CHAPTER

11

Screwed Joints

11.1 Introduction

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.

11.2 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 adopted to various operating conditions.
4. Screws are relatively cheap to produce due to standardisation and highly efficient manufacturing processes.

Disadvantages

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

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

11.3 Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. 11.1, are important from the subject point of view :

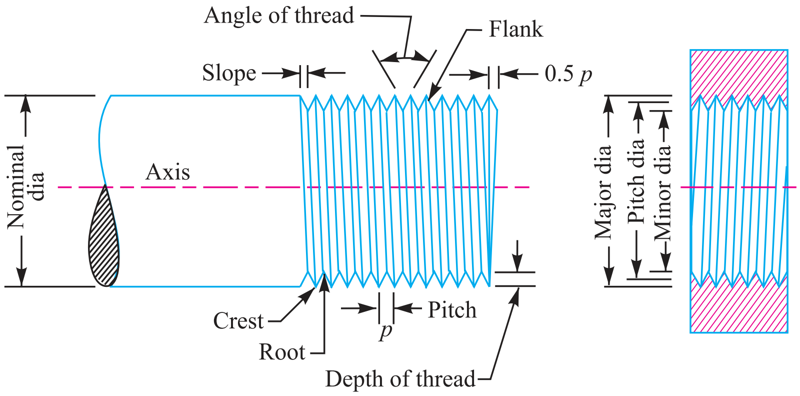


Fig. 11.1. Terms used in screw threads.

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 perpendicular distance between the crest and root.

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

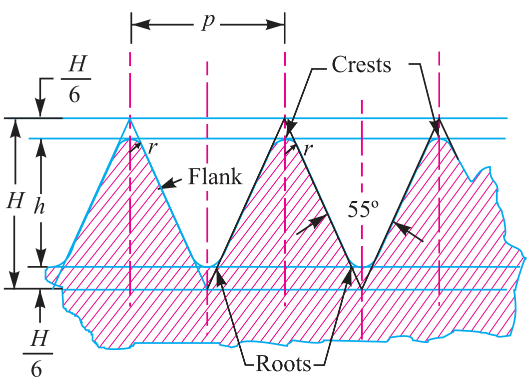
10. *Angle of thread.* It is the angle included by the flanks of the thread.

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

11.4 Forms of Screw Threads

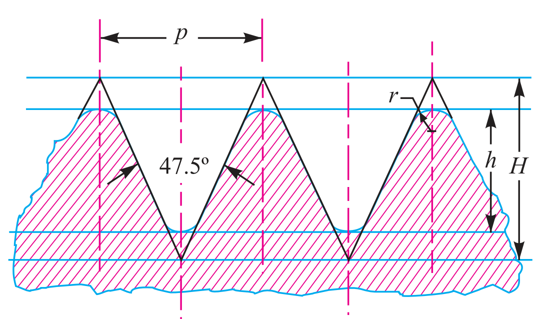
The following are the various forms of screw threads.

1. *British standard whitworth (B.S.W.) thread.* This is a British standard thread profile and has coarse pitches. It is a symmetrical *V*-thread in which the angle between the flankes, measured in an axial plane, is 55°. These threads are found on bolts and screwed fastenings for special purposes. The various proportions of B.S.W. threads are shown in Fig. 11.2.



$H = 0.96\,p$; $h = 0.64\,p$; $r = 0.1373\,p$

Fig. 11.2. British standard whitworth (B.S.W) thread.



$H = 1.13634\,p$; $h = 0.6\,p$; $r = 0.18083\,p$

Fig. 11.3. British association (B.A.) thread.

The British standard threads with fine pitches (B.S.F.) are used where great strength at the root is required. These threads are also used for line adjustments and where the connected parts are subjected to increased vibrations as in aero and automobile work.

The British standard pipe (B.S.P.) threads with fine pitches are used for steel and iron pipes and tubes carrying fluids. In external pipe threading, the threads are specified by the bore of the pipe.

2. *British association (B.A.) thread.* This is a B.S.W. thread with fine pitches. The proportions of the B.A. thread are shown in Fig. 11.3. These threads are used for instruments and other precision works.

3. *American national standard thread.* The American national standard or U.S. or Seller's thread has flat crests and roots. The flat crest can withstand more rough usage than sharp *V*-threads. These threads are used for general purposes *e.g.* on bolts, nuts, screws and tapped holes. The various proportions are shown in Fig. 11.4.

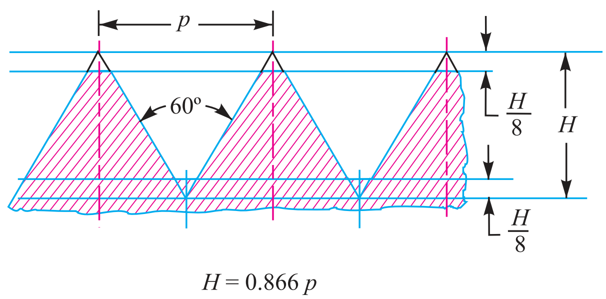


Fig. 11.4. American national standard thread.

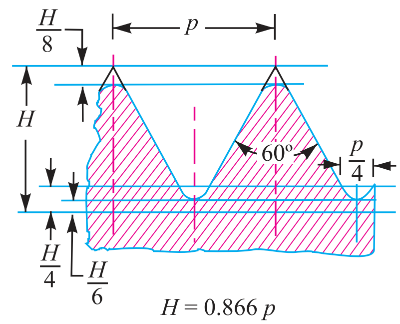


Fig. 11.5. Unified standard thread.

4. *Unified standard thread*. The three countries *i.e.*, Great Britain, Canada and United States came to an agreement for a common screw thread system with the included angle of 60°, in order to facilitate the exchange of machinery. The thread has rounded crests and roots, as shown in Fig. 11.5.

5. *Square thread*. The square threads, because of their high efficiency, are widely used for transmission of power in either direction. Such type of threads are usually found on the feed mechanisms of machine tools, valves, spindles, screw jacks etc. The square threads are not so strong as V-threads but they offer less frictional resistance to motion than Whitworth threads. The pitch of the square thread is often taken twice that of a B.S.W. thread of the same diameter. The proportions of the thread are shown in Fig. 11.6.

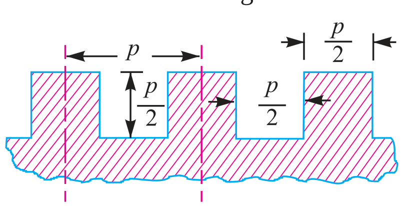


Fig. 11.6. Square thread.

