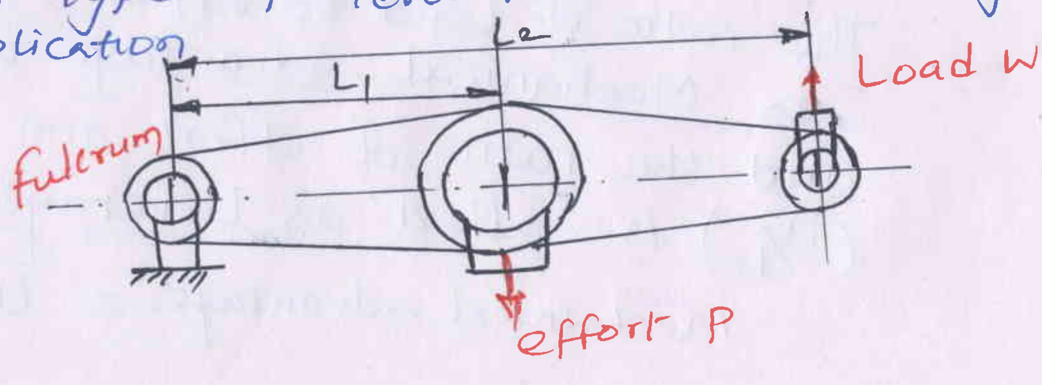


- In these levers also M.A. is greater than one
- Lever of the boiler safty valve.

(iii) Effort is between the fulcrum and Load

- In these levers M.A is less than one or effort arm is smaller than the Load arm

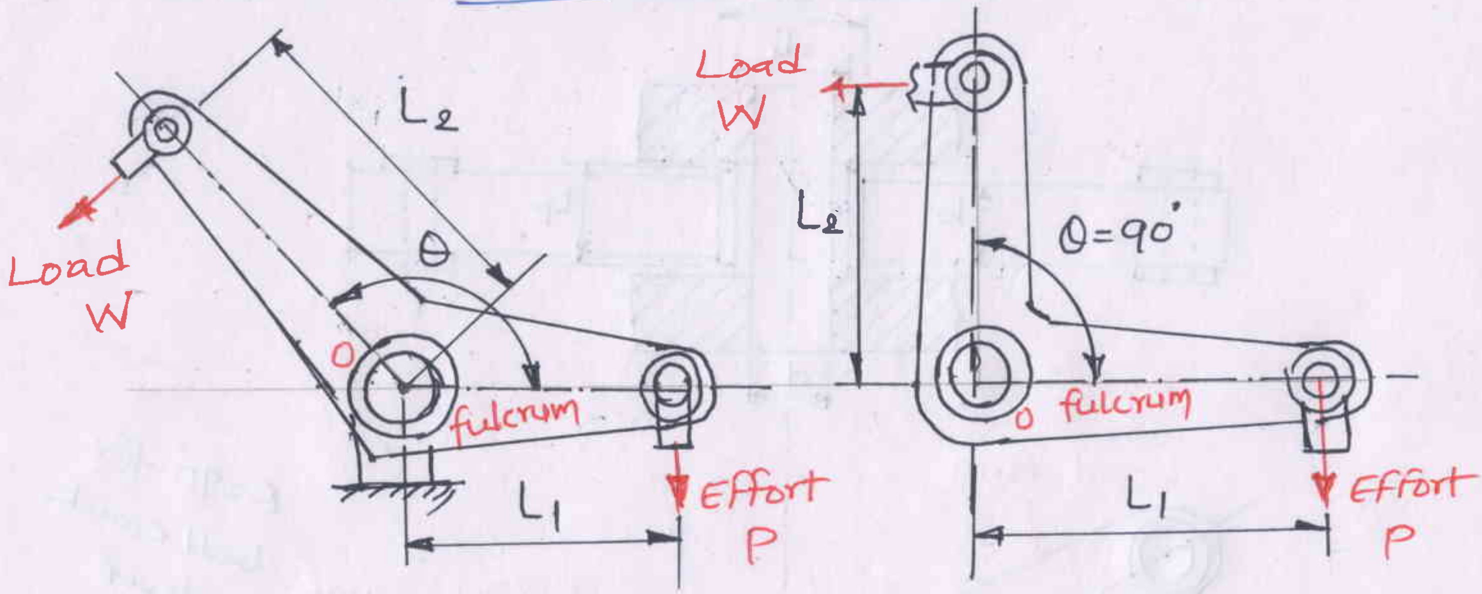
This type of lever is not used in engineering application



(iv) Levers in which load arm and effort arm are inclined.

