Important Topics of Machine Design

STRESS CONCENTRATION

Stress concentration has been discussed in earlier lessons. However, it is important to realize that stress concentration affects the fatigue strength of machine parts severely and therefore it is extremely important that this effect be considered in designing machine parts subjected to fatigue loading. This is done by using fatigue stress concentration factor defined as

$$k_f = \frac{\textbf{endurance limit of a notch free specimen}}{\textbf{endurance limit of a notched specimen}}$$

The notch sensitivity 'q' for fatigue loading can now be defined in terms of K_f and the theoretical stress concentration factor K_t and this is given by

$$q = \frac{K_f - 1}{K_t - 1}$$

The value of q is different for different materials and this normally lies between 0 to 0.7. The index is small for ductile materials and it increases

<u>Method of stress concentration reduction:-</u>

Although it is not possible to completely eliminate the effect of stress concentration, there are methods to reduce stress concentrations. This is achieved by providing a specific geometric shape to the component.

(a) Additional Notches and Holes in Tension Member:

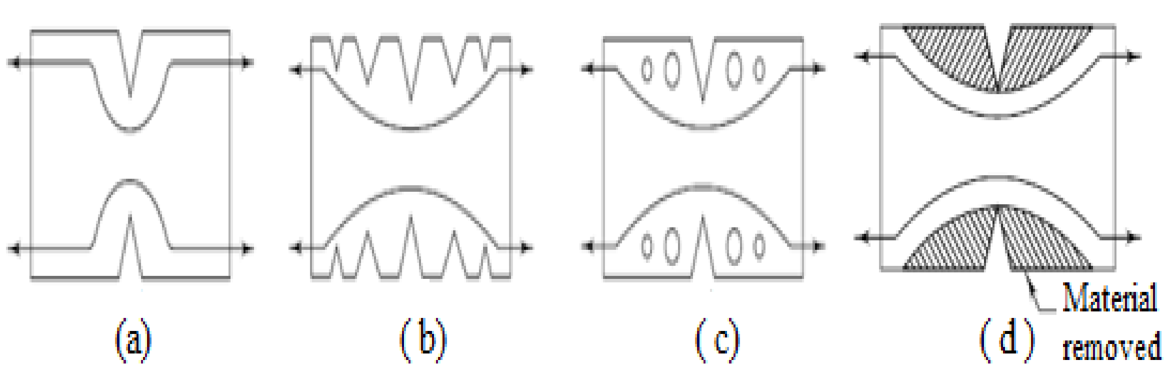
A flat plate with a V-notch subjected to tensile force is shown in Fig. (a). It is observed that a single notch results in a high degree of stress concentration.

The severity of stress concentration is reduced by three methods:

(I) Use of multiple notches as shown in fig (b)

(II) Drilling additional holes as shown in fig (c)

(III) Removal of undesired material as shown in fig (c)



(a) (b) (c) (d)

(b) Fillet Radius, Undercutting and Notch for Member in Bending

A bar of circular cross-section with a shoulder and subjected to bending moment is shown in Fig. (a).

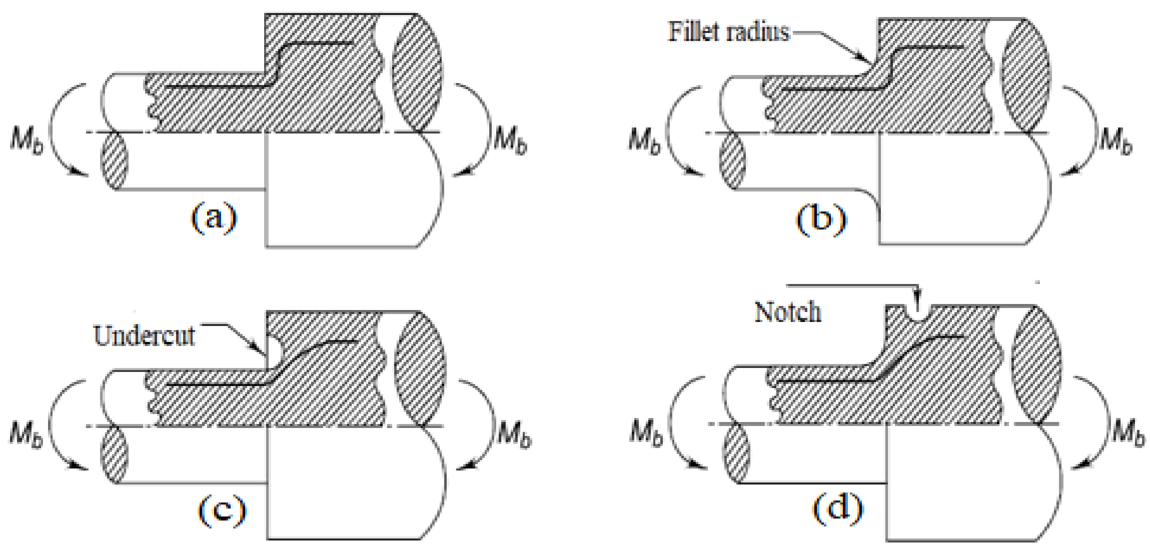
Ball bearings, gears or pulleys are mounted against this shoulder. The shoulder creates a change in cross-section of the shaft, which results in stress concentration.

There are three methods to reduce stress concentration at the base of this shoulder.

(I) By providing fillet radius at shoulder as shown Fig (b).

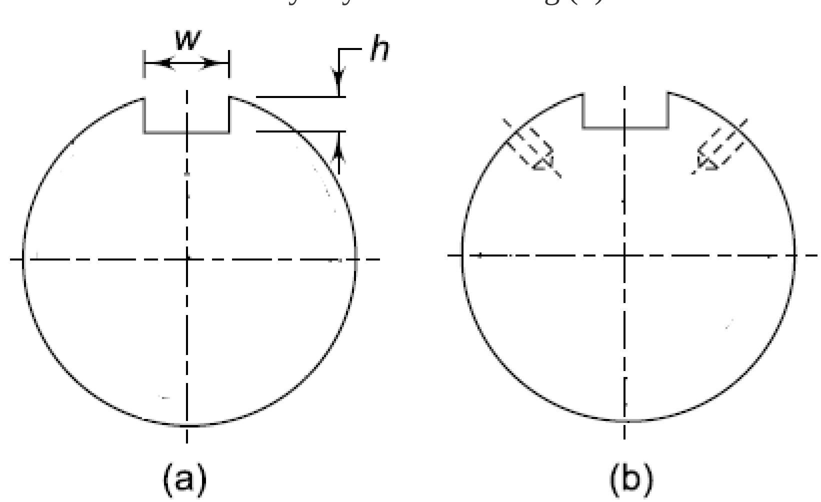
(II) By undercutting the shoulder as illustrated in Fig. (c).

(III) By additional notch at the top of shoulder as shown in Fig. (d).



(c) Drilling Additional Holes for Shaft:

A transmission shaft with a keyway is shown in Fig. (a). The keyway is a discontinuity and results in stress concentration at the corners of the keyway , therefore holes are drilled to minimise the stress concentration at the corners of keyway as shown in fig (b).

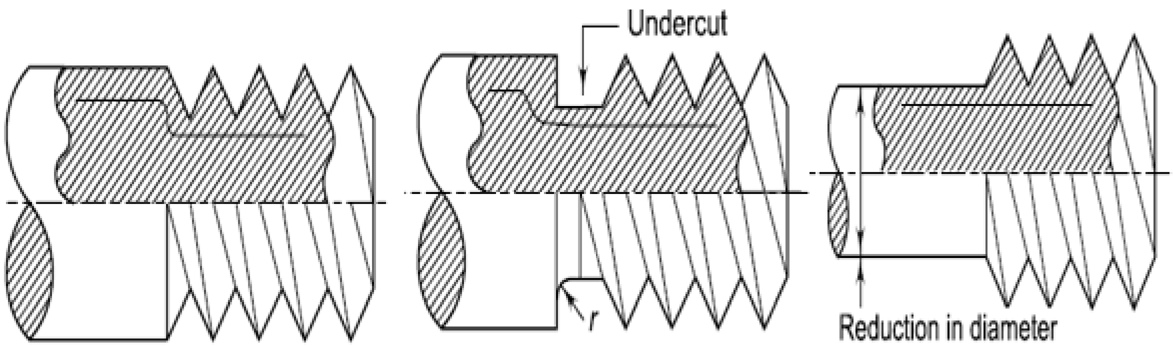


(d) Reduction of Stress Concentration in Threaded Members

A threaded component is shown in Fig. (a). There are three methods to reduce stress concentration

(I) undercutting as shown in Fig. (b)

(II) Reduction in Shank Diameter shown in Fig. (c)



(a) (b) (c)

https://www.youtube.com/watch?v=w4Dxc4LItt4

THEORIES OF FAILURE

When a machine element is subjected to a system of complex stress system, it is important to predict the mode of failure so that the design methodology may be based on a particular failure criterion. Theories of failure are essentially a set of failure criteria developed for the ease of design.

In machine design an element is said to have failed if it ceases to perform its function. There are basically two types of mechanical failure:

(a) **Yielding**- This is due to excessive inelastic deformation rendering the Machine part unsuitable to perform its function. This mostly occurs in ductile materials.

(b) **Fracture**- in this case the component tears apart in two or more parts. This mostly occurs in brittle materials.

There is no sharp line of demarcation between ductile and brittle materials.

However a rough guideline is that if percentage elongation is less than 5% then the material may be treated as brittle and if it is more than 15% then the material is ductile. However, there are many instances when a ductile material may fail by fracture. This may occur if a material is subjected to

(a) Cyclic loading.

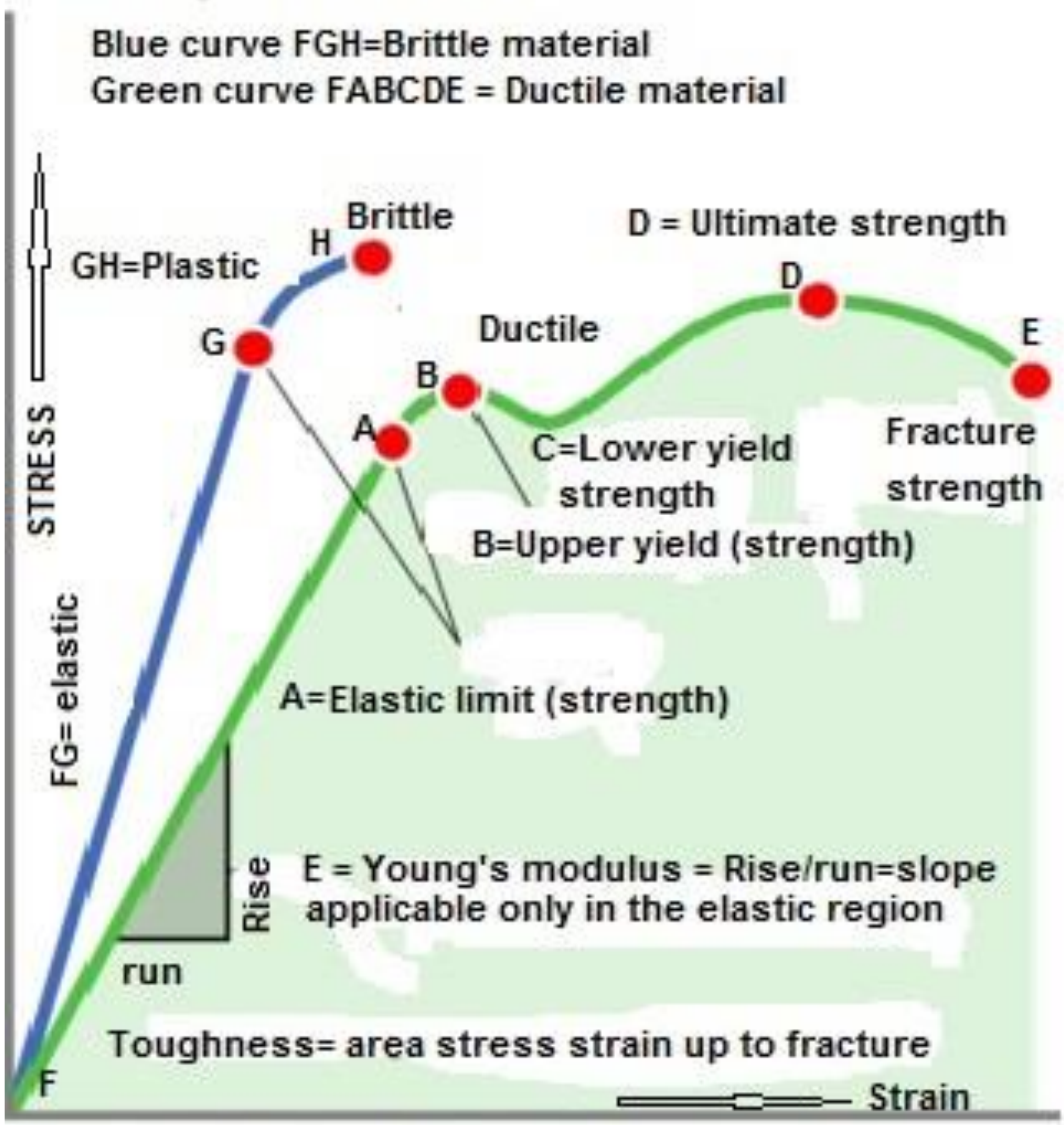
(b) Long term static loading at elevated temperature.

(c) Impact loading.

