R. K. Institute of Engineering and Technology

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LECTURE NOTE

ON

DESIGN OF MACHINE ELEMENTS

(DIPLOMA 5th SEM)

DEPARTMENT OF MECHANICAL ENGINEERING

PREPARED BY: Er. SATYABRATA NAYAK
Lecturer in Department of Mechanical Engineering
RKIET, Niali, Cuttack
Mail ID-: satyaautomech.pupun@gmail.com

R.K. INSTITUTE OF ENGINEERING AND TECHNOLOGY

NIALI, CUTTACK

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Chapter-1
Introduction

Introduction:

Machine Design:

Definition:

Machine Design is the creation of new and better machines and also improving of the existing ones.in Design new or better machine are invented or discovered which is more economical in the overall cost of production and operation.

I. The process of design is a long and time consuming one which includes the study of existing ideas and a new idea has to be conceived.
II. The idea keeping in mind as most economical & to give Proper shape, Size and form of drawings. The Preparation of drawing is done with availability of resources in money, men and Material required for successful completion of new idea in to an actual reality
III. In designing a machine component, it is necessary to have a good knowledge of many subjects such as Mathematics, Engineering Mechanics, Strength of Materials, Theory of Machines, Workshop Processes and Engineering Drawing.

❖ **Types of Design:**

The machine design may be classified as follows:

1. Adaptive design.

- ✓ In most cases, the designer's work is concerned with adaptation of existing designs.
- ✓ This type of design needs no special knowledge or skill and can be attempted by designers of ordinary technical training.
- ✓ The designer only makes minor alternation or modification in the existing designs of the product.

2. Development design.

- ✓ This type of design needs considerable 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 may differ quite markedly from the original product.

3. New design.

- ✓ This type of design needs lot of research, technical ability and creative thinking.
- ✓ Only those designers who have personal qualities of a sufficiently high order can take up the work of a new design.

Depending upon the Methods used it may be classified as following

1. Rational design.

This type of design depends upon mathematical formulae of principle of mechanics.

2. Empirical design.

This type of design depends upon empirical formulae based on the practice and past experience.

3. Industrial design.

This type of design depends upon the production aspects to manufacture any machine component in the industry.

4. Optimum design.

It is the best design for the given objective function under the specified constraints.

It may be achieved by minimizing the undesirable effects.

5. System design.

It is the design of any complex mechanical system like a motor car.

6. Element design.

It is the design of any element of the mechanical system like piston, crankshaft, connecting rod, etc.

7. Computer aided design.

This type of design depends upon the use of computer systems to assist in the creation, modification, analysis and optimization of a design.

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

During design of Machine element the knowledge of material & their properties is a significance for a design Engineer.

The machine elements should be made of such a material which has properties suitable for the conditions of operation. In addition to this, a design engineer must be familiar with the effects which the manufacturing processes and heat treatment have on the properties of the materials.

The engineering Materials are commonly used are classified as

1. Metals and their alloys, such as iron, Steel, Cupper, Aluminium, etc.
2. Non Metals such as Glass, rubber, Plastic, etc.

The Metals may be further classified as

a. Ferrous Metals :

The ferrous metals are those which have the iron as their main constituent, such as cast iron, wrought iron and steel.

a. Non Ferrous Metals :

The non-ferrous metals are those which have a metal other than iron as their main constituent, such as copper, aluminium, brass, tin, zinc, etc.

Selection of Material for Engineering Purpose:

The best material is one which serve the desired objective at the minimum cost.

The following factors should be considered while selecting the material:

- ✓ Availability of the materials,
- ✓ Suitability of the materials for the working conditions in service, and
- ✓ The cost of the materials.

The important properties, which determine the utility of the material are physical, chemical and mechanical properties.

Physical Properties of Metals

The physical properties of the metals include

- Luster
- Colour
- Size and shape
- Density
- Electric and Thermal conductivity
- Melting point.

The following table shows the important physical properties of some pure metals.

Table 2.1. Physical properties of metals.

Metal	*Density (kg/m³)*	*Melting point (°C)*	*Thermal conductivity (W/m°C)*	*Coefficient of linear expansion at 20°C (μm/m/°C)*
Aluminium	2700	660	220	23.0
Brass	8450	950	130	16.7
Bronze	8730	1040	67	17.3
Cast iron	7250	1300	54.5	9.0
Copper	8900	1083	393.5	16.7
Lead	11 400	327	33.5	29.1
Monel metal	8600	1350	25.2	14.0
Nickel	8900	1453	63.2	12.8
Silver	10 500	960	420	18.9
Steel	7850	1510	50.2	11.1
Tin	7400	232	67	21.4
Tungsten	19 300	3410	201	4.5
Zinc	7200	419	113	33.0
Cobalt	8850	1490	69.2	12.4
Molybdenum	10 200	2650	13	4.8
Vanadium	6000	1750	—	7.75

❖ **Mechanical Properties of Metals:**

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. These mechanical properties of the metal include

- ✓ Strength
- ✓ Stiffness
- ✓ Elasticity
- ✓ Plasticity
- ✓ Ductility
- ✓ Brittleness
- ✓ Malleability
- ✓ Toughness
- ✓ Resilience
- ✓ Creep
- ✓ Hardness

1. Strength.

✓ It is the ability of a material to resist the externally applied forces without breaking or yielding.

✓ 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.
- ✓ It may be noted that 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, in stamping images on coins and in ornamental work.

5. Ductility.

- ✓ It is the property of a material enabling it to be drawn into wire with the application of a tensile force. A ductile material must be both strong and plastic.
- ✓ The ductility is usually measured by the terms, percentage elongation and percentage reduction in area.
- ✓ The ductile material commonly used in engineering

6. Brittleness.

- ✓ It is the property of a material opposite to ductility. It is the property of breaking of a material with little permanent distortion.
- ✓ Brittle materials when subjected to tensile loads, snap off without giving any sensible elongation. Cast iron is a brittle material.

8. Malleability.

- ✓ It is a special case of ductility which permits materials to be rolled or hammered into thin sheets.
- ✓ A malleable material should be plastic but it is not essential to be so strong.
- ✓ The malleable materials commonly used in engineering practice (in order of diminishing malleability) are lead, soft steel, wrought iron, copper and aluminium.

9. Toughness.

- ✓ It is the property of a material to resist fracture due to high impact loads like hammer blows.
- ✓ The toughness of the material decreases when it is heated.
- ✓ 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.
- ✓ This property is desirable in parts subjected to shock and impact loads.

10. 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 thrust 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.

11. 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.

12. 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.

13. Fatigue.

✓ When a material is subjected to repeated stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as

✓ *fatigue. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This property is considered in designing shafts, connecting rods, springs, gears, etc.

14. Hardness.

✓ It is a very important property of the metals and has a wide variety of meanings.

✓ It embraces many different properties such as resistance to wear, scratching, deformation and Machinability etc.

✓ It also means the ability of a metal to cut another metal.

✓ The hardness is usually expressed in numbers which are dependent on the method of making the test.

The hardness of a metal may be determined by the following tests:

- Brinell hardness test,
- Rockwell hardness test,
- Vickers hardness (also called Diamond Pyramid) test, and
- Shore scleroscope.

NB:

Steels Designated on the Basis of Chemical Composition

According to Indian standard, IS: 1570 (Part II/Sec I)-1979 (Reaffirmed 1991), the carbon steels are designated in the following order:

(a) Figure indicating 100 times the average percentage of carbon content,

(b) **Letter 'C',** and

(c) Figure indicating 10 times the average percentage of manganese content. The figure after multiplying shall be rounded off to the nearest integer.

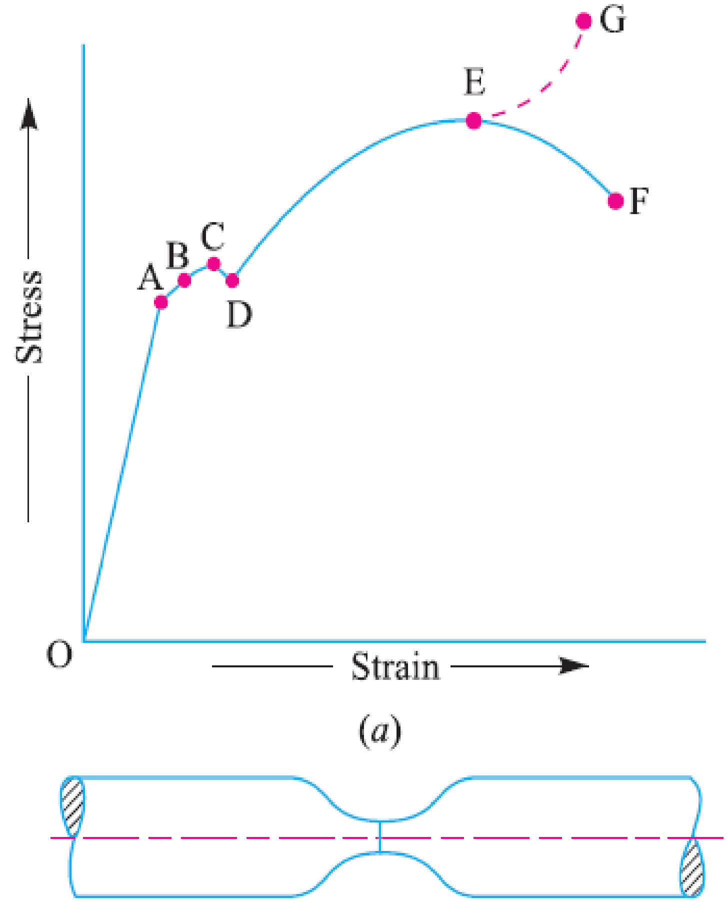
Table 2.6. Indian standard designation of carbon steel according to IS : 1570 (Part II/Sec 1) – 1979 (Reaffirmed 1991).

Indian standard designation	*Composition in percentages*		*Uses as per IS : 1871 (Part II)–1987 (Reaffirmed 1993)*
	Carbon (C)	*Manganese (Mn)*	
4C2	0.08 Max.	0.40 Max.	It is a dead soft steel generally used in electrical industry.
5C4 7C4 10C4	0.10 Max. 0.12 Max. 0.15 Max.	0.50 Max. 0.50 Max. 0.30 – 0.60	These steels are used where cold formability is the primary requirement. In the rimming quality, they are used as sheet, strip, rod and wire especially where excellent surface finish or good drawing qualities are required, such as automobile body, and fender stock, hoods, lamps, oil pans and a multiple of deep drawn and formed products. They are also used for cold heading wire and rivets and low carbon wire products. The killed steel is used for forging and heat treating applications.
10C4 14C6	0.15 Max. 0.10 – 0.18	0.30 – 0.60 0.40 – 0.70	The case hardening steels are used for making camshafts, cams, light duty gears, worms, gudgeon pins, spindles, pawls, ratchets, chain wheels, tappets, etc.
15C4	0.20 Max.	0.30 – 0.60	It is used for lightly stressed parts. The material, although easily machinable, is not designed specifically for rapid cutting, but is suitable where cold web, such as bending and riveting may be necessary.

For example **20C8** means a carbon steel containing 0.15 to 0.25 per cent (0.2 per cent on an average) **carbon** and 0.60 to 0.90 per cent (0.75 per cent rounded off to 0.8 per cent on an average) **manganese.**

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

- ✓ In designing various parts of a machine,
- ✓ it is necessary to know how the material will function in service. For this, certain characteristics or properties of the material should be known.
- ✓ The mechanical properties mostly used in mechanical engineering practice are commonly determined from a standard tensile test.
- ✓ This test consists of gradually loading a standard specimen of a material and noting the corresponding values of load and elongation until the specimen fractures.
- ✓ The load is applied and measured by a testing machine. The stress is determined by dividing the load values by the original cross-sectional area of the specimen.
- ✓ The elongation is measured by determining the amounts that two reference points on the specimen are moved apart by the action of the machine.
- ✓ The original distance between the two reference points is known as **gauge length.**
- ✓ The strain is determined by dividing the elongation values by the gauge length.
- ✓ The values of the stress and corresponding strain are used to draw the stress-strain diagram of the material tested.



(*a*)

(*b*) Shape of specimen after elongation.

Stress- Strain Diagram for Ductile Material

- ✓ A stress-strain diagram for a mild steel under tensile test is shown in Fig. The various properties of the material are discussed below:

❖ **Modes of Failure (By elastic deflection, general yielding & fracture)**

1. **Proportional limit.**

- ✓ We see from the diagram that from point O to A is a straight line, which represents that the stress is proportional to strain. Beyond point A, the curve slightly deviates from the straight line.
- ✓ It is thus obvious, that Hooke's law holds good up to point A and it is known as proportional limit.
- ✓ It is defined as that stress at which the stress-strain curve begins to deviate from the straight line.

2. **Elastic limit.**

- ✓ It may be noted that even if the load is increased beyond point A upto the point B, the material will regain its shape and size when the load is removed.

- ✓ This means that the material has elastic properties up to the point B. This point is known as **elastic limit.**

Definition:

- ✓ It is defined as the stress developed in the material without any permanent set.

Note: Since the above two limits are very close to each other, therefore, for all practical purposes these are taken to be equal.

3. Yield point.

- ✓ If the material is stressed beyond point B, the plastic stage will reach i.e. on the removal of the load, the material will not be able to recover its original size and shape. A little consideration will show that beyond point B, the strain increases at a faster rate with any increase in the stress until the point C is reached.
- ✓ At this point, the material yields before the load and there is an appreciable strain without any increase in stress. In case of mild steel, it will be seen that a small load drops to D, immediately after yielding commences.
- ✓ Hence there are two yield points C and D. The points C and D are called the upper and lower yield points respectively. The stress corresponding to yield point is known as yield point stress.

4. Ultimate stress.

- ✓ At D, the specimen regains some strength and higher values of stresses are required for higher strains, than those between A and D. The stress (or load) goes on increasing till the point E is reached.
- ✓ The gradual increase in the strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area.
- ✓ The work done, during stretching the specimen, is transformed largely into heat and the specimen becomes hot. At E, the stress, which attains its maximum value is known as ultimate stress.

Definition:

- ✓ It is defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

5. Breaking stress.

- ✓ After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen, as shown in Fig. A little consideration will show that the stress (or load) necessary to break away the specimen, is less than the maximum stress.
- ✓ The stress is, therefore, reduced until the specimen breaks away at point F. The stress corresponding to point F is known as breaking stress.

Note:

- ✓ The breaking stress (i.e. stress at F which is less than at E) appears to be somewhat misleading. As the formation of a neck takes place at E which reduces the cross-sectional area, it causes the specimen suddenly to fail at F.
- ✓ If for each value of the strain between E and F, the tensile load is divided by the reduced cross-sectional area at the narrowest part of the neck, then the true stress-strain curve will follow the dotted line EG.
- ✓ However, it is an established practice, to calculate strains on the basis of original cross-sectional area of the specimen.

Stress –Strain Curve for C.I.:

I. Proportional limit.

- ✓ We see from the diagram that from point O to A is a straight line, which represents that the stress is proportional to strain. Beyond point A, the curve slightly deviates from the straight line.

