UNIT-2

DESIGN OF BOLTED AND WELDED JOINTS

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

1. Screwed joints are highly reliable in operation.

2. Screwed joints are convenient to assemble and disassemble.

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

4. 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 weldedjoints.

Important Terms Used in Screw Threads

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

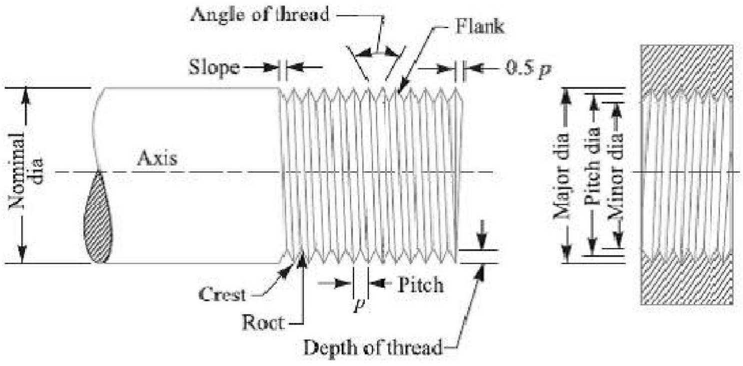


Fig.1 Terms used in screw threads

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

2. Minor diameter. It is the smallest diameter of an external or internal screw thread. It is also known as **core** or **root diameter.**

3. Pitch diameter. It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an **effective diameter.** In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.

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

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

5. Lead. It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.

6. Crest. It is the top surface of the thread.

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

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

9. Flank. It is the surface joining the crest and root.

10. Angle of thread. It is the angle included by the flanks of the thread.

11. Slope. It is half the pitch of the thread.

Stresses in Screwed Fastening due to Static Loading

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

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

Initial Stresses due to Screwing up Forces

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

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

$$Pi = 2840 \, d \text{ N}$$

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,

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

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

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

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

(b) 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,

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

Combined tension and shear stress. When the bolt is subjected to both tension and shearloads, 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 dueto 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**Stresses due to combined loading and Design of cylindrical cover plates:**

Stress due to Combined Forces

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

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

2. The external load, and

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

When the connected members are 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 the 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 resultant load will be either the initial tension or the external load, whichever is greater. The actual conditions usually lie between the two extremes. In order to determine the resultant axial load (*P*) on the bolt, the following equation may be used :

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

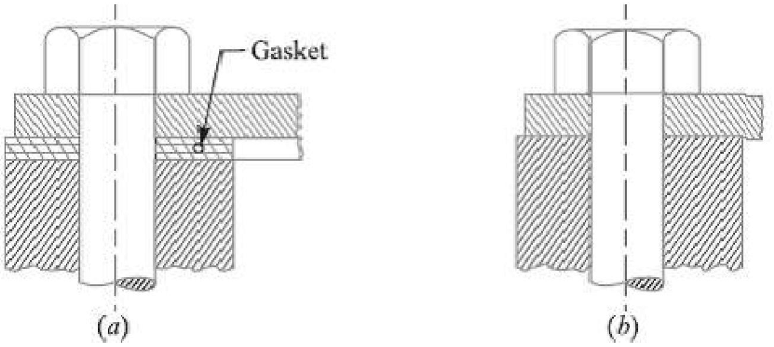


Fig.1 Where P_1 =

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 surfaces 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 obtain 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 the 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 secured by means of bolts or studs, but studs are preferred. 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:

1. Design of bolts or studs

In order to find the size and number of bolts or studs, the following procedure may beadopted.

Let D = Diameter of the cylinder,

p = Pressure in the cylinder,

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 acting on the cylinder cover,

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

This force is resisted by n number of bolts or studs provided on the cover.

Resisting force offered by n number of bolts or studs,

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

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

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

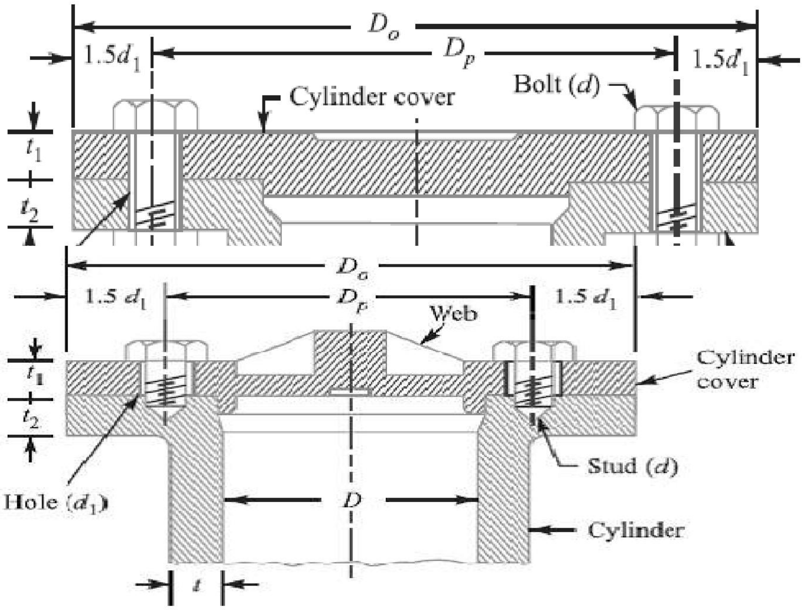


Fig. 2.

