Pipes and Pipe Joints

Introduction

The pipes are used for transporting various fluids like water, steam, different types of gases, oil and other chemicals with or without pressure from one place to another. Cast iron, wrought iron, steel and brass are the materials generally used for pipes in engineering practice. The use of cast iron pipes is limited to pressures of about 0.7 N/mm 2 because of its low resistance to shocks which may be created due to the action of water hammer. These pipes are best suited for water and sewage systems. The wrought iron and steel pipes are used chiefly for conveying steam, air and oil. Brass pipes, in small sizes, finds use in pressure lubrication systems on prime movers. These are made up and threaded to the same standards as wrought iron and steel pipes. Brass pipe is not liable to corrosion. The pipes used in petroleum industry are generally seamless pipes made of heat-resistant chrome- molybdenum alloy steel. Such type of pipes can resist pressures more than 4 N/mm 2 and temperatures greater than 440°C.

Stresses in Pipes

The stresses in pipes due to the internal fluid pressure are determined by Lame's equation as discussed in the previous chapter (Art. 7.9). According to Lame's equation, tangential stress at any radius x,

$$\sigma_t = \frac{p\,(r_i)^2}{(r_o)^2 - (r_i)^2}\left[1 + \frac{(r_o)^2}{x^2}\right] \quad ...(i)$$

and radial stress at any radius x,

$$\sigma_r = \frac{p\,(r_i)^2}{(r_o)^2 - (r_i)^2}\left[1 - \frac{(r_o)^2}{x^2}\right] \quad ...(ii)$$

where p = Internal fluid pressure in the pipe,
r_i = Inner radius of the pipe, and
r_o = Outer radius of the pipe.

The tangential stress is maximum at the inner surface (when $x = r_i$) of the pipe and minimum at the outer surface (when $x = r_o$) of the pipe. Substituting the values of $x = r_i$ and $x = r_o$ in equation (i), we find that the maximum tangential stress at the inner surface of the pipe,

$$\sigma_{t(max)} = \frac{p\,[(r_o)^2 + (r_i)^2]}{(r_o)^2 - (r_i)^2}$$

and minimum tangential stress at the outer surface of the pipe,

$$\sigma_{t(min)} = \frac{2\,p\,(r_i)^2}{(r_o)^2 - (r_i)^2}$$

The radial stress is maximum at the inner surface of the pipe and zero at the outer surface of the pipe. Substituting the values of $x = r_i$ and $x = r_o$ in equation (ii), we find that maximum radial stress at the inner surface,

$$\sigma_{r(max)} = -p \text{ (compressive)}$$

and minimum radial stress at the outer surface of the pipe,

$$\sigma_{r(min)} = 0$$

The thick cylindrical formula may be applied when

(a) the variation of stress across the thickness of the pipe is taken into account,

(b) the internal diameter of the pipe (D) is less than twenty times its wall thickness (t), i .e. D/t < 20, and

(c) the allowable stress (σ_t) is less than six times the pressure inside the pipe (p) i.e .

$\sigma_t / p < 6$.

According to thick cylindrical formula (Lame's equation), wall thickness of pipe,

$$t = R\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right]$$

where R = Internal radius of the pipe.

The following table shows the values of allowable tensile stress (σ_t) to be used in the above relations:

Table 8.1. Values of allowable tensile stress for pipes of different materials.

S.No.	*Pipes*	*Allowable tensile stress (σ_t) in MPa or N/mm²*
1.	Cast iron steam or water pipes	14
2.	Cast iron steam engine cylinders	12.5
3.	Lap welded wrought iron tubes	60
4.	Solid drawn steel tubes	140
5.	Copper steam pipes	25
6.	Lead pipes	1.6

Design of Pipes

The design of a pipe involves the determination of inside diameter of the pipe and its wall thickness as discussed below:

1. Inside diameter of the pipe. The inside diameter of the pipe depends upon the quantity of fluid to be delivered.

Let D = Inside diameter of the pipe,

v = Velocity of fluid flowing per minute, and

Q = Quantity of fluid carried per minute.

