G H Raisoni College of Engineering and Management, Ahmednagar

"Unit 1: Design of Simple Machine Elements"

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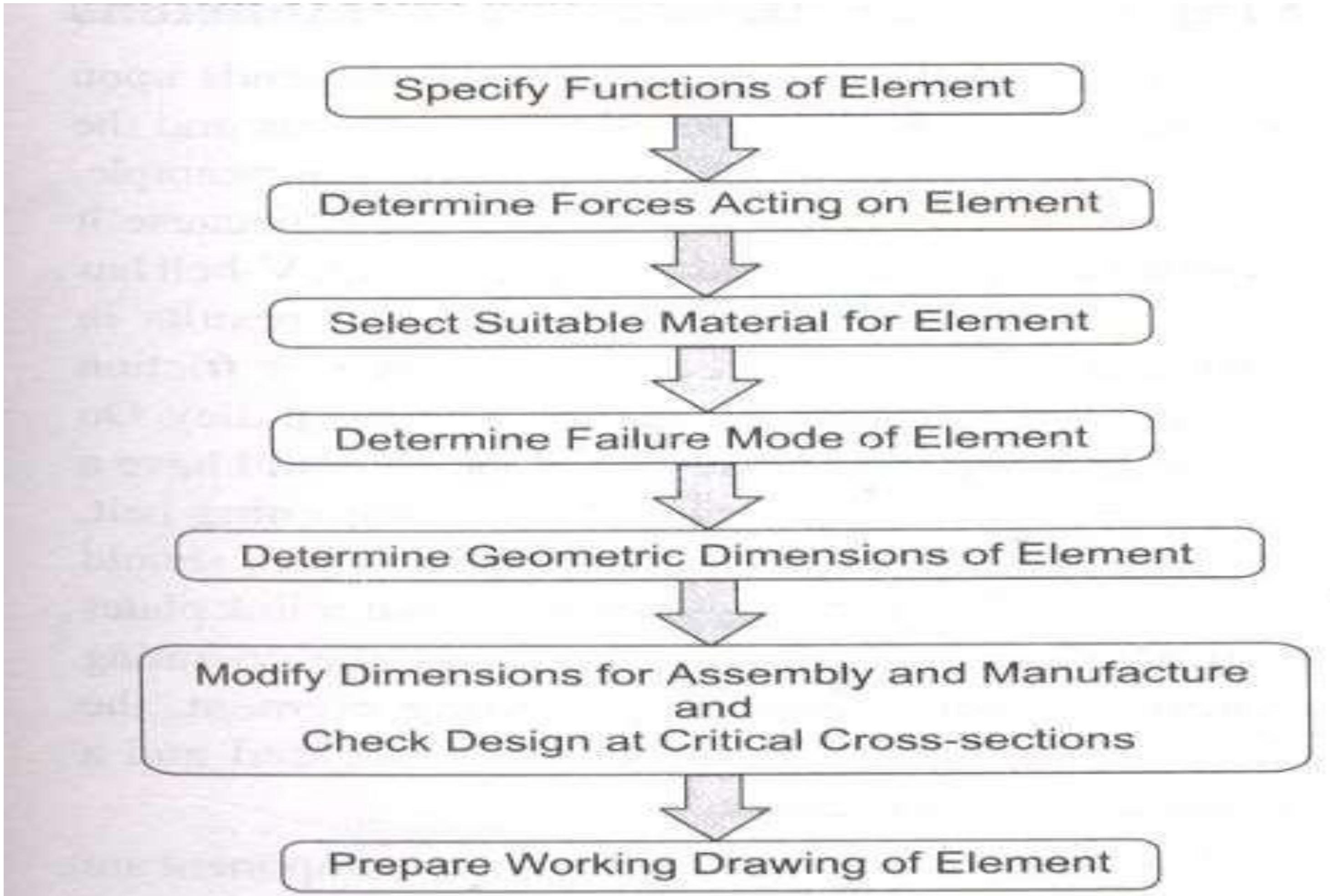
Machine Design

- Machine design is defined as the use of scientific principles, technical information & imagination in the description of a machine or a mechanical system to perform specific functions with maximum economy & efficiency.

- Machine Design is defined as the creation of new design or improving the exist one.

Basic Procedure



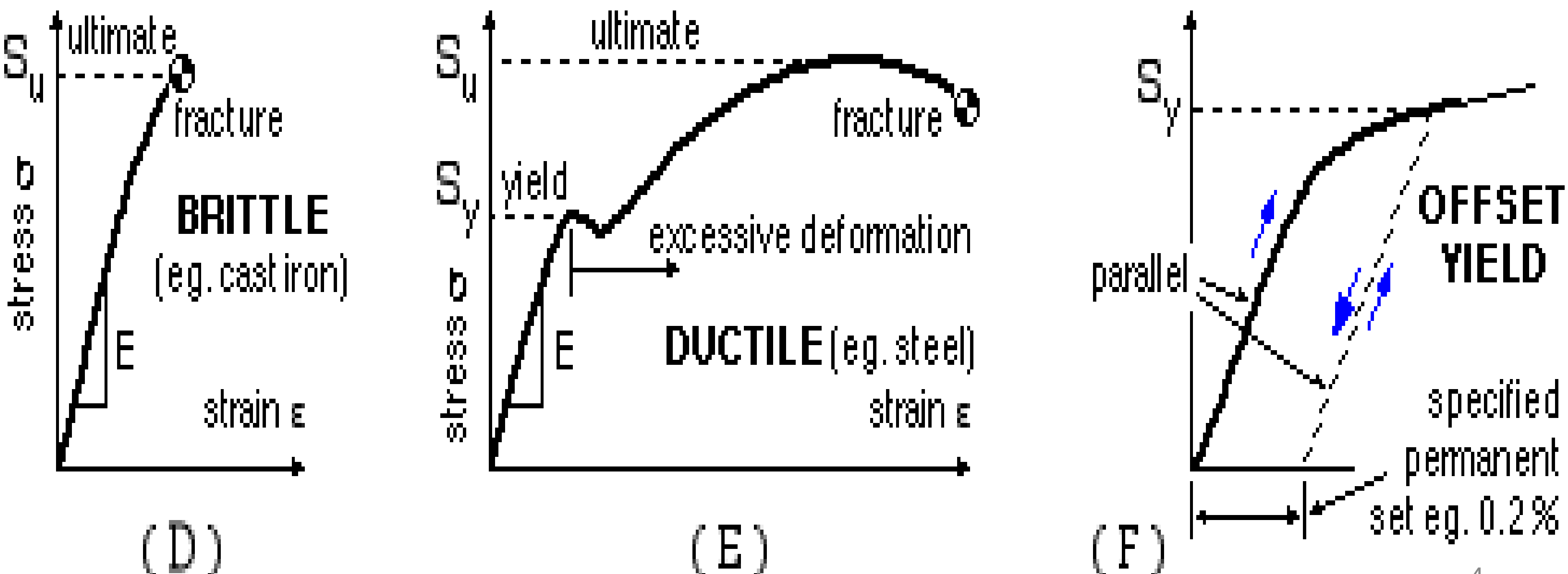
Factor of Safety

$$(fs) = \frac{\text{failure stress}}{\text{allowable stress}}$$

$$(fs) = \frac{\text{failure load}}{\text{working load}}$$

❖ **Ductile Material** $\sigma = \frac{S_{yt}}{(fs)}$

❖ **Brittle Material** $\sigma = \frac{S_{ut}}{(fs)}$



Factor of Safety

- **Effect of Failure:**
  - Low factor of safety = Little inconvenience or loss of time. Eg. failure of the ball bearing in a gearbox.
  - Higher factor of safety = Substantial financial loss or danger to the human life. Eg. Failure of the valve in a pressure vessel.

- **Type of Load:**
  - Low factor of safety = When the external force is static (a load which does not vary in magnitude or direction with respect to time).
  - Higher factor of safety = When the machine element is subjected to impact load.

Factor of Safety

- **Degree of Accuracy in Force Analysis:**
  - Low factor of safety = When the forces acting on the machine component are precisely determined
  - Higher factor of safety = When the machine component is subjected to a force whose magnitude or direction is uncertain and unpredictable.

- **Material of Component:**
  - Low factor of safety = Homogeneous ductile material
  - Higher factor of safety = Brittle material

- **Reliability**:
  - Component In certain applications like continuous process equipment, power stations or defense equipment, high reliability of components is expected.
  - The factor of safety increases with increasing reliability.

Factor of Safety

- **Cost of Component:**
  - As the factor of safety increases, dimensions of the component, material requirement and cost increase.
  - The factor of safety is low for cheap machine parts.

- **Testing of Machine Element:**
  - Low factor of Safety = Tested under actual conditions of service and operation.
  - Higher factor of safety: Not possible to test or deviation between test conditions and actual service conditions.

- **Service Conditions:**
  - Higher factor of safety = Corrosive atmosphere or high temperature environment

- **Quality of Manufacture:**
  - Low factor of Safety = Quality of manufacture is high.
  - Higher factor of safety = Quality of manufacture is poor.

Service Factor

- Overload capacity built into a component, device, engine, motor, etc., as a safety factor.
- It is expressed usually a number greater than one.
- As SF of 1.15 means the item can take 15 percent more load than its rated capacity without breakdown.
- Service Factor = $\frac{Maximun\ Torque\ (Maximum\ Load)}{Average\ Torque\ (Average\ Load)}$
- Ka = $\frac{Tmax}{Tavg}$

Standardization

- Standardization is the obligatory norms to which various characteristics of a product should conform.
- The characteristics include materials, dimensions and shape of the component, method of testing and method of marketing, packing and storing of the product.