(d) Work hardening.

(e) Severe quenching.

Yielding and fracture can be visualized in a typical tensile test as shown in the clipping- Typical engineering stress-strain relationship from simple tension tests for same engineering materials are shown in **figure**

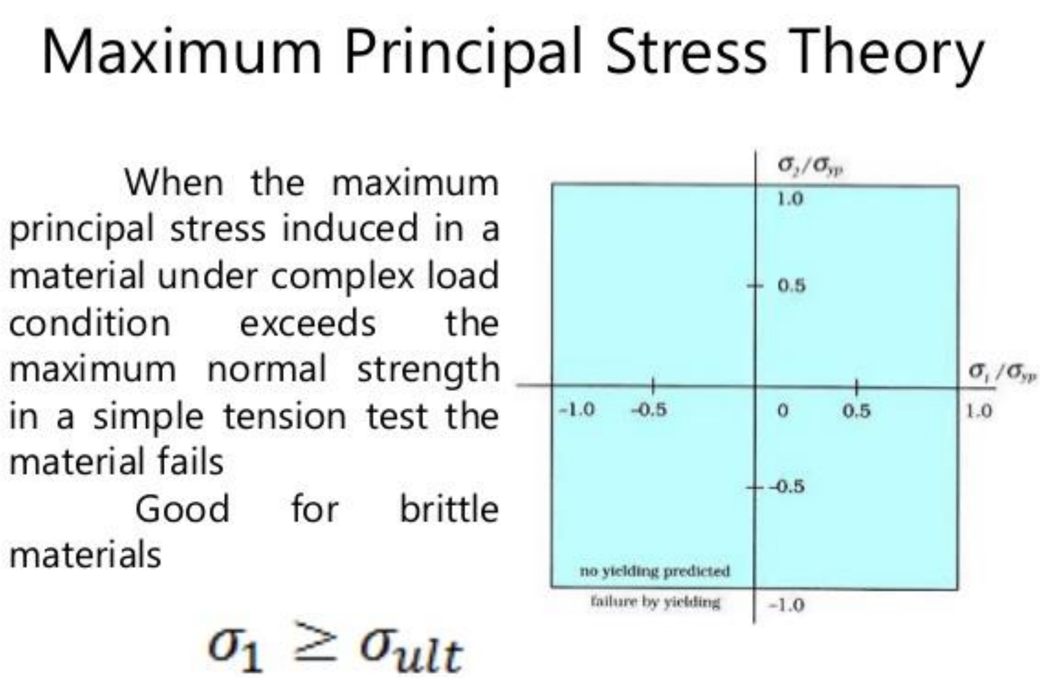


Stress strain curve for ductile brittle materials

Maximum principal stress theory (Rankine theory)

According to this, if one of the principal stresses σ1 (maximum principal stress), σ2 (minimum principal stress) or σ3 exceeds the yield stress, yielding would occur. In a two dimensional loading situation for a ductile material where tensile and compressive yield stress are nearly of same magnitude

$$\sigma_1 = \pm\, \sigma_y$$
$$\sigma_2 = \pm\sigma_y$$



This theory of yielding has very poor agreement with experiment. However, the theory has been used successfully for brittle materials.

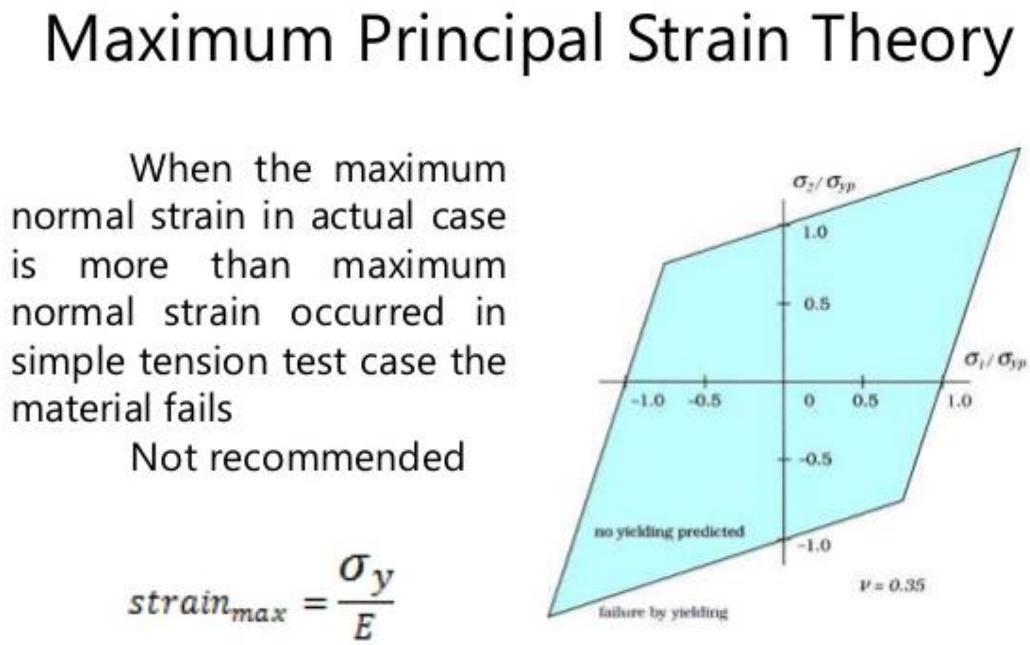
Maximum principal strain theory (St. Venant's theory)

According to this theory, yielding will occur when the maximum principal strain just exceeds the strain at the tensile yield point in either simple tension or compression. If ε_1 and ε_2 are maximum and minimum principal strains corresponding to σ_1 and σ_2, in the limiting case

$$\varepsilon 1 = \frac{1}{E}(\sigma 1 - \mu\sigma 2)$$

$$\varepsilon 2 = \frac{1}{E}(\sigma 2 - \mu\sigma 1)$$

The boundary of a yield surface in this case is thus given as shown in **figure-**

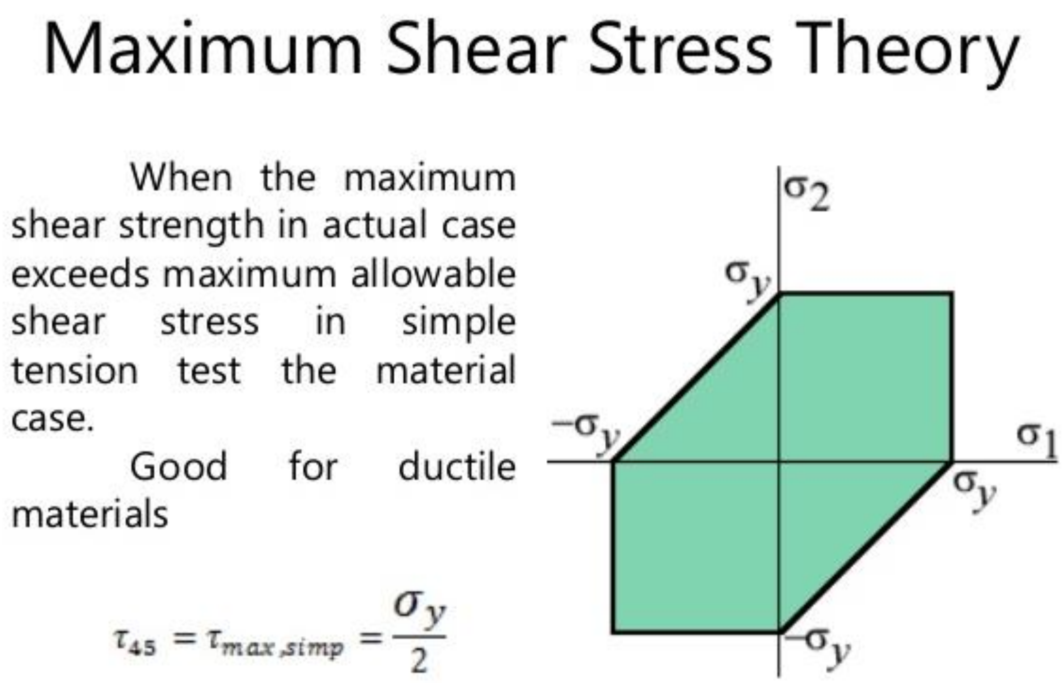


Maximum shear stress theory (Tresca theory)

According to this theory, yielding would occur when the maximum shear stress just exceeds the shear stress at the tensile yield point. At the tensile yield point $\sigma_2 = \sigma_3 = 0$ and thus maximum shear stress is $\sigma_y/2$. This gives us six conditions for a three-dimensional stress situation:

$$\sigma_1 - \sigma_2 = \pm\sigma_y$$
$$\sigma_2 - \sigma_3 = \pm\sigma_y$$
$$\sigma_3 - \sigma_1 = \pm\sigma_y$$

This criterion agrees well with experiment.



Maximum strain energy theory (Beltrami's theory)

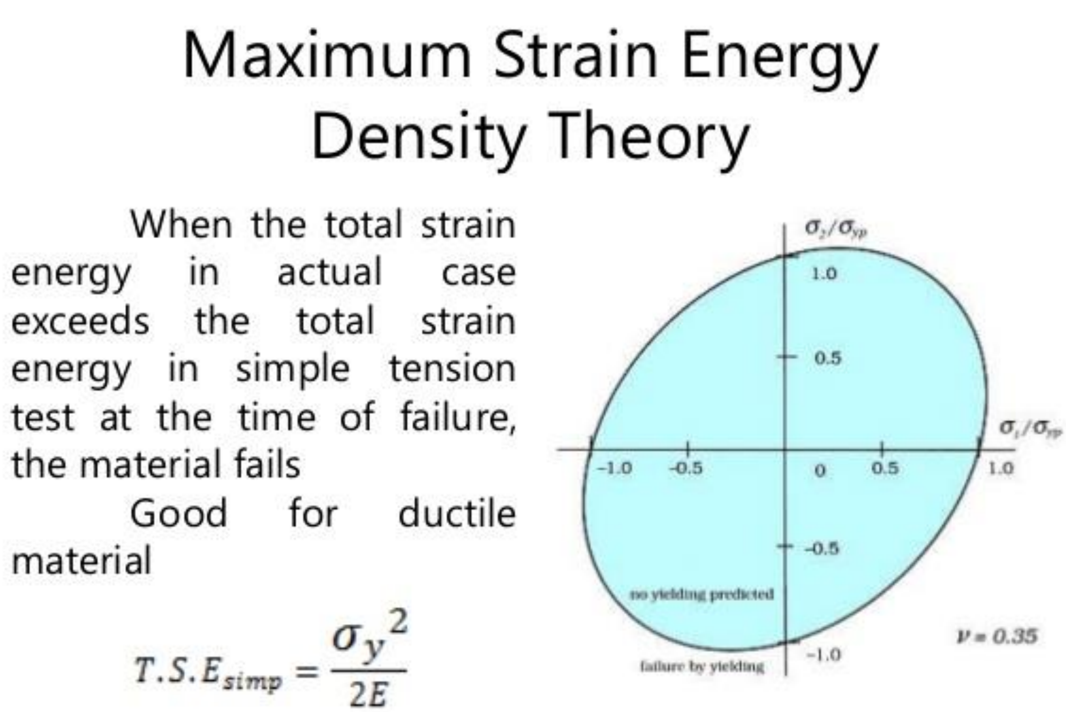
According to this theory failure would occur when the total strain energy absorbed at a point per unit volume exceeds the strain energy absorbed per unit volume at the tensile yield point. This may be given:

$$\frac{1}{2}\left(\sigma_1\varepsilon_1 + \sigma_2\varepsilon_2 + \sigma_3\varepsilon_3\right) = \frac{1}{2}\left(\sigma_y\varepsilon_y\right)$$

Substituting, ε_1, ε_2 , ε_3 and ε_y in terms of stresses we haveType equation here.

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1) = \sigma_y^2$$

This is the equation of an ellipse and the yield surface is shown in **figure:**



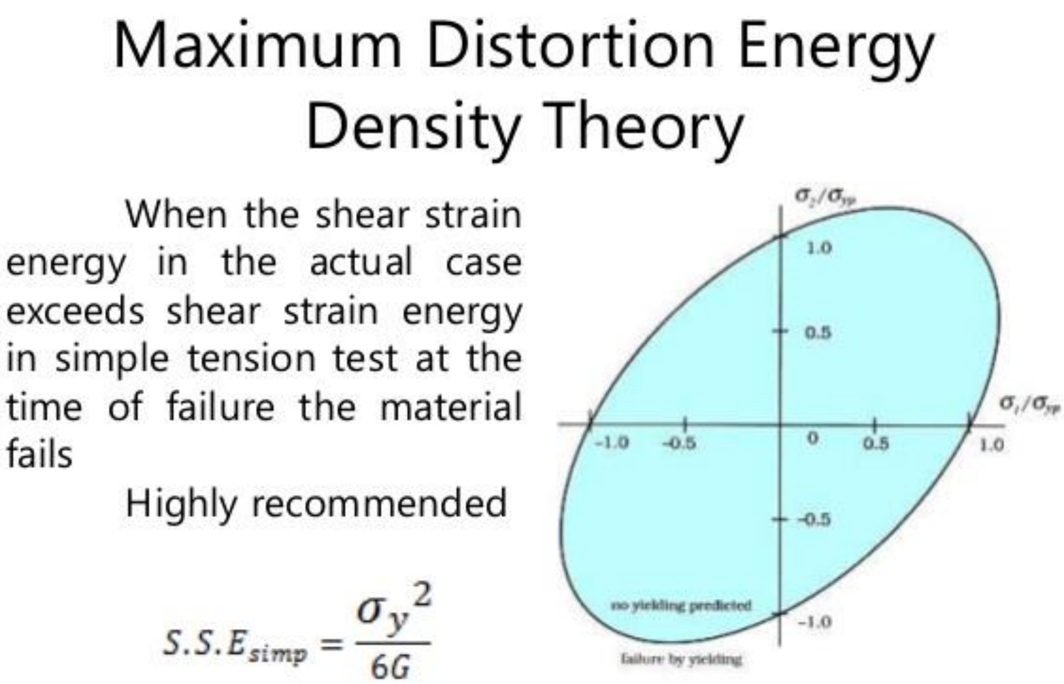
Distortion energy theory (Von Mises yield criterion)

According to this theory yielding would occur when total distortion energy absorbed per unit volume due to applied loads exceeds the distortion energy absorbed per unit volume at the tensile yield point.

