2.1.Design of Cotter and Knuckle joint

Q.1) State applications of Cotter Joint,Knuckle joint & turn Buckle

Cotter Joint :

1) It is used in bicycle to connect pedal to sprocket wheel.
2) It is also used to connect piston rod in cross-head.
3) It is used to connect piston rod with its extension.

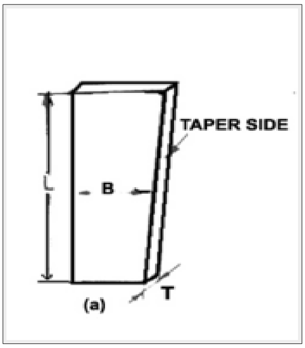
Knuckle Joint:

1) It is used in air braking arrangement in locomotive and track shifting mechanism of railway.
2) It is used in structures for suspension link.
3) It is used in coupling trolleys with tractors.
4) It is used in links of bicycle chain

Turn Buckle:

1) It is used in tie rod in crane
2) It is used in electric poles to support it.
3) It is used in telegraph wires
4) it is used in automobile chassis.
5) It is used in railway track changing mechanism.

Q.2) What is a Cotter ? Why taper is provided on cotter? How much taper is provided?



Cotter is a flat wedge shaped piece of steel which is used to connect two rods which transmit the force/motion but without rotation. Cotter is fitted in the slot and remains in its position by wedge action.

Taper is provided to,

i) It is easy to remove the cotter & dismantle the joint
ii) It ensures tightness of the joint in operation & prevents loosening of the parts.

Value of taper on cotter is 1 in 48 to 1 in 24.

Q.3) Differentiate between Key and Cotter

The main difference between keys and cotters are as follows:

(a) Keys are driven parallel to the axis whereas cotters are driven perpendicular to the axis.

(b) Keys are used in parts subjected to torque whereas cotters are used in parts subjected to tensile or compressive force.

(c) Keys resist shear over a longitudinal section whereas cotters resist shear over two transverse sections.

Q.4) Differentiate between Knuckle joint and cotter joint (Four points)

Ans :

Sr.No	Knuckle Joint	Cotter Joint
1	Can take only tensile load	Can take tensile & compressive load
2	Can permit angular movement between rods	Cannot permit angular movement
3	Subjected to bearing failure	Not subjected to bearing failure
4	No taper or clearance provided	Taper or clearance provided
5	Application tie bar links of bicycle chain joint for rail shifting mechanism	Application cotter foundation bolt joining two rods with a pipe joining piston rod with c/s head

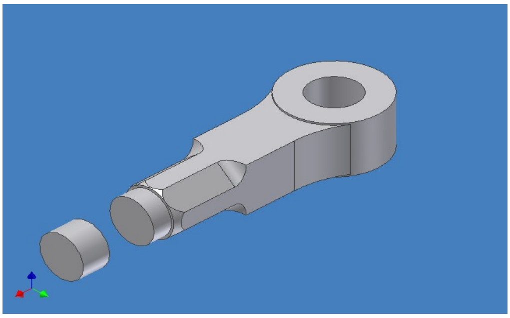
Design Procedure of Knuckle Joint

Notations -

P = Tension in Rod, d = diameter of rod, d1 = diameter of pin, d2 = Outer diameter of eye, d3 = diameter of Pin head, t = thickness of rod end (single eye), t1 = thickness of forked end (double eye), t2 = thickness of Pin head and thickness of collar

Design steps -

Step 1. Design of Rod (d) -



*Tensile failure

P = Area x Stress

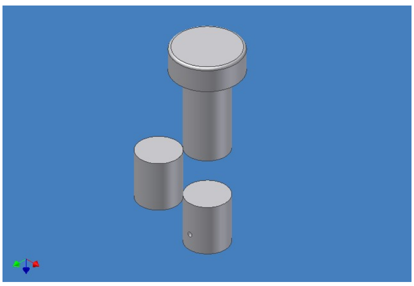
$$P=\frac{\pi}{4}\times d^2\times\sigma_t$$

Find d.

Step 2. Decide other dimensions using empirical relations

Dia. of knuckle pin =	d1 = d
Outer dia of eye =	d2 = 2d
Dia. of knuckle pin head & collar	d3 = 1.5 d
Thickness of single eye or rod end	t = 1.25 d
Thickness of fork	t1 = 0.75 d
Thickness of pin head	t2 = 0.5 d

Step 3. Check stresses in pin



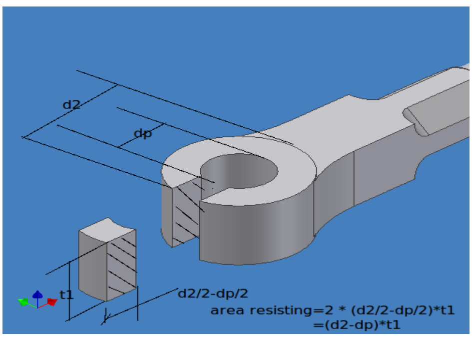
Double shear failure of pin,

$$P=2\times\frac{\pi}{4}\times d_1^2\times\sigma_s$$

Check σ_s

If fs(induced)<fs (allowable)................Design is safe

Step 4. Check stress in single eye

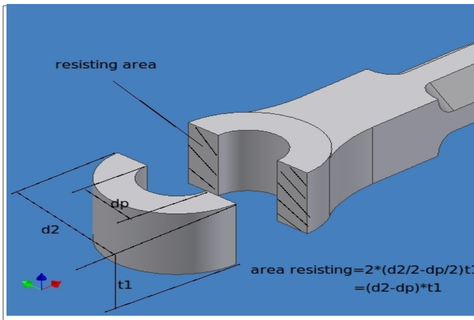


(1) Shear failure

$$P=(d_2-d_1)\times t\times\sigma_s$$

Check σ_s

If σ_s (induced)< σ_s (allowable).....................Design is safe



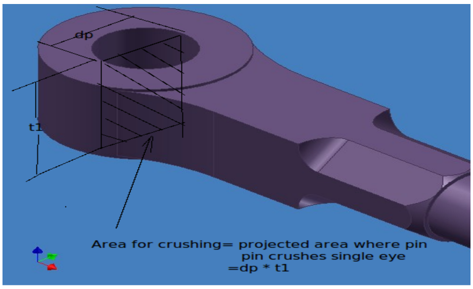
(2) Tensile failure

P = A x Stress

$$P=(d_2-d_1)\times t\times \sigma_t$$

Check $\boldsymbol{\sigma_t}$

If σ_t (induced)< σ_t (allowable)..................Design is safe



(3) Crushing failure -

$$P=(d_1\times t)\times \sigma_c$$

Check $\boldsymbol{\sigma_c}$

If σ_c (induced)< σ_c (allowable)................Design is safe

Step 5.Check stresses in forked end or double eye

(1) Shear failure

$$P=2\times(d_2-d_1)\times t_1\times \sigma_s$$

Check $\boldsymbol{\sigma_s}$

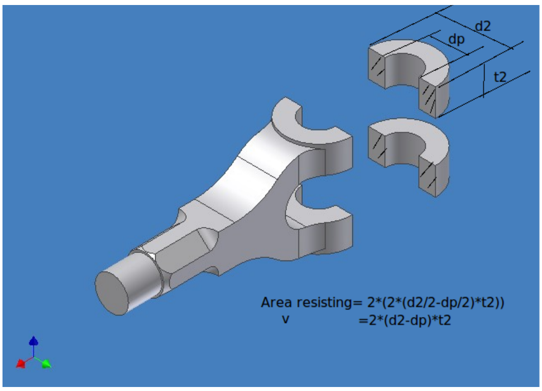
If σ_s (induced)< σ_s (allowable).....................Design is safe