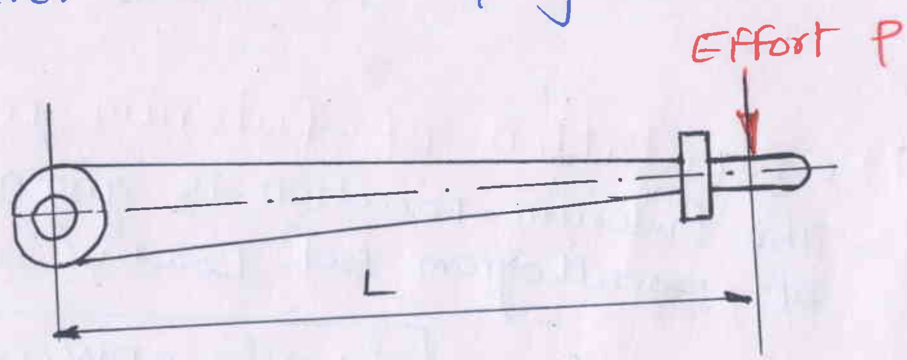
If the angle between the load arm and effort arm is other than 180° then these levers are calls as angular levers

- If the angle between the load arm and effort arm is 90° then the lever is called as bell crank lever



(v) One arm lever (Hand/Foot lever)

- These levers have only one arm and that is effort arm
- This lever is used to apply external torque