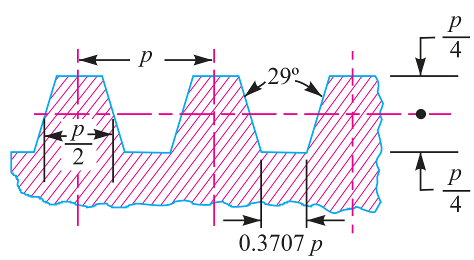


Fig. 11.7. Acme thread.

6. *Acme thread*. It is a modification of square thread. It is much stronger than square thread and can be easily produced. These threads are frequently used on screw cutting lathes, brass valves, cocks and bench vices. When used in conjunction with a split nut, as on the lead screw of a lathe, the tapered sides of the thread facilitate ready engagement and disengagement of the halves of the nut when required. The various proportions are shown in Fig. 11.7.

7. ***Knuckle thread.*** It is also a modification of square thread. It has rounded top and bottom. It can be cast or rolled easily and can not economically be made on a machine. These threads are used for rough and ready work. They are usually found on railway carriage couplings, hydrants, necks of glass bottles and large moulded insulators used in electrical trade.

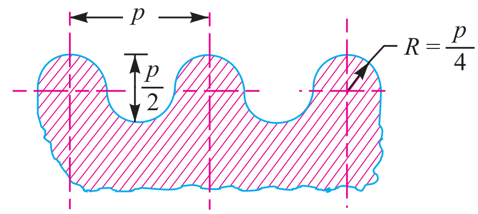
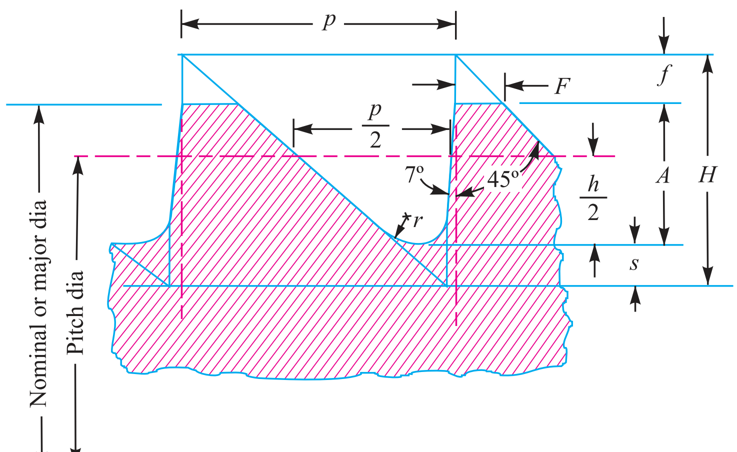


Fig. 11.8. Knuckle thread.

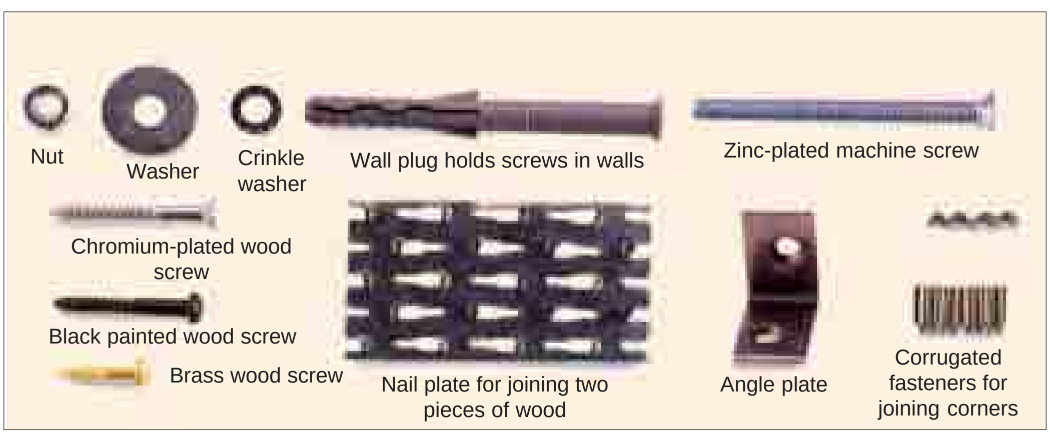
8. ***Buttress thread.*** It is used for transmission of power in one direction only. The force is transmitted almost parallel to the axis. This thread units the advantage of both square and V-threads. It has a low frictional resistance characteristics of the square thread and have the same strength as that of V-thread. The spindles of bench vices are usually provided with buttress thread. The various proportions of buttress thread are shown in Fig. 11.9.



$H = 0.89064\,p$; $A = 0.50286\,p$; $f = 0.24532\,p$;
$s = 0.13946\,p$; $F = 0.27544\,p$; $r = 0.12055\,p$.

Fig. 11.9. Buttress thread.

9. ***Metric thread.*** It is an Indian standard thread and is similar to B.S.W. threads. It has an included angle of 60° instead of 55°. The basic profile of the thread is shown in Fig. 11.10 and the design profile of the nut and bolt is shown in Fig. 11.11.



Simple Machine Tools.