We know that the quantity of fluid flowing per minute,

$$Q = \text{Area} \times \text{Velocity} = \frac{\pi}{4} \times D^2 \times v$$

$$\therefore \quad D = \sqrt{\frac{4}{\pi} \times \frac{Q}{v}} = 1.13\sqrt{\frac{Q}{v}}$$

2. Wall thickness of the pipe. After deciding upon the inside diameter of the pipe, the thickness of the wall (t) in order to withstand the internal fluid pressure (p) may be obtained by using thin cylindrical or thick cylindrical formula. The thin cylindrical formula may be applied when

(a) the stress across the section of the pipe is uniform,

(b) the internal diameter of the pipe (D) is more than twenty times its wall thickness (t), i.e. D/t > 20, and

(c) the allowable stress (σ_t) is more than six times the pressure inside t he pipe (p) ,

i .e . σ_t / p > 6.

According to thin cylindrical formula, wall thickness of pipe,

$$t = \frac{p.D}{2\sigma_t} \quad \text{or} \quad \frac{p.D}{2\sigma_t \eta_l}$$

where η_l = Efficiency of longitudinal joint.

A little consideration will show that the thickness of wall as obtained by the above relation is too small. Therefore for the design of pipes, a certain constant is added to the above relation. Now the relation may be written as

$$t = \frac{p.D}{2\sigma_t} + C$$

The value of constant '*C*', according to Weisback, are given in the following table.

Table 8.2. Values of constant '*C*'.

Material	*Cast iron*	*Mild steel*	*Zinc and Copper*	*Lead*
Constant (*C*) in mm	9	3	4	5

Pipe Joints

The pipes are usually connected to vessels from which they transport the fluid. Since the length of pipes available are limited, therefore various lengths of pipes have to be joined to suit any particular installation. There are various forms of pipe joints used in practice, but most common of them are discussed below.

1. Socket or a coupler joint . The most common method of joining pipes is by means of a socket or a coupler as shown in Fig. 8.2. A socket is a small piece of pipe threaded inside. It is screwed on half way on the threaded end of one pipe and the other pipe is then screwed in the remaining half of socket. In order to prevent leakage, jute or hemp is wound around the threads at the end of each pipe. This type of joint is mostly used for pipes carrying water at low pressure and where the overall smallness of size is most essential.

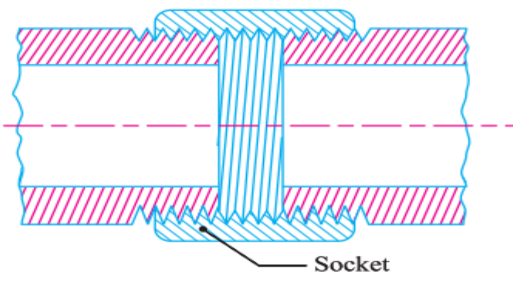


Fig. 8.2. Socket or coupler joint.

2. Nipple joint. In this type of joint, a nipple which is a small piece of pipe threaded outside is screwed in the internally threaded end of each pipe, as shown in Fig. 8.3. The disadvantage of this joint is that it reduces the area of flow.

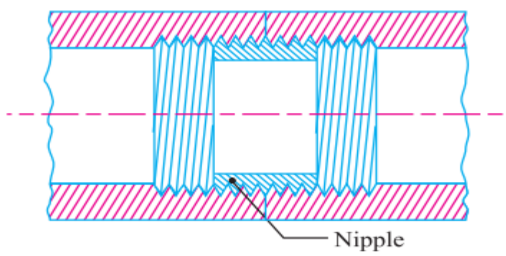


Fig. 8.3. Nipple joint.

3. Union joint. In order to disengage pipes joined by a socket, it is necessary to unscrew pipe from one end. This is sometimes inconvenient when pipes are long. The union joint, as shown in Fig. 8.4, provide the facility of disengaging the pipes by simply unscrewing a coupler nut.

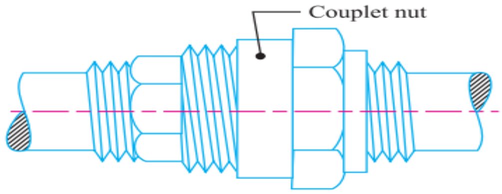


Fig. 8.4. Union joint.

4. Spigot and socket joint. A spigot and socket joint as shown in Fig. 8.5, is chiefly used for pipes which are buried in the earth. Some pipe lines are laid straight as far as possible. One of the important features of this joint is its flexibility as it adopts itself to small changes in level due to settlement of earth which takes place due to climate and other conditions. In this type of joint, the spigot end of one pipe fits into the socket end of the other pipe. The remaining space between the two is filled with a jute rope and a ring of lead. When the lead solidifies, it is caulked-in tightly.

