COURSE CONTENT

SUBECT: DESIGN OF MACHINE ELEMENT

UNIT-III

FACULTY: SWESHAREEFA MAHAKUL

ASST. PROF. ME DEPT

Introduction to Riveted Joints

A rivet is a short cylindrical bar with a head integral to it. The cylindrical portioon of the rivet is called **shank** or **body** and lo wer 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 us ed 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.

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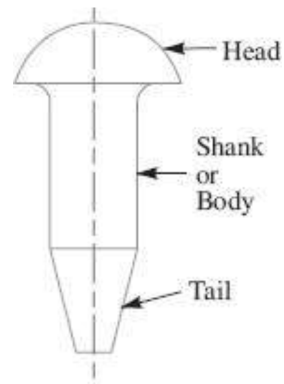


Fig. Rivet pa rts.

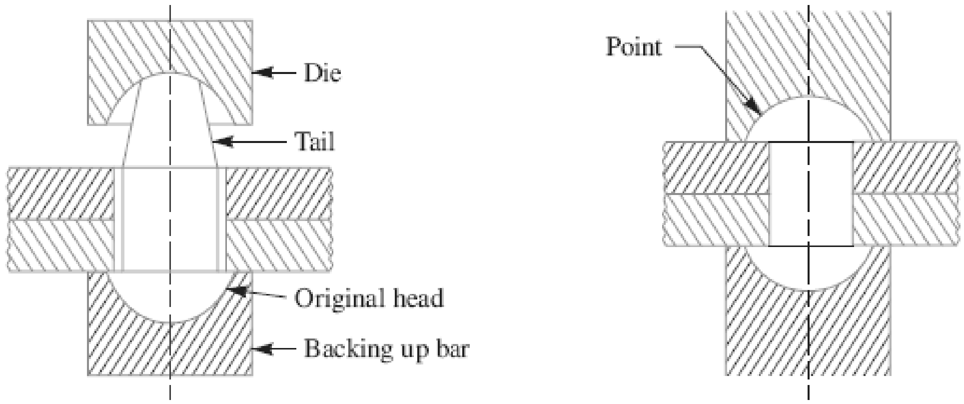
The **permanent fastenings** ar e those fastenings which cannot be disassem bled without destroying the connecting comp onents. The examples of permanent fastening s in order of strength are soldered, brazed, welded and riveted joints.

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

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 necessar y in order to contribute to strength and to pre ent 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. (a), the holes in the plates are punched and reamed or drilled. Punching is the cheapest method an d 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 nomin al diameter of the rivet.



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

Fig. Methods of riveting.

The plates are drilled tog ether and then separated to remove any burrs or chips so as to have a tight flush joint betwee n the plates. A cold rivet or a red hot rivet is in troduced into the plates and the **point** (i.e. se cond head) is then formed. When a cold rivet is used, the process is known as **cold riveti ng** 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 do ne 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 **p oint** as shown in Fig.(b). As the rivet cools, it tends to contract. The lateral contraction will be slight, but there will be a longitudinal ten sion introduced in the rivet which holds the plates firmly together.

In machine riveting, th e die is a part of the hammer which is ope rated 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 rivetin g process is used.

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

Types of Rivet Heads

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

1. Rivet heads for general purp oses (below 12 mm diameter) as shown in Fig.

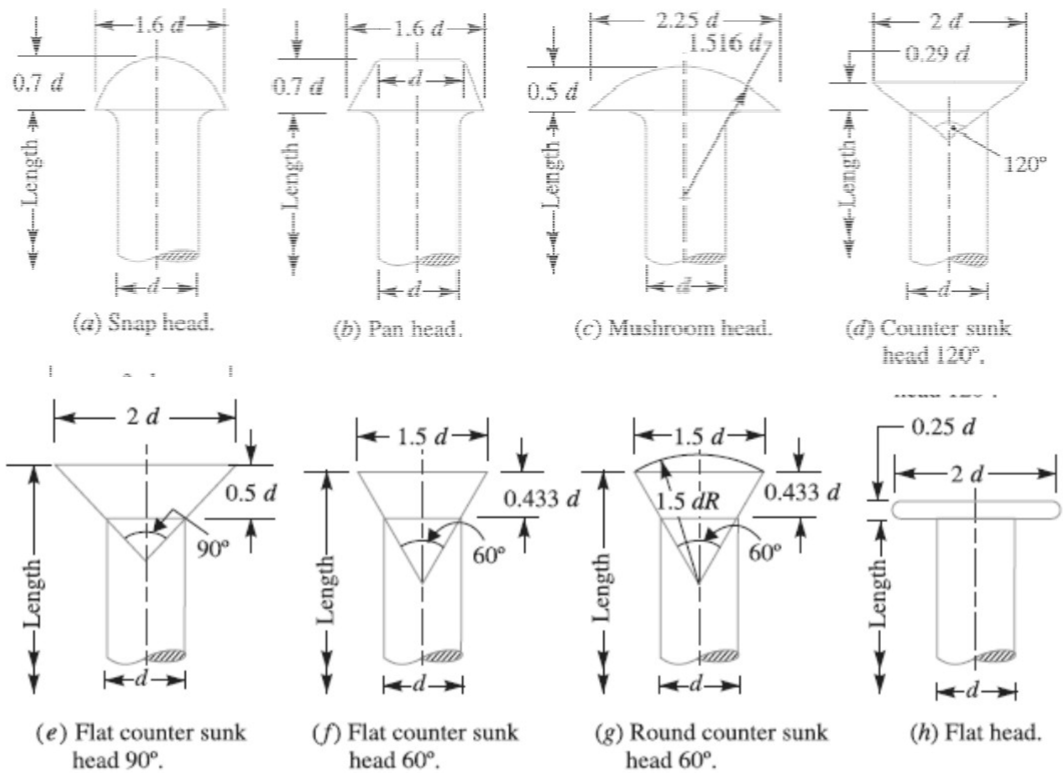
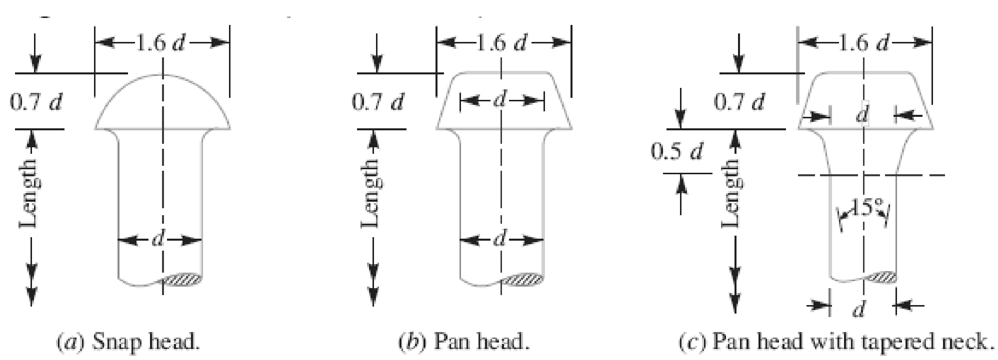
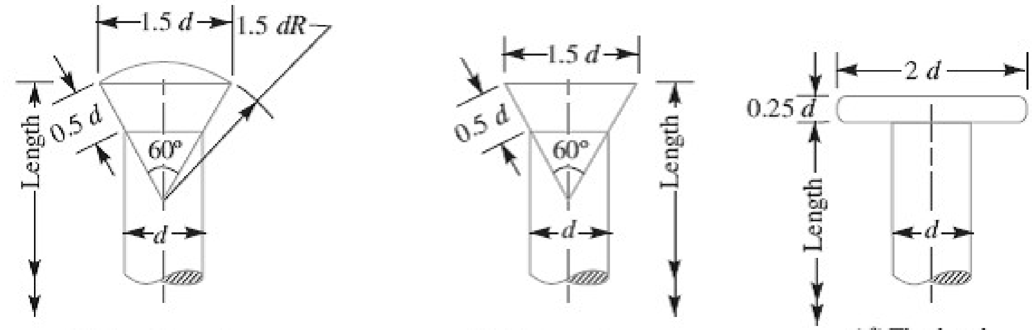


Fig. Rivet heads for general purposes (below 12 mm diameter).

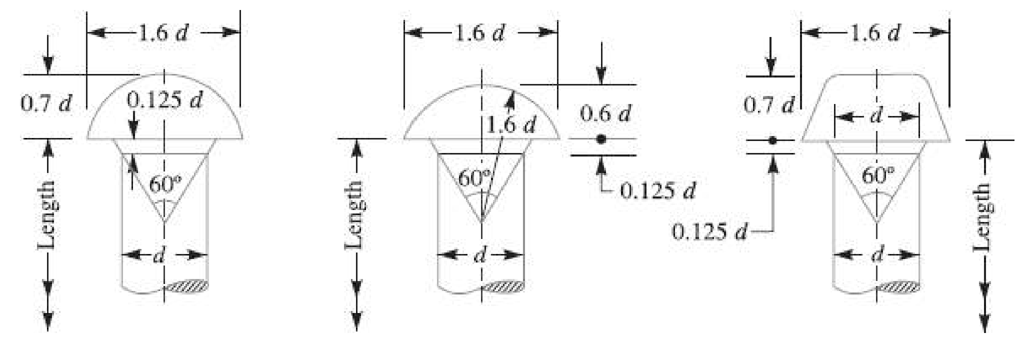
2. Rivet heads for general purp oses (From 12 mm to 48 mm diameter) as shown in Fig.



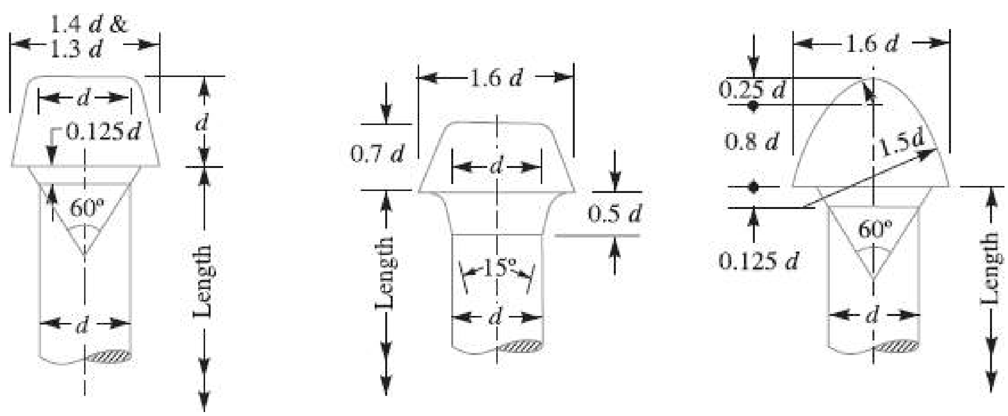


(d) Round counter sunk head 60°. (e) Flat counter sunk head 60°. (f) Flat head.

Fig. Rivet heads for g eneral 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.



(a) Snap head. (b) Ellipsoid head. (c) Pan head (Type I).



1.4 d for rivets under 24 mm. (e) Pan head with tapered neck. (f) Steeple h ead.

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.

1. Lap Joint

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

2. 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.

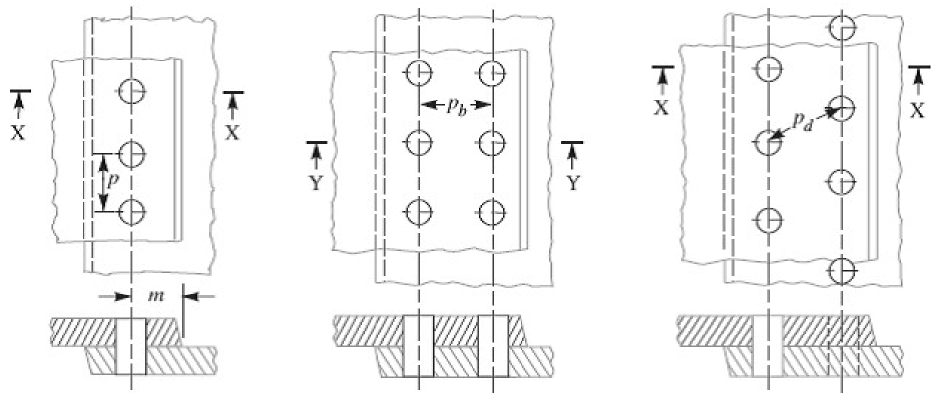
In a **single strap butt joint,** the edges of the main plates butt against each other and only one cover plate is placed on one side of the main plates and then riveted together. In a **double strap butt joint, t**he edges of the main plates butt against each other and two cover plates are placed on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

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, and

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



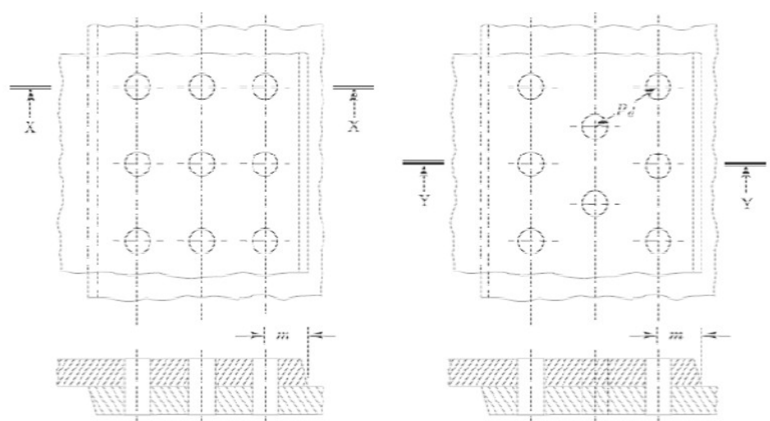
(a) Single riveted lap joint. (b) Double riveted lap joint (Chain riveting). (c) Double rivete d lap Joint (Zig-zag riveting).

Fig. Single and double riveted lap joints.

Similarly the joints may be **triple riveted** or **quadruple riveted.**

Notes: 1. when the rivets in the various rows are opposite to each other, as sho wn in Fig. (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. (c), then the joint is said to be z**ig-zag riveted.**

2. Since the plates overlap in lap joints, therefore the force P, P acting on the pl ates are not in the same straight line but they are at a distance equal to the thickness of the plate. These forces will form a couple which may bend the joint. Hence the lap joints may be used only where small loads are to be transmitted. On the other hand, the forces P, P in a butt joint act in the same straight line, therefore there will be no couple. Hence the butt jo ints are used where heavy loads are to be tran smitted.



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

Fig. 9.7. Triple riveted lap joint.

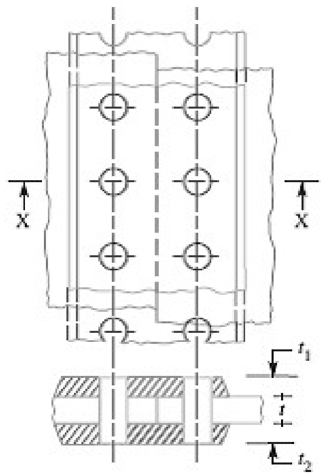
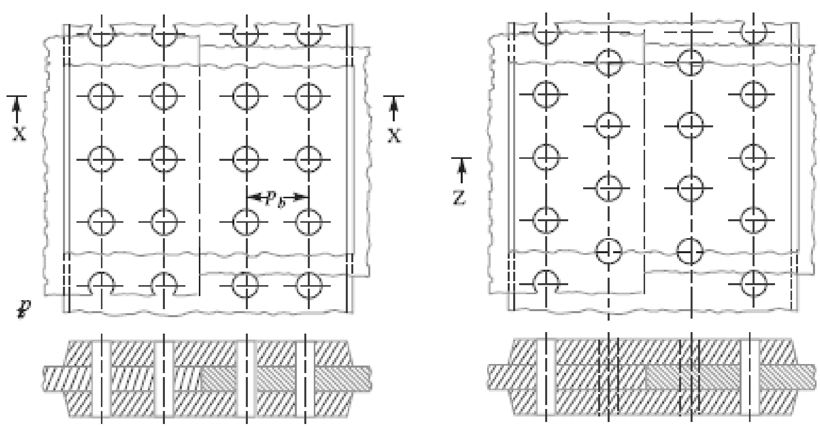


Fig. Single riveted double strap butt joint.



(a) Chain a ri veting. (b) Zig-zag riveting **Fig.** Double riveted double strap (equal) butt joints.