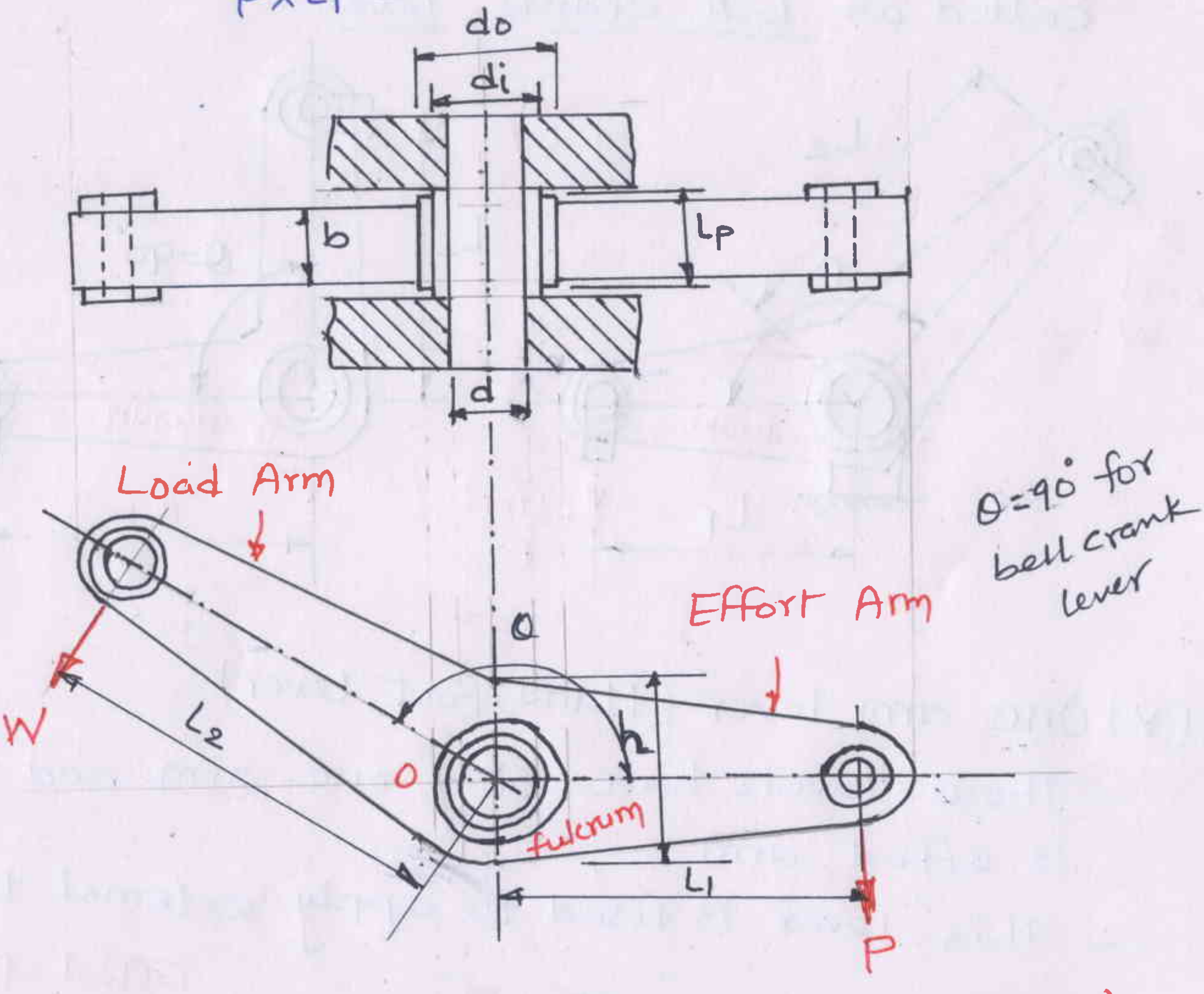


Design of Angular Lever

i) Calculation of effort (P)

- The Load W, length of Load arm L_2 and the length of effort arm L_1 is known.
- Calculate the effort P by considering the moment about the fulcrum.

$$P \times L_1 = W \times L_2$$



ii) Calculation of fulcrum reaction (F_R)

- The fulcrum reaction is given by the Law of parallogram betⁿ. Load W and effort P

$$F_R = \sqrt{P^2 + W^2 + 2PW\cos\theta}$$

θ = Angle between the lines of action of W & P

iii) Fulcrum pin dimensions

→ During the operation of lever, the fulcrum pin supports the lever and make it to oscillate. It means, there is relative motion between the pin and the lever.

- The fulcrum pin is subjected to bearing pressure and direct shear stress.

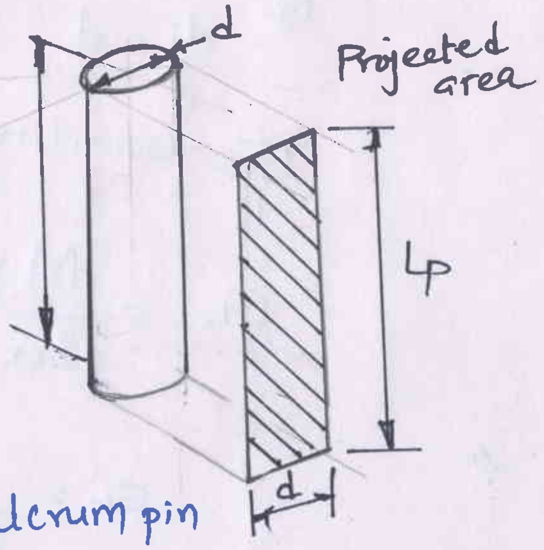
Let d = diameter of Fulcrum pin in mm

l_p = Supporting length of fulcrum pin in mm = 1d to 1.5 d

P_b = Permissible bearing pressure in N/mm^2

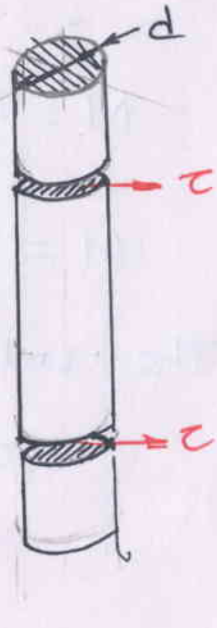
the bearing pressure betn. the Fulcrum pin and the boss of lever

