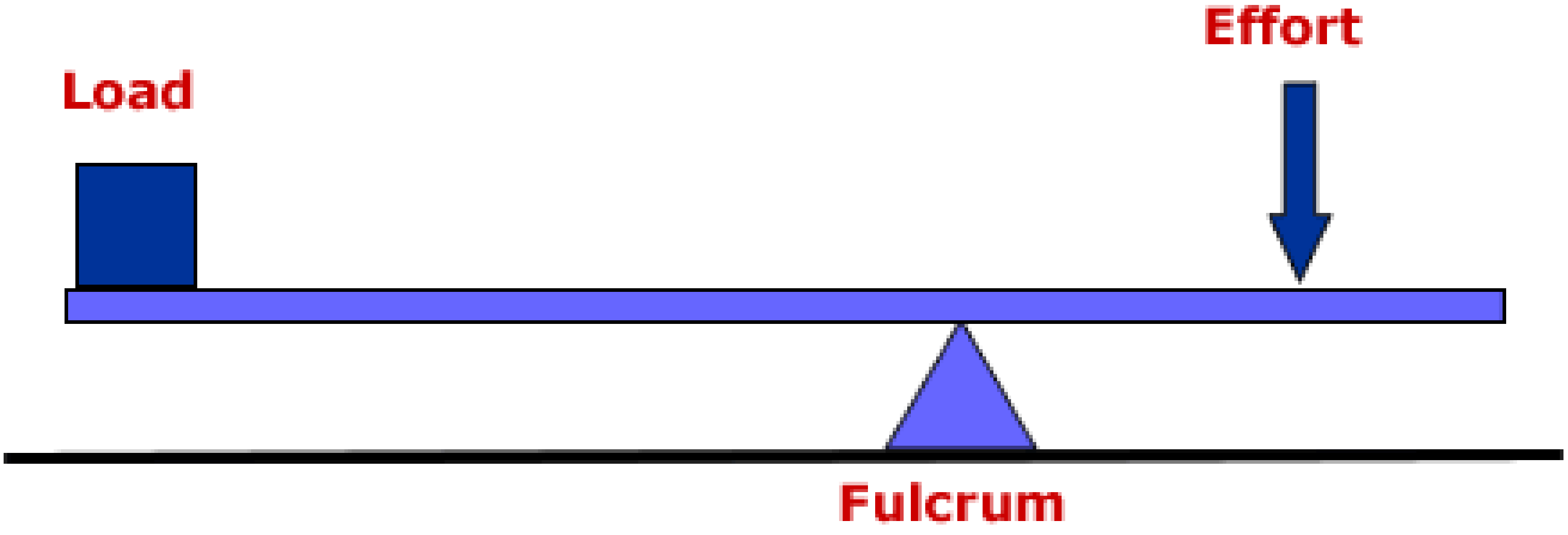
UNIT 02

LEVERS AND ITS DESIGN

- Introduction
- Types of levers
- Design of Hand lever
- Design of Foot Lever
- Design of Bell Crank lever

Introduction

- A lever is a rigid rod or bar pivoted at a point and capable for turning about the pivot point called fulcrum.
- The levers are used to lift a load with the small effort.
- **The ratio of lifted load to an effort called mechanical advantages.**
- **The ratio of length of effort arm to the length of load arm is called as leverage.**


Effort
Load
Fulcrum

Types of Levers

- According to the application of load and effort, the levers are classified as

1) One Arm Lever
2) Two Arm Lever
3) Angular Lever
4) Bell Crank Lever

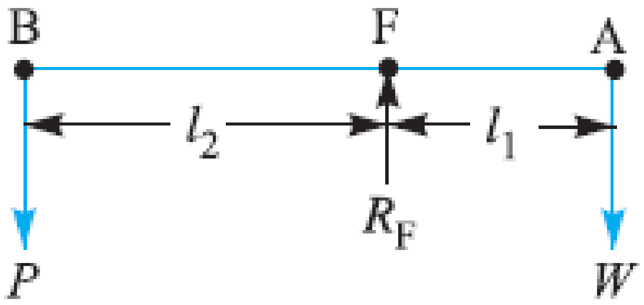
1. One Arm Lever

- The one arm lever is an example of hand lever, foot lever and cranking lever.
- It has only one arm and that is effort arm.
- This type of lever is used to apply external torque.

2. Two Arm Lever

- Depending upon the position of fulcrum pin, load and effort the two arm lever are of three types as explain below.

1) Fig. Shows two arm lever, in which the load arm and effort arm are of equal length.

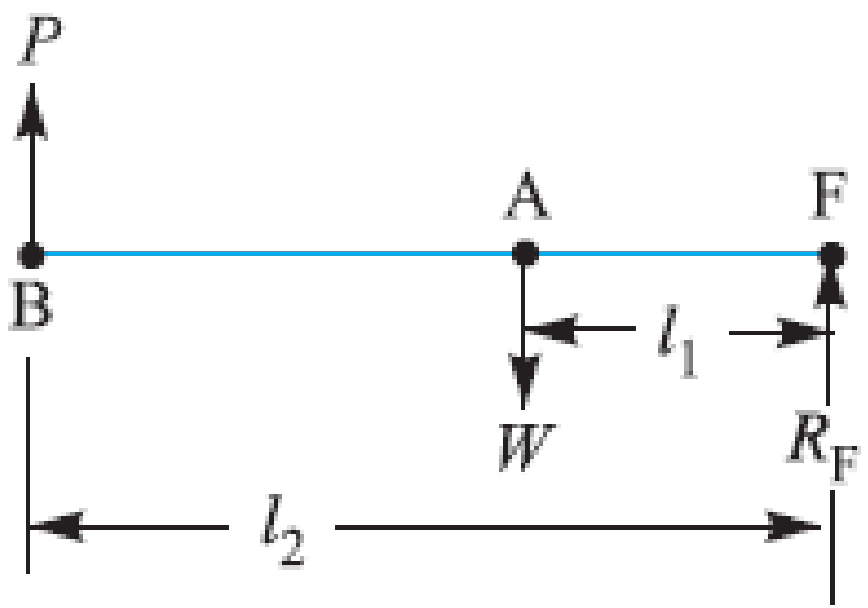


The fulcrum pin is pivoted in between the load and effort arm.

The examples of such lever are rocket arm of I.C. engine, beam of a balance, handle of hand pump. These lever have mechanical advantage is equal to one.

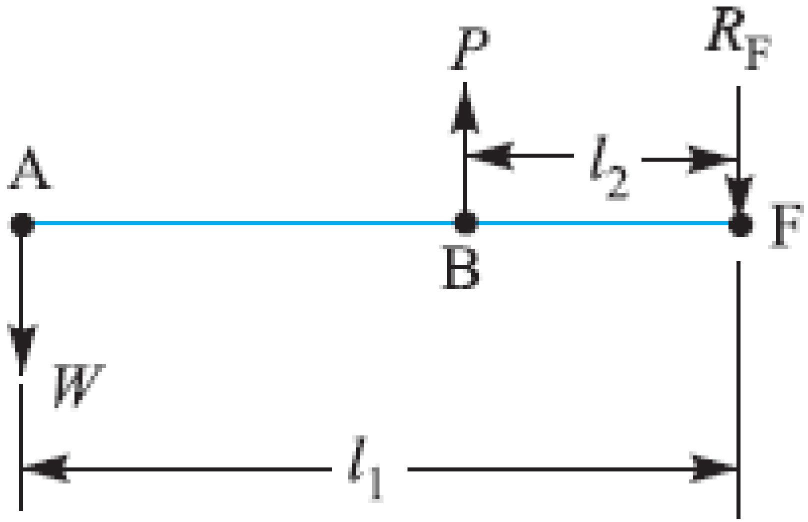
2) Fig. shows the two arm lever in which the effort arm is longer than the load arm and mechanical advantages than one.

Such types of lever used in boiler safety valve.



3) Fig. shows the two arm lever in which the effort arm is smaller than the load arm and mechanical advantage is less than one.

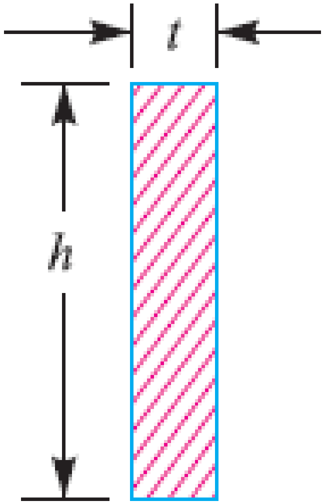
Such types of levers are used in stapler and forceps.



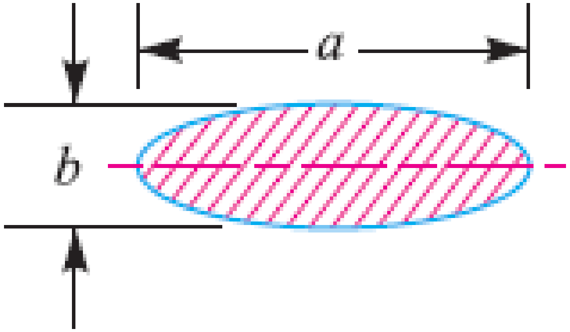
Why levers are tapered at the end?

- The thickness of lever is kept uniform throughout.
- The width of the lever is tapered from boss to the handle because the arm is subjected to varying bending moment which is maximum near the boss and decreases to the end and also for easy gripping of the hand on lever.

Cross Section of Levers / Handles



Rectangular section.



Elliptical section.

Section modulus for different sections

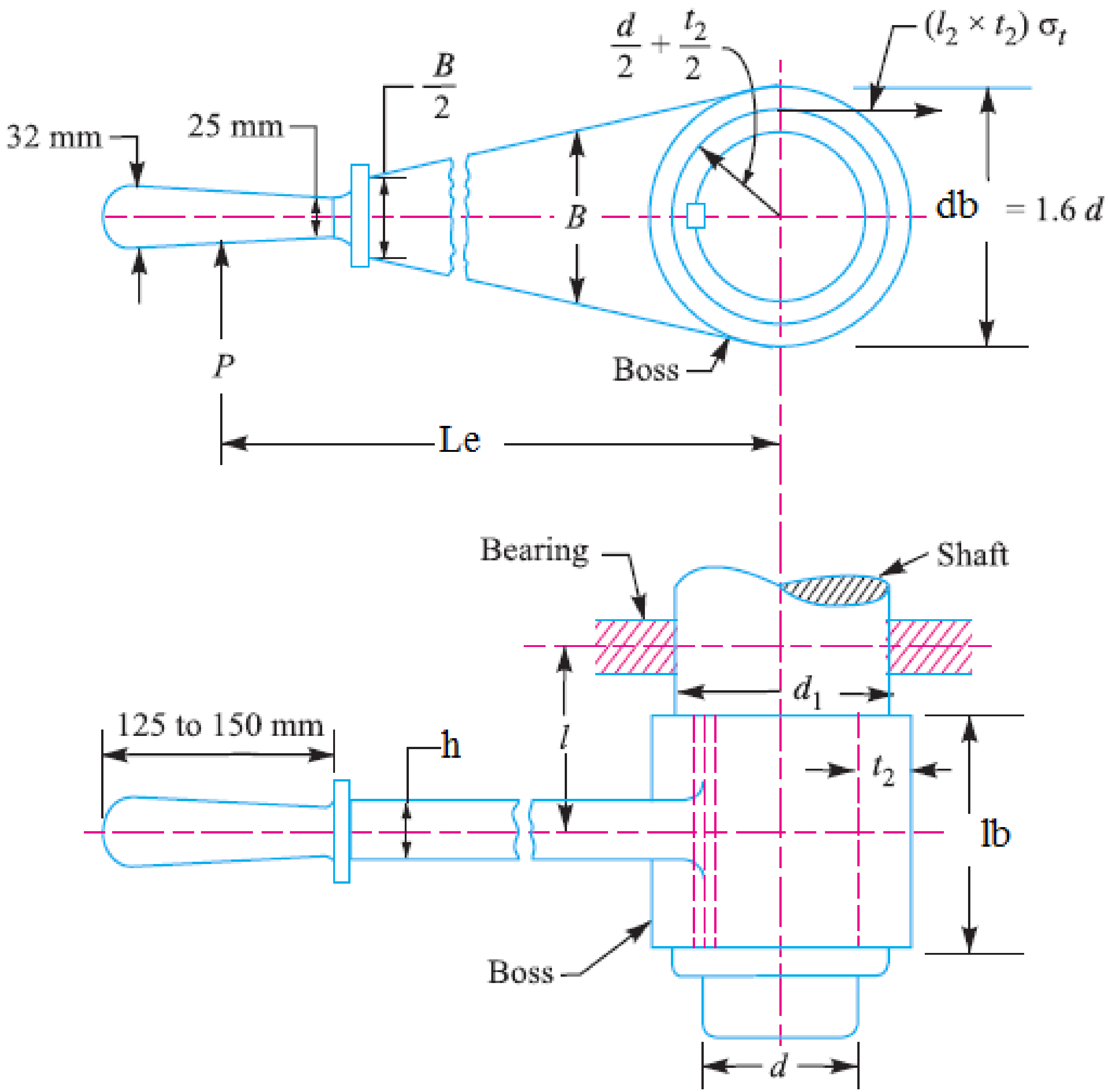
For rectangular section

$$Z = \frac{th^2}{6}$$

For Elliptical section

$$Z = \frac{\pi}{32} \times b \times a^2$$

Design of Hand Lever



$$Let, P = Force\,of\ effort\ applied\ at\,the\,handle\,in\ N$$

$$L_e = Effective\ length\ of\ the\,lever\ arm\,in\,mm.$$

$$l = Overhead\,length\ of\ the\,shaft\,in\,mm.$$

$$\sigma_t = Permissible\,tensile\ stress\,of\ the\,lever\ in\ N / mm^2$$

$$\tau_s = Permissible\,shear\ stress\,of\ the\,lever\ in\ N / mm^2$$

1) The maximum effort or force applied by a man may be assume as 300 N to 400 N

2) Due to the force applied at handle at a length 'L_e' the shaft is subjected to twisting moment (torque).