The failure criterion is

$$(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 = 2\sigma_y^2$$

This theory agrees very well with experimental results and is widely used for ductile materials.



Shaft

Shaft is a common and important machine element. It is a rotating member, in general, has a circular cross-section and is used to transmit power. The shaft may be hollow or solid. The shaft is supported on bearings and it rotates a set of gears or pulleys for the purpose of power transmission. The shaft is generally acted upon by bending moment, torsion and axial force. Design of shaft primarily involves in determining stresses at critical point in the shaft that is arising due to aforementioned loading. Other two similar forms of a shaft are axle and spindle.
Axle is a non-rotating member used for supporting rotating wheels etc. and do not transmit any torque. Spindle is simply defined as a short shaft. However, design method remains the same for axle and spindle as that for a shaft.

Design considerations for shaft

For the design of shaft following two methods are adopted,

Design based on Strength Criterion:-

In this method, design is carried out so that stress at any location of the shaft should not exceed the material yield stress. However, no consideration for shaft deflection and shaft twist is included.

Design based on Stiffness Criterion:-

Basic idea of design in such case depends on the allowable deflection and twist of the shaft.

Design based on Strength

The stress at any point on the shaft depends on the nature of load acting on it. The stresses which may be present are as follows.

Basic stress equations:

Bending stress

$$\sigma_b = \frac{32M}{\pi d_o^3(1 - k^4)}$$

Where,
M: Bending moment at the point of interest

d_o: Outer diameter of the shaft

k: Ratio of inner to outer diameters of the shaft (k = 0 for a solid shaft because inner diameter is zero)

Stress due to torsion

$$\tau = \frac{16T}{\pi d_o^3(1 - k^4)}$$

Where,
T: Torque on the shaft
τ: Shear stress due to torsion
d_o: Outer diameter of the shaft

k : Ratio of inner to outer diameters of the shaft (k = 0 for a solid shaft because inner diameter is zero)

Maximum shear stress theory

Design of the shaft mostly uses maximum shear stress theory. It states that a machine member fails when the maximum shear stress at a point exceeds the maximum allowable shear stress for the shaft material. Therefore,

$$\tau_{max} = \tau_{allowable} = \frac{1}{2}\sqrt{(\sigma^2} - 4\tau^2)$$

Design based on Stiffness

In addition to the strength, design may be based on stiffness. In the context of shaft, design for stiffness means that the lateral deflection of the shaft and/or angle of twist of the shaft should be within some prescribed limit. Therefore, design for stiffness is based on lateral stiffness and torsional rigidity.

Torsional rigidity

To design a shaft based on torsional rigidity, the limit of angle of twist should be known. The angle of twist is given as follows,

$$\theta = \frac{TL}{GI_p}$$

$$d_o = \sqrt[4]{\frac{584TL}{G(1 - k^4)\theta}}$$

Where,
θ = angle of twist
L = length of the shaft
G = modulus of rigidity
I_p = Polar moment of inertia

Types of shafts

Shaft:
A **ROTATING** member used for the transmission of power.

Axle:-
Generally a **STATIONARY** member used as a support for rotating members such as bearings, wheels, idler gears, etc.

Spindle:-
A short shaft, usually of small diameter, usually rotating, e.g. valve spindle for gate valve, but consider also the **HEADSTOCK SPINDLE** of a lathe, which is quite large and usually has a hole right through its centre

Stub shaft:-
A shaft which is integral with an engine, motor or prime mover and is of suitable size, shape and projection to allow its easy connection to other shafts.

Line shaft (or power transmission shaft):-
A shaft connected to a prime mover which transmits power to a number of machines – now mostly superseded by machines having individual motors.

Jack shaft:-
A short shaft used to connect a prime mover to a machine or another shaft. May also be a short shaft placed as an intermediate shaft between a prime mover and driven machine.

Flexible shaft:-
Permits the transmission of power between two shafts (e.g. motor shaft and machine shaft) whose rotational axes are at an angle or where the angle between the shafts may change.

<u>Design of key</u>

A key is a piece of mild steel inserted between the shaft and hub or boss of the pulley to connect these together in order to prevent relative motion between them. It is always inserted parallel to the axis of the shaft. Keys are used as temporary fastenings and are subjected to considerable crushing and shearing stresses. A keyway is a slot or recess in a shaft and hub of the pulley to accommodate a key.

Types of Keys

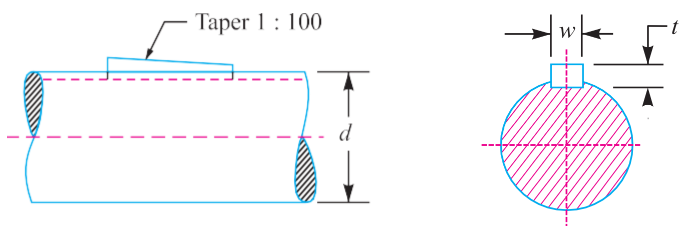
The following types of keys are important from the subject point of view :

1. Sunk keys
2. Saddle keys
3. Tangent keys
4. Round keys
5. Splines

Sunk Keys

The sunk keys are provided half in the keyway of the shaft and half in the keyway of the hub or boss of the pulley. The sunk keys are of the following types:

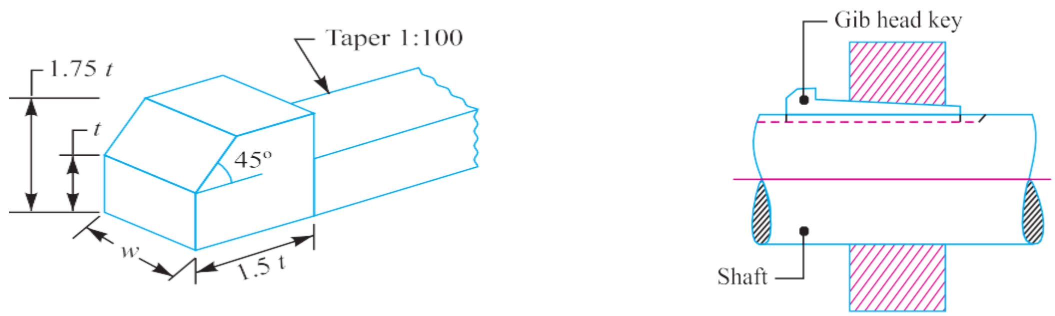
- ***Rectangular sunk key*.** A rectangular sunk key is shown in Fig. The usual proportions of this key are :
Width of key, $w = d / 4$; and thickness of key, $t = 2w / 3 = d / 6$
where d = Diameter of the shaft or diameter of the hole in the hub.
The key has taper 1 in 100 on the top side only.



- **Square sunk key.** The only difference between a rectangular sunk key and a square sunk key is that its width and thickness are equal,
i.e. w = t = d / 4

- ***Parallel sunk key*.** The parallel sunk keys may be of rectangular or square section uniform in width and thickness throughout. It may be noted that a parallel key is a taperless and is used where the pulley, gear or other mating piece is required to slide along the shaft.

- ***Gib-head key.*** It is a rectangular sunk key with a head at one end known as ***gib head.*** It is usually provided to facilitate the removal of key. A gib head key is shown in Fig.

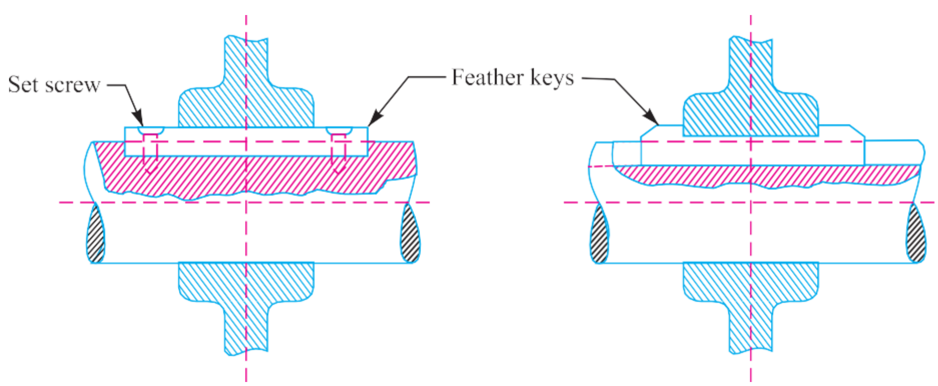


The usual proportions of the gib head key are :

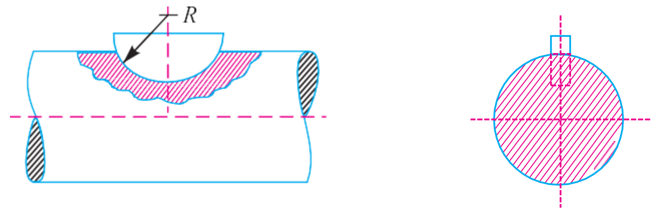
Width, $w = d / 4$;

and thickness at large end, $t = 2w / 3 = d / 6$

- **Feather key.** A key attached to one member of a pair and which permits relative axial movement is known as ***feather key.*** It is a special type of parallel key which transmits a turning moment and also permits axial movement. It is fastened either to the shaft or hub, the key being a sliding fit in the key way of the moving piece.



- ***Woodruff key.*** The woodruff key is an easily adjustable key. It is a piece from a cylindrical disc having segmental cross-section in front view as shown in Fig. 13.4. A woodruff key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made. This key is largely used in machine tool and automobile construction.



The main advantages of a woodruff key are as follows:

1. It accommodates itself to any taper in the hub or boss of the mating piece.
2. It is useful on tapering shaft ends. Its extra depth in the shaft *prevents any tendency to turn over in its keyway.

The disadvantages are:

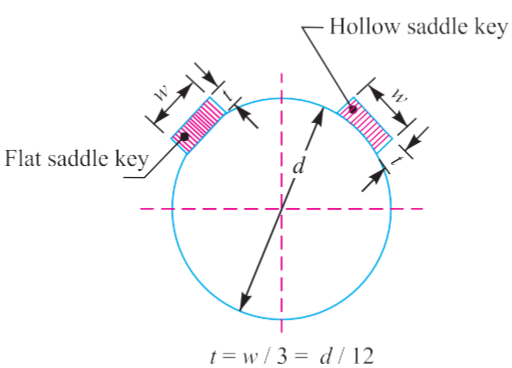
1. The depth of the keyway weakens the shaft.
2. It cannot be used as a feather

Saddle keys

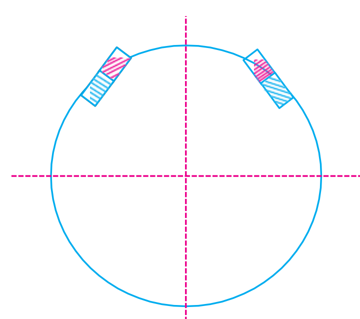
The saddle keys are of the following two types:

1. Flat saddle key,
2. Hollow saddle key.

A ***flat saddle key*** is a taper key which fits in a keyway in the hub and is flat on the shaft as shown in Fig. It is likely to slip round the shaft under load. Therefore it is used for comparatively light loads.



Saddle key



Tangent key

A ***hollow saddle key*** is a taper key which fits in a keyway in the hub and the bottom of the key is shaped to fit the curved surface of the shaft. Since hollow saddle keys hold on by friction, therefore these are suitable for light loads. It is usually used as a temporary fastening in fixing and setting eccentrics, cams etc.

Tangent Keys

The tangent keys are fitted in pair at right angles as shown in Fig. Each key is to withstand torsion in one direction only. These are used in large heavy duty shafts.

Riveted Joints

A rivet is a short cylindrical bar with a head integral to it. The cylindrical portion of 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 ***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.

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.

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.

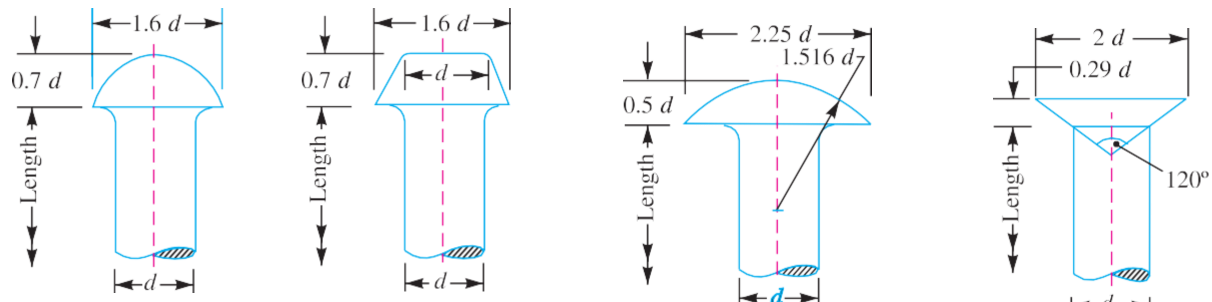
The rivets for general purposes shall be manufactured from steel conforming to the following Indian Standards :

- IS : 1148–1982 (Reaffirmed 1992) – Specification for hot rolled rivet bars (up to 40 mm diameter) for structural purposes; or
- IS : 1149–1982 (Reaffirmed 1992) – Specification for high tensile steel rivet bars for structural purposes.