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 even number is adopted. The bolts or studs are screwed up tightly, along with metal gasket or asbestos packing, in order to provide a leak proof joint. We have already discussed that due to the tightening of bolts, sufficient tensile stress is produced in the bolts or studs. This 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 depends upon the circumferential pitch of the bolts or studs. The circumferential pitch should be between 20 d_1 and 30 d_1, where d_1 is the diameter of the hole in mm for bolt or stud. The 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 cylinder wall.

2. Design of cylinder cover plate

The thickness of the cylinder cover plate (t_1) and the thickness of the cylinder flange (t_2) may be determined as discussed below:

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 and the point Y is the centre of internal pressure.

We know that the bending moment at A-A,

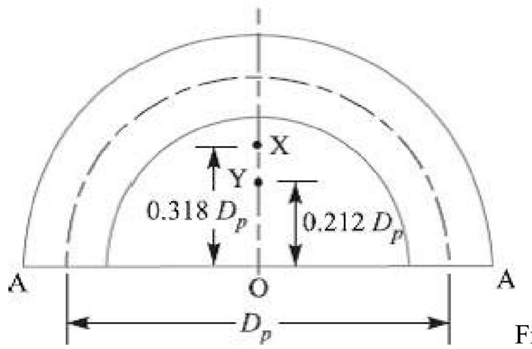


Fig.3

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

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

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

Where w = Width of plate

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

$= D_0 - 2d_1$

Knowing the tensile stress for the cover plate material, the value of t_1 may be determined by using the bending equation, *i.e.*, $\sigma t = M / Z$.

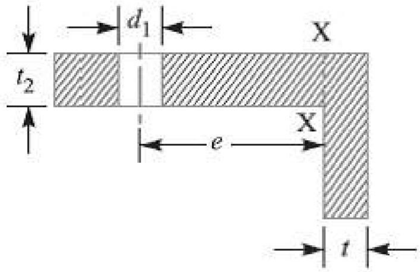


Fig.4

3. Design of cylinder flange

The thickness of the cylinder flange (t_2) may be determined from bending consideration. A portion of the cylinder flange under 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 figure, we find that eccentricity of the load from section *X-X* is

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

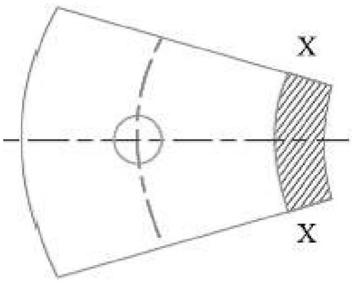


Fig.5

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

Bending moment, M = Load 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 byusing the bending equation *i.e.* $\sigma_t = M / Z$.

Problems on cylinder cover plates design

Problem:

A steam engine cylinder has an effective diameter of 350 mm and the maximum steam pressure acting on the cylinder cover is 1.25 N/mm². 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/mm² ; $\sigma_t = 33$ MPa $= 33$ N/mm²

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,

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

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,

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

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/mm². Design the cover plate along with the bolts. Assume allowable tensile stress for mild steel as 60 MPa and for bolt material as 40

MPa.

Solution. Given : $D = 120$ mm or $r = 60$ mm ; $p = 6$ N/mm^2 ; $\sigma_t = 60$ MPa $= 60$ N/mm^2 ; $\sigma_{tb} = 40$ 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,

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

We know that bending (tensile) stress,

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

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

Design of bolted joints under eccentric loading-1:

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

Consider a bracket having a rectangular 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

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

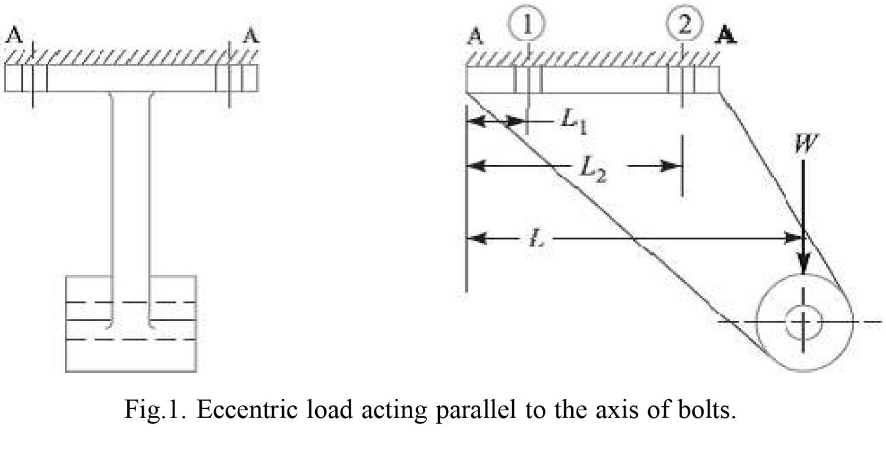


Fig.1. Eccentric load acting parallel to the axis of bolts.