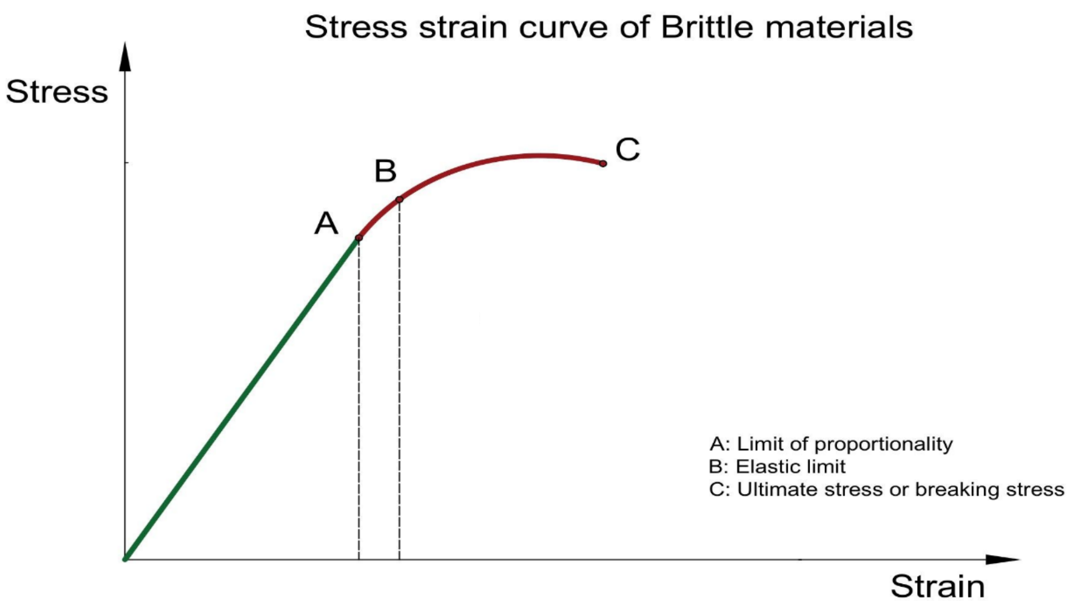
- ✓ It is thus obvious, that Hooke's law holds good up to point A and it is known as proportional limit.
- ✓ It is defined as that stress at which the stress-strain curve begins to deviate from the straight line.

II. Elastic limit.

- ✓ It may be noted that even if the load is increased beyond point A upto the point B, the material will regain its shape and size when the load is removed.
- ✓ This means that the material has elastic properties up to the point B. This point is known as **elastic limit.**

III. Ultimate Point:

After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen, as shown in Fig. A little consideration will show that the stress (or load) necessary to break away the specimen, is less than the maximum stress.



❖ **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 satisfactory.

❖ **Factor of Safety**

It is defined, in general, as the ratio of the maximum stress to the working stress**.** Mathematically,
Working or design stress

$$\textbf{Factor of Safety} = \frac{\textbf{Maximum Stress}}{\textbf{Working or Design Stress}}$$

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,

$$\textbf{Factor of Safety} = \frac{\textbf{Yield Stress}}{\textbf{Working or Design Stress}}$$

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.

$$\textbf{Factor of Safety} = \frac{\textbf{Ultimate Stress}}{\textbf{Working or Design Stress}}$$

This relation may also be used for ductile materials.

Note: The above relations for factor of safety are for static loading.

Selection of Factor of Safety

The selection of a proper factor of safety to be used in designing any machine component depends upon a number of considerations, such as the material, mode of manufacture, type of stress, general service conditions and shape of the parts. Before selecting a proper factor of safety, a design engineer should consider the following points:

1. The reliability of the properties of the material and change of these properties during service;
2. The reliability of test results and accuracy of application of these results to actual machine parts;
3. The reliability of applied load;
4. The certainty as to exact mode of failure;
5. The extent of simplifying assumptions;
6. The extent of localized stresses;
7. The extent of initial stresses set up during manufacture;
8. The extent of loss of life if failure occurs; and
9. The extent of loss of property if failure occurs.

Each of the above factors must be carefully considered and evaluated. The high factor of safety results in unnecessary risk of failure. The values of factor of safety based on ultimate strength for different materials and type of load are given in the following table:

Material	*Steady load*	*Live load*	*Shock load*
Cast iron	5 to 6	8 to 12	16 to 20
Wrought iron	4	7	10 to 15
Steel	4	8	12 to 16
Soft materials and alloys	6	9	15
Leather	9	12	15
Timber	7	10 to 15	20

❖ **Factors Governing the Design of Machine Elements:**

Following are the general considerations in designing a machine component:

1. **Type of load and stresses caused by the load.**

   The load, on a machine component, may act in several ways due to which the internal stresses are set up.

   1. Dead Load/ Steady Load
   2. Live Load/Variable Load
   3. Impact Load
   4. Suddenly Applied or Shock Load

2. **Motion of the parts or kinematics of the machine.**

The successful operation of any machine depends largely upon the simplest arrangement of the parts which will give the motion required.

The motion of the parts may be:
(a) Rectilinear motion which includes unidirectional and reciprocating motions.
(b) Curvilinear motion which includes rotary, oscillatory and simple harmonic.
(c) Constant velocity.
(d) Constant or variable acceleration

3. Selection of materials.

- ✓ It is essential that a designer should have a thorough knowledge of the properties of the materials and their behaviour under working conditions.
- ✓ Some of the important characteristics of materials are: strength, durability, flexibility, weight, resistance to heat and corrosion, ability to cast, welded or hardened, machinability, electrical conductivity, etc.

4. Form and size of the parts.

- ✓ The form and size are based on judgment. The smallest practicable cross-section may be used, but it may be checked that the stresses induced in the designed cross-section are reasonably safe.
- ✓ In order to design any machine part for form and size, it is necessary to know the forces which the part must sustain.
- ✓ It is also important to anticipate any suddenly applied or impact load which may cause failure.

5. Frictional resistance and lubrication.

- ✓ There is always a loss of power due to frictional resistance and it should be noted that the friction of starting is higher than that of running friction.
- ✓ It is, therefore, essential that a careful attention must be given to the matter of lubrication of all surfaces which move in contact with others, whether in rotating, sliding, or rolling bearings.

6. Convenient and economical features.

- ✓ In designing, the operating features of the machine should be carefully studied. The starting, controlling and stopping levers should be located on the basis of convenient handling.
- ✓ The adjustment for wear must be provided employing the various take up devices and arranging them so that the alignment of parts is preserved.
- ✓ If parts are to be changed for different products or replaced on account of wear or breakage, easy access should be provided and the necessity of removing other parts to accomplish this should be avoided if possible.
- ✓ The economical operation of a machine which is to be used for production, or for the processing of material should be studied, in order to learn whether it has the maximum capacity consistent with the production of good work.

7. Use of standard parts.

- ✓ The use of standard parts is closely related to cost, because the cost of standard or stock parts is only a fraction of the cost of similar parts made to order.
- ✓ The standard or stock parts should be used whenever possible; parts for which patterns are already in existence such as gears, pulleys and bearings and parts which may be selected from regular shop stock such as screws, nuts and pins. Bolts and studs should be as few as possible to avoid the delay caused by changing drills, reamers and taps and also to decrease the number of wrenches required.

8. Safety of operation.

- ✓ Some machines are dangerous to operate, especially those which are speeded up to insure production at a maximum rate. Therefore, any moving part of a machine which is within the zone of a worker is considered an accident hazard and may be the cause of an injury.

- ✓ It is, therefore, necessary that a designer should always provide safety devices for the safety of the operator.
- ✓ The safety appliances should in no way interfere with operation of the machine.

9. Workshop facilities.

- ✓ A design engineer should be familiar with the limitations of his employer's workshop, in order to avoid the necessity of having work done in some other workshop.
- ✓ It is sometimes necessary to plan and supervise the workshop operations and to draft methods for casting, handling and machining special parts.

10. Number of machines to be manufactured.

- ✓ The number of articles or machines to be manufactured affects the design in a number of ways.
- ✓ The engineering and shop costs which are called fixed charges or overhead expenses are distributed over the number of articles to be manufactured.
- ✓ If only a few articles are to be made, extra expenses are not justified unless the machine is large or of some special design.
- ✓ An order calling for small number of the product will not permit any undue expense in the workshop processes, so that the designer should restrict his specification to standard parts as much as possible.
- ✓

11. Cost of construction.

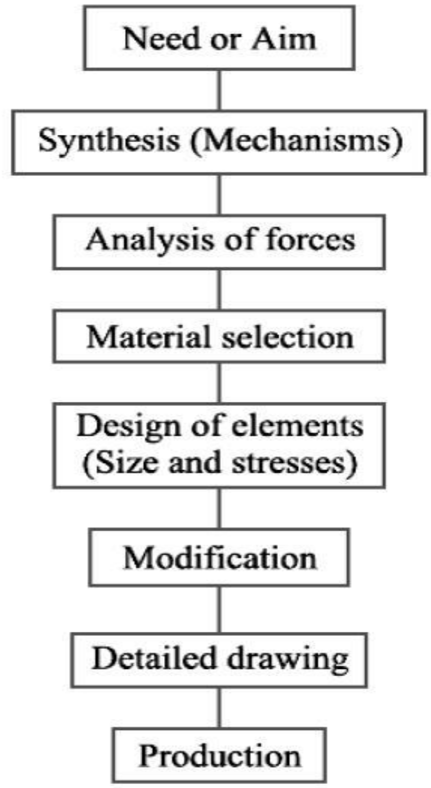
- ✓ The cost of construction of an article is the most important consideration involved in design. In some cases, it is quite possible that the high cost of an article may immediately bar it from further considerations.
- ✓ If an article has been invented and tests of handmade samples have shown that it has commercial value, it is then possible to justify the expenditure of a considerable sum of money in the design and development of automatic machines to produce the article, especially if it can be sold in large numbers.
- ✓ The aim of design engineer under all conditions, should be to reduce the manufacturing cost to the minimum.

12. Assembling.

- ✓ Every machine or structure must be assembled as a unit before it can function.
- ✓ Large units must often be assembled in the shop, tested and then taken to be transported to their place of service.
- ✓ The final location of any machine is important and the design engineer must anticipate the exact location and the local facilities for erection.

❖ **General Procedure in Machine Design:**

In designing a machine component, there is no rigid rule.
The problem may be attempted in several ways. However, the general procedure to solve a design problem is as follows:



1. Recognition of need.
First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.

2. Synthesis (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 (Size and Stresses).
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 than the permissible limit.

6. Modification.
Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing 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 suggested.

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.

Design of Fastening Element
Chapter - 2

Fastening

Classification:

1. Permanent Fastening
2. Temporary Fastening

Permanent Fastening

- ✓ The permanent fastenings are those fastenings which cannot be disassembled without destroying the connecting components.
- ✓ The examples of permanent fastenings in order of strength are soldered, brazed, welded and riveted joints.

Temporary Fastening

- ✓ The temporary or detachable fastenings are those fastenings which can be disassembled without destroying the connecting components.
- ✓ The examples of temporary fastenings are screwed, keys, cotters, pins and splined joints.

Welding Joint:

- ✓ 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.
- ✓ 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).
- ✓ Welding is extensively used in fabrication as an alternative method for casting or forging and as a replacement for bolted and riveted joints.
- ✓ It is also used as a repair medium *e.g*. to reunite metal at a crack, to build up a small part that has broken off such as gear tooth or to repair a worn surface such as a bearing surface.

Advantages and Disadvantages of Welded Joints over Riveted Joints

Following are the advantages and disadvantages of welded joints over riveted joints.

Advantages:

1. The welded structures are usually lighter than riveted structures. This is due to the reason, that in welding, gussets or other connecting components are not used.

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. In welded connections, the tension members are not weakened as in the case of riveted joints.

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

7. 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.

8. The welding provides very rigid joints. This is in line with the modern trend of providing rigid frames.

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

10. 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.

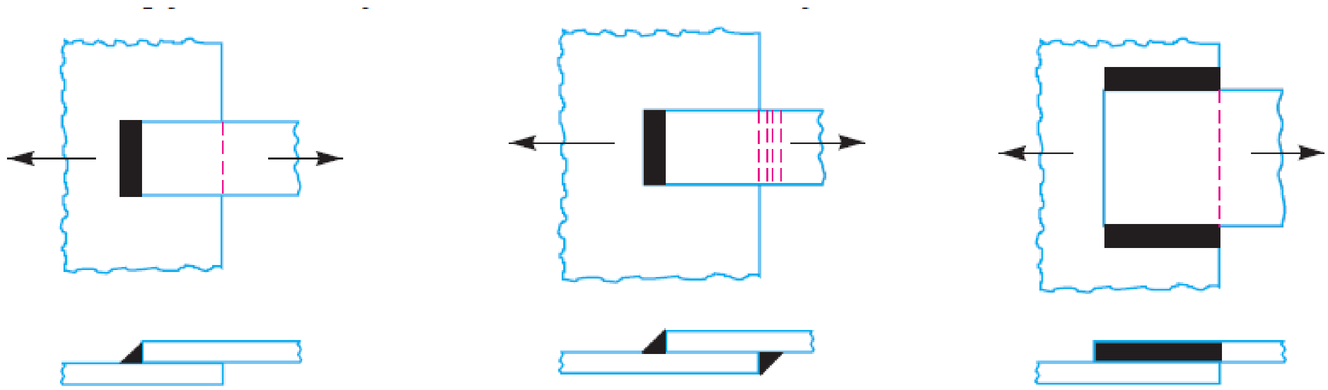
3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

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

Types of Welding Joint:

Following two types of welded joints are important from the subject point of view:

1. Lap joint or fillet joint, and **2.** Butt joint.



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

Fig. 10.2. Types of lap or fillet joints.

Lap Joint:

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

1. Single transverse fillet, **2.** Double transverse fillet, and **3.** Parallel fillet joints.

The fillet joints are shown in Fig. 10.2. A single transverse fillet joint has the disadvantage that the edge of the plate which is not welded can buckle or warp out of shape.

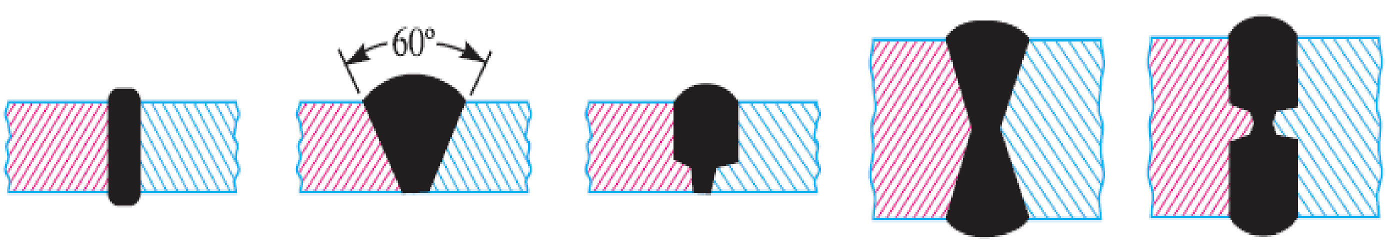
Butt Joint

The butt joint is obtained by placing the plates edge to edge as shown in Fig. 10.3. In butt welds, the plate edges do not require bevelling if the thickness of plate is less than 5 mm. 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.

The butt joints may be

1. Square butt joint, **2.** Single V-butt joint **3.** Single U-butt joint,

4. Double V-butt joint, and **5.** Double U-butt joint. These joints are shown in Fig. 10.3.



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

Fig. 10.3. Types of butt joints.