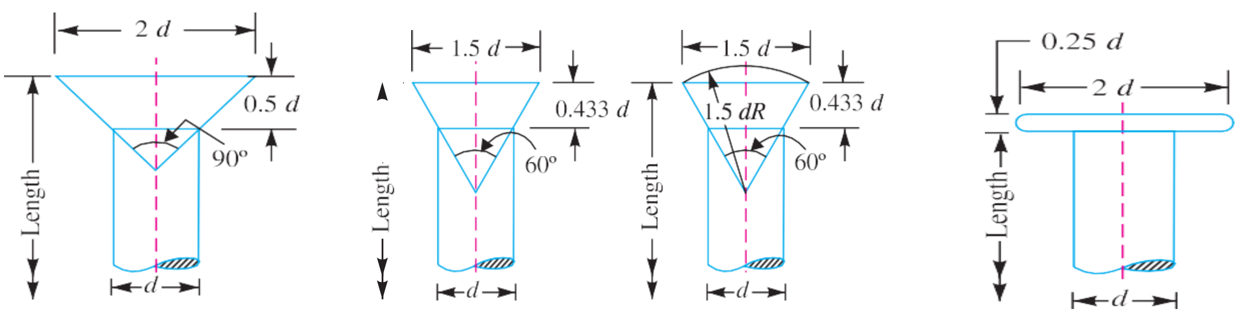
Types of Rivet Heads

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. 9.3, according to IS : 2155 – 1982 (Reaffirmed 1996)

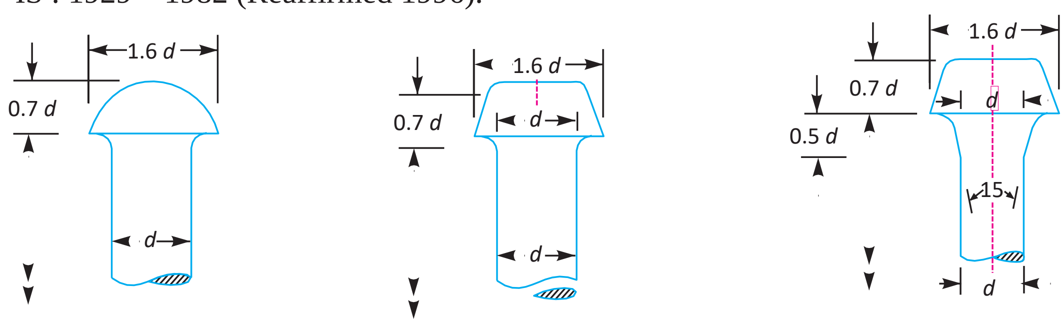


(*a*) Snap head. (*b*) Pan head. (*c*) Mushroom head. (*d*) Counter sunk head 120°.



(*e*) Flat counter sunk head 90°. (*f*) Flat counter sunk head 60°. (*g*) Round counter sunk head 60°. (*h*) Flat head.

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).



(a) Snap head (b) Pan head. (c) Pan head with tapered neck.

Types of Riveted Joints

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

1. Lap joint, and

2. Butt joint

Lap Joint

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

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:

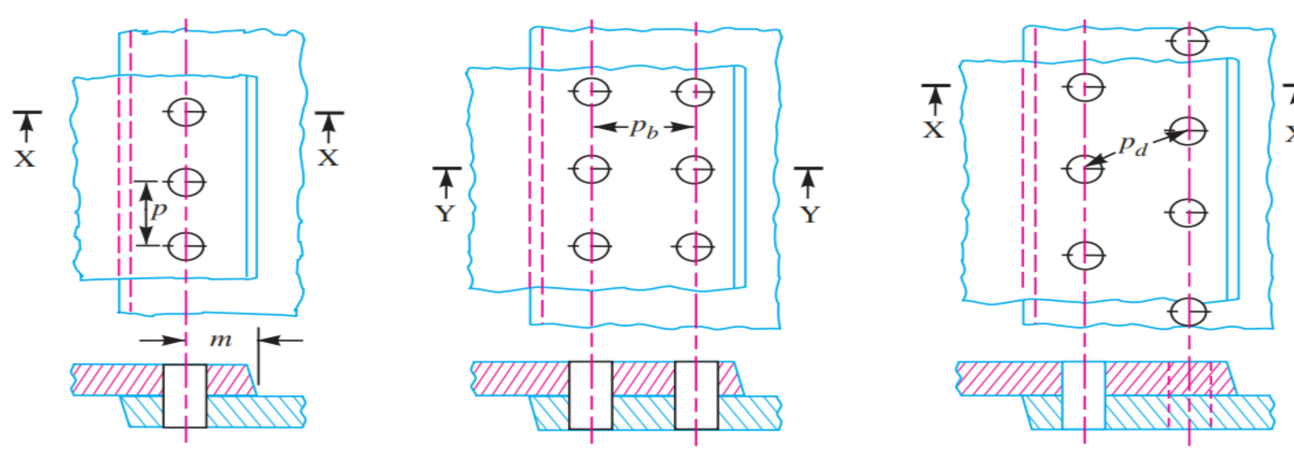
1. Single strap butt joint, and **2.** Double strap butt joint.

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

In a ***double strap butt joint,*** the edges of the main plates butt against each other and two cover plates are placed on both sides of the main plates and then riveted together.

A ***single riveted joint*** is that in which there is a single row of rivets in a lap joint and there is a single row of rivets on each side in a butt joint.

A ***double riveted joint*** is that in which there are two rows of rivets in a lap joint and there are two rows of rivets on each side in a butt joint.



(*a*) Single riveted lap joint.

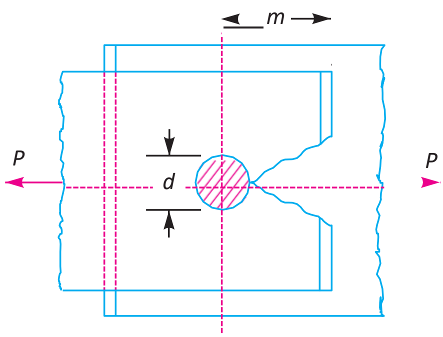
(*b*) Double riveted lap joint (Chain riveting).

(*c*) Double riveted lap joint (Zig-zag riveting).

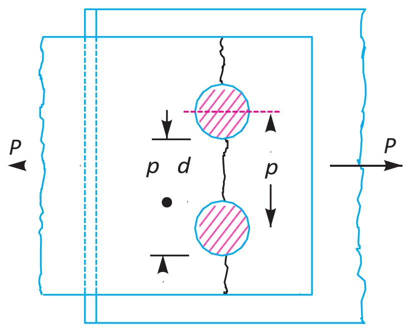
Failures of a Riveted Joint

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. This can be avoided by keeping the margin, $m = 1.5d$, where d is the diameter of the rivet hole.



Tearing of the plate at an edge.



Tearing of the plate across the

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 p = Pitch of the rivets,

d = Diameter of the rivet hole,

t = Thickness of the plate, and

σ_t = Permissible tensile stress for the plate material.

We know that tearing area per pitch length,

$A_t = (p - d)\, t$

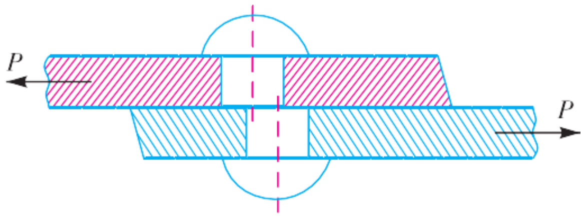
Therefore, Tearing resistance or pull required to tear off the plate per pitch length,

$P_t = A_t . \sigma_t = (p - d)\, t . \sigma_t$

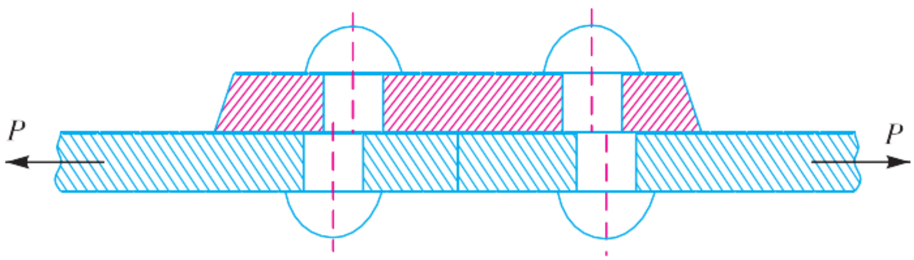
When the tearing resistance (P_t) is greater than the applied load (P) per pitch length, then this type of failure will not occur.

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

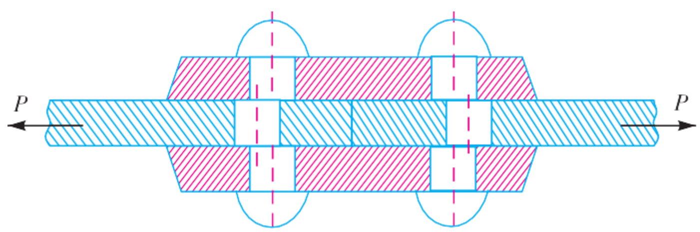
It may be noted that the rivets are in *single shear in a lap joint and in a single cover butt joint, as shown in Fig. But the rivets are in double shear in a double cover butt joint as shown in Fig. The resistance offered by a rivet to be sheared off is known as ***shearing resistance*** or ***shearing strength*** or ***shearing value*** of the rivet.



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



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



Shearing off a rivet in double cover butt joint.

Shearing resistance or pull required to shear off the rivet per pitch length :-

$P_s = n\frac{\pi}{4}d^2 * \tau$ ………………………………….for single shear

$P_s = n * 2 * \frac{\pi}{4}d^2 * \tau$ ………………………..……….for double shear

4. **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.

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

Let d = Diameter of the rivet hole,

t = Thickness of the plate,

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

n = Number of rivets per pitch length under crushing.

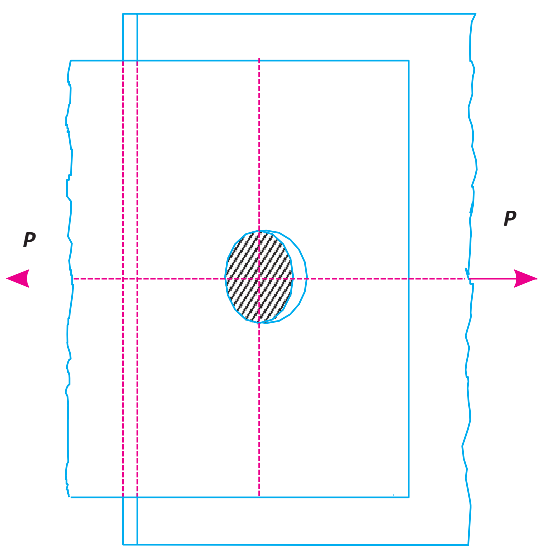
We know that crushing area per rivet (*i.e.* projected area per rivet)

$\boldsymbol{A_C = d.t}$

Total crushing area = $\mathbf{n.d.t}$

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

$\boldsymbol{P_C = n.d.t.\sigma_C}$



Crushing of a rivet.

Strength of a Riveted Joint

The strength of a joint may be defined as the maximum force, which it can transmit, without causing it to fail. We have seenthat P_t, P_S and P_C are the pulls required to tear off the plate, shearing off the rivet and crushing off the rivet. A little consideration will show that if we go on increasing the pull on a riveted joint, it will fail when the least of these three pulls is reached, because a higher value of the other pulls will never reach since the joint has failed, either by tearing off the plate, shearing off the rivet or crushing off the rivet.

If the joint is ***continuous*** as in case of boilers, the strength is calculated ***per pitch length*.** But if the joint is ***small,*** the strength is calculated for the ***whole length*** of the plate.

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.

We have already discussed that strength of the riveted joint

= Least of P_t, P_S and P_C

Strength of the un-riveted or solid plate per pitch length,

$\boldsymbol{P = p \times t \times \sigma_t}$

Efficiency of the riveted joint,

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

where,

p = Pitch of the rivets,

t = Thickness of the plate, and

σ_t = Permissible tensile stress of the plate material

Welded Joints

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). The latter method is extensively used because of greater speed of welding.

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.

Welding Processes

The welding processes may be broadly classified into the following two groups:

1. Welding processes that use heat alone *e.g.* fusion welding.
2. Welding processes that use a combination of heat and pressure *e.g.* forge welding.

Fusion Welding

In case of fusion welding, the parts to be jointed are held in position while the molten metal is supplied to the joint. The molten metal may come from the parts themselves (i.e. parent metal) or filler metal which normally have the composition of the parent metal. The joint surface become plastic or even molten because of the heat from the molten filler metal or other source. Thus, when the molten metal solidifies or fuses, the joint is formed.