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

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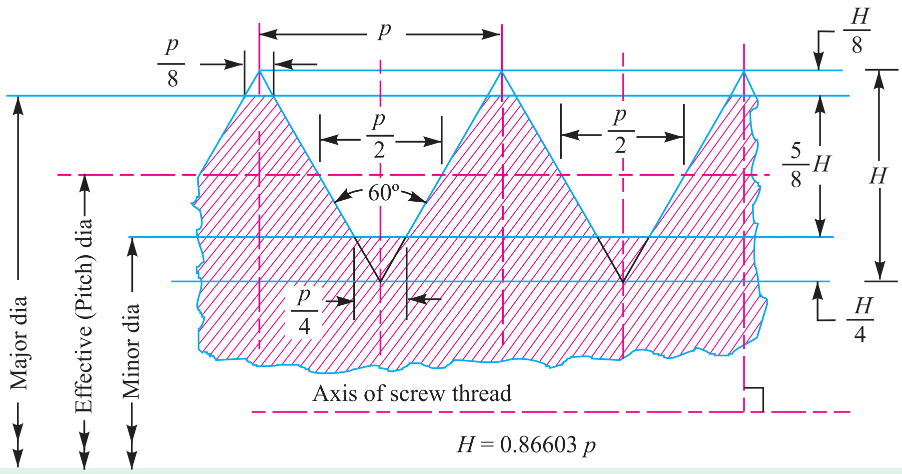
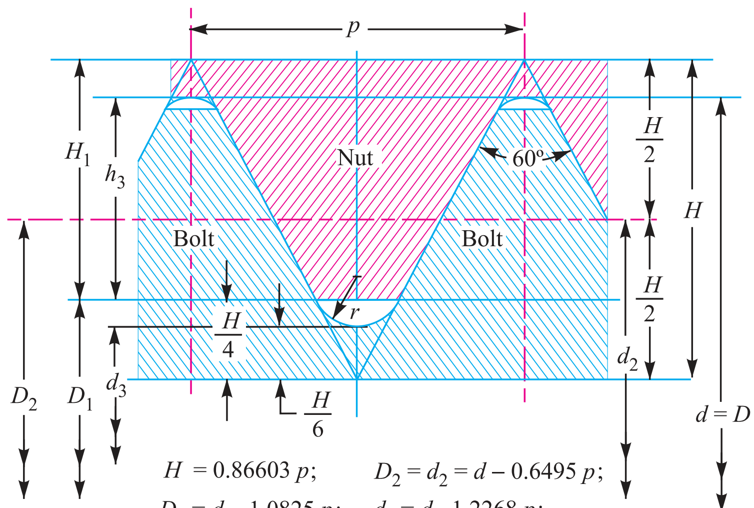


$H = 0.86603\ p$

Fig. 11.10. Basic profile of the thread.



$H = 0.86603\ p$; $\quad D_2 = d_2 = d - 0.6495\ p$;

$D_1 = d - 1.0825\ p$; $\quad d_3 = d - 1.2268\ p$;

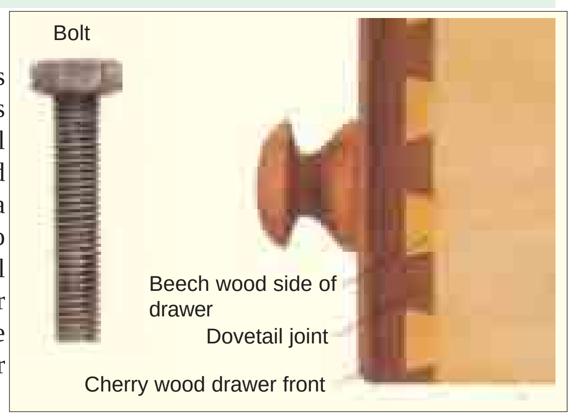
$H_1 = \dfrac{5}{8}H$; $\quad h_3 = \dfrac{17}{24}H$; $\quad r = \dfrac{H}{6}$

d = Diameter of nut; $\quad D$ = Diameter of bolt.

Fig. 11.11. Design profile of the nut and bolt.

11.5 Location of Screwed Joints

The choice of type of fastenings and its location are very important. The fastenings should be located in such a way so that they will be subjected to tensile and/or shear loads and bending of the fastening should be reduced to a minimum. The bending of the fastening due to misalignment, tightening up loads, or external loads are responsible for many failures. In order to relieve fastenings of bending stresses, the use of clearance spaces, spherical seat washers, or other devices may be used.



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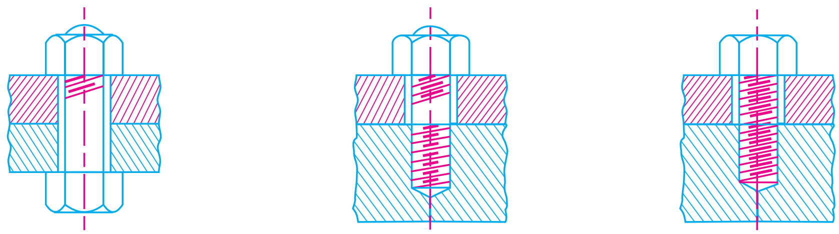
11.6 Common Types of Screw Fastenings

Following are the common types of screw fastenings :

1. ***Through bolts.*** A through bolt (or simply a bolt) is shown in Fig. 11.12 (*a*). It is a cylindrical bar with threads for the nut at one end and head at the other end. The cylindrical part of the bolt is known as **shank.** It is passed through drilled holes in the two parts to be fastened together and clamped them securely to each other as the nut is screwed on to the threaded end. The through bolts may or may not have a machined finish and are made with either hexagonal or square heads. A through bolt should pass easily in the holes, when put under tension by a load along its axis. If the load acts perpendicular to the axis, tending to slide one of the connected parts along the other end thus subjecting it to shear, the holes should be reamed so that the bolt shank fits snugly there in. The through bolts according to their usage may be known as **machine bolts, carriage bolts, automobile bolts, eye bolts etc.**



(*a*) Through bolt. (*b*) Tap bolt. (*c*) Stud.

Fig. 11.12

2. ***Tap bolts.*** A tap bolt or screw differs from a bolt. It is screwed into a tapped hole of one of the parts to be fastened without the nut, as shown in Fig. 11.12 (*b*).

3. ***Studs.*** A stud is a round bar threaded at both ends. One end of the stud is screwed into a tapped hole of the parts to be fastened, while the other end receives a nut on it, as shown in Fig. 11.12 (*c*). Studs are chiefly used instead of tap bolts for securing various kinds of covers *e.g.* covers of engine and pump cylinders, valves, chests etc.

Deck-handler crane is used on ships to move loads

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

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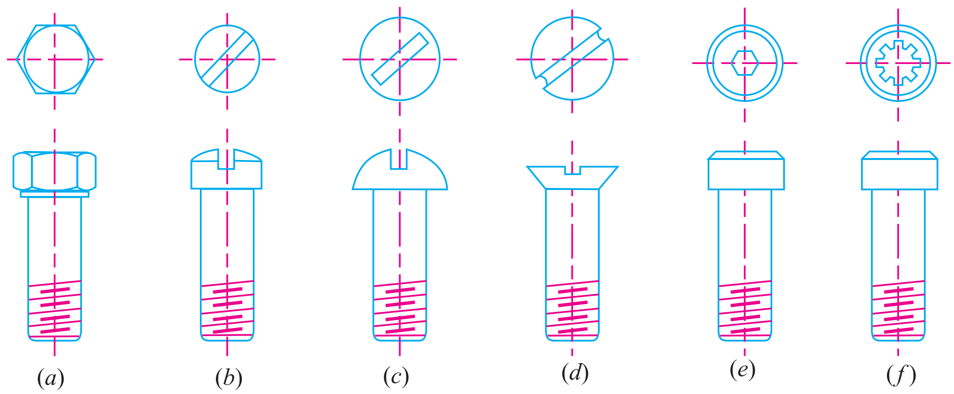
This is due to the fact that when tap bolts are unscrewed or replaced, they have a tendency to break the threads in the hole. This disadvantage is overcome by the use of studs.

