Subject- Design Of Machine Elements

Prepared by:- Mrs. Rutuparna Sahu

MACHINE DESIGN CHAPTER -1(INTRODUCTION)

SYLLABUS:-

1.1 Introduction to Machine Design and Classify it.
1.2 Different mechanical engineering materials used in design with their uses and their mechanical and physical properties
1.3 Define working stress, yield stress, ultimate stress & factor of safety and stress- strain curve for M.S & C.I.
1.4 Modes of Failure (By elastic deflection, general yielding & fracture)
1.5 State the factors governing the design of machine elements.
1.6 Describe design procedure.

1.1 Introduction to Machine Design and classify it:-

Machine Design:-

1 Machine Design is the creation of new and better machines and improving the existing ones.
2 Machine design is defined as the use of scientific principle, technical information & imagination in the description of a mechanical system to perform specific functions with maximum economic and efficiency.

Classification of Machine Design:-

1. **Adaptive design**:- Here designer's work is concerned with adaptation of existing designs. This type of design needs no special knowledge or skill.The designer only makes minor alternation or modification in the existing designs of the product.
2. **Development design**:-This type of design needs scientific training and design ability in order to modify the existing designs into a new idea by adopting a new material or different method of manufacture. In this case, though the designer starts from the existing design, but the final product will be quite different from the original product.
3. **New design**:-This type of design needs lot of research, technical ability and creative thinking. From this design completely new product will be formed.

The designs, depending upon the methods used, may be classified as follows:

- **Rational design**: - This type of design depends upon mathematical formulae of principle of mechanics.
- **Industrial design**: - This type of design depends upon the production aspects to manufacture any machine component in the industry.
- **System design**:-It is the design of any complex mechanical system like a motor car.
- **Element design: -** It is the design of any element of the mechanical system like piston, crankshaft, connecting rod, etc.
- **Computer aided design: -** This type of design depends upon the use of computer systems to assist in the creation, modification & analysis of a design.

LOAD: -_It is defined as any external force acting upon a machine part. The following four types of the load are:-

1**. Dead or steady load**:-A load is said to be a dead or steady load, when it does not change in magnitude or direction.

2. **Live or variable load**:-A load is said to be a live or variable load, when it changes continually.

3. **Suddenly applied or shock loads**: - A load is said to be a suddenly applied or shock load, when it is suddenly applied or removed.

4. **Impact load**:-A load is said to be an impact load, when it is applied with some initial velocity.

1.2 Different mechanical engineering materials used in design with their uses and their mechanical and physical properties:-

Mechanical Properties of Metals:-

1. Strength: - It is the ability of a material to resist the externally applied forces without breaking. The internal resistance offered by a part to an externally applied force is called stress.

2. Stiffness:-It is the ability of a material to resist deformation under stress. The modulus of elasticity is the measure of stiffness.

3. Elasticity: - It is the property of a material to regain its original shape after deformation when the external forces are removed. This property is desirable for materials used in tools and machines. Steel is more elastic than rubber.

4. Plasticity: - It is property of a material which retains the deformation produced under load permanently. This property of the material is necessary for forgings, rolling process.

5. Ductility: - It is the property of a material enabling it to be drawn into wire with the application of a tensile force. The ductility is usually measured by the terms, percentage elongation and percentage reduction in area. The ductile material commonly used in engineering practice are mild steel, copper, aluminium, nickel, zinc, tin and lead. The ductility of a material is commonly measured by means of percentage elongation and percentage reduction in area in a tensile test

Percentage elongation:-

Let l = original length, and

L = final length.

∴ Elongation = L – l

And percentage elongation = ((L–l)/l) * 100

Percentage reduction in area:-

Let A = Original cross-sectional area

a = Cross-sectional area at the neck.

Then reduction in area = A – a

Percentage reduction in area = ((A-a)/A)*100

6. Brittleness:-It is the property of a material opposite to ductility. It is the property of breaking of a material with little permanent deformation. Cast iron is a brittle material.

7. Malleability:-It is a special case of ductility which permits materials can be converted into thin sheets by the application of compressive force. The malleable materials commonly used in engineering practice are lead, soft steel, wrought iron, copper and aluminium.

8. Toughness:-It is the property of a material to resist fracture due to high impact loads like hammer blows. It is measured by the amount of energy that a unit volume of the material has absorbed after being stressed up to the point of fracture.

9. Machinability:-It is the property of a material which refers to a relative case with which a material can be cut. The machinability of a material can be measured in a number of ways such as comparing the tool life for cutting different materials or force required to remove the material at some given rate or the energy required to remove a unit volume of the material. It may be noted that brass can be easily machined than steel.

10. Resilience: - It is the property of a material to absorb energy and to resist shock and impact loads. It is measured by the amount of energy absorbed per unit volume within elastic limit. This property is essential for spring materials.

11. Creep: - When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called creep. This property is considered in designing internal combustion engines, boilers and turbines.

12. Fatigue:-When a material is subjected to repeated or fluctuating stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as fatigue. The failure consists of initiation & propagation of crack. This property is considered in designing shafts, connecting rods, springs, gears, etc.

13. Hardness:-It is the property by virtue of which material can resist wear, penetration, deformation. It also means the ability of a metal to cut another metal. Hardness is determined by the following tests:

(a) Brinell hardness test

(b) Rockwell hardness test

(c) Vickers hardness (also called Diamond Pyramid) test

(d) Shore scleroscope.

Physical Properties of Metals:- The physical properties of the metals include colour, size and shape, density, electric and thermal conductivity, and melting point.

1.3 Define working stress, yield stress, ultimate stress & factor of safety and stress-strain curve for M.S & C.I.

Yield stress: - It is defined as the maximum stress at which increase in elongation occurs without increase in load. After yield point on removal of the load the material will not be able to recover its original shape and size. Stress corresponding to yield point is known as yield point stress.

Ultimate stress: - The stress, which attains its maximum value is known as ultimate stress. It is obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

Working Stress: - When designing machine parts, it is desirable to keep the stress lower than the maximum or ultimate stress at which failure of the material takes place. This stress is known as the working stress or design stress. It is also known as safe or allowable stress.

(**Note**: By failure it is not meant actual breaking of the material. Some machine parts are said to fail when they have plastic deformation set in them, and they no more perform their function satisfactorily)

Factor of Safety:-

It is defined, as the ratio of the maximum stress to the working stress. Mathematically,

Factor of safety = (Maximum stress)/(Working or design stress)

In case of **ductile materials** e.g. mild steel, where the yield point is clearly defined, the factor of safety is based upon the **yield point stress**. In such cases,

Factor of safety =

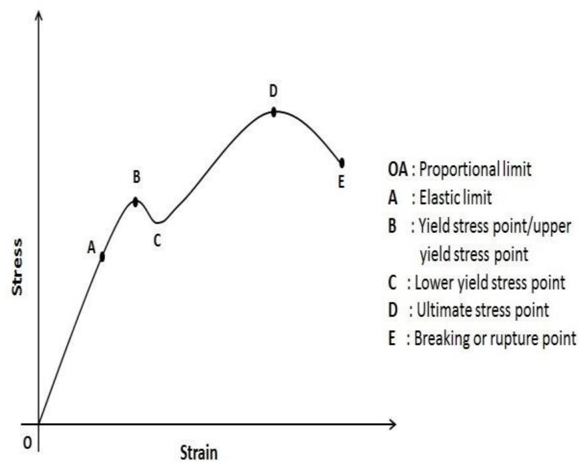
(Yield point stress)/(Working or design stress)

In case of **brittle materials** e.g. cast iron, the yield point is not well defined as for ductile materials.

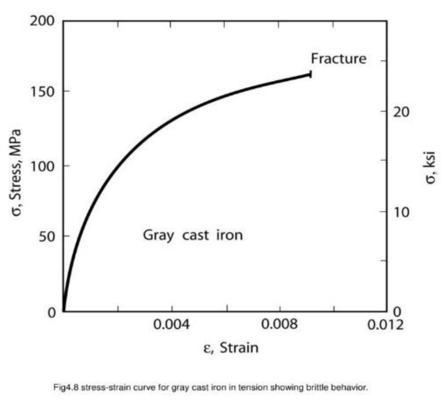
Therefore, the factor of safety for brittle materials is based on **ultimate stress**.