$$\therefore T = P \times L_e \quad ----(1)$$

The diameter of shaft (d) is obtained by considering the shaft under pure torsion.

$$\therefore T = \frac{\pi}{16} \times \tau_s \times d^3 \quad ----(2)$$

Equating equation (1) & (2), the diameter of shaft (d) is obtained

3) The diameter of shaft at the center of bearing (d_1) is obtained by considering the shaft in combined twisting and bending.

$$\therefore B.M. = M = P \times l \ \ and$$

$$Twisting\ moment = T = P \times L_e$$

$$\therefore Equivalent\ twisting\ moment = T_e = \sqrt{M^2 + T^2}$$

$$T_e = \sqrt{(Pl)^2 + (PL_e)^2} = P \times \sqrt{l^2 + L_e^2} \ ----(3)$$

$$Also\ equivalent\ twisting\ moment\ is$$

$$T_e = \frac{\pi}{16} \times \tau_s \times d_1^3 \ ------(4)$$

$$From\ equation\ (3)\ and\ (4), d_1\ is\ obtained$$

4) Diameter of the boss of the lever = $d_b = 1.6d$

5) Length of boss = l_b = d or 1.5d

6) Dimensions of key –

$$Let, w_k = Width\ of\ key\ in\ mm$$

$$t_k = Thickness\ of\ key\ in\ mm$$

$$l_k = l_b = Length\ of\ key\ in\ mm = length\ of\ boss$$

$$a)\ For\ rectangular\ key, w_k = \frac{d}{4}\ and\ t_k = \frac{2}{3} w_k$$

$$b)\ for\ square\ key, w_k = t_k = \frac{d}{4}$$

$i)$ *Considering key under shear failure due to torsional moment*

$$T = l_k \times w_k \times \tau_k \times \frac{d}{2}$$

$ii)$ *Considering key under crushing failure*

$$T = l_k \times \frac{t_k}{2} \times \sigma_{cr} \times \frac{d}{2}$$

From above equation w_k *and* t_k *is obtained*

7) Dimensions of lever cross section.

$$Consider\ the\ rectangular\ cross-section\ of\ lever.$$

$$Let,\ b = width\ of\ lever\ in\ mm.$$

$$t = depth\ or\ thickness\ of\ the\ lever\ in\ mm.$$

$$Taking\ b = 2t\ or\ 3t$$

$$The\ lever\ is\ subjected\ to\ bending\ moment.$$

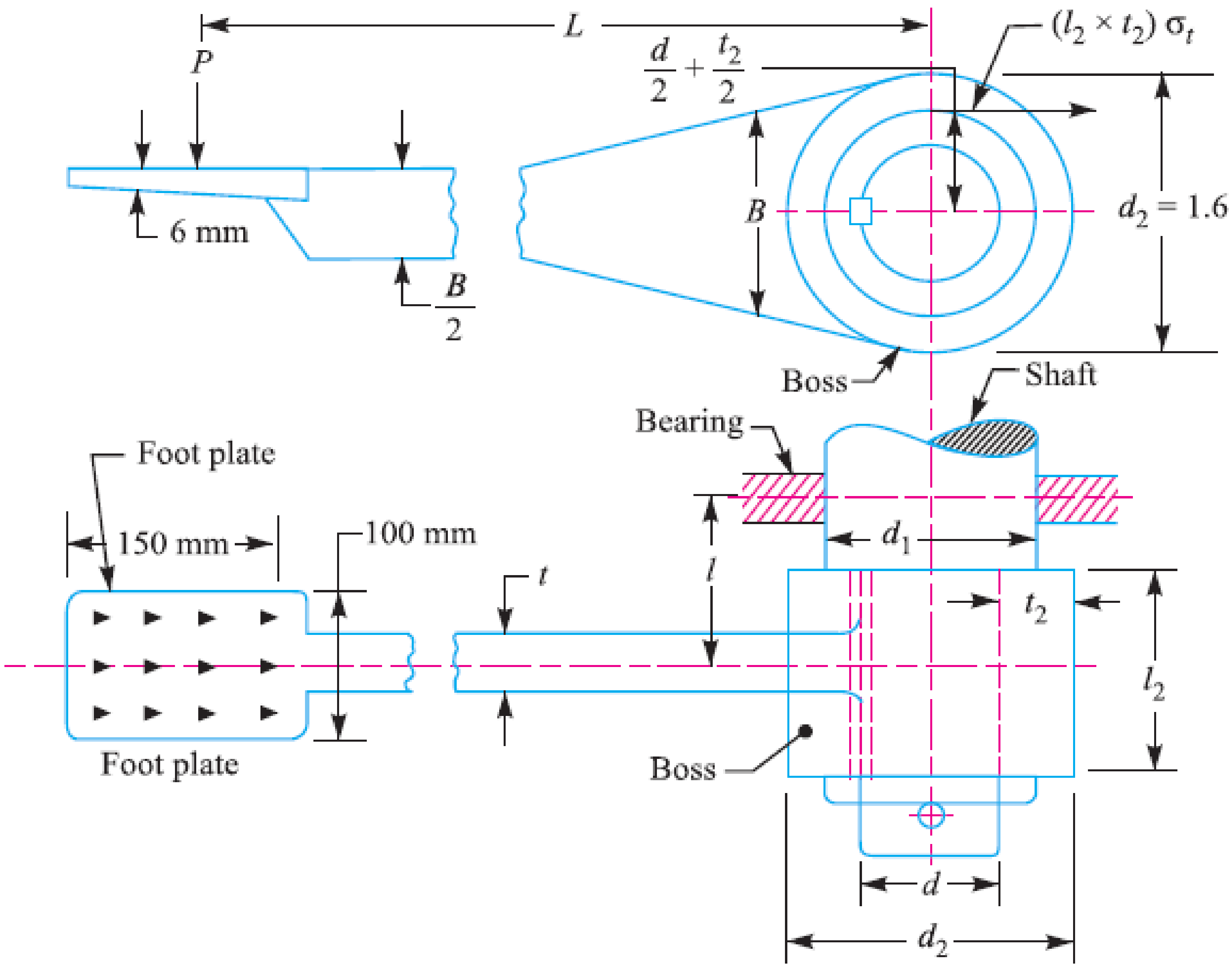
$$The\ maximum\ B.M.\ on\ the\ lever\ is\ taken\ near\ the\ boss.$$

$$M = P \times (L_e - \frac{d_b}{2})$$

$$\therefore \sigma_b = \frac{M}{Z} = \frac{P \times (L_e - \frac{d_b}{2})}{\frac{bt^2}{6}} = \frac{6P(L_e - \frac{d_b}{2})}{bt^2}$$

Design of Foot Lever

- The foot lever is design and designated in the same way as the hand lever.
- Only the difference is that hand is replace by foot plate.
- The force exerted by a single person by foot is taken as 600 N to 800 N.



L
P
$\frac{d}{2} + \frac{t_2}{2}$
$(l_2 \times t_2)\,\sigma_t$
6 mm
B
$\frac{B}{2}$
$d_2 = 1.6$
Boss
Shaft
Bearing
Foot plate
150 mm
100 mm
t
l
d_1
t_2
l_2
Foot plate
Boss
d
d_2

Problems

1) A foot lever is 1 m long from centre of shaft to point of application of 800 n load. Find

   i) Diameter of shaft ii) Dimensions of key

   iii) Dimensions of rectangular arm of foot lever

   at 60 mm from the centre of shaft assuming width of arm as 3 times thickness.

   Allowable tensile stress may be taken as

   73 N/mm^2 and shear stress as 70 N/mm^2.

2) Draw a neat labelled diagram of a hand lever and state how diameter of shaft and boss is calculated.

3) A foot lever is 1m from the centre of the shaft to the point of application of 800 N load. Find diameter of shaft if shear stress is 70 MPa,. (4M)

$$We\,know\,that, torque\,on\,the\,lever\,is$$

$$T = P \times L = 800 \times 1000$$

$$\therefore T = 800 \times 10^3\,Nmm$$

$$Also\,we\,know\,that,$$

$$T = \frac{\pi}{16} \times \tau \times d^3$$

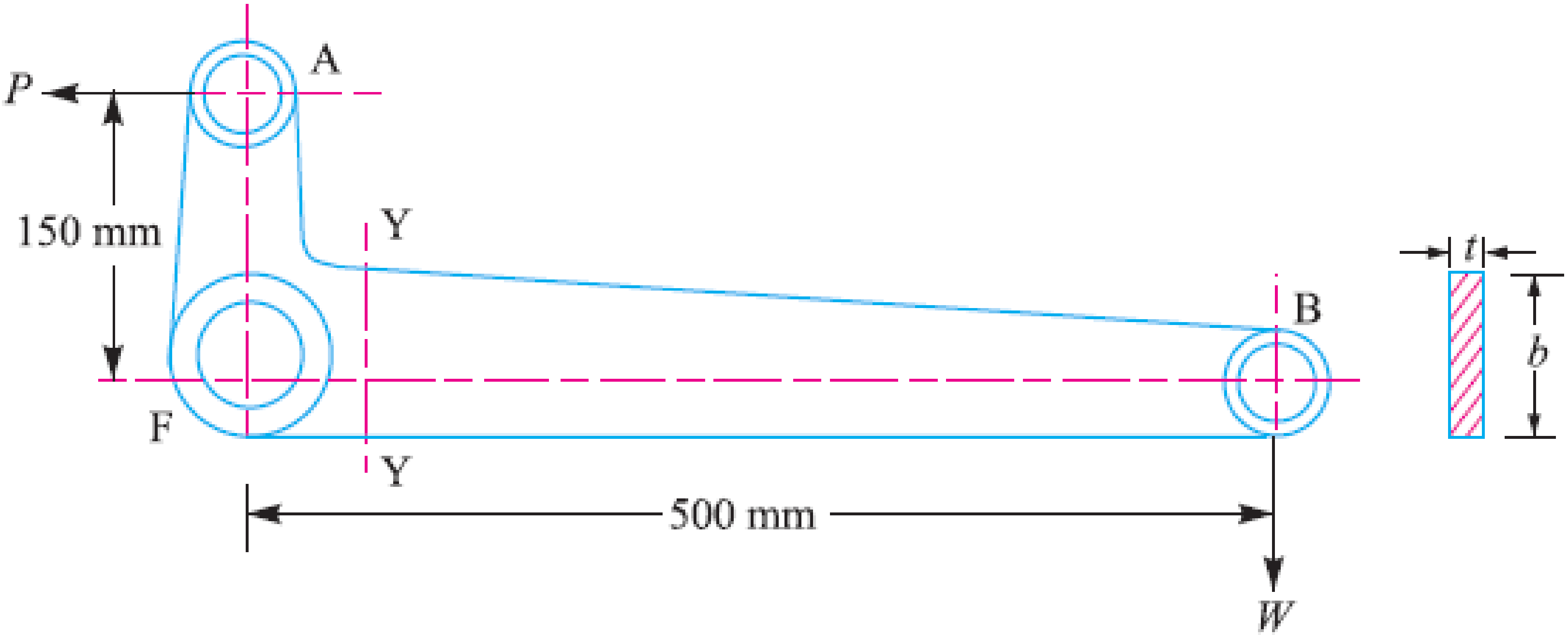
$$800 \times 10^3 = \frac{\pi}{16} \times 70 \times d^3$$

$$\therefore d = 40\,mm$$

4) Design a foot brake lever from the following data (8M)

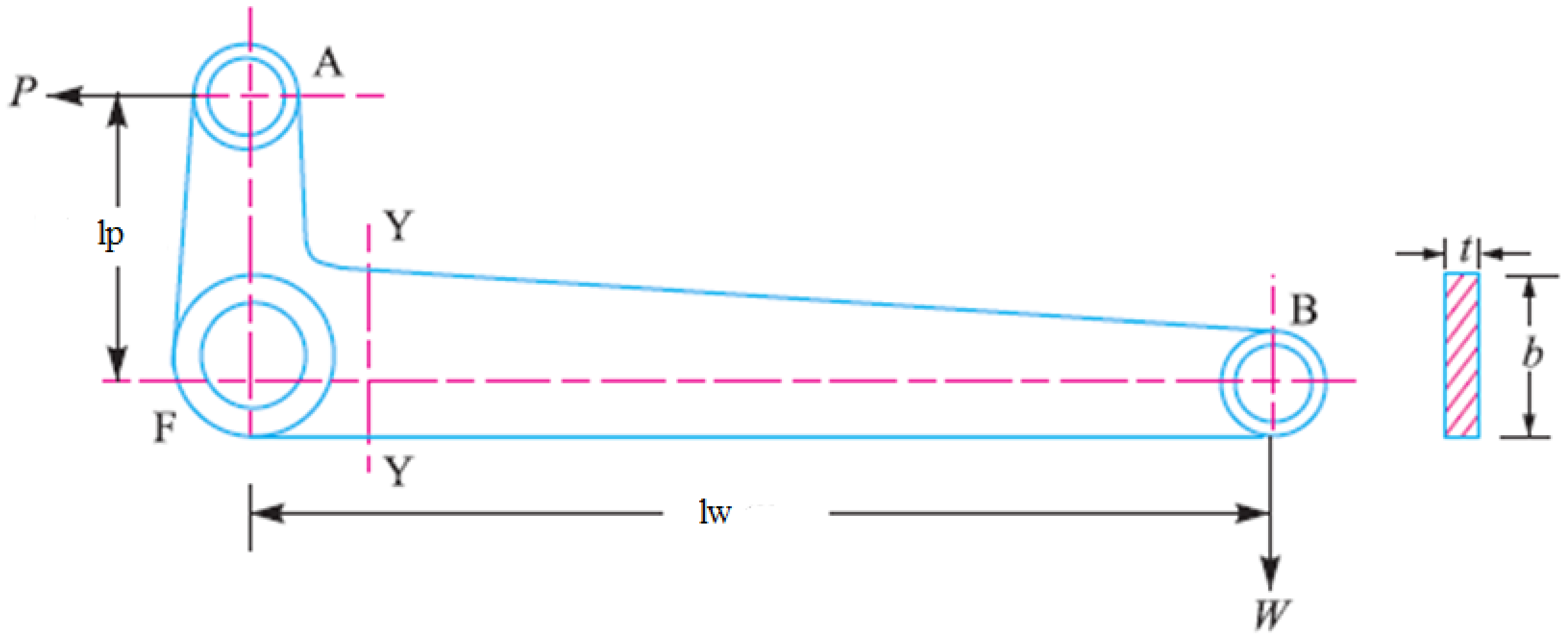
i) Length of lever from the C.G. of spindle to the point of application of load = 1m
ii) Maximum load on the foot plate = 800 N
iii) Overhang from nearest bearing = 100 mm
iv) Permissible tensile & shear stress = 70 MPa

Bell Crank Lever



In a bell crank lever, the two arms of the lever are at right angles. Such type of levers are used in railway signalling, governors of Hartnell type, the drive for the air pump of condensors etc. The bell crank lever is designed in a similar way as discussed earlier. The arms of the bell crank lever may be assumed of rectangular, elliptical or I-section.