The fusion welding, according to the method of heat generated, may be classified as:

1. Thermit welding
2. Gas welding
3. Electric arc welding.

Thermit Welding

In thermit welding, a mixture of iron oxide and aluminium called ***thermit*** is ignited and the iron oxide is reduced to molten iron. The molten iron is poured into a mould made around the joint and fuses with the parts to be welded. A major advantage of the thermit welding is that all parts of weld section are molten at the same time and the weld cools almost uniformly. This results in a minimum problem with residual stresses. It is fundamentally a melting and casting process.

Gas Welding

A gas welding is made by applying the flame of an oxy-acetylene or hydrogen gas from a welding torch upon the surfaces of the prepared joint. The intense heat at the white cone of the flame heats up the local surfaces to fusion point while the operator manipulates a welding rod to supply the metal for the weld. A flux is being used to remove the slag. Since the heating rate in gas welding is slow, therefore it can be used on thinner materials.

Electric Arc Welding

In electric arc welding, the work is prepared in the same manner as for gas welding. In this case the filler metal is supplied by metal welding electrode. The operator, with his eyes and face protected, strikes an arc by touching the work of base metal with the electrode.

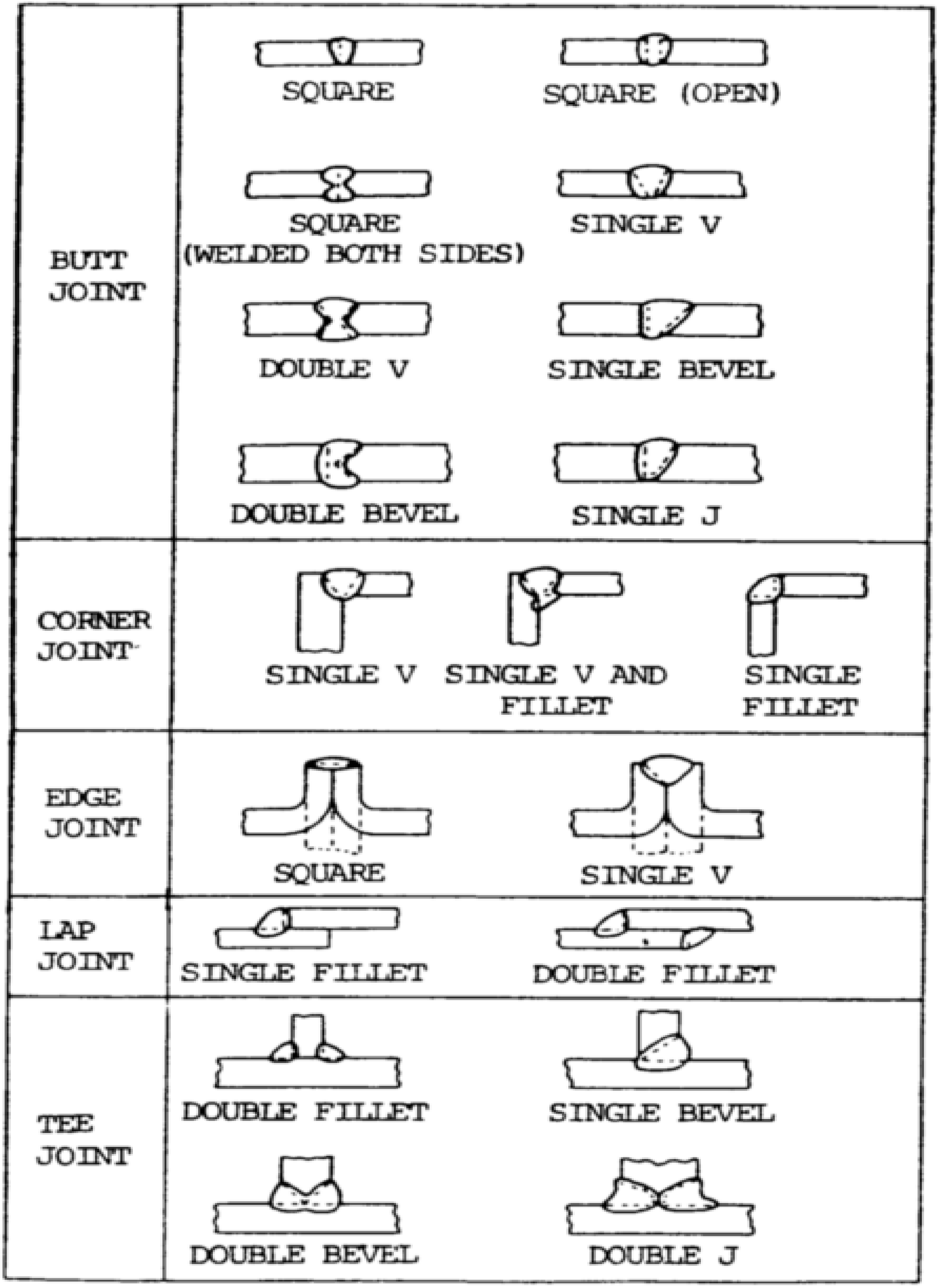
Forge Welding

In forge welding, the parts to be jointed are first heated to a proper temperature in a furnace or forge and then hammered. This method of welding is rarely used now-a-days. An **e*lectric-resistance welding*** is an example of forge welding.

Types of Welded Joints

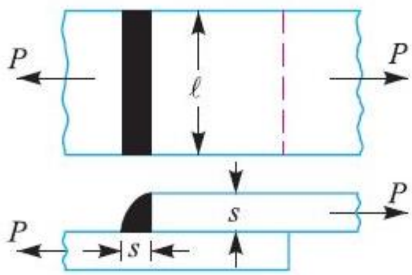
Following two types of welded joints are important from the subject point of view:

1. Lap joint or fillet joint
2. Butt joint

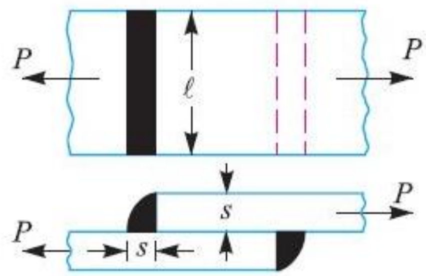

BUTT JOINT
SQUARE
SQUARE (OPEN)
SQUARE (WELDED BOTH SIDES)
SINGLE V
DOUBLE V
SINGLE BEVEL
DOUBLE BEVEL
SINGLE J
CORNER JOINT
SINGLE V
SINGLE V AND FILLET
SINGLE FILLET
EDGE JOINT
SQUARE
SINGLE V
LAP JOINT
SINGLE FILLET
DOUBLE FILLET
TEE JOINT
DOUBLE FILLET
SINGLE BEVEL
DOUBLE BEVEL
DOUBLE J

Strength of Transverse Fillet Welded Joints

We have already discussed that 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.

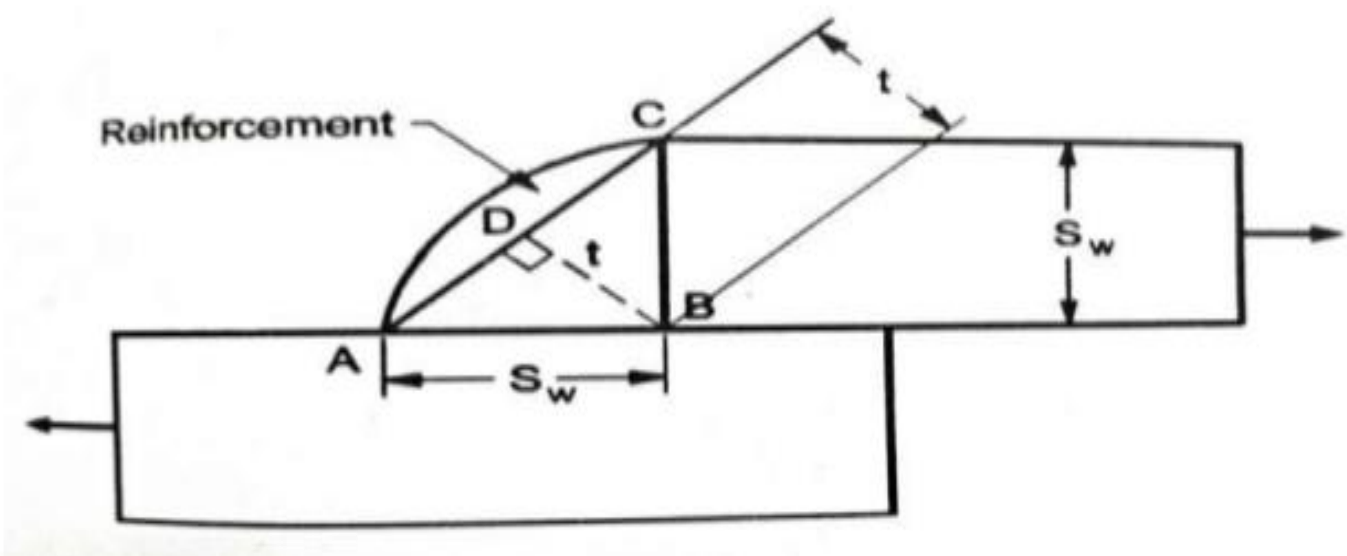


(*a*) Single transverse fillet weld.



(*b*) Double transverse fillet weld.

In order to determine the strength of the fillet joint, it is assumed that the section of fillet is a right angled triangle *ABC* with hypotenuse *AC* making equal angles with other two sides *AB* and *BC*. The enlarged view of the fillet is shown in Fig. 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.



Let, t = Throat thickness (BD),

S_w = Leg or size of weld,

= Thickness of plate, and

l = Length of weld

$t = s \times \sin 45° = 0.707\ s$

Minimum area of the weld or throat area,

A = Throat thickness × Length of weld

$= t \times l = 0.707\ s \times l$

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

P = Throat area × Allowable tensile stress = $0.707\ s \times l \times \sigma_t$

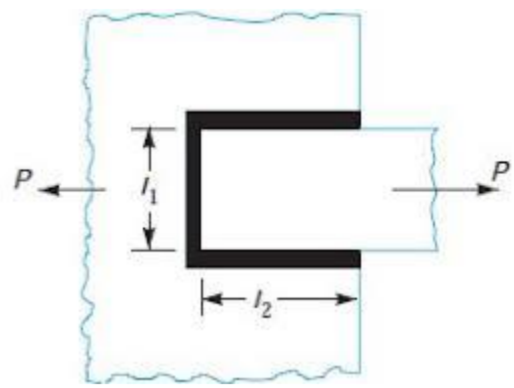
and tensile strength of the joint for double fillet weld,

$P = 2 \times 0.707\ s \times l \times \sigma_t = 1.414\ s \times l \times \sigma_t$

Strength of Combined Parallel and Transverse Fillet Welded Joints:-

The parallel fillet welded joints are designed for shear strength. Consider a double parallel fillet welded joint as shown in Fig. We have already discussed in the previous article, that the minimum area of weld or the throat area

$\boldsymbol{A = 0.707\ s \times l}$



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

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

and shear strength of the joint for double parallel fillet weld

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

Cotter Joint

A cotter is a flat wedge shaped piece of rectangular cross-section and its width is tapered (either on one side or both sides) from one end to another for an easy adjustment. The taper varies from 1 in 48 to 1 in 24 and it may be increased up to 1 in 8, if a locking device is provided. The locking device may be a taper pin or a set screw used on the lower end of the cotter.

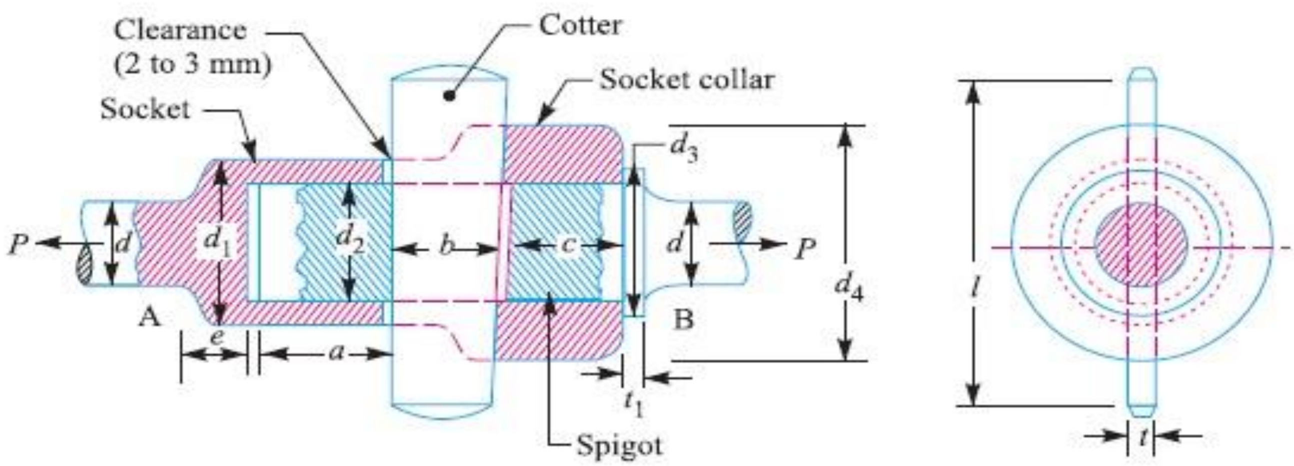
Types of Cotter Joints

Following are the three commonly used cotter joints to connect two rods by a cotter:

1. Socket and spigot cotter joint
2. Sleeve and cotter joint
3. Gib and cotter joint

Socket and Spigot Cotter Joint:-

In a socket and spigot cotter joint, one end of the rods (say *A*) is provided with a socket type of end as shown in Fig. and the other end of the other rod (say *B*) is inserted into a socket. The end of the rod which goes into a socket is also called ***spigot.*** *A* rectangular hole is made in the socket and spigot. *A* cotter is then driven tightly through a hole in order to make the temporary connection between the two rods. The load is usually acting axially, but it changes its direction and hence the cotter joint must be designed to carry both the tensile and compressive loads. The compressive load is taken up by the collar on the spigot.