∴ Factor of safety =

(Ultimate stress)/(Working or design stress)



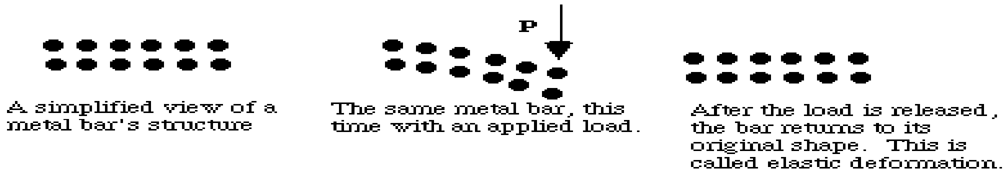
(Stress-strain curve for mild steel)



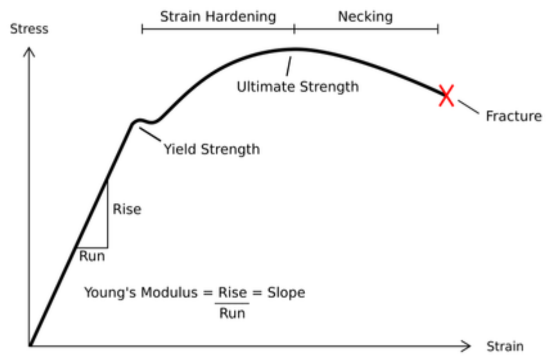
(Stress-strain curve for cast iron)

1.4 Modes of Failure (By elastic deflection, general yielding & fracture):-

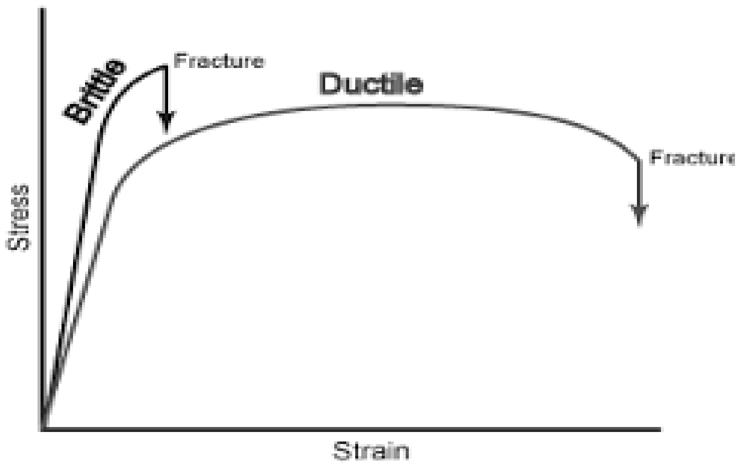
1. By elastic deformation: - This type of deformation is reversible. When the force is removed the object will return to its original shape. Elastomer like rubber shows large elastic deformation. But in metals, ceramics show smaller elastic range.



2. **Yielding: -** Yield strength is defined as the stress at which material begins to deform plastically. Prior to yield point the material will deform elastically & will return to original state after the removal of load. But once the yield point is attained some fraction of deformation will be permanent & non reversible.



3. **Fracture**: - Under tensile stress plastic deformation is characterized by a strain hardening region and a necking region & finally fracture occurs.



1.5 State the factors governing the design of machine elements:-

1**. Strength**: - A machine part shouldn’t fail under the effect of forces acting on it. It should have sufficient strength to avoid failure due to fracture or yielding.

2**. Rigidity**: - Machine component should be rigid enough, so that it will not deflect or bend due to forces or moment acting on it. A transmission shaft is designed on the basis of lateral rigidity & torsional rigidity.

3. **Wear resistance**: - A machine component should be wear resistant because wear reduces accuracy of machine tool along with its life cycle. Surface hardening will increase the wear resistance of the machine component.

4. **Safety: -** The shape and dimension of the machine part should ensure safety to the operator of the machine.

5. **Minimum dimension and weight**: - A machine should have minimum possible dimension & weight which will reduce the material cost.

6. **Conformance to the standard**: - Machine part should conform to the national & international standards covering the dimension, profile & material.

7. **Minimum life cycle cost**: - Total cost i.e. to be paid for purchasing the parts, operating & maintaining it over its life span should be minimum.

1.6 Describe design procedure:-

The general procedure to solve a design problem is as follows:

1. **Recognition of need**:-First of all the need, aim or purpose for which the machine is to be designed should be recognised.

2**. Mechanisms**:-Select the possible mechanism or group of mechanisms which will give the desired motion.

3**. Analysis of forces**: - Find the forces acting on each member of the machine and the energy transmitted by each member.

4**. Material selection**:-Select the material best suited for each member of the machine.

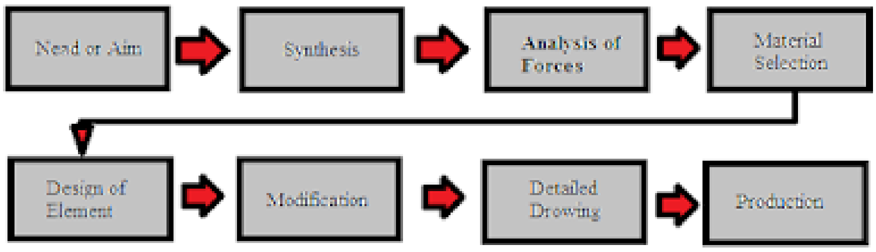
5. **Design of elements**: - Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform more than the permissible limit.

6**. Modification**: - Modify the size of the member to reduce overall cost.

7**. Detailed drawing**:-Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes.

8. **Production**: - The component, as per the drawing, is manufactured in the workshop.

The flow chart for the general procedure in machine design is shown in Fig.



General Procedure in Machine Design

DESIGN OF MACHINE ELEMENTS

CHAPTER-2 :-(DESIGN OF FASTENING ELEMENTS)

PREPARED BY:-RUTUPARNA SAHU

SYLLABUS:-

2.1 Joints and their classification.

2.2 State types of welded joints.

2.3 State advantages of welded joints over other joints.

2.4 Design of welded joints for eccentric loads.

2.5 State types of riveted joints and types of rivets.

2.6 Describe failure of riveted joints.

2.7 Determine strength & efficiency of riveted joints.

2.8 Design riveted joints for pressure vessel.

2.9 Solve numerical on Welded Joint and Riveted Joints

2.1 Joints and their classification:-

A machine or a structure is made of a large number of parts and they need to be joined suitably for the machine to operate satisfactorily. Parts are joined by fasteners and fasteners are conveniently classified as permanent or detachable fasteners. Again permanent and detachable joints are classified as:-

Permanent fasteners: - The permanent fastenings are those fastenings which cannot be disassembled without destroying the connecting components. The examples of permanent fastenings are soldered, brazed, welded and riveted joints.

Detachable fasteners:-The temporary or detachable fastenings are those fastenings which can be disassembled without destroying the connecting components. The examples of temporary fastenings are screws, nuts, bolts, keys, cotters, pins and knuckle joint.

2.2 State types of welded joints:-

a) A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material.

b) The heat required for the fusion of the material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding).

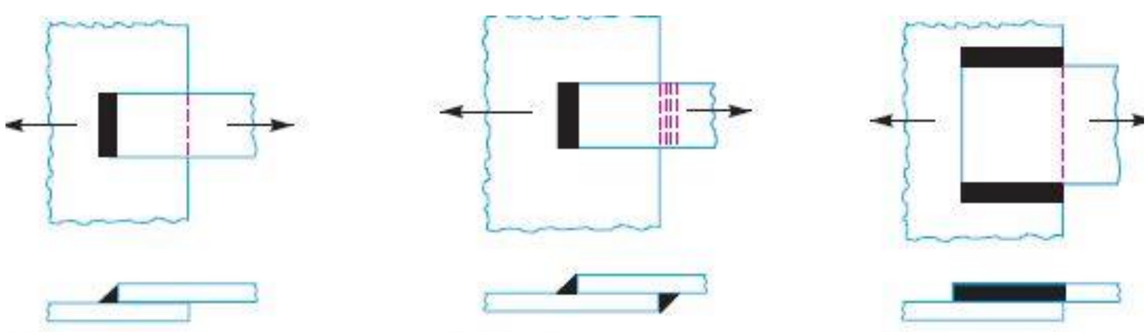
Types of welded joints: - There are two types of welded joints.

1) Lap joint 2) Butt joint

1) Lap Joint/Fillet Joint:-

a) The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates.
b) The cross-section of the fillet is approximately triangular.
c) The fillet joints may be:-

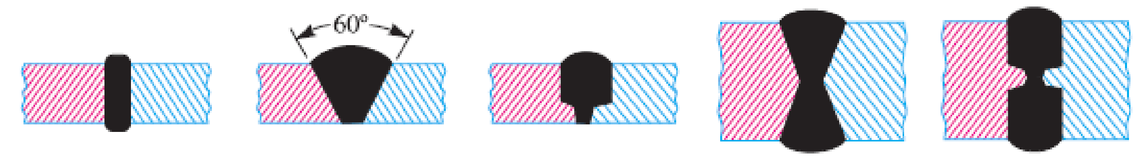
1. Single transverse fillet
2. Double transverse fillet
3. Parallel fillet joints.



(a) Single transverse. (b) Double transverse. (c) Parallel fillet.

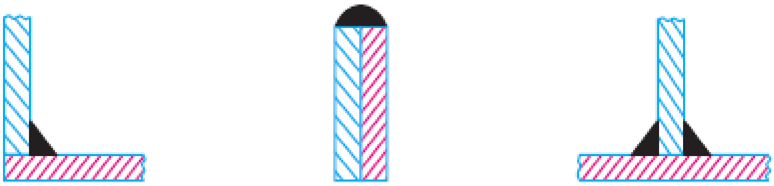
2) Butt Joint:-

a) The butt joint is obtained by placing the plates edge to edge.
b) In butt welds, the plate edges do not require bevelling if the thickness of plate is less than 5 mm.
c) On the other hand, if the plate thickness is 5 mm to 12.5 mm, the edges should be bevelled to V or U-groove on both sides.
d) The butt joints may be
1. Square butt joint
2. Single V-butt joint
3. Single U-butt joint
4. Double V-butt joint
5. Double U-butt joint.