(2) Tensile failure -

$$P=2\times(d_2-d_1)\times t_1\times \sigma_t$$

Check $\boldsymbol{\sigma_t}$

If σ_t (induced)< σ_t (allowable)..................Design is safe



(3) Crushing failure -

$$P=2\times(d_1\times t_1)\times \sigma_c$$

check $\boldsymbol{\sigma_c}$

If σ_c (induced)< σ_c (allowable)................Design is safe

Steps In short :

Part	Failure	Equation	To find
1) Rod	Tensile	$P=\frac{\pi}{4}\times d^2\times\sigma_t$	d=
2) Decide other dimensions	Emperical relations	d1 = d d2 = 2d. d3 = 1.5 d t = 1.25 d t1 = 0.75 d t2 = 0.5 d	d1,d2,d3,t,t1,t2
3) Check Pin	Double shear	$P=2\times\frac{\pi}{4}\times d_1^2\times\sigma_s$	σ_s check
4) Check single eye (Rod end)	Shear	$P=(d_2-d_1)\times t\times\sigma_s$	σ_s check
	Tensile	$P=(d_2-d_1)\times t\times\sigma_t$	σ_t check
	Crushing	$P=(d_1\times t)\times\sigma_c$	σ_c check
5) Check double eye (Fork end)	double Shear	$P=2\times(d_2-d_1)\times t_1\times$	check σ_s
	tensile	$P=(d_2-d_1)\times t_1\times\sigma_t$ 2(d2-d1)	check σ_t
	Crushing	$P=2\times(d_1\times t_1)\times\sigma_c$	check σ_c

Numerical Problems

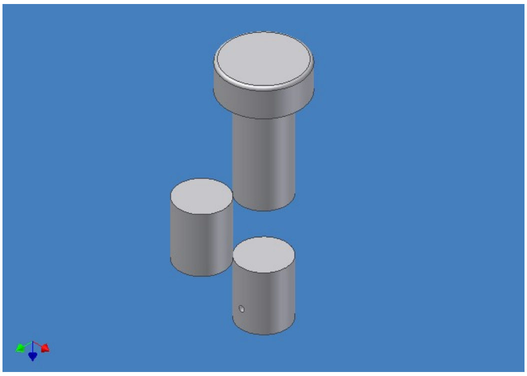
1.Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.

Example 2. 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.

3.Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable
stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.

Que. Explain with sketch what are the modes of Knuckle pin failure. Or Explain the bending failure of Knuckle Pin

1. Failures Of Knuckle Pin In Double Shear



Cross section area of pin

$$A=\frac{\pi}{4}\times d1^2$$

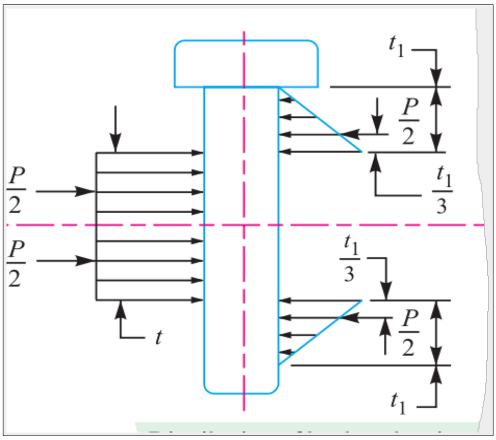
Where , d1 diameter of knuckle pin and the shear strength

$$=\frac{2\times\pi}{4}\times d1^2\times T$$

Equating this to the load acting on the rod, we have

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

2. Failures Of Knuckle Pin In Bending



In case, the stress due to bending is taken into account

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

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

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

and sections of modules

$$Z=\frac{\pi}{32}\times d_1^3$$

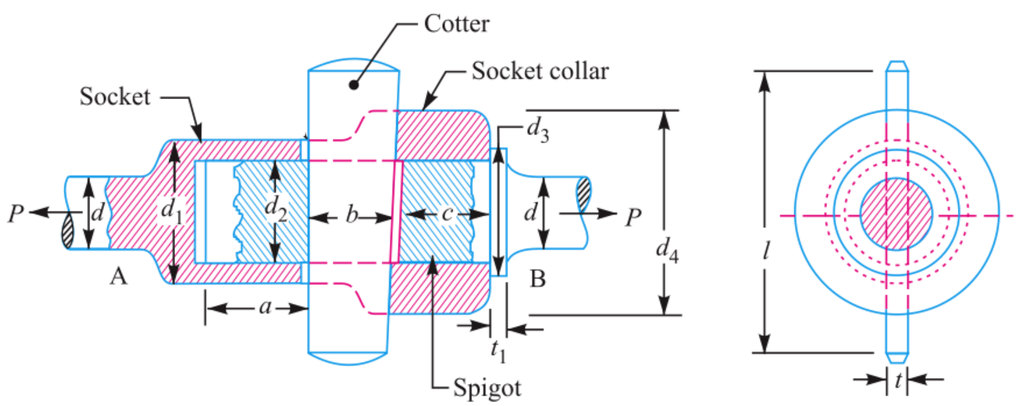
Maximum bending (tensile) stress

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

$$\sigma_b=\frac{\frac{p}{2}\left(\frac{t_1}{3}-\frac{t}{4}\right)}{\frac{\pi}{32}\times d_1}$$

Using this relation the induced bending stress in the pin can be checked and it should be less than the allowable bending stress.

Design Procedure of Cotter Joint



Notations -

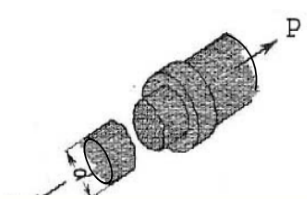
d= Diameter of the rod,d1= outer diameter of socket,d2= Diameter of spigot or inside diameter of socket
d3= Outside diameter of spigot collar ,d4= Diameter of socket collar,t1= Thickness of spigot collar
a = Distance from the end of the slot to end of spigot,c= thickness of socket collar
b,t,l= width , thickness and length of cotter.

Parts of cotter Joint

1. Cotter {b,t,l}
2. Spigot {d2,d3,t1,a)
3. Socket {d1,d4,c}

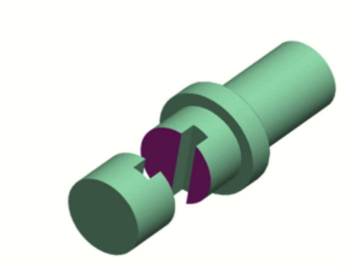
Steps in Design of Cotter Joint

1.Failure of the rods in tension



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

From this equation diameter of the rods (d) may be determined.