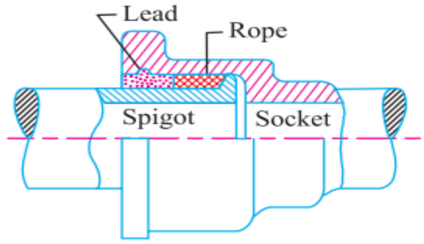


Fig. 8.5. Spigot and socket joint.

5. Expansion joint. The pipes carrying steam at high pressures are usually joined by means of expansion joint. This joint is used in steam pipes to take up expansion and contraction of pipe line due to change of temperature. In order to allow for change in length, steam pipes are not rigidly clamped but supported on rollers. The rollers may be arranged on wall bracket, hangers or floor stands. The expansion bends, as shown in Fig. 8.6 (a) and (b), are useful in a long pipe line. These pipe bends will spring in either direction and readily accommodate themselves to small movements of the actual pipe ends to which they are attached.

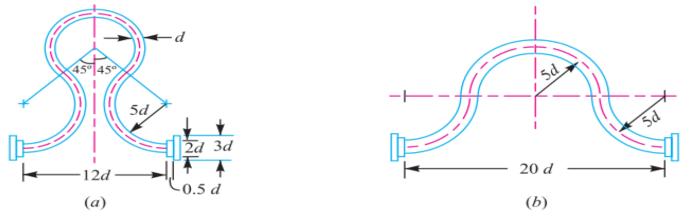


Fig. 8.6. Expansion bends.

6. Flanged joint. It is one of the most widely used pipe joint. A flanged joint may be made with flanges cast integral with the pipes or loose flanges welded or screwed. Fig. 8.8 shows two cast iron pipes with integral flanges at their ends. The flanges are connected by means of bolts. The flanges have seen standardized for pressures upto 2 N/mm 2. The flange faces are machined to ensure correct alignment of the pipes. The joint may be made leak proof by placing a gasket of soft material, rubber or canvass between the flanges. The flanges are made thicker than the pipe walls, for strength. The pipes may be strengthened for high pressure duty by increasing the thickness of pipe for a short length from the flange, as shown in Fig. 8.9.

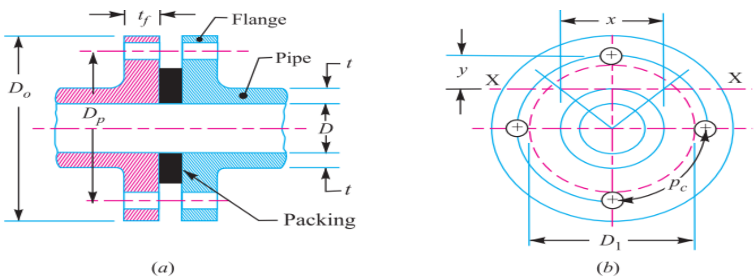


Fig. 8.8. Flanged joint.

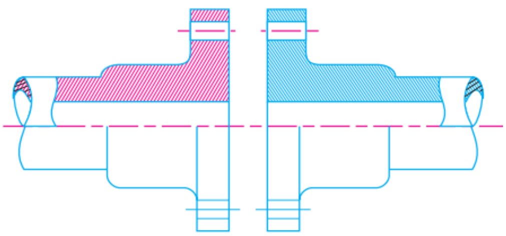
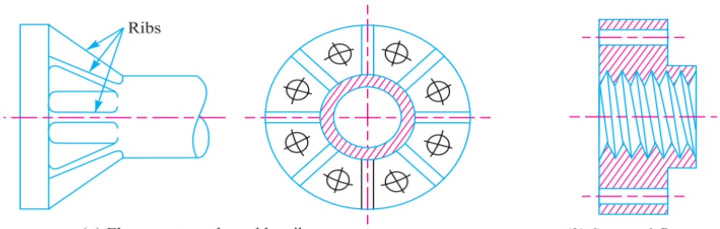


Fig. 8.9. Flanged joint.

For even high pressure and for large diameters, the flanges are further strengthened by ribs or stiffness as shown in Fig. 8.10 (a). The ribs are placed between the bolt holes. For larger size pipes, separate loose flanges screwed on the pipes as shown in Fig. 8.10 (b) are used instead of integral flanges.