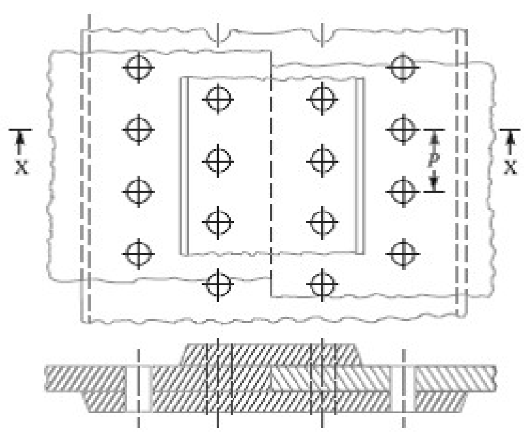


Fig. Double riveted double strap (unequal) butt joint with zig-zag riveting.

Important Terms Used in Riveted Joints

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

1. Pitch. It is the distance from the centre of one rivet to the centre of the next r ivet measured parallel to the seam as shown in Fig.1 It is usually denoted by p.

2. Back pitch. It is the perpendi cular distance between the centre lines of the successive rows as shown in Fig.1. It is usually denoted by p_b.

3. Diagonal pitch. It is the distance between the centers of the rivets in adj acent rows of zigzag riveted joint as shown in Fig. It is usually denoted by p_d.

4. Margin or marginal pitch. It is the distance between the centres of rive t hole to the nearest edge of the plate as shown in Fig. 9.6. It is usually denoted by m.

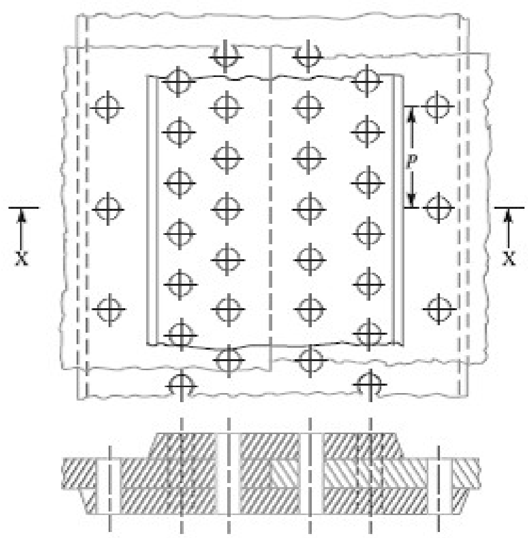
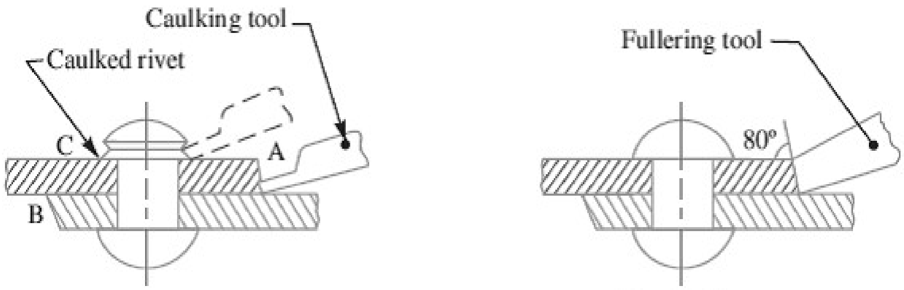


Fig.1. Triple riveted double strap (unequal) butt joint.

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 proces s 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 thhe edge of the plate, which is planed to a level 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.2 (a) forming a metal to metal j oint. In actual practice, both the edges at A and B are caulked. 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 tak en to prevent injury to the plate below the tool.



(*a*) Caulking. (*b*) Fullering.

Fig.2. Caulking and fullering.

A more satisfactory way of making the joints staunch is known as **fuller ing** which has largely superseded caulking. In this case, a fullering tool with a thickness at thee end equal to that of the plate is used in such a way that the greatest pressure due to the blo ws occur near the joint, giving a clean finish, with less risk of damaging the plate. A fullering process is shown in Fig. (b).

<u>Failures of a Riveted Joint</u>

A riveted joint may fail in the following ways:

1. Tearing of the plate at an ed ge. A joint may fail due to tearing of the plate at an edge as shown in Fig.3. 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 m ay tear off across a row of rivets as shown in Fig. In such cases, we consider only one pitch length of the plate, since every rivet is respo nsible for that much length of the plate only.

The resistance offered by the pla te against tearing is known as **tearing resistan ce** 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

$_t$ = Permissible tensile stress for the plate material.

We know that tearing area per pitch length,

$$A_t = (p\text{-}d)t$$

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

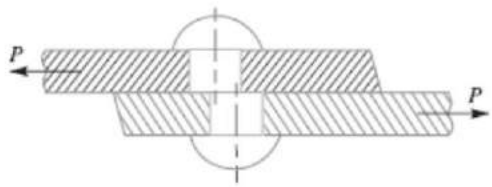
$$P_t = A_t \ . \ \sigma_t = (p - d)\ t \ . \ \sigma_t$$

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

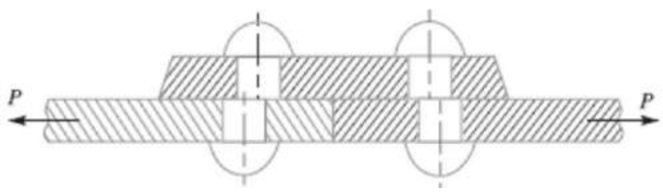
3. Shearing of the rivets. The plates which are connected by the rivets exert te nsile stress on the rivets, and if the rivets are un able to resist the stress, they are sheared off as shown in Fig.

5.

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 o ffered 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) Shearin g off a rivet in a single cover butt

joint. **Fig. 5.** Shearing of rivets.

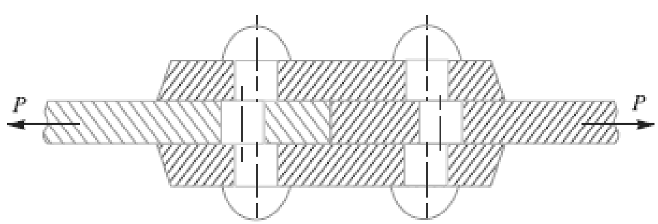


Fig.6. Shea ing off a rivet in double cover butt joint.

Let d = Diameter of the rivet hole,

=Safepermissibleshearstressforthe rivet material, and

n = Number of rivets per pitch length.

We know that shearing area,

$$A_s \quad \frac{\;}{4} d^2$$ ...(In single shear)

$$2 \quad \frac{\;}{4} d^2$$ ... (Theoretically, in double shear)

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

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

$$P_s \quad n \quad \frac{\;}{4} d^2 \quad \tau$$ ... (In single shear)

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

As we discussed earlier, when the shearing takes place at one cross-section of the rivet, then the rivets are said to be in **single shear.** Similarly, when the shearing takes place at two cross-sections of the rivet, then the rivets are said to be in **double shear.**

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

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

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. 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 diametric 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,

$_c$ = 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),

$$A_c = d.\ t$$

Total crushing area = $n\ .\ d\ .\ t$

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

$$P_c = n\ .\ d\ .\ t\ .\ _c$$

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

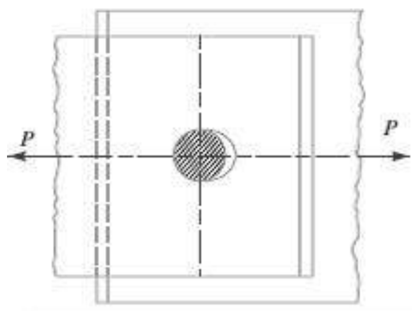


Fig. 7. Crushing of a rivet.

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 seen that 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,

$$P = p \cdot t \cdot {}_t$$

Efficiency of the riveted joint,

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

Where p = Pitch of the rivets,

t = Thickness of the plate, and

σ_t = Permissible tensile stress of the plate material.

Problem:

A double riveted lap joint is ma de between 15 mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushi ng, find the minimum force per pitch which w ill rupture the joint. If the above joint is subje cted to a load such that the factor of safety is 4 , find out the actual stresses developed in the p lates and the rivets.

Solution. Given : $t = 15$ mm ; $d = 25$ mm ; $p = 75$ mm ; $\sigma_{tu} = 400$ MPa = 400 N/mm^2 ; $\tau_u = 320$ MPa = 320 N/mm^2 ; $\sigma_{cu} = 640$ MPa = 640 N/mm^2

Minimum force per pitch which will rupture the joint

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

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

Ultimate shearing resistance of the rivets per pitch,

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

and ultimate crushing resistance of the rivets per pitch,

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

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

Actual stresses produced in the plates and rivets

Since the factor of safety is 4, therefore safe load per pitch length of the joint

$$= 300\,000/4 = 75\,000 \text{ N}$$

Let σ_{ta}, τ_a and σ_{ca} be the actual tearing, shearing and crushing stresses produced with a safe load of 75 000 N in tearing, shearing and crushing.

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

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

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

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

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

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

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

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

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

Problem

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 she aring stress in rivets = 90 MPa Permissible cru shing stress in rivets = 180 MPa.

Solution. Given : $t = 6$ mm ; $d = 20$ mm ; $\sigma_t = 120$ MPa = 120 N/mm^2 ; $\tau = 90$ MPa = 90 N/mm^2 ; $\sigma_c = 180$ MPa = 180 N/mm^2

1. ***Efficiency of the first joint***

Pitch, $p = 50$ mm ...(Given)

First of all, let us find the tearing resistance of the plate, shearing and crushing resistances of the rivets.

(*i*) ***Tearing resistance of the plate***

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

$$P_t = (p - d)\, t \times \sigma_t = (50 - 20)\, 6 \times 120 = 21\ 600 \text{ N}$$

(*ii*) ***Shearing resistance of the rivet***

Since the joint is a single riveted lap joint, therefore the strength of one rivet in single shear is taken. We know that shearing resistance of one rivet,

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

(*iii*) ***Crushing resistance of the rivet***

Since the joint is a single riveted, therefore strength of one rivet is taken. We know that crushing resistance of one rivet,

$$P_c = d \times t \times \sigma_c = 20 \times 6 \times 180 = 21\ 600 \text{ N}$$

$\therefore$ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 21\ 600 \text{ N}$$

We know that strength of the unriveted or solid plate,

$$P = p \times t \times \sigma_t = 50 \times 6 \times 120 = 36\ 000 \text{ N}$$

$\therefore$ Efficiency of the joint,

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

2. ***Efficiency of the second joint***

Pitch, $p = 65$ mm ...(Given)

(*i*) ***Tearing resistance of the plate,***

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

$$P_t = (p - d)\ t \times \sigma_t = (65 - 20)\ 6 \times 120 = 32\ 400 \text{ N}$$

(*ii*) ***Shearing resistance of the rivets***

Since the joint is double riveted lap joint, therefore strength of two rivets in single shear is taken. We know that shearing resistance of the rivets,

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

(*iii*) ***Crushing resistance of the rivet***

Since the joint is double riveted, therefore strength of two rivets is taken. We know that crushing resistance of rivets,

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

$\therefore$ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 32\,400 \text{ N}$$

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

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

$\therefore$ Efficiency of the joint,

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

Design of boiler joints according to IBR

Design of Boiler Joints

The boiler has a longitudinal joint as well as circumferential joint. The ***longitudinal joint*** is used to join the ends of the plate to get the required diameter of a boiler. For this purpose, a butt joint with two cover plates is used. The ***circumferential joint*** is used to get the required length of the boiler. For this purpose, a lap joint with one ring overlapping the other alternately is used.

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.

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.*** First of all, the thickness of the boiler shell is determined by using the thin cylindrical formula, *i.e*.

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

Where 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.

Indian Boiler Regulations (I.B.R.) allows 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.

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

i.e. $d = 6\,t$ (when t is greater than 8 mm)

But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the rivets to crushing resistance. In no case,

the diameter of rivet hole should not be less than the thickness of the plate, because there will be danger of punch crushing.

3. ***Pitch of rivets.*** The pitch of the rivets is obtained by equating the tearing resistance of the plate to the shearing resistance of the rivets. It may noted that **(*a*)** The pitch of the rivets should not be less than 2*d*, which is necessary for the formation of head.

(*b*) The maximum value of the pitch of rivets for a longitudinal joint of a boiler as per I.B.R. is $p_{max} = C \times t + 41.28$ mm where t = Thickness of the shell plate in mm, and C = Constant. The value of the constant C may be taken from DDB. If the pitch of rivets as obtained by equating the tearing resistance to the shearing resistance is more than p_{max}, then the value of p_{max} is taken.

4. ***Distance between the rows of rivets.*** The distance between the rows of rivets as specified by Indian Boiler Regulations is as follows:

(a) For equal number of rivets in more than one row for lap joint or butt joint, the distance between the rows of rivets (p_b) should not be less than 0.33 *p* + 0.67 *d*, for zig-zig riveting, and 2 *d*, for chain riveting.

(b) For joints in which the number of rivets in outer rows is ***half*** the number of rivets in inner rows and if the inner rows are chain riveted, the distance between the outer rows and the next rows should not be less than 0.33 *p* + 0.67 or 2 *d*, whichever is greater. The distance between the rows in which there are full number of rivets shall not be less than 2*d*.

(c) For joints in which the number of rivets in outer rows is ***half*** the number of rivets in inner rows and if the inner rows are zig-zig riveted, the distance between the outer rows and the next rows shall not be less than 0.2 *p* + 1.15 *d*. The distance between the rows in which there are full number of rivets (zig-zag) shall not be less than 0.165 *p* + 0.67 *d*.

Note : In the above discussion, *p* is the pitch of the rivets in the outer rows.

5. ***Thickness of butt strap.*** According to I.B.R., the thicknesses for butt strap (t_1) are as given below:

(a) The thickness of butt strap, in no case, shall be less than 10 mm.

(b) $t_t = 1.125\, t$, for ordinary (chain riveting) single butt strap.

$$t_1 = 1.125\, t \left(\frac{p - d}{p - 2d} \right)$$

For single butt straps, every alternate rivet in outer rows being omitted.

$t_1 = 0.625\, t$, for double butt-straps of equal width having ordinary riveting (chain riveting).

$$t_1 = 0.625\, t \left(\frac{p-d}{p-2d} \right)$$

For double butt straps of equal width having every alternate rivet in the outer rows being omitted.

(c) For unequal width of butt straps, the thicknesses of butt strap are $t_1 = 0.75\, t$, for wide strap on the inside, and
$t_1 = 0.625\, t$, for narrow strap on the outside.

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

Note: The above procedure may also be applied to ordinary riveted joints.

Design of eccentric loaded riveted joints and Proble**Eccentric Loaded Riveted Joint**

When the line of action of the load does not pass through the centroid of the rivet system and thus all rivets are not equally loaded, then the joint is said to be an ***eccentric loaded riveted joint***, as shown in Fig. 1(a). The eccentric loading results in secondary shear caused by the tendency of force to twist the joint about the centre of gravity in addition to direct shear or primary shear.

Let P = Eccentric load on the joint, and

e = Eccentricity of the load *i.e.* the distance between the line of action of the load and the centroid of the rivet system *i.e.* G.

The following procedure is adopted for the design of an eccentrically loaded riveted joint.

Note: This picture is given as additional information and is not a direct example of the current chapter.

1. First of all, find the centre of gravity G of the rivet system. Let A = Cross-sectional area of each rivet,

x_1, x_2, x_3 etc. = Distances of rivets from OY,

and y=, y_2, y_3 etc. = Distances of rivets from OX.