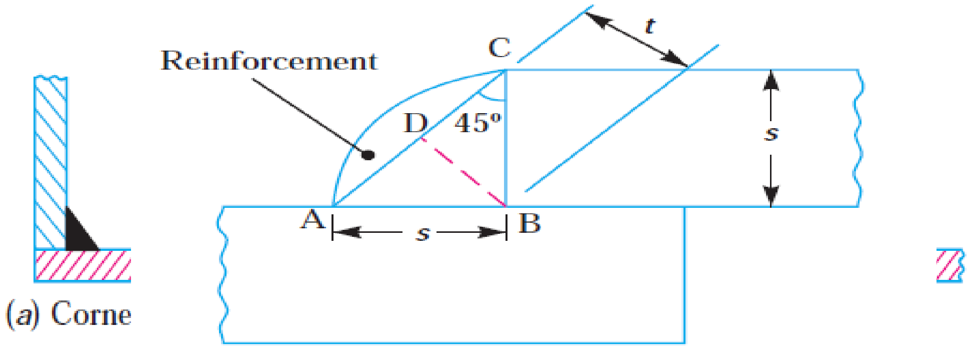


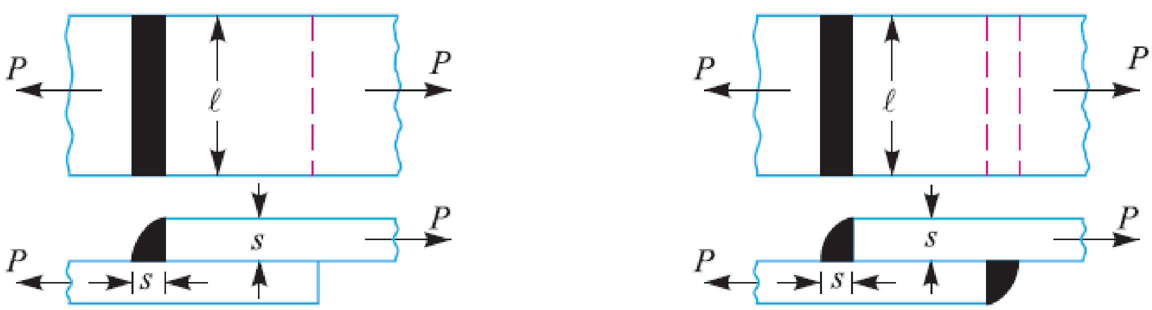
Fig. 10.7. Enlarged view of a fillet weld.

The other type of welded joints are corner joint, edge joint and T-joint as shown in Fig. 10.4.
The main considerations involved in the selection of weld type are:

1. The shape of the welded component required,
2. The thickness of the plates to be welded, and
3. The direction of the forces applied.

Strength of Transverse Fillet Welded Joints:

We have the fillet or lap joint is obtained by overlapping the plates and then welding the edges of the plates.
The transverse fillet welds are designed for tensile strength.
Let us consider a single and double transverse fillet welds as shown in Fig. 10.6 (*a*) and (*b*) respectively.



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

Fig. 10.6. Transverse fillet welds.

- ✓ 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 Fig. 10.7.
- ✓ 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 obtained at the throat BD, which is given by the product of the throat thickness and length of weld.

The minimum area of weld is obtained at the throat BD

Area of Throat

$$A = l \times t$$

From Δ ABD we have

$$\sin\theta = \frac{P}{h} = \frac{BD}{AB} = \frac{t}{s}$$

$$\Rightarrow \mathbf{t = S \times \sin 45 = 0.707S}$$

Now Area of Single Fillet

$$A = t \times l = 0.707S \times l$$

Area of Double Fillet

$$A = 2t \times l = 2 \times 0.707S \times l = \mathbf{1.414\,S \times \boldsymbol{l}}$$

Strength of Welding Joint is calculated as the tensile load acting on single transverse Joint is given by

σ_t − Tensile stress acting on plates

Tensile Load acting on Transverse Joint

$$\mathbf{P_{\boldsymbol{t}} = \sigma_t \times A = \sigma_t \times 0.707S \times \boldsymbol{l} \ldots\ldots\ldots\ldots\ldots Single\ Transverse}$$

$$\mathbf{P_{\boldsymbol{t}} = \sigma_t \times A = \sigma_t \times 1.414S \times \boldsymbol{l} \ldots\ldots\ldots\ldots\ldots Double\ Transverse}$$

Strength of Parallel Fillet Welded Joints: Strength of Parallel Fillet Weld Joint as shown in Fig. below

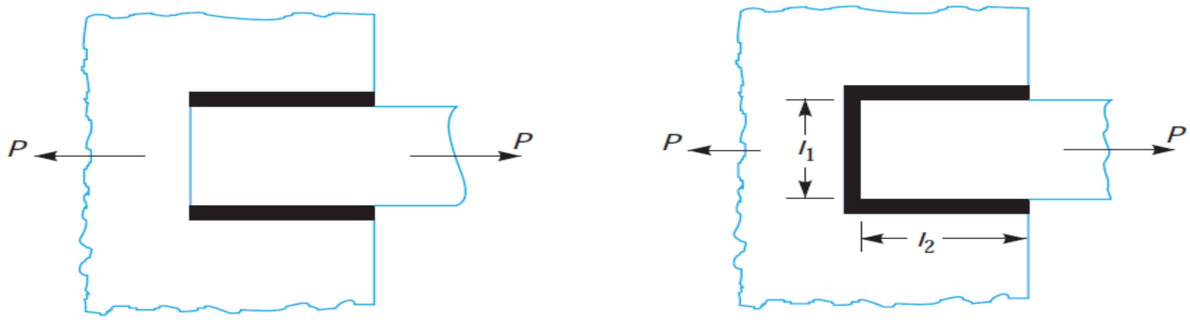
We have to calculate the minimum area of weld or the throat area (i.e. Maximum stress is induced at minimum area)

Now Area of Single Fillet

$$A = t \times l = 0.707S \times l$$

Area of Double Fillet

$$A = 2t \times l = 2 \times 0.707S \times l = \mathbf{1.414\,S \times l}$$



(*a*) Double parallel fillet weld.

(*b*) Combination of transverse and parallel fillet weld.

Fig. 10.8

In Parallel fillet welding joint Shear Stress is induced in joint

Let

l = length of Parallel fillet weld

t = thickness of throat

P – shear load acting on weld

τ = Shear Stress induced on joint

Shear strength of weld

$$P_s = \tau \times A = \tau \times t \times l$$

$$\mathbf{P_s = \tau \times 0.707S \times l \ldots\ldots\ldots\ldots\ldots\ldots\ldots..\text{Single Parallel Fillet weld}}$$

$$\mathbf{P_s = \tau \times 1.414S \times l \ldots\ldots\ldots\ldots\ldots\ldots\ldots..\text{Double Parallel Fillet weld}}$$

Notes: 1. if there is a combination of single transverse and double parallel fillet welds as shown in Fig. 10.8 (*b*), then the strength of the joint is given by the sum of strengths of single transverse and double parallel fillet welds.

Mathematically,

Strength of weld joint is the algebraic sum of strength of parallel & transverse joint

$$P = P_t + P_s = \boldsymbol{\sigma_t \times 0.707S \times l_1 + \tau \times 1.414S \times l_2}$$

2. In order to allow for starting and stopping of the bead, 12.5 mm should be added to the length of each weld obtained by the above expression. $\mathbf{(l = l + 12.5)}$

3. For reinforced fillet welds, the throat dimension may be taken as 0.85 *t*.

Eccentrically Loaded Welded Joints

✓ 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 (see case 1), then maximum stresses are as follows:
Let's consider a welded 'T' joint as shown in diagram
Let
$\mathrm{P} = \text{eccentric load acting on joint}$
$\mathrm{e} = \text{eccentricity of Load}$
$\mathrm{S} = \text{Size of Weld}$
$l = \text{Length of Weld}$
$\mathrm{t} = \text{Thickness of weld}$

Case-1:
'T' Joint fixed at one end subjected to eccentric load as shown in fig.
During calculation of strength of weld joint for this case stresses developed in joint are as

1. Direct shear stress due to shear force p acting at the throat (Parallel fillet Welding)
2. Bending stress due to eccentric load by Moment produced by it

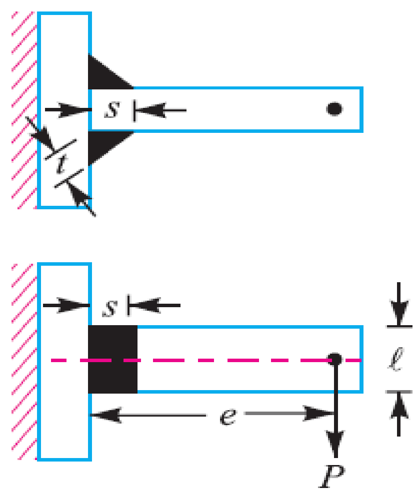


Fig. 10.22. Eccentrically loaded welded joint.

$\mathrm{P}_s - \text{shear load acting on weld}$
$\tau = \text{Shear Stress induced on joint}$

Shear stress on weld:

$$\mathrm{P_s} = \tau \times \mathrm{A} = \tau \times \mathrm{t} \times l$$

$$\mathrm{P_s} = \tau \times 0.707\mathrm{S} \times l \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{Single Parallel Fillet weld}$$

$$\mathrm{P_s} = \tau \times 1.414\mathrm{S} \times l \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{Double Parallel Fillet weld}$$

Now shear stress acting on weld

$$\boldsymbol{\tau} = \frac{\mathbf{P_s}}{\mathbf{1.414S} \times \boldsymbol{l}} \ldots\ldots\ldots\ldots\ldots\ldots (i) \textit{Direct Shear Stress on Weld}$$

Bending Stress on Weld:

The Weld joint is subjected to Turning moment by the action of eccentric load 'P'
From Bending Equation

$$\frac{\mathrm{M}}{\mathrm{I}} = \frac{\sigma_\mathrm{b}}{\mathrm{y}}$$

$$\Rightarrow \sigma_\mathrm{b} = \frac{\mathrm{M}}{\mathrm{Z}}$$

Section Modulus of weld joint for Single Fillet Joint

$$\mathrm{Z} = \frac{\mathrm{t} \times l^2}{6} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{For Single Fillet Joint}$$

$$\mathrm{Z} = \frac{2 \times \mathrm{t} \times l^2}{6} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{For Double Fillet Joint}$$

Moment produced by eccentric load

$$M = P \times e$$

Bending Stress induced in joint

$$\Rightarrow \sigma_b = \frac{M}{Z} = \frac{P \times e}{\frac{2 \times t \times l^2}{6}}$$

$$\boldsymbol{\sigma_b = \frac{6 \times P \times e}{1.414 \times S \times l^2}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(ii)Bending stress due to eccentric load}$$

We have two stresses are acting on joint for this we have to consider the combined effect of both Stresses by Principal stress theory
According to Maximum Shear stress Theory

1. $$\boldsymbol{\tau_{Max} = \frac{1}{2}\sqrt{(\sigma_b{}^2 + 4\tau^2)}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \textbf{Maximum Shear Stress}$$
2. $$\boldsymbol{\sigma_{n_{Max}} = \frac{\sigma_b}{2} + \frac{1}{2}\sqrt{(\sigma_b{}^2 + 4\tau^2)}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \textbf{Maximum Normal Stress}$$

Case-2: Double Parallel Fillet 'T' Joint is subjected to eccentric load

From Fig of Double Parallel Fillet 'T' Joint is subjected to eccentric load at an eccentricity of 'e' from C.G. of the Weld at Point.

It is observed that due to eccentric load on Welding joint as shown in fig. there are two following types of stresses are induced in it.

1. Direct shear stress or Primary Shear stress (Parallel fillet Weld)
2. Torsional shear stress due to turning moment produced by eccentric load.

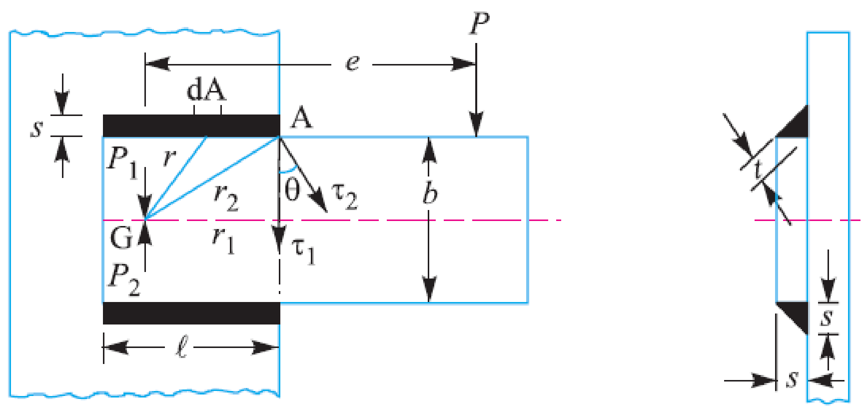


Fig. 10.23. Eccentrically loaded welded joint.

❖ Direct shear stress or Primary Shear stress (Parallel fillet Weld):
We have for Parallel fillet welding
Area of throat

$$A = 2 \times t \times l = 1.414S \times l$$

Shear Stress induced in welded joint

$$\boldsymbol{\tau_1 = \frac{P}{1.414S \times l}}$$

Generally the shear stress produced is assumed to be uniform over the entire length of weld

$$\boldsymbol{\tau_1} = \frac{\mathbf{P}}{\mathbf{1.414S} \times \boldsymbol{l}}$$

❖ Torsional Shear Stress due to eccentric load 'P'

We know that due to application of eccentric load. Due to this it tends to rotate the joint about center of gravity 'G' of Weld system.

Due to the turning moment shear stress 'τ_2' is produced

$$T = P \times e \ldots \ldots \ldots \ldots \ldots \ldots (i)$$

Shear stress produce due to turning moment at any section is proportional to its radial distance from 'G' (Concept of Polar Modulus)

From torsion equation shear stress is uniform throughout the length of weld.

$$\frac{\tau}{R} = \frac{\tau_2}{r_2}$$

For a layer of fillet weld at a distance of 'r' equivalent shear stress

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

$r -$ radius of small strip 'dA' from C. G

$\tau -$ Shear stress induced in small strip

$\tau_2 -$ Torsional shear stress at point 'A'

$r_2 -$ Distance of torsional shear stress from C. G.

Considering small strip at radius 'r' from 'G' point

Shear force on small strip

$$df = \tau \times dA$$

Turning moment produced by force

$$dT = df \times r$$

$\therefore$ Total turning moment produced on whole weld

$$T = \int dT = \int df \times r$$

$$= \int \tau \times dA \times r$$

$$= \int \frac{\tau_2}{r_2} dA.r.r$$

$$T = \frac{\tau_2}{r_2} \int r^2 \, dA$$

$$T = \frac{\tau_2}{r_2} \times J \ldots \ldots \ldots \ldots \ldots \ldots \ldots (ii)$$

Equating equation (i) and (ii)

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

$$\Rightarrow \boldsymbol{\tau_2} = \frac{\mathbf{P \times e}}{\mathbf{J}} \times \mathbf{r_2} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \textbf{Torsional Shear Stress}$$

Now the resultant shear stress produced in weld joint

∴ Resultant Shear Stress 'A' According Parallelogram law of Forces

$$\cos\theta = \frac{r_1}{r_2}$$

Magnitude of Shear Stress

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

Value of Polar Moment of Inertia

$$J = \frac{tl3b^2}{6} + \frac{tl^3}{6}$$

$l = \text{length of weld}$

$b - \text{depth of transverse plate}$

Problem:

A welded joint as shown in Fig. 10.24, is subjected to an eccentric load of 2 KN. Find the size of weld, if the maximum shear stress in the weld is 25 M Pa.