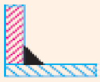
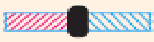
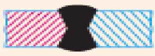
(a) Square butt joint. (b) Single V-butt joint. (c) Single U-butt joint. (d) Double V-butt joint. (e) Double U-butt joint.

Some other types of welded joints:-



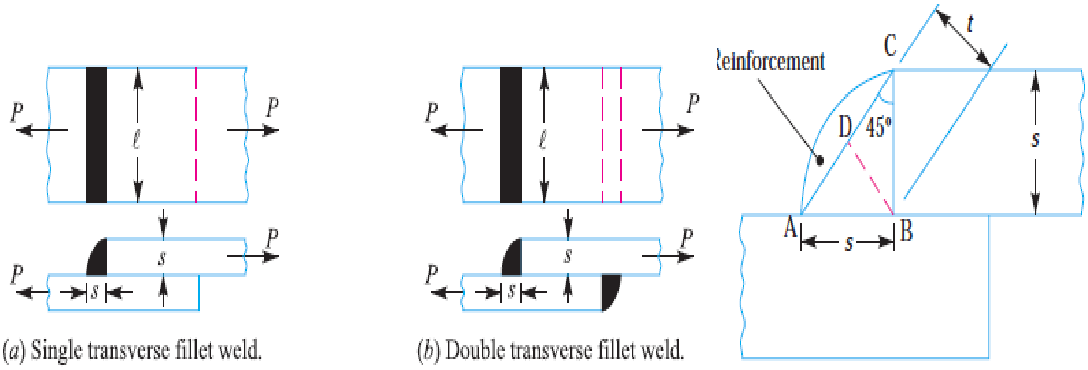
(a) Corner joint. (b) Edge joint. (c) T-joint.

Some Basic Weld Symbols Are:-

S. No.	Form of weld	Sectional representation	Symbol
1.	Fillet		
2.	Square butt		
3.	Single-V butt		
4.	Double-V butt		
5.	Single-U butt		
6.	Double-U butt		

Strength of Transverse Fillet Welded Joints:-

The transverse fillet welds are designed for tensile strength. Let us consider a single and double transverse fillet welds as shown in figure.



(a) Single transverse fillet weld. (b) Double transverse fillet weld.

(Enlarged view of fillet weld)

In order to determine the strength of the fillet joint, it is assumed that the section of fillet is a right angled triangle ABC with hypotenuse AC making equal angles with other two sides AB and BC. The enlarged view of the fillet is shown in figure. The length of each side is known as **leg or size** of the weld and the perpendicular distance of the hypotenuse from the intersection of legs (i.e. BD) is known as **throat thickness**. The **minimum area** of the weld is given by the **product of the throat thickness and length of weld.**

Let t = Throat thickness (BD)

s = Leg or size of weld= Thickness of plate

l = Length of weld,

From Figure we find that the throat thickness,

$t = s \times \sin 45° = 0.707\, s$

∴ Minimum area of the weld or throat area,

A = Throat thickness × Length of weld = $t \times l = 0.707\, s \times l$

If σt is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld,

P = Throat area × Allowable tensile stress = $0.707\, s \times l \times \sigma t$

And tensile strength of the joint for double fillet weld,

$P = 2 \times 0.707\, s \times l \times \sigma t = 1.414\, s \times l \times \sigma t$

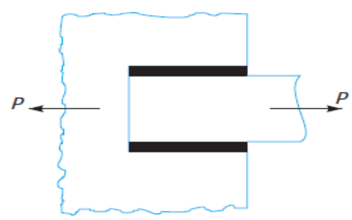
Strength of Parallel Fillet Welded Joints:-

The parallel fillet welded joints are designed for shear strength. Consider a double parallel fillet welded joint as shown in Figure (a)

We have already discussed in the previous case, that the minimum area of weld or the throat area, **$A = 0.707\, s \times l$**

If τ is the allowable shear stress for the weld metal, then the shear strength of the joint for single parallel fillet weld,

P = Throat area × Allowable shear stress = $0.707\, s \times l \times \tau$



(a) Double parallel fillet weld.

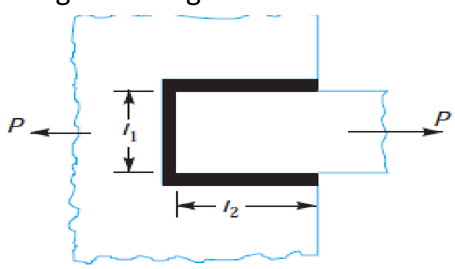
Shear strength of the joint for double parallel fillet weld,

$P = 2 \times 0.707 \times s \times l \times \tau = 1.414\, s \times l \times \tau$

<u>Notes:-</u> If there is a combination of single transverse and double parallel fillet welds as shown in Figure (b) then the strength of the joint is given by the sum of strengths of single transverse and double parallel fillet welds.

Mathematically,

$P = 0.707s \times l1 \times \sigma t + 1.414\, s \times l2 \times \tau$



(b) Combination of transverse and parallel fillet weld.

Strength of Butt Joints:-

The butt joints are designed for tension or compression. In case of butt joint, the length of leg or size of weld is equal to the throat thickness which is equal to thickness of plates.

Tensile strength of the butt joint (single-V or square butt joint):-

$P = t \times l \times \sigma t$

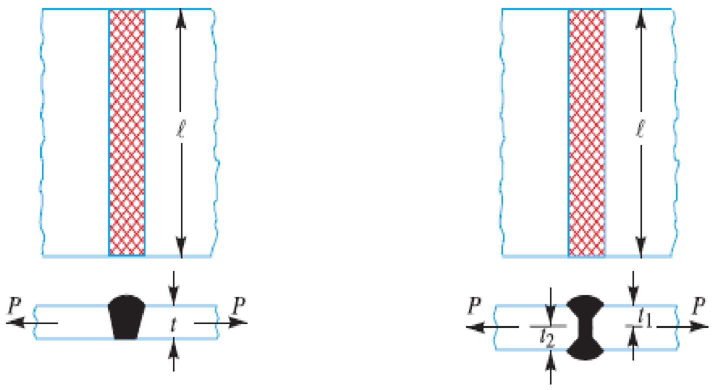
Where, l = Length of weld. It is generally equal to the width of plate.

Tensile strength for double-V butt joint as shown in Fig (b) is given by

$P = (t1 + t2)\ l \times \sigma t$

Where t1 = Throat thickness at the top

t2 = Throat thickness at the bottom.



(a) Single V-butt joint. (b) Double V-butt joint.

Q) A plate 100 mm wide and 10 mm thick is to be welded to another plate by means of double parallel fillets. The plates are subjected to a static load of 80 kN. Find the length of weld if the permissible shear stress in the weld does not exceed 55 MPa.

Ans.:- Given: Width = 100 mm; Thickness = 10 mm; P = 80 KN = 80 × 1000N;

τ = 55 MPa = 55 N/mm2 Let L=Length of weld, and s = Size of weld = Plate thickness = 10 mm

We know that maximum load which the plates can carry for double parallel fillet weld

$P= 80 \times 1000 = 1.414 \times s \times L \times \tau = 1.414 \times 10 \times L \times 55 = 778\ L$

∴ L= 80000 / 778 = 103 mm

Q) A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of double parallel fillet welds. The plates are subjected to a static load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 56 MPa.

Ans: - Given: Width = 100 mm; Thickness = 12.5 mm; P = 50 KN = 50 × 1000N;

τ = 56 MPa = 56 N/mm2 Length of weld for static loading

Let L = Length of weld, and

s = Size of weld = Plate thickness = 12.5 mm ... (Given)

We know that the maximum load which the plates can carry for double parallel fillet welds

$P=50 \times 1000=1.414\ s \times L \times \tau= 1.414 \times 12.5 \times L\times 56 = 990\ L$

∴ L = 50000/ 990 = 50.5 mm

2.3 State advantages of welded joints over other joints:-

Advantages:-

1. The welded structures are usually lighter than riveted structures.

2. The welded joints provide maximum efficiency (may be 100%) which is not possible in case of riveted joints.

3. Alterations and additions can be easily made in the existing structures.

4. As the welded structure is smooth in appearance, therefore it looks pleasing.

5. A welded joint has a great strength. Often a welded joint has the strength of the parent metal itself.

6. Sometimes, the members are of such a shape (i.e. circular steel pipes) that they afford difficulty for riveting. But they can be easily welded.

7. It is possible to weld any part of a structure at any point. But riveting requires enough clearance.

8. The process of welding takes less time than the riveting.

Disadvantages:-

1. Since there is an uneven heating and cooling during fabrication, therefore the members may get distorted or additional stresses may develop.