We know that $\bar{x} = \frac{A_1 x_1 + A_2 x_2 + A_3 x_3 + \ldots}{A_1 + A_2 + A_3 + \ldots} = \frac{A x_1 + A x_2 + A x_3 + \ldots}{n.A}$

$= \frac{x_1 + x_2 + x_3 \ldots}{n}$ ....(where n = Number of rivets)

Similarly, $\bar{y} = \frac{y1 + y2 + y3 + \ldots}{n}$

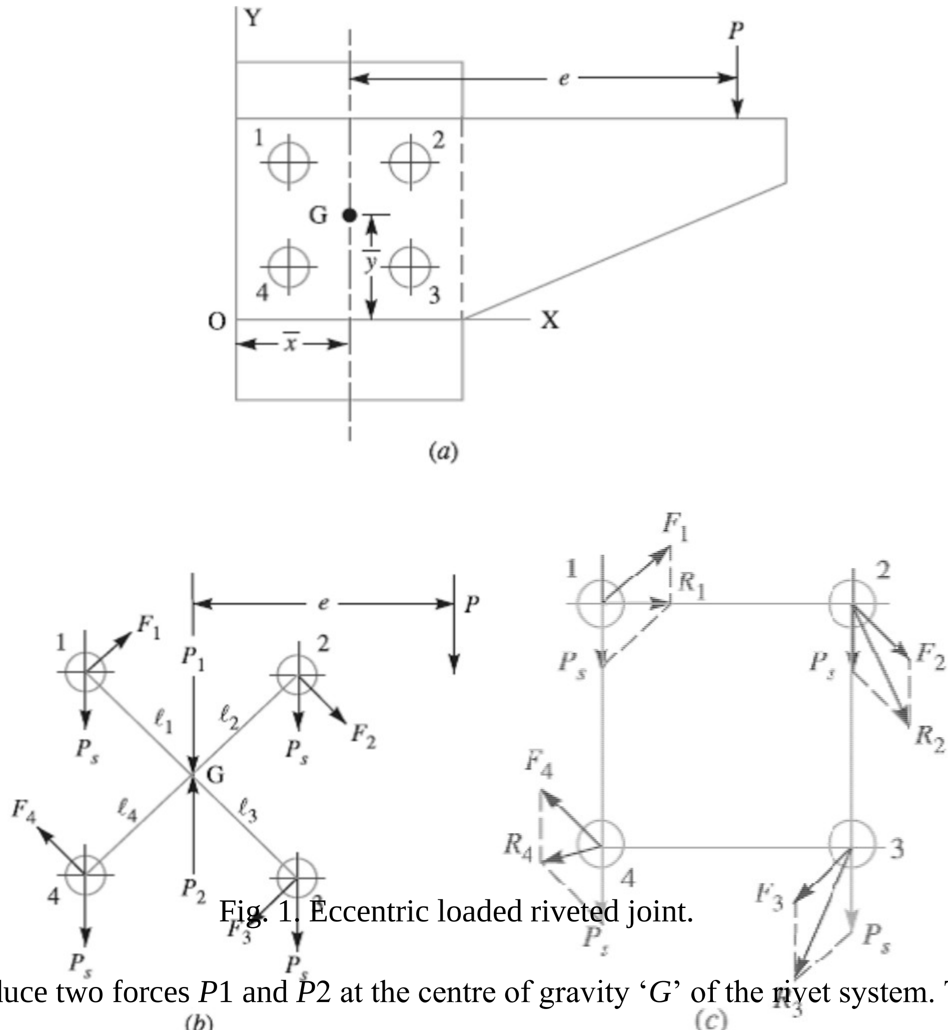


Fig. 1. Eccentric loaded riveted joint.

2. Introduce two forces *P*1 and *P*2 at the centre of gravity '*G*' of the rivet system. These forces are equal and opposite to *P* as shown in Fig.(*b*).

3. Assuming that all the rivets are of the same size, the effect of $P_1 = P$ is to p roduce direct shear load on each rivet of equal magnitude. Therefore, direct shear load on each rivet,

$$P_s = \frac{P}{n}, \text{ acting parallel to the load P,}$$

4. The effect of $P_2 = P$ is to pr oduce a turning moment of magnitude $P \times e$ w hich tends to rotate the joint about the centre of gravity '*G*' of the rivet system in a clockw ise direction. Due to the turning moment, sec ondary shear load on each rivet is produced. In order to find the secondary shear load, the following two assumptions are made:

(a) The secondary shear load is proportional to the radial distance of the rivet under consideration from the centre of gravity of the rivet system.

(b) The direction of secondary shear load is perpendicular to the line joining the centre of the rivet to the centre of gravity of the rivet system..

Let F_1, F=, F_3 ... = Secondary shear loads on the rivets 1, 2, 3...etc.

F_1, F_2, F_3 ... = Radial distance of the rivets 1, 2, 3 ...etc. from the centre of gravity '*G*' of the rivet system.

From assumption *a*(), $F_1 \quad l_1$; $F_2 \quad l_2$ and so on

or $$\frac{F_1}{l_1} \quad \frac{F_2}{l_2} \quad \frac{F_3}{l_3} ...$$

$$F_2 \quad F_1 \frac{l_2}{l_1} \text{, and } F_3 \quad F_1 \frac{l_3}{l_1}$$

We know that the sum of the external turning moment due to the eccentric load and of internal resisting moment of the rivets must be equal to zero.

$$P.e = F_1.l_1 + F_2.l_2 + F_3.l_3 + \dots$$

$$F_1 l_1 \quad F_1 \frac{l_2}{l_1} l_2 \quad F_1 \frac{l_3}{l_1} l_3 \quad ...$$

$$\frac{F_1}{l_1} \quad l_1^2 \quad l_2^2 \quad l_3^2 ...$$

From the above expression, the value of F_1 may be calculated and hence F_2 and F_3 etc. are known. The direction of these forces are at right angles to the lines joining the centre of rivet to the centre of gravity of the rivet system, as shown in Fig. 1(*b*), and should produce the moment in the same direction (*i.e.* clockwise or anticlockwise) about the centre of gravity, as the turning moment ($P \times e$).

5. The primary (or direct) and secondary shear load may be added vectorially to determine the resultant shear load (*R*) on each rivet as shown in Fig.1 (*c*). It may also be obtained by using the relation

$$R = \sqrt{P_s^{\,2} + F^2 + 2P_s F \cos\theta}$$

Where = Angle between the p rimary or direct shear load (P_s)

And secondary shear load (F).

When the secondary shear load on each rivet is equal, then the heavil y loaded rivet will be one in which the included angle between the direct shear load and se condary shear load is minimum. The maximu m loaded rivet becomes the critical one for determining the strength of the riveted joint. Knowing the permissible shear stress ( ), the diameter of the rivet hole may be obtained by using the relation,

Maximum resultant shea load (R) = $\frac{}{4}$ d^2

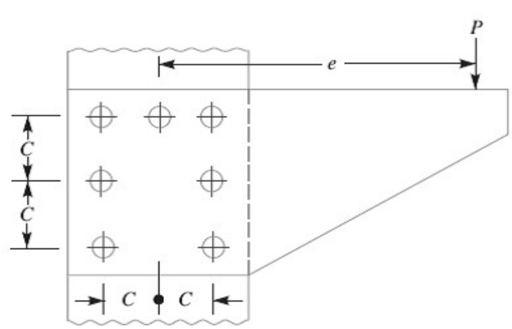
From DDB, the standard diam eter of the rivet hole (d) and the rivet dia meter may be specified

Notes : 1. In the solution of a problem, the primary and shear loads ma y be laid off approximately to scale and generally the rivet having the maximum resultant s hear load will be apparent by inspection. The values of the load for that rivet may then be calc ulated.

2. When the thickness of the plate is given, then the diameter of the rivet hole may be checked against crushing.

3. When the eccentric load P is inclined at some angle, then the same procedur e as discussed above may be followed to find th e size of rivet.

Problem: An eccentrically load ed lap riveted joint is to be designed for a ste el bracket as shown in Fig. 2. The bracket plate is 25 mm thick. All rivets are to be of the same size. Load on the bracket, P = 50 kN ; rivet spacing, C = 100 mm; load arm, e = 400 m m. Permissible shear stress is 65 MPa and cr ushing stress is 120 MPa. Determine the size of the rivets to be used for the joint.



Solution. Given: t = 25 mm ; P = 50 kN = 50 × 103 N ; e = 400 mm ; n = 7 ; = 65 MPa = 65 N/mm$_2$; c = 120 MPa = 120 N/mm^2.

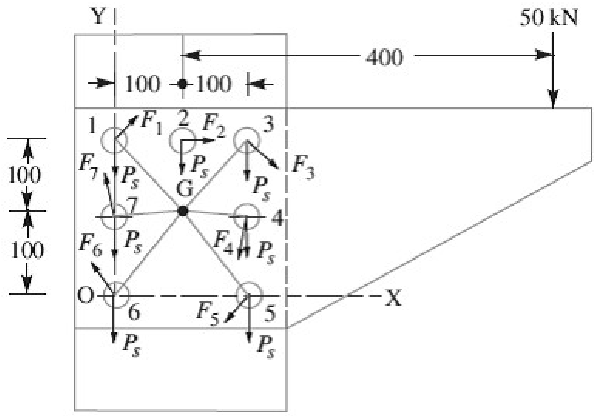


Fig.2

First of all, let us find the centre of gravity (G) of the rivet system.

Let x = Distance of centre of gravity from OY,

y = Distance of centre of gravity from OX,

x_1, x_2, x_3... = Distances of centre of gravity of each rivet from OY,
and y_1, y_2, y_3... = Distances of centre of gravity of each rivet from OX.

We know that

$$\bar{x} = \frac{x_1 + x_2 + x_3 + x_4 + x_5 + x_6 + x_7}{n}$$

$$= \frac{100 + 200 + 200 + 200}{7} = 100 \text{ mm} \qquad ...(\because x_1 = x_6 = x_7 = 0)$$

$$\bar{y} = \frac{y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7}{n}$$

$$= \frac{200 + 200 + 200 + 100 + 100}{7} = 114.3 \text{ mm} \quad ...(\because y_5 = y_6 = 0)$$

The centre of gravity (G) of the rivet system lies at a distance of 100 mm from OY and 114.3 mm from OX, as shown in Fig. 2.

We know that direct shear load o n each rivet,

$$P_s = \frac{P}{n} = \frac{50 \times 10^3}{7} = 7143 \text{ N}$$

The direct shear load acts paralle l to the direction of load *P i.e.* vertically downw ard as shown in Fig. 2. Turning moment produ ced by the load *P* due to eccentricity (*e*)

$$= P \times e = 50 \times 103 \times 400 = 20 \times 10^6 \text{ N-mm}$$

This turning moment is resisted by seven rivets as shown in Fig.2.

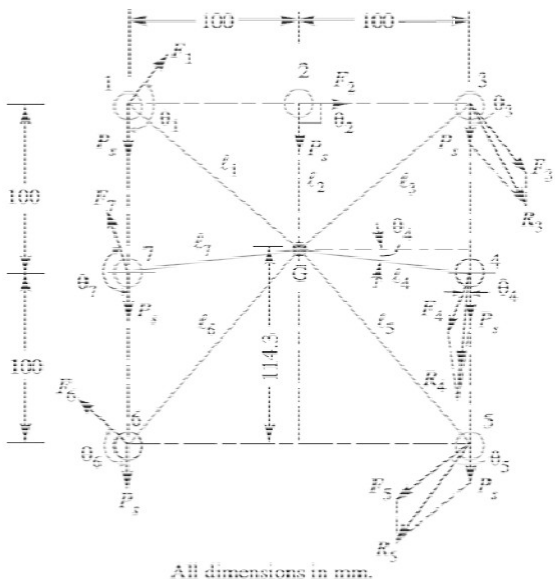


Fig. 3

Let F_1, F_2, F_3, F_4, F_5, F_6 and F_7 be the secondary shear load on the rivets 1, 2, 3, 4, 5, 6 and 7 placed at distances l_1, l_2, l_3, l_4, l_5 , l_6 and l_7 respectively from the centre of gravity of the rivet system as shown in Fig. 3.

From the geometry of the figure, we find that

$$l_1 = l_3 = \sqrt{(100)^2 + (200 - 114.3)^2} = 131.7 \text{ mm}$$

$$l_2 = 200 - 114.3 = 85.7 \text{ mm}$$

$$l_4 = l_7 = \sqrt{(100)^2 + (114.3 - 100)^2} = 101 \text{ mm}$$

$$l_5 = l_6 = \sqrt{(100)^2 + (114.3)^2} = 152 \text{ mm}$$

Now equating the turning mome nt due to eccentricity of the load to the resistin g moment of the rivets, we have

$$P \times e = \frac{F_1}{l_1}\left[(l_1)^2 + (l_2)^2 + (l_3)^2 + (l_4)^2 + (l_5)^2 + (l_6)^2 + (l_7)^2\right]$$

$$= \frac{F_1}{l_1}\left[2(l_1)^2 + (l_2)^2 + 2(l_4)^2 + 2(l_5)^2\right]$$

$$\text{....}(\because l_1 = l_3; l_4 = l_7 \text{ and } l_5 = l_6)$$

$$50 \times 10^3 \times 400 = \frac{F_1}{131.7}\left[2(131.7)^2 + (85.7)^2 + 2(101)^2 + 2(152)^2\right]$$

$$20 \times 10^6 \times 131.7 = F_1(34\ 690 + 7345 + 20\ 402 + 46\ 208) = 108\ 645\ F_1$$

$$F_1 = 20 \times 10^6 \times 131.7 / 108\ 645 = 24\ 244 \text{ N}$$

Since the secondary shear loads are proportional to their radial distances from the centre of gravity, therefore

$$F_2 = F_1 \times \frac{l_2}{l_1} = 24\ 244 \times \frac{85.7}{131.7} = 15\ 776 \text{ N}$$

$$F_3 = F_1 \times \frac{l_3}{l_1} = F_1 = 24\ 244 \text{ N} \qquad \text{...}(\because l_1 = l_3)$$

$$F_4 = F_1 \times \frac{l_4}{l_1} = 24\ 244 \times \frac{101}{131.7} = 18\ 593 \text{ N}$$

By drawing the direct and secondary shear loads on each rivet, we see that the rivets 3, 4 and 5 are heavily loaded. Let us now find the angles between the direct and secondary shear load for these three rivets. From the geometry of Fig.3, we find that

$$\cos\theta_3 = \frac{100}{l_3} = \frac{100}{131.7} = 0.76$$

$$\cos\theta_4 = \frac{100}{l_4} = \frac{100}{101} = 0.99$$

$$\cos\theta_5 = \frac{100}{l_5} = \frac{100}{152} = 0.658$$

Now resultant shear load on rivet 3,

$$R_3 = \sqrt{(P_s)^2 + (F_3)^2 + 2P_s \times F_3 \times \cos\theta_3}$$

$$= \sqrt{(7143)^2 + (24\ 244)^2 + 2 \times 7143 \times 24\ 244 \times 0.76} = 30\ 033 \text{ N}$$

Resultant shear load on rivet 4,

$$= \sqrt{(7143)^2 + (18\ 593)^2 + 2 \times 7143 \times 18\ 593 \times 0.99} = 25\ 684 \text{ N}$$

And resultant shear load on rivet 5,

$$R_5 = \sqrt{(P_s)^2 + (F_5)^2 + 2\ P_s \times F_5 \times \cos\theta_5}$$

$$= \sqrt{(7143)^2 + (27\ 981)^2 + 2 \times 7143 \times 27\ 981 \times 0.658} = 33\ 121\ \text{N}$$

The resultant shear load may be determined graphically, as shown in Fig.3.

From above we see that the maximum resultant shear load is on rivet 5. If *d* is the diameter of rivet hole, then maximum resultant shear load (R_5),

$$33\ 121 = \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4} \times d^2 \times 65 = 51\ d^2$$

$$d^2 = 33\ 121 / 51 = 649.4 \quad \text{or} \quad d = 25.5\ \text{mm}$$

From DDB, we see that according the standard diameter of the rivet hole (*d)* is 25.5 mm and the corresponding diameter of rivet is 24 mm.

Let us now check the joint for crushing stress. We know that

$$\text{Crushing stress} = \frac{\text{Max. load}}{\text{Crushing area}} = \frac{R_5}{d \times t} = \frac{33\ 121}{25.5 \times 25}$$

$$= 51.95\ \text{N/mm}^2 = 51.95\ \text{MPa}$$

Since this stress is well below the given crushing stress of 120 MPa, therefore the design is satisfactory.

Introduction to Welded Joints

Introduction

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. 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 hea ting and cooling during fabrication, therefore the members may get distorted or additional st resses may develop.

2. It requires a highly skilled labour and supervision.

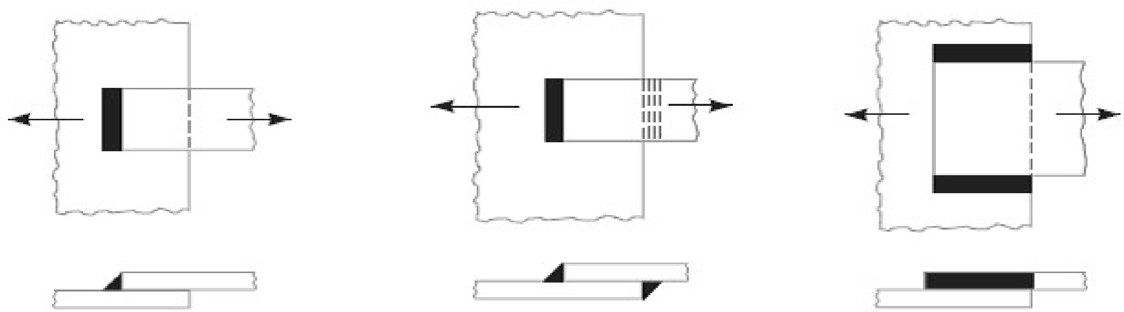
3. Since no provision is kept for expansion and contraction in the frame, thereffore there is a possibility of cracks developing in it.

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

Types of Welded Joints

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.1. Types of Lab and Butt Joints

Lap Joint

The lap joint or the fillet joint is obtained by overlapping the plates and the n 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.** Dou ble transverse fillet and **3.** Parallel fillet joints.