Ans.:

Given Data:

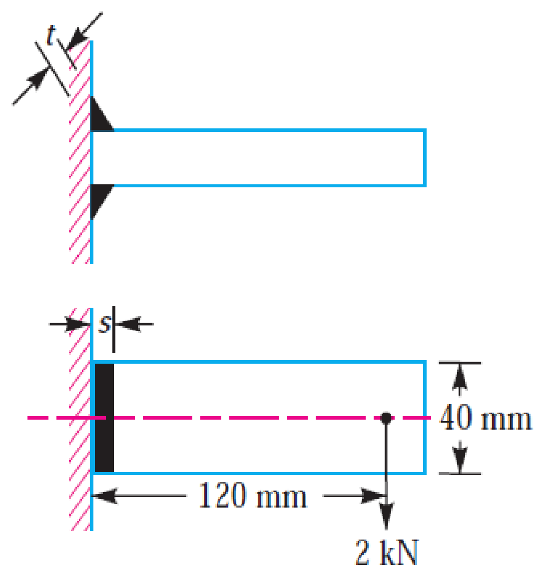
$P = 2\text{ KN} = 2000\text{ N}$

$S = ?$

$\tau_{Max} = 25\text{ MPa}$

$e = 120\text{ mm}$

$l = 40\text{ mm}$



1. Primary or Direct Shear stress acting on joint:

$$\tau = \frac{P_s}{1.414S \times l} = \frac{2000}{1.414 \times 40 \times l}$$

$$\tau = \frac{35.4}{S}$$

2. Section Modulus of weld through the throat.

$$Z = \frac{2tl^2}{6} = \frac{1.414\,S \times (40)^2}{6}$$

$$Z = 377s\text{ mm}^3$$

3. Bending Moment produced by eccentric load

$$M = P \times e = 2000 \times 120 = 24 \times 10^4\text{ Nmm}$$

4. Bending Stress induced in joint

$$\sigma_b = \frac{M}{Z} = \frac{240 \times 10^3}{377.S} = \frac{636.6}{S}\ {}^{N}\!/_{mm^2}$$

We have from Maximum Shear Stress Theory

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

$$\Rightarrow \mathbf{25} = \frac{1}{2}\sqrt{\left(\frac{636.6^2}{S} + \frac{35.4^2}{S}\right)}$$

$$\Rightarrow 25 = \frac{320.3}{S}$$

$$\Rightarrow S = \frac{320.3}{25} = 12.8 \text{ mm}$$

$$\mathbf{S = 12.8\ mm \ldots\ldots\ldots Size\ of\ Weld}$$

Q. A bracket carrying a load of 15 KN is to be welded as shown in Fig. Find the size of weld required if the allowable shear stress is not to exceed 80 M Pa

Given Data:

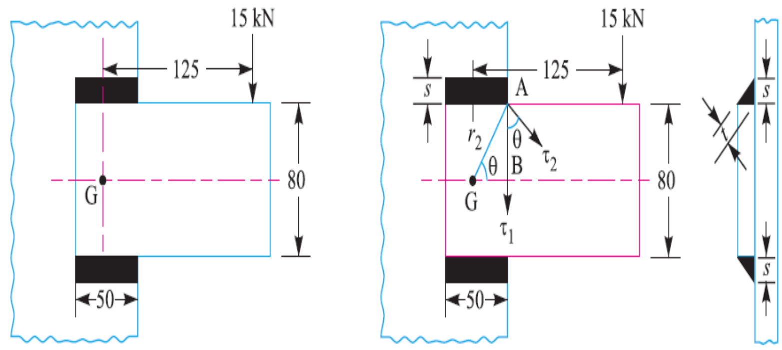
$P = 15 \text{ KN} = 15000 \text{ N}$

$\tau_r = 80 \text{ MPa}$

$e = 125 \text{ mm}$

$l = 50 \text{ mm}$

$b = 80 \text{ mm}$



All dimensions in mm.

Area of throat

$A = 2 \times t \times l = 1.414S \times l = 1.414 \times S \times 50$
$= 70.7S\ mm^2$

We know that

Direct or Primary shear stress

$$\tau_1 = \frac{P}{1.414S \times l} = \frac{15000}{70.7S} = \frac{212}{S}\ N/mm^2$$

Value of Polar Moment of Inertia

$$J = \frac{tl3b^2}{6} + \frac{tl^3}{6}$$

$$= \frac{0.707\,S \times 50[3 \times (80^2) + (50^2)\,]}{6}\ mm^4$$

$$J = 127850\ S\ mm^4$$

$\therefore$ *Maximum Radius of the weld*

$r_2 = \sqrt{(AB^2 + BG^2)} = \sqrt{(40^2 + 25^2)} = 47\ mm$

Shear Stress due to turning Moment i.e. Secondary Shear Stress

$$\tau_2 = \frac{P \times e}{J} \times r_2 = \frac{15000 \times 125 \times 47}{127850\ S} = \frac{689.3}{S}\ N/mm^2$$

$$\cos\theta = \frac{r_1}{r_2} = \frac{25}{47} = 0.53$$

The resultant shear stress produced in welding Joint

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

$$\Rightarrow 80 = \sqrt{\left(\frac{212}{S}^2 + \frac{689.3}{S}^2 + 2 \times \frac{212}{S} \times \frac{689.3}{S} \times 0.532\right)} = \frac{822}{S}$$

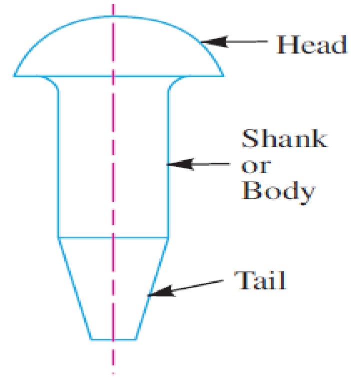
$$\Rightarrow 80 = \frac{822}{S}$$

$$\Rightarrow \mathbf{S = 10.3\ mm} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. \boldsymbol{Size\ of\ Weld}$$

Rivet Joint

Introduction

- ✓ 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,** as shown in Fig.
- ✓ The rivets are used to make permanent fastening between the plates such as in structural work, ship building, bridges, tanks and boiler shells.
- ✓ The riveted joints are widely used for joining light metals.



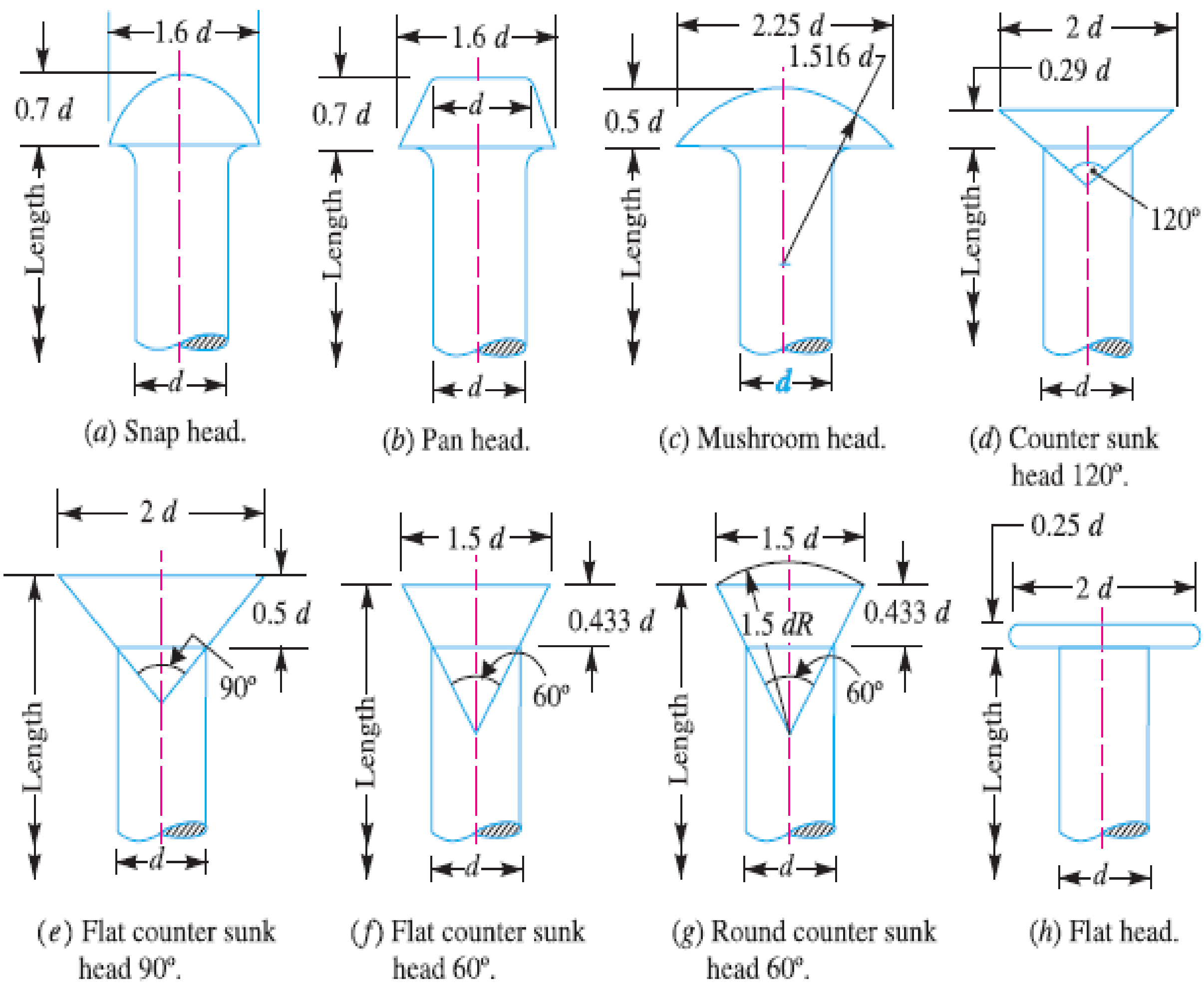
The fastenings (i.e. joints) may be classified into the following two groups:

1. Permanent fastenings, and
2. Temporary or detachable fastenings.

Types of Rivet:

According to Indian standard specifications, the rivet heads are classified into the following three types:

1. **Rivet heads for general purposes (below 12 mm diameter) as shown in Fig., according to IS: 2155 – 1982 (Reaffirmed 1996)**



. Rivet heads for general purposes (below 12 mm diameter).

2. **Rivet heads for general purposes (From 12 mm to 48 mm diameter) as shown in Fig. 9.4, according to IS: 1929 – 1982 (Reaffirmed 1996).**

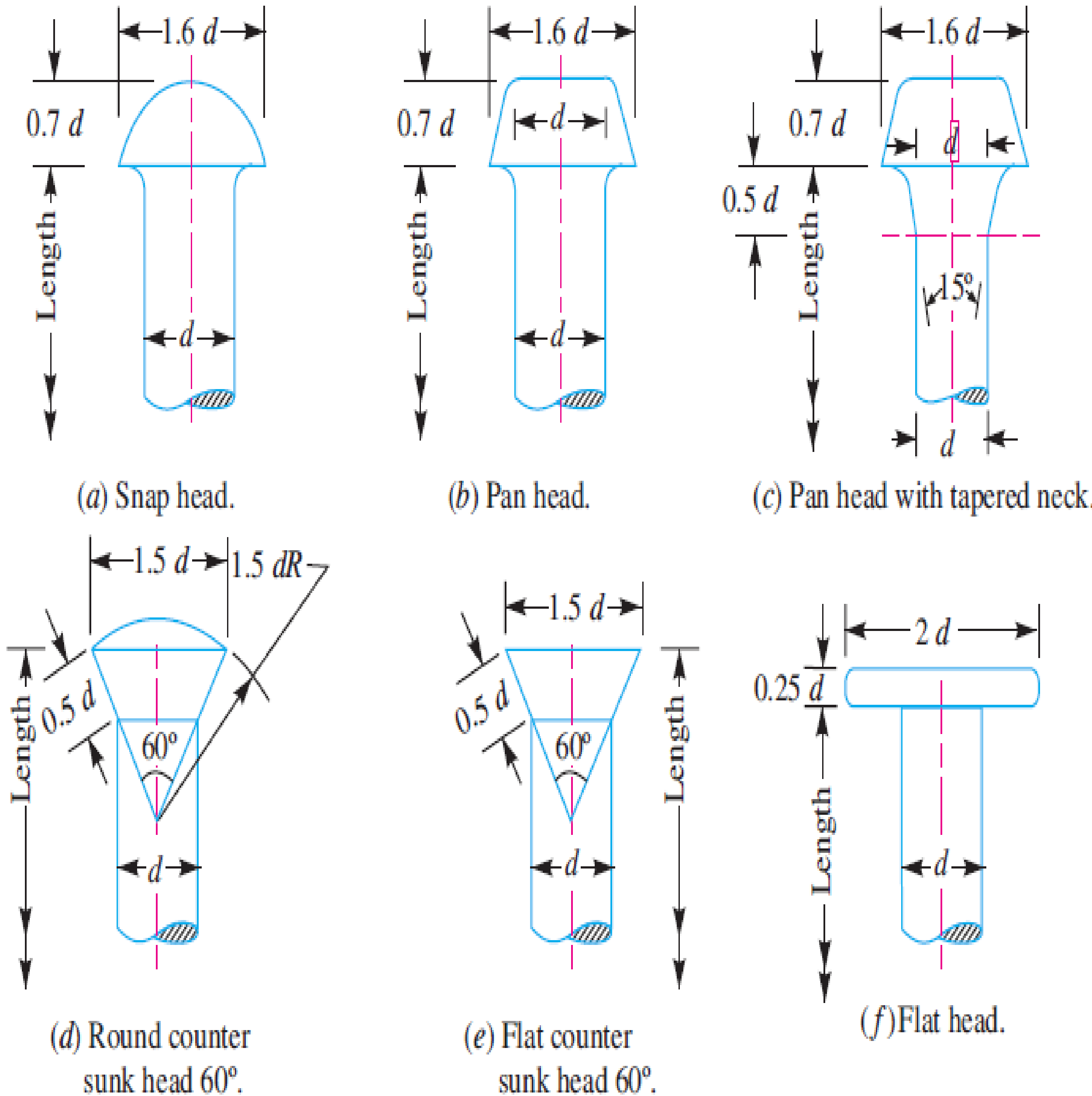


Fig. 9.4. Rivet heads for general purposes (from 12 mm to 48 mm diameter)

3. **Rivet heads for boiler work (from 12 mm to 48 mm diameter, as shown in Fig. 9.5, according to IS: 1928 – 1961 (Reaffirmed 1996).**

The snap heads are usually employed for structural work and machine riveting.
The counter sunk heads are mainly used for ship building where flush surfaces are necessary.
The conical heads (also known as conoidal heads) are mainly used in case of hand hammering.
The pan heads have maximum strength, but these are difficult to shape.