Further the load W tends to rotate the bracket about the edge A-A. Due to this, each bolt is stretched by an amount that depends upon its distance from the tilting edge. Since the stress is a function of elongation, therefore each bolt will experience a different load which also depends upon the distance from the tilting edge. For convenience, all the bolts 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 from 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

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

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

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

... (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

$$= W.L \quad \text{...}(ii)$$

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

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

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,

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

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

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

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

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

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 stress in the bolt material is 60 MPa. The distances are: L_1 = 80 mm, L_2 = 250 mm, and L = 500 mm.

Solution. Given : W = 30 kN ; σ_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 mmand the corresponding size of the bolt is M 33. **Ans.**

Eccentric Load Acting Perpendicular to the Axis of Bolts

A wall bracket carrying an eccentric load perpendicular to the axis of the bolts is shown inFig.2.

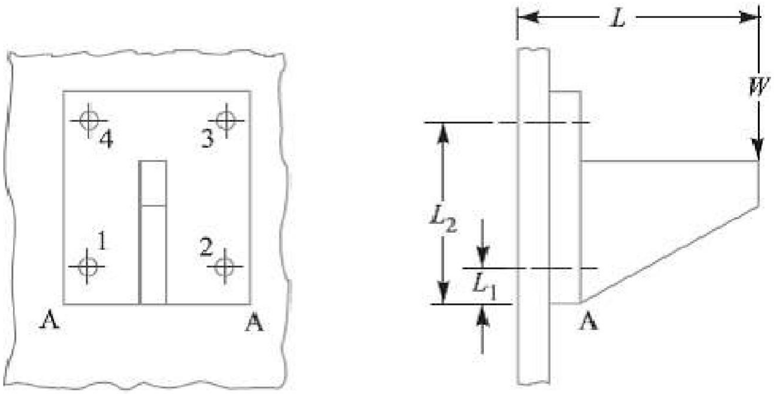


Fig. 2. Eccentric load perpendicular to the axis of bolts.

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 number of bolts.

A little consideration will show that the eccentric load W will try to tilt the bracket in the clockwise direction about the edge *A-A*. As discussed earlier, the bolts will be 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 equivalent loads may be determined by the following relations:

Equivalent tensile load,

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

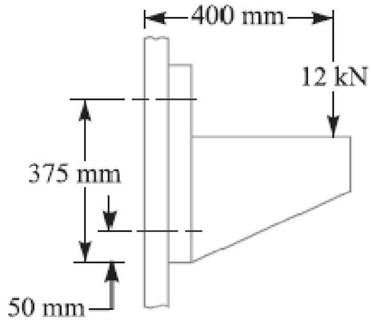
And equivalent shear load,

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

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 vertically 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 subjected to tensile load due to turning moment. The 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 equivalent 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 with Circular Base

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

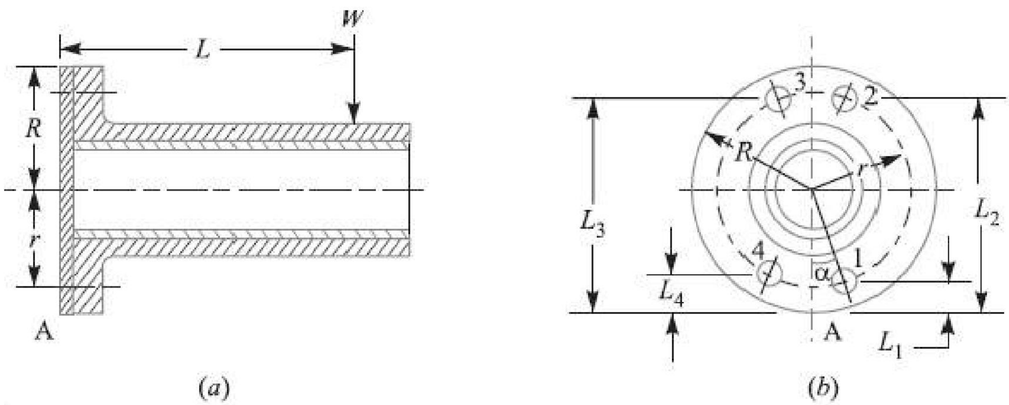


Fig.1. Ecce tric load on a bracket with circular base.