4. ***Cap screws.*** The cap screws are similar to tap bolts except that they are of small size and a variety of shapes of heads are available as shown in Fig. 11.13.



(*a*) Hexagonal head; (*b*) Fillister head; (*c*) Round head; (*d*) Flat head;
(*e*) Hexagonal socket; (*f*) Fluted socket.

Fig. 11.13. Types of cap screws.

5. ***Machine screws.*** These are similar to cap screws with the head slotted for a screw driver. These are generally used with a nut.

6. ***Set screws.*** The set screws are shown in Fig. 11.14. These are used to prevent relative motion between the two parts. A set screw is screwed through a threaded hole in one part so that its point (*i.e.* end of the screw) presses against the other part. This resists the relative motion between the two parts by means of friction between the point of the screw and one of the parts. They may be used instead of key to prevent relative motion between a hub and a shaft in light power transmission members. They may also be used in connection with a key, where they prevent relative axial motion of the shaft, key and hub assembly.

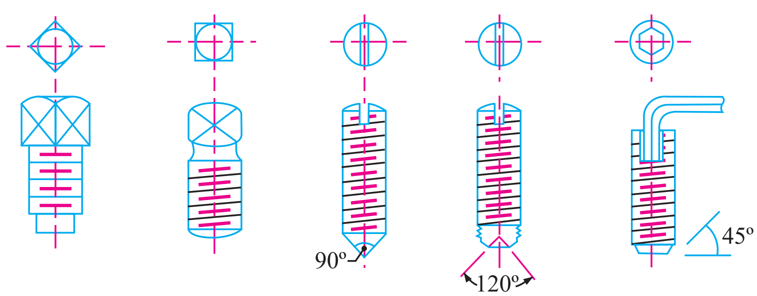


Fig. 11.14. Set screws.

The diameter of the set screw (d) may be obtained from the following expression:

$$d = 0.125\, D + 8 \text{ mm}$$

where D is the diameter of the shaft (in mm) on which the set screw is pressed.

The tangential force (in newtons) at the surface of the shaft is given by

$$F = 6.6\, (d)^{2.3}$$

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∴ Torque transmitted by a set screw,

$$T = F \times \frac{D}{2} \text{ N-m} \qquad \text{... (}D\text{ is in metres)}$$

and power transmitted (in watts), $P = \dfrac{2\pi N.T}{60}$, where N is the speed in r.p.m.

11.7 Locking Devices

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

1. *Jam nut or lock nut.* A most common locking device is a jam, lock or check nut. It has about one-half to two-third thickness of the standard nut. The thin lock nut is first tightened down with ordinary force, and then the upper nut (*i.e.* thicker nut) is tightened down upon it, as shown in Fig. 11.15 (*a*). The upper nut is then held tightly while the lower one is slackened back against it.

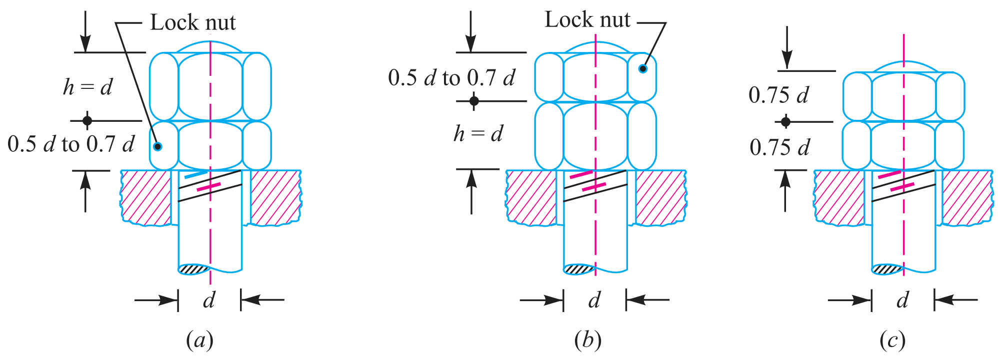

Fig. 11.15. Jam nut or lock nut.

In slackening back the lock nut, a thin spanner is required which is difficult to find in many shops. Therefore to overcome this difficulty, a thin nut is placed on the top as shown in Fig. 11.15 (*b*).

If the nuts are really tightened down as they should be, the upper nut carries a greater tensile load than the bottom one. Therefore, the top nut should be thicker one with a thin nut below it because it is desirable to put whole of the load on the thin nut. In order to overcome both the difficulties, both the nuts are made of the same thickness as shown in Fig. 11.15 (*c*).

2. *Castle nut.* It consists of a hexagonal portion with a cylindrical upper part which is slotted in line with the centre of each face, as shown in Fig. 11.16. The split pin passes through two slots in the nut and a hole in the bolt, so that a positive lock is obtained unless the pin shears. It is extensively used on jobs subjected to sudden shocks and considerable vibration such as in automobile industry.

3. *Sawn nut.* It has a slot sawed about half way through, as shown in Fig. 11.17. After the nut is screwed down, the small screw is tightened which produces more friction between the nut and the bolt. This prevents the loosening of nut.

4. *Penn, ring or grooved nut.* It has a upper portion hexagonal and a lower part cylindrical as shown in Fig. 11.18. It is largely used where bolts pass through connected pieces reasonably near their edges such as in marine type connecting rod ends. The bottom portion is cylindrical and is recessed to receive the tip of the locking set screw. The bolt hole requires counter-boring to receive the cylindrical portion of the nut. In order to prevent bruising of the latter by the case hardened tip of the set screw, it is recessed.

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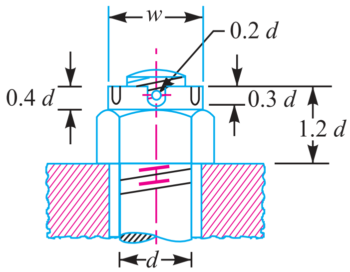

Fig. 11.16. Castle nut.