Standards Used in Design

1. **Standards for materials, their chemical compositions, Mechanical properties & heat treatment**

   Eg. Indian standard IS 210 specifies seven grades of grey cast iron designated as FG 150, FG 200 (Number indicates ultimate tensile strength in N/mm2).

2. **Standards for Shapes and Dimensions of commonly used Machine Elements**

   Eg. IS 2494 (Part 1) specifies dimensions and shape of the cross-section of endless V-belts for power transmission.

Standards Used in Design

3. Standards for Fits, Tolerances and Surface Finish of Component

Eg. Selection of the type of fi t for different applications is illustrated in IS 2709 on ‘Guide for selection of fits’.

4. Standards for Testing of Products (Codes)

Eg. The method of testing of pressure vessels is explained in IS 2825 on ‘Code for unfired pressure vessels’.

5. Standards for Engineering Drawing of Components

Eg. SP46 prepared by Bureau of Indian Standards on ‘Engineering Drawing Practice for Schools and Colleges’ which covers all standards related to engineering drawing.

Standard and Code

- **Standard:**
  - A set of specifications for parts, materials or processes.
  - The objective of a standard is to reduce the variety and limit the number of items to a reasonable level.
- **Code:**
  - A set of specifications for the analysis, design, manufacture, testing and erection of the product.
  - The purpose of a code is to achieve a specified level of safety.

Types of Standards

- **Company standards:**
  - Set by company or a group of sister concerns.
- **National standards:**
  - Set by national apex body and normally followed throughout the country.
  - Eg. Bureau of Indian Standards (BIS), American Society of Mechanical Engineers (ASME)
- **International standards:**
  - Set by international apex body and normally followed throughout the world.
  - Eg. International Standards Organization (ISO).

Preferred Numbers

- Preferred numbers are used to specify the 'size' of the product.

- The size of product is general term, which includes different parameters like power transmitting capacity, load carrying capacity, speed, and dimensions of the component such as height, length, width and volume of product.

- These parameters expressed numerically, e.g. 5 kw, 10 kw, or 1000 rpm

Preferred Numbers

- French engineer Charles Renard first introduced preferred numbers in the 19 th century.
- The system is based on the use geometric progression to develop a set of numbers.
- R5, R10, R20, R40, and R80 series which increases in steps of 56%, 26%, 12%, 6 % and 3 % respectively.
- Each series has its own series factor as shown below

R5 Series	$\sqrt[5]{10} = 1.58$
R10 Series	$\sqrt[10]{10} = 1.26$
R20 Series	$\sqrt[20]{10} = 1.12$
R40 Series	$\sqrt[40]{10} = 1.06$
R80 Series	$\sqrt[80]{10} = 1.03$

Preferred Numbers

❖ Eg.

Example 1.5.2 **SPPU- May 18, 6 Marks**

A manufacturer is interested to start the business with five different models of machines ranging from 7.5 kW to 75 kW. Specify the power capacities of the five models.

Solution :

Given : $P_{min} = 7.5$ kW, $P_{max} = 75$ kW, $n = 5$.

1. Step ratio

- The range of the power capacity is given by,

$$R = \frac{P_{max}}{P_{min}} = \frac{75}{7.5} = 10$$

- The power capacities are as per geometric step ratio given by,

$$a = [R]^{\frac{1}{(n-1)}} = [10]^{\frac{1}{(5-1)}}$$

$$= [10]^{\frac{1}{4}} \quad \text{i.e.} \quad \text{R4 series}$$

$$\text{or} \quad a = 1.778$$

2. Power capacities

- The power capacities are as follows

$$P_1 = P_{min} = 7.5 \text{ kW}$$

$$P_2 = a P_1 = 1.778 \times 7.5$$

$$= 13.33 \text{ kW or } 13 \text{ kW}$$

$$P_3 = a^2 P_1 = (1.778)^2 \times 7.5$$

$$= 23.71 \text{ kW or } 24 \text{ kW}$$

$$P_4 = a^3 P_1 = (1.778)^3 \times 7.5$$

$$= 42.16 \text{ kW or } 42 \text{ kW}$$

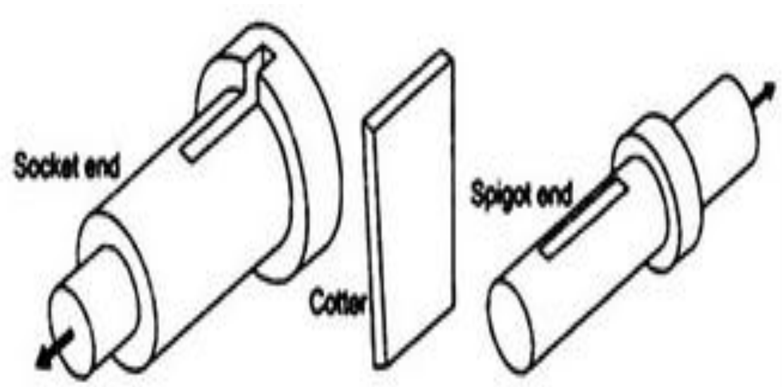
$$P_5 = a^4 P_1 = (1.778)^4 \times 7.5$$

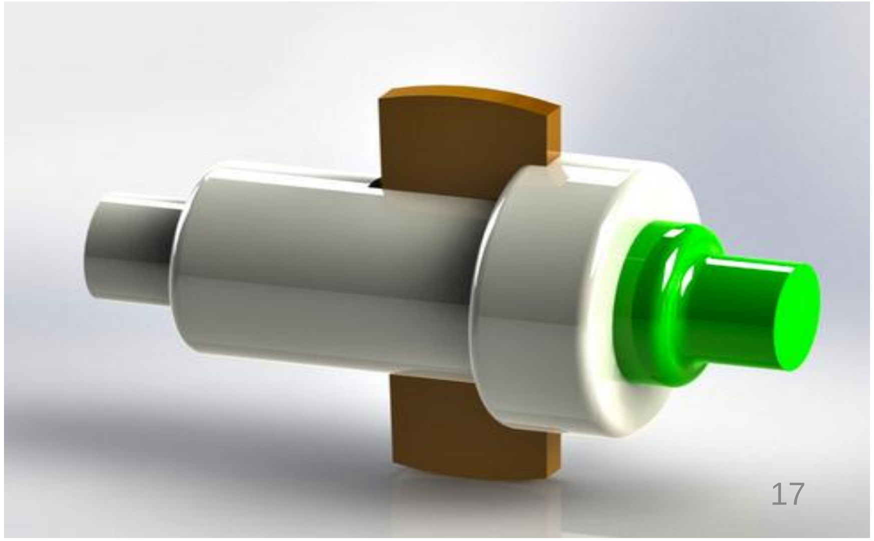
$$= 75 \text{ kW}$$

- **The power capacities of six model are : 7.5 kW, 13 kW, 24 kW, 42 kW and 75 kW.** **...Ans.**

Cotter Joint

- To connect two rods subjected to axial tensile or compressive loads and not suitable to connect rotating shafts which transmit torque.
- Axes of the rods to be joined should be collinear. There is no relative angular movement between rods.
- Cotter joint is widely used to connect the piston rod and crosshead of a steam engine.