The fillet joints are shown in Fig.1. 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 pla cing the plates edge to edge as shown in Fig.2. In butt welds, the plate edges do not require b eveling 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 bevel ed to V or U-groove on both sides.

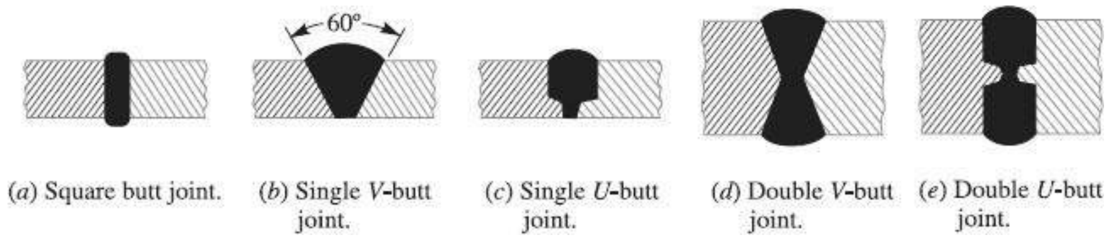


Fig. 2. Types of Butt joints

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.** D uble U-butt

joint. These joints are shown in Fig. 2.

The other type of welded joints a re corner joint, edge joint and T-joint as shown in Fig. 3.

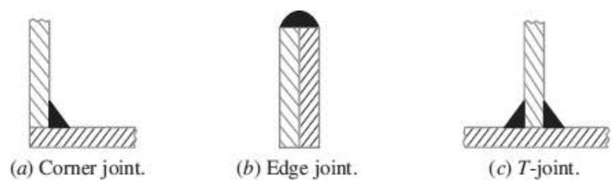
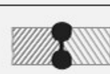
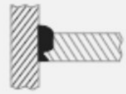


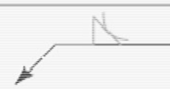
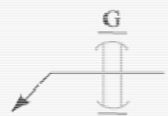
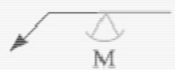
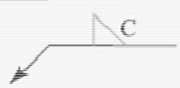
Fig. 3. Other types of Joints

Basic Weld Symbols

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

S. No.	*Form of weld*	*Sectional representation*	*Symbol*
9.	Single-*J* butt		
10.	Double-*J* butt		
11.	Bead (edge or seal)		
12.	Stud		
13.	Sealing run		
14.	Spot		
15.	Seam		
16.	Mashed seam	 Before After	
17.	Plug		
18.	Backing strip		
19.	Stitch		
20.	Projection	 Before After	
21.	Flash	 Rod or bar Tube	
22.	Butt resistance or pressure (upset)	 Rod or bar Tube	

Supplementary Weld Symbols

S. No.	*Particulars*	*Drawing representation*	*Symbol*
1.	Weld all round		
2.	Field weld		
3.	Flush contour		
4.	Convex contour		
5.	Concave contour		
6.	Grinding finish		G
7.	Machining finish		M
8.	Chipping finish		C

Elements of a welding symbol

Elements of a Welding Symbol

A welding symbol consists of th e following eight elements:

1. Reference line, **2.** Arrow,

3. Basic weld symbols, **4.** Dimen sions and other data,

5. Supplementary symbols, **6.** Finish symbols,

7. Tail, and **8.** Specification, pro cess or other references.

Standard Location of Element s of a Welding Symbol

The arrow points to the location of weld, the basic symbols with dimensions a re located on one or both sides of reference li ne. The specification if any is placed in the tail of arrow. Fig. 1. shows the standard locations o f welding symbols represented on drawing.

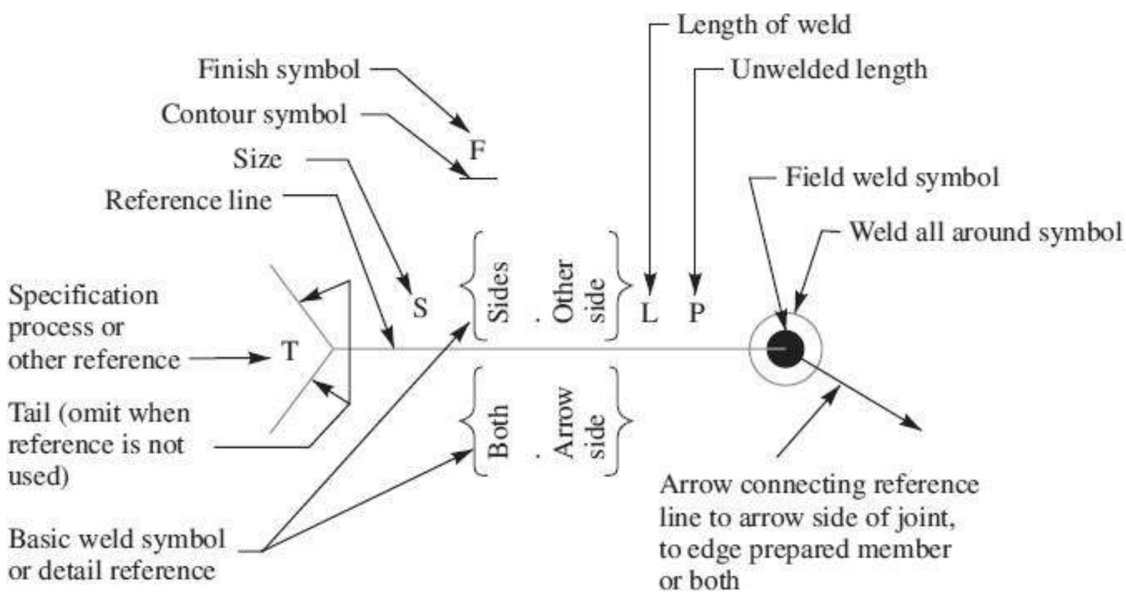
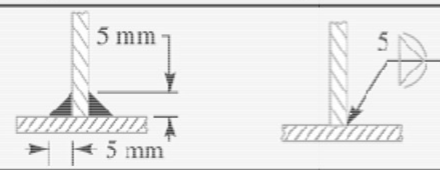
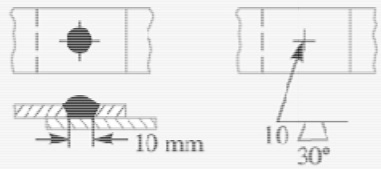
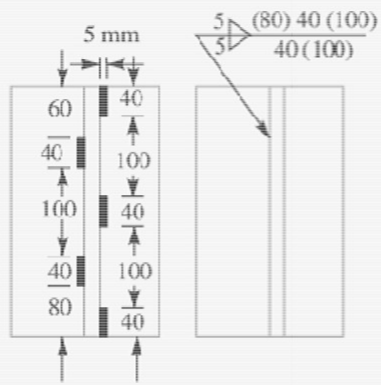


Fig.1 Standard location of weld symbols.

Some of the examples of weldin g symbols represented on drawing are shown in the following table.

Representation of welding symbols.

S. No.	Desired weld	Represeniation on drawing
1.	Fillet-weld each side of Tee- convex contour	
2.	Single V-butt weld -machining finish	
3.	Double V- butt weld	
4.	Plug weld - 30° Groove-angle-flush contour	
5.	Staggered intermittent fillet welds	

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Contents: Design of Welded Join ts

Strength of Transverse Fillet W elded 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 design ed for tensile strength. Let us consider a single and double transverse fillet welds as shown in Fig. 1(*a*) and (*b*) respectively.

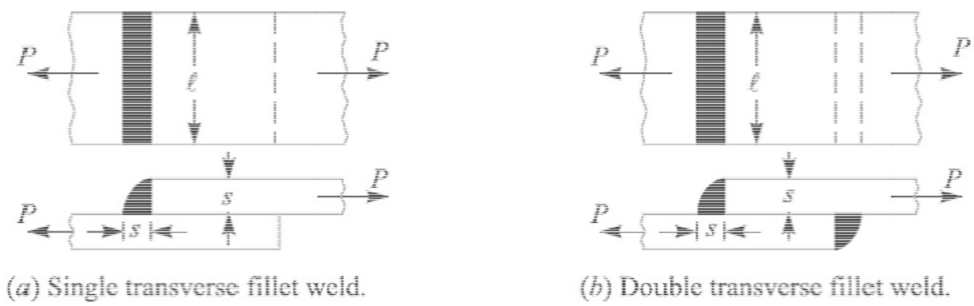


Fig.1 Transverse fillet welds.

The length of each side is known as ***leg*** or ***size of the weld*** and the perpendicular distance of the hypotenuse from the inters ection of legs (*i.e. BD*) is known as ***throat th ickness.*** 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 we ld.

Let t = Throat thickness (*BD*),

s = Leg or size of weld,

= Thickness of plate,

and l = Length of weld,

From Fig.2, we find that the throat thickness,

$$t = s \times \sin 45° = 0.707\, s$$

Therefore, Minimum area of the weld or throat area,

$$A = \text{Throat thickness} \times \text{Length of weld}$$

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

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

P = Throat area × Allowa ble tensile stress = $0.707\, s \times l \times \sigma_t$ And tensile strength of the joint for double fillet weld,

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

Note: Since the weld is weaker than the plate due to slag and blow holes, there fore the weld is given a reinforcement which m ay be taken as 10% of the plate thickness.

Strength of Parallel Fillet Wel ded Joints

The parallel fillet welded joints are designed for shear strength. Consider a double parallel fillet welded joint as shown in Fig.3 (*a*). We have already discussed in the pr evious article, that the minimum area of weld or the throat area,

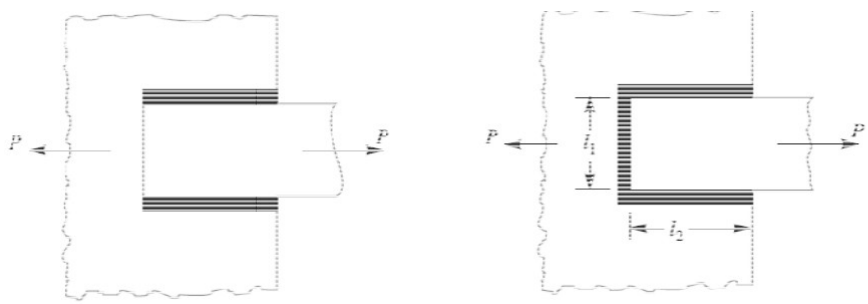
$$A = 0.707\ s \times l$$

If τ is the allowable shear stres s for the weld metal, then the shear strength o f the joint for single parallel fillet weld,

$$P = \text{Throat area} \times \text{Allowable shear stress} = 0.707\ s \times l \times \tau$$

And shear strength of the joint fo r double parallel fillet weld,

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



(*a*) Double parallel fillet weld.

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

Fig.3

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

$$P = 0.707s \times l_1 \times \sigma_t + 1.414\ s \times l_2 \times \tau$$

Where l_1 is normally the width of the plate.

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.

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

Problem:

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

Solution. Given: *Width = 100 mm ; Thickness = 10 mm ; P = 80 kN = 80 × 10^3 N ; τ = 55 MPa = 55 N/mm^2

Let l = Length of weld, and

s = Size of weld = Plate thickness = 10 mm ... (Given)

We know that maximum load which the plates can carry for double parallel fillet weld (P),

$$80 \times 10^3 = 1.414 \times s \times l \times \tau = 1.414 \times 10 \times l \times 55 = 778\, l$$

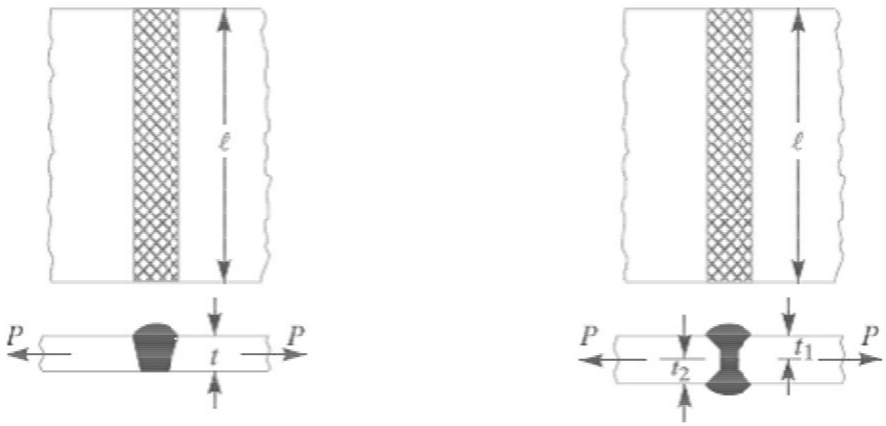
$$\therefore \quad l = 80 \times 10^3 / 778 = 103 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l = 103 + 12.5 = 115.5 \text{ mm} \quad \textbf{Ans.}$$

Strength of Butt Joints

The butt joints are designed for tension or compression. Consider a single V-butt joint as shown in Fig. 4(*a*).



(*a*) Single *V*-butt joint.

(*b*) Double *V*-butt joint.

Fig.4. Butt Joints

In case of butt joint, the length o f leg or size of weld is equal to the throat thickness which is equal to thickness of plates. Therefore, Tensile strength of the butt joint (single-*V* or square butt joint),

$$P = t \times l \times \sigma_1$$

Where l = Length of weld. It is generally equal to the width of plate. And tensile strength for double-*V* butt joint as shown in F ig. 4(*b*) is given by

$$P = (t_1 + t_2)\, l \times \sigma_t$$

Where t_1 = Throat thickness at th e top, and

t_2 = Throat thickness at th e bottom.

It may be noted that size of the weld should be greater than the thickness of the plate, but it may be less. The following table shows recommended minimum size of the welds.

Stresses for Welded Joints

The stresses in welded joints are difficult to determine because of the variable and unpredictable parameters like homogenuity of the weld metal, thermal stresses in the welds, changes of physical properties due to high rate of cooling etc. The stresses are obtained, on the following assumptions:

1. The load is distributed uniformly along the entire length of the weld, and
2. The stress is spread uniformly over its effective section.

The following table shows the stresses for welded joints for joining ferrous metals with mild steel electrode under steady and fatigue or reversed load.

Stress Concentration Factor for Welded Joints

The reinforcement provided to the weld produces stress concentration at the junction of the weld and the parent metal. When the parts are subjected to fatigue loading, the stress concentration factors should be taken into account.

Problem:

A plate 100 mm wide and 12.5 m m thick is to be welded to another plate by me ans of parallel fillet welds. The plates are subj cted to a load of 50 kN. Find the length of the weld so that the maximum stress does not ex ceed 56 MPa. Consider the joint first under static loading and then under fatigue loading.

Solution. Given: *Width = 100 mm ; Thickness = 12.5 mm ; P = 50 kN = 50×10^3 N ; τ = 56 MPa = 56 N/mm^2

Length of weld for static loading

Let l = Length of weld, and

s = Size of weld = Plate thickness

= 12.5 mm ... (Given)

We know that the maximum load which the plates can carry for double parallel fillet welds (P),

$$50 \times 10^3 = 1.414\, s \times l \times \tau$$

$$= 1.414 \times 12.5 \times l \times 56 = 990\, l$$

$$\therefore \quad l = 50 \times 10^3 / 990 = 50.5 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l = 50.5 + 12.5 = 63 \text{ mm} \quad \textbf{Ans.}$$

Length of weld for fatigue loading

From Table 10.6, we find that the stress concentration factor for parallel fillet welding is 2.7.

$\therefore$ Permissible shear stress,

$$\tau = 56 / 2.7 = 20.74 \text{ N/mm}^2$$

We know that the maximum load which the plates can carry for double parallel fillet welds (P),

$$50 \times 10^3 = 1.414\, s \times l \times \tau = 1.414 \times 12.5 \times l \times 20.74 = 367\, l$$

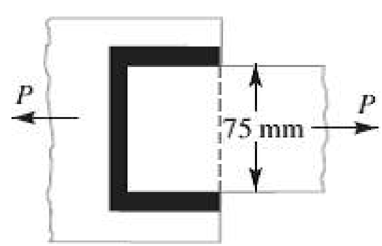
$$\therefore \quad l = 50 \times 10^3 / 367 = 136.2 \text{ mm}$$

Adding 12.5 for starting and stopping of weld run, we have

$$l = 136.2 + 12.5 = 148.7 \text{ mm} \quad \textbf{Ans.}$$

Problem:

A plate 75 mm wide and 12.5 mm thick is joined with another plate by a single trans verse weld and a double parallel fillet weld as shown in Fig. The maximum tensile and shear stresses are 70 MPa and 56 MPa respectively. Find the length of each parallel fillet weld, if the joint is subjected to both static and fatigue loading.