Let R = Radius of the column flange,

r = Radius of the bolt pitch circle,

w = Load per bolt per unit distance from the tilting edge,

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

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

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

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

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

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

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

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

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 relation to the bolts as in the case of pillar crane. But if the direction of load is fixed, then the maximum load on the bolts may be reduced by locating the bolts in such a way that two of them are equally stressed as shown in Fig.2. In such a case, maximum load is given by

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

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 bolts are designed for tensile load only.

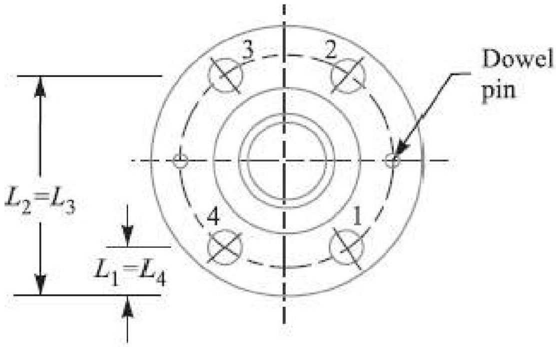


Fig.2.

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 bolts, taking safe tensile stress as 60 MPa for the material of the bolts.

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

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

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

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

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

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,

From DDB, we find that the standard core diameter of the bolt is 45.795 mm andcorresponding size of the bolt is M 52.

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 thesame procedure may be followed as discussed for eccentric loaded riveted joints.

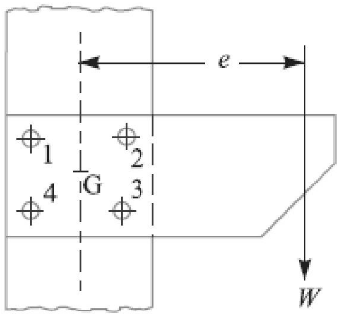


Fig. 1. Eccentric load in the plane containing the bolts.

Problem:

Fig.2 shows a solid forged bracket to carry a vertical load of 13.5 kN applied through the centre of hole. The square flange is secured to the flat side of a vertical stanchion through four bolts. Calculate suitable diameter D and d for the arms of the bracket, if the permissible stresses are 110 MPa in tenstion and 65 MPa in shear. Estimate also the tensile load on each top bolt and the maximum shearing 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

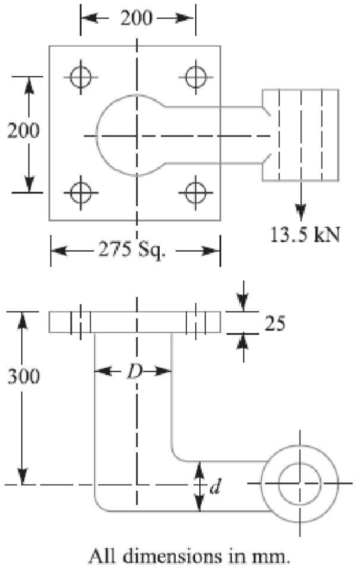


Fig.2

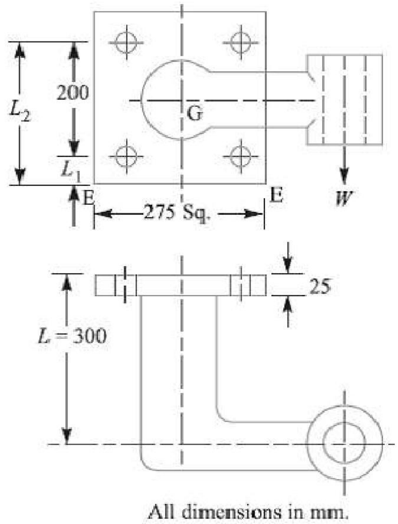


Fig.3

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

$\therefore$ 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),

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

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

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, $Z = \dfrac{\pi}{32} \times d^3 = 0.0982\, d^3$

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

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

$\therefore$ $d^3 = 29.2 \times 10^6 / 110 = 265.5 \times 10^3$ or $d = 64.3$ say 65 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 $E–E$

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

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

and turning moment of the load about the tilting edge

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

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

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

$\therefore$ Tensile load on each top bolt

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

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,

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

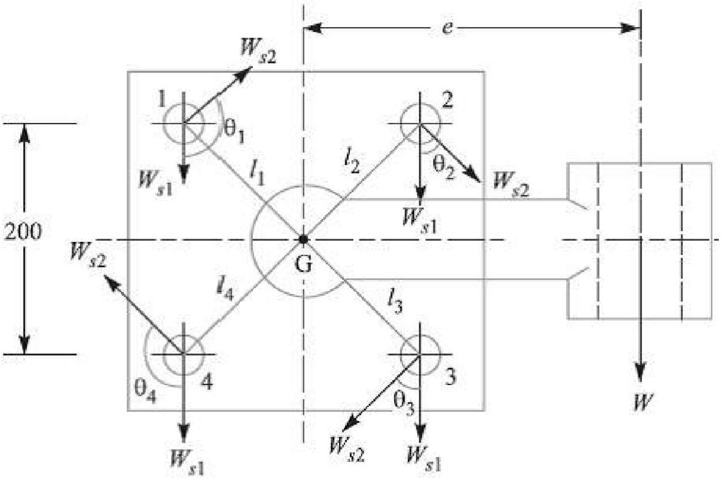


Fig.4

$\therefore$ 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 acts 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