Cotter Joint

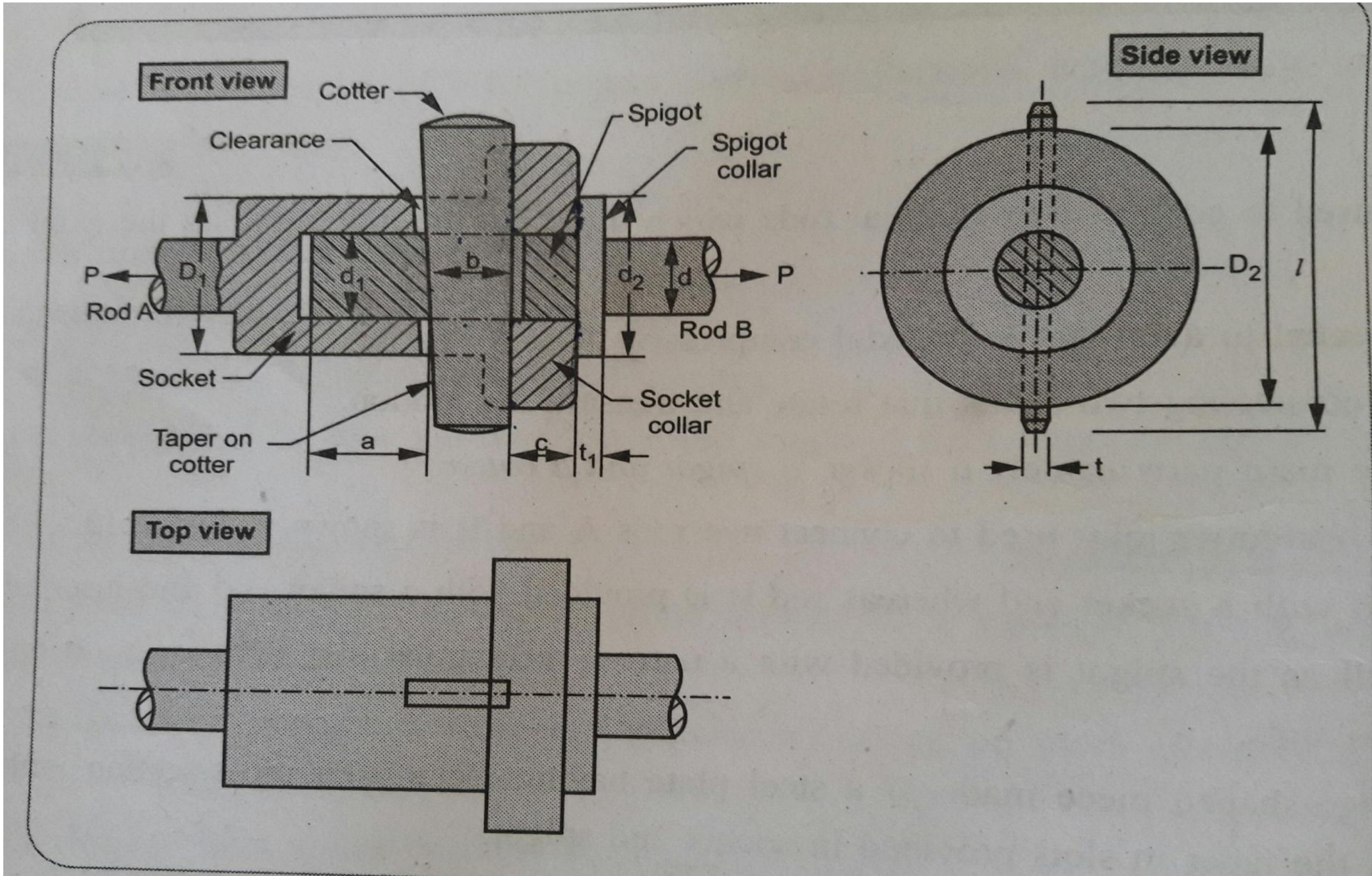


Fig. 1.14 : Cotter joint

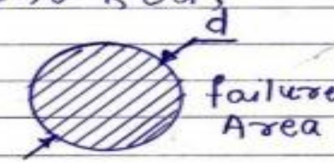
Design Procedure of Cotter Joint

Design Procedure

① Design of dia of rod (d)

Considering tensile failure of the Rod,

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



$\frac{1}{2}$ mark

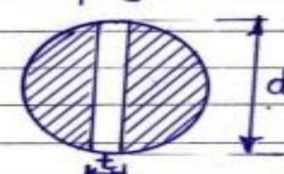
② Design of dia of spigot (d_1) & thickness of cotter (t)

ⓐ By emperical Relation

$$\boxed{t = 0.3d}$$

ⓑ Considering tensile failure of spigot.

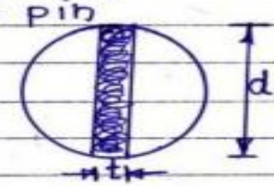
$$\boxed{\sigma_t = \frac{P}{\left[\frac{\pi}{4} d_1^2 - d_1 t\right]}}$$



$\frac{1}{2}$ mark

ⓒ Considering crushing failure of spigot area which is in connection with cotter pin

$$\boxed{\sigma_c = \frac{P}{d_1 t}}$$

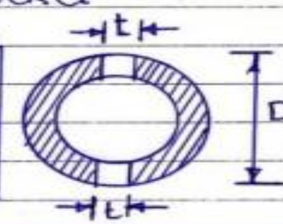


$\frac{1}{2}$ mark

③ Design of outside diameter of socket (D_1)

Considering tensile failure of socket

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

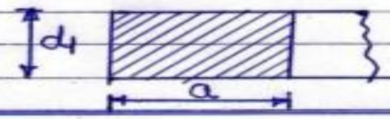


1 mark

Design Procedure of Cotter Joint

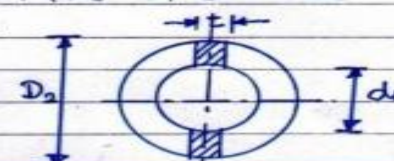
④ Design of distance from end of slot to the end of spigot (a)
Considering double shear failure along the two plane, as shown in fig. — $\frac{1}{2}$ mark

$$\boxed{\tau = \frac{P}{2 d_1 a}}$$



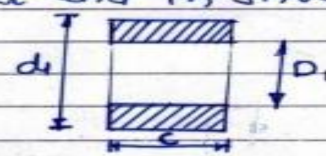
⑤ Design of Dia. of socket collar (D_2)
Considering crushing failure of socket collar as shown in fig. — 1 mark

$$\boxed{\sigma_c = \frac{P}{(D_2 - d_1)\, t}}$$



⑥ Design of thickness of socket collar (c)
Considering failure of socket end in shearing — 1 mark

$$\boxed{\tau = \frac{P}{2[D_2 - d_1]\, c}}$$

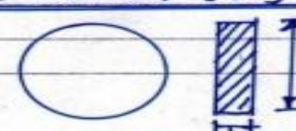


⑦ Design of Dia. of socket collar (d_2)
Considering crushing failure of spigot collar at the contact area between socket collar — 1 mark

$$\boxed{\sigma_c = \frac{P}{\frac{\pi}{4}[d_2^2 - d_1^2]}}$$

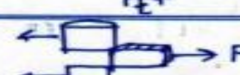
⑧ Design of thickness of spigot collar (t_1) — $\frac{1}{2}$ mark

$$\boxed{\tau = \frac{P}{\pi d_1 t_1}}$$

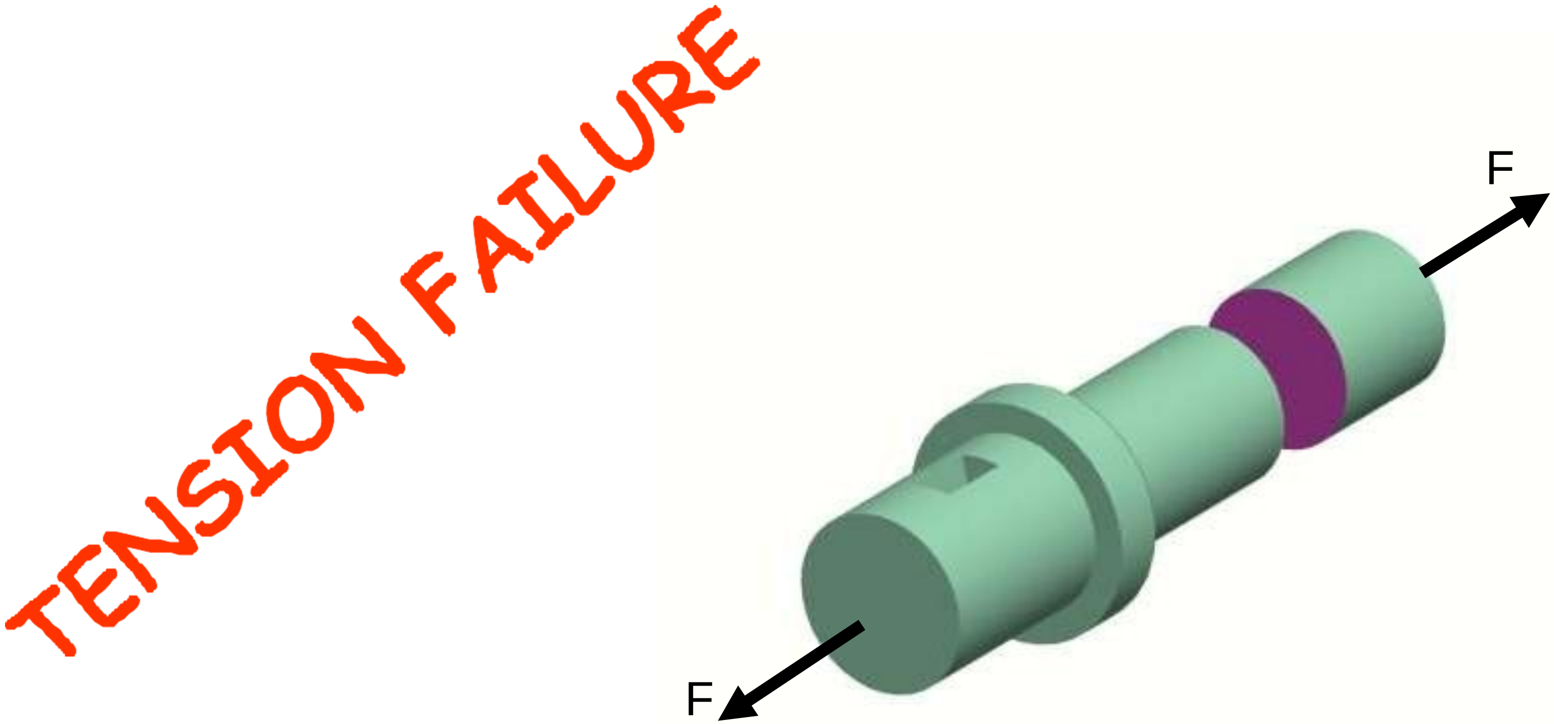


⑨ Design of width of cotter (b) — $\frac{1}{2}$ mark

$$\boxed{\tau = \frac{P}{2bt}}$$ — Double shear



Cotter Joint : Modes of Failure



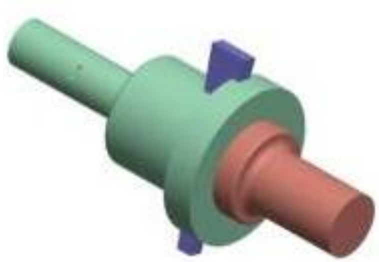
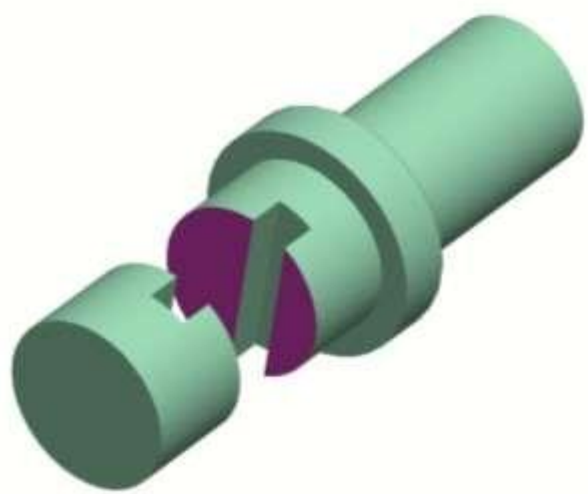