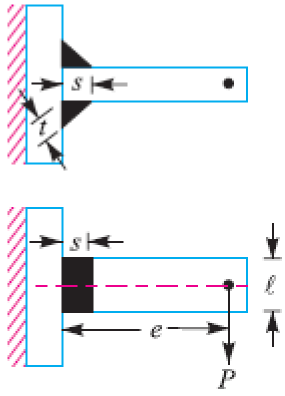
2. It requires a highly skilled labour and supervision.

3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

4. The inspection of welding work is more difficult than riveting work.

2.4 Design of welded joints for eccentric loads: - An eccentric load may be imposed on welded joints in many ways. The stresses induced on the joint may be of different nature or of the same nature. The induced stresses are combined depending upon the nature of stresses. When the shear and bending stresses are simultaneously present in a joint (case 1), then maximum stresses are as follows

CASE-1:-



(Eccentrically loaded welded joint)

Maximum normal stress,

$$\sigma_{t(max)} = \frac{\sigma_b}{2} + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

and maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

where σ_b = Bending stress, and

τ = Shear stress.

Consider a T-joint fixed at one end and subjected to an eccentric load P at a distance e as shown in Figure

Let P = Eccentric load,

e = Eccentricity i.e. perpendicular distance between the line of action of load and centre of gravity (G) of the throat section or fillets,

s = Size of weld, l = Length of weld, and t = Throat thickness.

The joint will be subjected to the following two types of stresses:

1. Direct shear stress due to the shear force P acting at the welds, and

2. Bending stress due to the bending moment P × e.

We know that area at the throat,

A = Throat thickness × Length of weld = t × l × 2

= 2 t × l ... (For double fillet weld) = 2 × 0.707 s × l = 1.414 s × l ... (t = s cos 45° = 0.707 s)

∴ Shear stress in the weld (assuming uniformly distributed),

$$\tau = \frac{P}{A} = \frac{P}{1.414\, s \times l}$$

Section modulus of the weld metal through the throat,

$$Z = \frac{t \times l^2}{6} \times 2 \qquad \text{...(For both sides weld)}$$

$$= \frac{0.707\, s \times l^2}{6} \times 2 = \frac{s \times l^2}{4.242}$$

Bending moment, $M = P \times e$

∴ Bending stress, $\sigma_b = \frac{M}{Z} = \frac{P \times e \times 4.242}{s \times l^2} = \frac{4.242\, P \times e}{s \times l^2}$

We know that the maximum normal stress,

$$\sigma_{t(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

maximum shear stress,

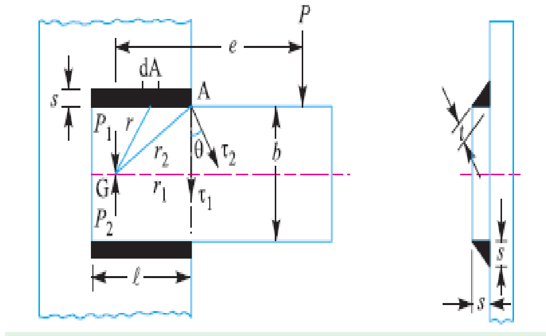
$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

CASE-2:-

When a welded joint is loaded eccentrically as shown in Figure the following two types of the stresses are induced:

1. Direct or primary shear stress, and

2. Shear stress due to turning moment.



Let P = Eccentric load,

e = Eccentricity i.e. perpendicular distance between the line of action of load and centre of gravity (G) of the throat section or fillets,

l = Length of single weld,

s = Size or leg of weld, and

t = Throat thickness.

Let two loads P1 and P2 (each equal to P) are introduced at the centre of gravity 'G' of the weld system. The effect of load P1 = P is to produce direct shear stress which is assumed to be uniform over the entire weld length. The effect of load P2 = P is to produce a turning

We know that the direct or primary shear stress,

$$\tau_1 = \frac{\text{Load}}{\text{Throat area}} = \frac{P}{A} = \frac{P}{2\,t \times l}$$

$$= \frac{P}{2 \times 0.707\, s \times l} = \frac{P}{1.414\, s \times l}$$

... ($\because$ Throat area for single fillet weld $= t \times l = 0.707\, s \times l$)

moment of magnitude P × e which tends of rotate the joint about the centre of gravity 'G' of the weld system. Due to the turning moment, secondary shear stress is induced.

We know that the direct or primary shear stress

Since the shear stress produced due to the turning moment (T = P × e) at any section is proportional to its radial distance from G, therefore stress due to P × e at the point A is proportional to AG (r2) and is in a direction at right angles to AG. Consider a section dA which is at a distance r from G. So stress at dA is proportional to r.

In order to find the resultant stress, the primary and secondary shear stresses are combined

∴ Resultant shear stress at A,

$$\frac{\tau_2}{r_2} = \frac{\tau}{r} = \text{Constant}$$

or
$$\tau = \frac{\tau_2}{r_2} \times r \qquad \text{...}(i)$$

where τ_2 is the shear stress at the maximum distance (r_2) and τ is the shear stress at any distance r.

Consider a small section of the weld having area dA at a distance r from G.

$\therefore$ Shear force on this small section

$$= \tau \times dA$$

and turning moment of this shear force about G,

$$dT = \tau \times dA \times r = \frac{\tau_2}{r_2} \times dA \times r^2 \qquad \text{... [From equation } (i)]$$

$\therefore$ Total turning moment over the whole weld area,

$$T = P \times e = \int \frac{\tau_2}{r_2} \times dA \times r^2 = \frac{\tau_2}{r_2} \int dA \times r^2$$

$$= \frac{\tau_2}{r_2} \times J \qquad \left(\because J = \int dA \times r^2\right)$$

where J = Polar moment of inertia of the throat area about G.

$\therefore$ Shear stress due to the turning moment *i.e.* secondary shear stress,

$$\tau_2 = \frac{T \times r_2}{J} = \frac{P \times e \times r_2}{J}$$

$$\tau_A = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\tau_1 \times \tau_2 \times \cos\theta}$$

where θ = Angle between τ_1 and τ_2, and

$$\cos\theta = r_1 / r_2$$

Q.A welded joint as shown in Figure, is subjected to an eccentric load of 2 kN. Find the size of weld, if the maximum shear stress in the weld is 25 MPa.

Ans.:- Given: P = 2kN = 2000 N; e = 120 mm;

l = 40 mm; τmax = 25 MPa = 25 N/mm^2

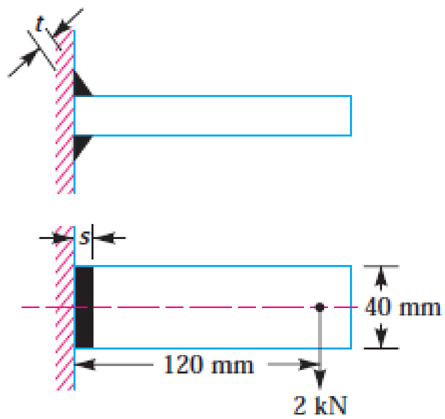
Let s = Size of weld in mm, and

t = Throat thickness.

The joint, as shown in Figure, will be subjected to direct shear stress due to the shear force, P = 2000 N and bending stress due to the bending moment of P × e.

We know that area at the throat,

A = 2t × l = 2 × 0.707 s × l = 1.414 s ×l = 1.414 s × 40 = 56.56 × s mm^2



$\therefore$ Shear stress, $\tau = \dfrac{P}{A} = \dfrac{2000}{56.56 \times s} = \dfrac{35.4}{s}$ N/mm^2

Bending moment, $M = P \times e = 2000 \times 120 = 240 \times 10^3$ N-mm

Section modulus of the weld through the throat,

$$Z = \frac{s \times l^2}{4.242} = \frac{s\,(40)^2}{4.242} = 377 \times s \text{ mm}^3$$

$\therefore$ Bending stress, $\sigma_b = \dfrac{M}{Z} = \dfrac{240 \times 10^3}{377 \times s} = \dfrac{636.6}{s}$ N/mm^2

We know that maximum shear stress (τ_{max}),

$$25 = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\,\tau^2} = \frac{1}{2}\sqrt{\left(\frac{636.6}{s}\right)^2 + 4\left(\frac{35.4}{s}\right)^2} = \frac{320.3}{s}$$

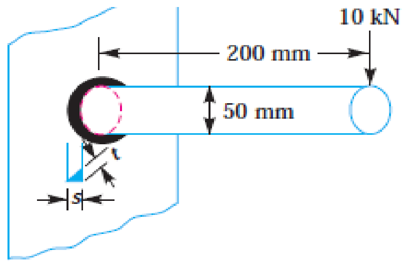
$\therefore$ $s = 320.3 / 25 = 12.8$ mm **Ans.**

Q.A 50 mm diameter solid shaft is welded to a flat plate as shown in Figure If the size of the weld is 15 mm, find the maximum normal and shear stress in the weld.

Ans.:- Given: D = 50 mm; s = 15 mm; P = 10 kN = 10 000 N; e = 200 mm

Let t = Throat thickness.