(*a*) Flanges strengthened by ribs. (*b*) Screwed flange.

Fig. 8.10

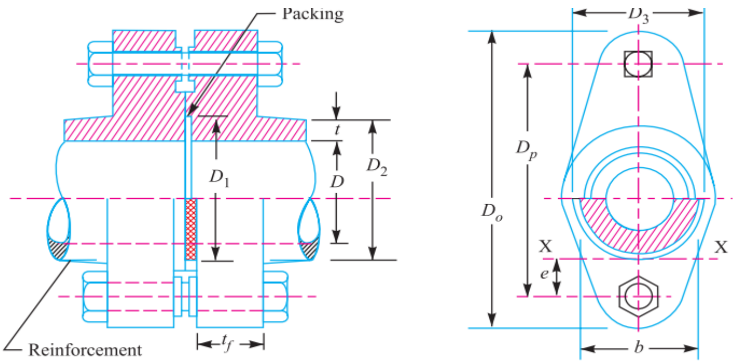


Fig. 8.11. Hydraulic pipe joint.

7. Hydraulic pipe joint. **This type of joint has oval flanges and are fastened by means of two bolts, as shown in Fig. 8.11. The oval flanges are usually used for small pipes, upto 175 mm diameter. The flanges are generally cast integral with the pipe ends. Such joints are used** to carry fluid **pressure varying from 5 to 14 N/mm2. Such a high pressure is found in hydraulic applications like riveting, pressing, lifts etc. The hydraulic machines used in these installations are pumps, accumulators, intensifiers etc.**

Standard Pipe Flanges for Steam

The Indian boiler regulations (I.B.R.) 1950 (revised 1961) have standardised all dimensions of pipe and flanges based upon steam pressure. They have been divided into five classes as follows:

Class I : For steam pressures up to 0.35 N/mm 2 and water pressures up to 1.4 N/mm 2. This is not suitable for feed pipes and shocks.

Class II : For steam pressures over 0.35 N/mm2 but not exceeding 0.7 N/mm2.

Class III : For steam pressures over 0.7 N/mm 2 but not exceeding 1.05 N/mm2.

Class IV : For steam pressures over 1.05 N/mm2 but not exceeding 1.75 N/mm 2.

Class V : For steam pressures from 1.75 N/mm 2 to 2 .45 N/mm 2.

Hydraulic Pipe Joint for High Pressures : The pipes and pipe joints for high fluid pressure are classified as follows:

1. **For hydraulic pressures up to 8.4 N/mm 2 and pipe bore from 50 mm to 175 mm, the flanges and pipes are cast integrally from remelted cast iron. The flanges are made elliptical and secured by two bolts. The proportions of these pipe joints have been standardized from 50 mm to 175 mm, the bore increasing by 2 5 mm. This category is further split up into two classes:**

(a) Class A : For fluid pressures from 5 to 6.3 N/mm ^{2}, and

(b) Class B : For fluid pressures from 6.3 to 8.4 N/mm ^{2} .

The flanges in each of the above classes may be of two types. Type I is suitable for pipes of 50 to 100 mm bore in class A , and for 50 to 175 mm bore in class B . The flanges of type II are stronger than those of Type I and are usually set well back on the pipe. 2. For pressures above 8.4 N/mm2 with bores of 50 mm or below, the piping is of wrought steel, solid drawn, seamless or rolled. The flanges may be of cast iron, steel mixture or forged steel. These are screwed or welded on to the pipe and are square in elevation secured by four bolts. These joints are made for pipe bores 12 .5 mm to 50 mm rising in increment of 3 mm from 12 .5 to 17.5 mm and by 6 mm from 17.5 to 50 mm. The flanges and pipes in this category are strong enough for service under pressures ranging up to 47.5 N/mm ^{2}.In all the above classes, the joint is of the spigot and socket type made with a jointing ring of gutta-percha.

Notes: The hydraulic pipe joints for high pressures differ from those used for low or medium pressure in the following ways:

1. The flanges used for high pressure hydraulic pipe joints are heavy oval or square in form, They use two or four bolts which is a great advantage while assembling and disassembling the joint especially in narrow space.

2. The bolt holes are made square with sufficient clearance to accomodate square bolt heads and to allow for small movements due to setting of the joint.