Design of Bell Crank Lever



1) **Determination of effort/load as per problem.**

Let W be the load and P is the effort at the load arm of length l_w and effort arm l_e respectively. Final load/effort can be calculated by taking moment about the fulcrum.

$$W \times l_w = P \times l_e$$

2) **Determination of resultant fulcrum reaction (R_F)**

$$R_F = \sqrt{W^2 + P^2}$$

3) Design of fulcrum pin –

The fulcrum pin is supports the lever and allows to oscillate. Due to relative motion of lever on pin, fulcrum pin is subjected to bearing pressure and direct shear stress.

a) Fulcrum pin is designed by considering under bearing pressure

$$Assume \frac{l_p}{d_p} = 1.25$$

$$l_p = length\ of\ pin$$

$$d_p = diameter\ of\ fulcrum\ pin$$

$$P_b = \frac{R_F}{l_p \times d_p}$$

$$Find\ the\ diameter\ of\ pin\ (d_p)$$

b) Direct shear stress –

Fulcrum pin is subjected to double shear.

$$\tau = \frac{R_F}{2A} = \frac{R_F}{2 \times \frac{\pi}{4} d_p^2}$$

This equation is used for checking the shear stress.
If calculated shear stress is less than given shear stress then the design of fulcrum pin is safe.

4) Diameter of Boss of Lever –

The boss of lever is subjected to bending stress due to bending moment of lever.

d_o= outer diameter of the boss of lever

d_i= inner diameter of the boss of lever

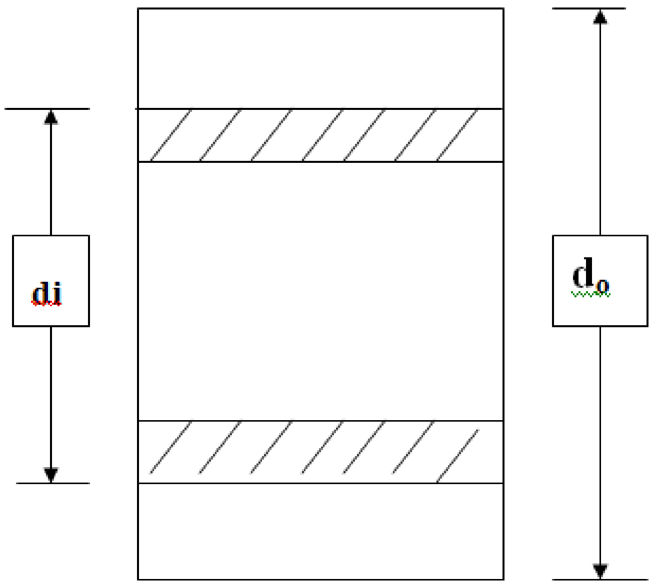
l_b= length of boss

There is relative motion between the fulcrum pin and the boss of a lever, a brass bush of 2 mm to 3 mm thickness should be insert in the boss of fulcrum lever as a bearing so that renewal became simple when wear occurs.

$d_i = d_p + (2 \times 3)$ if bush of 3 mm is used.

$d_i = d_p$ if bush is not used.

$d_o = 2d_p$



- The maximum bending stress induced in the boss of the lever is

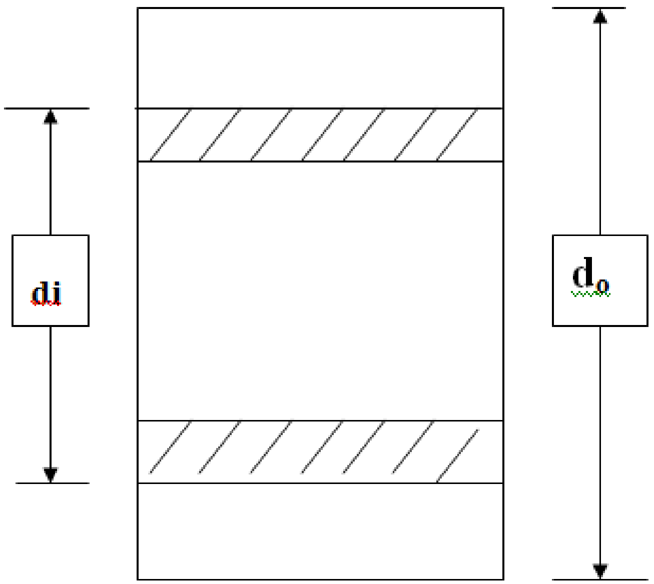
$$\sigma_b = \frac{M}{Z} = \frac{M}{(\frac{I}{y})}$$

$$\sigma_b = \frac{M \times y}{I}$$

$$\sigma_b = \frac{M \times \frac{d_o}{2}}{\frac{1}{12} \times l_b [d_o^3 - d_i^3]}$$

$$\sigma_b = \frac{6Md_o}{l_b [d_o^3 - d_i^3]}$$

$$where\ B.M. = M = W \times l_w = P \times l_e$$



From this equation outer diameter of boss can be obtained or bending stress should be checked

5) Design of Lever to find dimensions –

The lever is subjected to bending moment

The maximum bending moment acts near the boss .

$$M = P \times \left(l_e - \frac{d_o}{2} \right) = W \times \left(l_e - \frac{d_o}{2} \right)$$

The maximum bending stress induced is

$$\sigma_b = \frac{M}{Z}$$

Where Z is the section modulus of cross section of lever that may be rectangular or elliptical

a) Consider the rectangular C/s of the lever -

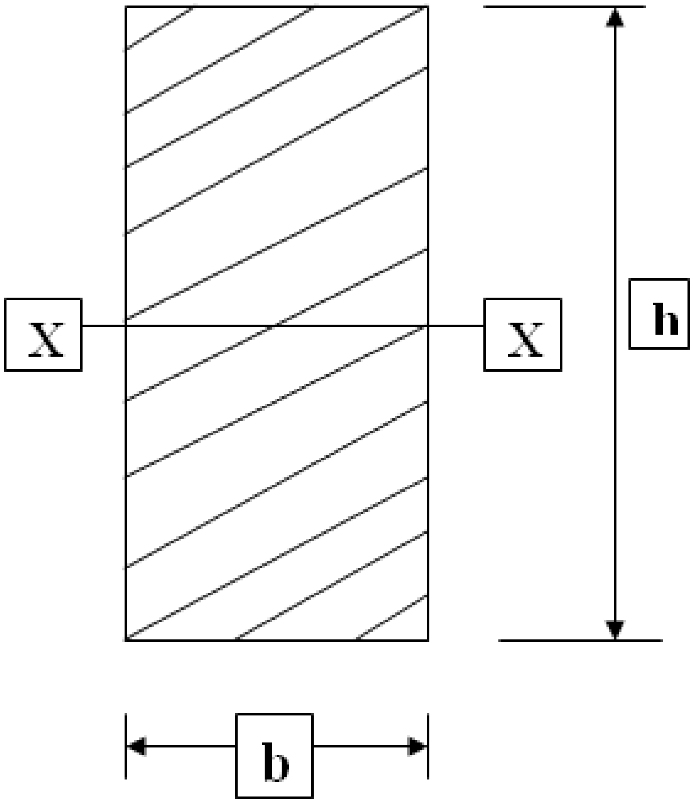
$$h = depth\ of\ lever$$

$$b = thickness\ of\ lever$$

$$Assume\ h = 2b\ to\ 4b$$

$$\therefore Z = \frac{I}{y} = \frac{\frac{1}{12}bh^3}{\frac{h}{2}}$$

$$\therefore Z = \frac{bh^2}{6}$$

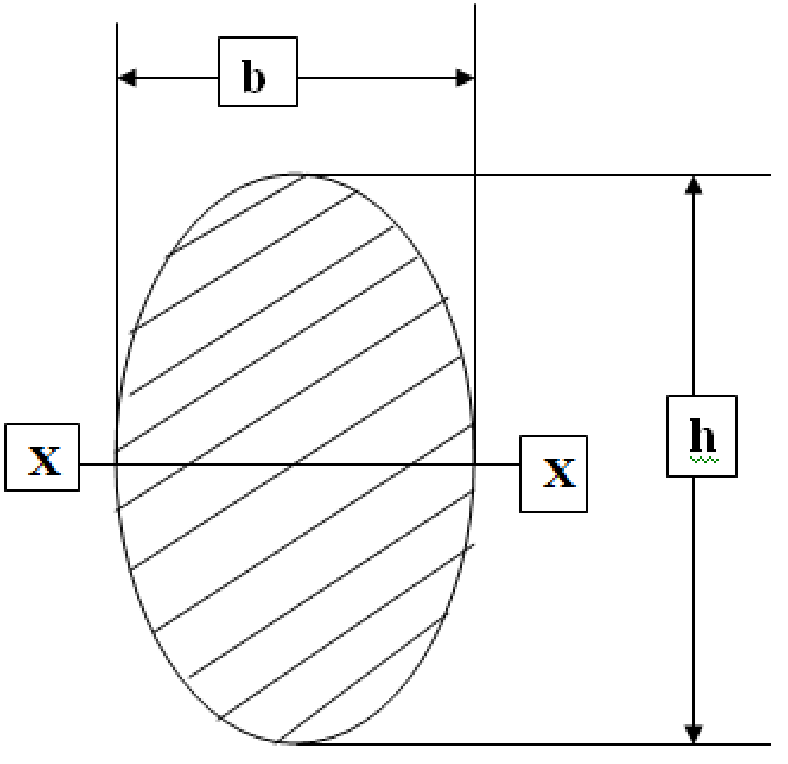


b) For Elliptical Section –

$$where, b = thickness\ of\ the\ lever\ i.e. \min or\ axis$$

$$h = Depth\ or\ height\ of\ the\ lever\ i.e. major\ axis$$

$$h = 2b\ to\ 2.5b$$



$$Z = \frac{I}{y}$$

$$Z = \frac{\left(\frac{\pi b h^3}{64}\right)}{\left(\frac{h}{2}\right)}$$

$$Z = \frac{\pi b h^2}{32}$$

Problems

1) A right angled bell crank lever having one 500 mm long and another arm is 150 mm is used to lift a load of 5 KN. The permissible stresses for pin and lever is 80 MPa in tension and compression and 60 MPa in shear. The bearing pressure on pin is not to exceed 10 MPa. Determine the dimensions of rectangular cross section of the lever and pin diameter.

2) A right angle bell crank lever having one arm 700 mm and other 400 mm long. The load of 1.75 KN is to be raised acting on a pin at the end of 700 mm arm and effort is applied at the end of 400 mm arm. The lever consists of a steel forgings, turning on a point at the fulcrum. The permissible stresses are in tension and compression are 80 N/mm2 and 60 N/mm2 in shear. The bearing pressure on the pin is not to exceed 6 N/mm2.

Find

i) Diameter and length of fulcrum pin.

ii) Thickness (t) and depth (b) of rectangular C/s of the lever (Assume b = 3t)

Design of C clamp & offset link

- Some machine component are subjected to two or three types of stresses such a combination of stress known as combine stresses.
- When the line of action of an external load is parallel but non co-axial with the centroidal axis of the component, this type of load called as eccentric load and the distance between the two axis is called as eccentricity (e).

Design procedure of such types

1) **Direct stress –**

The magnitude of the direct stress induced in the machine component.

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

$$where\ A = cross\ \sec tional\ area$$

2) Bending Stress –

The magnitude of bending stress induced in machine component is given by

$$\sigma_b = \frac{M}{\frac{I}{y}} = \frac{M}{Z}$$

$$where\, Z = \frac{I}{y} = Section\ \mathrm{mod}\, ulus$$

$$M = P \times e = bending\ moment$$

3) Resultant stress –

The resultant stress are obtained by the principle of super position.

$$\sigma_R = -\sigma_d + \sigma_b = -(\frac{P}{A}) + \frac{M}{Z}$$

$$\sigma_R = -\sigma_d - \sigma_b = -(\frac{P}{A}) - (\frac{M}{Z})$$

The positive sign indicate the tensile stress while negative sign indicate the compressive stress.