The joint, as shown in Figure, is subjected to direct shear stress and the bending stress. We know that the throat area for a circular fillet weld,



$$A = t \times \pi D = 0.707\, s \times \pi D$$
$$= 0.707 \times 15 \times \pi \times 50$$
$$= 1666 \text{ mm}^2$$

$\therefore$ Direct shear stress,

$$\tau = \frac{P}{A} = \frac{10\,000}{1666} = 6 \text{ N/mm}^2$$

We know that bending moment,

$$M = P \times e = 10\,000 \times 200 = 2 \times 10^6 \text{ N-mm}$$

From Table 10.7, we find that for a circular section, section modulus,

$$Z = \frac{\pi\, t\, D^2}{4} = \frac{\pi \times 0.707\, s \times D^2}{4} = \frac{\pi \times 0.707 \times 15\,(50)^2}{4} = 20\,825 \text{ mm}^3$$

$\therefore$ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{2 \times 10^6}{20\,825} = 96 \text{ N/mm}^2 = 96 \text{ MPa}$$

(Refer table no. 10.7 in R.S KHURMI book)

We know that maximum normal stress,

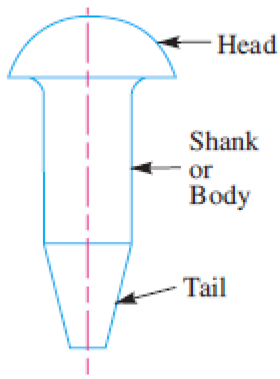
$$\sigma_{t(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} = \frac{1}{2} \times 96 + \frac{1}{2}\sqrt{(96)^2 + 4 \times 6^2}$$
$$= 48 + 48.4 = 96.4 \text{ MPa} \quad \textbf{Ans.}$$

We know that the maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} = \frac{1}{2}\sqrt{(96)^2 + 4 \times 6^2} = 48.4 \text{ MPa}$$ (Ans)

2.5 State types of riveted joints and types of rivets:-

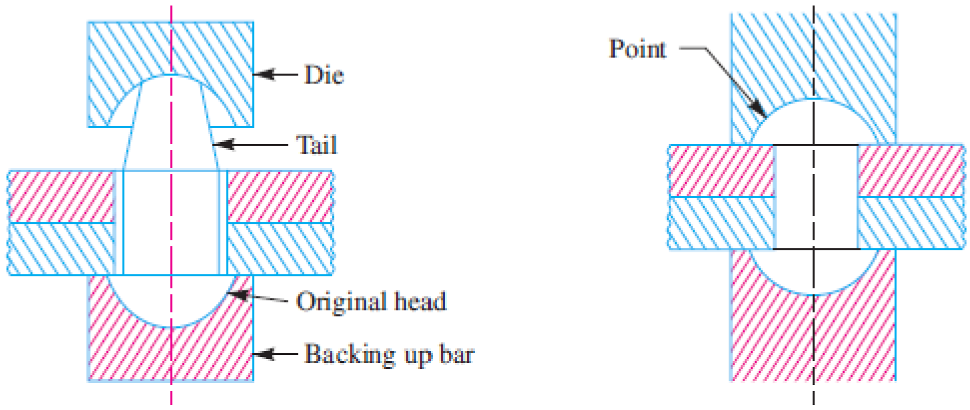
<u>Rivet & Application of rivet:-</u>



- A rivet is a short cylindrical bar with a head integral to it. The cylindrical portion of the rivet is called shank or body and lower portion of shank is known as tail.
- The rivets are used to make permanent fastening between the plates such as in structural work, ship building, bridges, tanks and boiler shells.

METHODS OF RIVETING:-

- The function of rivets in a joint is to make a connection that has strength and tightness. The strength is necessary to prevent failure of the joint. The tightness is necessary in order to prevent leakage.
- The plates are drilled together and then separated to remove any burrs or chips so as to have a tight joint between the plates.
- A cold rivet or a red hot rivet is introduced into the plates and the point (i.e. second head) is then formed. When a cold rivet is used, the process is known as cold riveting and when a hot rivet is used, the process is known as hot riveting.
- The riveting may be done by hand or by a riveting machine. In hand riveting, the original rivet head is backed up by a hammer or heavy bar and then the die or set is placed against the end to be headed and the blows are applied by a hammer. This causes the shank to expand thus filling the hole and the tail is converted into a point.
- In machine riveting, the die is a part of the hammer which is operated by air, hydraulic or steam pressure.



(a) Initial position. (b) Final position.

Material of Rivets:-

The material of the rivets must be tough and ductile. They are usually made of steel (low carbon steel or nickel steel), brass, aluminium or copper.

Types of riveted joints:-

Following are the two types of riveted joints, depending upon the way in which the plates are connected.

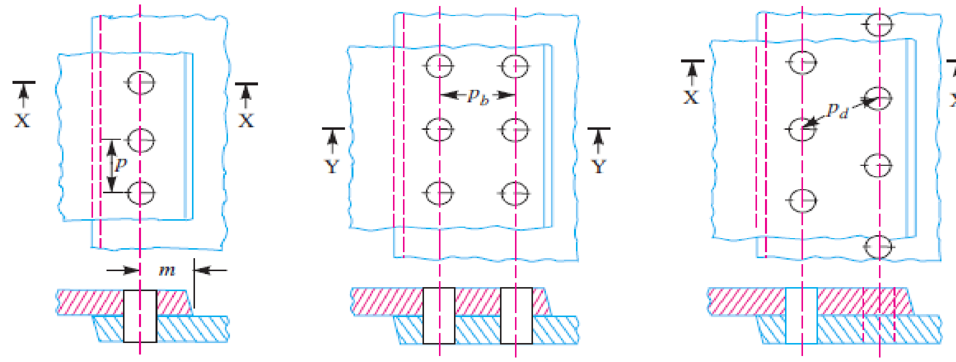
1. Lap joint, and 2. Butt joint.

Lap Joint: - A lap joint is that in which one plate overlaps the other and the two plates are then riveted together.

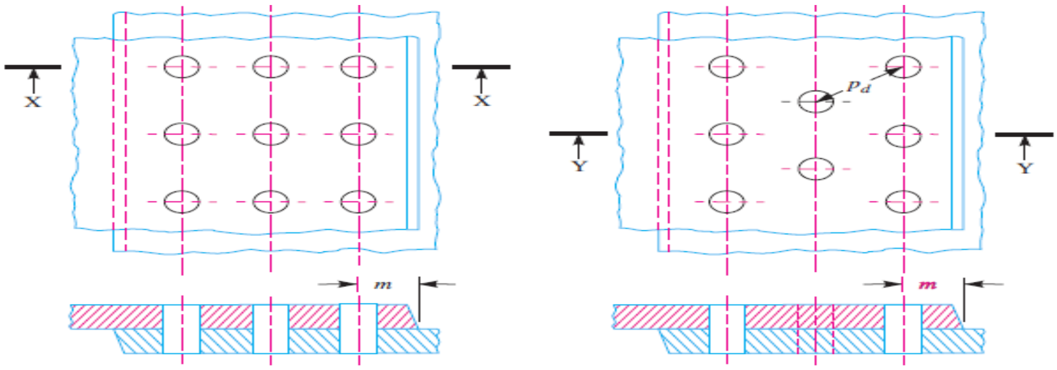
Depending upon the number of rows of the rivets lap joints are of following types:-

1. Single riveted lap joint, and 2. Double riveted lap joint.

- In a single riveted lap joint Plates are held together by one row of rivets.
- A double riveted lap joint is that in which plates are held together by two rows of rivets.
- Joints may be triple riveted or quadruple riveted.
- In double and triple riveted lap joints, the rivets can be arranged either in chain pattern or zig-zag pattern.
- When the rivets in the various rows are opposite to each other, then the joint is said to be chain riveted.
- If the rivets in the adjacent rows are arranged in such a way that every rivet is in the middle of the two rivets of the opposite row then the joint is said to be zig-zag riveted.



(*a*) Single riveted lap joint. (*b*) Double riveted lap joint (Chain riveting). (*c*) Double riveted lap joint (Zig-zag riveting).



(*a*) Chain riveting. (*b*) Zig-zag riveting.

(Triple riveted lap joint)

<u>Butt Joint: -</u> A butt joint is that in which the main plates are kept in alignment butting (i.e. touching) each other and a cover plate (i.e. strap) is placed either on one side or on both sides of the main plates. The cover plate is then riveted together with the main plates.

Following are the types of butt joints depending upon the number of rows of the rivets.

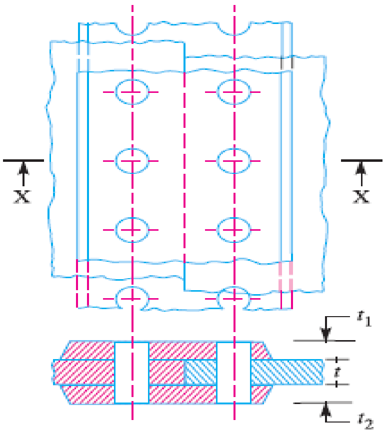
1. Single riveted butt joint and 2. Double riveted butt joint.