Solution. Given : Width = 75 mm ; Thickness = 12.5 mm ; $\sigma_\tau = 70$ MPa = 70 N/mm^2 ; $\tau = 56$ MPa = 56 N/mm^2.

The effective length of weld (l_1) for the transverse weld may be obtained by subtracting 12.5 mm from the width of the plate.

$$\therefore \quad l_1 = 75 - 12.5 = 62.5 \text{ mm}$$

Length of each parallel fillet for static loading

Let l_2 = Length of each parallel fillet.

We know that the maximum load which the plate can carry is

$$P = \text{Area} \times \text{Stress} = 75 \times 12.5 \times 70 = 65\ 625 \text{ N}$$

Load carried by single transverse weld,

$$P_1 = 0.707\, s \times l_1 \times \sigma_t = 0.707 \times 12.5 \times 62.5 \times 70 = 38\ 664 \text{ N}$$

and the load carried by double parallel fillet weld,

$$P_2 = 1.414\, s \times l_2 \times \tau = 1.414 \times 12.5 \times l_2 \times 56 = 990\, l_2 \text{ N}$$

$\therefore$ Load carried by the joint (P),

$$65\ 625 = P_1 + P_2 = 38\ 664 + 990\, l_2 \quad \text{or} \quad l_2 = 27.2 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l_2 = 27.2 + 12.5 = 39.7 \text{ say } 40 \text{ mm} \quad \textbf{Ans.}$$

Length of each parallel fillet for fatigue loading

From Table 10.6, we find that the stress concentration factor for transverse welds is 1.5 and for parallel fillet welds is 2.7.

$\therefore$ Permissible tensile stress,

$$\sigma_t = 70 / 1.5 = 46.7 \text{ N/mm}^2$$

and permissible shear stress,

$$\tau = 56 / 2.7 = 20.74 \text{ N/mm}^2$$

Load carried by single transverse weld,

$$P_1 = 0.707\, s \times l_1 \times \sigma_t = 0.707 \times 12.5 \times 62.5 \times 46.7 = 25\ 795 \text{ N}$$

and load carried by double parallel fillet weld,

$$P_2 = 1.414\, s \times l_2 \times \tau = 1.414 \times 12.5\, l_2 \times 20.74 = 366\, l_2 \text{ N}$$

$\therefore$ Load carried by the joint (P),

$$65\ 625 = P_1 + P_2 = 25\ 795 + 366\, l_2 \quad \text{or} \quad l_2 = 108.8 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l_2 = 108.8 + 12.5 = 121.3 \text{ mm} \quad \textbf{Ans.}$$

Fig.2.Circular fillet weld subjected to Ben

Contents: Special fillet welded j oints

Special Cases of Fillet Welded Joints

The following cases of fillet welded joints are important from the subject point o f view.

1. ***Circular fillet weld subjected to torsion.*** Consider a circular rod connected to a rigid plate by a fillet weld as shown in Fig. 1.

Let d = Diameter of rod, r = Radius of rod,

T = Torque acting on the rod, s = Size (or leg) of weld,

t = Throat thickness,

J = Polar moment of inertia of the weld section $= \dfrac{\pi t d^3}{4}$

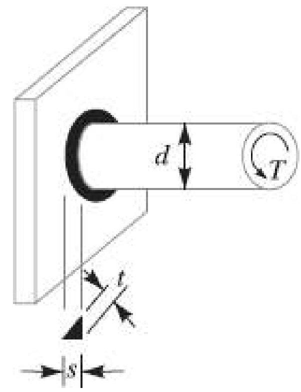


Fig. 1. Circular fillet weld subjecte d to torsion.

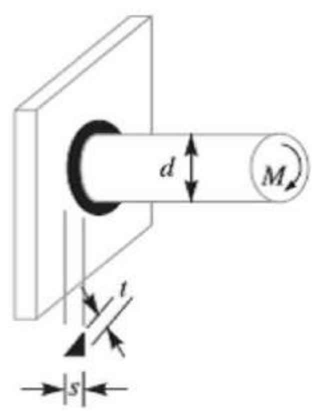
We know that shear stress for th e material,

$$\tau = \frac{T.r}{J} = \frac{T \times d/2}{J} = \frac{T \times d/2}{\pi t d^3/4} = \frac{2T}{\pi t d^2}$$

This shear stress occurs in a horizontal plane along a leg of the fillet weld. T he maximum shear occurs on the throat of weld which is inclined at 45° to the horizontal plane. Length of throat, $t = s \sin 45° = 0.707\, s$ and maximum shear stress,

$$\tau_{max} = \frac{2T}{\pi \times 0.707\, s \times d^2} = \frac{2.83\, T}{\pi\, s\, d^2}$$

2. Circular fillet weld subjected to bending moment. Consider a circular rod connect ed to a rigid plate by a fillet weld as shown in Fig.2.



Let d = Diameter of rod,

M = Bending moment acting on the rod, s = Size (or leg) of weld,

t = Throat thickness,

Z = Section modulus of the weld section

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

We know that the bending stress

$$\sigma_b = \frac{M}{Z} = \frac{M}{\pi t d^2/4} = \frac{4M}{\pi t d^2}$$

This bending stress occurs in a horizontal plane along a leg of the fillet weld. The maximum bending stress occurs on the throat of the weld which is inclined at 45° to the horizontal plane.

Length of throat, t = s sin 45° = 0.707 s and maximum bending stress,

$$\sigma_{b(max)} = \frac{4M}{\pi \times 0.707 s \times d^2} = \frac{5.66 M}{\pi s d^2}$$

3. Long fillet weld subjected to torsion. Consider a vertical plate attached to a horizontal plate by two identical fillet welds as shown in Fig.3.

Let T = Torque acting on the vertical plate, l = Length of weld,

s = Size (or leg) of weld, t = Throat thickness, and

J = Polar moment of inertia of the weld section

$$= 2 \times \frac{t \times l^3}{12} = \frac{t \times l^3}{6} \ldots$$

It may be noted that the effect of the applied torque is to rotate the vertical plate about the Z-axis through its mid point. This rotation is resisted by shearing stresses developed between two fillet welds and the horizontal plate. It is assumed that these horizontal shearing stresses vary from zero at the Z-axis and maximum at the ends of the plate. This variation of shearing stress is analogous to the variation of normal stress over the depth (l) of a beam subjected to pure bending.

Therefore, Shear stress,

$$\tau = \frac{T \times l/2}{t \times l^3/6} = \frac{3T}{t \times l^2}$$

The maximum shear stress occurs at the throat and is given by

$$\tau_{max} = \frac{3T}{0.707\, s \times l^2} = \frac{4.242\, T}{s \times l^2}$$

Axially Loaded Unsymmetrical WeldedSections

Sometimes unsymmetrical secti ons such as angles, channels, *T*-sections etc., welded on the flange edges are loaded axially a s shown in Fig. In such cases, the lengths of weld should be proportioned in such a way that the sum of resisting moments of the welds about the gravity axis is zero. Consider an angle s ection as shown in Fig. Let l_a = Length of weld at the to p,

l_b = Length of weld at the bottom ,

l = Total length of weld = $l_a + l_b$

P = Axial load,

a = Distance of top weld from gravity axis,

b = Distance of bottom weld fro m gravity axis, and

f = Resistance offered by the wel d per unit length.

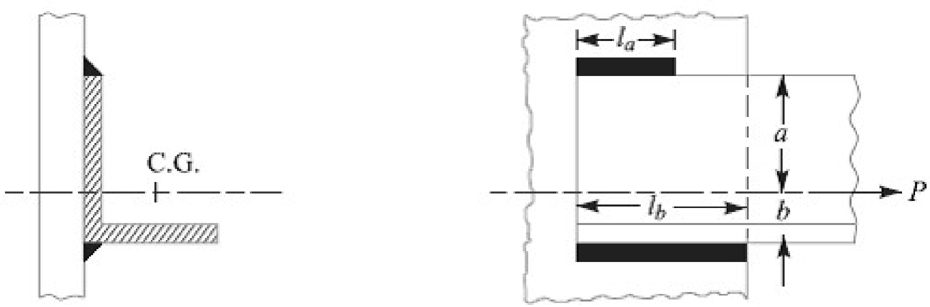


Fig. Axial ly loaded unsymmetrical welded section

Moment of the top weld about g ravity axis

$$= l_a \times f \times a$$

And moment of the bottom weld about gravity axis

$$= l_b \times f \times b$$

Since the sum of the moments of the weld about the gravity axis must be zero, therefore,

$$l_a \times f \times a - l_b \times f \times b = 0$$

or $l_a \times a = l_b \times b$ ...(***i***)

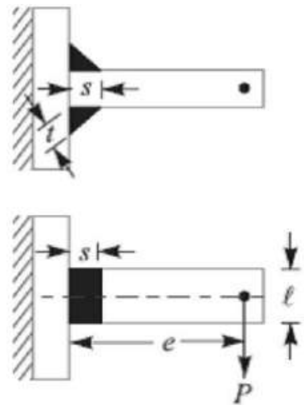
We know that $l = l_a + l_b$ ...(***ii***)

From equations **(*i*)** and **(*ii*),** we have

$$l_a = \frac{l \times b}{a + b}, \quad \text{and} \quad l_b = \frac{l \times a}{a + b}$$

Eccentrically Loaded Welded Joints

An eccentric load may be impo sed on welded joints in many ways. The stress es induced on the joint may be of different nat ure 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 foll ows: Maximum normal stress,



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

And Maximum shear stress,

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

Where σ_b = Bending stress, and

τ = Shear stress. Fig.1. Eccentrically loaded welded joint When the stresses are of the same nature, these may be combined vectorially (see case 2). We shall now discuss the two cases of eccentric loading as follows:

Case 1

Consider a *T*-joint fixed at one end and subjected to an eccentric load *P* at a distance *e* as shown in Fig. 1

Let *s* = Size of weld,

l = Length of weld, and

t = Throat thickness.

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

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

2. Bending stress due to the bend ing moment $P \times e$.

We know that area at the throat,

$$A = \text{T hroat thickness} \times \text{Length of weld}$$

$$= t \times l \times 2 = 2\,t \times l \text{ ... (For double fillet weld)}$$

$$= 2 \times 0.707\,s \times l = 1.414\,s \times l \text{ ... (since, t} = s \cos 45° = 0.707\,s)$$

Shear stress in the weld (assumi g uniformly distributed),

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

Section modulus of the weld met al through the throat,

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

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

Bending moment, $M = P \times e$

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

We know that the maximum normal stress,

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

And maximum shear stress,

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

Case 2

When a welded joint is loaded e ccentrically as shown in Fig.2, the following two types of the stresses are induced:

1. Direct or primary shear stress, and

2. Shear stress due to turning moment.

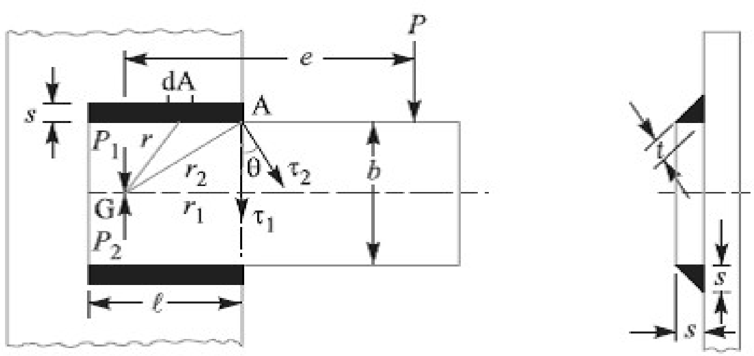


Fig.2 eccentrically loaded welded joint.

Let P = Eccentric load,

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

l = Length of single weld,

s = Size or leg of weld, a nd

t = Throat thickness.

Let two loads P_1 and P_2 (each e qual to P) are introduced at the centre of gravity 'G' of the weld system. The effect of load P_1 = P is to produce direct shear stress which is assumed to be uniform over the entire wel d length. The effect of load P_2 = P is to produce a turning moment of magnitude P × e whi ch tends of rotate the joint about the centre of gravity 'G' of the weld system. Due to the turning moment, secondary shear stress is induced. We know that the direct or prima ry shear stress,

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

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

Since the shear stress produced due to the turning moment (T = P × e) at a ny section is proportional to its radial distance from G, therefore stress due to P × e at t he point A is proportional to AG (r2) and is in a direction at right angles to AG. In other words,

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

$$\tau = \frac{\tau_2}{r_2} \times r \qquad \ldots(i)$$

Where τ_2 is the shear stress at the maximum distance (r_2) and τ is the shear stress at any distance r. Consider a small section of the weld having area dA at a distance r fr om G. Shear force on this small section

$$= \tau \times dA$$

And turning moment of this shea r force about G,

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

Total turning moment over the w hole weld area,

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

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

Where J = Polar moment of inertia of the throat area about G.
Shear stress due to the turning moment i.e. secondary shear stress,

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

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

Resultant shear stress at A,

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

θ = Angle between τ_1 and τ_2, and

$\cos\theta = r_1 / r_2$

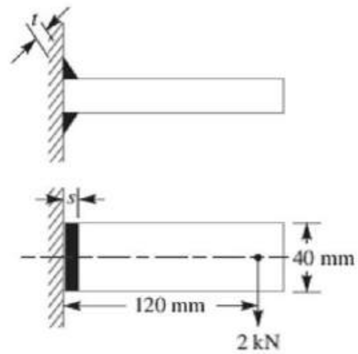
Problem:

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 st ress in the weld is 25 MPa.

Solution. Given: P = 2kN = 2000 N ; e = 120 mm ; l = 40 mm ; τ_{max} = 25 MPa = 25 N/mm^2

Let s = Size of weld in mm, and

t = Throat thickness.



The joint, as shown in Fig. 10.24, will be subjected to direct shear stress due to the shear force, P = 2000 N and bending stress due to the bending moment of $P \times e$.

We know that area at the throat,

$$A = 2t \times l = 2 \times 0.707\, s \times l = 1.414\, s \times l = 1.414\, s \times 40 = 56.56 \times s \text{ mm}^2$$

$\therefore$ Shear stress,

$$\tau = \frac{P}{A} = \frac{2000}{56.56 \times s} = \frac{35.4}{s} \text{ N/mm}^2$$

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

Section modulus of the weld through the throat,

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

$\therefore$ Bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{240 \times 10^3}{377 \times s} = \frac{636.6}{s} \text{ N/mm}^2$$

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

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

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

Problem:

A bracket carrying a load of 15 k N is to be welded as shown in Fig. Find the siz e of weld required if the allowable shear stress is not to exceed 80 MPa.

Solution. Given : P = 15 kN = 15 × 10^3 N ; τ = 80 MPa = 80 N/mm^2 ; b = 80 mm ; l = 50 mm; e = 125 mm

Let s = Size of weld in mm, and

t = Throat thickness.

We know that the throat area,

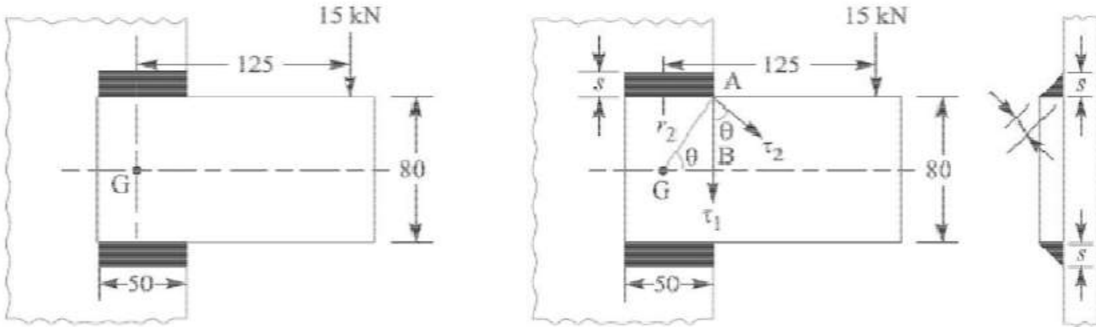
$$A = 2 \times t \times l = 2 \times 0.707\, s \times l = 1.414\, s \times l = 1.414 \times s \times 50 = 70.7\, s \text{ mm}^2$$

∴ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{15 \times 10^3}{70.7\ s} = \frac{212}{s}\ \text{N/mm}^2$$

$$J = \frac{t.l\,(3b^2 + l^2)}{6} = \frac{0.707\ s \times 50\,[3\,(80)^2 + (50)^2]}{6}\ \text{mm}^4$$

$$= 127\ 850\ s\ \text{mm}^4 \qquad \ldots (\because\ t = 0.707\ s)$$



All dimensions in mm.