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Design of Knuckle Joint

- A knuckle joint is used to connect two rods which are under the action of tensile load, when small amount of flexibility or angular moment is necessary.
- The line of action of load is always axial.
- The knuckle joint consist of three major parts.

a) Single eye b) Double eye c) Knuckle pin

- The single eye is formed at the one end and double eye is formed at the other end of the rod.
- The single eye fit into fork or double eye, both the parts are connect by pin inserted through eye.
- The knuckle pin has a head at one end and collar and taper pin or split pin at other end.
- The ends of the rod are of octagonal forms for improving the gripping at the time of repairs.
- Knuckle Joint Video

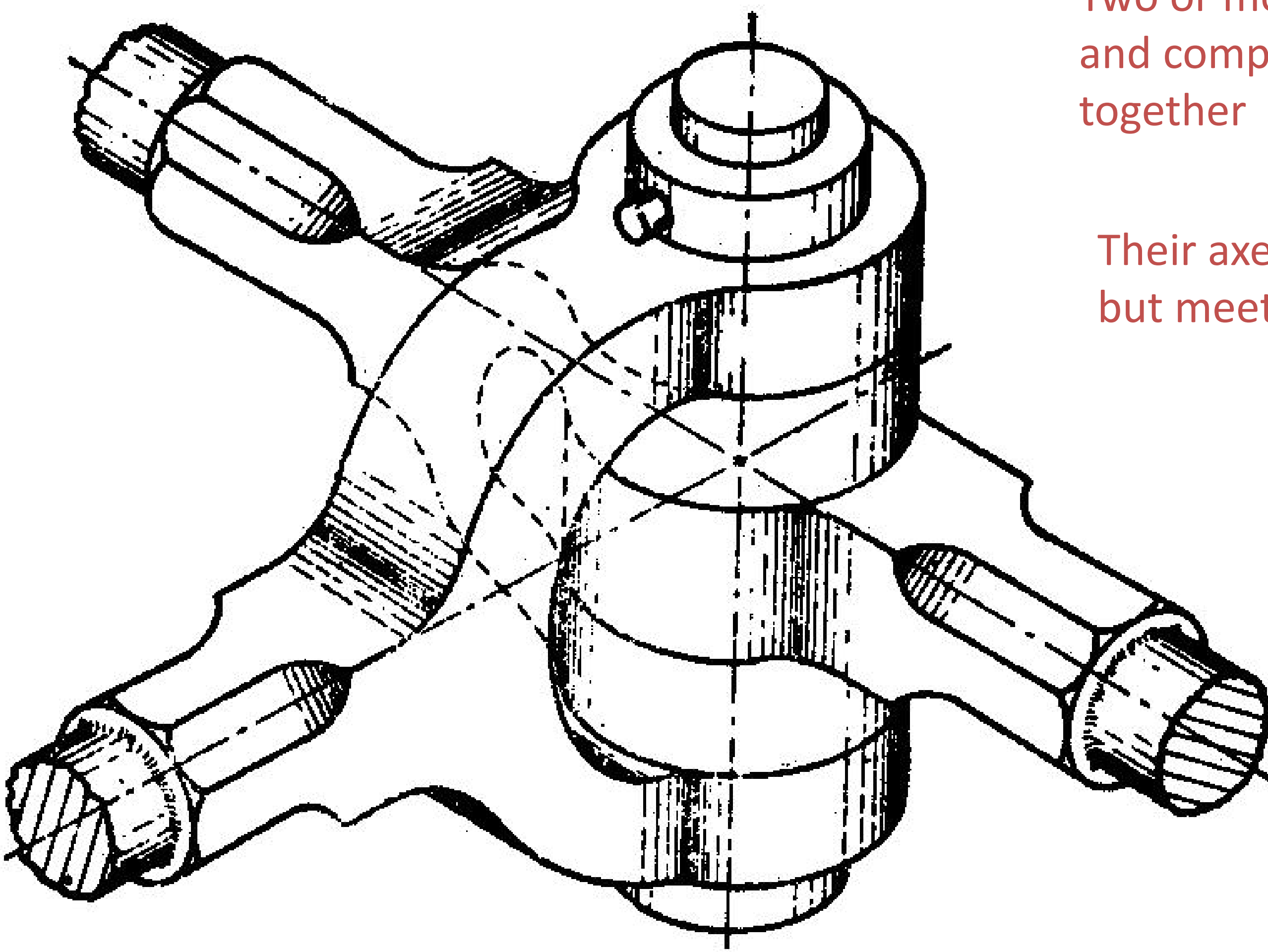
- **Function of Split pin –**

It holds collar and prevent lifting or ejecting the knuckle pin from the joint.

Applications of knuckle Joint –

1. Tie rod of roof truss
2. Link of roller chain
3. Tension link in bridge structure
4. Tie rod joint of jib crane.

Knuckle joint



Two or more rods subjected to tensile and compressive forces are fastened together

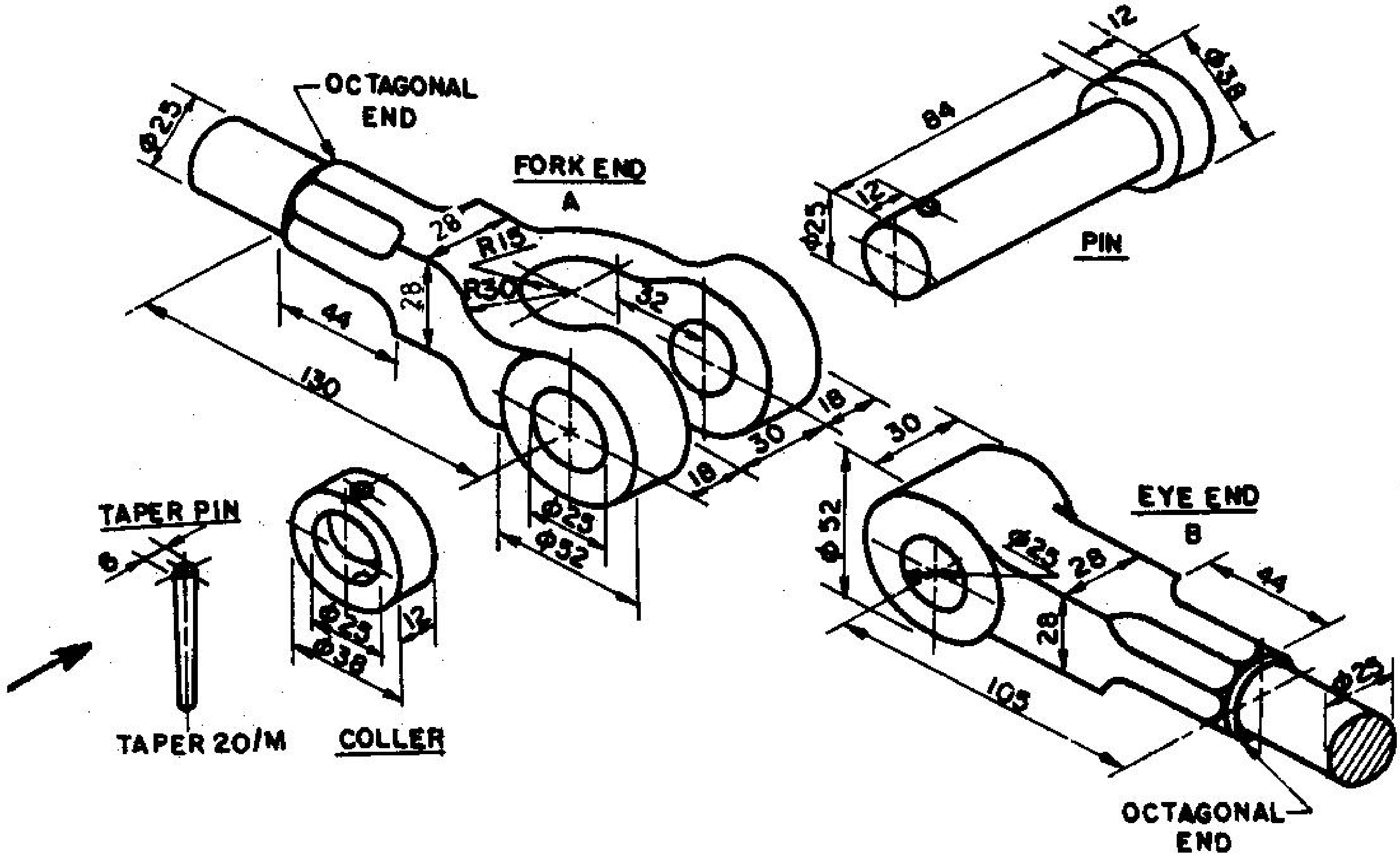
Their axes are not in alignments but meet in a point

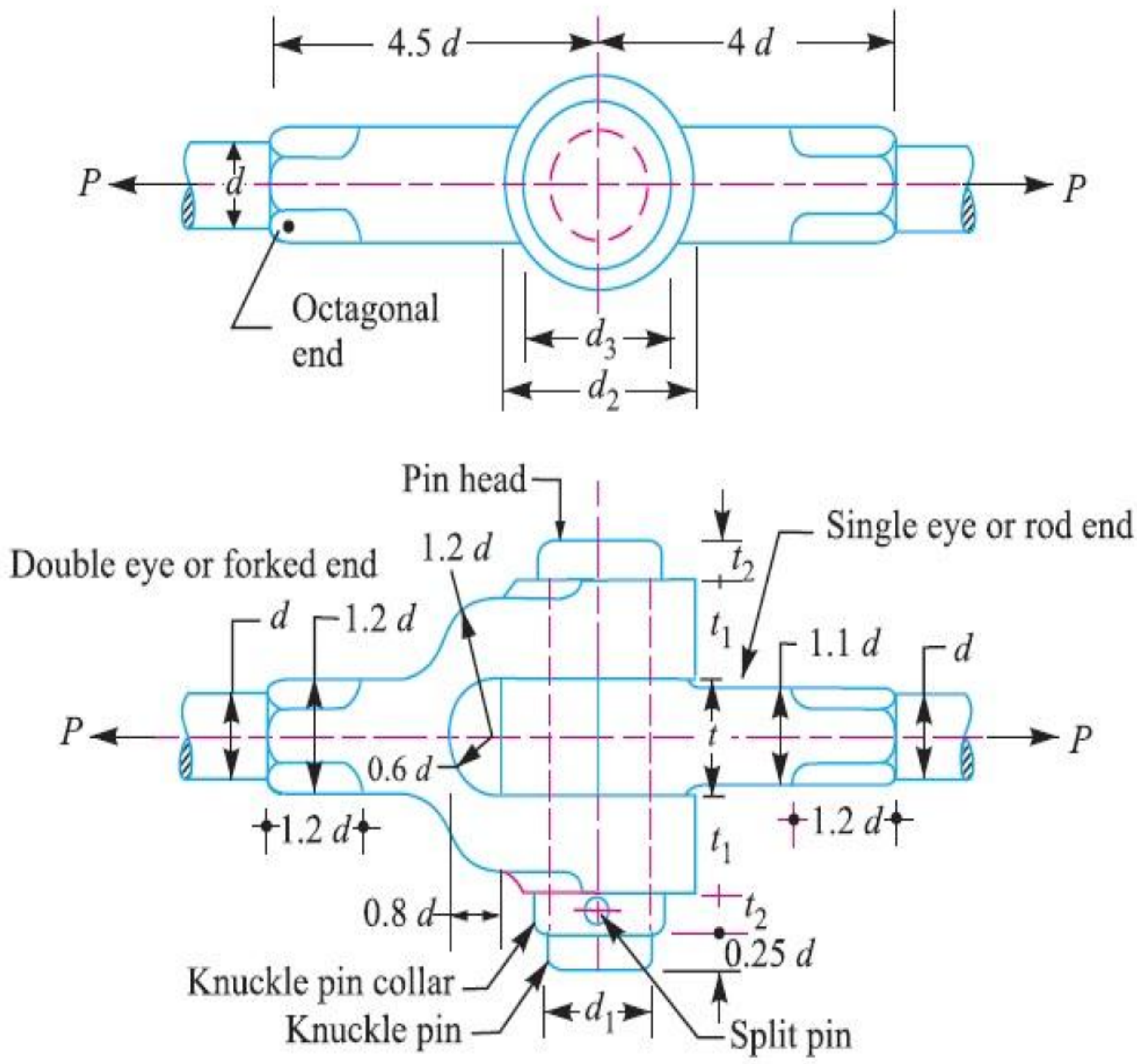
The joint allows a small angular moment of one rod relative to another

It can be easily connected and disconnected

Applications: Elevator chains, valve rods, etc

Knuckle joint





4.5 d
4 d
P
d
P
Octagonal end
d3
d2
Pin head
Single eye or rod end
1.2 d
Double eye or forked end
t2
d
1.2 d
t1
1.1 d
d
P
t
P
0.6 d
1.2 d
1.2 d
t1
0.8 d
t2
0.25 d
Knuckle pin collar
Knuckle pin
d1
Split pin

- Let P = tensile load acting on rod

d = diameter of rod

d_1 = diameter of pin

d_2 = outer diameter of eye.

d_3 = diameter of knuckle pin head & collar.

t = thickness of single eye.

t_1 = thickness of fork.

t_2 = thickness of pin head.

Empirical Relations

Diameter of pin = d_1 = d

Outer diameter of eye = d_2 = 2d

Diameter of knuckle pin head & collar = d_3 = 1.5d

Thickness of single eye = t = 1.25d

Thickness of fork = t_1 = 1.75d

Thickness of pin head t_2 = 0.5d

Design Procedure

1) Failure of rod in tension

$$\sigma_t = \frac{P}{\frac{\pi}{4} d^2} ----(1)$$

$$From\, this\, equation\ d\ is obtained$$

2) Failure of knuckle pin

a) Failure of knuckle in double shear.