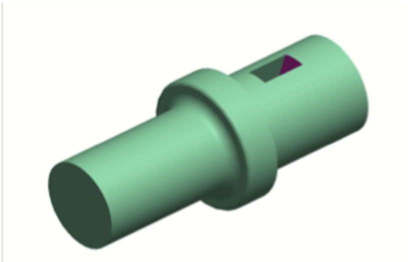


2.Failure of spigot in tension (across the weakest section (or slot))

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

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

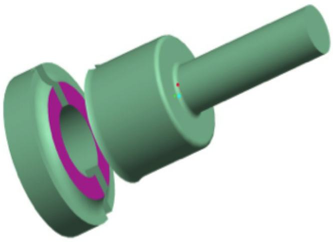
3.Failure of the cotter in crushing in spigot

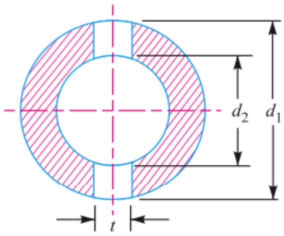


$$P=d_2\times t\times\sigma_c$$

From this equation, the induced crushing stress may be checked.

4. Failure of the socket in tension across the slot



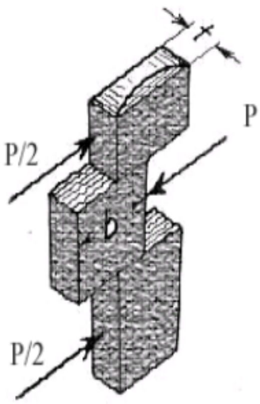


$$P=\left\{\frac{\pi}{4}[(d_1)^2-(d_2)^2]-(d_1-d_2)t\right\}\sigma_t$$

From this equation, outside diameter of socket (d_1) may be determined.

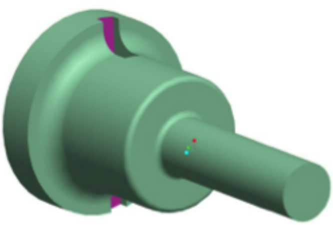
5.Failure of cotter in shear

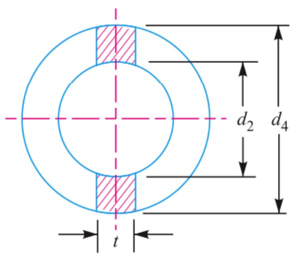
Double shear



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

From this equation, width of cotter (b) is determined.

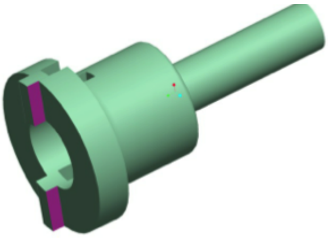




6.Failure of the socket collar in crushing.

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

From this equation, the diameter of socket collar (d_4) may be obtained.

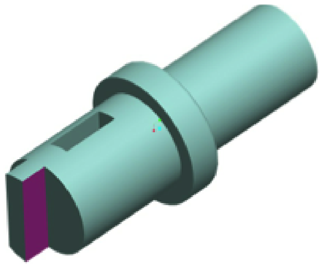


7.Failure of socket end in shearing {double shear}

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

From this equation, the thickness of socket collar (c) may be obtained.

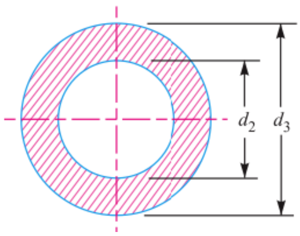
8.Failure of rod end(spigot) in shear {Double shear}

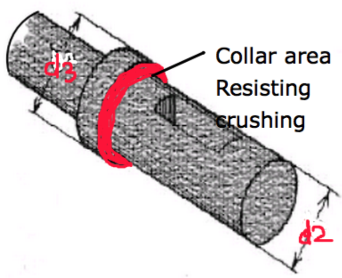


$P=2\,(a\times d_2)\times\sigma_s$

From this equation, the distance from the end of the slot to the end of the rod (a) may be obtained.

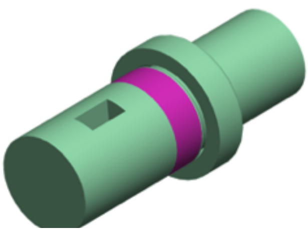
9.Failure of spigot collar in crushing





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

From this equation, the diameter of the spigot collar (d_3) may be obtained.



10.Failure of the spigot collar in shearing

$P=\pi d_2\times t_1\times\sigma_s$

From this equation, the thickness of spigot collar (t_1) *may be obtained.*

11.The length of cotter (l)

Emperical relation

$l=4d$

STEPS IN SHORT

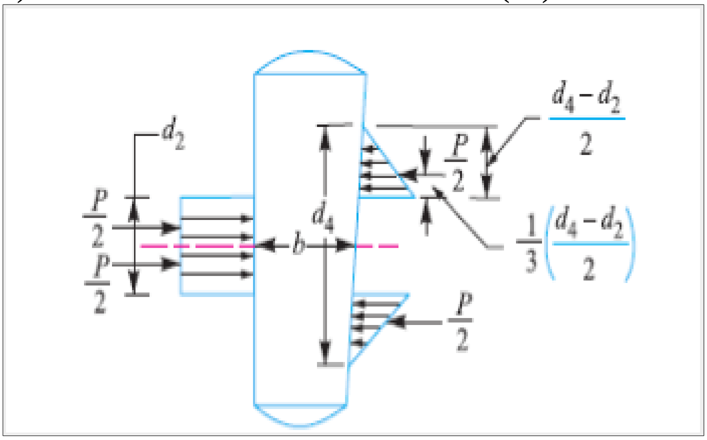
No	Failure	Equation	To find
1	Tensile Failure of rod	$P=\frac{\pi}{4}\times d^2\times\sigma_t$	$d=$
2	Tensile failure of spigot	$P=[\frac{\pi}{4}(d_2)^2-d_2\times t]\sigma_t$ take t=d2/4	d_2 =
3	Crushing of cotter in spigot	$P=d_2\times t\times\sigma_c$	σ_c check
4	Tensile failure of socket	$P=\left\{\frac{\pi}{4}[(d_1)^2-(d_2)^2]-(d_1-d_2)t\right\}\sigma_t$	$d_1=$
5	Shear Failure of cotter	$P=2b\times t\times\sigma_s$	b=
6	Crushing of Socket collar against cotter	$P=(d_4-d_2)t\times\sigma_c$	d4=
7	Shear of Socket end	$P=2(d_4-d_2)c\times\sigma_s$	c=
8	Shear of spigot end	$P=2(a\times d_2)\times\sigma_s$	a=
9	Crushing of spigot collar	$P=\frac{\pi}{4}[(d_3)^2-(d_2)^2]\sigma_c$	d3=
10	Shearing of spigot collar	$P=\pi d_2\times t_1\times\sigma_s$	t1=
11	Empirical length of cotter	$l=4d$	$l=$

Numerical Problems

Problem 1): Design and draw a cotter joint to support a load varying from 30kN in compression to 30kN 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 = 50MPa; shear stress = 35MPa and crushing stress = 190MPa.