3. The surfaces forming the joint make contact only through a gutta-percha ring on the small area provided by the spigot and recess. The tightening up of the bolts squeezes the ring into a triangular shape and makes a perfectly tight joint capable of withstanding pressure up to 47.5 N/mm ^{2} .

4. In case of oval and square flanged pipe joints, the condition of bending is very clearly defined due to the flanges being set back on the pipe and thickness of the flange may be accurately determined to withstand the bending action due to tightening of bolts.

Design of Circular Flanged Pipe Joint

Consider a circular flanged pipe joint as shown in Fig. 8.8. In designing such joints, it is assumed that the fluid pressure acts in between the flanges and tends to separate them with a pressure existing at the point of leaking. The bolts are required to take up tensile stress in order to keep the flanges together. The effective diameter on which the fluid pressure acts, just at the point of leaking, is the diameter of a circle touching the bolt holes. Let this diameter be D_1. If d_1 is the diameter of bolt hole and Dp is the pitch circle diameter, then

$$D_1 = D_p - d_1$$

∴ Force trying to separate the two flanges,

$$F = \frac{\pi}{4}(D_1)^2\ p \qquad ...(i)$$

Let n = Number of bolts,

d_c = Core diameter of the bolts, and

σ_t = Permissible stress for the material of the bolts.

∴ Resistance to tearing of bolts

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

Assuming the value of dc , the value of n may be obtained from equations (i) and (ii). The number of bolts should be even because of the symmetry of the section. The circumferential pitch of the bolts is given by

$$p_c = \frac{\pi D_p}{n}$$

In order to make the joint leakproof, the value of p_c should be between 20 $\sqrt{d_1}$ to 30 $\sqrt{d_1}$, where d_1 is the diameter of the bolt hole. Also a bolt of less than 16 mm diameter should never be used to make the joint leakproof.

The thickness of the flange is obtained by considering a segment of the flange as shown in Fig. 8.8 (b). In this it is assumed that each of the bolt supports one segment. The effect of joining of these segments on the stresses induced is neglected. The bending moment is taken about the section X-X, which is tangential to the outside of the pipe. Let the width of this segment is x and the distance of this section from the centre of the bolt is y .

∴ Bending moment on each bolt due to the force F

$$= \frac{F}{n} \times y \qquad \text{...(iii)}$$

and resisting moment on the flange

$$= \sigma_b \times Z \qquad \text{...(iv)}$$

where σ_b = Bending or tensile stress for the flange material, and

Z = Section modulus of the cross-section of the flange $= \frac{1}{6} \times x \, (t_f)^2$

Equating equations (iii) and (iv), the value of t_f may be obtained.

The dimensions of the flange may be fixed as follows:

Nominal diameter of bolts, **d = 0.75 t + 1 0 mm**

Number of bolts, **n = 0.02 75 D + 1 .6** **...(D is in mm)**

Thickness of flange, **t_f = 1.5 t + 3 mm**

Width of flange, **B = 2.3 d**

Outside diameter of flange, **Do = D + 2 t + 2B**

Pitch circle diameter of bolts, **Dp = D + 2 t + 2 d + 12 mm**

The pipes may be strengthened by providing greater thickness near the flanges $\left(\text{equal to } \frac{t + t_f}{2}\right)$ as shown in Fig. 8.9. The flanges may be strengthened by providing ribs equal to thickness of $\frac{t + t_f}{2}$, as shown in Fig. 8.10 (*a*).

Example . A flanged pipe with internal diameter as 200 mm is subjected to a fluid pressure of 0.35 N/mm 2. The elevation of the flange is shown in Fig. 8.12. The flange is connected by means of eight M 16 bolts. The pitch circle diameter of the bolts is 290 mm. If the thickness of the flange is 20 mm, find the working stress in the flange.

Solution.

Given : D = 200 mm ; p = 0.35 N/mm 2 ; n = 8 ; * d = 16 mm ; D_p = 290 mm ; t_f = 20 mm

First of all, let us find the thickness of the pipe. Assuming the pipe to be of cast iron, we find from Table 8.1 that the allowable tensile stress for cast iron, σ_t = 14 N/mm2 and from Table 8.2, C = 9 mm.