Fig. Steel Spigot Breaking in Tension Outside the Joint

Cotter Joint : Modes of Failure



TENSION FAILURE

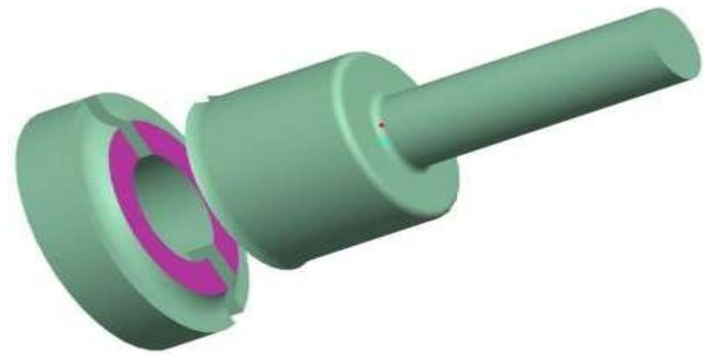


Fig. Spigot Breaking in Tension Across Slot

Fig. Socket Breaking in Tension Across Slot

Cotter Joint : Modes of Failure

SHEAR FAILURE

Fig. Double Shearing of Cotter Pin

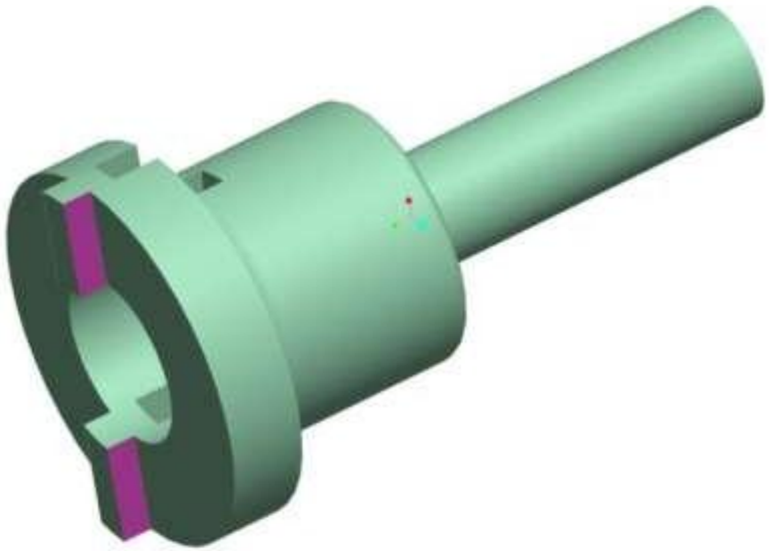
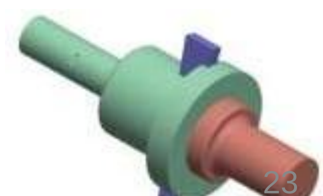


Fig. Double Shearing of Socket End



Cotter Joint : Modes of Failure

SHEAR FAILURE

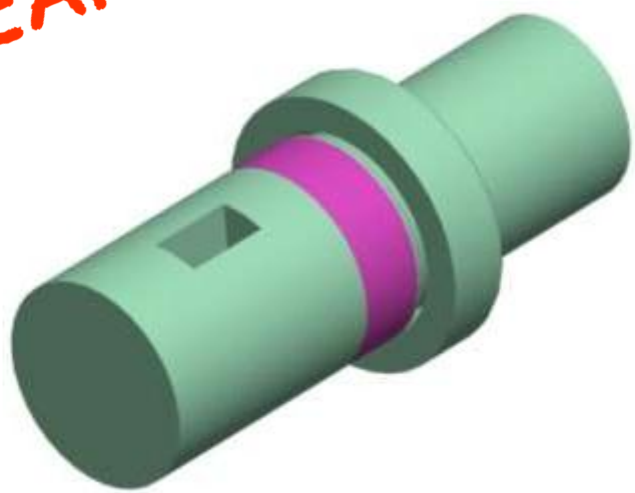


Fig. Shearing Away of the Collar in the Spigot

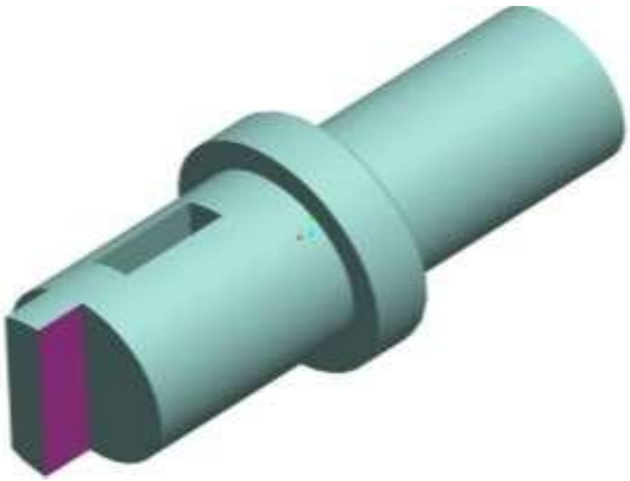
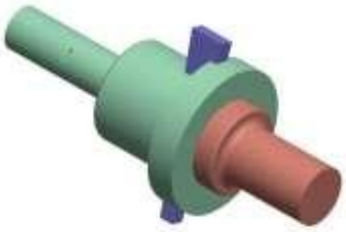