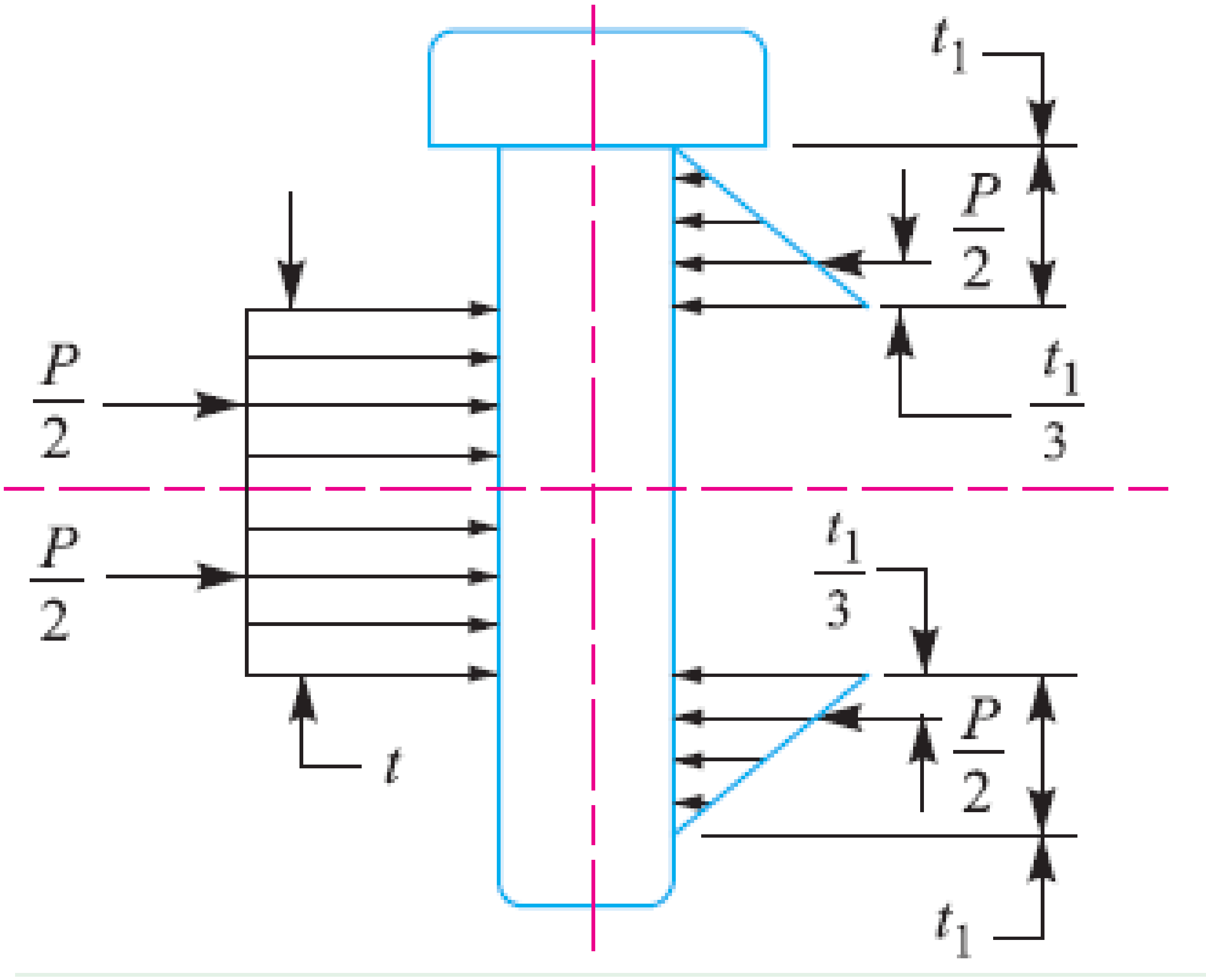
$$\tau = \frac{P}{2 \times \frac{\pi}{4} d_1^2} ---(2)$$

$$From\, this\ equation\ diameter\ of\ knuckle\, pin$$

$$(d_1)\, is\ obtained$$

This assume that, there is no slack or clearance, but in actual practice pin is loose in fork to permit angular moment of one with respect to other. So it is subjected to bending moment in addition to shear

b) Considering bending failure of knuckle pin



$$M = \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2}\right) - \frac{P}{2} \times \frac{t}{4}$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2} - \frac{t}{4}\right)$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)$$

$$Section \bmod ulus = Z = \frac{\pi}{32} d_1^3$$

$$\therefore \sigma_t = \frac{M}{Z} = \frac{\frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)}{\frac{\pi}{32} d_1^3} --- (3)$$

From this equation (d_1) *is obtained*

3) Failure of single eye end in tension

- Single eye end may tear of due to tension.

$$\sigma_t = \frac{P}{(d_2 - d_1) \times t} - - - - (4)$$

$$From\, this\, equation\, (\sigma_t)\ for\, the \sin gle$$

$$eye\, end\ may\, be\, checked.$$

4) Failure of Single eye end in shearing

- Considering the shear failure of single eye end.

$$\tau = \frac{P}{(d_2 - d_1) \times t} ---- (5)$$

$$From\, this\, equation\, shear\, stress\, (\tau)\, for$$

$$\sin gle\, eye\, end\, may\, be\, checked.$$

5) Failure of single eye end in crushing

- Considering the crushing failure of single eye end.

$$\sigma_{cr} = \frac{P}{d_1 \times t} ----(6)$$

$$From\, this\, equation\, crushing\, stress\, (\sigma_{cr})$$

$$for\, the\, \sin gle\, eye\, or\, pin\, may\, be\, checked.$$

6) Failure of forked end in tension

- Considering the tensile failure of double eye of forked end.

$$\sigma_t = \frac{P}{(d_2 - d_1) \times 2t_1} ---(7)$$

$$From\, this\; equation\; tensile\; stress\,(\sigma_t)$$

$$for\; the\;\; forked\, end\;\, may\, be\, checked.$$

7) Failure of forked end in shear

- Considering the shear failure of forked end.

$$\tau = \frac{P}{(d_2 - d_1) \times 2t_1} - - - (8)$$

From this equation shear stress (τ) for the forked end may be checked.

8) Failure of forked in crushing

- Considering the crushing failure of forked end.

$$\sigma_{cr} = \frac{P}{d_1 \times 2t_1} ---(9)$$

From this equation, the crushing stress in the forked may be chacked.

1. First of all, find the diameter of the rod by considering the failure of the rod in tension. We know that tensile load acting on the rod,

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

where d = Diameter of the rod, and

σ_t = Permissible tensile stress for the material of the rod.

2. After determining the diameter of the rod, the diameter of pin (d_1) may be determined by considering the failure of the pin in shear. We know that load,

$$P = 2 \times \frac{\pi}{4} (d_1)^2 \tau$$

A little consideration will show that the value of d_1 as obtained by the above relation is less than the specified value (*i.e.* the diameter of rod). So fix the diameter of the pin equal to the diameter of the rod.

3. Other dimensions of the joint are fixed by empirical relations as discussed in Art. 12.13.

4. The induced stresses are obtained by substituting the empirical dimensions in the relations as discussed in Art. 12.14.

In case the induced stress is more than the allowable stress, then the corresponding dimension may be increased.

Sr. No.	Knuckle Joint	Cotter Joint
1	It takes only tensile load	It takes tensile as well as compressive load.
2	It allows angular movement between rods	It can not allow angular movement
3	It is subjected to baring failure.	It is not subjected to bearing failure.
4	No taper or clearance provided on knuckle pin.	Taper or clearance will be provide on cotter.
5	Ex – tie bar, links od bicycle chain, joint for rail shifting mechanism	Ex- cotter foundation bolt, joining of two rods with a pipe, joining piston rod with c/s head.

Problems

Example 12.7. *Design a knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.*

Cotter joint

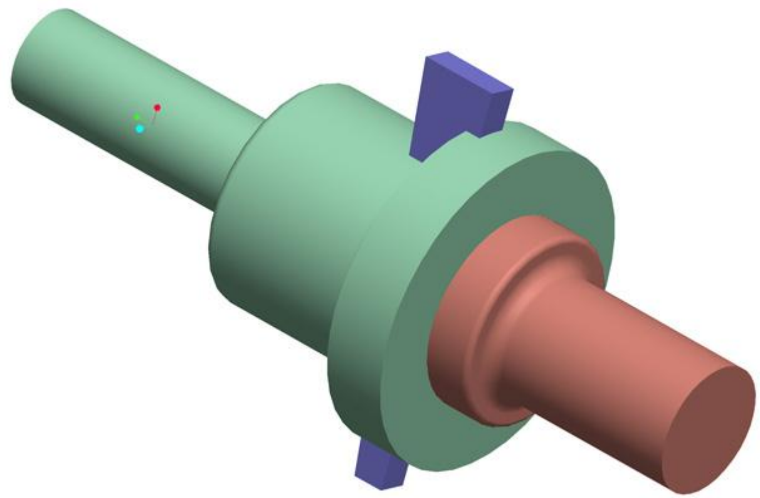


FIG 01: ASSEMBLED COTTERED JOINT

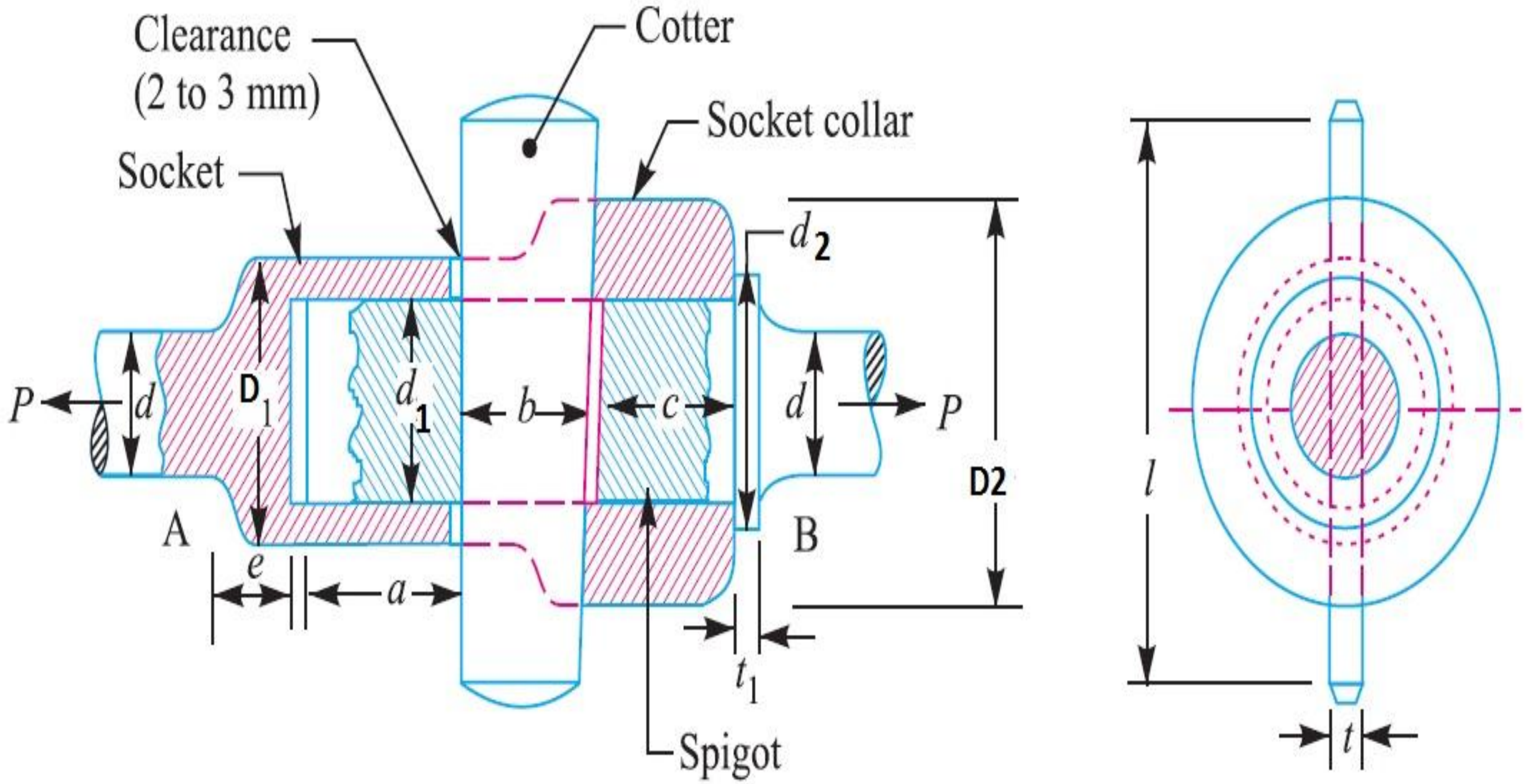
- Cotter Joint Video

- Cotter is flat wedge shape piece of rectangular c/s and width is tapered (either one side or both side) from one end to another for an easy adjustment.
- The taper varies from 1 in 48 to 1 in 24 and it may be increased to 1 in 8 if locking is provided.
- A cotter joint is temporary fastening and using to connect rapidly two co-axial rods or bars which are subjected to tensile or compressive forces.
- Ex- Connection between piston rod and cross head of a steam engine, valve rod and its steam, steam engine connecting rod strap ends, etc.

Design Of Cotter Joint

- Cotter joint is also called as socket and spigot joint.
- It mainly consists of three parts.