$$P_b = \frac{F_R}{d l_p}$$



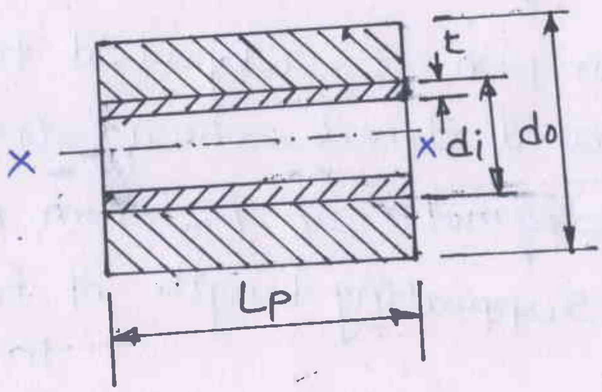
Considering the failure of fulcrum pin in double shear.

$$\tau = \frac{F_R}{2 \times \frac{\pi}{4} d^2}$$



iv) Dimension of boss of lever.

→ Due to bending moment, the boss of lever is subjected to bending stress (σ_b)



d_o = out diameter of boss

d_i = inner diameter of boss

$$d_i = d + 2t = d + 2 \times 3$$

or $d_i = d$ (if bush is not used)

The bending stress in the boss of lever is

$$\sigma_b = \frac{M.y}{I_{xx}} = \frac{M \times d_o/2}{\frac{L_p}{12}(d_o^3 - d_i^3)}$$

$$\sigma_b = \frac{6 M d_o}{L_p(d_o^3 - d_i^3)}$$

M = Bending moment on the boss

$$M = W \times L_2 = P \times L_1$$

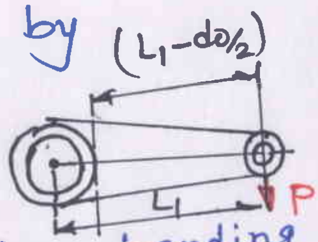
The outer diameter of the boss

$$d_o = 2d$$

V) Dimension of lever Cross-Section

- The arms of a Lever are subjected to bending moment. The maximum bending moment acts on the cross-section adjacent to the boss

- This max. bending moment is given by

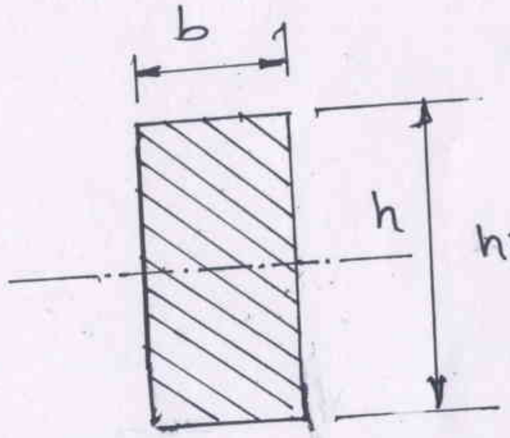


$$M_{max} = P\left(L_1 - \frac{d_o}{2}\right)$$

- The bending stress corresponding to Max. bending moment in lever Cross-section

$$\sigma_b = \frac{M_{max} \cdot y}{I_{xx}}$$

- For rectangular section

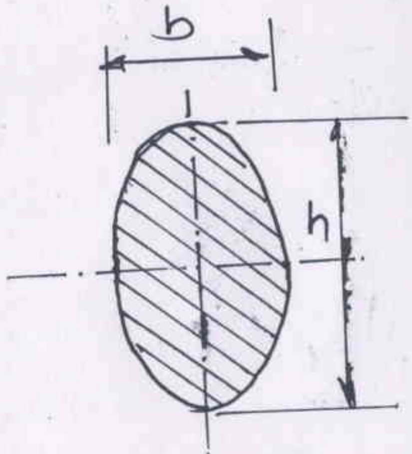


$h = 2b$ to $4b$

$$\sigma_b = \frac{M_{max} \times h/2}{bh^3/12}$$

$$\sigma_b = \frac{6 M_{max}}{bh^2}$$

- For elliptical section



$$\sigma_b = \frac{M_{max} \cdot y}{I_{xx}} = \frac{M_{max} \times h/2}{\pi bh^3/64}$$

$$\sigma_b = \frac{32 M_{max}}{\pi bh^2}$$

$h = 2b$ to $2.5b$