∴ Maximum radius of the weld,

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

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

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{15 \times 10^3 \times 125 \times 47}{127\ 850\ s} = \frac{689.3}{s}\ \text{N/mm}^2$$

and $$\cos\theta = \frac{r_1}{r_2} = \frac{25}{47} = 0.532$$

We know that resultant shear stress,

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

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

∴ $$s = 822 / 80 = 10.3\ \text{mm}\ \ \textbf{Ans.}$$

Introduction to 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.**

A screwed joint is mainly composed of two elements i.e. a bolt and nut. The screwed joints are widely used where the machine parts are required to be readily connected or disconnected without damage to the machine or the fastening. This may be for the purpose of holding or adjustment in assembly or service inspection, repair, or replacement or it may be for the manufacturing or assembly reasons. The parts may be rigidly connected or provisions may be made for predetermined relative motion.

Advantages and Disadvantages of Screwed Joints

Following are the advantages and disadvantages of the screwed joints.

Advantages

3. Screwed joints are highly reliable in operation.

4. Screwed joints are convenient to assemble and disassemble.

5. A wide range of screwed joints may be adapted to various operating conditions.

6. Screws are relatively cheap to produce due to standardization and highly efficient manufacturing processes.

Disadvantages

The main disadvantage of the screwed joints is the stress concentration in the threaded portions which are vulnerable points under variable load conditions.

Note : The strength of the screwed joints is not comparable with that of riveted or welded joints.

Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. 1, are important from the subject point of view:

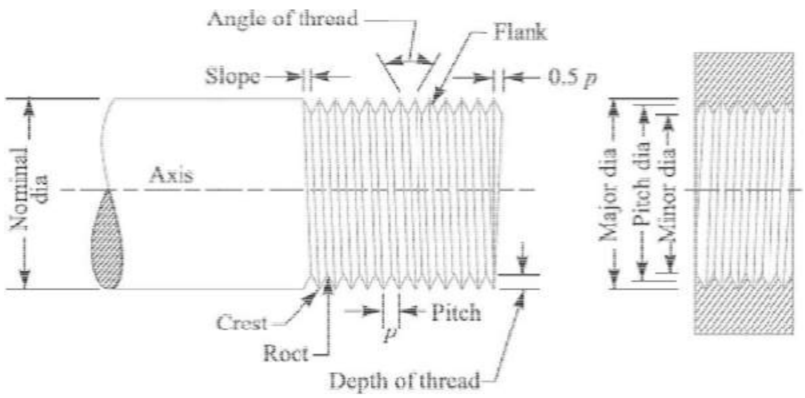


Fig .1 Terms used in screw threads

3. Major diameter. It is the largest diameter of an external or internal scre w thread. The screw is specified by this diameter. It is also known as **outside** or **nominal diam eter.**

4. Minor diameter. It is the sm allest diameter of an external or internal scre w thread. It is also known as **core** or **root diam eter.**

5. Pitch diameter. It is the diam eter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make eq ual the width of the thread and the width of the spaces between the threads. It is also calle d 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.

6. 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. Mathematically,

$$\text{Pitch} = \frac{1}{\text{No. of threads per unit length of screw}}$$

7. Lead. It is the distance betwee n two corresponding points on the same helix. It may also be defined as the distance which a s crew 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.

2 **Crest.** It is the top surface of the thread.

3 **Root.** It is the bottom surface created by the two adjacent flanks of the thread.

4 **Depth of thread.** It is the per endicular distance between the crest and root.

5 **Flank.** It is the surface joining the crest and root.

2. **Angle of thread.** It is the angle included by the flanks of the thread.

3. **Slope.** It is half the pitch of the thread.

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:

3. Internal stresses due to screwing up forces,

4. Stresses due to external forces, and

5. Stress due to combination of stresses at (1) and (2).

Initial Stresses due to Screwing up Forces

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

3. **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

$$P_i = 2840\ d\ \text{N}$$

Where Pi = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm.

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint, then the initial tension in a bolt may be reduced to half of the above value. In such cases

$$P_i = 1420\ d\ \text{N}$$

The small diameter bolts may fail during tightening, therefore bolts of smaller diameter (less than M 16 or M 18) are not permitted in making fluid tight joints. If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it, is given by

P = Permissible stress × Cross-sectional area at bottom of the thread

$$\text{Stress area} = \frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$$

Where d_p = Pitch diameter, and

d_c = Core or minor diameter.

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. Let

d_c = Root or core diameter of the thread, and

σ_t = Permissible tensile stress for the bolt material.

We know that external load applied,

$$d_c - \sqrt{\frac{4P}{\pi \sigma_t}}$$

Notes: (a) if the external load is taken up by a number of bolts, then

$$P = \frac{\pi}{4} (d_c)^2 \sigma_t \times n$$

5. In case the standard table is not available, then for coarse threads, $d_c = 0.84$ d, where d is the nominal diameter of bolt.

2. Shear stress. Sometimes, the bolts are used to prevent the relative movement of two or more parts, as in case of flange coupling, and 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.

Shearing load carried by the bolts,

$$P_s = \frac{\pi}{4} \times d^2 \times \tau \times n \quad \text{or} \quad d = \sqrt{\frac{4\, P_s}{\pi \tau n}}$$

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,

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

And maximum principal tensile stress,

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2} \sqrt{(\sigma_2)^2 + 4\tau^2}$$

These stresses should not exceed the safe permissible values of stresses.

<u>Stress due to Combined Forces</u>

The resultant axial load on a bolt depends upon the following factors:

6. The initial tension due to tightening of the bolt,

7. The external load, and

8. The relative elastic yielding (springiness) of the bolt and the connected members.

When the connected members a re very yielding as compared with the bolt, which is a soft gasket, as shown in Fig. 1 (*a*), then the resultant load on the bolt is approximately equal to the sum of the initial tension and th e external load. On the other hand, if the bolt is very yielding as compared with the connected members, as shown in Fig. 1(*b*), then the resul tant load will be either the initial tension or th e external load, whichever is greater. The act ual conditions usually lie between the two extre mes. In order to determine the resultant axial lo ad (*P*) on the bolt, the following equation may be used :

$$P = P_1 + \frac{a}{1+a} \times P_2 = P_1 + K.P_2 \qquad \ldots\left(\text{Substituting } \frac{a}{1+a} = K\right)$$

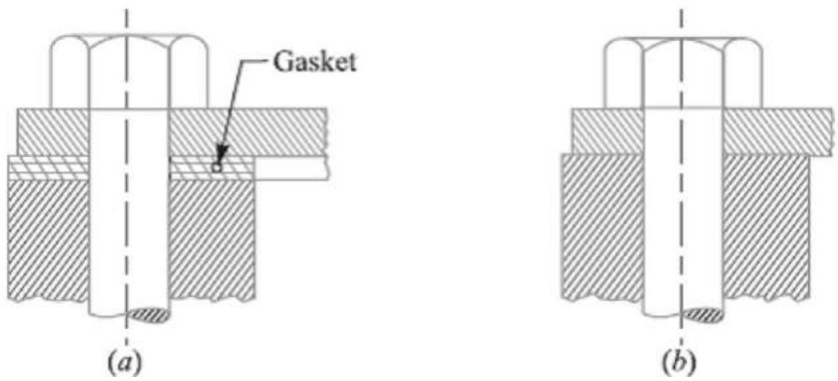


Fig.1

Where P_1 = Initial tension due to tightening of the bolt, P_2 = External load on the bolt, and

a = Ratio of elasticity of connected parts to the elasticity of bolt.

For soft gaskets and large bolts, the value of *a* is high and the value of a/ (1+a) is approximately equal to unity, so that the resultant load is equal to the sum of the initial tension and the external load. For hard gaskets or metal to metal contact surf aces and with small bolts, the value of a is small and the resultant load is mainly due to the initial tension (or external load, in rare case it is greater than initial tension). The value of '*a*' may be estimated by the designer to obt ain an approximate value for the resultant load. The values of a/(1+ *a)* (*i.e*. *K*) for various type of joints are shown in the following table. The designer thus has control over the influence on the resultant load on a bolt by proportioning th e sizes of the connected parts and bolts and by specifying initial tension in the bolt.

Values of K for various types of joints.

Type of joint	$K = \frac{a}{1+a}$
Metal to metal joint with through bolts	0.00 to 0.10
Hard copper gasket with long through bolts	0.25 to 0.50
Soft copper gasket with long through bolts	0.50 to 0.75
Soft packing with through bolts	0.75 to 1.00
Soft packing with studs	1.00

Design of Cylinder Covers

The cylinder covers may be sec ured by means of bolts or studs, but studs are p referred. The possible arrangement of securin the cover with bolts and studs is shown in Fig . 2 (*a*) and (*b*) respectively. The bolts or studs, cylinder cover plate and cylinder flange may be designed as discussed below:

(c) Design of bolts or studs

In order to find the size and n umber of bolts or studs, the following proce dure may be adopted.

Let D = Diameter of the cyli der,

p = Pressure in the cylind er,

d_c = Core diameter of the bolts or studs,

n = Number of bolts or studs, and

σ_{tb} = Permissible tensile stress for the bolt or stud material.

We know that upward force acti ng on the cylinder cover,

$$P = \frac{\pi}{4}(D^2)\, p \qquad ...(i)$$

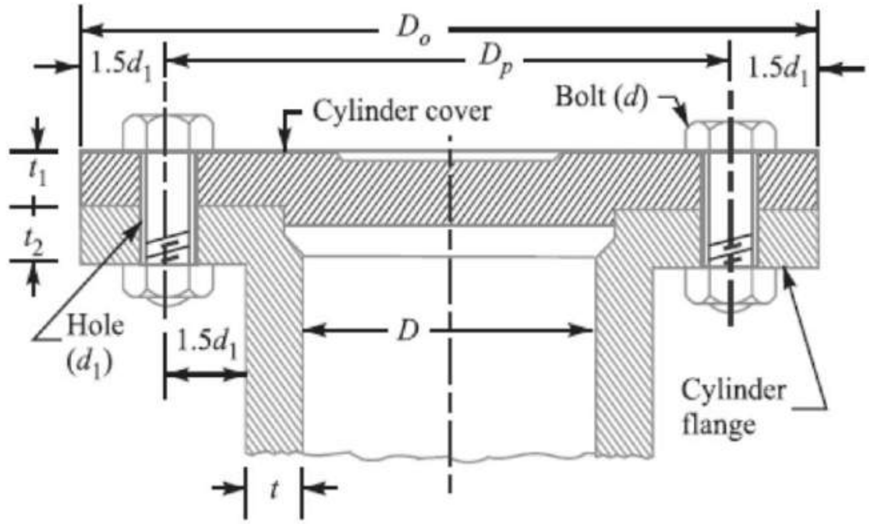
This force is resisted by *n* numbe r of bolts or studs provided on the cover.

Resisting force offered by *n* num ber of bolts or studs,

$$P = \frac{\pi}{4}(d_c)^2\, \sigma_{tb} \times n \qquad ...(ii)$$

From equations **(*i*)** and **(*ii*),** we have

$$\frac{\pi}{4}(D^2)\,p = \frac{\pi}{4}(d_c)^2\,\sigma_{tb} \times n \qquad ...(ii)$$



(*a*) Arrangement of securing the cylinder cover with bolts.

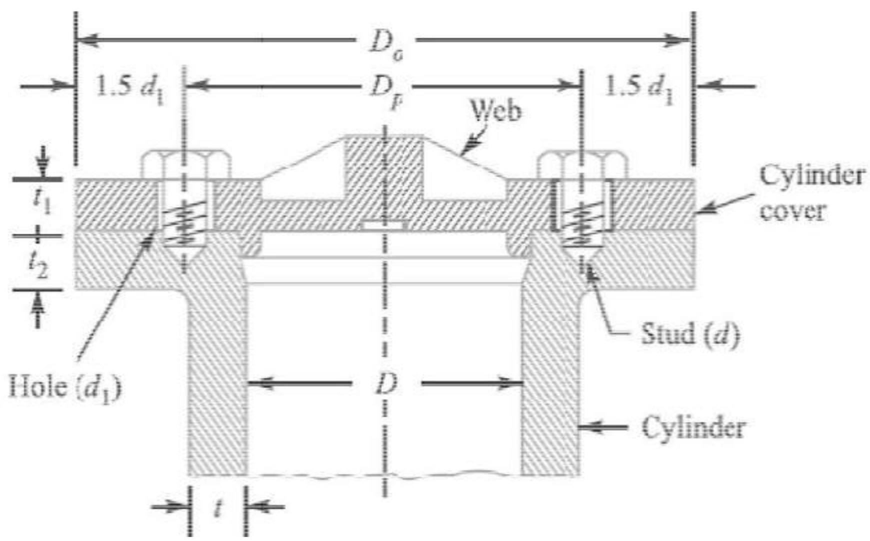


Fig. 2.

From this equation, the number of bolts or studs may be obtained, if the size of the bolt or stud is known and *vice-versa*. Usually the size of the bolt is assumed. If the value of n as obtained from the above relation is odd or a fraction, then next higher ev en number is adopted. The bolts or studs ar e screwed up tightly, along with metal gaske t or asbestos packing, in order to provide a leak proof joint. We have already discussed th at due to the tightening of bolts, sufficient te nsile stress is produced in the bolts or studs. Thhis may break the bolts or studs, even before any load due to internal pressure acts upon them . Therefore a bolt or a stud less than 16 mm diameter should never be used.

The tightness of the joint also d epends upon the circumferential pitch of the b olts or studs. The circumferential pitch should be between 20 d_1 and 30 d_1 , where d_1 is the d iameter of the hole in mm for bolt or stud. Th e pitch circle diameter (D_p) is usually taken as $D + 2t + 3d_1$ and outside diameter of the cover is kept as

$$D_0 = Dp + 3d_1 = D + 2t +$$

$6d_1$ where t = Thickness of the cylin der wall.

2. Design of cylinder cover plate

The thickness of the cylinder co ver plate (t_1) and the thickness of the cylinder flange (t_2) may be determined as discussed belo w:

Let us consider the semi-cover plate as shown in Fig. 3. The internal pressure in the cylinder tries to lift the cylinder cover while the bolts or studs try to retain it in its position. But the centres of pressure of these two loads do not coincide. Hence, the cover plate is subjected to bending stress. The point X is the centre of pressure for bolt load a nd the point Y is the centre of internal pressure.

We know that the bending mom ent at A-A,

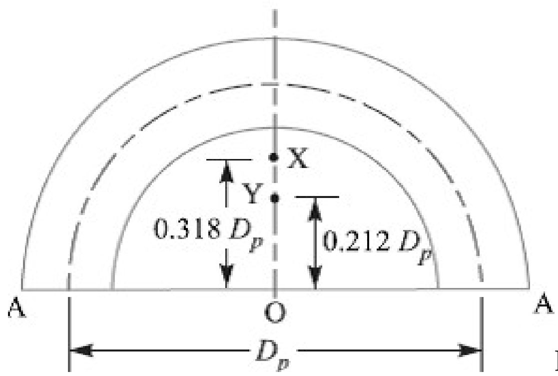


Fig.3

$$M = \frac{\text{Total bolt load}}{2}(OX - OY) = \frac{P}{2}(0.318\ D_P - 0.212\ D_P)$$

$$= \frac{P}{2} \times 0.106\ D_p = 0.053\ P \times D_p$$

$$Z = \frac{1}{6} w\,(t_1)^2$$

Where w = Width of plate

5.Outside dia. of cover plate – 2 × dia. of bolt hole

6.$D_0 - 2d_1$

Knowing the tensile stress for th e cover plate material, the value of t_1 may be d etermined by using the bending equation,

$$i.e., \sigma t = M / Z.$$

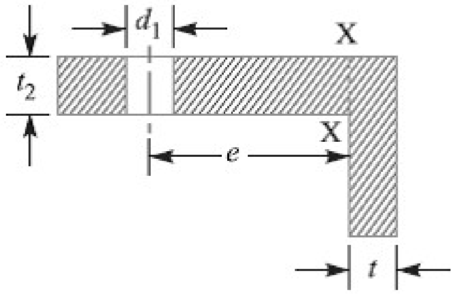


Fig.4

3. Design of cylinder flange

The thickness of the cylinder flange (t_2) may be determined from bending con sideration. A portion of the cylinder flange un der the influence of one bolt is shown in Fig. 4. The load in the bolt produces bending stress in the section *X*-*X*. From the geometry of the fi gure, we find that eccentricity of the load from section *X*-*X* is

e = Pitch circle radius – (Radius of bolt hole +Thickness of cylinder wall)

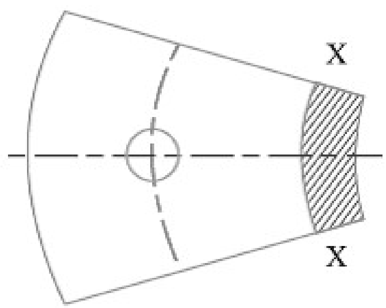


Fig.5

$$= \frac{D_p}{2} - \left(\frac{d_1}{2} + t\right)$$

Bending moment, *M* = L oad on each bolt × *e*

$$= \frac{P}{n} \times e$$

R = Cylinder radius + Thickness of cylinder wall

$$= \frac{D}{2} + t$$

Width of the section *X*-*X*,

$$w = \frac{2\pi R}{n},$$ Where *n* is the number of bolts.