Q. Explain with sketch bending Failure of COTTER PIN

It is assumed that the load is uniformly distributed over the various cross- sections of the joint. But in 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 rod end is uniformly distributed while in the socket end it varies from zero at the outer diameter (d4) and maximum at the inner diameter (d2) as shown in Fig



The maximum bending moment occurs at the center of the cotter and is given by

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

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

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

We know the sections modules of the cotter

$$z = t\,X\,\frac{b^2}{6}$$

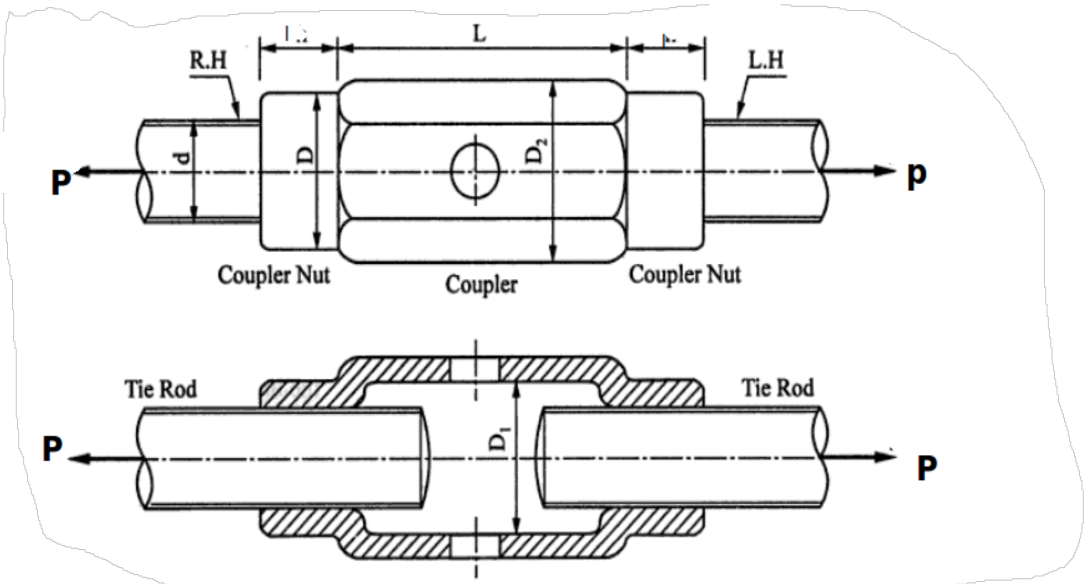
Bending stress induced in the cotter

$$6b = \frac{M\,max}{Z} = \frac{\frac{P}{2}\left(\frac{(d4 - d2)}{6} + \frac{d2}{4}\right)}{t\,X\,\frac{b^2}{6}}$$

This bending stress induced in the cotter should be less than the allowable bending stress of the cotter.

2.2.Design of Turn Buckle

Design Procedure of Turn Buckle



Turn Buckle

Notations -

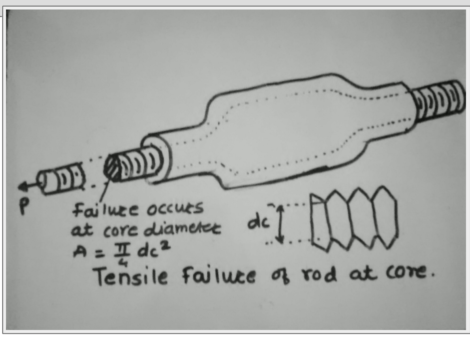
dc = Core diameter of rod, d = Outer diameter of rod, D = Outer diameter of coupler at end.

D1 = Inner diameter of coupler at centre, D2 = Outer diameter of coupler at centre. l = Length of threaded portion, L = Total length of coupler.

Parts of Turn Buckle

1.Tie rod (dc,d) 2. Coupler Nut (D,l) 3.Coupler (D1,D2,L)

Design steps



Step 1. Decide the design Load (Pd)

For threaded portion the design load is taken 30% more than the given load to account for friction in threads

$$Pd = 1.3 \times P$$

Step2. Design of tie rod (dc,d) -

Tensile failure of rod at threads

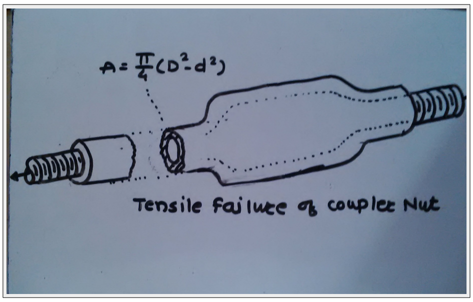
Pd = Area x Stress

$$Pd = \frac{\Pi}{4} dc^2 \times \sigma_t$$ Find dc.

Outer diameter of threaded portion

d=1.15 * dc

Step 3. Design of coupler nut (l,D)



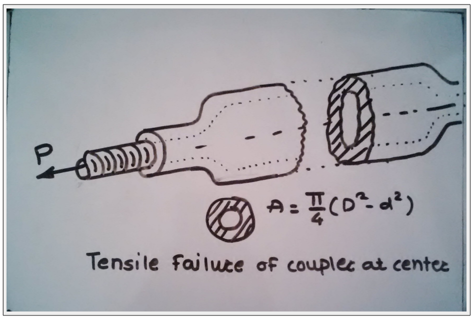
Tensile failure of Coupler nut

$$pd = \frac{\pi}{4}(D^2 - d^2) * \sigma_t$$ Find D

Shear failure of threads

$$Pd = (\pi . d_c . l) \sigma_s$$ Find l

Step 4. Design of coupler (D1,D2,L)



Empirical relation

$$D1 = d + 6mm$$

Tensile failure of couple

$$pd = \frac{\pi}{4}(D2^2 - D1^2) * \sigma_t$$ Find D2

Empirical relation L=6d

Steps in Short

Part	Failure	Equation	To find
Design load	Empirical	Pd=1.3 × P	pd
Design of rod (dc,d)	Tensile Empirical	$Pd = \frac{\Pi}{4} dc^2 \times \sigma_t$ d=1.15 * dc	dc d
Design of Coupler nut (D,l)	Tensile Shear	$pd = \frac{\pi}{4}(D^2 - d^2) * \sigma_t$ $Pd = (\pi . d_c . l)\sigma_s$	D l
Design of Coupler (D1,D2,L)	Empirical Tensile Empirical	$D1 = d + 6mm$ $pd = \frac{\pi}{4}(D2^2 - D1^2) * \sigma_t$ L=6d	D1= D2= L=

SOLVED Numerical Problems

1. The pull in the tie rod of an iron roof truss is 50 kN. Design a suitable adjustable screwed joint. The permissible stresses are 75 MPa in tension, 37.5 MPa in shear and 90 MPa in crushing.