Design of Different Type of Cotter Joint:-

Let P = Load carried by the rods,

d = Diameter of the rods,

d_1 = Outside diameter of socket,

d_2 = Diameter of spigot or inside diameter of socket,

d_3 = Outside diameter of spigot collar,

t_1 = Thickness of spigot collar,

d_4 = Diameter of socket collar,

c = Thickness of socket collar,

b = Mean width of cotter,

t = Thickness of cotter,

l = Length of cotter,

a = Distance from the end of the slot to the end of rod,

σ_t = Permissible tensile stress for the rods material,

τ = Permissible shear stress for the cotter material, and

σ_t = Permissible crushing stress for the cotter material.

The dimensions for a socket and spigot cotter joint may be obtained by considering the various modes of failure as discussed below:

1. **Failure of the rods in tension**

The rods may fail in tension due to the tensile load P.

Area resisting tearing = $\frac{\pi}{4} * d^2$

Tearing strength of the rods = $\frac{\pi}{4} * d^2 * \sigma_t$

Equating this to load P = $\frac{\pi}{4} * d^2 * \sigma_t$

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

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

Since the weakest section of the spigot is that section which has a slot in it for the cotter, as shown in Fig. therefore

Area resisting tearing of the spigot across the slot = $\frac{\pi}{4} * (d_2)^2 - d_2 * t$

Tearing strength of the spigot across the slot = $(\frac{\pi}{4} * (d_2)^2 - d_2 * t)\sigma_t$

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

3. **Failure of the rod or cotter in crushing**

We know that the area that resists crushing of a rod or cotter

A = $d_2 * t$

P = $d_2 * t * \sigma_C$

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

4. **Failure of the socket in tension across the slot**

The resisting area of the socket across the slot

= $\frac{\pi}{4}\lfloor(d_1)^2 - (d_2)^2 - (d_1 - d_2)t\rfloor$

P = $\frac{\pi}{4}\lfloor(d_1)^2 - (d_2)^2 - (d_1 - d_2)t\rfloor \sigma_t$

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

5. **Failure of cotter in shear**

Since the cotter is in double shear, therefore shearing area of the cotter

= 2b*t
P = 2b*t* τ
From this equation, width of cotter (b) is determined

6. **Failure of the socket collar in crushing**

Area that resists crushing of socket collar

= $(d_4 - d_2)t$

P = $(d_4 - d_2)t* \sigma_C$

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

7. **Failure of socket end in shearing**

Since the socket end is in double shear, therefore area that resists shearing of socket collar

= $2(d_4 - d_2)c$

P = $2(d_4 - d_2)c * \tau$

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

8. **Failure of rod end in shear**

Since the rod end is in double shear, therefore the area resisting shear of the rod end

$= 2 * a * d_2$

P = $(2 * a * d_2)\tau$

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

9. **Failure of spigot collar in crushing**

Area that resists crushing of the collar

$= \frac{\pi}{4}\lfloor(d_3)^2 - (d_2)^2\rfloor$

P = $\frac{\pi}{4}\lfloor(d_3)^2 - (d_2)^2\rfloor\ \sigma_C$

10. **Failure of the spigot collar in shearing**

Area that resists shearing of the collar

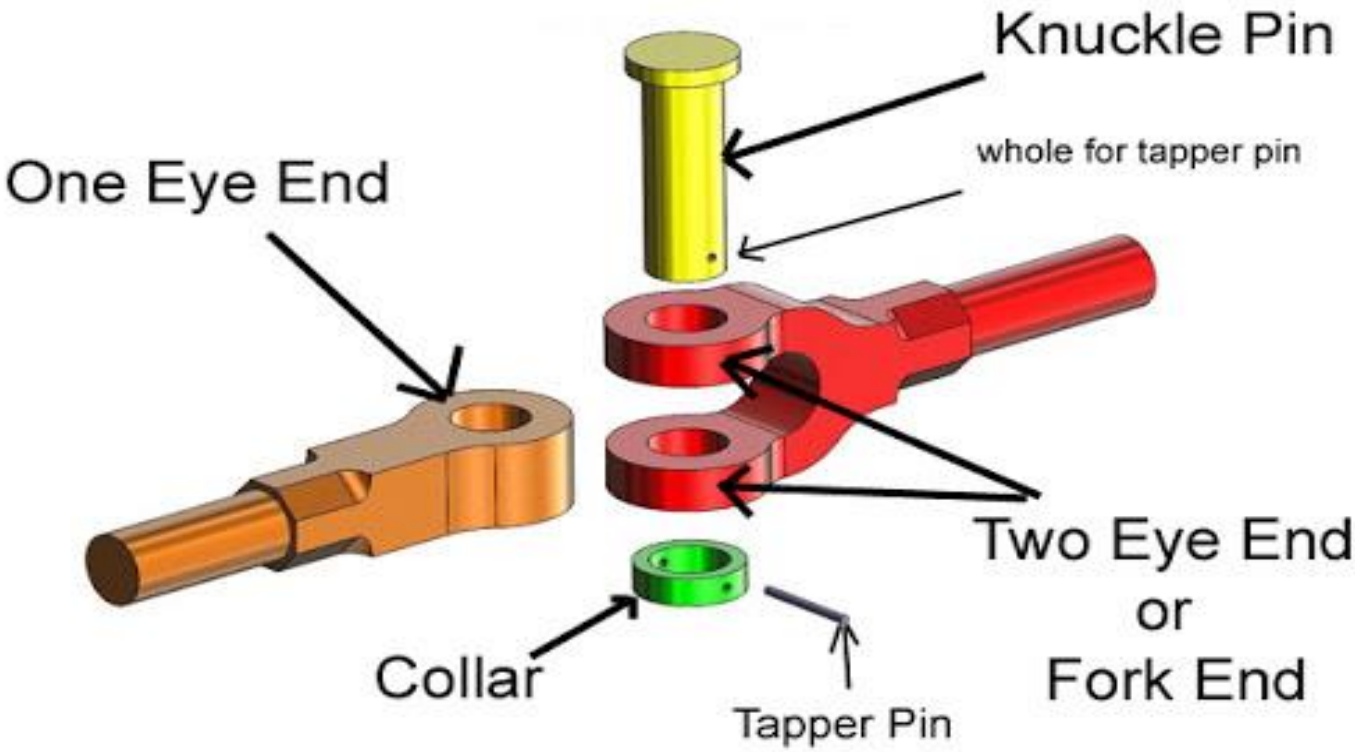
$= \pi d_2 * t_1$

P = $\pi d_2 * t_1 * \tau$

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

<u>Knuckle Joint</u>

A knuckle joint is used to connect two rods which are under the action of tensile loads. However, if the joint is guided, the rods may support a compressive load. A knuckle joint may be readily disconnected for adjustments or repairs. Its use may be found in the link of a cycle chain, tie rod joint for roof truss, valve rod joint with eccentric rod, pump rod joint, tension link in bridge structure and lever and rod connections of various types.



Dimensions of Various Parts of the Knuckle Joint:-

If *d* is the diameter of rod, then diameter of pin

$d_1 = d$,

Outer diameter of eye,

$d_2 = 2d$,

Diameter of knuckle pin head and collar,

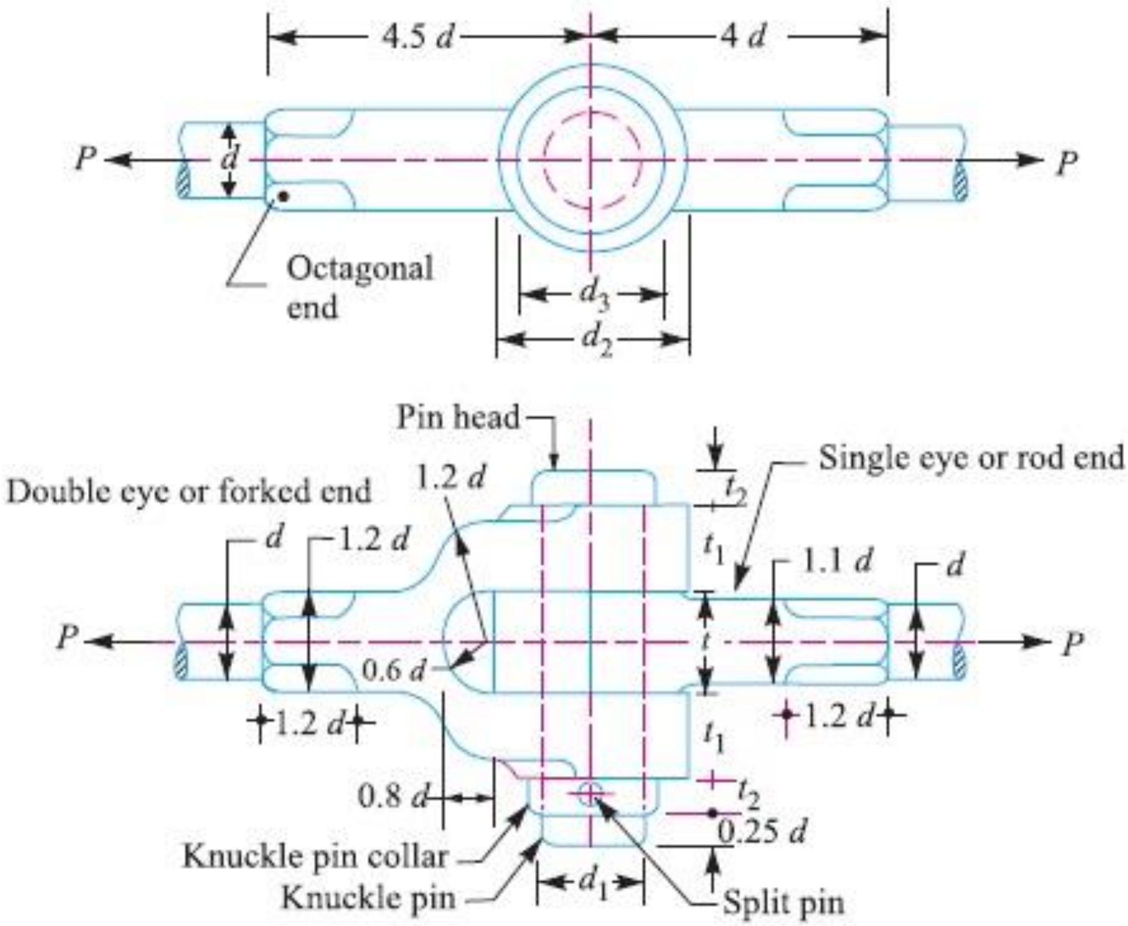
$d_3 = 1.5d$

Thickness of single eye or rod end,

$t = 1.25\,d$

Thickness of fork, $t_1 = 0.75\,d$

Thickness of pin head, $t_2 = 0.5\,d$



Methods of Failure of Knuckle Joint:-

Consider a knuckle joint as shown in Fig. 12.16.

Let P = Tensile load acting on the rod,

d = Diameter of the rod,

d_1 = Diameter of the pin,

d_2 = Outer diameter of eye,

t = Thickness of single eye,

t_1 = Thickness of fork.

σ_t, τ and σ_c = Permissible stresses for the joint material in tension, shear and crushing respectively.

In determining the strength of the joint for the various methods of failure, it is assumed that there is no stress concentration, and the load is uniformly distributed over each part of the joint.

1. **Failure of the solid rod in tension**

Since the rods are subjected to direct tensile load, therefore tensile strength of the rod

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

From this equation, diameter of the rod (d) is obtained.

2. **Failure of the knuckle pin in shear**

Since the pin is in double shear, therefore cross-sectional area of the pin under shearing

$= 2\frac{\pi}{4} * d^2$

$P = 2\frac{\pi}{4} * d^2 * \tau$

From this equation, diameter of the knuckle pin (d_1) is obtained

3. **Failure of the single eye or rod end in tension**

The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing

$= (d_1 - d_2)t$

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

From this equation, the induced tensile stress (σ_t) for the single eye or rod end may be checked. In case the induced tensile stress is more than the allowable working stress, then increase the outer diameter of the eye (d_2).

4. **Failure of the single eye or rod end in shearing**

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing

$= (d_1 - d_2)t$

$P = (d_1 - d_2)t * \tau$

From this equation, the induced shear stress (τ) for the single eye or rod end may be checked.

5. **Failure of the single eye or rod end in crushing**

The single eye or pin may fail in crushing due to the tensile load. We know that area resisting crushing

$A = d_1 * t$

P = $d_1 * t * \sigma_C$

From this equation, the induced crushing stress (σ_C) for the single eye or pin may be checked. In case the induced crushing stress in more than the allowable working stress, then increase the thickness of the single eye (t).

6. Failure of the forked end in tension

The forked end or double eye may fail in tension due to the tensile load. We know that area resisting tearing

= $(d_1 - d_2)2t_1$

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

From this equation, the induced tensile stress for the forked end may be checked.