Section modulus,

$$Z = \frac{1}{6} w (t_2)^2$$

Knowing the tensile stress for the cylinder flange material, the value of t_2 may be obtained by using the bending equation *i.e.* $\sigma_t = M / Z$.

Problem:

A steam engine cylinder has an effective diameter of 350 mm and the ma ximum steam pressure acting on the cylinder c over is 1.25 N/mm^2. Calculate the number and size of studs required to fix the cylinder cover , assuming the permissible stress in the studs as 33 MPa.

Solution. Given: $D = 350$ mm ; $p = 1.25$ N/mm^2 ; $\sigma_t = 33$ MPa $= 33$ N/mm^2

Let d = Nominal diameter of studs,

d_c = Core diameter of studs, and

n = Number of studs.

We know that the upward force acting on the cylinder cover,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (350)^2\ 1.25 = 120\ 265 \text{ N} \quad ...(i)$$

Assume that the studs of nominal diameter 24 mm are used. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the stud is 20.32 mm.

∴ Resisting force offered by n number of studs,

$$P = \frac{\pi}{4} \times (d_c)^2\ \sigma_t \times n = \frac{\pi}{4} (20.32)^2\ 33 \times n = 10\ 700\ n \text{ N} \quad ...(ii)$$

From equations (*i*) and (*ii*), we get

$$n = 120\ 265 / 10\ 700 = 11.24 \text{ say } 12 \textbf{ Ans.}$$

Taking the diameter of the stud hole (d_1) as 25 mm, we have pitch circle diameter of the studs,

$$D_p = D_1 + 2t + 3d_1 = 350 + 2 \times 10 + 3 \times 25 = 445 \text{ mm}$$

...(Assuming $t = 10$ mm)

∴ *Circumferential pitch of the studs

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 445}{12} = 116.5 \text{ mm}$$

We know that for a leak-proof joint, the circumferential pitch of the studs should be between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of stud hole in mm.

∴ Minimum circumferential pitch of the studs

$$= 20\sqrt{d_1} = 20\sqrt{25} = 100 \text{ mm}$$

and maximum circumferential pitch of the studs

$$= 30\sqrt{d_1} = 30\sqrt{25} = 150 \text{ mm}$$

Since the circumferential pitch of the studs obtained above lies within 100 mm to 150 mm, therefore the size of the bolt chosen is satisfactory.

∴ Size of the bolt = M 24 **Ans.**

Problem:

A mild steel cover plate is to be designed for an inspection hole in the shell of a pressure vessel. The hole is 120 mm in diameter and the pressure inside the vessel is 6 N/mm ^{2}. Design the cover plate along with the bo lts. Assume allowable tensile stress for mild st eel as 60 MPa and for bolt material as 40 MPa.

Solution. Given : D = 120 mm or r = 60 mm ; p = 6 N/mm^2 ; σ_t = 60 MPa = 60 N/mm^2 ; σ_{tb} = 40 MPa = 40 N/mm^2

First for all, let us find the thickness of the pressure vessel. According to Lame's equation, thickness of the pressure vessel,

$$t = r\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right] = 60\left[\sqrt{\frac{60+6}{60-6}} - 1\right] = 6 \text{ mm}$$

Let us adopt $t = 10$ mm

Design of bolts

Let d = Nominal diameter of the bolts,

d_c = Core diameter of the bolts, and

n = Number of bolts.

We know that the total upward force acting on the cover plate (or on the bolts),

$$P = \frac{\pi}{4}(D)^2\ p = \frac{\pi}{4}(120)^2 6 = 67\ 867 \text{ N} \quad ...(i)$$

Let the nominal diameter of the bolt is 24 mm. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the bolt is 20.32 mm.

∴ Resisting force offered by n number of bolts,

$$P = \frac{\pi}{4}(d_c)^2\ \sigma_{tb} \times n = \frac{\pi}{4}(20.32)^2\ 40 \times n = 67\ 867 \text{ N} = 12\ 973\ n \text{ N} \quad ...(ii)$$

From equations (i) and (ii), we get

$$n = 67\ 867 / 12\ 973 = 5.23 \text{ say } 6$$

Taking the diameter of the bolt hole (d_1) as 25 mm, we have pitch circle diameter of bolts,

$$D_p = D + 2t + 3d_1 = 120 + 2 \times 10 + 3 \times 25 = 215 \text{ mm}$$

∴ Circumferential pitch of the bolts

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 215}{6} = 112.6 \text{ mm}$$

We know that for a leak proof joint, the circumferential pitch of the bolts should lie between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of the bolt hole in mm.

∴ Minimum circumferential pitch of the bolts

$$= 20\sqrt{d_1} = 20\sqrt{25} = 100 \text{ mm}$$

and maximum circumferential pitch of the bolts

$$= 30\sqrt{d_1} = 30\sqrt{25} = 150 \text{ mm}$$

Since the circumferential pitch of the bolts obtained above is within 100 mm and 150 mm, therefore size of the bolt chosen is satisfactory.

$\therefore$ Size of the bolt = M 24 **Ans.**

Design of cover plate

Let t_1 = Thickness of the cover plate.

The semi-cover plate is shown in Fig. 11.27.

We know that the bending moment at *A-A*,

$$M = 0.053\, P \times D_p = 0.053 \times 67\,860 \times 215 = 773\,265 \text{ N-mm}$$

Outside diameter of the cover plate,

$$D_o = D_p + 3d_1 = 215 + 3 \times 25 = 290 \text{ mm}$$

Width of the plate,

$$w = D_o - 2d_1 = 290 - 2 \times 25 = 240 \text{ mm}$$

$\therefore$ Section modulus,

$$Z = \frac{1}{6} w (t_1)^2 = \frac{1}{6} \times 240 (t_1)^2 = 40 (t_1)^2 \text{ mm}^3$$

Contents: Design of bolted joints under eccentric loading-

We know that bending (tensile) stress,

$$\sigma_t = M/Z \quad \text{or} \quad 60 = 773\,265 / 40 (t_1)^2$$

$$\therefore \quad (t_1)^2 = 773\,265 / 40 \times 60 = 322 \quad \text{or} \quad t_1 = 18 \text{ mm}$$ **Ans.**

1 **<u>Eccentric Load Acting Parallel to the Axis of Bolts</u>**

Consider a bracket having a rect angular base bolted to a wall by means of four bolts as shown in Fig.1. A little consideration will show that each bolt is subjected to a direct tensile load of

$$W_{t1} = \frac{W}{n}, \text{ where } n \text{ is the number of bolts.}$$

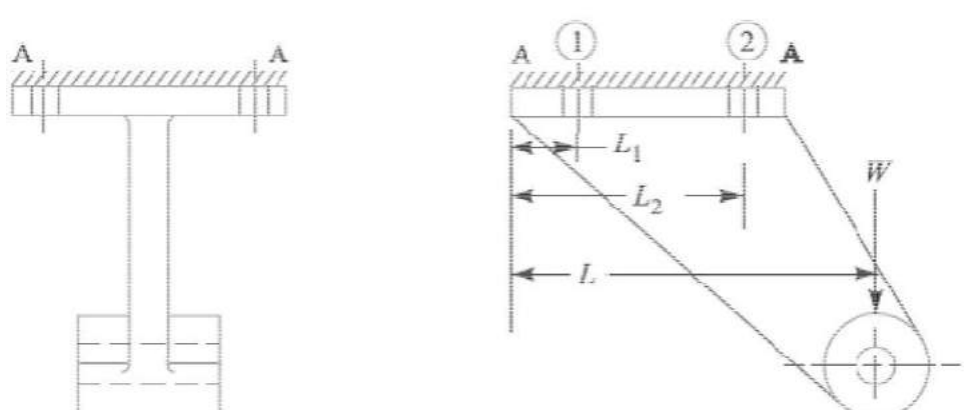


Fig.1. Eccentric load acting parallel to the axis of bolts.

Further the load *W* tends to rota te the bracket about the edge *A-A*. Due to this , each bolt is stretched by an amount that depe nds upon its distance from the tilting edge. Sinc e the stress is a function of elongation, there fore each bolt will experience a different load which also depends upon the distance from the tilting edge. For convenience, all the bolt s are made of same size. In case the flange is heavy, it may be considered as a rigid body.

Let w be the load in a bolt per unit distance due to the turning effect of the bracket and let W_1 and W_2 be the loads on each of the bolts at distances L_1 and L_2 fr om the tilting edge.

Load on each bolt at distance L_1,

$$W_1 = w.L_1$$

And moment of this load about the tilting edge

66 $w.L_1 \times L_1 = w\,(L_1)^2$

Similarly, load on each bolt at distance L_2,

$$W_2 = w.L_2$$

And moment of this load about the tilting edge

$$= w.L_2 \times L_2 = w\,(L_2)^2$$

So, Total moment of the load on the bolts about the tilting edge

$$= 2w\,(L_1)^2 + 2w\,(L_2)^2 \quad \textbf{...(}\boldsymbol{i}\textbf{)}$$

... (Since, there are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

2. $W.L$ **... (*ii*)**

From equations **(*i*)** and **(*ii*)**, we have

$$W.L = 2w\,(L_1)^2 + 2w(L_2)^2 \qquad \text{or} \qquad w = \frac{W.L}{2\,[(L_1)^2 + (L_2)^2]} \qquad ...(iii)$$

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance $L2$ are heavily loaded.

So, Tensile load on each bolt at distance L_2,

$$W_{t2} = W_2 = w.L_2 = \frac{W.L.L_2}{2[(L_1)^2 + (L_2)^2]} \qquad \text{... [From equation } (iii)]$$

And the total tensile load on the most heavily loaded bolt,

$$W_t = W_{t1} + W_{t2} \quad \textbf{... (}\boldsymbol{iv}\textbf{)}$$

If d_c is the core diameter of the olt and σt is the tensile stress for the bolt mate rial, then total tensile load,

$$W_t = \frac{\pi}{4}\,(d_c)^2\,\sigma_t \qquad ...(v)$$

From equations **(*iv*)** and **(*v*),** the value of dc may be obtained.

Problem:

A bracket, as shown in Fig.1, supports a load of 30 kN. Determine the size of bolts, if the maximum allowable tensile stre ss in the bolt material is 60 MPa. The distance s are: L_1 = 80 mm, L_2 = 250 mm, and L = 500 mm.

Solution. Given : $W = 30$ kN ; $\sigma_t = 60$ MPa $= 60$ N/mm^2 ; $L_1 = 80$ mm ; $L_2 = 250$ mm ; $L = 500$ mm

We know that the direct tensile load carried by each bolt,

$$W_{t1} = \frac{W}{n} = \frac{30}{4} = 7.5 \text{ kN}$$

and load in a bolt per unit distance,

$$w = \frac{W.L}{2\,[(L_1)^2 + (L_2)^2]} = \frac{30 \times 500}{2\,[(80)^2 + (250)^2]} = 0.109 \text{ kN/mm}$$

Since the heavily loaded bolt is at a distance of L_2 mm from the tilting edge, therefore load on the heavily loaded bolt,

$$W_{t2} = w.L_2 = 0.109 \times 250 = 27.25 \text{ kN}$$

∴ Maximum tensile load on the heavily loaded bolt,

$$W_t = W_{t1} + W_{t2} = 7.5 + 27.25 = 34.75 \text{ kN} = 34\ 750 \text{ N}$$

Let d_c = Core diameter of the bolts.

We know that the maximum tensile load on the bolt (W_t),

$$34\ 750 = \frac{\pi}{4}(d_c)^2\ \sigma_t = \frac{\pi}{4}(d_c)^2\ 60 = 47\ (d_c)^2$$

$$\therefore \quad (d_c)^2 = 34\ 750 / 47 = 740$$

or $$d_c = 27.2 \text{ mm}$$

From DDB (coarse series), we find that the standard core diameter of the bolt is 28.706 mm and the corresponding size of the bolt is M 33. **Ans.**

Eccentric Load Acting Perpen dicular to the Axis of Bolts

A wall bracket carrying an eccen tric load perpendicular to the axis of the bolts is shown in Fig.2.

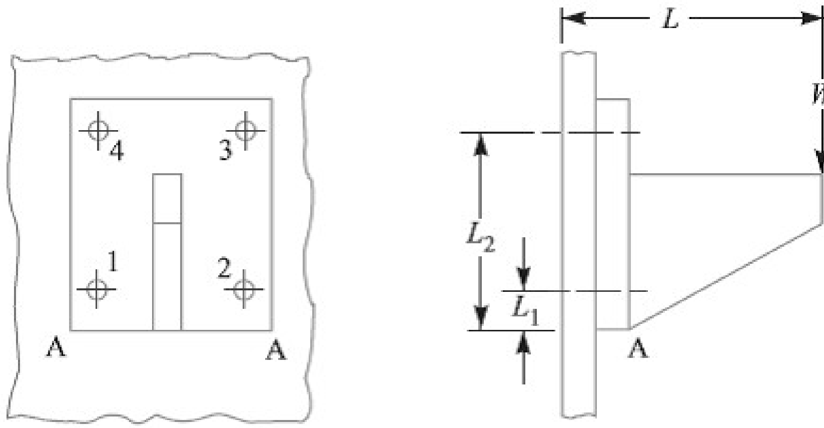


Fig. 2. Eccentric load perpendicular to the axis of bolts.

In this case, the bolts are subjected to direct shearing load which is equally shared by all the bolts. Therefore direct shear load on each bolts,

$W_s = W/n$, where n is num ber of bolts.

A little consideration will show that the eccentric load W will try to tilt the bracket in the clockwise direction about the e dge A-A. As discussed earlier, the bolts will bee subjected to tensile stress due to the turning moment. The maximum tensile load on a heavily loaded bolt (W_t) may be obtained in the similar manner as discussed in the previous article. In this case, bolts 3 and 4 are heavily loaded.

Maximum tensile load on bolt 3 or 4,

$$W_{t2} = W_t = \frac{W.L.L_2}{2\left[(L_1)^2 + (L_2)^2\right]}$$

When the bolts are subjected to shear as well as tensile loads, then the equival ent loads may be determined by the following relations:

Equivalent tensile load,

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

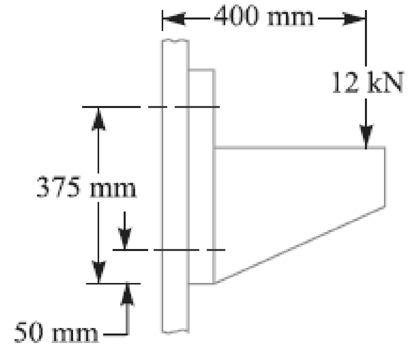
And equivalent shear load,

$$W_{se} = \frac{1}{2}\left[\sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

Knowing the value of equivalent loads, the size of the bolt may be determined for the given allowable stresses.

Problem:

For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in Fig. The maximum load that comes on the bracket is 12 kN acting verticall y at a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for the bolt material is 84 MPa. Also find the cross-section of the arm of the bracket which is rectangular.



Solution. Given : $W = 12$ kN $= 12 \times 10^3$ N ; $L = 400$ mm ; $L_1 = 50$ mm ; $L_2 = 375$ mm ; $\sigma_t = 84$ MPa $= 84$ N/mm^2 ; $n = 4$

We know that direct shear load on each bolt,

$$W_s = \frac{W}{n} = \frac{12}{4} = 3 \text{ kN}$$

Since the load *W* will try to tilt the bracket in the clockwise direction about the lower edge, therefore the bolts will be subje cted to tensile load due to turning moment. T he maximum loaded bolts are 3 and 4 (See Fig.1), because they lie at the greatest distance from the tilting edge *A–A* (*i.e.* lower edge).

We know that maximum tensile load carried by bolts 3 and 4,

$$W_t = \frac{W.L.L_2}{2\,[(L_1)^2 + (L_2)^2]} = \frac{12 \times 400 \times 375}{2\,[(50)^2 + (375)^2]} = 6.29 \text{ kN}$$

Since the bolts are subjected to shear load as well as tensile load, therefore equiivalent tensile load,

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right] = \frac{1}{2}\left[6.29 + \sqrt{(6.29)^2 + 4 \times 3^2}\right] \text{kN}$$

$$= \frac{1}{2}(6.29 + 8.69) = 7.49 \text{ kN} = 7490 \text{ N}$$

Size of the bolt

Let d_c = Core diameter of the bolt.