Fig 10: Shearing of Spigot End



Cotter Joint : Modes of Failure

BEARING FAILURE

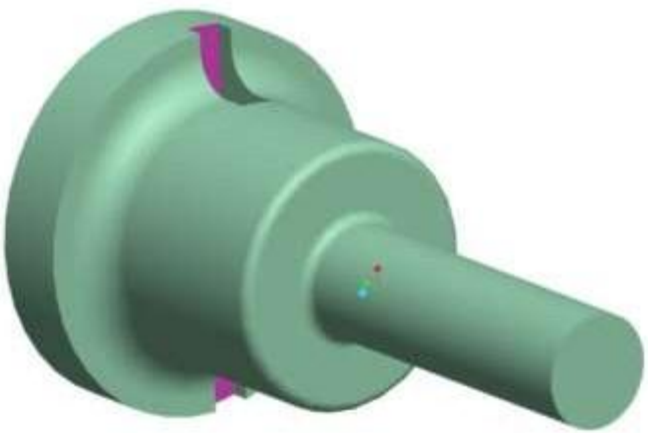


Fig. Crushing of Cotter Pin
Against Socket

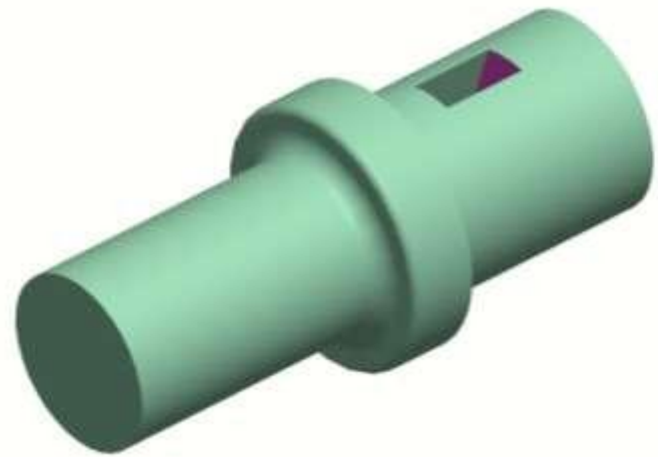
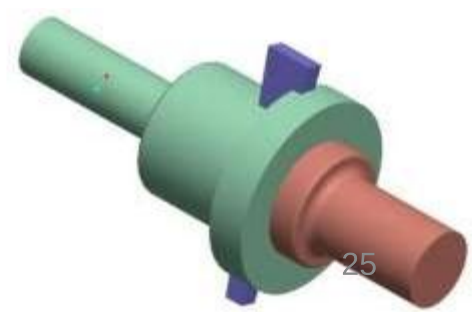
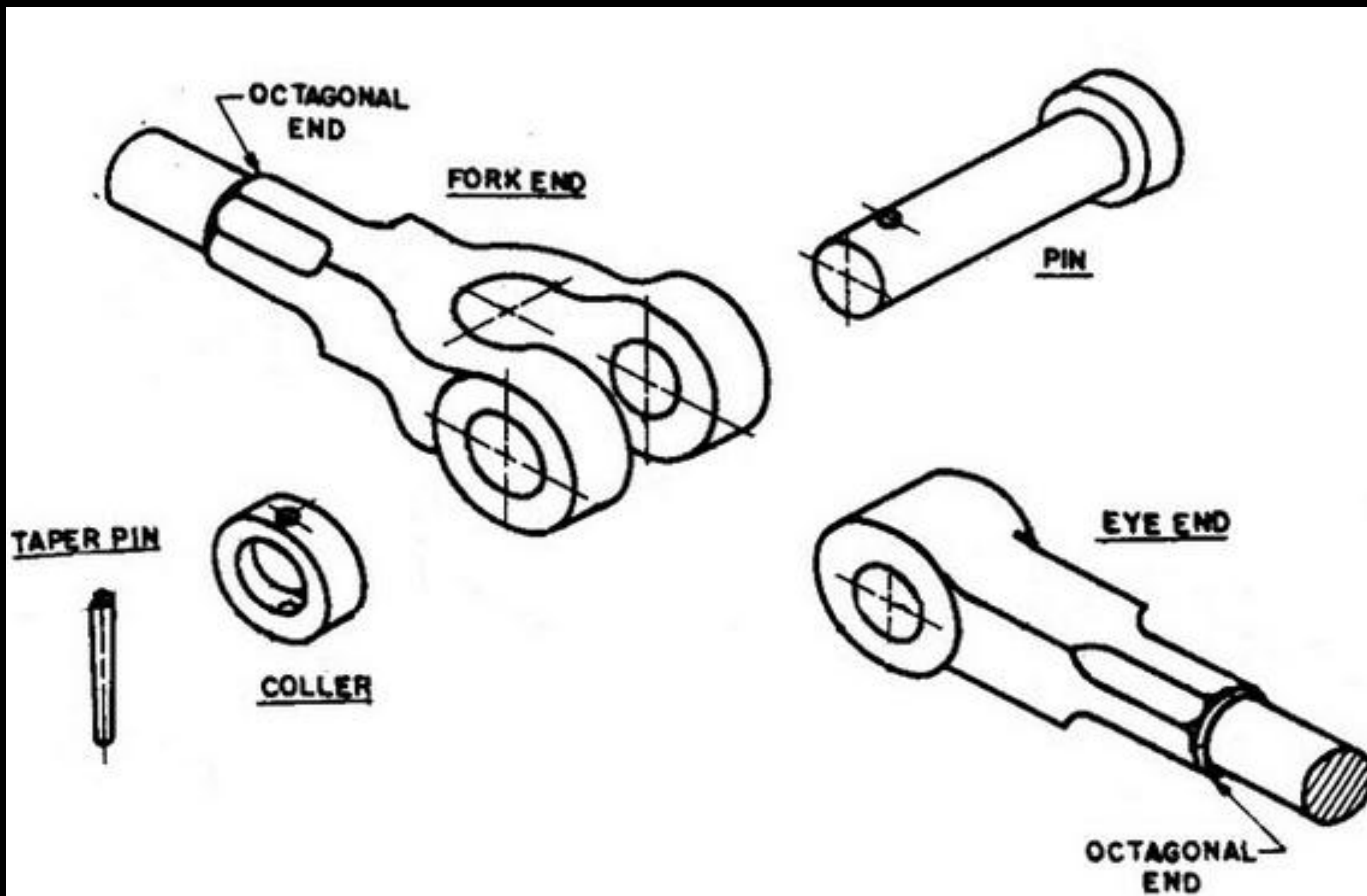


Fig. Crushing of Cotter Pin
Against Rod End



Knuckle Joint



Knuckle Joint

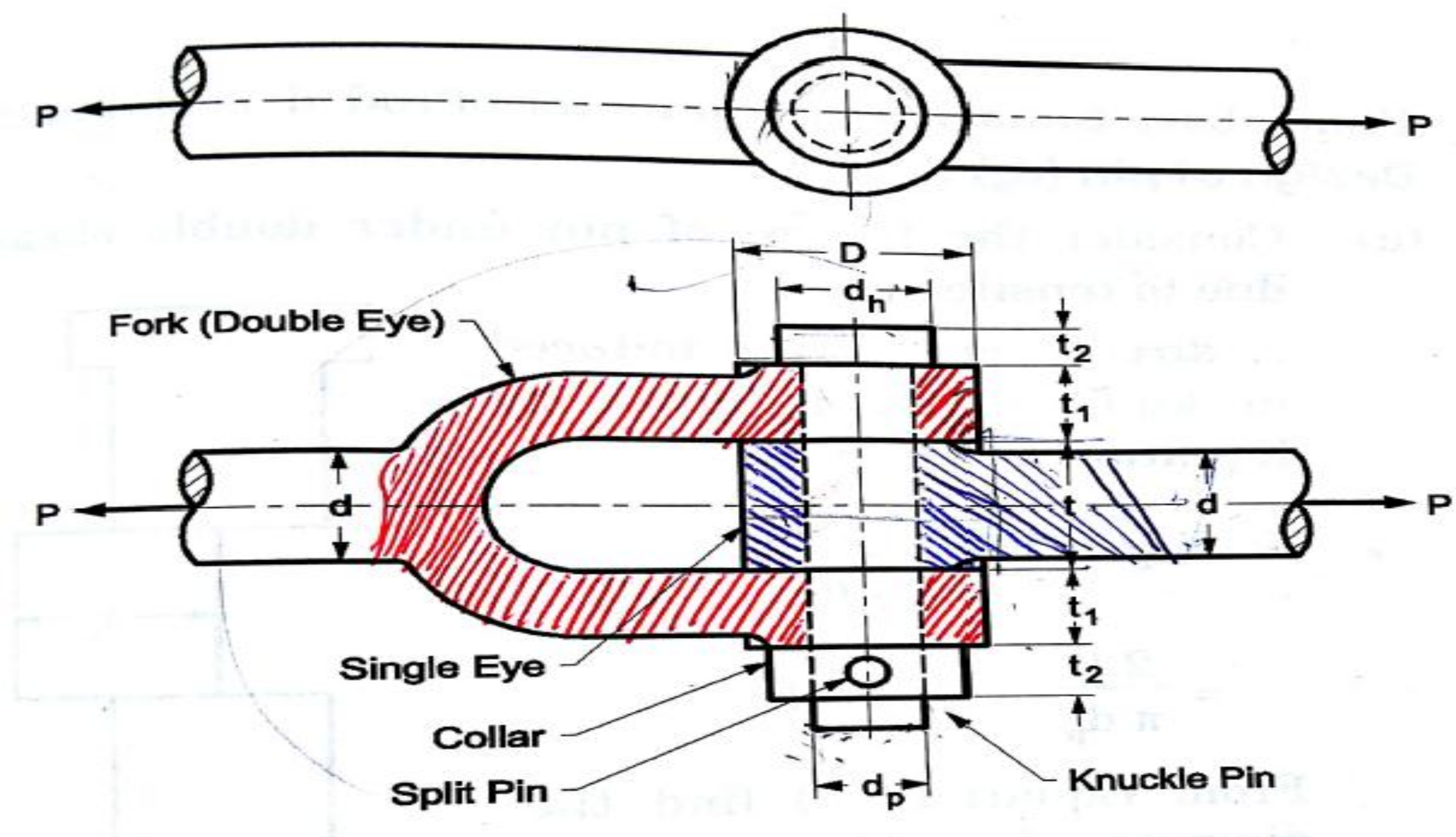
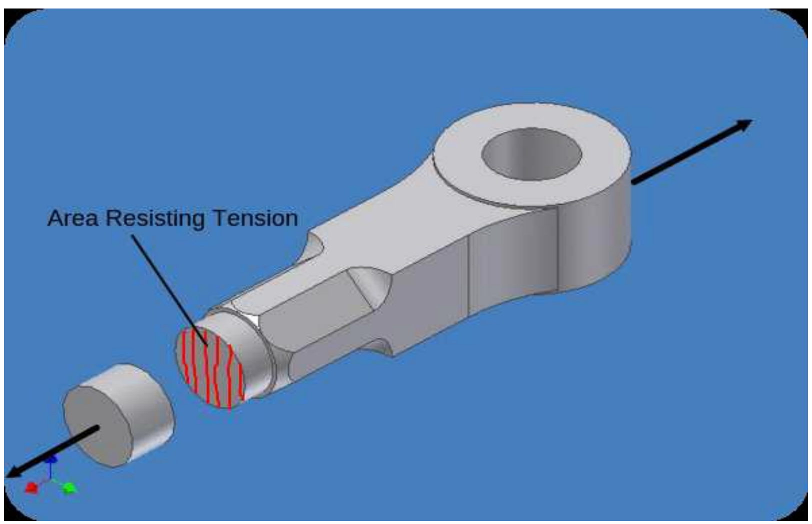


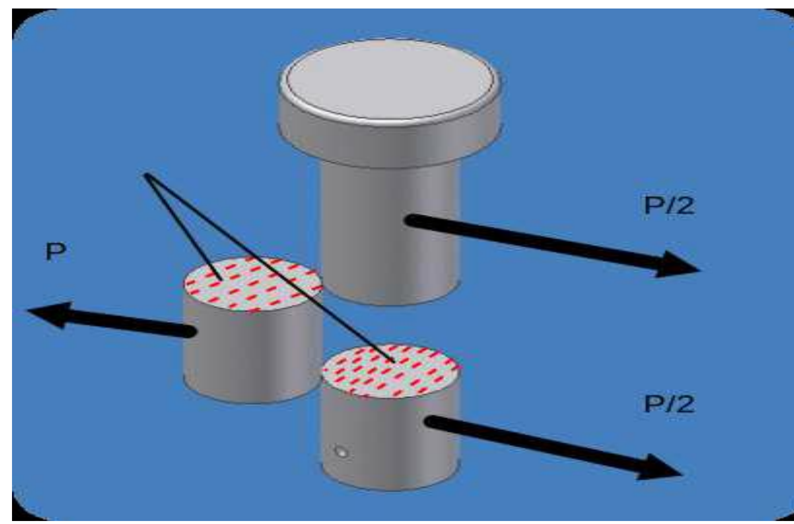
Fig. 2.2.1 : Knuckle joint

Design Procedure of Knuckle Joint

Sr. No.	Parameter to be Calculated	Stress Induced	Equation
1.	**Diameter of rod (d)**	Tensile stress (σ_t)	$\sigma_t = \frac{P}{(\pi d^2/4)}$
2.	**Diameter of knuckle pin (d_p)**	Direct shear stress (t)	$\tau = \frac{P}{2\,(\pi d_p^2)\,/\,4}$