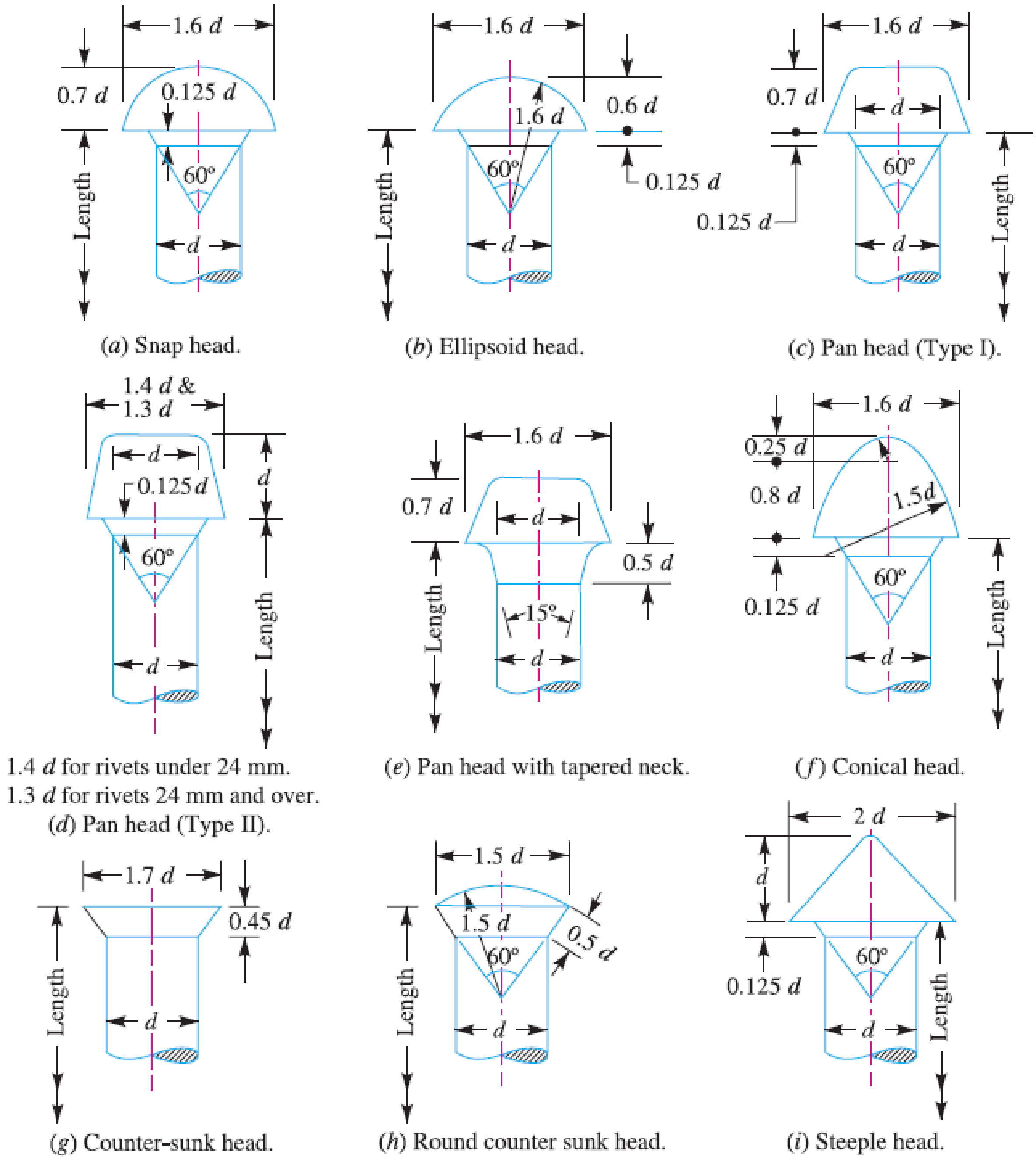
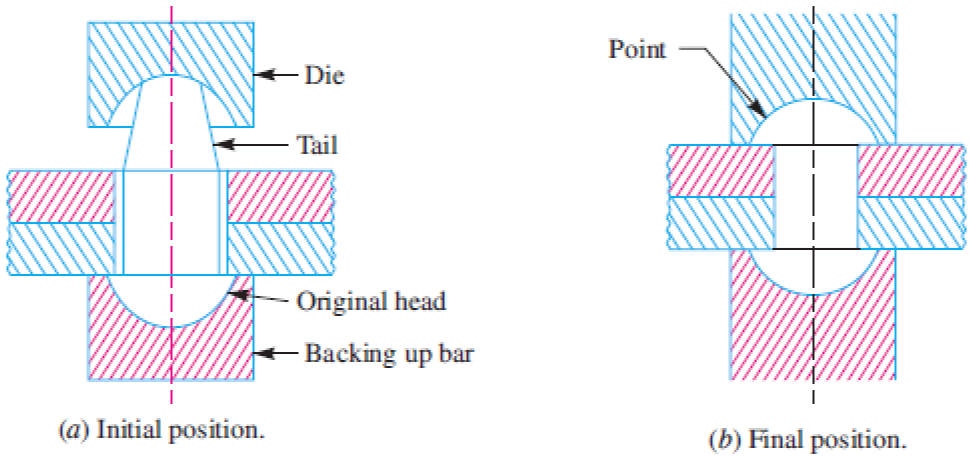


Fig. 9.5. Rivet heads for boiler work.

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 contribute to strength and to prevent leakage as in a boiler or in a ship hull.
- ✓ When two plates are to be fastened together by a rivet as shown in Fig. the holes in the plates are punched and reamed or drilled.
- ✓ Punching is the cheapest method and is used for relatively thin plates and in structural work. Since punching injures the material around the hole, therefore drilling is used in most pressure-vessel work.
- ✓ In structural and pressure vessel riveting, the diameter of the rivet hole is usually 1.5 mm larger than the nominal diameter of the rivet.



Methods of riveting.

- ✓ The plates are drilled together and then separated to remove any burrs or chips so as to have a tight flush 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 cold riveting process is used for structural joints while hot riveting is used to make leak proof joints.
- 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, as shown in Fig. (a), 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** as shown in Fig. 9.2 (b). As the rivet cools, it tends to contract.
- The lateral contraction will be slight, but there will be a longitudinal tension introduced in the rivet which holds the plates firmly together.
- In machine riveting, the die is a part of the hammer which is operated by air, hydraulic or steam pressure.

Notes:

1. For steel rivets up to 12 mm diameter, the cold riveting process may be used while for larger diameter rivets, hot riveting process is used.

2. In case of long rivets, only the tail is heated and not the whole shank.

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, but when strength and a fluid tight joint is the main consideration, then the steel rivets are used.
- ✓ The rivets for general purposes shall be manufactured from steel conforming to the following Indian Standards:

(*a*) IS: 1148–1982 (Reaffirmed 1992) – Specification for hot rolled rivet bars (up to 40 mm diameter) for structural purposes; or

(b) IS: 1149–1982 (Reaffirmed 1992) – Specification for high tensile steel rivet bars for structural purposes.

The rivets for boiler work shall be manufactured from material conforming to IS: 1990 – 1973 (Reaffirmed 1992) – Specification for steel rivets and stay bars for boilers.

Note:

The steel for boiler construction should conform to IS: 2100 – 1970 (Reaffirmed 1992) – Specification for steel billets, bars and sections for boilers.

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.

9.8 Lap Joint

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

9.9 Butt Joint

✓ 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.

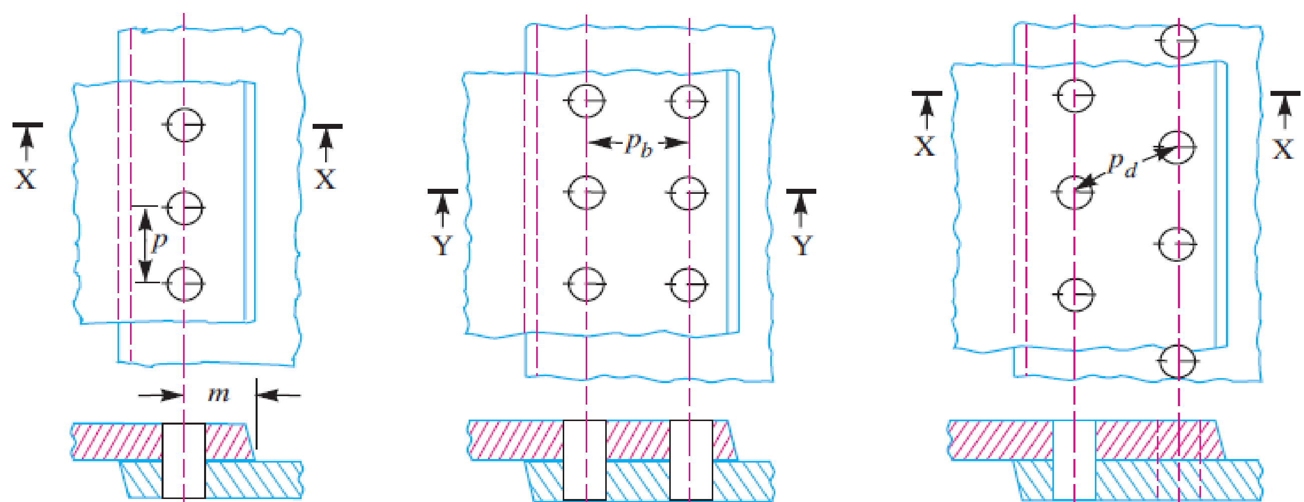
✓ The cover plate is then riveted together with the main plates.

Butt joints are of the following two types:

1. Single strap butt joint

2. Double strap butt joint

1. 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.
2. 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.



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

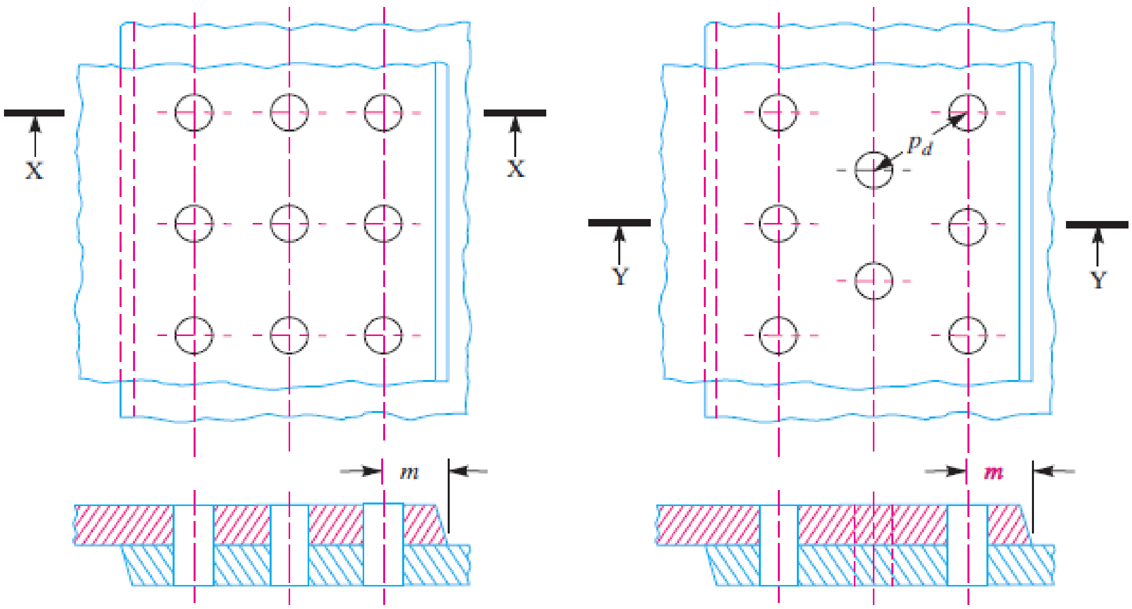
Fig. 9.6. Single and double riveted lap joints.

In addition to the above, following are the types of riveted joints depending upon the number of rows of the rivets.

1. Single riveted joint

2. Double riveted joint.

- A single riveted joint is that in which there is a single row of rivets in a lap joint as shown in Fig. 9.6 (a) and there is a single row of rivets on each side in a butt joint as shown in Fig. 9.8.
- A double riveted joint is that in which there are two rows of rivets in a lap joint as shown in Fig. 9.6 (b) and (c) and there are two rows of rivets on each side in a butt joint as shown in Fig. 9.9.



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

Fig. 9.7. Triple riveted lap joint.

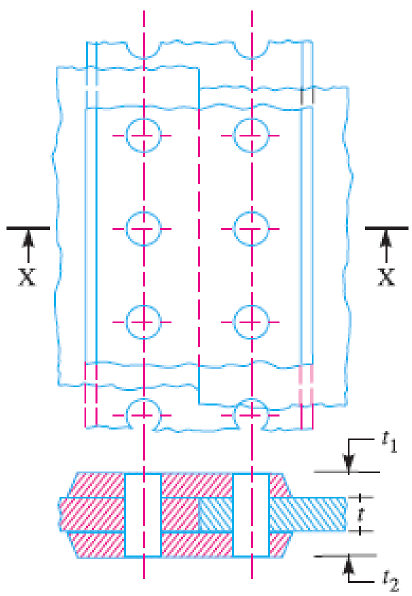


Fig. 9.8. Single riveted double strap butt joint.

Notes:

1. When the rivets in the various rows are opposite to each other, as shown in Fig. 9.6 (b), then the joint is said to be chain riveted. On the other hand, if the rivets in the adjacent rows are staggered in such a way that every rivet is in the middle of the two rivets of the opposite row as shown in Fig. 9.6 (c), then the joint is said to be zig-zag riveted.

Important Terms Used in Riveted Joints:

The following terms in connection with the riveted joints are important from the subject point of view:

1. **Pitch.**

   It is the distance from the center of one rivet to the center of the next rivet measured parallel to the seam as shown in Fig. 9.6.

   It is usually denoted by p.

2. **Back pitch.**

   It is the perpendicular distance between the center lines of the successive rows

   It is usually denoted by p_b.

3. **Diagonal pitch.**

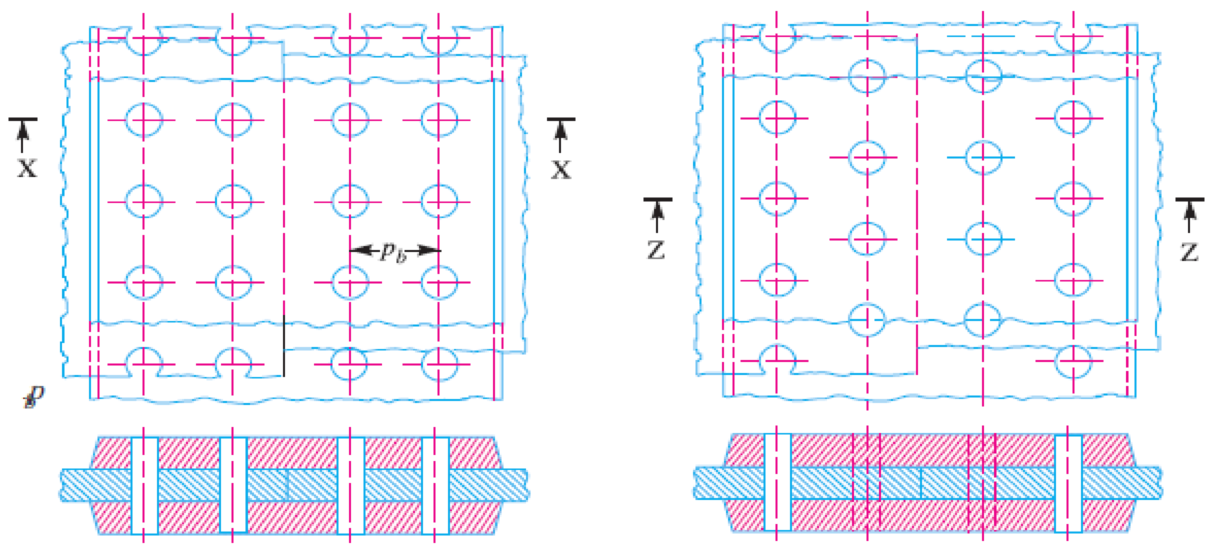
   It is the distance between the centers of the rivets in adjacent rows of zig-zag riveted joint.