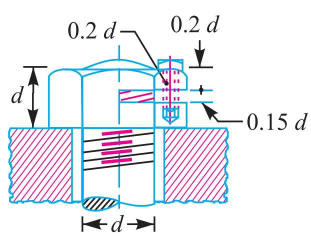

Fig. 11.17. Sawn nut.

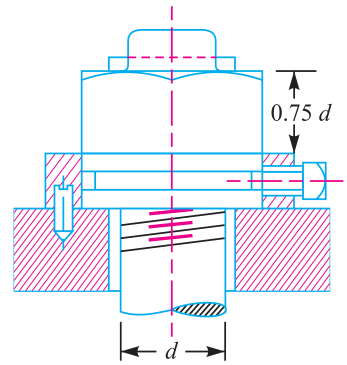

Fig. 11.18. Penn, ring or grooved nut.

5. *Locking with pin*. The nuts may be locked by means of a taper pin or cotter pin passing through the middle of the nut as shown in Fig. 11.19 (*a*). But a split pin is often driven through the bolt above the nut, as shown in Fig. 11.19 (*b*).

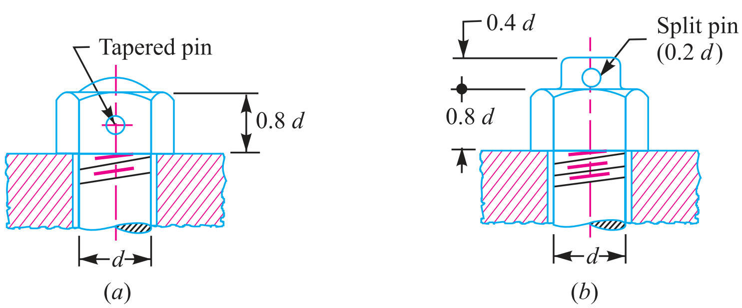

Fig. 11.19. Locking with pin.

6. *Locking with plate*. A form of stop plate or locking plate is shown in Fig. 11.20. The nut can be adjusted and subsequently locked through angular intervals of 30° by using these plates.

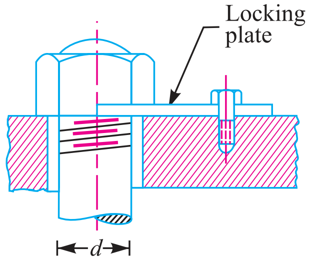

Fig. 11.20. Locking with plate.

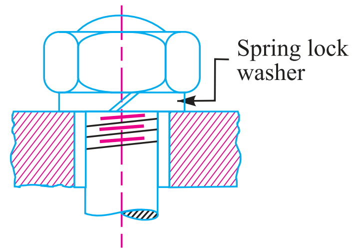

Fig. 11.21. Locking with washer.

7. *Spring lock washer*. A spring lock washer is shown in Fig. 11.21. As the nut tightens the washer against the piece below, one edge of the washer is caused to dig itself into that piece, thus increasing the resistance so that the nut will not loosen so easily. There are many kinds of spring lock washers manufactured, some of which are fairly effective.

11.8 Designation of Screw Threads

According to Indian standards, IS : 4218 (Part IV) 1976 (Reaffirmed 1996), the complete designation of the screw thread shall include

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1. *Size designation.* The size of the screw thread is designated by the letter `*M*' followed by the diameter and pitch, the two being separated by the sign ×. When there is no indication of the pitch, it shall mean that a coarse pitch is implied.

2. *Tolerance designation.* This shall include

(*a*) A figure designating tolerance grade as indicated below:
'7' for fine grade, '8' for normal (medium) grade, and '9' for coarse grade.

(*b*) A letter designating the tolerance position as indicated below :
'*H*' for unit thread, '*d*' for bolt thread with allowance, and '*h*' for bolt thread without allowance.

For example, A bolt thread of 6 mm size of coarse pitch and with allowance on the threads and normal (medium) tolerance grade is designated as *M*6-8*d*.

11.9 Standard Dimensions of Screw Threads

The design dimensions of I.S.O. screw threads for screws, bolts and nuts of coarse and fine series are shown in Table 11.1.

Table 11.1. Design dimensions of screw threads, bolts and nuts according to IS : 4218 (Part III) 1976 (Reaffirmed 1996) (Refer Fig. 11.1)

Designation	*Pitch mm*	*Major or nominal diameter Nut and Bolt (d = D) mm*	*Effective or pitch diameter Nut and Bolt (d_p) mm*	*Minor or core diameter (d_c) mm*		*Depth of thread (bolt) mm*	*Stress area mm²*
				Bolt	*Nut*		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Coarse series							
M 0.4	0.1	0.400	0.335	0.277	0.292	0.061	0.074
M 0.6	0.15	0.600	0.503	0.416	0.438	0.092	0.166
M 0.8	0.2	0.800	0.670	0.555	0.584	0.123	0.295
M 1	0.25	1.000	0.838	0.693	0.729	0.153	0.460
M 1.2	0.25	1.200	1.038	0.893	0.929	0.158	0.732
M 1.4	0.3	1.400	1.205	1.032	1.075	0.184	0.983
M 1.6	0.35	1.600	1.373	1.171	1.221	0.215	1.27
M 1.8	0.35	1.800	1.573	1.371	1.421	0.215	1.70
M 2	0.4	2.000	1.740	1.509	1.567	0.245	2.07
M 2.2	0.45	2.200	1.908	1.648	1.713	0.276	2.48
M 2.5	0.45	2.500	2.208	1.948	2.013	0.276	3.39
M 3	0.5	3.000	2.675	2.387	2.459	0.307	5.03
M 3.5	0.6	3.500	3.110	2.764	2.850	0.368	6.78
M 4	0.7	4.000	3.545	3.141	3.242	0.429	8.78
M 4.5	0.75	4.500	4.013	3.580	3.688	0.460	11.3
M 5	0.8	5.000	4.480	4.019	4.134	0.491	14.2
M 6	1	6.000	5.350	4.773	4.918	0.613	20.1