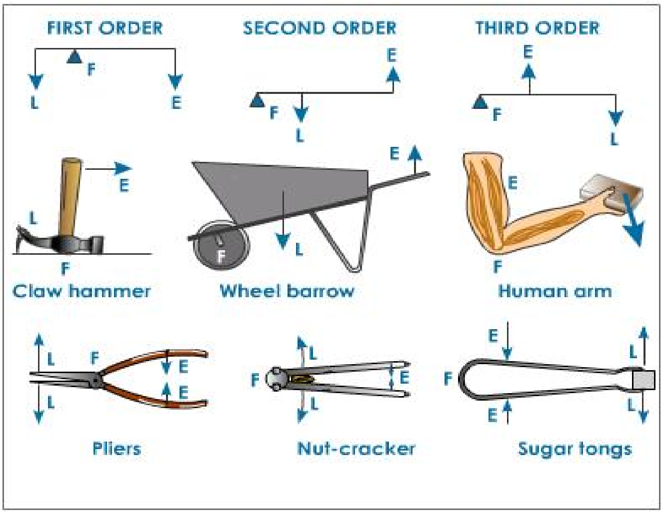
2. The pull in the tie rod of a roof truss is 44 kN. Design a suitable adjustable screw joint. The permissible tensile and shear stresses are 75 MPa and 37.5 MPa respectively.
[Ans. d = 36 mm ; l = 15.58 mm ; D = 48mm ; D1=42,D2 = 53 mm]

3] The maximum pull in the tie rod of turn buckle is 4.5 kN. The tie rod made up of steel 40 C 8 (syt = 380 N/mm2) design the adjustable joint taking F.O.S. = 5 & allowable shear stress 0.4 times allowable tensile stress.

2.3 Design of Levers , Hand/Foot Lever

Q.5) What is a lever ? what are the different types of levers?

A lever is one of the simplest mechanical devices. A lever consists of a beam or stick or rod. However, a lever by itself is not effective. It must have something on which to pivot. This pivot is called a fulcrum. A lever helps to lift weights with less effort.



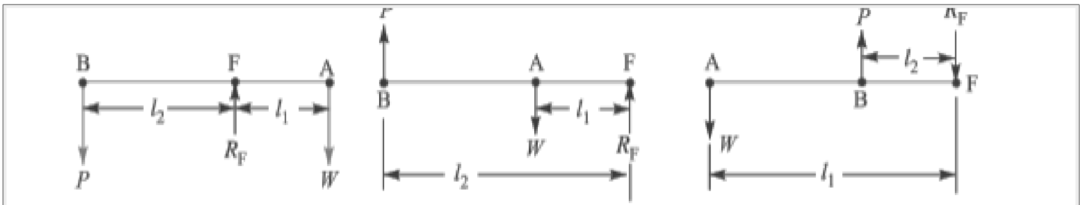
First-class levers have the fulcrum placed between the load and the effort, as in the seesaw, claw hammer and scissors are examples of this type.

Second-class levers have the load between the effort and the fulcrum. A wheelbarrow is a second-class lever. The wheel's axle is the fulcrum, the handles take the effort, and the load is placed between them. The effort always travels a greater distance and is less than the load.

Third-class levers have the effort placed between the load and the fulcrum. The effort always travels a shorter distance and must be greater than the load. A example of a third-class lever is the human forearm: the fulcrum is the elbow, the effort is applied by the biceps muscle, and the load is in the hand.

Q.6) State the three different ways of application of levers in engineering practice.

Ans:



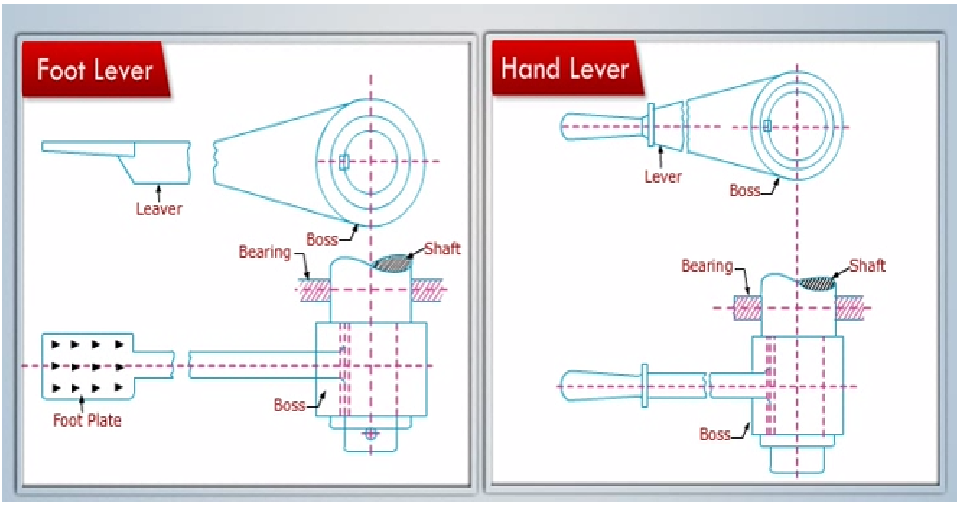
(*a*) First type of lever. (*b*) Second type of lever. (*c*) Third type of lever.

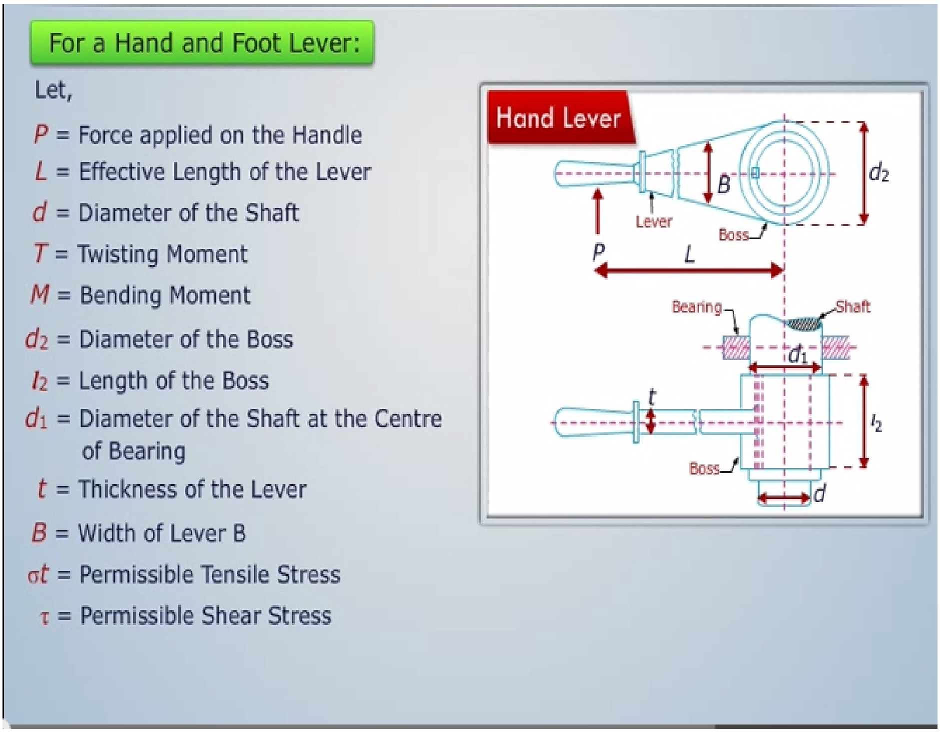
First type of levers, the fulcrum is in between the load and effort. In this case, the effort arm is greater than load arm, therefore mechanical advantage obtained is more than one. Such type of levers are commonly found in bell cranked levers used in railway signaling arrangement, rocker arm in internal combustion engines, handle of a hand pump, hand wheel of a punching press, beam of a balance, foot lever etc.