   It is usually denoted by p_d.

4. **Margin or marginal pitch.**

   It is the distance between the centers of rivet hole to the nearest edge of the plate

   It is usually denoted by m.



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

Fig. 9.9. Double riveted double strap (equal) butt joints.

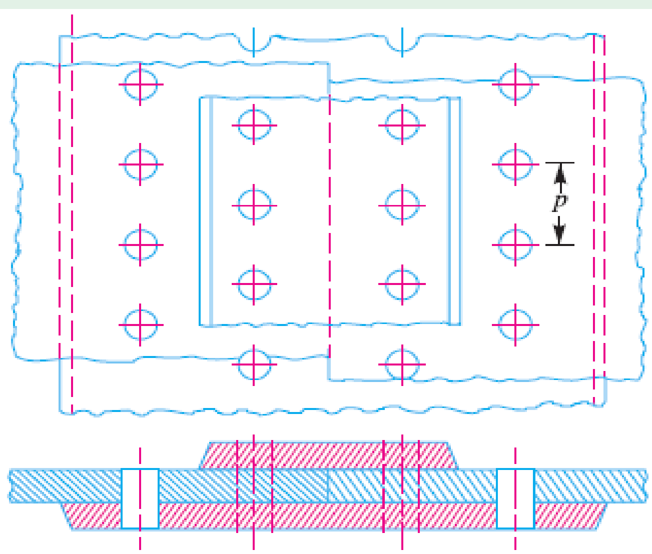


Fig. 9.10. Double riveted double strap (unequal) butt joint with zig-zag riveting.

❖ **Caulking and Fullering:**

- In order to make the joints leak proof or fluid tight in pressure vessels like steam boilers, air receivers and tanks etc. a process known as ***caulking*** is employed.
- In this process, a narrow blunt tool called caulking tool, about 5 mm thick and 38 mm in breadth, is used. The edge of the tool is ground to an angle of 80°.
- The tool is moved after each blow along the edge of the plate, which is planned to a Bevel of 75° to 80° to facilitate the forcing down of edge.
- It is seen that the tool burrs down the plate at *A* in Fig. 9.12 (*a*) forming a metal to metal joint. In actual practice, both the edges at *A* and B are caulked.

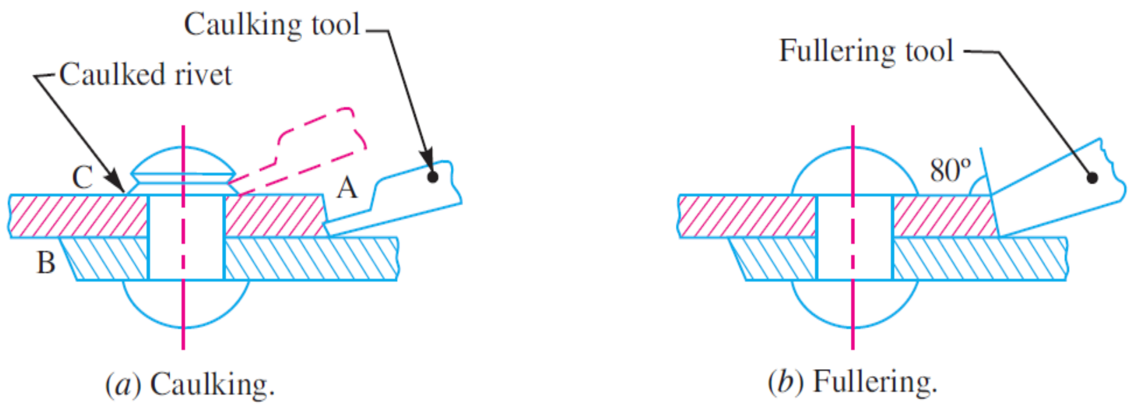


Fig. 9.12. Caulking and fullering.

- The head of the rivets as shown at C are also turned down with a caulking tool to make a joint steam tight. A great care is taken to prevent injury to the plate below the tool.
- A more satisfactory way of making the joints staunch is known as fullering which has largely superseded caulking.
- In this case, a fullering tool with a thickness at the end equal to that of the plate is used in such a way that the greatest pressure due to the blows occur near the joint, giving a clean finish, with less risk of damaging the plate.
- A fullering process is shown in Fig. 9.12 (b).

Failures of a Riveted Joint

A riveted joint may fail in the following ways:

1. Tearing of the plate at an edge. A joint may fail due to tearing of the plate at an edge as shown in Fig. 9.13. 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 Fig. 9.14.

✓ 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.

Let

$d = \text{diameter of rivet}$

t – Thickness of plate

P – Pitch of Rivet rows

σ_t – Tearing Stress or Tensile stress induced in rivet joint

P_t – Tearing Resistance of rivet Joint from it is observed that

Tearing area per pitch length

$A_t = (P - d)t$

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

$$\mathbf{P_t = A_t \times \sigma_t = (P - d)t \times \sigma_t}$$

Note:

If tearing resistance is greater than tensile load failure may not occur.

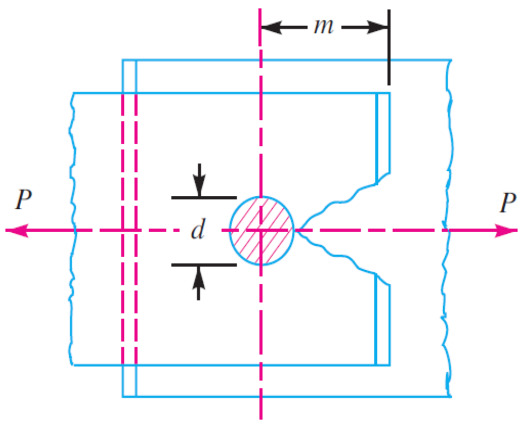


Fig. 9.13. Tearing of the plate at an edge.

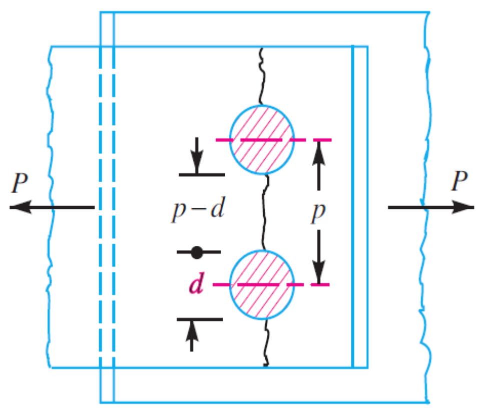


Fig. 9.14. Tearing of the plate across the rows of rivets.

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 Fig. 9.15.
- ✓ It may be noted that the rivets are in *single shear in a lap joint and in a single cover butt joint, as shown in Fig. 9.15.
- ✓ But the rivets are in double shear in a double cover butt joint as shown in Fig. 9.16.
- ✓ 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

P_s = shearing resistance of rivet joint

n – No of rivet per pitch length

τ – Shear Stress induced in rivet

d = diameter of rivet

We know that Shearing area

$$A_s = \frac{\pi}{4} \times d^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (\text{Single Shear})$$

$$= 2 \times \frac{\pi}{4} \times d^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (\text{in a Double Shear})$$

$$= 1.875 \times \frac{\pi}{4} \times d^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (\text{in a Double Shear, according to indian Boiler Regulation})$$

Shearing Resistance or pull required to shear off the rivet per pitch length load by rivet Joint

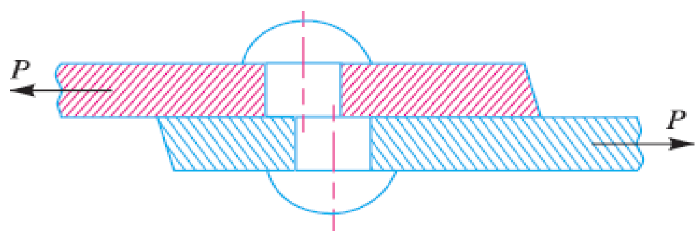
$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau \ldots\ldots\ldots\ldots\ldots \text{For Single Shear (Single Rivetting) } n = 1$$

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau \ldots\ldots\ldots\ldots\ldots \text{For Double Shear (Doble Rivetting) } n = 2$$

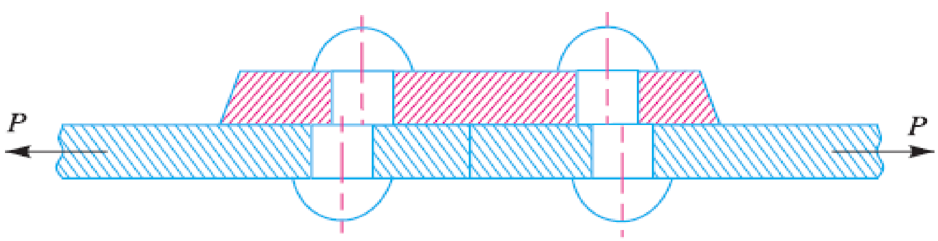
According to IBR act the general formula used for calculation of shear load on rivet joint

$$\mathbf{P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau \ldots\ldots\ldots \text{For Double strap Butt joint}}$$

When the shearing resistance is greater than the applied load per pitch length. Then this type of failure will occur



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



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

Fig. 9.15. Shearing of rivets.

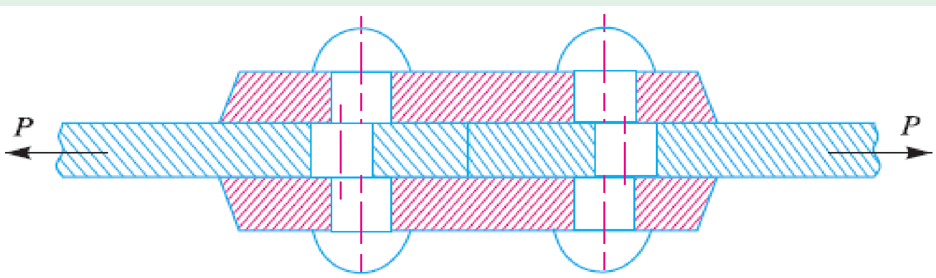


Fig. 9.16. Shearing off a rivet in double cover butt joint.

Crushing of the plate or rivets.

- ✓ Sometimes, the rivets do not actually shear off under the tensile stress, but are crushed as shown in Fig. 9.17. Due to this, the rivet hole becomes of an oval shape and hence the joint becomes loose.
- ✓ The failure of rivets in such a manner is also known as *bearing failure*. The area which resists this action is the projected area of the hole or rivet on diametral plane.

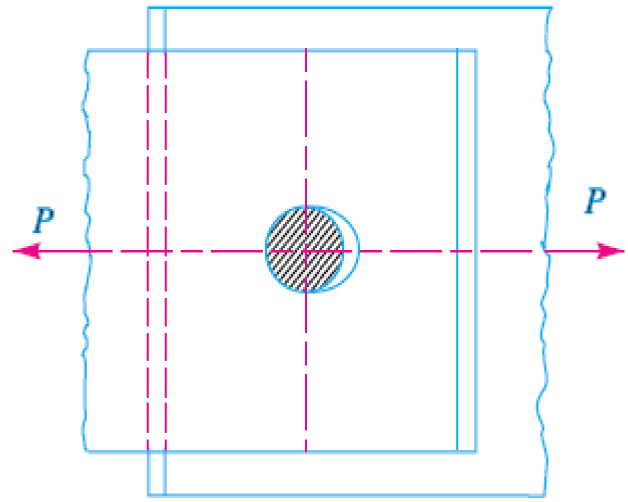


Fig. 9.17. Crushing of a rivet.

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

$d =$ Diameter of rivet hole
$t =$ thickness of plate
$\sigma_c -$ Safe or permissible Crushing stress
$n -$ No of rivet per pitch length

We know that crushing area per rivet (i.e. projected area per rivet)

$$A_c = d.t$$

$\therefore$ Total Crushing area $= n \times d \times t$
and crushing resistance or pull required to crush the rivet per pitch length

$$\mathbf{P_c = n \times d \times t \times \sigma_c}$$

When the crushing resistance is greater than the applied load per pitch length. Then this type of failure will occur

Strength of a Riveted Joint:

- ✓ It may be defined as the maximum force which can be transmit without causing it to fail.
- ✓ In earlier we have calculated three loads like Tearing load, crushing and Shearing load are required to tear off the plates, shearing off the rivets and crushing of rivet plates
- ✓ With increase in tensile load on rivet joint it will fail when the least of these three pulls are reached

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.
It is denote by symbol "η"
We have already discussed that strength of the rivet joint

$$= \text{ Least of } P_t, P_s, P_c$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t$$

$\therefore$ Efficiency of the rivet Joint,

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100\ \%$$

$p\ =$ Pitch of the rivets,
$t\ =$ Thickness of the plate, and
$\sigma_t\ =$ Permissible tensile stress of the plate material.