∴ Thickness of the pipe,

$$t = \frac{p.D}{2\,\sigma_t} + C = \frac{0.35 \times 200}{2 \times 14} + 9 = 11.5 \text{ say } 12\,\text{mm}$$

Since the diameter of the bolt holes (d_1) is taken larger than the nominal diameter of the bolts (d), therefore let us take diameter of the bolt holes,

d_1= d + 2 mm = 1 6 + 2 = 1 8 mm

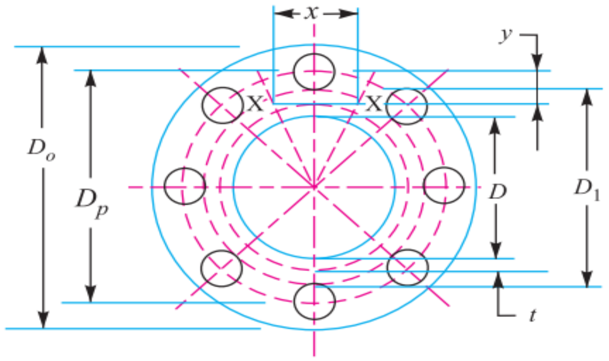


Fig. 8.12

and diameter of the circle on the inside of the bolt holes,

$$D_1 = D_p - d_1 = 290 - 18 = 272 \text{ mm}$$

∴ Force trying to separate the flanges *i.e.* force on 8 bolts,

$$F = \frac{\pi}{4}(D_1)^2\ p = \frac{\pi}{4}(272)^2\ 0.35 = 20\ 340 \text{ N}$$

Now let us find the bending moment about the section X-X which is tangential to the outside of the pipe. The width of the segment is obtained by measuring the distance from the drawing. On measuring, we get

x = 90 mm

and distance of the section X-X from the centre of the bolt,

$$y = \frac{D_p}{2} - \left(\frac{D}{2} + t\right) = \frac{290}{2} - \left(\frac{200}{2} + 12\right) = 33 \text{ mm}$$

Let σ_b = Working stress in the flange.

We know that bending moment on each bolt due to force F

$$= \frac{F}{n} \times y = \frac{20340}{8} \times 33 = 83\ 900 \text{ N-mm} \quad \ldots(i)$$

and resisting moment on the flange

$$= \sigma_b \times Z = \sigma_b \times \frac{1}{6} \times x\,(t_f)^2$$

$$= \sigma_b \times \frac{1}{6} \times 90\,(20)^2 = 6000\,\sigma_b \text{ N-mm} \quad \ldots(ii)$$

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

$$\sigma_b = 83\ 900 / 6000$$

$= 13.98\ N/mm^2 = 13.98$ MPa **Ans.**

Design of Oval Flanged Pipe Joint

Consider an oval flanged pipe joint as shown in Fig. 8.11. A spigot and socket is provided for locating the pipe bore in a straight line. A packing of trapezoidal section is used to make the joint leak proof. The thickness of the pipe is obtained as discussed previously. The force trying to separate the two flanges has to be resisted by the stress produced in the bolts. If a length of pipe, having its ends closed somewhere along its length, be considered, then the force separating the two flanges due to fluid pressure is given by

$$F_1 = \frac{\pi}{4} \times D^2 \times p$$

where D = Internal diameter of the pipe.

The packing has also to be compressed to make the joint leak proof. The intensity of pressure should be greater than the pressure of the fluid inside the pipe. For the purposes of calculations, it is assumed that the packing material is compressed to the same pressure as that of inside the pipe. Therefore the force tending to separate the flanges due to pressure in the packing is given by

$$F_2 = \frac{\pi}{4} \times \left[(D_1)^2 - (D)^2 \right] p$$

where D_1 = Outside diameter of the packing.