1) Socket
2) Spigot
3) Cotter


Clearance
(2 to 3 mm)
Cotter
Socket
Socket collar
d_2
P
d
D_1
d_1
b
c
d
P
D2
A
B
e
a
t_1
Spigot
l
t

P= axial tensile or compressive force in rod.

d= diameter of rod in mm

d_1= diameter of spigot end or inside diameter of socket in mm

d_2= diameter of spigot collar in mm

D_1=outside diameter of socket in mm

D_2=diameter of socket collar in mm.

b=mean width of cotter in mm

C=thickness of socket collar in mm

t_1= thickness of spigot collar in mm

a= distance from the end of slot to the end of the spigot in mm

$$\sigma_t = permissible\, tensile\ stress\ for\ socket, spigot\ and\ \cot ter$$

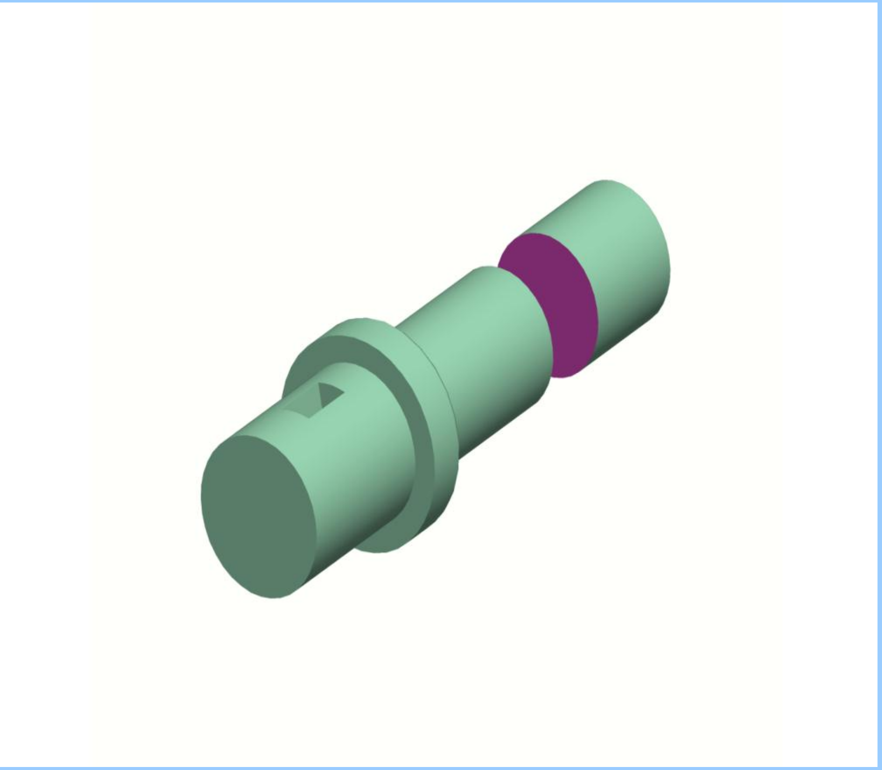
$$\sigma_c = \sigma_{cr} = permissible\, crushing\ stress\ for\ socket, spigot\ and\ \cot ter$$

$$\tau = permissible\, shear\ stress\ for\ socket, spigot\ and\ \cot ter$$

1) Consider the failure of rod in tension or compression due to axial force.

- Tensile stress induced in the rod is given by

$$\sigma_t = \frac{P}{(\frac{\pi}{4})d^2} - - - - - - - (1)$$



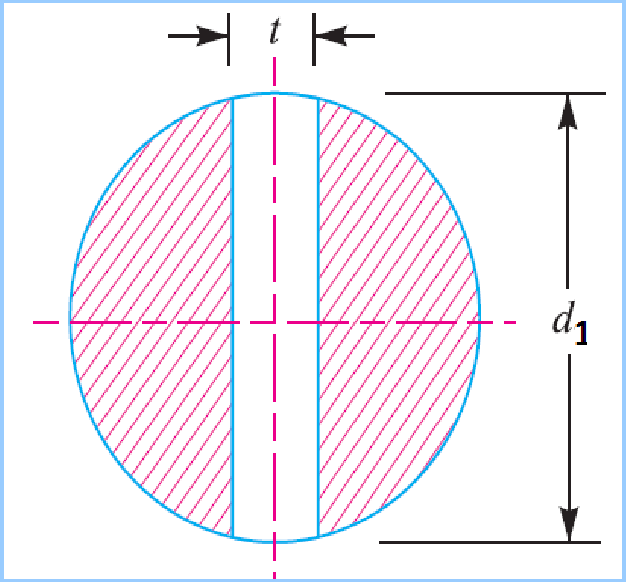
2) Design of Spigot End

a) Failure of spigot end in tension

$$\sigma_t = \frac{P}{\frac{\pi}{4} d_1^2 - d_1 t} - - - (2)$$

In actual practice thickness of cot*ter is*

$t = 0.3d$



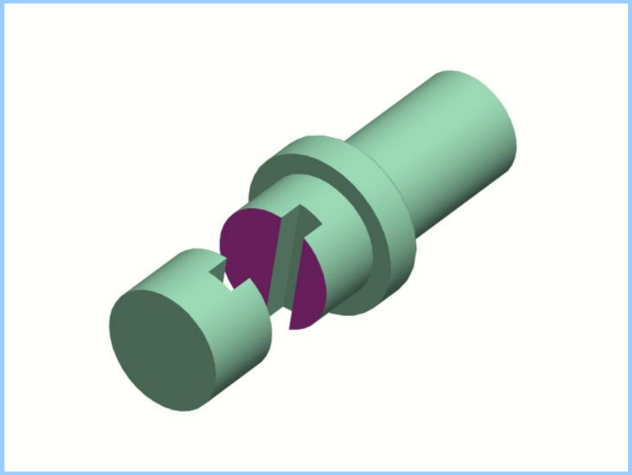
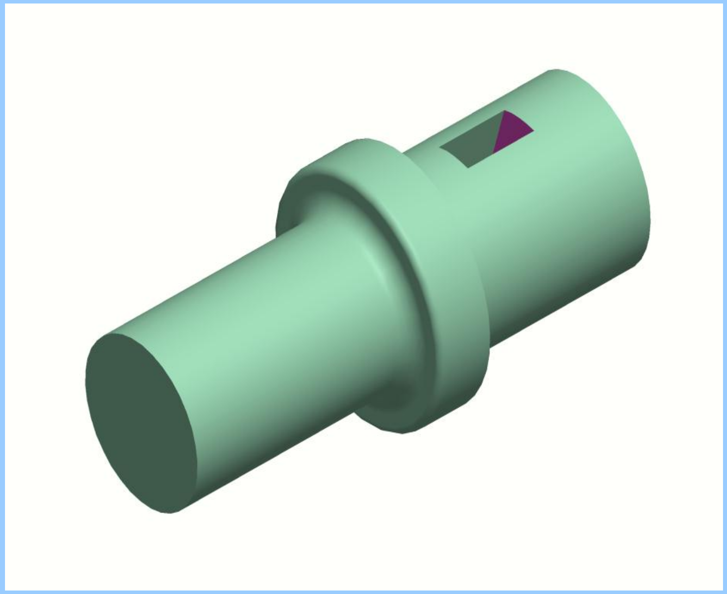


FIG 05: SPIGOT BREAKING IN TENSION ACROSS SLOT

b) Failure of spigot end under crushing in the slot of cotter

$$\sigma_{cr} = \frac{P}{d_1 t} - - - - - (3)$$

Select larger value of d1 from the eq.2 and 3

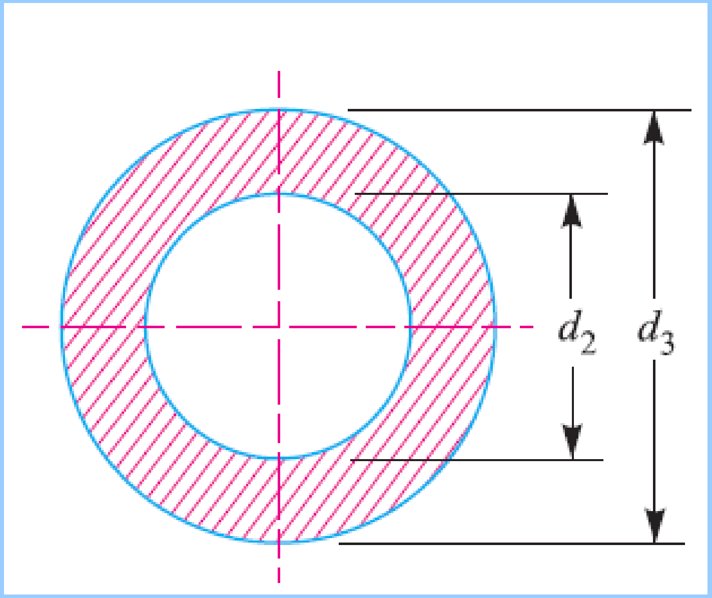


3) Design of spigot Collar

a) Crushing failure of spigot collar at the area between spigot collar and socket collar

$$\sigma_{cr} = \frac{P}{\frac{\pi}{4}(d_2^2 - d_1^2)} ----(4)$$

From above equation d_2 *is obtained*

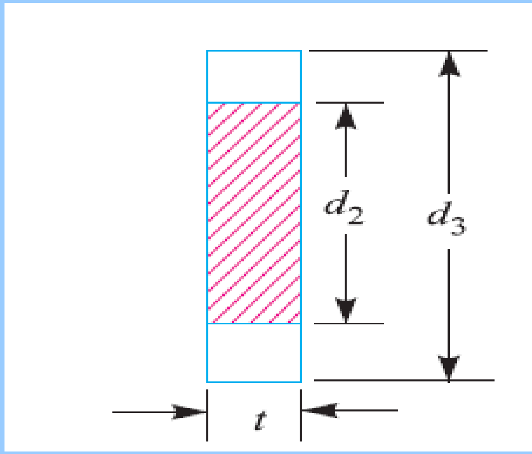


b) Failure of spigot collar in shear

Due to axial tensile load, the spigot collar will be subjected to direct shear stress along its circumference.

$$\tau = \frac{P}{\pi d_1 t_1} --- (5)$$

From above equation t_1 *is obtained*



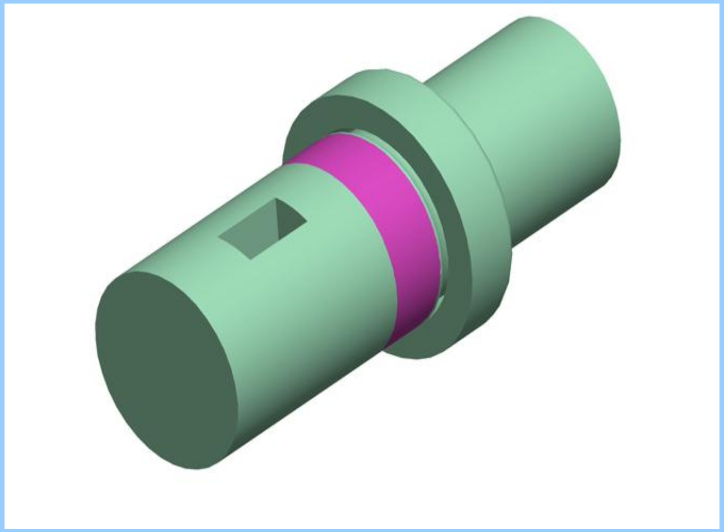


FIG 11: SHEARING AWAY OF THE COLLAR IN THE SPIGOT

c) Distance from end of slot to end of spigot

The spigot end is subjected to double shear

$$\tau = \frac{P}{2d_1 a} - - - (6)$$

From above equation a is obtained

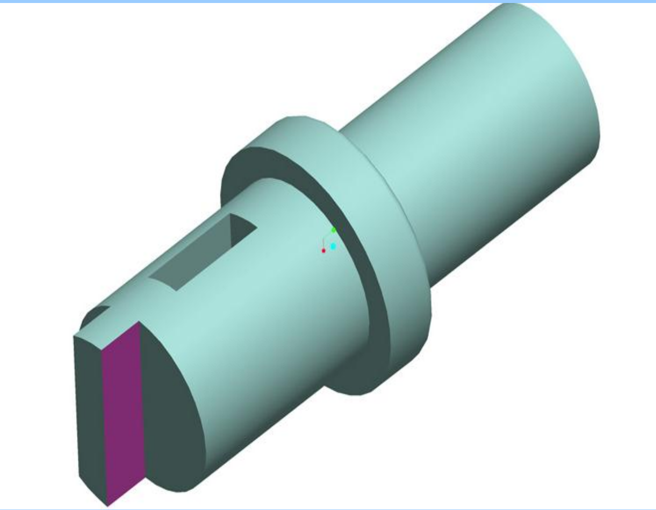
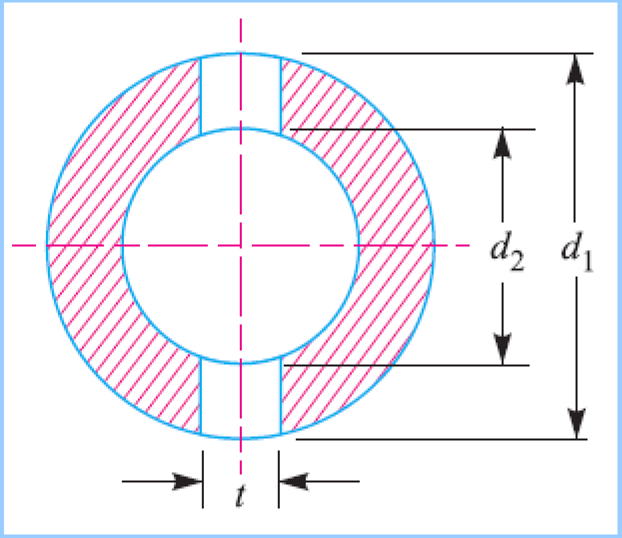