Second type of levers, the load is in between the fulcrum and effort. In this case, the effort arm is more than load arm, therefore the mechanical advantage is more than one. The application of such type of levers is found in levers of loaded safety valves.

Third type of levers, the effort is in between the fulcrum and load. Since the effort arm, in this case, is less than the load arm, therefore the mechanical advantage is less than one. The use of such type of levers is not recommended in engineering practice. However a pair of tongs, the treadle of a sewing machine etc. are examples of this type of lever.

Design of Hand Lever and Foot Lever





l= Effective overhang of the lever from nearest bearing

Step 1 :Design Of shaft and Boss :(d,d1,d2,l)

1. Diameter of shaft d {subjected to torque only}

Torque Acting on Shaft $T = P \times L$

Diameter of Shaft obtained on basis of shear stress

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

Find diameter of shaft d using this equation

2) Diameter of shaft at centre of bearing d1 {Subjected to torque and Bending moment}

Bending Moment acting on shaft

$$T=P\times l$$

Equivalent torque acting on shaft

$$T=\sqrt{T^2+M^2}$$

Diameter of Shaft at centre of bearing on the basis of shear strength

$$T_e=\frac{\pi}{16}\times\tau\times d_1^3$$

Find diameter of shaft d1 using this equation

3) Dimensions of Boss (Which holds the shaft)

$$d_2=1.6\,d \quad , \quad l_2=1.5\,d$$

Step 2 : Design Of Key :(t_k, l_k, w_k)

- Using a square Key

$$t_k=d/4 \quad , w_k=d/4 \quad ,$$

Shear Failure of key

$$\frac{Torque}{Radius}=area\times stress \quad , \quad \frac{T}{d/2}=w_k\times l_k\times\tau$$

Find lk using eqn. above

Step 3 : Design Of cross-section of lever (B,t)

- Using bending equation ,

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

Using this equation find B and t of lever.{Assume B=3t or any other relation }

- The lever is subjected to bending moment,

$$M=P\times(L-\frac{d_2}{2})$$

- Moment of inertia

$$I=\frac{t.B^3}{12}$$

- $\sigma_b=$ Allowable stress for lever material
- Distance of extreme fiber form N-A

$$y=\frac{B}{2}$$

Steps in Short

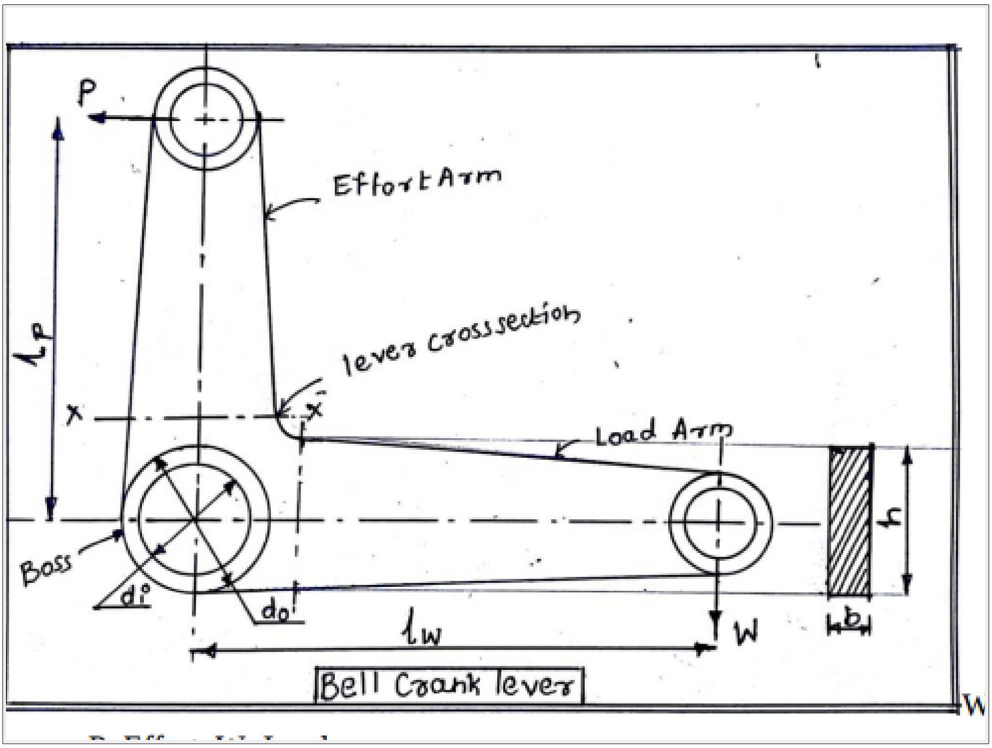
Part	Failure	Equation	To find
Shaft {d,d1}	Torsional shear stress Torsional shear stress	T= P× L $T = \frac{\pi}{16} \times \sigma_s \times d^3$ M = P × l $T_e = \sqrt{T^2 + M^2}$ $T_e = \frac{\pi}{16} \times \sigma_s \times d_1^3$	d= d1=
Boss {d2,l2}	Emperical relations	$d_2 = 1.6\,d$ $l_2 = 1.25\,d$	d2= l2=
Key {Wk,tk,lk}	Emperical relation {sq key} Shear Failure of Key	$t_k = d/4 \quad w_k = d/4$ $\frac{T}{d/2} = w_k \times l_k \times \sigma_s$	$t_k =$ $w_k =$ $l_k =$
Lever Cross section {t,B}		$M = P \times (L - \frac{d_2}{2})$ $I = \frac{t.B^3}{12} \quad y = \frac{B}{2}$ $\frac{M}{I} = \frac{\sigma_b}{y}$	t= B=

Problems:

1. ***A hand lever has a length of 1100 mm from the centre of shaft to the point of application of 800N load. The effective overhang from the nearest bearing is 150 mm. If the allowable tensile stress and allowable shear are 73MPa and 60MPa respectively,Find i) diameter of shaft: and ii) section of lever, if B/t = 2***