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: $\sigma t = 120 \text{ MPa}$; $\tau = 100 \text{ MPa}$; $\sigma c = 150 \text{ MPa}$ Find the efficiency of joint, taking the strength of the rivet in double shear as twice than that of single shear.
Ans.
Given Data:
$t = 20 \text{ mm}$
$d = 25 \text{ mm}$
$P = 100 \text{ mm}$
$\sigma_t = 120 \text{ MPa} = 120 \ {}^{N}/_{mm^2}$
$\sigma_c = 150 \text{ MPa} = 150 \ {}^{N}/_{mm^2}$
$\tau = 150 \text{ MPa} = 100 \ {}^{N}/_{mm^2}$

1. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 120 \times (100 - 25) \times 20 = 180{,}000 \text{N}$$

2. Shearing Resistance of rivet

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

3. Crushing Resistance of rivets

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

Strength of the rivet joint

$$= \text{ Least of } P_t, P_s, P_c$$
$$\mathbf{P_c = 150,000N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 100 \times 20 \times 120 = 240000N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100 \%$$
$$= \frac{\mathbf{P_c}}{P} \times 100\%$$
$$= \frac{150000}{240000} \times 100 \%$$
$$\boldsymbol{\eta} = \mathbf{62.50} \%$$

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 65 mm.
Assume Permissible tensile stress in plate = 120 MPa , Permissible shearing stress in rivets = 90 MPa , Permissible crushing stress in rivets = 180 MPa

Given Data:
$t = 06 \text{ mm}$
$d = 20 \text{ mm}$
$P = 50 \text{ mm}$
$\sigma_t = 120 \text{ MPa} = 120 \ {}^{N}/_{mm^2}$

$\sigma_c = 150 \text{ MPa} = 180 \; {}^{\text{N}}\!/_{\text{mm}^2}$

$\tau = 90 \text{ MPa} = 90 \; {}^{\text{N}}\!/_{\text{mm}^2}$

❖ **Single riveted lap joint**

i. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 120 \times (50 - 20) \times 6 = 21600\text{N}$$

ii. Shearing Resistance of rivet

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau = 1 \times \frac{\pi}{4} \times 25^2 \times 90 = 28278 \text{ N}$$

iii. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 1 \times 6 \times 20 \times 180 = 21600N$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s, P_c$$
$$\mathbf{P_c = 21600N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 120 \times 50 \times 6 = 36000N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100\ \%$$
$$= \frac{\mathbf{P_c}}{P} \times 100\%$$
$$= \frac{21600}{36000} \times 100\ \%$$
$$\boldsymbol{\eta} = \mathbf{60}\ \%$$

Double riveted lap joint:

$t = 06 \text{ mm}$
$d = 20 \text{ mm}$
$P = 65 \text{ mm}$

i. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 120 \times (65 - 20) \times 6 = 32400\text{N}$$

ii. Shearing Resistance of rivet

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times \frac{\pi}{4} \times 20^2 \times 90 = 56556 \text{ N}$$

iii. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 2 \times 6 \times 20 \times 180 = 43200N$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s, P_c$$
$$\mathbf{P_c = 43200N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 120 \times 65 \times 6 = 46800N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100\ \%$$
$$= \frac{\mathbf{P_c}}{P} \times 100\%$$
$$= \frac{32400}{46800} \times 100\ \%$$
$$\boldsymbol{\eta} = \mathbf{69.20}\ \%$$

❖ **Design of Boiler Joints:-**

A boiler is a device which is used to store and supply of steam in a power plant. It is a cylindrical element having length & diameter.

It consists of two joint

1. Longitudinal Butt joint
2. Circumferential Lap Joint

Longitudinal joint is used to join the ends of the plate to get the required diameter of Boiler. It is type of edge to edge joint (Butt Joint).

Circumferential Joint is used to get the required length of the boiler. It is done by overlapping of ends of plate. (Lap Joint).

Since a boiler is made up of number of rings, therefore the longitudinal joints are staggered for convenience of connecting rings at places where both longitudinal and circumferential joints occur

❖ **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. The assumption implies that the shell and plate are rigid and that all the deformation of the joint takes place in the rivets themselves.
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

According to Indian Boiler Regulations (I.B.R), the following procedure should be adopted for the design of longitudinal butt joint for a boiler.

1. Thickness of Boiler Shell (t)
2. Diameter of rivet (d)
3. Pitch of rivets (P)
4. Distance between rows of rivet (P_b)
5. Thickness of butt Strap (Equal or Unequal width)
6. Margin (m)

1. Thickness of Boiler Shell:

We know that circumferential stress in case of thin cylinder

$$\sigma_t = \frac{PD}{2t\eta_l}$$

$$\Rightarrow \mathbf{t} = \frac{\mathbf{PD}}{\mathbf{2} \times \boldsymbol{\sigma}_{\mathbf{t}} \times \boldsymbol{\eta}_{l}} + \mathbf{1}\ \boldsymbol{mm\ as\ corrosoin\ allowance}$$

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.

The following points may be noted:

(*a*) The thickness of the boiler shell should not be less than 7 mm.

(*b*) The efficiency of the joint may be taken from the following table.

Table 9.1. Efficiencies of commercial boiler joints.

Lap joints	*Efficiency (%)*	**Maximum efficiency*	*Butt joints (Double strap)*	*Efficiency (%)*	**Maximum efficiency*
Single riveted	45 to 60	63.3	Single riveted	55 to 60	63.3
Double riveted	63 to 70	77.5	Double riveted	70 to 83	86.6
Triple riveted	72 to 80	86.6	Triple riveted (5 rivets per pitch with unequal width of straps)	80 to 90	95.0
			Quadruple riveted	85 to 94	98.1

* The maximum efficiencies are valid for ideal equistrength joints with tensile stress = 77 MPa, shear stress = 62 MPa and crushing stress = 133 MPa.

Indian Boiler Regulations (I.B.R.) allow a maximum efficiency of 85% for the best joint.
(*c*) According to I.B.R., the factor of safety should not be less than 4. The following table shows the values of factor of safety for various kind of joints in boilers.

2. Diameter of Rivet (d):

If the thickness of plate is greater than 8 mm. then diameter of rivet can be found by using **Unwin's** Empirical formula

$$d = 6\sqrt{t}\,, t > 8 \text{ mm}$$

If the thickness of plate is less than 8 mm found out by equating loads of shearing and crushing Mathematically

$$P_s = P_c$$

$$\Rightarrow \mathbf{n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = n \times d \times t \times \sigma_c}$$

Table 9.3. Size of rivet diameters for rivet hole diameter as per IS : 1928 – 1961 (Reaffirmed 1996).

Basic size of rivet mm	12	14	16	18	20	22	24	27	30	33	36	39	42	48
Rivet hole diameter (min) mm	13	15	17	19	21	23	25	28.5	31.5	34.5	37.5	41	44	50

According to IS : 1928 – 1961 (Reaffirmed 1996), the table on the next page (Table 9.4) gives the preferred length and diameter combination for rivets.

3. **Pitch of Rivet:**

   The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets
   We have

$$P_s = P_t$$
$$\Rightarrow \mathbf{n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

$$\mathbf{P_{Max} = C \times t + 41.28\ mm}$$

where

$t -$ thickness of shell plate in mm

$C -$ constant

Table 9.5. Values of constant *C*.

Number of rivets per pitch length	*Lap joint*	*Butt joint (single strap)*	*Butt joint (double strap)*
1	1.31	1.53	1.75
2	2.62	3.06	3.50
3	3.47	4.05	4.63
4	4.17	–	5.52
5	–	–	6.00

NB:- if the pitch of rivet is obtained is morethan $\mathbf{P_{Max}}$ then the value of $\mathbf{P_{Max}}$ is taken

4. **Distance between rows of rivet (P_b)**

For equal number of rivets morethan one row for lap and butt joint. Distance between rows of rivet

$$P_b = 0.33\ P + 0.67\ d \ldots \ldots \ldots \ldots \text{Zig} - \text{Zag Rivetting}$$

$$P_b = 2d \ldots \ldots \ldots \ldots . \ldots \ldots \ldots \ldots \text{Chain Rivetting}$$

5. **Thickness of Butt Strap:-**
   i. Thickness of Buttstrap(t_1) in no case shall be less than 10 mm.
      $t_1 = 1.125\ t \ldots \ldots \ldots \ldots \ldots \ldots \ldots$ Single Butt Strap
      $t_1 = 0.625\ t \ldots \ldots \ldots \ldots \ldots \ldots \ldots .$ Equal width of butt strap having Ordinary rivetting
   ii. For unequal width of Butt Straps
      $t_1 = 0.75t \ldots \ldots \ldots \ldots \ldots \ldots$ for inside butt strap
      $t_1 = 0.625\ t \ldots \ldots \ldots \ldots . .$ narrow butt strap outside
6. **Margin:**

   We have to calculate the value of margin as
   $m = 1.5\ d, d =$ diameter of rivet

Q.A double riveted lap joint with zig − zag riveting is to be designed for 13 mm thick plates. Assume σt = 80 MPa ; τ = 60 MPa ; and σc = 120 MPa State how the joint will fail and find the efficiency of the joint.

Solution.

Given :

$t = 13\,mm$

$\sigma_t = 80\text{ MPa} = 80\ {}^{N}/_{mm^2}$

$\sigma_c = 120\text{ MPa} = 120\ {}^{N}/_{mm^2}$

$\tau = 60\text{ MPa} = 60\ {}^{N}/_{mm^2}$

Diameter of Rivet (d):

Diameter rivet

Since $t > 8\text{ mm}$

$d = 6\sqrt{t} = 6\sqrt{13} = 21.6\,mm$

From Table 9.3, we find that according to IS: 1928 – 1961 (Reaffirmed 1996), the standard size of the rivet hole (d) is 23 mm and the corresponding diameter of the rivet is 22 mm.

Pitch of Rivet:

The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets We have

$$P_s = P_t$$

$$\Rightarrow \Rightarrow \mathbf{n \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

$$\Rightarrow 2 \times \frac{\pi}{4} \times 23^2 \times 60 = (P - 23) \times 13 \times 80$$

$$\Rightarrow \mathbf{P = 71\ mm}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

From Table 9.5, we find that for 2 rivets per pitch length, the value of *C* is 2.62

$$P_{Max} = C \times t + 41.28\text{ mm}$$

$$P_{Max} = 2.62 \times 13 + 41.28 = 75.28\text{ mm}$$

Since P_{Max} is more than *p*, therefore we shall adopt

$$\mathbf{P = 71mm}$$

Distance between the rows of rivet:

For equal number of rivets morethan one row for lap. Distance between rows of rivet

$$P_b = 0.33\,P + 0.67\,d \ldots\ldots\ldots\ldots \text{Zig} - \text{Zag Rivetting}$$

$$P_b = 0.33 \times 71 + 0.67 \times 23 = 38.8\text{ mm Say } 40\text{ mm}$$

Margin:

We know that the margin,

$$m = 1.5\,d = 1.5 \times 23 = 34.5\ say\ 35\ mm$$

Failure of the joint:

$t = 13\ \text{mm}$
$d = 23\ \text{mm}$
$P = 71\ \text{mm}$

i. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 80 \times (71 - 23) \times 13 = 49920\text{N}$$

ii. Shearing Resistance of rivet

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times \frac{\pi}{4} \times 23^2 \times 60 = 49864\ \text{N}$$

iii. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 2 \times 23 \times 13 \times 120 = 71760N$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s, P_c$$

$$\mathbf{P_s = 49864N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 71 \times 13 \times 80 = 73840\ N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100\ \%$$

$$= \frac{P_s}{P} \times 100\%$$

$$= \frac{49864}{73840} \times 100\ \%$$

$$\boldsymbol{\eta = 67.50}\ \%$$

Q. Two plates of 7 mm thick are connected by a triple riveted lap joint of zig-zag pattern. Calculate the rivet diameter, rivet pitch and distance between rows of rivets for the joint. Also state the mode of failure of the joint. The safe working stresses are as follows:

$$\boldsymbol{\sigma t = 90\ \text{MPa}\ ;\ \tau = 60\ \text{MPa}\ ;\ \text{and}\ \sigma c = 120\ \text{MPa}}$$

Given Data
Type of Joint: Lap Joint
Form of Joint: Triple Riveted
$\sigma t = 90\ \text{MPa}$
$\tau = 60\ \text{MPa}$
$\sigma c = 120\ \text{MPa}$
$t = 7\ \text{mm}$
$n = 3$

1. **Diameter of Rivet (d):**

For Triple riveted Lap Joint
No of Rivet per Pitch length n = 3

$$\text{we have Crushing resistance of rivets } P_c = n \times d \times t \times \sigma_c = 3 \times d \times 8 \times 120 = \mathbf{2520d\ N} \ldots\ldots\ldots\ldots (\mathbf{i})$$

$$\text{shearing Resistance of Rivets } P_s = n \times \frac{\pi}{4} \times d^2 \times \tau = 3 \times \frac{\pi}{4} \times d^2 \times 60 = 141.4d^2\ \text{N} \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (\text{ii})$$

Equating loads of shearing and crushing (i) & (ii)
Mathematically

$$P_s = P_c$$
$$141.4d^2 = 2520d$$
$$d = \frac{2520}{141.4} = 17.8 \text{ mm}$$
From Table 9.3, we see that according to IS: 1928 – 1961 (Reaffirmed 1996), the standard diameter of rivet hole (*d)* is 19 mm and the corresponding diameter of rivet is 18 mm.

Pitch of Rivet:

The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets

We have

$$P_s = P_t$$
$$\Rightarrow \mathbf{n \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

$$\Rightarrow 3 \times \frac{\pi}{4} \times 19^2 \times 60 = (P - 19) \times 7 \times 90$$
$$\Rightarrow \mathbf{P = 100\ mm}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

From Table 9.5, we find that for 2 rivets per pitch length, the value of *C* is 3.47

$$P_{Max} = C \times t + 41.28 \text{ mm}$$

$$P_{Max} = 3.47 \times 7 + 41.28 = 66 \text{ mm}$$

Since P_{Max} is less than *p*, therefore we shall adopt

$$\mathbf{P} = P_{Max} = \mathbf{66\ mm}$$

Distance between the rows of rivet:

For equal number of rivets morethan one row for lap. Distance between rows of rivet

$$P_b = 0.33\, P + 0.67\, d \ldots\ldots\ldots\ldots \text{Zig} - \text{Zag Rivetting}$$

$$P_b = 0.33 \times 66 + 0.67 \times 19 = 34.5 \text{ mm}$$

Failure of the joint:

$t = 7$ mm
$d = 19$ mm
$P = 66$ mm

i. Tearing resistance of Plate
$$P_t = (P - d)t \times \sigma_t = 90 \times (66 - 19) \times 7 = 29610 \text{ N}$$

ii. Shearing Resistance of rivet

$$P_s = n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times \frac{\pi}{4} \times 19^2 \times 60 = 51045 \text{ N}$$

iii. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 3 \times 19 \times 7 \times 120 = 47880\ N$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s, P_c$$
$$\mathbf{P_t = 29610\ N}$$

From above we see least value of three loads and it is observed that the joint will fail due to tearing off plate**.**

Q. Two plates of 10 mm thickness each are to be joined by means of a single riveted double strap butt joint. Determine the rivet diameter, rivet pitch, strap thickness and efficiency of the joint. Take the working stresses in tension and shearing as 80 MPa and 60 MPa respectively

Given Data:
Type of Joint: Butt Joint
Form of Joint: Single Riveted
No of Strap: Double Butt Strap
$t = 10\ mm$
$\sigma_t = 80\ \text{MPa}$
$\tau = 60\ \text{MPa}$

Diameter of Rivet (d):

Diameter rivet

Since $t > 8\ \text{mm}$

$d = 6\sqrt{10} = 6\sqrt{10} = 18.97\ mm$

From Table 9.3, we find that according to IS: 1928 – 1961 (Reaffirmed 1996), the standard size of the rivet hole (d) is 19 mm and the corresponding diameter of the rivet is 18 mm.

Pitch of Rivet:

The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets

We have

$$P_s = P_t$$
$$\Rightarrow \Rightarrow \mathbf{n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

$$\Rightarrow 2 \times \frac{\pi}{4} \times 19^2 \times 60 = (P - 19) \times 10 \times 80$$
$$\Rightarrow \mathbf{P = 58.87\ mm\ Say\ 60\ mm}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

From Table 9.5, we find that for 2 rivets per pitch length, the value of *C* is 1.75

$$P_{Max} = C \times t + 41.28\ \text{mm}$$

$$P_{Max} = 1.75 \times 10 + 41.28 = 58.78\ \text{mm Say } 60\ \text{mm}$$

Since P_{Max} is more than *p*, therefore we shall adopt

$$\mathbf{P = P_{Max} = 60mm}$$

Thickness of Butt Strap:-

Thickness of Buttstrap(t_1) in no case shall be less than 10 mm.