- A single riveted butt joint is that in which there is a single row of rivets on each side in a butt joint.
- A double riveted butt joint is that in which there are two rows of rivets on each side in a butt joint.

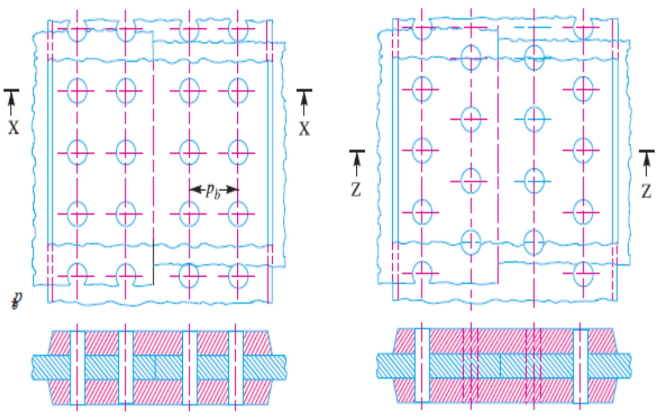
Butt joints are of the following two types on the basis of cover plates (straps):-

1. Single strap butt joint, and 2. Double strap butt joint.

- In a single strap butt joint, the edges of the main plates butt against each other and only one cover plate is placed on one side of the main plates and then riveted together.
- In a double strap butt joint, the edges of the main plates butt against each other and two cover plates are placed on both sides of the main plates and then riveted together.



(Single riveted double strap butt joint)



(*a*) Chain riveting.   (*b*) Zig-zag riveting.

(Double riveted double strap butt joint)

<u>Important Terms Used in Riveted Joints:-</u>

1. Pitch: - It is the distance from the centre of one rivet to the centre of the next rivet in the same row. It is usually denoted by p.

2. Back pitch: - It is the perpendicular distance between the centre lines of the successive rows of rivets. It is usually denoted by Pb. It is also known as transverse pitch or row pitch.

3. **Diagonal pitch**: - It is the distance between the centres of the rivets in adjacent rows of zig-zag riveted joint. It is usually denoted by Pd.

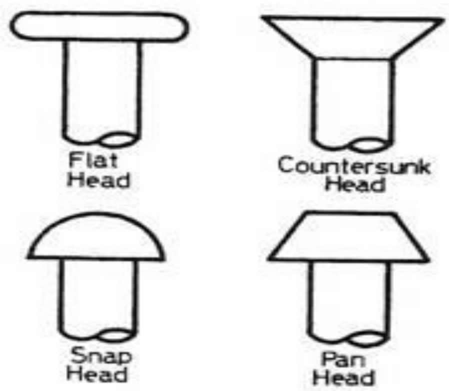
4. Margin or marginal pitch: - It is the distance between the centre of rivet hole to the nearest edge of the plate. It is usually denoted by m.

<u>Types of rivets:-</u>

Rivets are classified on the basis of their heads. Such as:-

Snap head rivet, Pan head rivet, Counter sunk head rivet, Flat head rivet, Half counter sunk head rivet

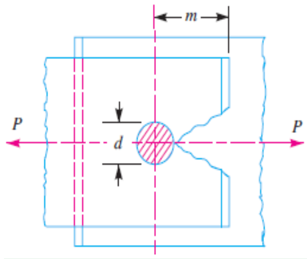
- Snap head or the button head rivets are used in boilers, pressure vessels.



- Pan head rivets are also known as cone head rivets. They are used in boilers & mainly suited for corrosive atmosphere.
- Counter sunk head rivets are used in structural work & ships.
- In flat head rivet the height of the protruding head is less than that of snap head & pan head rivets. They are used in light sheet metal work such as manufacture of buckets, steel boxes.
- Half counter sunk head rivet is the combination of snap head & counter sunk head rivets. It is used for joining steel plates up to 4 mm thickness.

2.6 Describe failure of riveted joints:-

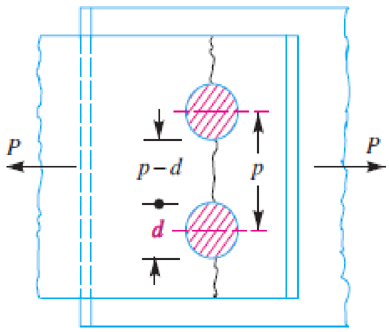
A riveted joint may fail in the following ways**:-**



Tearing of the plate at an edge.

1. Tearing of the plate at an edge:-A joint may fail due to tearing of the plate at an edge as shown in figure. This can be avoided by keeping the margin, m = 1.5d, where d is the diameter of the rivet hole.

2. Tearing of the plate across a row of rivets: - Due to the tensile stresses in the main plates, the main plate or cover plates may tear off across a row of rivets as shown in figure In such cases, we consider only one pitch length of the plate, since every rivet is responsible for that much length of the plate only. The resistance offered by the plate against tearing is known as tearing resistance or tearing strength or tearing value of the plate.



Tearing of the plate across the rows of rivets.

Let p = Pitch of the rivets,

d = Diameter of the rivet hole,

t = Thickness of the plate, and

σt = Permissible tensile stress for the plate material.

We know that tearing area per pitch length,

At = (p – d) t

∴ Tearing resistance or pull required to tear off the plate per pitch length,

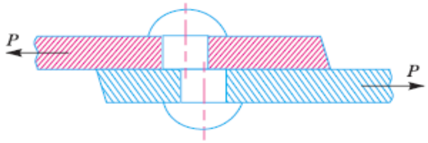
Pt = At.σt = (p – d) t.σt

When the tearing resistance (Pt) is greater than the applied load (P) per pitch length, then this type of failure will not occur.

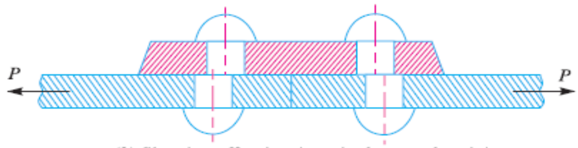
3. Shearing of the rivets: - The plates which are connected by the rivets exert tensile stress on the rivets, and if the rivets are unable to resist the stress, they are sheared off as shown in figure.

It may be noted that the rivets are in **single shear** in a lap joint and in a single cover butt joint. But the rivets are in **double shear** in a double cover butt joint.

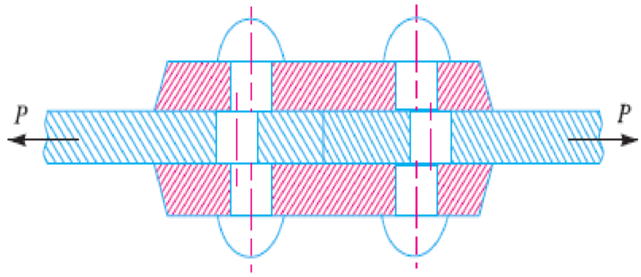
(When the shearing takes place at one cross-section of the rivet, then the rivets are said to be in single shear. Similarly, when the shearing takes place at two cross-sections of the rivet, then the rivets are said to be in double shear.)



(*a*) Shearing off a rivet in a lap joint.



(*b*) Shearing off a rivet in a single cover butt joint.



Shearing off a rivet in double cover butt joint.

The resistance offered by a rivet to be sheared off is known as shearing resistance or shearing strength or shearing value of the rivet.

Let d = Diameter of the rivet hole,

τ = Safe permissible shear stress for the rivet material, and

n = Number of rivets per pitch length.

We know that shearing area,

$$A_s = \frac{\pi}{4} \times d^2 \quad \text{...(In single shear)}$$

$$= 2 \times \frac{\pi}{4} \times d^2 \quad \text{...(Theoretically, in double shear)}$$

$$= 1.875 \times \frac{\pi}{4} \times d^2 \quad \text{...(In double shear, according to Indian Boiler Regulations)}$$

$\therefore$ Shearing resistance or pull required to shear off the rivet per pitch length,

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{...(In single shear)}$$

$$= n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{...(Theoretically, in double shear)}$$

$$= n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{...(In double shear, according to Indian Boiler Regulations)}$$

When the shearing resistance (Ps) is greater than the applied load (P) per pitch length, then this type of failure will occur.

4. Crushing of the plate or rivets:-This type of failure occurs when the compressive stress between the shank of the rivet and the plate exceeds the yield stress in compression. The failure results in elongating the rivet hole in the plate and loosening of the joint. Due to this, the rivet hole becomes of an oval shape.

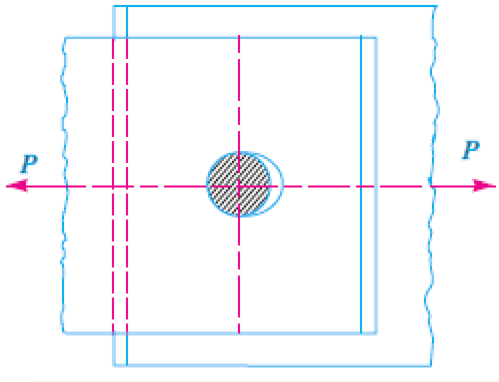
The resistance offered by a rivet to be crushed is known as crushing resistance or crushing strength or bearing value of the rivet.