2. ***The effective length of hand lever is 1 meter. The effective overhang from the nearest bearing is 150mm. The lever and shaft are made of alloy steel for which tensile yield strength is 460N/mn^2 . If the maximum force exented at the handle is 300N, design the lever and the shaft with a safety with a safety factor of 4.***
3. ***A hand lever for a brake is 0.8 m long from the centre of gravity of the spindle to the point of application of the pull of 300 N. The effective overhang from the nearest bearing is 100 mm. If the permissible stress in tension, shear and crushing is not to exceed 66 MPa, design the spindle, key and lever. Assume the arm of the lever to be rectangular having width twice of its thickness.***

2.4 Design Of bell crank lever



P= Effort, W=load, lw= length of load arm, lp=length of effort arm,

Rf=Fulcrum reaction, d= diameter of pin,lp=length of fulcrum pin =1.25.db

lb= length of boss 1.25 d, Pb=bearing pressure, do=Outer idameter of boss,

di= Inside diameter of boss

Consider a brass bush of 3 mm thickness is to fit into the boss

di=d+2*2

di=d+6 mm

M= Bending moment, b= width of lever crosssection ,h= depth of lever cross section

Design procedure

1] $\begin{array}{l} Detrmination\, of\ effort\,(P) \\ W \times L_w = P \times l_p \end{array}$

2] Determine of fulcrum reaction (RF)

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

3] Design of fulcrum pin

$lp = lb = 1.25\,d$

a) $Considering\,bearing\;pressure\;of\;futerum\;ping$

$$Pb = \frac{R_F}{rb \times d}$$

b) $Considering\,double\,shear\;failure\;of\;\;pin$

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

Check the shear stress induced in pin with above formula

4] Design of boss of lever

$d_i = d + 6\ldots.\;Emperical\,relation$

$Considering\,bending\;stress\,acting\;on\,the\,boss$

$$6\,b = \frac{M\,y}{I_{xx}}$$

$$M = lp \times p\,,\;y = \frac{do}{2}$$

$$I_{xx} = \frac{1}{12} \times lb \times [d_0^3 - d_1^3]$$

5] Design of lever-arm cross section near to boss

$Considering\,bending\,failure$

$$6\,b = \frac{M \times y}{I_{xx}}$$

$$M = P \times [l_p - \frac{do}{2}]$$

$$y = \frac{h}{2}$$

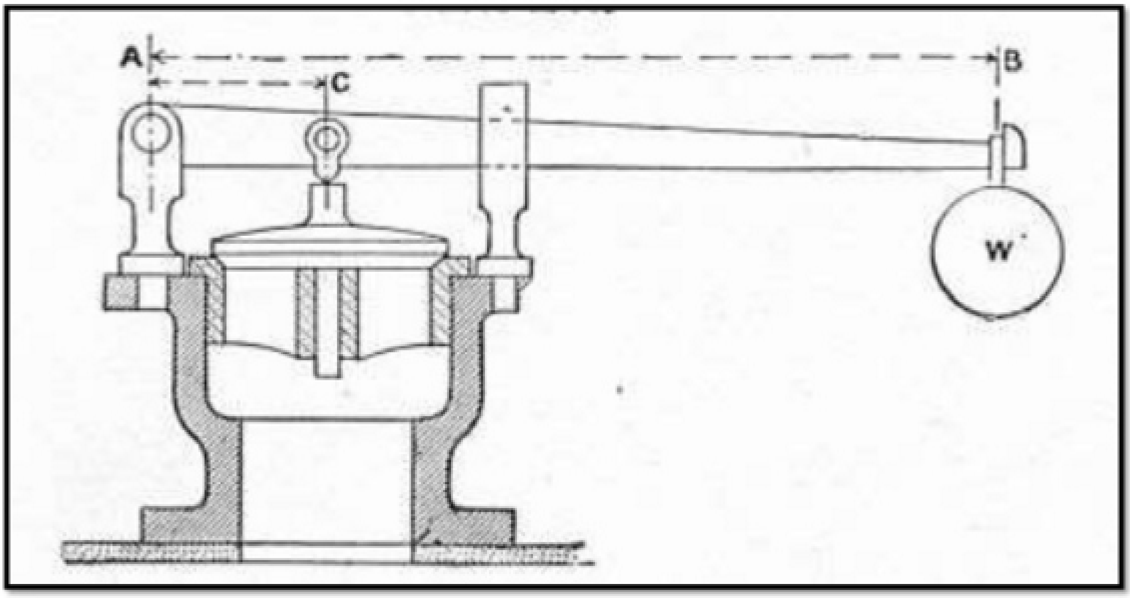
$$I_{xx} = \frac{bh^3}{12}$$

Find b and h using the bending equation

Numerical Problems

1) A bell crank lever having one am 500 mm and another aram 150 mm is sused to tilt a load of 5 kN. The permissible stresses for lever and pin materials in shear and tension are 60 Mpa and 80 Mpa respectively. The bearing pressure on the pin is limited to 10 Mpa, Determine 1) Diameter of Pin 2) Diameter of boss of lever and 3) Dimensions of the rectangular crosssection of lever

2.5 Design of Lever for safety valve



➢ Procedure

- Weight due to steam pressure

  $Ps = \frac{W}{A}$

- Determination of effort

$P \times L_p = W \times L_w$

- Determination of pin at valve spindle

$P_b = \frac{W}{l_b * d}$

Shear stress = $\frac{W}{2A}$

4. Determination of boss of lever

Bending stress $= \frac{6 * p * (lp - lw - \frac{do}{2})}{3 * b^3}$

- Inner diameter
  di = do + 6
- Outer diameter
  do = 2d
- Take larger diameter

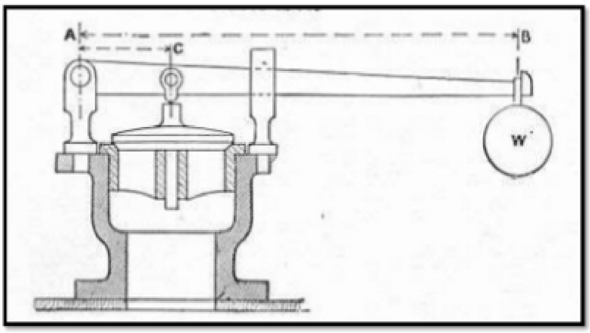
5. Dimensions of lever arm

$-R_f + W - P = 0$

$R_f = W - p$

Bending stress $= \frac{6 * p * (lp - lw - \frac{do}{2})}{3 * b^3}$

EX :- A lever loaded safety valve of 70 mm diameter is required to flow off at a pressure of 1mpa. Design a lever of rectangular cross section (b × 3b) and pin at valve spindle distance between the weight and fulcrum is equal to 880 mm , distance between the fulcrum and valve spindle is 80 mm. Tensile stress is 70 Mpa.