7. Failure of the forked end in shear

The forked end may fail in shearing due to the tensile load. We know that area resisting shearing

= $(d_1 - d_2)2t_1$

P = $(d_1 - d_2)2t_1 * \tau$

From this equation, the induced shear stress for the forked end may be checked. In case, the induced shear stress is more than the allowable working stress, then thickness of the fork (t_1) is increased.

8. Failure of the forked end in crushing

The forked end or pin may fail in crushing due to the tensile load. We know that area resisting crushing

= $d_1 * 2t_1$

P = $d_1 * 2t_1 * \sigma_C$

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

Flange coupling

Shafts are usually available up to 7 metres length due to inconvenience in transport. In order to have a greater length, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

Shaft couplings are used in machinery for several purposes, the most common of which are the following:

- To provide for the connection of shafts of units that is manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.
- To provide for misalignment of the shafts or to introduce mechanical flexibility.
- To reduce the transmission of shock loads from one shaft to another.
- To introduce protection against overloads.
- It should have no projecting parts.

Shaft couplings are divided into two main groups as follows:

Rigid coupling. It is used to connect two shafts which are perfectly aligned. Following types of rigid coupling are important from the subject point of view:

- Sleeve or muff coupling.
- Clamp or split-muff or compression coupling, and
- Flange coupling.

Flexible coupling. It is used to connect two shafts having both lateral and angular misalignment. Following types of flexible coupling are important from the subject point of view:

- Bushed pin type coupling,
- Universal coupling, and
- Oldham coupling

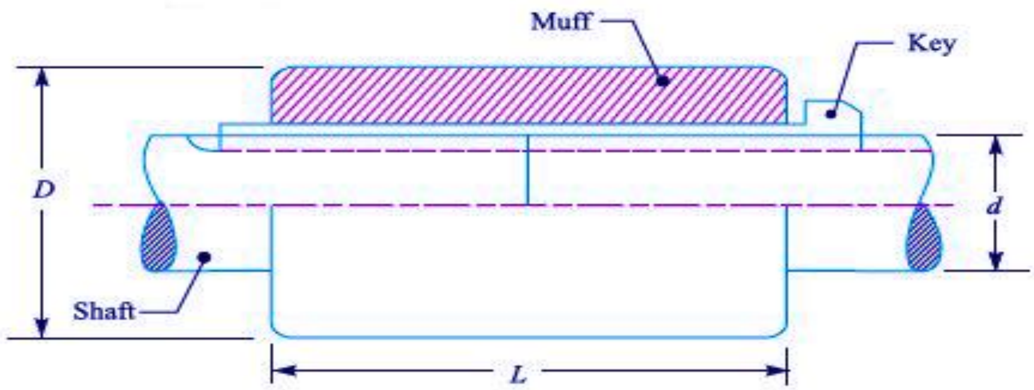
Sleeve or Muff-coupling:-

It is the simplest type of rigid coupling, made of cast iron. It consists of a hollow cylinder whose inner diameter is the same as that of the shaft. It is fitted over the ends of the two shafts by means of a gib head key, as shown in Fig. The power is transmitted from one shaft to the other shaft by means of a key and a sleeve. It is, therefore, necessary that all the elements must be strong enough to transmit the torque. The usual proportions of a cast iron sleeve coupling are as follows:

Outer diameter of the sleeve, $D = 2d + 13$ mm and length of the sleeve,

$L = 3.5\ d$

Where d is the diameter of the shaft



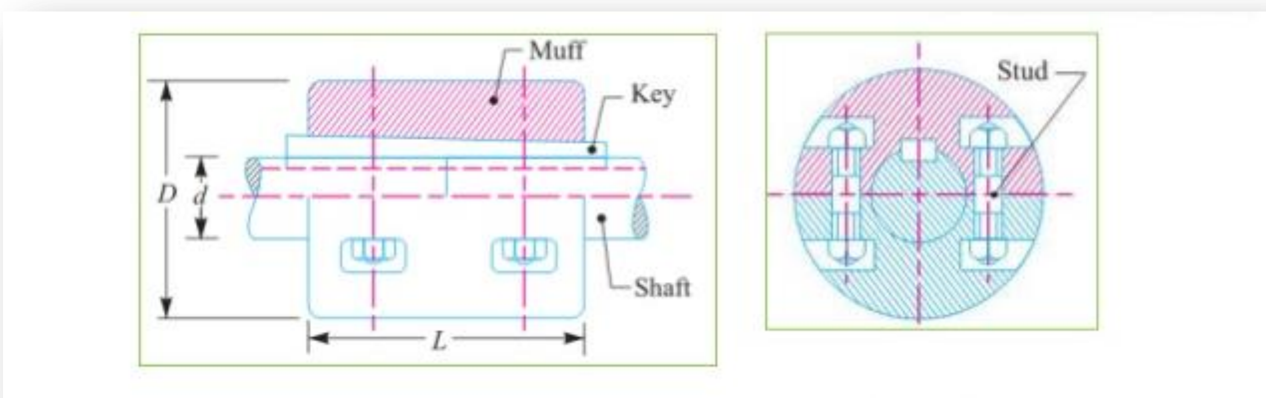
Clamp or Compression Coupling:-

It is also known as ***split muff coupling.*** In this case, the muff or sleeve is made into two halves and are bolted together as shown in Fig. The halves of the muff are made of cast iron. The shaft ends are made to a butt each other and a single key is fitted directly in the keyways of both the shafts. One-half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts. The number of bolts may be two, four or six. The nuts are recessed into the bodies of the muff castings. This coupling may be used for heavy duty and moderate speeds. The advantage of this coupling is that the position of the shafts need not be changed for assembling or disassembling of the coupling. The usual proportions of the muff for the clamp or compression coupling are:

Diameter of the muff or sleeve, $D = 2d + 13$ mm

Length of the muff or sleeve, $L = 3.5\ d$

Where d = Diameter of the shaft



Flange Coupling

A flange coupling usually applies to a coupling having two separate cast iron flanges. Each flange is mounted on the shaft end and keyed to it. The faces are turned up at right angle to the axis of the shaft. One of the flange has a projected portion and the other flange has a corresponding recess. This helps to bring the shafts into line and to maintain alignment. The two flanges are coupled together by means of bolts and nuts. The flange coupling is adopted to heavy loads and hence it is used on large shaft- ing. The flange couplings are of the following three types:

Unprotected type flange coupling. In an unprotected type flange coupling, as shown in Fig. each shaft is keyed to the boss of a flange with a counter sunk key and the flanges are coupled together by means of bolts. Generally, three, four or six bolts are used.

If d is the diameter of the shaft or inner diameter of the hub, then Outside diameter of hub,

$D = 2\,d$

Length of hub, $L = 1.5\,d$

Pitch circle diameter of bolts,

$D_1 = 3d$

Outside diameter of flange,

$D_2 = D_1 + (D_1 - D) = 2\,D_1 - D = 4\,d$

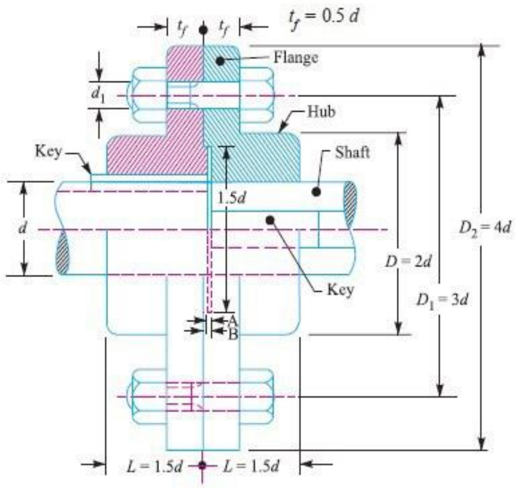
Thickness of flange, $t_f = 0.5\,d$

Number of bolts

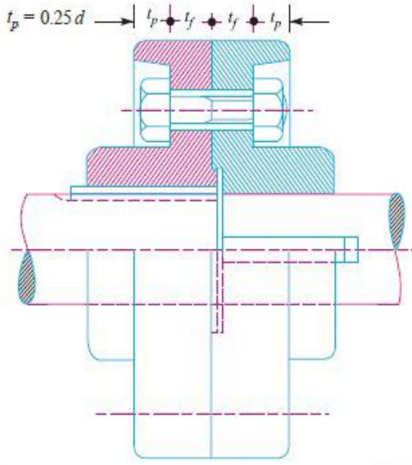
= 3, for d upto 40 mm

= 4, for d upto 100 mm

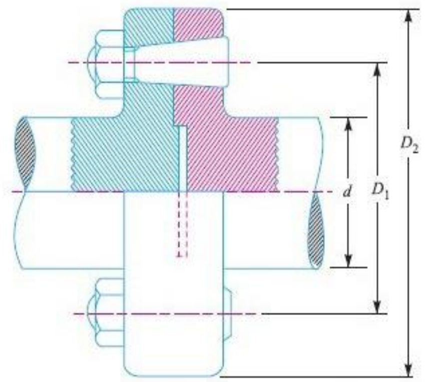
= 6, for d upto 180 mm



UNPROTECTED TYPE FLANGE COUPLING



PROTECTED TYPE FLANGE COUPLING



Thickness of flange	$= d/3$
Taper of bolt	= 1 in 20 to 1 in 4
Pitch circle diameter of bolts,	$D_1 = 1.6\ d$
Outside diameter of flange,	$D_2 = 2.2\ d$

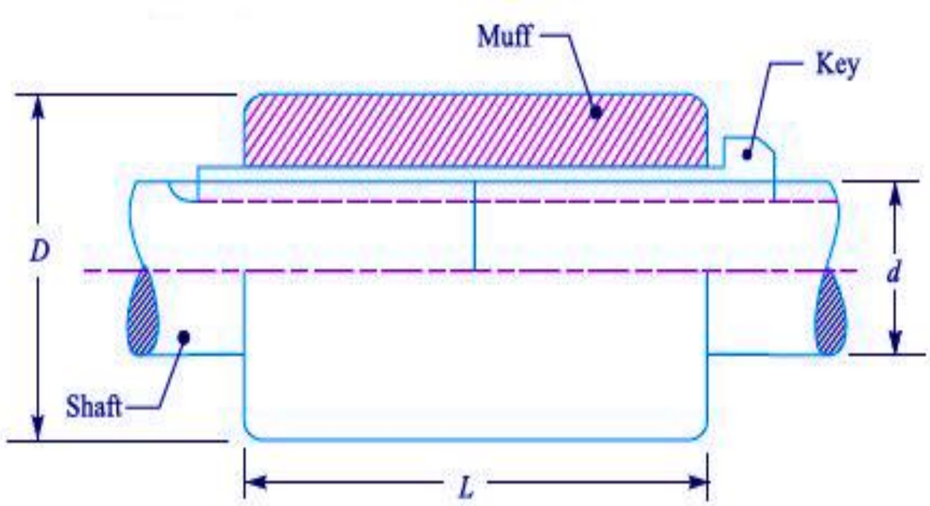
MARINE TYPE FLANGE COUPLIN

Protected type flange coupling. In a protected type flange coupling, as shown in Fig. the protruding bolts and nuts are protected by flanges on the two halves of the coupling, in order to avoid danger to the workman.

The thickness of the protective circumferential flange (t_p) is taken as 0.25 d. The other proportions of the coupling are same as for unprotected type flange coupling

Marine type flange coupling. In a marine type flange coupling, the flanges are forged integral with the shafts as shown in Fig. The flanges are held together by means of tapered headless bolts, numbering from four to twelve depending upon the diameter of shaft

Design of Muff Coupling



1. **Design for sleeve:-**

The sleeve is designed by considering it as a hollow shaft.

Let T = Torque to be transmitted by the coupling, and

τ_C = Permissible shear stress for the material of the sleeve which is cast iron. The safe value of shear stress for cast iron may be taken as 14 MPa.

We know that torque transmitted by a hollow section,

$$T = \frac{\pi}{16} * \tau_c \left(\frac{D^4 - d^4}{D}\right)$$

2. **Design for key**

The length of the coupling key is atleast equal to the length of the sleeve (*i.e.* 3.5 *d*). The coupling key is usually made into two parts so that the length of the key in each shaft

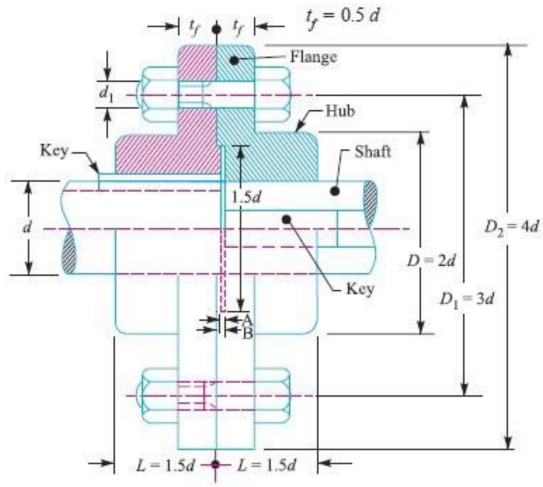
$$l = \frac{L}{2} = \frac{3.5d}{2}$$

The torque transmitted

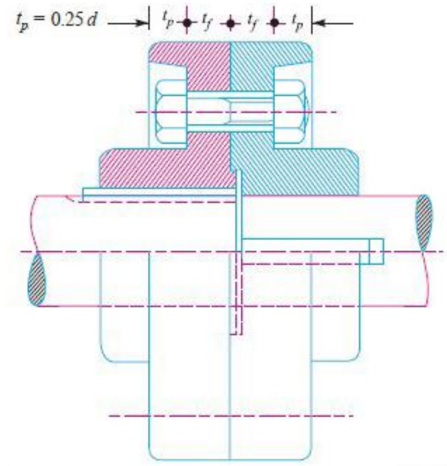
$= l * w * \tau * \frac{d}{2}$ (Considering shearing of the key)

$= l * w * \sigma_c * \frac{d}{2}$ (Considering crushing of the key)

Design of Flange Coupling:-



UNPROTECTED TYPE FLANGE COUPLING



PROTECTED TYPE FLANGE COUPLING

Consider a flange coupling as shown in Fig.