$$\theta 1 = \theta 4 = 135°, \text{ and } \theta 2 = \theta 3 = 45°$$

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

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

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

Introduction to Welded Joints

A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without the application of pressure and a filler material. The heat required for the fusion of the material may be obtained by burning of gas (in case of gas welding) or by an electric arc (in case of electric arc welding). The latter method is extensively used because of greater speed of welding. Welding is extensively used in fabrication as an alternative method for casting or forging and as a replacement for bolted and riveted joints. It is also used as a repair medium *e.g.* to reunite metal at a crack, to build up a small part that has broken off such as gear tooth or to repair a worn surface such as a bearing surface.

Advantages and Disadvantages of Welded Joints over Riveted Joints

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

Advantages

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

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

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

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

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

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

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

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

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

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

Disadvantages

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

2. It requires a highly skilled labour and supervision.

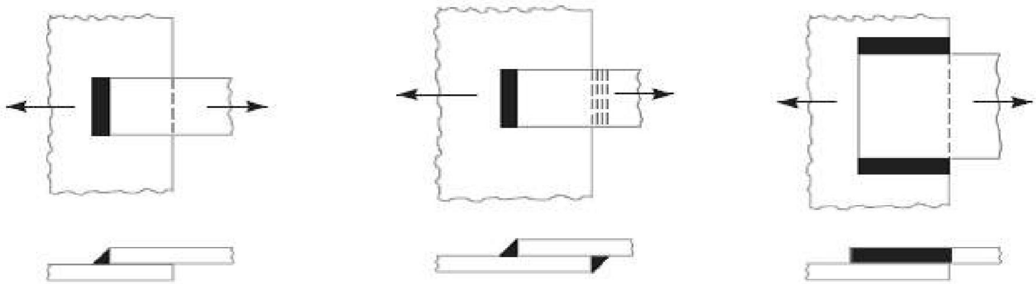
3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

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

Types of 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

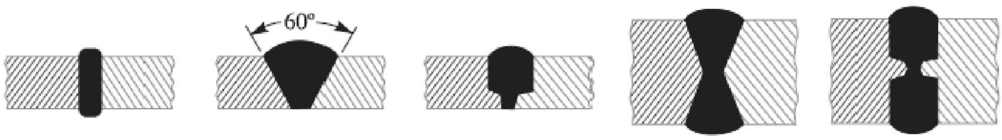
The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates. The cross-section of the fillet is approximately triangular. The fillet joints may be

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

The fillet joints are shown in Fig.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 placing the plates edge to edge as shown in Fig.2. In butt welds, the plate edges do not require beveling 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 beveled to V or U-groove on both sides.



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

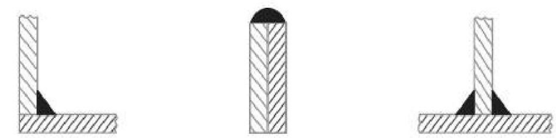
Fig. 2. Types of Butt joints

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.

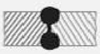
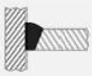
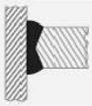
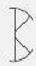
The other type of welded joints are corner joint, edge joint and T-joint as shown in Fig. 3.

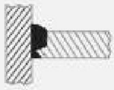
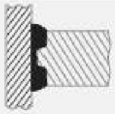
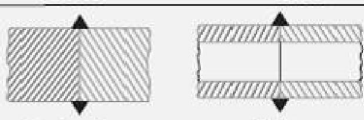
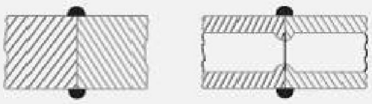


(*a*) Corner joint. (*b*) Edge joint. (*c*) *T*-joint.

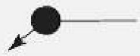
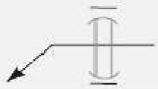
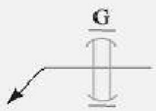
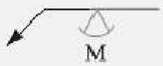
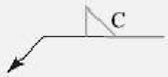
Fig. 3. Other types of Joints

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

<u>**References:**</u>

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

Elements of a Welding Symbol

A welding symbol consists of the following eight elements:

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

3. Basic weld symbols, **4.** Dimensions and other data,

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

7. Tail, and **8.** Specification, process or other references.

Standard Location of Elements of a Welding Symbol

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

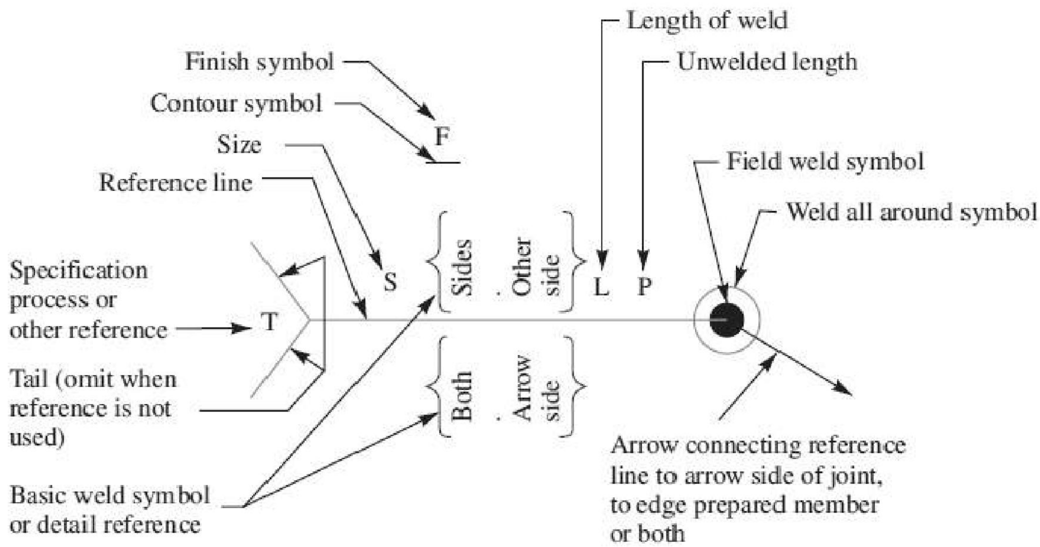
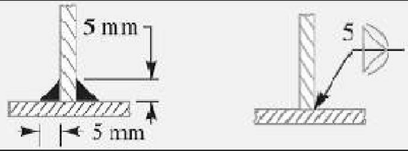
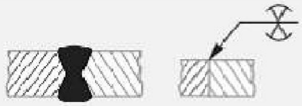
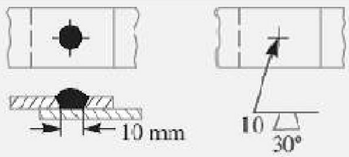
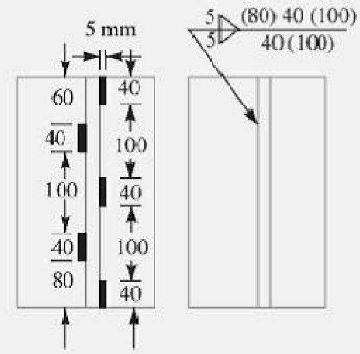


Fig.1 Standard location of weld symbols.

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

Representation of welding symbols.

S. No.	Desired weld	Representation 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	

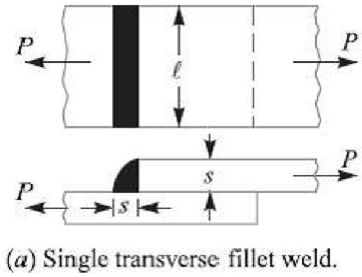
References:

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

Design of Welded Joints:

Strength of Transverse Fillet Welded Joints

We have already discussed that the fillet or lap joint is obtained by overlapping the plates and then welding the edges of the plates. The transverse fillet welds are designed for tensile strength. Let us consider a singl and double transverse fillet welds as shown in Fig. 1(*a*) and (*b*) respectively.



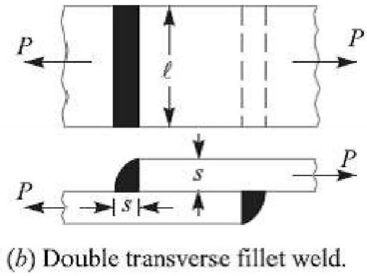


Fig.1 Transverse fillet welds.

The length of each side is known as ***leg*** or ***size of the weld*** and the perpendicular distance of the hypotenuse from the intersection of legs (*i.e. BD*) is known as ***throat thickness.*** The minimum area of the weld is obtained at the throat *BD*, which is given by the product of the throat thickness and length of weld.

Let t = Throat thickness (BD),

s = 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 stress for the weld metal, then the tensile strength of the joint for single fillet weld,

$$P = \text{Throat area} \times \text{Allowable tensile stress} = 0.707\, s \times l \times \sigma_t$$

And tensile strength of the joint for double fillet weld,

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

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

Strength of Parallel Fillet Welded Joints

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

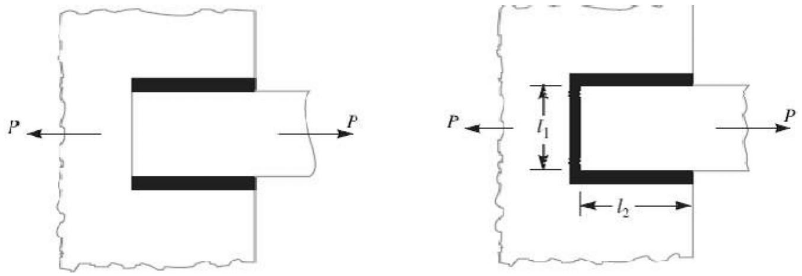
$$A = 0.707\, s \times l$$

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

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

And shear strength of the joint for double parallel fillet weld,

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



(*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 strengths of single transverse and double parallel fillet welds. Mathematically,