∴ Total force trying to separate the two flanges,

$$F = F_1 + F_2$$

$$= \frac{\pi}{4} \times D^2 \times p + \frac{\pi}{4} \left[(D_1)^2 - (D)^2 \right] p = \frac{\pi}{4} (D_1)^2 p$$

Since an oval flange is fastened by means of two bolts, therefore load taken up by each bolt is $F_b = F/2$. If d_c is the core diameter of the bolts, then

$$F_b = \frac{\pi}{4} (d_c)^2 \sigma_{tb}$$

where σ_{tb} is the allowable tensile stress for the bolt material. The value of σ_{tb} is usually kept low to allow for initial tightening stress in the bolts. After the core diameter is obtained, then the nominal diameter of the bolts is chosen from * tables. It may be noted that bolts of less than 12 mm diameter should never be used for hydraulic pipes, because very heavy initial tightening stresses may be induced in smaller bolts. The bolt centers should be as near the centre of the pipe as possible to avoid bending of the flange. But sufficient clearance between the bolt head and pipe surface must be provided for the tightening of the bolts without damaging the pipe material. The thickness of the flange is obtained by considering the flange to be under bending stresses due to the forces acting in one bolt. The maximum bending stress will be induced at the section X-X. The bending moment at this section is given by

$$M_{xx} = F_b \times e = \frac{F}{2} \times e$$

and section modulus, $Z = \frac{1}{6} \times b\, (t_f)^2$

where b = Width of the flange at the section X-X, and

t_f = Thickness of the flange.

Using the bending equation, we have

$$M_{xx} = \sigma_b . Z$$

or $F_b \times e = \sigma_b \times \frac{1}{6} \times b\, (t_f)^2$

where σ_b = Permissible bending stress for the flange material.

From the above expression, the value of t_f may be obtained, if b is known. The width of the flange is estimated from the lay out of the flange. The hydraulic joints with oval flanges are known as ***Armstrong's pipe joints.*** The various dimensions for a hydraulic joint may be obtained by using the following empirical relations:

Nominal diameter of bolts, $d = 0.75\, t + 10$ mm

Thickness of the flange, $t_f = 1.5\, t + 3$ mm

Outer diameter of the flange,

$$D_o = D + 2t + 4.6\, d$$

Pitch circle diameter, $D_p = D_o - (3\, t + 20 \text{ mm})$

Design of Square Flanged Pipe Joint : The design of a square flanged pipe joint, as shown in Fig. 8.14, is similar to that of an oval flanged pipe joint except that the load has to be divided into four bolts. The thickness of the flange may be obtained by considering the bending of the flange about one of the sections A-A, B-B, or C-C. A little consideration will show that the flange is weakest in bending about section A-A. Therefore the thickness of the flange is calculated by considering the bending of the flange, about section A-A.

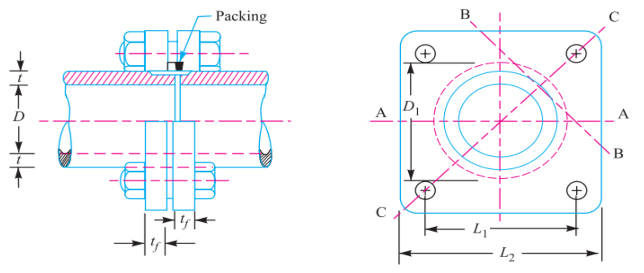


Fig. 8.14. Square flanged pipe joint.

Example . Design a square flanged pipe joint for pipes of internal diameter 50 mm subjected to an internal fluid pressure of 7 N/mm 2. The maximum tensile stress in the pipe material is not to exceed 2 1 MPa and in the bolts 2 8 MPa

Solution.

Given: D = 50 mm or R = 25 mm ; p = 7 N/mm2 ; σ_t = 21 MPa = 21 N/mm 2 ; σ_{tb} = 28 MPa = 28 N/mm 2 First of all, let us find the thickness of the pipe. According to Lame's equation, we know that thickness of the pipe,.

$$t = R\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right] = 25\left[\sqrt{\frac{21+7}{21-7}} - 1\right] = 10.35 \text{ say } 12 \text{ mm}$$

Assuming the width of packing as 10 mm, therefore outside diameter of the packing,

D_1 = 50 + 2 × Width of packing = 50 + 2 × 10 = 70 mm

Force trying to separate the flanges,

$$F = \frac{\pi}{4}(D_1)^2\ p = \frac{\pi}{4}(70)^2\ 7 = 26\ 943 \text{ N}$$

Since this force is to be resisted by four bolts, therefore force on each bolt,

F_b = F / 4 = 26 943 / 4 = 6735.8 N

Let d_c = Core diameter of the bolts.