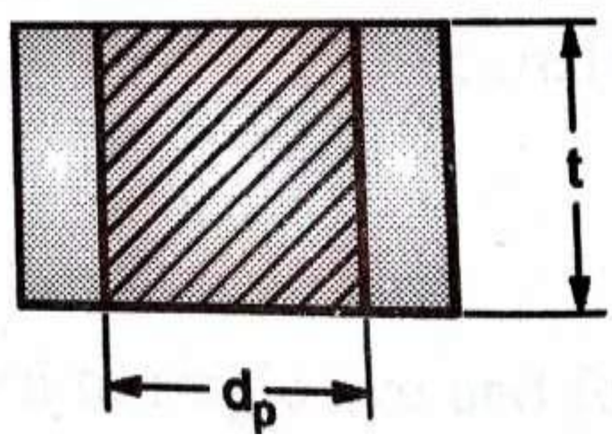
Diameter of rod (d)



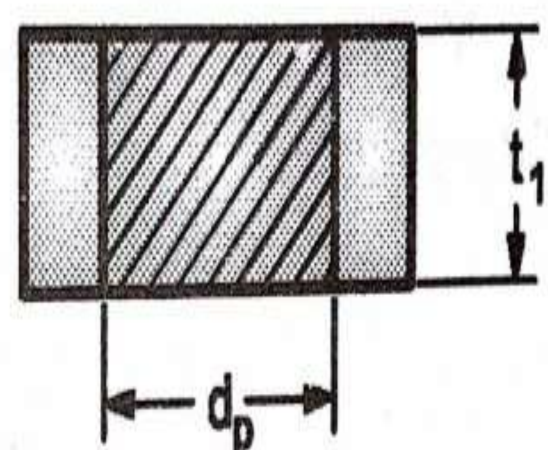
Diameter of knuckle pin (dp)

Design Procedure of Knuckle Joint

3.	**Thickness of single eye (t)**	(i) Crushing stress (σ_c) or bearing pressure (P_b) (ii) By proportion	σ_c or $P_b = \frac{P}{d_p t}$ $t = 2.25\ d$	't' is taken as larger of two values
4.	**Thickness of fork (t_1)**	(i) Crushing stress (σ_c) or bearing pressure (P_b) (ii) By proportion	σ_c or $P_b = \frac{P}{2\ d_p\ t_1}$ $t_1 = 0.75\ d$	't_1' is taken as larger of two values



Thickness of single eye (t)



Thickness of fork (t1)

Design Procedure of Knuckle Joint

5.	**Outside diameter of eye (D)**	(i) Tensile stress (σ_t) (ii) Direct shear stress (τ)	$\sigma_t = \frac{P}{(D - d_p)\, t}$ $\tau = \frac{P}{(D - d_p)\, t}$	'**D**' is taken as larger of two values.
6.	**Stresses in fork**	(i) Tensile stress (σ_t) (ii) Shear stress (τ)	$\sigma_t = \frac{P}{2\,(D - d_p)\, t_1}$ $\tau = \frac{P}{2\,(D - d_p)\, t_1}$	For safety : $\sigma_t < \sigma_{tall}$ $\tau < \tau_{all}$

Applications of Knuckle Joint

Levers

- Lever is defined as a mechanical device in the form of a rigid bar pivoted about the fulcrum to multiply or transfer the force.

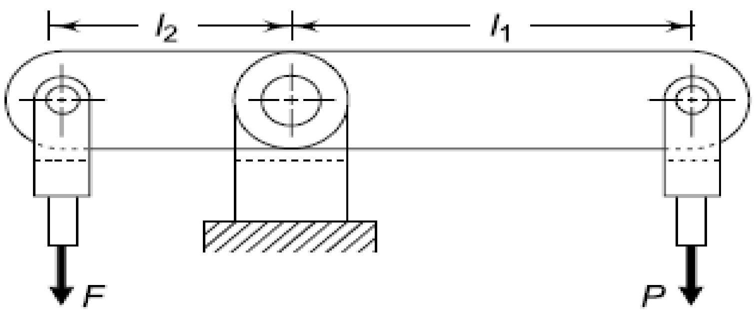


Fig. 4.42 *Construction of Lever*

- The ratio of load to effort (F/P) is called the 'mechanical advantage' of the lever.
- The ratio of the effort arm to the load arm (l1/l2) is called the 'leverage'.
- Taking moment of forces about the fulcrum,

$$F \times l_2 = P \times l_1$$

or

$$\frac{F}{\rho} = \frac{l_1}{l_2}$$

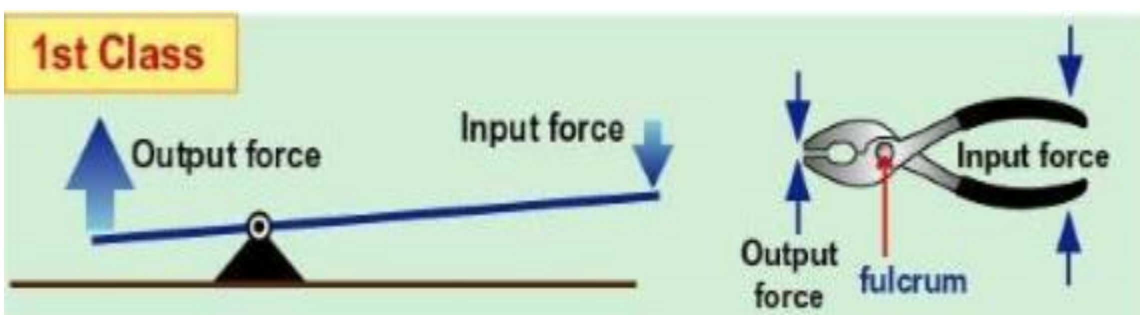
Types of Lever

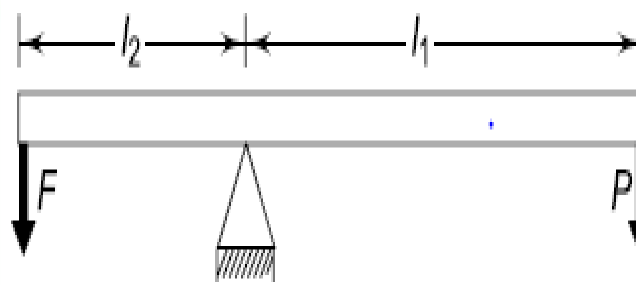
❖ **'First' type of lever:**

- The fulcrum is located between the load and the effort.
- In this case, the effort arm can be kept less than the load arm or equal to the load arm or more than the load arm.
- When $l1 < l2$,…..mechanical advantage < 1
- When $l1 = l2$,…..mechanical advantage = 1
- When $l1 > l2$,…..mechanical advantage > 1

❖ **Applications:**

- Rocker arm for the overhead valves of internal combustion engine
- Bell crank levers in railway signal mechanisms and
- Levers of hand pumps.

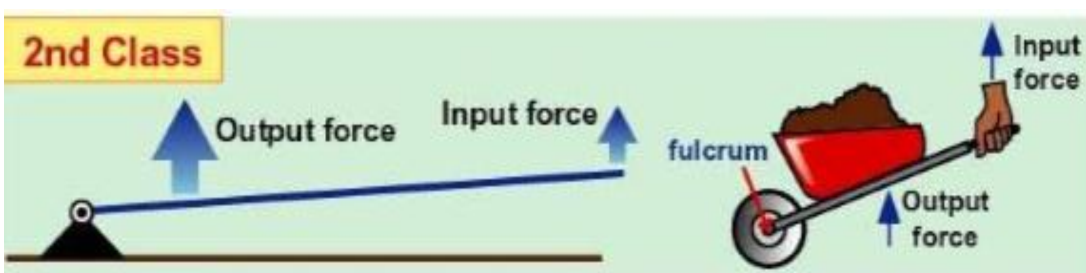


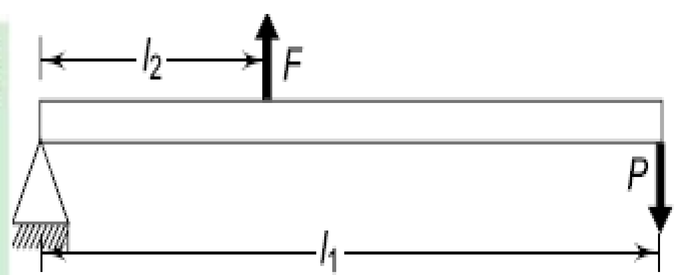


Types of Lever

- **‘Second’ type of lever:**
  - The load is located between the fulcrum and the effort.
  - In this case, the effort arm is always more than the load arm and the mechanical advantage is more than 1.

- **Application:**
  - Lever-loaded safety valves mounted on the boilers.