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

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 length 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),

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

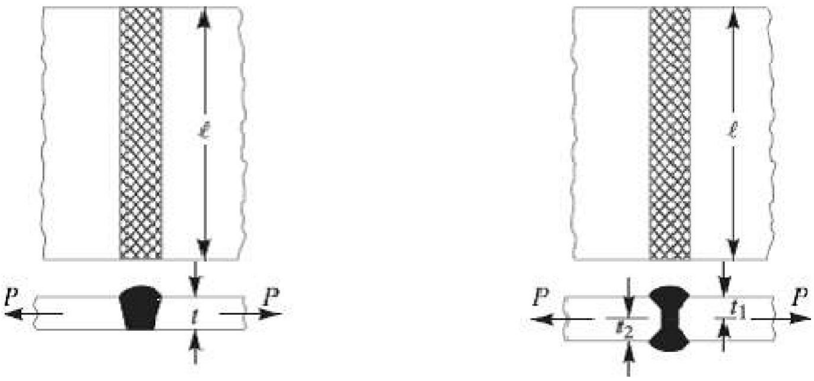
$$\therefore \quad l = 80 \times 10^3 / 778 = 103 \text{ mm}$$

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 of 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 Fig. 4(*b*) is given by

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

Where t_1 = Throat thickness at the top, and
t_2 = Throat thickness at t 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.

References:

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

Problem:

A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of parallel fillet welds. The plates are subjected to a load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 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 $= 50 \times 10^3$ N ; $\tau = 56$ 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),

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

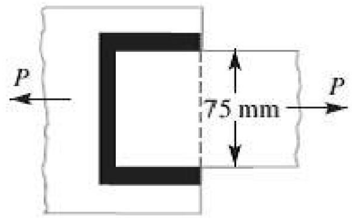
$$\therefore \quad l = 50 \times 10^3 / 367 = 136.2 \text{ mm}$$

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 transverse 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

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

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

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

References:

1. Machine Design - V.Bandari .

2. Machine Design – R.S. Khurmi

3. Design Data hand Book - S MD Jalaludin.

Special fillet welded joints:

Special Cases of Fillet Welded Joints

The following cases of fillet welded joints are important from the subject point of 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 $= \dfrac{\pi t d^3}{4}$

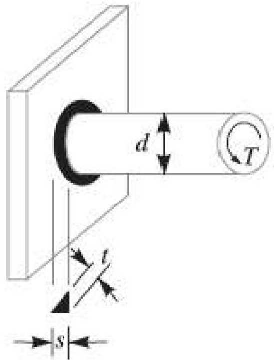


Fig. 1. Circular fillet weld subjected to torsion.

We know that shear stress for the 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. The 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 connected 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,

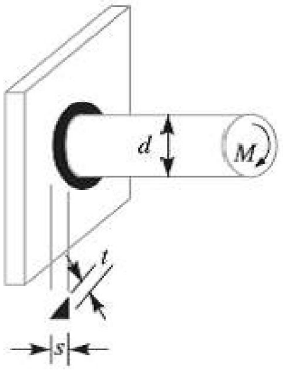


Fig.2.Circular fillet weld subjected to Bending moment.

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,

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

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

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

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,

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

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

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

References:

1. Machine Design - V.Bandari .

2. Machine Design – R.S. Khurmi

3. Design Data hand Book - S MD Jalaludin.

Unsymmetrical welded joints:

Axially Loaded Unsymmetrical WeldedSections

Sometimes unsymmetrical sections such as angles, channels, *T*-sections etc., welded on the flange edges are loaded axially as 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 section as shown in Fig.

Let l_a = Length of weld at the top,

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 from gravity axis, and

f = Resistance offered by the weld per unit length.

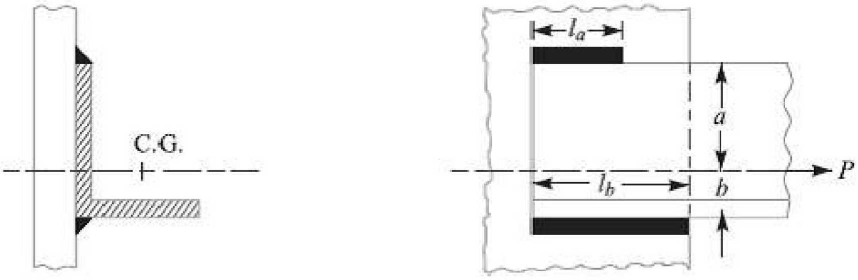


Fig. Axially loaded unsymmetrical welded section

Moment of the top weld about gravity 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,

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

$$\text{or } l_a \times a = l_b \times b \qquad \ldots(i)$$

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

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

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

<u>Eccentrically Loaded Welded Joints</u>

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

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.