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(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
M 7	1	7.000	6.350	5.773	5.918	0.613	28.9
M 8	1.25	8.000	7.188	6.466	6.647	0.767	36.6
M 10	1.5	10.000	9.026	8.160	8.876	0.920	58.3
M 12	1.75	12.000	10.863	9.858	10.106	1.074	84.0
M 14	2	14.000	12.701	11.546	11.835	1.227	115
M 16	2	16.000	14.701	13.546	13.835	1.227	157
M 18	2.5	18.000	16.376	14.933	15.294	1.534	192
M 20	2.5	20.000	18.376	16.933	17.294	1.534	245
M 22	2.5	22.000	20.376	18.933	19.294	1.534	303
M 24	3	24.000	22.051	20.320	20.752	1.840	353
M 27	3	27.000	25.051	23.320	23.752	1.840	459
M 30	3.5	30.000	27.727	25.706	26.211	2.147	561
M 33	3.5	33.000	30.727	28.706	29.211	2.147	694
M 36	4	36.000	33.402	31.093	31.670	2.454	817
M 39	4	39.000	36.402	34.093	34.670	2.454	976
M 42	4.5	42.000	39.077	36.416	37.129	2.760	1104
M 45	4.5	45.000	42.077	39.416	40.129	2.760	1300
M 48	5	48.000	44.752	41.795	42.587	3.067	1465
M 52	5	52.000	48.752	45.795	46.587	3.067	1755
M 56	5.5	56.000	52.428	49.177	50.046	3.067	2022
M 60	5.5	60.000	56.428	53.177	54.046	3.374	2360
Fine series							
M 8 × 1	1	8.000	7.350	6.773	6.918	0.613	39.2
M 10 × 1.25	1.25	10.000	9.188	8.466	8.647	0.767	61.6
M 12 × 1.25	1.25	12.000	11.184	10.466	10.647	0.767	92.1
M 14 × 1.5	1.5	14.000	13.026	12.160	12.376	0.920	125
M 16 × 1.5	1.5	16.000	15.026	14.160	14.376	0.920	167
M 18 × 1.5	1.5	18.000	17.026	16.160	16.376	0.920	216
M 20 × 1.5	1.5	20.000	19.026	18.160	18.376	0.920	272
M 22 × 1.5	1.5	22.000	21.026	20.160	20.376	0.920	333
M 24 × 2	2	24.000	22.701	21.546	21.835	1.227	384
M 27 × 2	2	27.000	25.701	24.546	24.835	1.227	496
M 30 × 2	2	30.000	28.701	27.546	27.835	1.227	621
M 33 × 2	2	33.000	31.701	30.546	30.835	1.227	761
M 36 × 3	3	36.000	34.051	32.319	32.752	1.840	865
M 39 × 3	3	39.000	37.051	35.319	35.752	1.840	1028

Note : In case the table is not available, then the core diameter (d_c) may be taken as 0.84 d, where d is the major diameter.

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11.10 Stresses in Screwed Fastening due to Static Loading

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

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

We shall now discuss these stresses, in detail, in the following articles.

11.11 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

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

where P_i = 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 (*i.e.* stress area)

The stress area may be obtained from Table 11.1 or it may be found by using the relation

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

where d_p = Pitch diameter, and

d_c = Core or minor diameter.

Simple machine tools.

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

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2. ***Torsional shear stress caused by the frictional resistance of the threads during its tightening.*** The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation. We know that

$$\frac{T}{J} = \frac{\tau}{r}$$

$$\therefore \quad \tau = \frac{T}{J} \times r = \frac{T}{\frac{\pi}{32}(d_c)^4} \times \frac{d_c}{2} = \frac{16\,T}{\pi\,(d_c)^3}$$

where τ = Torsional shear stress,

T = Torque applied, and

d_c = Minor or core diameter of the thread.

It has been shown during experiments that due to repeated unscrewing and tightening of the nut, there is a gradual scoring of the threads, which increases the torsional twisting moment (T).

3. ***Shear stress across the threads.*** The average thread shearing stress for the screw (τ_s) is obtained by using the relation :

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

where b = Width of the thread section at the root.

The average thread shearing stress for the nut is

$$\tau_n = \frac{P}{\pi d \times b \times n}$$

where d = Major diameter.

4. ***Compression or crushing stress on threads.*** The compression or crushing stress between the threads (σ_c) may be obtained by using the relation :

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

where d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement.

5. ***Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis.*** When the outside surfaces of the parts to be connected are not parallel to each other, then the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by

$$\sigma_b = \frac{x \,.\, E}{2l}$$

where x = Difference in height between the extreme corners of the nut or head,

l = Length of the shank of the bolt, and

E = Young's modulus for the material of the bolt.

Example 11.1. *Determine the safe tensile load for a bolt of M 30, assuming a safe tensile stress of 42 MPa.*

Solution. Given : d = 30 mm ; σ_t = 42 MPa = 42 N/mm^2

From Table 11.1 (coarse series), we find that the stress area *i.e.* cross-sectional area at the bottom of the thread corresponding to M 30 is 561 mm^2.

∴ Safe tensile load = Stress area × σ_t = 561 × 42 = 23 562 N = 23.562 kN **Ans.**

Note: In the above example, we have assumed that the bolt is not initially stressed.

Example 11.2. *Two machine parts are fastened together tightly by means of a 24 mm tap bolt. If the load tending to separate these parts is neglected, find the stress that is set up in the bolt by the initial tightening.*

Solution. Given : d = 24 mm

From Table 11.1 (coarse series), we find that the core diameter of the thread corresponding to M 24 is d_c = 20.32 mm.

Let σ_t = Stress set up in the bolt.

We know that initial tension in the bolt,

$$P = 2840\, d = 2840 \times 24 = 68\ 160 \text{ N}$$

We also know that initial tension in the bolt (P),

$$68\ 160 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(20.30)^2\,\sigma_t = 324\,\sigma_t$$

$$\therefore \quad \sigma_t = 68\ 160 / 324 = 210 \text{ N/mm}^2 = 210 \text{ MPa}$$ **Ans.**

11.12 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,

$$P = \frac{\pi}{4}(d_c)^2\,\sigma_t \quad \text{or} \quad d_c = \sqrt{\frac{4P}{\pi\,\sigma_t}}$$

Now from Table 11.1, the value of the nominal diameter of bolt corresponding to the value of d_c may be obtained or stress area $\left[\frac{\pi}{4}(d_c)^2\right]$ may be fixed.

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.

Simple machine tools.

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

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

Let d = Major diameter of the bolt, and

n = Number of bolts.