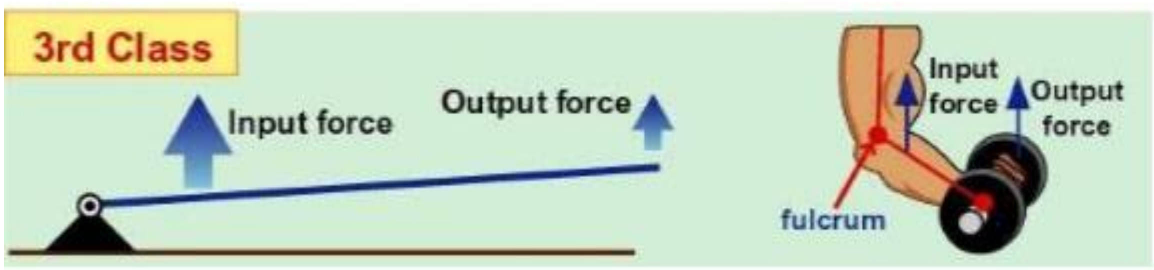


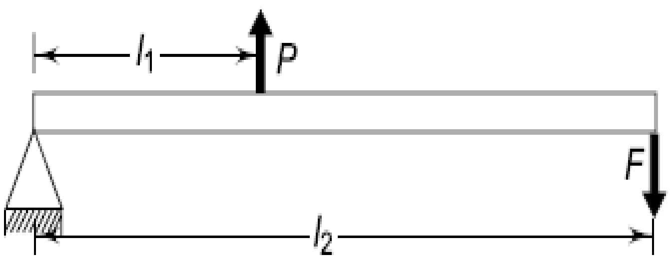


Types of Lever

❖ **'Third' type of lever:**
   - The effort is located between the load and the fulcrum.
   - In this case, the load arm is always greater than the effort arm and the mechanical advantage is less than 1.

❖ **Application:**
- Picking fork





Design of Angular Lever

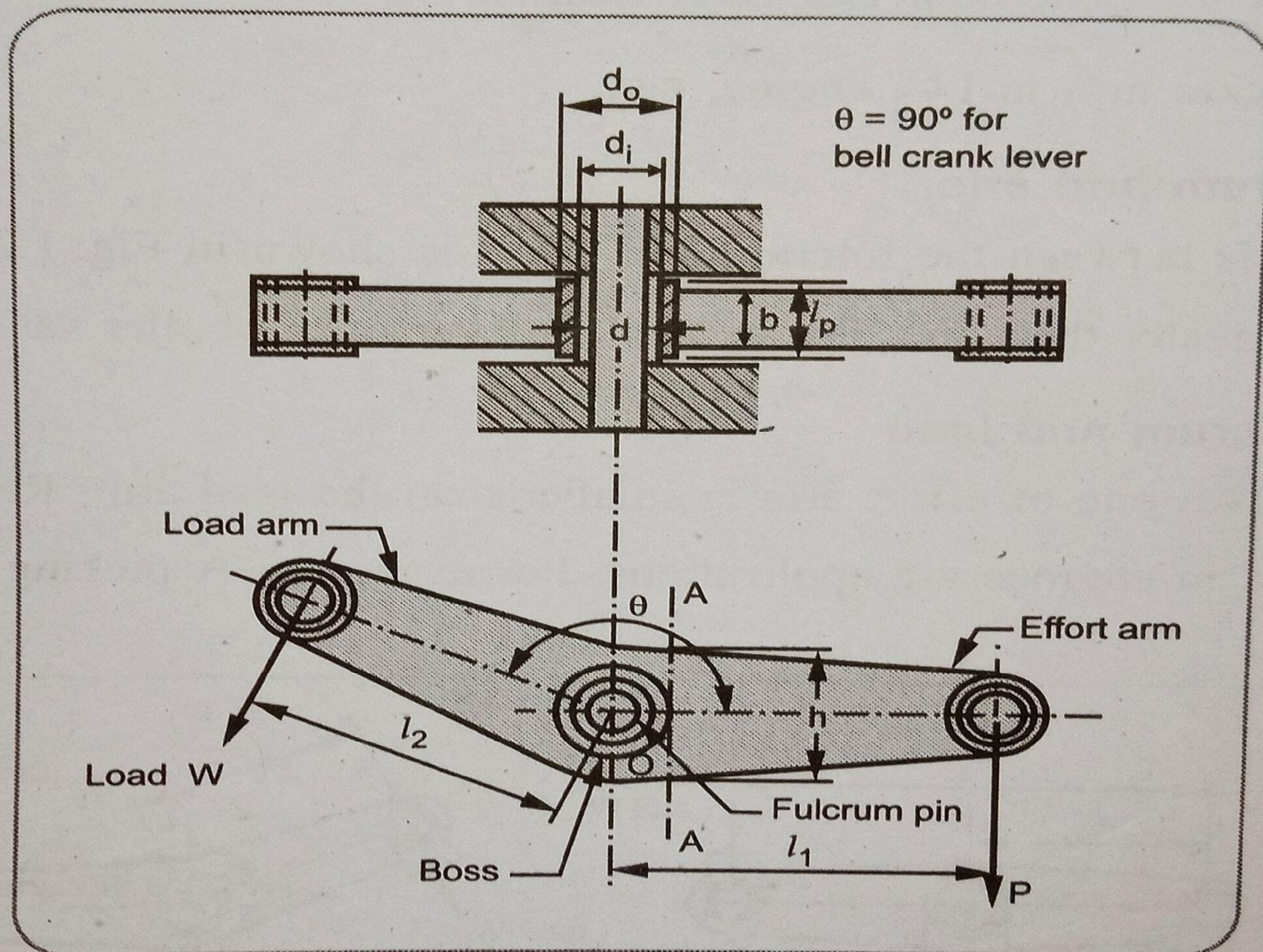


Fig. 1.22 : Angular lever

Design of Angular Lever

d = dia. of fulcrum pin (mm)
l_p = Supporting length of fulcrum pin (mm) = 1 d to 1.5 d
P_b = Permissible bearing pr. N/mm^2
d_o and d_i = outer and inner dia. of boss of lever (mm)

Design procedure :-

① Calculation of effort (P) :-
– considering moment about the fulcrum

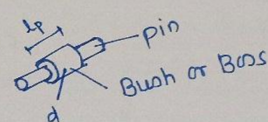
$$P \times l_1 = W \times l_2$$

② Calculation of fulcrum reaction (F_R)
– The F_R is given by law of parallelogram betn load w and P

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

θ :- Angle betn the lines of action of w and P.

③ Fulcrum pin dimensions :-
– considering bearing pr. betn fulcrum pin and boss of the lever



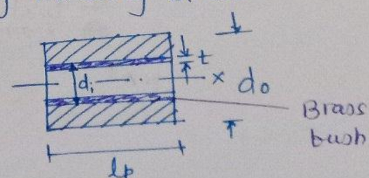
$$P_b = \frac{F_R}{d\, l_p}$$

– Failure of fulcrum pin in double shear $\tau = \dfrac{F_R}{2 \times \frac{\pi}{4} d^2}$

④ Dimensions of boss of lever :
i) Inner diameter (d_i) = $d + 2t$ & $t = 3$ mm with brass bush
$d_i = d$ w/o brass bush

ii) Outer dia : (d_o) Considering bending stress.

$$\sigma_b = \frac{M \cdot y}{I_{xx}} = \frac{M \times \frac{d_o}{2}}{\left[\frac{l_p}{12}(d_o^3 - d_i^3)\right]} \times$$



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

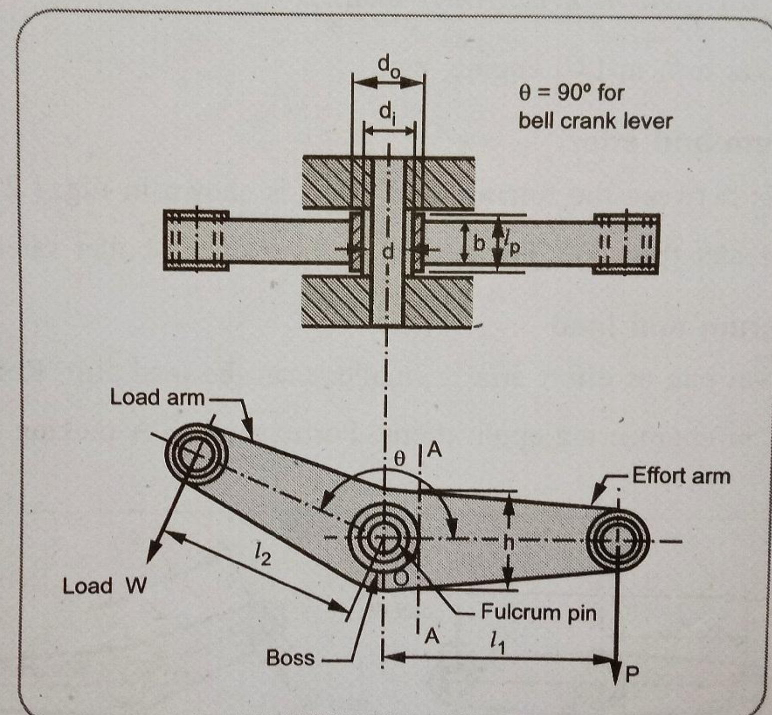


Fig. 1.22 : Angular lever

Design of Angular Lever

M = B.M on the boss of lever in N.mm

$M = W \times l_2 = P \times l_1$

Emperical relation $d_0 = 2d$

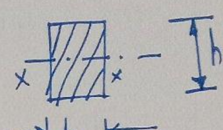
Select larger value of d_0

5) Dimensions of lever c/s :-

- Arms of levers are subjected to B.M
- Max. BM acts on the c/s adjacent to the boss

$$M_{max} = P\left(l_1 - \frac{d_0}{2}\right) \quad \therefore \sigma_b = \frac{M_{max} \times h/2}{bh^3/12}$$

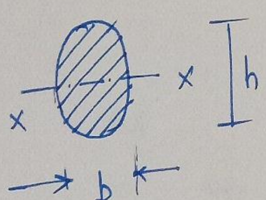
$$\sigma_b = \frac{6 M_{max}}{bh^2}$$ if rect. c/s



b - Breadth or thickness of lever (mm)

h - Depth or height of lever (mm) = 2b to 4b

For Elliptical section $\sigma_b = \dfrac{M_{max} \times h/2}{\pi b h^3/64} = \dfrac{32 M_{max}}{\pi b h^2}$



b = Minor axis of ellipse (mm)

h = Major axis of ellipse (mm)

= 2b to 2.5 b.