We know that the equivalent tensile load (W_{te}),

$$7490 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2 84 = 66(d_c)^2$$

$\therefore$ $(d_c)^2 = 7490 / 66 = 113.5$ or $d_c = 10.65$ mm

From Table 11.1 (coarse series), the standard core diameter is 11.546 mm and the corresponding size of the bolt is M 14. **Ans.**

Cross-section of the arm of the bracket

Let t and b = Thickness and depth of arm of the bracket respectively.

$\therefore$ Section modulus,

$$Z = \frac{1}{6} t.b^2$$

Assume that the arm of the bracket extends upto the face of the steel column. This assumption gives stronger section for the arm of the bracket.

$\therefore$ Maximum bending moment on the bracket,

$$M = 12 \times 10^3 \times 400 = 4.8 \times 10^6 \text{ N-mm}$$

We know that the bending (tensile) stress (σ_t),

$$84 = \frac{M}{Z} = \frac{4.8 \times 10^6 \times 6}{t.b^2} = \frac{28.8 \times 10^6}{t.b^2}$$

$\therefore$ $t.b^2 = 28.8 \times 10^6 / 84 = 343 \times 10^3$ or $t = 343 \times 10^3 / b^2$

Assuming depth of arm of the bracket, $b = 250$ mm, we have

$$t = 343 \times 10^3 / (250)^2 = 5.5 \text{ mm } \textbf{Ans.}$$

Eccentric Load on a Bracket w ith Circular Base

Sometimes the base of a bracket is made circular as in case of a flanged beari ng of a heavy machine tool and pillar crane etc. Consider a round flange bearing of a machin e tool having four bolts as shown in Fig. 1.

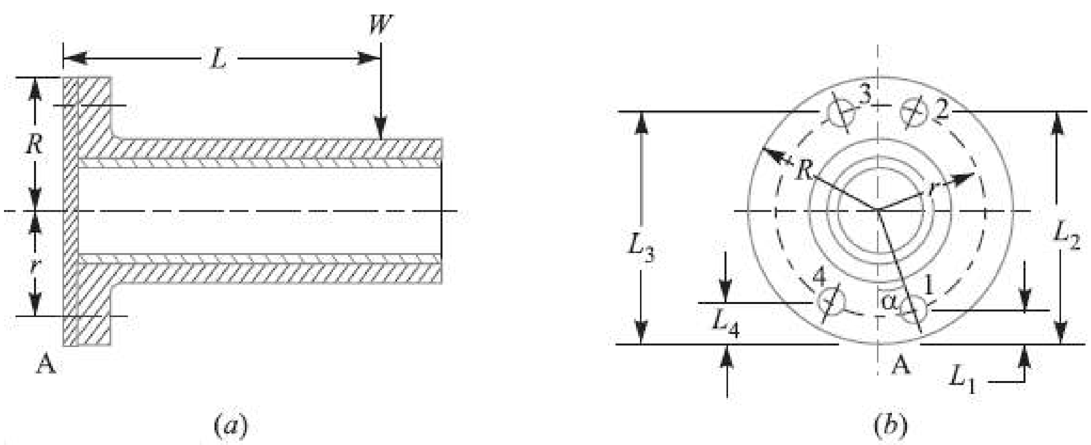


Fig.1. Ecce tric load on a bracket with circular base.

Let R = Radius of the column flange,

r = Radius of the bolt pit ch circle,

w = Load per bolt per uni t distance from the tilting edge,

L = Distance of the load from the tilting edge, and

L_1, L_2, L_3, and L_4 = Dista nce of bolt centers from the tilting edge A.

As discussed in the previous article, equating the external moment $W \times L$ to t he sum of the resisting moments of all the bolt s, we have,

$$W.L = w[(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2]$$

$$\therefore \quad w = \frac{W.L}{(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2} \quad \ldots(i)$$

Now from the geometry of the Fig. 1(b), we find that

$L_1 = R - r \cos \alpha$ $L_2 = R + r \sin \alpha$

$L_3 = R + r \cos \alpha$ and $L_4 = R - r \sin$

α Substituting these values in equation (i), we get

$$w = \frac{W.L}{4R^2 + 2r^2}$$

Load in the bolt situated at 1 = $w.L_1$ =

$$\frac{W.L.L_1}{4R^2 + 2r^2} = \frac{W.L(R - r\cos\alpha)}{4R^2 + 2r^2}$$

This load will be maximum when cos α is minimum *i.e.* when $\cos \alpha = -1$ or $\alpha = 180°$.

Maximum load in a bolt

$$= \frac{W.L(R + r)}{4R^2 + 2r^2}$$

In general, if there are n number of bolts, then load in a bolt

$$= \frac{2W.L(R - r\cos\alpha)}{n(2R^2 + r^2)}$$

And maximum load in a bolt,

$$W_t = \frac{2W.L(R + r)}{n(2R^2 + r^2)}$$

The above relation is used when the direction of the load W changes with relatioon to the bolts as in the case of pillar crane. But if the direction of load is fixed, then the maxi mum load on the bolts may be reduced by lo cating the bolts in such a way that two of the m are equally stressed as shown in Fig.2. In su ch a case, maximum load is given by

$$W_t = \frac{2\,W.L}{n}\left[\frac{R + r\cos\left(\frac{180}{n}\right)}{2R^2 + r^2}\right]$$

Knowing the value of maximum load, we can determine the size of the bolt.

Note: Generally, two dowel pins as shown in Fig. 2, are used to take up the shear load. Thus the bolts are relieved of shear stress and the bo lts are designed for tensile load only.

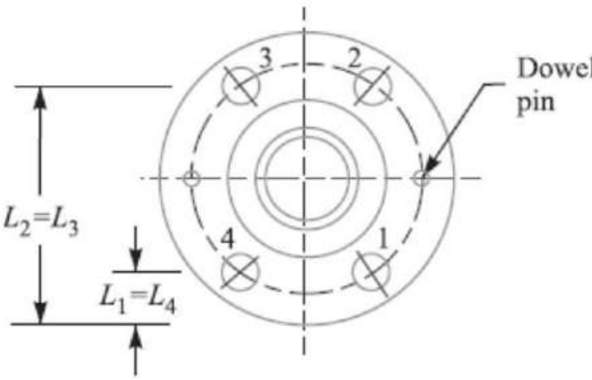


Fig.2.

Problem:

A flanged bearing, as shown in Fig.1, is fastened to a frame by means of four bolts spaced equally on 500 mm bolt circle. The diameter of bearing flange is 650 mm and a load of 400 kN acts at a distance of 250 mm from the frame. Determine the size of the bolt s, taking safe tensile stress as 60 MPa for the material of the bolts.

Solution. Given : $n = 4$; $d = 500$ mm or $r = 250$ mm ; $D = 650$ mm or $R = 325$ mm ; $W = 400$ kN $= 400 \times 10^3$ N ; $L = 250$ mm ; $\sigma_t = 60$ MPa $= 60$ N/mm^2

Let d_c = Core diameter of the bolts.

We know that when the bolts are equally spaced, the maximum load on the bolt,

$$W_t = \frac{2\,W.L}{n}\left[\frac{R + r\cos\left(\frac{180}{n}\right)}{2\,R^2 + r^2}\right]$$

$$= \frac{2 \times 400 \times 10^3 \times 250}{4}\left[\frac{325 + 250\cos\left(\frac{180}{4}\right)}{2\,(325)^2 + (250)^2}\right] = 91\,643 \text{ N}$$

We also know that maximum load on the bolt (W_t),

$$91\,643 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(d_c)^2\,60 = 47.13\,(d_c)^2$$

$\therefore \quad (d_c)^2 = 91\,643 / 47.13 = 1945 \quad$ or $\quad d_c = 44$ mm

From DDB, we find that the standard core diameter of the bolt is 45.795 mm and corresponding size of the bolt is M 52. **Ans.**

Eccentric Load Acting in the Plane Containing the Bolts

When the eccentric load acts in the plane containing the bolts, as shown in Fig.1, then the same procedure may be followed as discussed for eccentric loaded riveted joints.

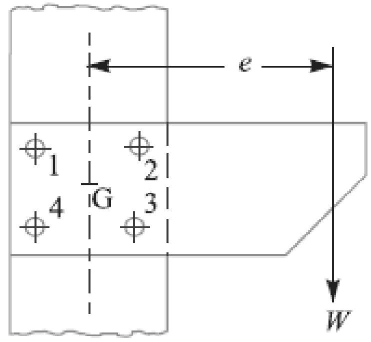
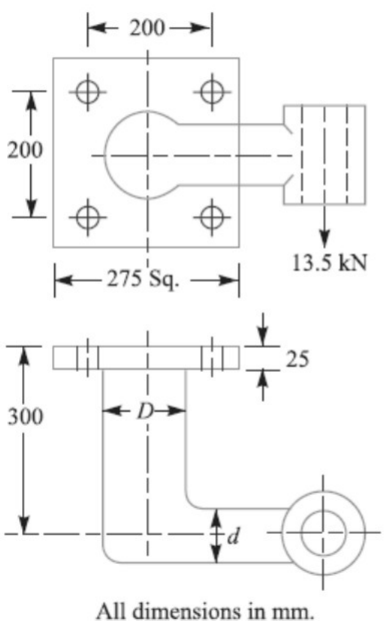


Fig. 1. Eccentric load in the plane containing the bolts.

Problem:

Fig.2 shows a solid forged brac ket to carry a vertical load of 13.5 kN applie d through the centre of hole. The square flang e is secured to the flat side of a vertical stancchion through four bolts. Calculate suitable diameter D and d for the arms of the bracket, if th e permissible stresses are 110 MPa in tenstion and 65 MPa in shear. Estimate also the tensile load on each top bolt and the maximum sheari ng force on each bolt.

Solution. Given : $W = 13.5$ kN = 13 500 N ; $\sigma_t = 110$ MPa = 110 N/mm^2 ; $\tau = 65$ MPa = 65 N/mm^2



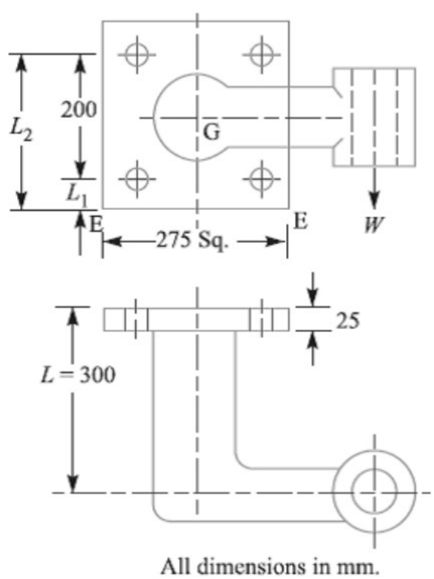


Fig.2

Fig.3

Diameter D for the arm of the bracket

The section of the arm having D as the diameter is subjected to bending moment as well as twisting moment. We know that bending moment,

$$M = 13\ 500 \times (300 - 25) = 3712.5 \times 10^3 \text{ N-mm}$$

and twisting moment, $T = 13\ 500 \times 250 = 3375 \times 10^3$ N-mm

∴ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3712.5 \times 10^3)^2 + (3375 \times 10^3)^2} \text{ N-mm}$$
$$= 5017 \times 10^3 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$5017 \times 10^3 = \frac{\pi}{16} \times \tau \times D^3 = \frac{\pi}{16} \times 65 \times D^3 = 12.76\, D^3$$

$$\therefore \quad D^3 = 5017 \times 10^3 / 12.76 = 393 \times 10^3$$

or $D = 73.24$ say 75 mm **Ans.**

Diameter (d) for the arm of the bracket

The section of the arm having d as the diameter is subjected to bending moment only. We know that bending moment,

$$M = 13\ 500 \left(250 - \frac{75}{2}\right) = 2868.8 \times 10^3 \text{ N-mm}$$

and section modulus, $Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3$

We know that bending (tensile) stress (σ_t),

$$110 = \frac{M}{Z} = \frac{2868.8 \times 10^3}{0.0982\, d^3} = \frac{29.2 \times 10^6}{d^3}$$

$$\therefore \quad d^3 = 29.2 \times 10^6 / 110 = 265.5 \times 10^3 \quad \text{or} \quad d = 64.3 \text{ say } 65 \text{ mm}$$ **Ans.**

Tensile load on each top bolt

Due to the eccentric load W, the bracket has a tendency to tilt about the edge $E–E$, as shown in Fig. 11.46.

Let w = Load on each bolt per mm distance from the tilting edge due to the tilting effect of the bracket.

Since there are two bolts each at distance L_1 and L_2 as shown in Fig. 11.46, therefore total moment of the load on the bolts about the tilting edge $E–E$

$$= 2\,(w.L_1)\,L_1 + 2(w.L_2)\,L_2 = 2w\,[(L_1)^2 + (L_2)^2]$$
$$= 2w\,[(37.5)^2 + (237.5)^2] = 115\ 625\, w \text{ N-mm} \qquad \text{...}(i)$$

...($\because L_1 = 37.5$ mm and $L_2 = 237.5$ mm)

and turning moment of the load about the tilting edge

$$= W.L = 13\ 500 \times 300 = 4050 \times 10^3 \text{ N-mm} \qquad \text{...}(ii)$$

From equations (*i*) and (*ii*), we have

$$w = 4050 \times 10^3 / 115\ 625 = 35.03 \text{ N/mm}$$

∴ Tensile load on each top bolt

$$= w.L_2 = 35.03 \times 237.5 = 8320 \text{ N}$$ **Ans.**

Maximum shearing force on each bolt

We know that primary shear load on each bolt acting vertically downwards,

$$W_{s1} = \frac{W}{n} = \frac{13\ 500}{4} = 3375 \text{ N} \qquad \text{...}(\because \text{ No. of bolts, } n = 4)$$

Since all the bolts are at equal distances from the centre of gravity of the four bolts (G), therefore the secondary shear load on each bolt is same.

Distance of each bolt from the centre of gravity (G) of the bolts,

$$l_1 = l_2 = l_3 = l_4 = \sqrt{(100)^2 + (100)^2} = 141.4 \text{ mm}$$

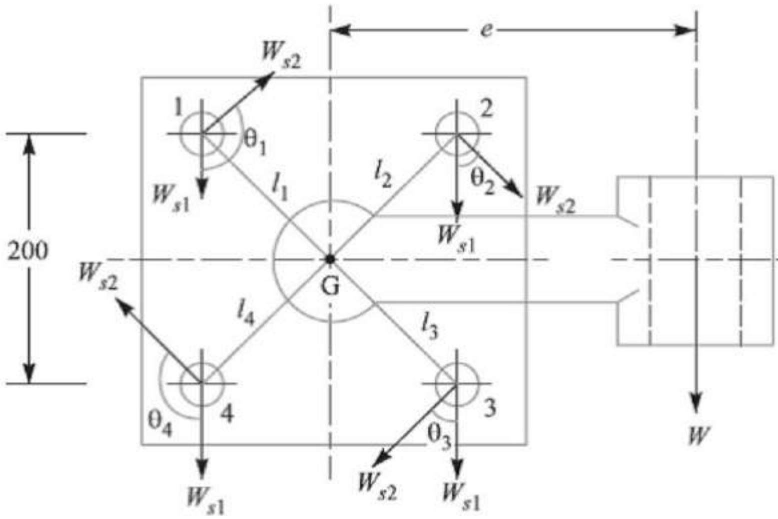


Fig.4

$\therefore$ Secondary shear load on each bolt,

$$W_{s2} = \frac{W.e.l_1}{(l_1)^2 + (l_2)^2 + (l_3)^3 + (l_4)^2} = \frac{13\ 500 \times 250 \times 141.4}{4\ (141\ 4)^2} = 5967 \text{ N}$$

Since the secondary shear load a cts at right angles to the line joining the centre of gravity of the bolt group to the centre of the bolt as shown in Fig. 4, therefore the resultant of the primary and secondary shear load on each bolt gives the maximum shearing force on each bolt. From the geometry of the Fig. 4, we find that

θ1 = θ4 = 135°, and θ2 = θ3 = 45°

Maximum shearing force on the bolts 1 and 4

$$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2\ W_{s1} \times W_{s2} \times \cos 135°}$$

$$= \sqrt{(3375)^2 + (5967)^2 - 2 \times 3375 \times 5967 \times 0.7071} = 4303 \text{ N } \textbf{Ans.}$$

And maximum shearing force on the bolts 2 and 3

$$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2\, W_{s1} \times W_{s2} \times \cos 45°}$$

$$= \sqrt{(3375)^2 + (5967)^2 + 2 \times 3375 \times 5967 \times 0.7071} = 8687 \text{ N } \textbf{Ans.}$$

References:

1. Machine Design - V.Bandari
2. Machine Design – R.S. Khurm i
3. Design Data hand Book - S MD Jalaludin.