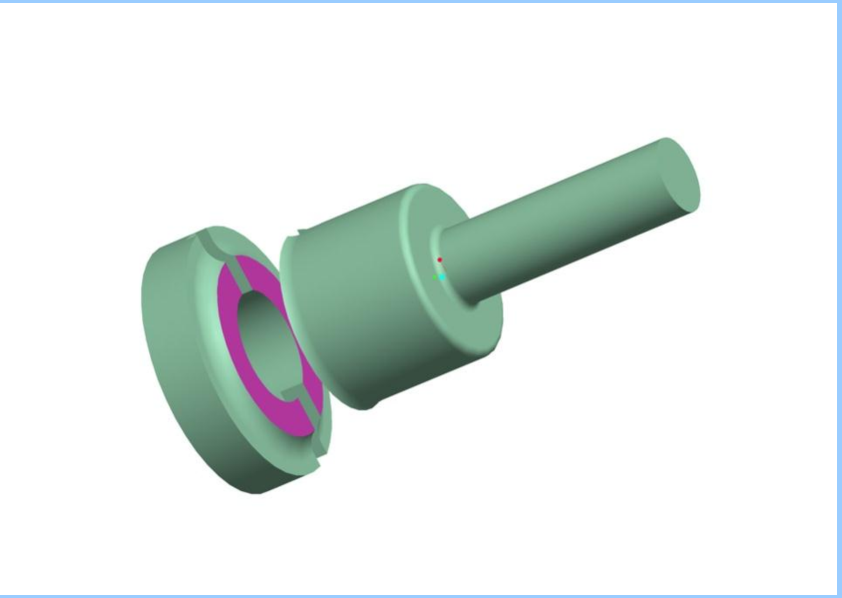
FIG 10: SHEARING OF SPIGOT END

4) Design of Socket

$$\sigma_t = \frac{P}{[\frac{\pi}{4}(D_1^2 - d_1^2)] - [(D_1 - d_1)t]} - - - (7)$$

$From\ above\ eq^n\ D_1\ is\ obtained$





5) Design of socket collar

Crushing failure of socket collar at the area between socket collar and cotter.

$$\sigma_{cr} = \frac{P}{(D_2 - d_1)t} ----(8)$$

$From\ above\ eq^n\ D_2\ is\ obtained$

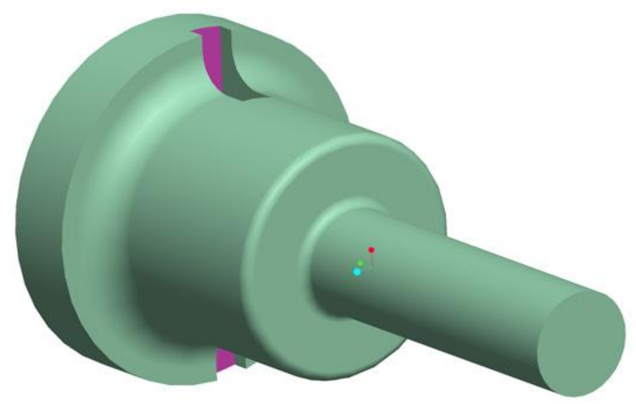
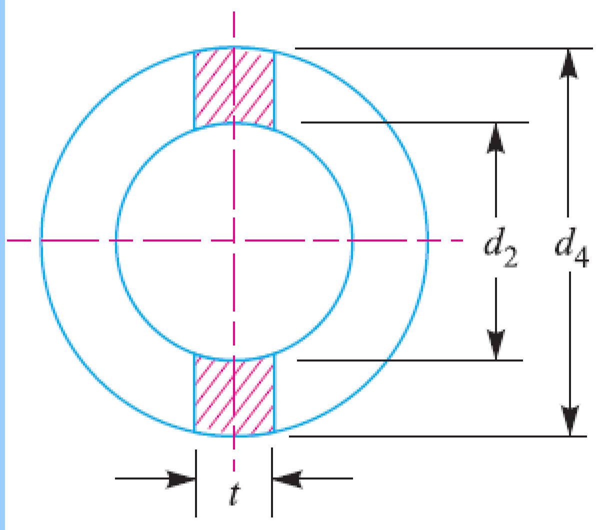


FIG 08: CRUSHING OF COTTER PIN AGAINST SOCKET

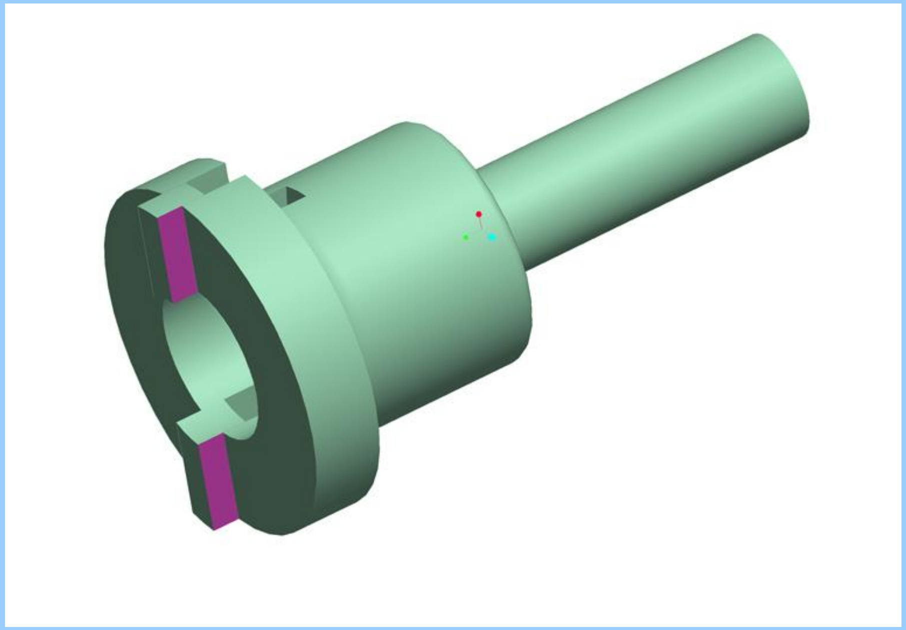


6) Thickness of socket collar

The socket collar is subjected to **double shear.**

$$\tau = \frac{P}{2(D_2 - d_1)C} ---(9)$$

$$From\,above\;eq^n\;C\;is\;obtained$$

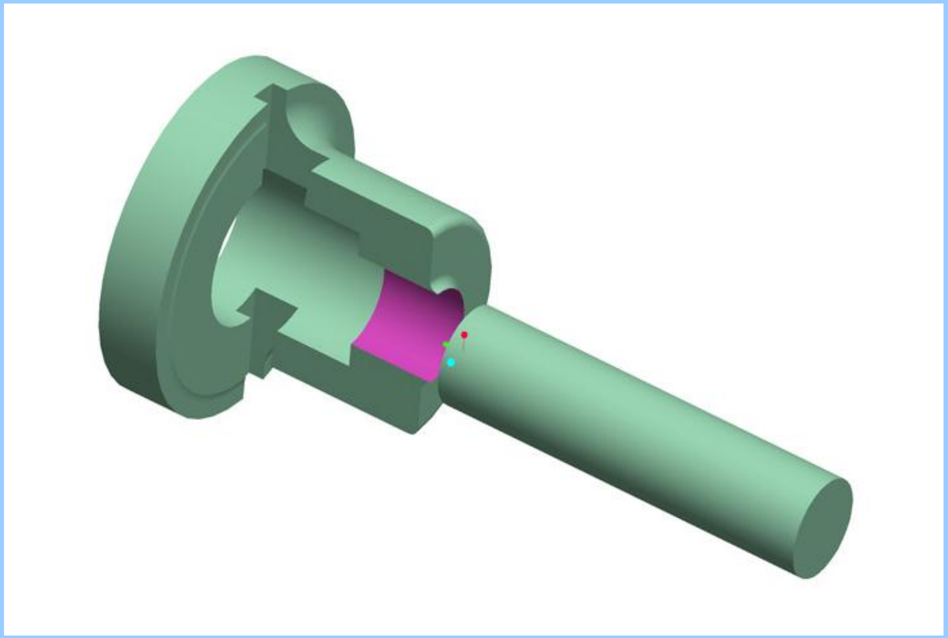


7) Find distance ‘e’ of the socket end

- The socket end is subjected to **shear failure.**

$$\tau = \frac{P}{\pi\, d\, e} - - - (10)$$

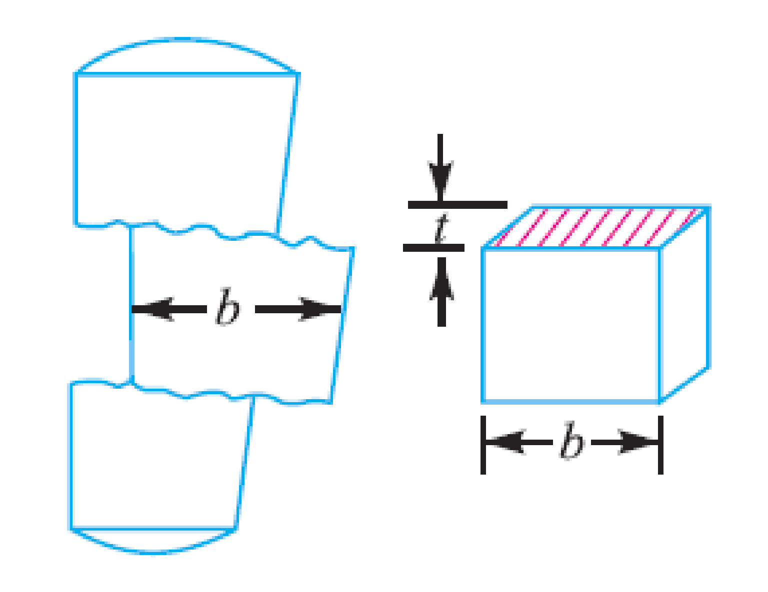
$$From\, above\, eq^{n}\, 'e'\, is\, obtaied$$