Given data

L_w = 80 Mpa

P_s = 1 Mpa

L_p = 880 Mpa

P_b = 25 Mpa

D_v= 70 mm

Tensile stress = 70 Mpa

Shear stress = 50 Mpa

1. Weight due to steam pressure

$P_s = \frac{W}{A}$

W = Ps × A

W = 3846.5 N

2. Determination of effort

$L_w \times W = L_p \times P$

P = 349.68 N

3. Determination of pin at valve spindle

L_b = 1.25 d = 1.25 × 12 = 15 mm

- Bering pressure

$P_b = \frac{W}{l_b * d}$

d = 11.0945 mm ~ 12 mm

- Double shear stress

Shear stress = $\frac{W}{2A}$

= 17.01 Mpa< 50 Mpa

4. Dimensions of the boss

di = d + 6 = 12+ 6 = 18 mm

M = p (Lp - Lw)

- Shear stress = $\frac{6*p*(lp-lw-\frac{do}{2})}{3*b^3}$

do ^3 - 1598.54 do - 5832 = 0

do = 42 mm

do = 2× d = 2×12 = 24 mm

- Dimensions of the lever arm

- Rf + W - P = 0

Rf = 3496.9 N

- The value of R_f is nearest about W so dimensions is same

- tensile stress = $\frac{p*\left(l_p-lw-\frac{d_0}{2}\right)*18}{27*b*b*b}$

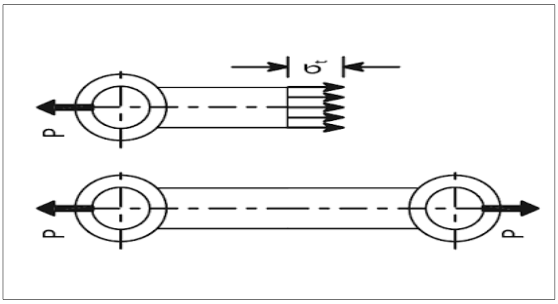
b = 13.72 mm

or b = 14 mm

h = 3×b = 42 mm

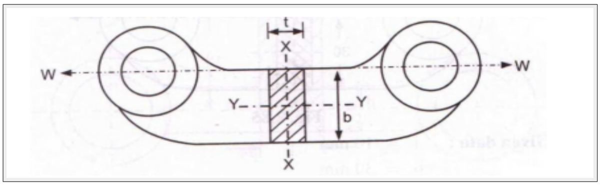
2.6 Design Offset Link, C-Clamp

Axial Link : It is subjected to only Direct tensile or compressive stress



$$\sigma_{max} = \frac{P}{A}$$

OFFSET LINK : It is subjected to both Direct and Bending stresses

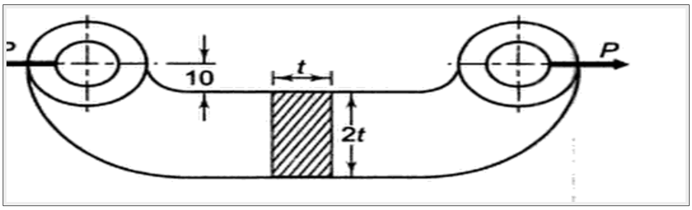


$$\sigma_{max} = \frac{P}{A} + \frac{P.e.y}{I}$$

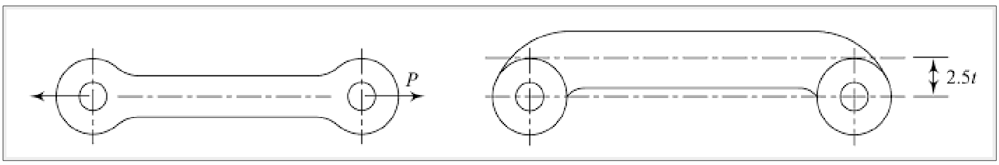
NUMERICAL PROBLEMS

1) Design an offset link for a load of 1000 N. Maximum permissible stress in tension for link material is 60 N/mm 2 . Assume b = 3t for rectangular cross section of the link.

2) An offset link subjected to a force of 25 kN as shown in figure is made up of grey cast iron FG300 and factor of safety is 3. Determine the dimensions of the cross section of the link.



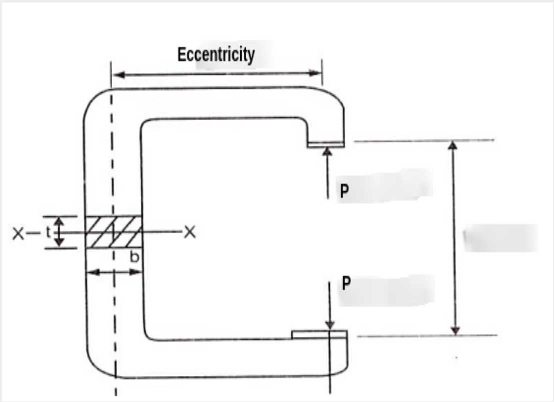
3) A symmetrical link shown in figure carries a tnesile force of 10 kN. It is made of Plain carbon steel 30c8 with syt=350 Mpa. If the b/t ratio is 4 and factor of safety is 4. Find the width and thickness t.



If the shape of the link is modified as shown in figure next to it, Determine the increase in width b and thickness t.

{Increase in b= 25.2, increase in t=6.3}

Design of C-clamp



It is subjected to both Direct and Bending stresses,

Max stress

$$\sigma_{max} = \frac{P}{A} + \frac{P.e.y}{I}$$

The above equation is used to find the cross-sectional area of the c Clamp.

NUMERICAL PROBLEMS

1) Design "C" clamp frame for a total clamping force of 20 kN. The cross-section of the frame is rectangular and width to thickness ratio is 2. The distance between the load line and neutral axis of rectangular cross section is 120 mm and the gap between two faces is 180 mm. The frame is made of cast steel for which maximum permissible tensile stress is 100 N/mm2.

{ans; t=34 mm, b=68 mm}

2) Design "C" clamp frame for a total clamping force of 25 kN. The cross-section of the frame is rectangular and width to thickness ratio is 2. The distance between the load line and neutral axis of rectangular cross section is 130 mm and the gap between two faces is 200 mm. The frame is made of cast steel for which maximum permissible tensile stress is 100 N/mm2.

3) The Spindle of a drilling machine as shown in figure No.1 is subjected to a maximum axial load of 10 kN during operation. Determine the diameter of the solid cast iron column if the permissible tensile stress is 40 N/mm2 . The distance between the axis of the spindle and the axis of the column is 350 mm.

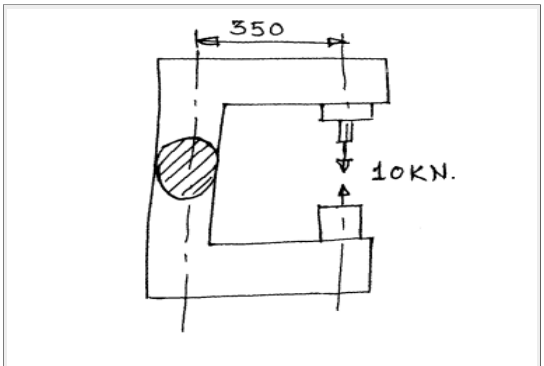


Fig. No. 1

4) Find the dimensions of a c clamp required to withstand a force of 50 KN. Allowable stress for clamp material is 80 MPa in tension. The force acts at an eccentricity of 200mm and width to thickness ratio is 3.