$t_1 = 0.625\ t = 0.625 \times 10 = 6.25\ \text{mm} \ldots\ldots$ Equal width of butt strap having Ordinary rivetting

Failure of the joint:

$t = 10\ \mathrm{mm}$
$d = 19\ \mathrm{mm}$
$P = 60\ \mathrm{mm}$

i. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 80 \times (60 - 19) \times 10 = 32800\mathrm{N}$$

ii. Shearing Resistance of rivet

$$P_s = n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = 1 \times 1.875 \times \frac{\pi}{4} \times 19^2 \times 60 = 31900\ \mathrm{N}$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s,$$
$$\mathbf{P_s = 31900\ N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 60 \times 10 \times 80 = 48000\ N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s}{P \times t \times \sigma_t} \times 100\ \%$$
$$= \frac{P_s}{P} \times 100\%$$
$$= \frac{31900}{48000} \times 100\ \%$$
$$\boldsymbol{\eta} = \mathbf{66.5}\ \%$$

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.

Given Data:

Name of Joint: Butt Joint
Form of Joint: Double riveted
No. of Strap: Double Strap

$D = 1.5\ m = 1500\ mm$

$Steam\ pressure\ P = 0.95\ {}^{N}/_{mm^2}$

$\boldsymbol{\eta_l} = \mathbf{75}\%$

$\sigma_t = 90\ \mathrm{MPa} = 90\ {}^{\mathrm{N}}/_{\mathrm{mm^2}}$

$\sigma_c = 140\ \mathrm{MPa} = 140\ {}^{\mathrm{N}}/_{\mathrm{mm^2}}$

$\tau = 60\ \mathrm{MPa} = 56\ {}^{\mathrm{N}}/_{\mathrm{mm^2}}$

- **Thickness of Boiler Shell:**

We know that circumferential stress in case of thin cylinder

$$\sigma_t = \frac{PD}{2t\eta_l}$$

$$\Rightarrow \mathbf{t = \frac{0.95 \times 1500}{2 \times 90 \times .75} + 1\,\boldsymbol{mm} = 11.6\ \boldsymbol{say}\ 12\ \boldsymbol{mm}}$$

- **Diameter of Rivet (d):**

Diameter rivet

Since $t > 8$ mm

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

From Table 9.3, we find that according to IS: 1928 – 1961 (Reaffirmed 1996), the standard size of the rivet hole (d) is 21 mm and the corresponding diameter of the rivet is 20 mm.

- **Pitch of Rivet:**

The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets

We have

$$P_s = P_t$$

$$\Rightarrow\Rightarrow \mathbf{n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

$$\Rightarrow 2 \times 1.875 \times \frac{\pi}{4} \times 21^2 \times 56 = (P - 21) \times 12 \times 90$$

$$\Rightarrow \mathbf{P = 88.35\ Say\ 90\ mm}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

From Table 9.5, we find that for 2 rivets per pitch length, the value of *C* is 3.50

$$P_{Max} = C \times t + 41.28\ mm$$

$$P_{Max} = 3.5 \times 12 + 41.28 = 83.28\ say\ 84\ mm$$

Since P_{Max} is less than *p*, therefore we shall adopt

$$\mathbf{P = 84\ m}$$

- **Failure of the joint:**

$t = 28$ mm
$d = 34.5$ mm
$P = 220$ mm

i. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 90 \times (84 - 21) \times 12 = 68040N$$

ii. Shearing Resistance of rivet

$$P_s = 2 \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = 2 \times 1.875 \times \frac{\pi}{4} \times 21^2 \times 56 = 72745\ N$$

iii. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 2 \times 21 \times 12 \times 140 = 70560\ N$$

Strength of the rivet joint

$$= \text{ Least of } P_t, P_s, P_c$$

$$\mathbf{P_t = 68040\ N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 84 \times 12 \times 90 = 90720\ N$$

Efficiency of Joint

$$\eta = \frac{\text{Least of } P_t, P_s}{P \times t \times \sigma_t} \times 100\ \%$$

$$= \frac{P_s}{P} \times 100\%$$

$$= \frac{68040}{90720} \times 100\ \%$$

$$\boldsymbol{\eta} = \mathbf{75}\ \%$$

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

- **Design of Circumferential Joint:**

  The following procedure is adopted for the design of circumferential lap joint of a boiler

- **Thickness of Boiler Shell:**

We know that circumferential stress in case of thin cylinder

$$\sigma_t = \frac{PD}{2t\eta_l}$$

$$\Rightarrow \mathbf{t} = \frac{\mathbf{PD}}{\mathbf{2 \times \sigma_t \times \eta_l}} + \mathbf{1}\ \boldsymbol{mm\ as\ corrosoin\ allowance}$$

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.

- **Diameter of Rivet (d):**

If the thickness of plate is greater than 8 mm. then diameter of rivet can be found by using **Unwin's** Empirical formula

$$\mathbf{d = 6\sqrt{t}\,, t > 8\ mm}$$

If the thickness of plate is less than 8 mm found out by equating loads of shearing and crushing Mathematically

$$P_s = P_c$$

$$\Rightarrow \mathbf{n \times \frac{\pi}{4} \times d^2 \times \tau = n \times d \times t \times \sigma_c}$$

- ❖ **Number of Rivet:**

  For lap joint there is single shear off takes place.

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

For Circumferential joint Pressure of Steam

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

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

Now equating $W_s = P_s$

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

$$\Rightarrow \mathbf{n = \frac{D^2}{d^2} \times \frac{P}{\tau}} \ldots\ldots\ldots\ldots\ldots..\boldsymbol{No\ of\ Rivet}$$

- ❖ **Pitch of Rivets:-**

  Pitch of rivet is obtained from efficiency of circumferential joint

$$\boldsymbol{\eta_c = \frac{P_1 - d}{P_1}}$$

$$P_1 - \text{pitch of rivet joint}$$

- ❖ **No. of Rows:**

  No of rows of rivet is obtained by following relation

$$\text{No of Rows} = \frac{\text{Total number of rivets}}{\text{No. of Rivets in one row}}$$

$$\text{No. of Rivets in a Row} = \mathbf{\frac{\pi(D + t)}{P_1}}$$

  After finding out the no. of rows type of joint will be decied (i.e. Single or Double Rivet etc.)

- ❖ **Overlap of plates:**

$$\mathbf{Overlap\ of\ plate\ = (No.\ of\ rows\ of\ \ rivets - 1)P_b + m}$$

  $\text{where } m = \text{Margin}$

Q. A steam boiler is to be designed for a working pressure of 2.5 N/mm2 with its inside diameter 1.6 m. Give the design calculations for the longitudinal and circumferential joints
For the following working stresses for steel plates and rivets:
In tension = 75 MPa; in shear = 60 MPa; in crushing = 125 MPa.
Draw the joints to a suitable scale.

Given Data:

Name of Joint: Butt Joint
Form of Joint: Double riveted
No. of Strap: Double Strap
$D = 1.6\, m = 1600\, mm$
$Steam\ pressure\ P = 2.5\ {}^{N}/_{mm^2}$
$\sigma_t = 90\text{ MPa} = 75\ {}^{\text{N}}/_{\text{mm}^2}$
$\sigma_c = 125\text{ MPa} = 125\ {}^{\text{N}}/_{\text{mm}^2}$
$\tau = 60\text{ MPa} = 60\ {}^{\text{N}}/_{\text{mm}^2}$

Design of Longitudinal Joint:

- **Thickness of Boiler Shell:**
  We know that circumferential stress in case of thin cylinder

$$\sigma_t = \frac{PD}{2t}$$
$$\Rightarrow \mathbf{t} = \frac{\mathbf{2.5 \times 1600}}{\mathbf{2 \times 75}} + \mathbf{1}\ \boldsymbol{mm} = \mathbf{27.6}\ \boldsymbol{say}\ \mathbf{28}\ \boldsymbol{mm}$$

- **Diameter of Rivet (d):**
  Diameter rivet
  Since $t > 8$ mm
  $d = 6\sqrt{28} = 6\sqrt{28} = 31.75\, mm$

From Table 9.3, we find that according to IS: 1928 – 1961 (Reaffirmed 1996), the standard size of the rivet hole (d) is 34.5 mm and the corresponding diameter of the rivet is 33 mm.

- **Pitch of Rivet:**
  The pitch of rivet is calculated by equating tearing resistance and shearing resistance of rivets
  We have

$$P_s = P_t$$
$$\Rightarrow \Rightarrow \mathbf{n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = (P - d)t \times \sigma_t}$$

$$\Rightarrow 2 \times 1.875 \times \frac{\pi}{4} \times 34.5^2 \times 60 = (P - 34.5) \times 12 \times 75$$
$$\Rightarrow \mathbf{P = 261.5\ mm}$$

Pitch of rivets should be less than 2d. which is necessary for formation of head.

The maximum value for pitch of rivet for longitudinal joint of boiler.

From Table 9.5, we find that for 2 rivets per pitch length, the value of *C* is 6

$$P_{Max} = C \times t + 41.28\text{ mm}$$

$$P_{Max} = 6 \times 28 + 41.28 = 209.28\text{ say } 220\text{ mm}$$

Since P_{Max} is more than *p*, therefore we shall adopt

$$P = P_{Max} = 220\text{ mm}$$

∴ Pitch of rivets in the inner row,

$$p' = 220 / 2 = 110 \text{ mm}$$

❖ **Distance between the rows of rivet:**

According to I.B.R., the distance between the outer row and the next row

$$P_b = 0.2\,P + 1.15\,d$$

$$P_b = 0.2 \times 220 + 1.15 \times 34.5 = 83.7 \text{ mm Say } 85 \text{ mm}$$

and the distance between the inner rows for zig-zig riveting

$$P_b = 0.165\,P + 0.67d$$

$$P_b = 0.165 \times 220 + 0.67 \times 34.5 = 59.4 \text{ mm Say } 60 \text{ mm}$$

❖ **Thickness of Butt Strap:-**

Thickness of Buttstrap(t_1) in no case shall be less than 10 mm.

i. For unequal width of Butt Straps

$t_1 = 0.75t = 0.75 \times 28 = 21 \text{ mm} \ldots\ldots\ldots\ldots\ldots\ldots$ for inside butt strap

$t_1 = 0.625\,t = 0.625 \times 28 = 17.5 \text{ say } 18 \text{ mm} \ldots\ldots\ldots\ldots..$ narrow butt strap outside

❖ **Margin:**

We know that the margin,

$$m = 1.5\,d = 1.5 \times 34.5 = 57.75 \text{ say } 52 \text{ mm}$$

❖ **Failure of the joint:**

$t = 28 \text{ mm}$
$d = 34.5 \text{ mm}$
$P = 220 \text{ mm}$

iv. Tearing resistance of Plate

$$P_t = (P - d)t \times \sigma_t = 75 \times (220 - 34.5) \times 28 = 389550\text{N}$$

v. Shearing Resistance of rivet

$$P_s = 2 \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau = 4 \times 1.875 \times \frac{\pi}{4} \times 34.5^2 \times 60 = 476820 \text{ N}$$

vi. Crushing Resistance of rivets

$$P_c = n \times d \times t \times \sigma_c = 5 \times 34.5 \times 28 \times 125 = 603750\,N$$

The joint may also fail by tearing off the plate between the rivets in the second row. This is only possible if the rivets in the outermost row gives way (*i.e.* shears). Since there are two rivet holes per pitch length in the second row and one rivet in the outermost row, therefore

Combined tearing and shearing resistance

$$= (P - 2d)t \times \sigma_t + \frac{\pi}{4} \times d^2 \times \tau$$

$$= (220 - 2 \times 34.5)28 \times 75 + \frac{\pi}{4} \times 34.5^2 \times 60 = 373196 \text{ N}$$

Strength of the rivet joint

$$= \text{Least of } P_t, P_s, P_c$$
$$P_t = 373196\text{N}$$

Strength of the uriveted or solid plate per pitch length

$$P = P \times t \times \sigma_t = 220 \times 28 \times 75 = 462000 \, N$$

❖ **Efficiency of Joint**

$$\eta = \frac{\text{Least of } P_t, P_s, P_c}{P \times t \times \sigma_t} \times 100 \,\%$$
$$= \frac{P_t}{P} \times 100\%$$
$$= \frac{373196}{462000} \times 100 \,\%$$
$$\boldsymbol{\eta} = \mathbf{80.8} \,\%$$

Design of Circumferential Joint:

The circumferential joint for a steam boiler may be designed as follows:

1. The thickness of the boiler shell (t)and diameter of rivet hole (d)will be same as for longitudinal joint,

$$\text{i.e. } t = 28 \text{ mm} ; \text{ and } d = 34.5 \text{ mm}$$

❖ **Number of Rivet:**

For lap joint there is single shear off takes place.

$$P_s = \mathbf{n} \times \frac{\boldsymbol{\pi}}{\mathbf{4}} \times \mathbf{d^2} \times \boldsymbol{\tau} \ldots\ldots\ldots\ldots\ldots (\mathbf{i})$$

For Circumferential joint Pressure of Steam

$$P = \frac{W_s}{\frac{\pi}{4} \times D^2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (\text{ii})$$

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

Now equating $W_s = P_s$

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

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

$$\Rightarrow \mathbf{n} = \frac{\mathbf{1600^2}}{\mathbf{34.5^2}} \times \frac{\mathbf{2.5}}{\mathbf{60}}$$

$$\mathbf{n = 89.6} \; \boldsymbol{say} \; \mathbf{90} \ldots\ldots\ldots\ldots\ldots.. \boldsymbol{No\ of\ Rivet}$$

❖ **Pitch of Rivets:-**

Assuming the joint to be double riveted lap joint with zig-zag riveting, therefore number of rivets per row

$$= 90 / 2 = 45$$

We know that the pitch of the rivets,

$$P_1 - \text{pitch of rivet joint}$$

$$\text{No. of Rivets in a Row}$$

$$\mathbf{P_1 = \frac{\pi(D+t)}{\mathit{n}} = \frac{\pi(1600+28)}{45} = 113.7\ \mathit{mm}}$$

Let us take pitch of the rivets

$$\mathbf{P_1} = 140 \text{ mm}$$

Efficiency of joint

We know that the efficiency of the circumferential joint,

$$\boldsymbol{\eta_c = \frac{140 - 34.5}{140} = 0.753\ or\ 75.3}\ \%$$

$$P_1 - \text{pitch of rivet joint}$$

- **No. of Rows:**

No of rows of rivet is obtained by following relation

$$\text{No of Rows} = \frac{\text{Total number of rivets}}{\text{No. of Rivets in one row}}$$

$$\text{No. of Rivets in a Row} = \frac{\mathbf{\pi(D+t)}}{\mathbf{P_1}}$$

After finding out the no. of rows type of joint will be decied (i.e. Single or Double Rivet etc.)

Distance between the rows of rivet:

For equal number of rivets morethan one row for lap. Distance between rows of rivet

$$P_b = 0.33\ P_1 + 0.67\ d \ldots\ldots\ldots\ldots \text{Zig} - \text{Zag Rivetting}$$

$$P_b = 0.33\ \times 140\ + 0.67\ \times 34.5 = 69.3 \text{ say } 70\text{mm}$$

- **Margin:**

We know that the margin,

$$m\ =\ 1.5\ d\ =\ 1.5\ \times\ 34.5\ =\ 51.75 \text{ say } 52 \text{ mm}$$