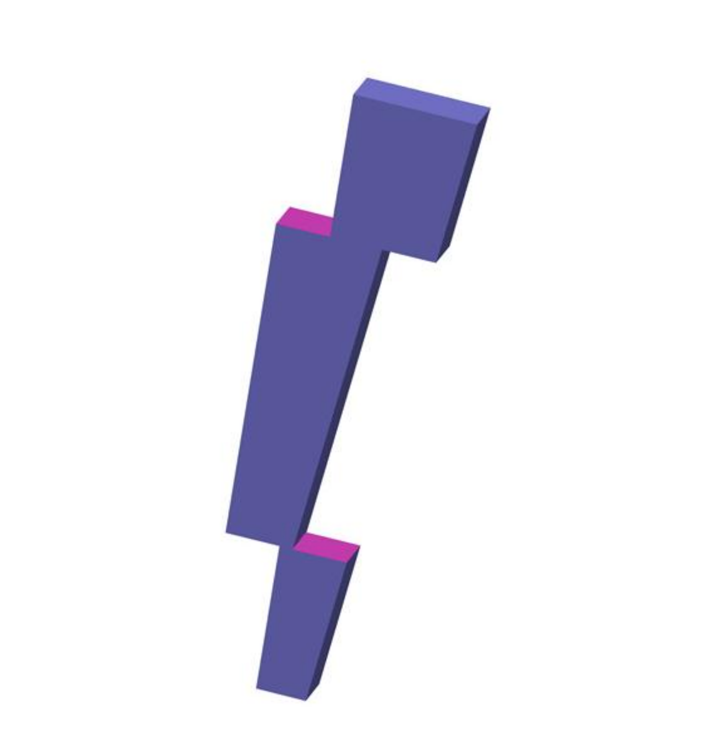


8) Design of cotter

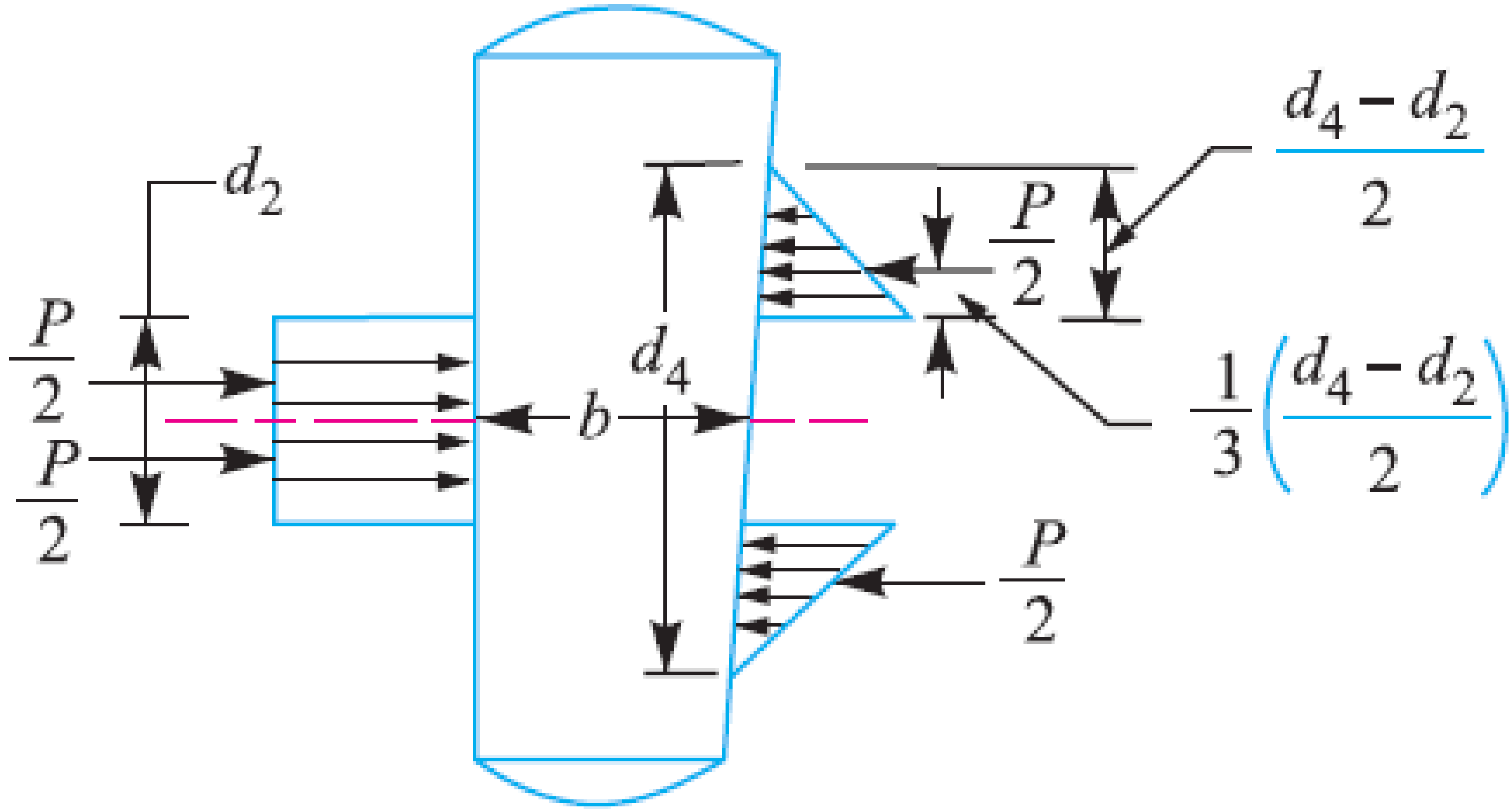
$$\tau = \frac{P}{2bt} - - - - (11)$$

$$From\,above\,eq^n\,b\,\,is\,obtained$$





<u>FIG 06: DOUBLE SHEARING OF COTTER PIN</u>



d_2
$\frac{P}{2}$
$\frac{P}{2}$
d_4
b
$\frac{d_4 - d_2}{2}$
$\frac{P}{2}$
$\frac{1}{3}\left(\frac{d_4 - d_2}{2}\right)$
$\frac{P}{2}$

b) Failure of cotter in bending

$$B.M. = \frac{P}{2}\left[\frac{1}{3}\times\left(\frac{D_2-d_1}{2}\right)+\frac{d_1}{2}\right]-\left(\frac{P}{2}\times\frac{d_1}{4}\right)$$

$$= \frac{P}{2}\left[\frac{D_2-d_1}{6}+\frac{d_1}{2}-\frac{d_1}{4}\right]$$

$$M_{\max} = \frac{P}{2}\left[\frac{D_2-d_1}{6}+\frac{d_1}{4}\right]$$

$$\therefore \sigma_b = \frac{M_{\max}}{Z}$$

$$= \frac{M_{\max}}{\left(\frac{t\,b^2}{6}\right)} = \frac{6M_{\max}}{t\,b^2}$$

$$\sigma_b = \frac{6\times\frac{P}{2}\left(\frac{D_2-d_1}{6}+\frac{d_1}{4}\right)}{t\,b^2}$$

$$\sigma_b = \frac{P(2D_2+d_1)}{4\,t\,b^2} --- (12)$$

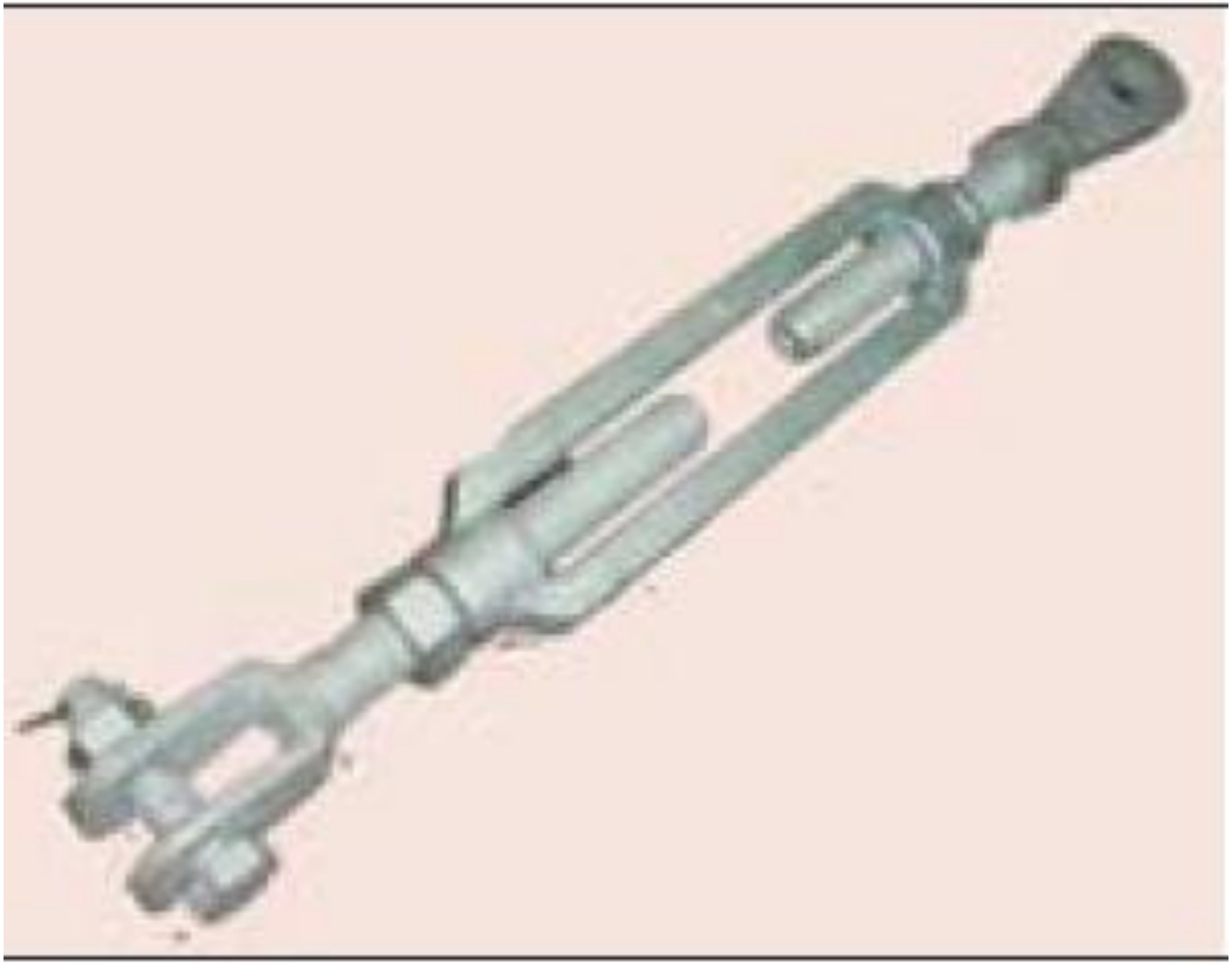
- From equation 12 ‘b’ is obtained.
- The larger value of ‘b’ is taken from equation 11 and 12.
- Length of cotter is taken as L=4d

Problems

Example 12.1. *Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically.*

Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress = 90 MPa.

Design of Turn Buckle



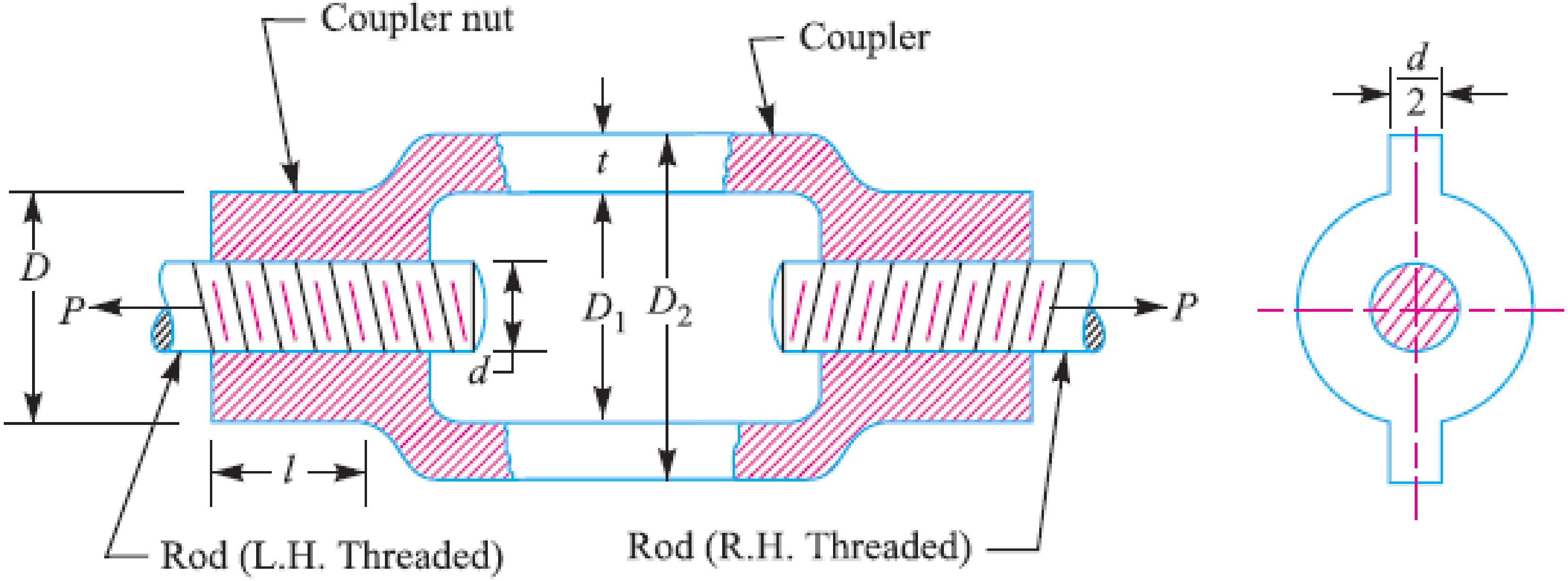
Introduction

- Turn buckle or coupler is a mechanical joint which is used to connect two members which are subjected to tensile loading which require slight adjustment of length or tension under loaded conditions.
- It consists of central hexagonal nut called coupler and tie rod having right hand and left hand threads.
- A coupler of hexagonal shape is to facilitate the turning of it with a spanner or sometimes a hole is provide in the nut so that Tommy bar can be inserted for rotating it.

- As coupler rotate, the tie rods are either pulled together or pushed apart depending upon the direction of the rotation.
- **Applications –**

1) To tighten the members of the roof truss.
2) Used to connect link in a mechanism to transfer motion.
3) Used between the two railway wagons or boggies.
4) To tighten the cable or stay ropes of electric distribution poles.

Design of Turn Buckle



- Design load = P_d = 1.25P

1) Diameter of rods (d_c) –

Considering tensile failure of rods.

$$\sigma_t = \frac{P_d}{\frac{\pi}{4}(d_c)^2} --- (1)$$

$$d_c = core\ diameter\ of\ the\ threads.$$

$$d_o = \frac{d_c}{0.84} --- (2)$$

$$d_o = outer\ diameter\ of\ threads$$

2) Length of Coupler Nut (l)

- Consider the direct shearing of the thread at the root of coupler nut and the screw.

a) The direct shear stress induced in screw threads.

$$\tau_s = \frac{P}{\pi \times d_c \times l} --(3)$$

b) The direct shear stress induced in nut

$$\tau_n = \frac{P}{\pi \times d_o \times l} --(4)$$

c) By Empirical Relation

$$l = d_o \, to \, 1.25 d_o \rightarrow for\ steel$$

$$l = 1.5 d_o \, to \, 2 d_o \rightarrow for\ C.I\ and\ soft\ maerial$$

d) Check the Threads for crushing

Considering crushing failure of threads

$$\sigma_{cr} = \frac{P_d}{\frac{\pi}{4}(d_o^2 - d_c^2) \times n \times l} - - - (5)$$

$n = l_n/p$

P = pitch of the thread

$From\ this\ equation\ \sigma_{cr}\ is\ obtained$

$$\sigma_{cr} \langle \sigma_{cr(given)}$$

3) Outside Diameter of Coupler Nut

- Considering the tensile failure of coupler nut.

$$\sigma_t = \frac{P_d}{\frac{\pi}{4}(D^2 - d_o^2)} ---(6)$$

$$By\ empirical\ relation$$

$$D = 1.25\,d_o\ to\ 1.5\,d_o ---(7)$$

4) Outside Diameter of Coupler

- Let D_1 = Inside dia. Of coupler = (d_o+6)
- D_2 = Outside dia. Of coupler.
- Considering the tensile failure of coupler.

$$\sigma_t = \frac{P_d}{\frac{\pi}{4}(D_2^2 - D_1^2)} --- (8)$$

By empirical relation

$$D_2 = 1.5\,d_o \; to \; 1.75\,d_o --- (9)$$

$$5)\, Length\, of\ coupler\ nut\,(L_n) = L_n = 6d_o$$

$$6)\, Thickness\, of\ the\, coupler = t = 0.75d_o$$

$$7)\, Thickness\, of\ the\, coupler\ nut = t_1 = 0.5d_o$$

THE END OF THIS DESIGN