Let d = Diameter of the rivet hole,

t = Thickness of the plate,

σc = Safe permissible crushing stress for the rivet or plate material, and

n = Number of rivets per pitch length under crushing.



We know that crushing area per rivet:-

Ac = d.t

∴ Total crushing area = n.d.t

And crushing resistance or pull required to crush the rivet per pitch length,

Pc = n.d.t.σc

When the crushing resistance (Pc) is greater than the applied load (P) per pitch length, then this type of failure will occur.

2.7 Determine strength & efficiency of riveted joints:-

Strength of a Riveted Joint:-

The strength of a joint may be defined as the force that the joint can withstand, without causing failure. Pt, Ps and Pc are the pulls required to tear off the plate, shearing off the rivet and crushing off the rivet. If we go on increasing the pull on a riveted joint, it will fail when the least of these three pulls is reached. So strength of the riveted joint is the least of Pt, Ps, and Pc.

Efficiency of a Riveted Joint:-

The efficiency of a riveted joint is defined as the ratio of the strength of riveted joint to the strength of the un-riveted or solid plate.

∴ Efficiency of the riveted joint,

$$\eta = \frac{\text{Least of } P_t,\ P_s \text{ and } P_c}{p \times t \times \sigma_t}$$

where p = Pitch of the rivets,

t = Thickness of the plate, and

σ_t = Permissible tensile stress of the plate material.

Q. A double riveted lap joint is made between 15 mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing, find the minimum force per pitch which will rupture the joint.

If the above joint is subjected to a load such that the factor of safety is 4, find out the actual stresses developed in the plates and the rivets.

Ans.:- Given : t = 15 mm ; d = 25 mm ; p = 75 mm ; σtu = 400 MPa = 400 N/mm2 ;

τu = 320 MPa = 320 N/mm2 ; σcu = 640 MPa = 640 N/mm2

Minimum force per pitch which will rupture the joint:-

Since the ultimate stresses are given, therefore we shall find the ultimate values of the resistances of the joint. We know that ultimate tearing resistance of the plate per pitch,

$$P_{tu} = (p - d)t \times \sigma_{tu} = (75 - 25)15 \times 400 = 300\ 000 \text{ N}$$

Ultimate shearing resistance of the rivets per pitch,

$$P_{su} = n \times \frac{\pi}{4} \times d^2 \times \tau_u = 2 \times \frac{\pi}{4} (25)^2\, 320 = 314\ 200 \text{ N} \quad ...(\because n = 2)$$

and ultimate crushing resistance of the rivets per pitch,

$$P_{cu} = n \times d \times t \times \sigma_{cu} = 2 \times 25 \times 15 \times 640 = 480\ 000 \text{ N}$$

From above we see that the minimum force per pitch which will rupture the joint is 300 000 N or 300 kN.(Ans)

Actual stresses produced in the plates and rivets:-

Since the factor of safety is 4, therefore safe load per pitch length of the joint = 300 000/4 = 75 000 N

Let σta, τa and σca be the actual tearing, shearing and crushing stresses produced with a safe load of 75 000 N in tearing, shearing and crushing.

We know that actual tearing resistance of the plates (P_{ta}),

$$75\ 000 = (p - d)\, t \times \sigma_{ta} = (75 - 25)15 \times \sigma_{ta} = 750\ \sigma_{ta}$$

$$\therefore \quad \sigma_{ta} = 75\ 000 / 750 = 100 \text{ N/mm}^2 = 100 \text{ MPa} \quad \textbf{Ans.}$$

Actual shearing resistance of the rivets (P_{sa}),

$$75\ 000 = n \times \frac{\pi}{4} \times d^2 \times \tau_a = 2 \times \frac{\pi}{4} (25)^2\, \tau_a = 982\ \tau_a$$

$$\therefore \quad \tau_a = 75000 / 982 = 76.4 \text{ N/mm}^2 = 76.4 \text{ MPa} \quad \textbf{Ans.}$$

and actual crushing resistance of the rivets (P_{ca}),

$$75\ 000 = n \times d \times t \times \sigma_{ca} = 2 \times 25 \times 15 \times \sigma_{ca} = 750\ \sigma_{ca}$$

$$\therefore \quad \sigma_{ca} = 75000 / 750 = 100 \text{ N/mm}^2 = 100 \text{ MPa} \quad \textbf{Ans.}$$

Q. Find the efficiency of the following riveted joints:

1. Single riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 50 mm.

2. Double riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 65mm.

Assume

Permissible tensile stress in plate = 120 MPa

Permissible shearing stress in rivets = 90 MPa

Permissible crushing stress in rivets = 180 MPa

Ans.:- Given: t = 6 mm; d = 20 mm; σt = 120 MPa = 120 N/mm2; τ = 90 MPa = 90 N/mm2;

σc = 180 MPa = 180 N/mm2

1. Efficiency of the first joint:-

Pitch (p) =50mm

(i) Tearing resistance of the plate:-

We know that the tearing resistance of the plate per pitch length,

Pt = (p – d) t × σt = (50 – 20) 6 × 120 = 21 600 N

(ii) Shearing resistance of the rivet:-

Since the joint is a single riveted lap joint, therefore the strength of one rivet in single shear is taken.

$$P_s = \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4} (20)^2 \, 90 = 28\,278 \text{ N}$$

(iii) Crushing resistance of the rivet:-

Since the joint is a single riveted, therefore strength of one rivet is taken.

Pc = d × t × σc = 20 × 6 × 180 = 21 600 N

∴ Strength of the joint

= Least of Pt, Ps and Pc = 21 600 N

We know that strength of the unriveted or solid plate,

P = p × t × σt = 50 × 6 × 120 = 36 000 N

∴ Efficiency of the joint:-

$$\eta = \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{21\,600}{36\,000} = 0.60 \text{ or } 60\%$$

2. Efficiency of the second joint:-

Pitch (p) =65mm

(i) Tearing resistance of the plate:-

Pt = (p – d) t × σt = (65 – 20) 6 × 120 = 32 400 N

(ii) Shearing resistance of the rivets:-

Since the joint is double riveted lap joint, therefore strength of two rivets in single shear is taken.

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times \frac{\pi}{4} (20)^2\, 90 = 56\,556 \text{ N}$$

(iii) Crushing resistance of the rivet:-

Since the joint is double riveted, therefore strength of two rivets is taken.

$P_c = n \times d \times t \times \sigma_c = 2 \times 20 \times 6 \times 180 = 43\,200$ N

∴ Strength of the joint = Least of P_t, P_s and P_c = 32 400 N

We know that the strength of the unriveted or solid plate,

$P = p \times t \times \sigma_t = 65 \times 6 \times 120 = 46\,800$ N

∴ Efficiency of the joint:-

$$\eta = \frac{\text{Least of } P_t,\, P_s \text{ and } P_c}{P} = \frac{32\,400}{46\,800} = 0.692 \text{ or } 69.2\%$$

Q. A double riveted double cover butt joint in plates 20 mm thick is made with 25 mm diameter rivets at 100 mm pitch. The permissible stresses are:

σ_t = 120 MPa; τ = 100 MPa; σ_c = 150 MPa

Find the efficiency of joint, taking the strength of the rivet in double shear as twice than that of single shear.

Ans: - Given: t = 20 mm; d = 25 mm; p = 100 mm; σ_t = 120 MPa = 120 N/mm2;

τ = 100 MPa = 100 N/mm2; σ_c = 150 MPa = 150 N/mm2

(i) Tearing resistance of the plate:-

$P_t = (p – d)\, t \times \sigma_t = (100 – 25)\, 20 \times 120 = 180\,000$ N

(ii) Shearing resistance of the rivets:-

Since the joint is double riveted butt joint, therefore the strength of two rivets in double shear is taken.

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times 2 \times \frac{\pi}{4} (25)^2\, 100 = 196\,375 \text{ N}$$

(iii) Crushing resistance of the rivets:-

Since the joint is double riveted, therefore the strength of two rivets is taken.

$P_c = n \times d \times t \times \sigma_c = 2 \times 25 \times 20 \times 150 = 150\,000$ N

∴ Strength of the joint

= Least of Pt, Ps and Pc

= 150 000 N

We know that the strength of the unriveted or solid plate,

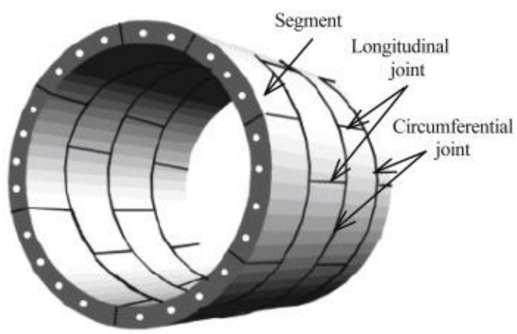
P = p × t × σt = 100 × 20 × 120 = 240 000 N

∴ Efficiency of the joint:-

$$= \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{150\ 000}{240\ 000}$$

= 0.625 or 62.5% (Ans)

2.8 Design riveted joints for pressure vessel:-



- Boiler & pressure vessels are cylindrical vessels & they are subjected to circumferential tensile stress and longitudinal tensile stress.
- The pressure vessel has a longitudinal joint as well as circumferential joint.
- The longitudinal joint is used to join the ends of the plate to get the required diameter of a boiler. For this purpose, a butt joint with two cover plates is used.
- Circumferential joint is used to get the required length of the boiler. For this purpose, a lap joint with one ring overlapping the other alternately is used.