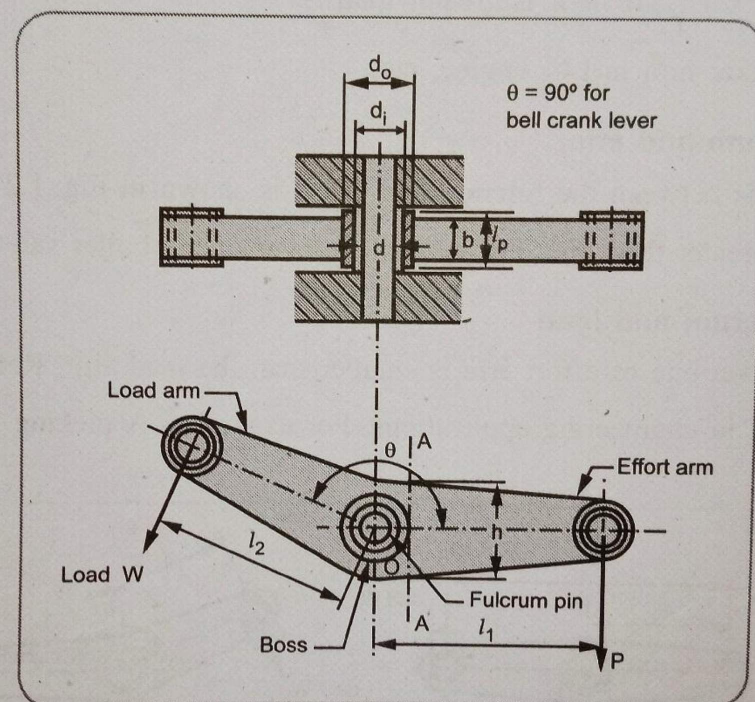


Fig. 1.22 : Angular lever

Eccentric Loading

❖ **Eccentric Load (P):**
When the line of action of an external axial load is parallel but not co-axial with the centroidal axis of component.

❖ **Eccentricity (e):**
The distance between the centroidal axis of the machine component and the axis of load.

❖ **Examples:**
- ➢ Frames of punching machine
- ➢ Clamps
- ➢ Brackets

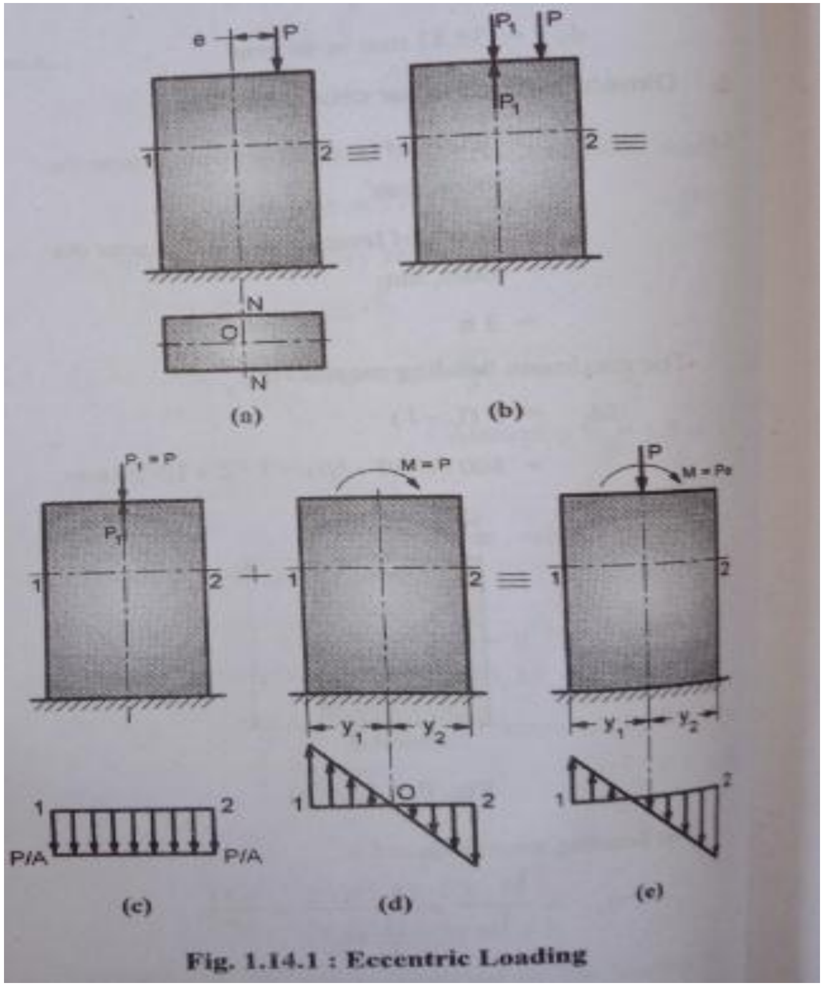


Fig. 1.14.1 : Eccentric Loading

Stresses induced in bar due to Eccentric Loading

- **Direct Compressive Stress:**

$$\sigma_d = \frac{P}{A} \text{(Compressive)} \qquad \ldots($$

where, σ_d = direct stress induced in a bar, N/mm^2

A = cross-sectional area of the bar, mm^2

- **Bending Stress:**

$$\sigma_b = \frac{M \cdot y}{I} \qquad \ldots$$

where, σ_b = bending stress at a distance y from the neutral axis (i.e. NN axis), N/mm^2

I = moment of inertia of cross-section about neutral axis (i.e. NN axis), mm^4

M = bending moment = Pe, N-mm

The bending stress σ_{b1} at point 1 is,

$$\sigma_{b1} = \frac{My_1}{I} \text{(tensile)}$$

and the bending stress σ_{b2} at point 2 is,

$$\sigma_{b2} = \frac{My_2}{I} \text{(compressive)}$$

- **Resultant Stress:**

The resultant stress σ_1 at point 1 is,

$$\sigma_1 = -\sigma_d + \sigma_{b2}$$

or $$\sigma_1 = -\frac{P}{A} + \frac{My_1}{I}$$

The resultant stress σ_2 at point 2 is,

$$\sigma_2 = -\sigma_d + \sigma_b$$

or $$\sigma_2 = -\frac{P}{A} - \frac{My_2}{I}$$

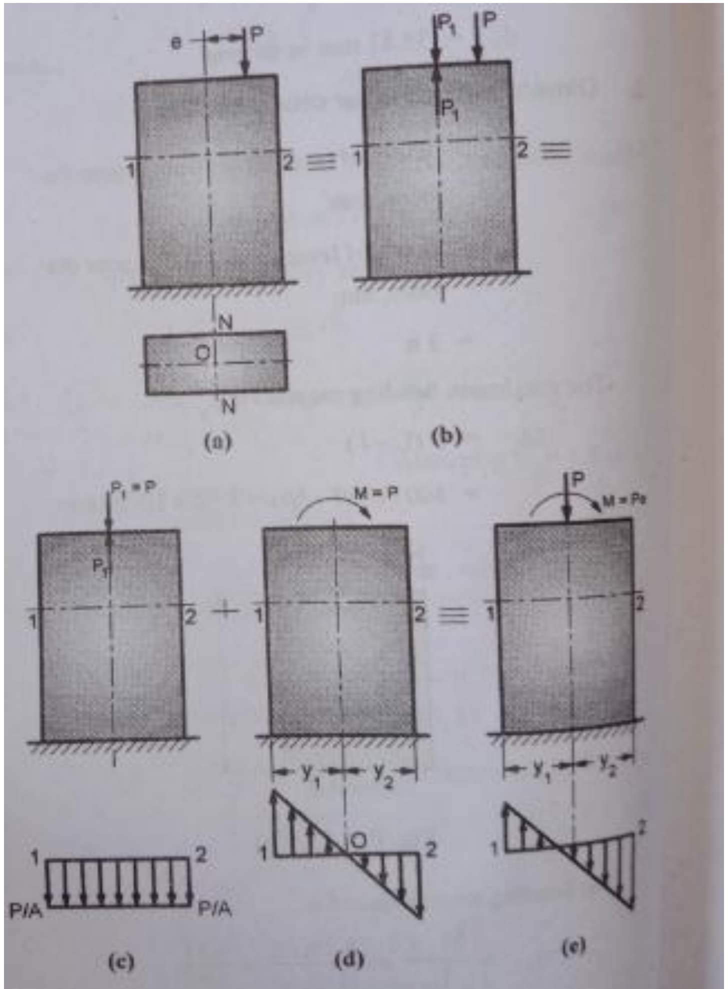


Fig. 1.14.1 : Eccentric Loading