∴ Shearing load carried by the bolts,

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

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

Maximum principal shear stress,

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

and maximum principal tensile stress,

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

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

Example 11.3. *An eye bolt is to be used for lifting a load of 60 kN. Find the nominal diameter of the bolt, if the tensile stress is not to exceed 100 MPa. Assume coarse threads.*

Solution. Given : P = 60 kN = 60 × 10^3 N ; σ_t = 100 MPa = 100 N/mm^2

An eye bolt for lifting a load is shown in Fig. 11.22.

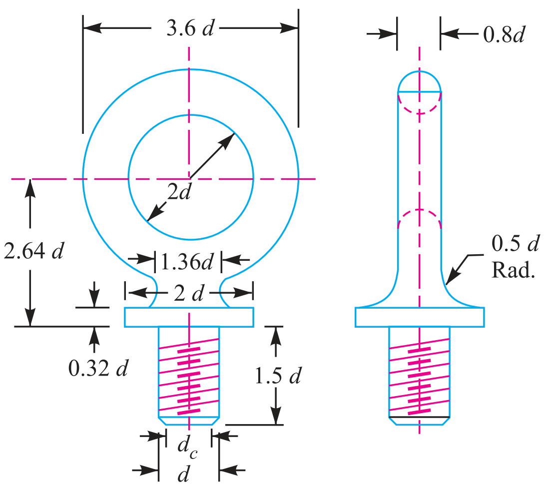

Fig. 11.22

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that load on the bolt (P),

$$60 \times 10^3 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2\, 100 = 78.55\,(d_c)^2$$

∴ $(d_c)^2$ = 600 × 10^3 / 78.55 = 764 or d_c = 27.6 mm

From Table 11.1 (coarse series), we find that the standard core diameter (d_c) is 28.706 mm and the corresponding nominal diameter (d) is 33 mm. **Ans.**

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Note : A lifting eye bolt, as shown in Fig. 11.22, is used for lifting and transporting heavy machines. It consists of a ring of circular cross-section at the head and provided with threads at the lower portion for screwing inside a threaded hole on the top of the machine.

Example 11.4. *Two shafts are connected by means of a flange coupling to transmit torque of 25 N-m. The flanges of the coupling are fastened by four bolts of the same material at a radius of 30 mm. Find the size of the bolts if the allowable shear stress for the bolt material is 30 MPa.*

Solution. Given : $T = 25$ N-m $= 25 \times 10^3$ N-mm ; $n = 4$; $R_p = 30$ mm ; $\tau = 30$ MPa $= 30$ N/mm^2

We know that the shearing load carried by flange coupling,

$$P_s = \frac{T}{R_p} = \frac{25 \times 10^3}{30} = 833.3 \text{ N} \quad ...(i)$$

Let d_c = Core diameter of the bolt.

∴ Resisting load on the bolts

$$= \frac{\pi}{4}(d_c)^2 \tau \times n = \frac{\pi}{4}(d_c)^2\ 30 \times 4 = 94{\cdot}26\ (d_c)^2 \quad ...(ii)$$

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

$$(d_c)^2 = 833.3 / 94.26 = 8.84 \quad \text{or} \quad d_c = 2.97 \text{ mm}$$

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 3.141 mm and the corresponding size of the bolt is M 4. **Ans.**

Example 11.5. *A lever loaded safety valve has a diameter of 100 mm and the blow off pressure is 1.6 N/mm^2. The fulcrum of the lever is screwed into the cast iron body of the cover. Find the diameter of the threaded part of the fulcrum if the permissible tensile stress is limited to 50 MPa and the leverage ratio is 8.*

Solution. Given : $D = 100$ mm ; $p = 1.6$ N/mm^2; $\sigma_t = 50$ MPa $= 50$ N/mm^2

We know that the load acting on the valve,

$$F = \text{Area} \times \text{pressure} = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4}(100)^2\ 1.6 = 12\ 568 \text{ N}$$

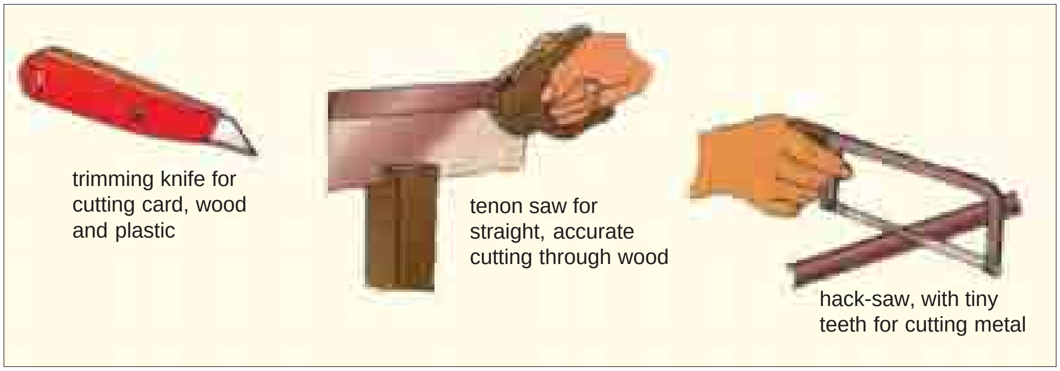
Since the leverage is 8, therefore load at the end of the lever,

$$W = \frac{12\ 568}{8} = 1571 \text{ N}$$

∴ Load on the fulcrum,

$$P = F - W = 12\ 568 - 1571 = 10\ 997 \text{ N} \quad ...(i)$$

Let d_c = Core diameter of the threaded part.



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

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∴ Resisting load on the threaded part of the fulcrum,

$$P = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(d_c)^2\,50 = 39.3\,(d_c)^2 \qquad \text{...(ii)}$$

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

$$(d_c)^2 = 10\,997 / 39.3 = 280 \qquad \text{or} \qquad d_c = 16.7 \text{ mm}$$

From Table 11.1 (fine series), we find that the standard core diameter is 18.376 mm and the corresponding size of the bolt is M 20 × 1.5. **Ans.**

11.13 Stress due to Combined Forces

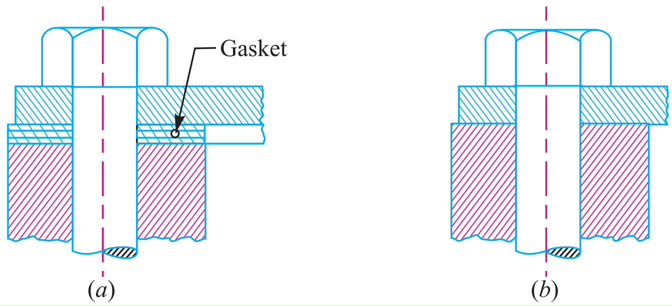

Fig. 11.23

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

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

2. The extenal 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. 11.23 (*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. 11.23 (*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 :

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

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.