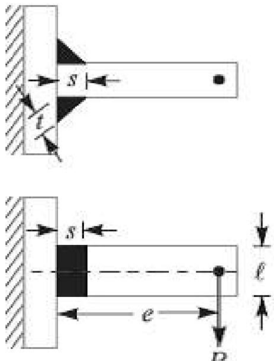


Fig.1. Eccentrically loaded welded joint

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 shear force *P* acting at the welds, and

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

We know that area at the throat,

A = Throat thickness × Length of weld

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

$= 2 \times 0.707\,s \times l = 1.414\,s \times l$ ... (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 metal through the throat,

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

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

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 eccentrically 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.

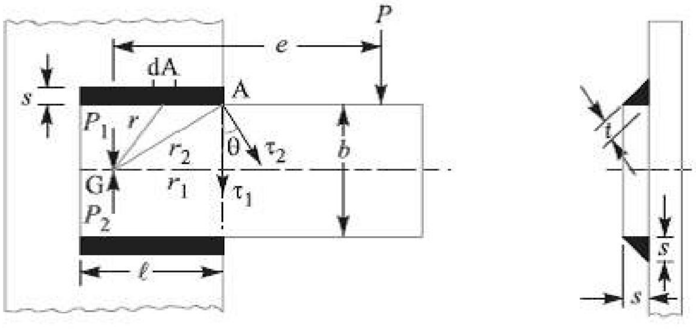


Fig.2 eccentrically loaded welded joint.

Let P = Eccentric load,

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

l = Length of single weld,

s = Size or leg of weld, and

t = Throat thickness.

Let two loads P_1 and P_2 (each equal 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 weld length. The effect of load P_2 = P is to produce a turning moment of magnitude P × e which tends of rotate the joint about the centre of gravity 'G' of the weld system. Due to the turning moment, secondary shear stress is induced.

We know that the direct or primary shear stress,

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

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

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

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

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

Where τ_2 is the shear stress at the maximum distance (r_2) and τ is the shear stress at any distance r. Consider a small section of the weld having area dA at a distance r from G.

Shear force on this small section

$$= \tau \times dA$$

And turning moment of this shear force about G,

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

Total turning moment over the whole weld area,

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

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

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

□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,

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

$$\theta = \text{Angle between } \tau_1 \text{ and } \tau_2 \text{, and}$$

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

References:

1. Machine Design - V.Bandari .

2. Machine Design – R.S. Khurmi

3. Design Data hand Book - S MD Jalaludin

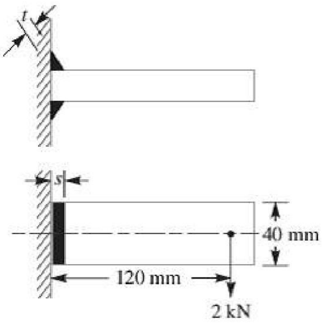
Problem:

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

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

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

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

Section modulus of the weld through the throat,

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

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

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

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

$$\therefore \quad s = 320.3 / 25 = 12.8 \text{ mm} \quad \textbf{Ans.}$$

Problem:

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

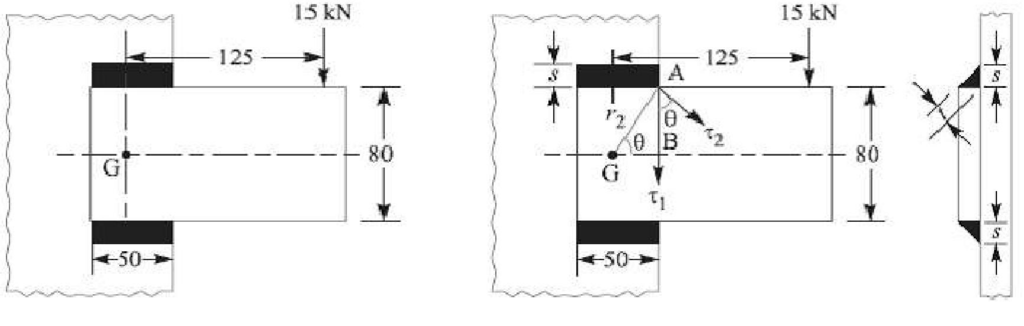
$$A = 2 \times t \times l = 2 \times 0.707\, s \times l = 1.414\, s \times l = 1.414 \times s \times 50 = 70.7\, s \text{ mm}^2$$

∴ 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,

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

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

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

References:

1. Machine Design - V.Bandari

2. Machine Design – R.S. Khurmi

3. Design Data hand Book - S MD Jalaludin.