We know that force on each bolt (F_b),

$$6735.8 = \frac{\pi}{4}(d_c)^2\ \sigma_{tb} = \frac{\pi}{4}(d_c)^2\ 28 = 22\ (d_c)^2$$

$$\therefore \quad (d_c)^2 = 6735.8/22 = 306 \quad \text{or} \quad d_c = 17.5 \text{ mm}$$

and nominal diameter of the bolts,

$$d = \frac{d_c}{0.84} = \frac{17.5}{0.84} = 20.9 \text{ say } 22 \text{ mm} \quad \textbf{Ans.}$$

The axes of the bolts are arranged at the corners of a square of such size that the corners of the nut clear the outside of the pipe.

Minimum length of a diagonal for this square,

L = Outside diameter of pipe + 2 × Dia. of bolt

= D + 2 t + 2 d = 50 + (2 × 12) + (2 × 22) = 118 mm

and side of this square,

$$L_1 = \frac{L}{\sqrt{2}} = \frac{118}{\sqrt{2}} = 83.5 \text{ mm}$$

The sides of the flange must be of sufficient length to accommodate the nuts and bolt heads without overhang. Therefore the length L_2 may be kept as $(L_1 + 2d)$ *i.e.*

$$L_2 = L_1 + 2d = 83.5 + 2 \times 22 = 127.5 \text{ mm}$$

The elevation of the flange is shown in Fig. 8.15. In order to find the thickness of the flange, consider the bending of the flange about section A-A. The bending about section A-A will take place due to the force in two bolts.

∴ Bending moment due to the force in two bolts (*i.e.* due to $2F_b$),

$$M_1 = 2F_b \times \frac{L_1}{2} = 2 \times 6735.8 \times \frac{83.5}{2} = 562\ 440 \text{ N-mm}$$

Water pressure acting on half the flange

$$= 2\ F_b = 2 \times 6735.8 = 13\ 472 \text{ N}$$

The flanges are screwed with pipe having metric threads of 4.4 threads in 10 mm (i.e. pitch of the threads is 10/4.4 = 2.28 mm).

Nominal or major diameter of the threads = Outside diameter of the pipe = D + 2 t = 50 + 2 × 12 = 74 mm

Nominal radius of the threads = 74 / 2 = 37 mm

Depth of the threads = 0.64 × Pitch of threads = 0.64 × 2.28 = 1.46 mm

Core or minor radius of the threads = 37 – 1.46 = 35.54 mm

Mean radius of the arc from A-A over which the load due to fluid pressure may be taken to be concentrated

$$= \frac{1}{2}(37 + 35.54) = 36.27 \text{ mm}$$

The centroid of this arc from *A-A*

$$= 0.6366 \times \text{Mean radius} = 0.6366 \times 36.27 = 23.1 \text{ mm}$$

∴ Bending moment due to the water pressure,

$$M_2 = 2\,F_b \times 23.1 = 2 \times 6735.8 \times 23.1 = 311\ 194 \text{ N-mm}$$

Since the bending moments M_1 and M_2 are in opposite directions, therefore

Net resultant bending moment on the flange about section A-A,

$$M = M_1 - M_2 = 562\ 440 - 311\ 194 = 251\ 246 \text{ N-mm}$$

Width of the flange at the section A-A,

$$b = L_2 - \text{Outside diameter of pipe} = 127.5 - 74 = 53.5 \text{ mm}$$

Let t_f = Thickness of the flange in mm.

∴ Section modulus,

$$Z = \frac{1}{6} \times b\,(t_f)^2 = \frac{1}{6} \times 53.5\,(t_f)^2 = 8.9\,(t_f)^2 \text{ mm}^3$$

We know that net resultant bending moment (M),

$$251\ 246 = \sigma_b . Z = 21 \times 8.9\,(t_f)^2 = 187\,(t_f)^2$$

∴ $(t_f)^2 = 251\ 246 / 187 = 1344$ or $t_f = 36.6$ say 38 mm **Ans.**

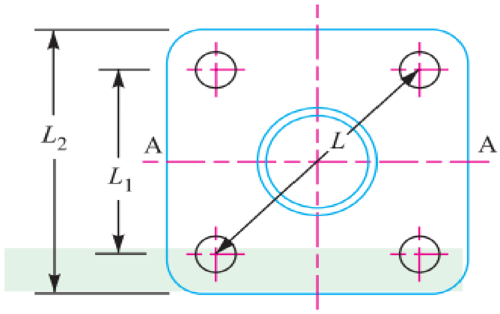


Fig. 8.15