Let d = Diameter of shaft or inner diameter of hub,

D = Outer diameter of hub,

d_1 = Nominal or outside diameter of bolt,

D_1 = Diameter of bolt circle,

n = Number of bolts,

t_f = Thickness of flange,

τ_s, τ_b and τ_k = Allowable shear stress for shaft, bolt and key material respectively

τ_c = Allowable shear stress for the flange material *i.e.* cast iron,

σ_{cb}, and σ_{ck} = Allowable crushing stress for bolt and key material respectively.

The flange coupling is designed as discussed below

1. **Design for hub**

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft

$$T = \frac{\pi}{16} * \tau_c \left(\frac{D^4 - d^4}{D}\right)$$

2. Design for key

The key is designed with usual proportions and then checked for shearing and crushing stresses. The material of key is usually the same as that of shaft. The length of key is taken equal to the length of hub.

3. Design for flange

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the torque transmitted

$$T = \frac{\pi D^2}{2} * \tau_c * t_f$$

The thickness of flange is usually taken as half the diameter of shaft. Therefore from the above relation, the induced shearing stress in the flange may be checked

4. Design for bolts

The bolts are subjected to shear stress due to the torque transmitted. The number of bolts (n) depends upon the diameter of shaft and the pitch circle diameter of bolts (D_1) is taken as 3 d

Torque transmitted, $T = \frac{\pi}{4} * d_1^2 * \tau_b * n * \frac{D_1}{2}$

From this equation, the diameter of bolt (d_1) may be obtained. Now the diameter of bolt may be checked in crushing

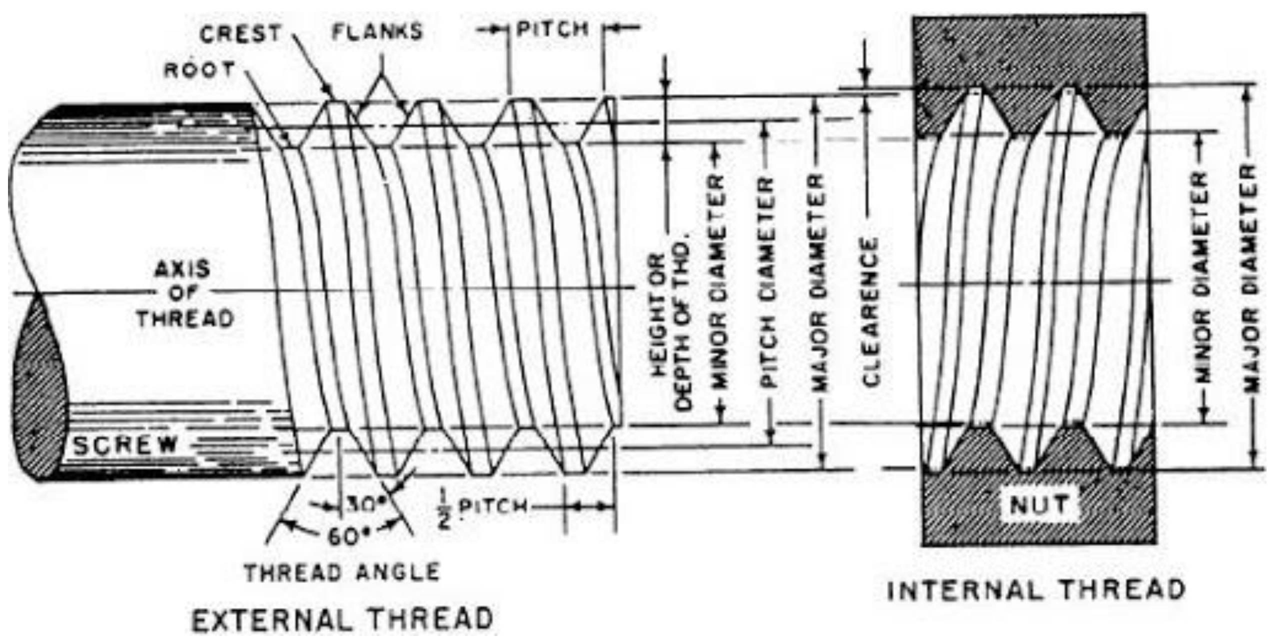
$$\text{Torque, } T = (n * d_1 * t_f * \sigma_{cb}) \frac{D_1}{2}$$

Screwed joints

A screw thread is formed by cutting a continuous helical groove on a cylindrical surface. A screw made by cutting a single helical groove on the cylinder is known as ***single threaded*** (or single-start) screw and if a second thread is cut in the space between the grooves of the first, a ***double threaded*** (or double-start) screw is formed. Similarly, triple and quadruple (*i.e.* multiple-start) threads may be formed. The helical grooves may be cut either ***right hand*** or ***left hand*.**

Important Terms Used in Screw Threads:-

The following terms used in screw threads, as shown in Fig. are important from the subject point of view



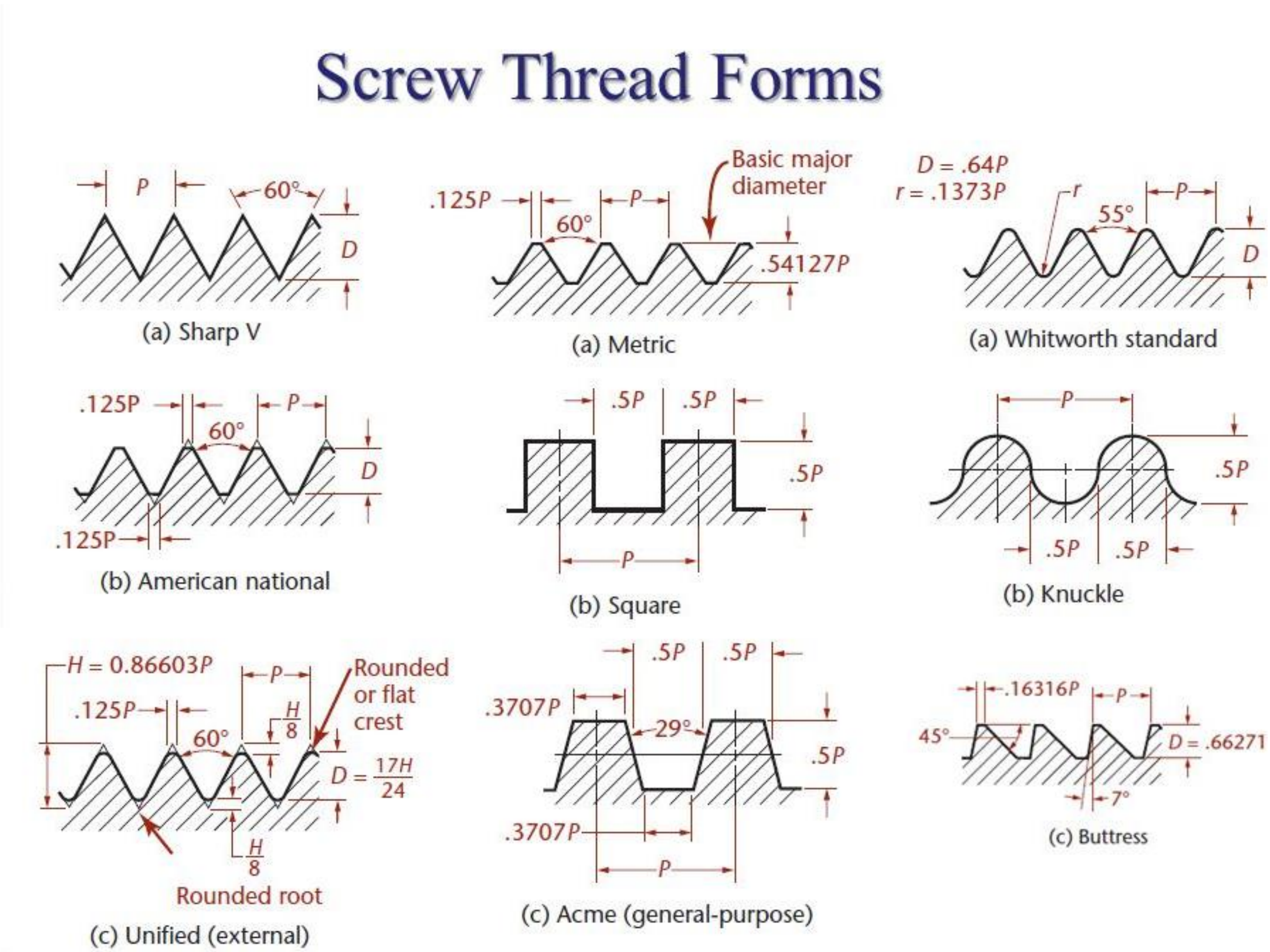
1. ***Major diameter*.** It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as ***outside*** or ***nominal diameter*.**
2. ***Minor diameter*.** It is the smallest diameter of an external or internal screw thread. It is also known as ***core*** or ***root diameter*.**
3. ***Pitch diameter*.** It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an ***effective diameter*.** In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.

4. ***Pitch*****.** It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane.

Pitch = 1/ (No. of threads per unit length of screw)

5. ***Lead*****.** It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.
6. ***Crest*****.** It is the top surface of the thread.
7. ***Root*****.** It is the bottom surface created by the two adjacent flanks of the thread.
8. ***Depth of thread*****.** It is the perpendicular distance between the crest and root.
9. ***Flank*****.** It is the surface joining the crest and root.
10. ***Angle of thread*****.** It is the angle included by the flanks of the thread.
11. ***Slope*****.** It is half the pitch of the thread

<u>Form of Screw threads</u>



Locking Devices

Ordinary thread fastenings, generally, remain tight under static loads, but many of these fastenings become loose under the action of variable loads or when machine is subjected to vibra- tions. The loosening of fastening is very dangerous and must be prevented. In order to prevent this, a large number of locking devices are available, some of which are discussed below:

***Jam nut or lock nut*.** A most common locking device is a jam, lock or check nut. It has about one-half to two-third thickness of the standard nut. The thin lock nut is first tightened down with ordinary force, and then the upper nut (*i.e*. thicker nut) is tightened down upon it, as shown in Fig. The upper nut is then held tightly while the lower one is slackened back against it.

***Castle nut*.** It consists of a hexagonal portion with a cylindrical upper part which is slotted in line with the centre of each face. The split pin passes through two slots in the nut and a hole in the bolt, so that a positive lock is obtained unless the pin shears. It is extensively used on jobs subjected to sudden shocks and considerable vibration such as in automobile industry.

***Sawn nut*.** It has a slot sawed about half way through. After the nut is screwed down, the small screw is tightened which produces more friction between the nut and the bolt. This prevents the loosening of nut.

Stresses in Screwed Fastening due to Static Loading:-

The following stresses in screwed fastening due to static loading are important from the subject point of view :

1. Internal stresses due to screwing up forces,
2. Stresses due to external forces, and
3. Stress due to combination of stresses at (1) and (2).

Initial Stresses due to Screwing up Forces

1. **Tensile stress due to stretching of bolt**

Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$P_i = 2840\, d$ N

where P_i = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm

2. **Torsional shear stress caused by the frictional resistance of the threads during its tightening.**

The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation

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

where τ = Torsional shear stress,

T = Torque applied, and

d_C = Minor or core diameter of the thread.

3. **Shear stress across the threads**

The average thread shearing stress for the screw (τ_S) is obtained by using the relation:

$$\tau_s = \frac{P}{\pi d_c * b * n}$$

4. **Compression or crushing stress on threads**

The compression or crushing stress between the threads (σ_c) may be obtained by using the relation

$$\sigma_c = \frac{P}{\pi[d^2 - d_c^2]n}$$

Stresses due to External Forces:-

The following stresses are induced in a bolt when it is subjected to an external load

1. **Tensile stress**

The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt

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

d_c= core or root diameter of the thread

σ_t= Permissible tensile stress for the bolt, stud or screw material

2. **Shear stress**

Sometimes, the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, then the shear stress is induced in the bolts. The shear stresses should be avoided as far as possible. It should be noted that when the bolts are subjected to direct shearing loads, they should be located in such a way that the shearing load comes upon the body (*i.e.* shank) of the bolt and not upon the threaded portion. In some cases, the bolts may be relieved of shear load by using shear pins. When a number of bolts are used to share the shearing load, the finished bolts should be fitted to the reamed holes.

Let d = Major diameter of the bolt, and

n = Number of bolts.

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

3. Combined tension and shear stress

When the bolt is subjected to both tension and shear loads, as in case of coupling bolts or bearing, then the diameter of the shank of the bolt is obtained from the shear load and that of threaded part from the tensile load. A diameter slightly larger than that required for either shear or tension may be assumed and stresses due to combined load should be checked for the following principal stresses.

Maximum principal shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2}$$

Maximum principal tensile stress

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