Assumptions in Designing Boiler Joints:-

The following assumptions are made while designing a joint for boilers:-

1. The load on the joint is equally shared by all the rivets.

2. The tensile stress is equally distributed over the section of metal between the rivets.

3. The shearing stress in all the rivets is uniform.

4. The crushing stress is uniform.

5. There is no bending stress in the rivets.

6. The holes into which the rivets are driven do not weaken the member.

7. The rivet fills the hole after it is driven.

8. The friction between the surfaces of the plate is neglected.

Design of Longitudinal Butt Joint for a Boiler:-

1. Thickness of boiler shell:-

$$t = \frac{P.D}{2\,\sigma_t \times \eta_l} + 1 \text{ mm as corrosion allowance}$$

t = Thickness of the boiler shell,

P = Steam pressure in boiler,

D = Internal diameter of boiler shell,

σ_t = Permissible tensile stress, and

η_l = Efficiency of the longitudinal joint.

2. Diameter of rivets:-

After finding out the thickness of the boiler shell (t), the diameter of the rivet hole (d) may be determined by using Unwin's empirical formula, i.e.

$$d = 6\sqrt{t} \qquad \text{(when } t \text{ is greater than 8 mm)}$$

But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the rivets to crushing resistance.

3. Pitch of rivets:-

The pitch of the rivets is obtained by equating the tearing resistance of the plate to the shearing resistance of the rivets.

Pt = (p – d) t × σt = $P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$

Here p is the optimum pitch.

(a) The pitch of the rivets should not be less than 2d.

(b) The maximum value of the pitch of rivets for a longitudinal joint:-

$$p_{max} = C \times t + 41.28 \text{ mm}$$

t = Thickness of the shell plate and

C = Constant.

If optimum pitch is more than maximum pitch then maximum pitch is considered as the pitch of the rivet.

4. Distance between the rows of rivets:-

The distance between the rows of rivets (pb) should not be less than 0.33 p + 0.67 d, for

Zig-zig riveting, and 2 d, for chain riveting.

5. Thickness of butt strap:-

t1 = 0.625 t, for double butt-straps of equal width

For unequal width of butt straps, the thicknesses of butt strap are

t1 = 0.75 t, for wide strap on the inside, and

t2 = 0.625 t, for narrow strap on the outside

6. Margin: - The margin (m) is taken as 1.5 d.

Design of Circumferential Lap Joint for a Boiler:-

1. Thickness of the shell and diameter of rivets:-

$$t = \frac{P.D}{2\,\sigma_t \times \eta_l} + 1 \text{ mm as corrosion allowance}$$

t = Thickness of the boiler shell,

P = Steam pressure in boiler,

D = Internal diameter of boiler shell,

σ_t = Permissible tensile stress, and

η_l = Efficiency of the longitudinal joint.

$$d = 6\sqrt{t} \qquad \text{(when } t \text{ is greater than 8 mm)}$$

But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the rivets to crushing resistance.

3. Pitch of rivets:-

The pitch of the rivets is obtained by equating the tearing resistance of the plate to the shearing resistance of the rivets

$$Pt = (p - d)\, t \times \sigma t = \quad P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$$

The maximum value of the pitch of rivets for a longitudinal joint:-

$$p_{max} = C \times t + 41.28 \text{ mm}$$

t = Thickness of the shell plate and

C = Constant.

If optimum pitch is more than maximum pitch then maximum pitch is considered as the pitch of the rivet.

4. Number of rivets: - Since it is a lap joint, therefore the rivets will be in single shear.

∴ Shearing resistance of the rivets

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

Where n = Total number of rivets.

Knowing the inner diameter of the boiler shell (D), and the pressure of steam (P), the total Shearing load acting on the circumferential joint,

$$W_s = \frac{\pi}{4} \times D^2 \times P$$

Now, Ps = Ws

$$n \times \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4} \times D^2 \times P$$

$$n = \left(\frac{D}{d}\right)^2 \frac{P}{\tau}$$

5. Number of rows: - The number of rows of rivets for the circumferential joint may be obtained from the following relation:-

$$\text{Number of rows} = \frac{\text{Total number of rivets}}{\text{Number of rivets in one row}}$$

And the number of rivets in one row

$$= \frac{\pi (D + t)}{p_1}$$

Where D = Inner diameter of shell.

6. Distance between the rows of rivets:-

The distance between the rows of rivets (pb) should not be less than 0.33 p + 0.67 d, for Zig-zig riveting, and 2 d, for chain riveting.

7. Margin: - The margin (m) is taken as 1.5 d.

8. Overlap = (No. of rows of rivets – 1) pb + m

Where m = Margin.

Q. Design a double riveted butt joint with two cover plates for the longitudinal seam of a boiler shell 1.5 m in diameter subjected to a steam pressure of 0.95 N/mm2. Assume joint efficiency as 75%, allowable tensile stress in the plate 90 MPa; compressive stress 140 MPa; and shear stress in the rivet 56 MPa. (C=3.50).Take double shear is 1.875 times single shear. Consider the rivets are arranged in zig zag manner.

Ans:-Given: D = 1.5 m = 1500 mm; P = 0.95 N/mm2; ηl = 75% = 0.75; σt = 90 MPa

= 90 N/mm2; σc = 140 MPa = 140 N/mm2; τ = 56 MPa = 56 N/mm2; C=3.5

1) Thickness of boiler shell plate:-

$$t = \frac{P.D}{2\sigma_t \times \eta_l} + 1\,\text{mm} = \frac{0.95 \times 1500}{2 \times 90 \times 0.75} + 1 = 11.6 \text{ say } 12\,\text{mm}$$

2) Diameter of rivet:-

Since the thickness of the plate is greater than 8 mm, therefore the diameter of the rivet hole,

$$d = 6\sqrt{t} = 6\sqrt{12} = 20.8\,\text{mm}$$

= 21 mm

3) Pitch of rivets:-

Let p = Pitch of rivets.

The pitch of the rivets is obtained by equating the tearing resistance of the plate to the shearing resistance of the rivets.

Pt = (p – d) t × σt = (p – 21)12 × 90 = 1080 (p – 21) N

$$P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times 1.875 \times \frac{\pi}{4}\,(21)^2 \times 56\,\text{N}$$
$$= 72\,745\,\text{N}$$

Pt = Ps

1080 (p – 21) = 72 745

∴ p – 21 = 72 745 / 1080 = 67.35 or p = 67.35 + 21 = 88.35mm

pmax = C × t + 41.28 mm

From Table 9.5, we find that for a double riveted double strap butt joint and two rivets per pitch length, the value of C is 3.50.

∴ pmax = 3.5 × 12 + 41.28 = 83.28 say 84 mm

Since the value of p is more than pmax, therefore we shall adopt pitch of the rivets,

p = pmax = 84 mm

4. Distance between rows of rivets:-

Assuming zig-zag riveting, the distance between the rows of the rivets

pb = 0.33 p + 0.67 d = 0.33 × 84 + 0.67 × 21 = 41.8 say 42 mm

5. Thickness of cover plates:-

The thickness of each cover plate of equal width is

$t_1 = 0.625\ t = 0.625 \times 12 = 7.5$ mm

6. Margin:-

We know that the margin,

$m = 1.5\ d = 1.5 \times 21 = 31.5$ say 32 mm

Let us now find the efficiency for the designed joint.

Tearing resistance of the plate:-

$P_t = (p - d)\ t \times \sigma_t = (84 - 21)12 \times 90 = 68\ 040$ N

Shearing resistance of the rivets:-

$$P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times 1.875 \times \frac{\pi}{4}\ (21)^2 \times 56 = 72\ 745\ \text{N}$$

Crushing resistance of the rivets,

$P_c = n \times d \times t \times \sigma_c = 2 \times 21 \times 12 \times 140 = 70\ 560$ N

Since the strength of riveted joint is the least value of P_t, P_s or P_c, therefore strength of the riveted joint,

$P_t = 68\ 040$ N

We know that strength of the un-riveted plate,

$P = p \times t \times \sigma_t = 84 \times 12 \times 90 = 90\ 720$ N

∴ Efficiency of the designed joint,

$$\eta = \frac{P_t}{P} = \frac{68\ 040}{90\ 720} = 0.75 \text{ or } 75\%$$

Since the efficiency of the designed joint is equal to the given efficiency of 75%, therefore the design is satisfactory.