Simple machine tools.

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

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For soft gaskets and large bolts, the value of a is high and the value of $\frac{a}{1+a}$ 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 $\frac{a}{1+a}$ (*i.e.* K) for various type of joints are shown in Table 11.2. 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.

Table 11.2. 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

11.14 Design of Cylinder Covers

The cylinder covers may be secured by means of bolts or studs, but studs are preferred. The possible arrangement of securing the cover with bolts and studs is shown in Fig. 11.24 (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 be adopted.

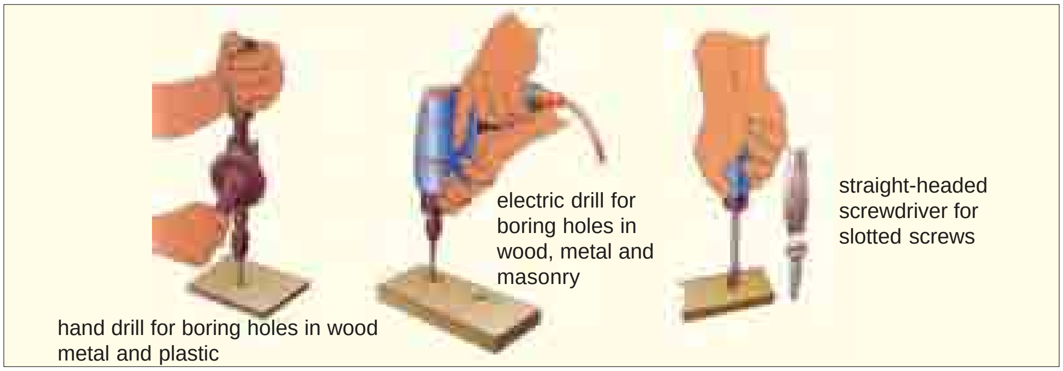
Let D = Diameter of the cylinder,

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.



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

We know that upward force acting on the cylinder cover,

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

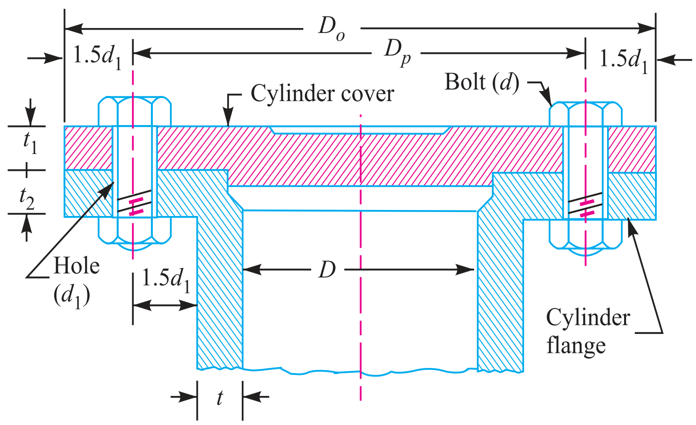
This force is resisted by n number of bolts or studs provided on the cover.

∴ Resisting force offered by n number of bolts or studs,

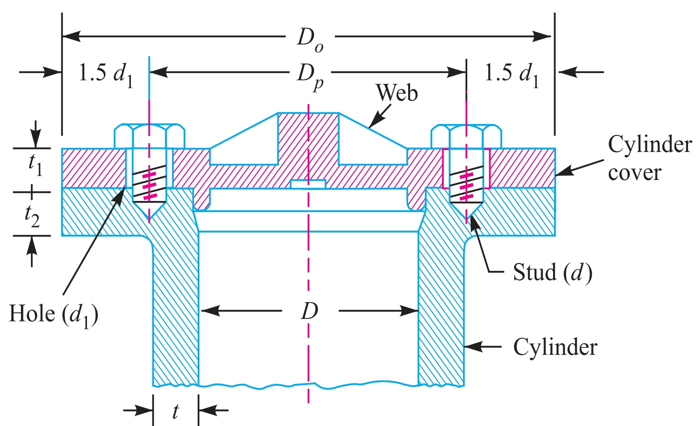
$$P = \frac{\pi}{4} (d_c)^2\ \sigma_{tb} \times n \qquad ...(ii)$$

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

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



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



(*b*) Arrangement of securing the cylinder cover with studs.

Fig. 11.24

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 $20\sqrt{d_1}$ and $30\sqrt{d_1}$, 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

$$D_o = D_p + 3d_1 = D + 2t + 6d_1$$

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. 11.25. 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.

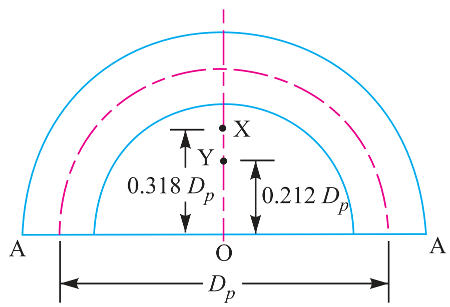


Fig. 11.25. Semi-cover plate of a cylinder.

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

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

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

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

where w = Width of plate

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

= $D_o - 2d_1$

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

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. 11.26.

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)

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

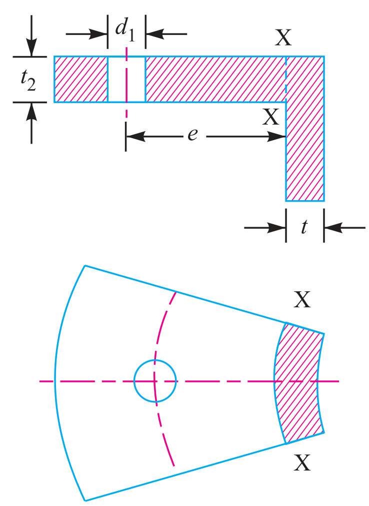


Fig. 11.26. A portion of the cylinder flange.

∴ Bending moment, $M = \text{Load on each bolt} \times e = \dfrac{P}{n} \times e$

Radius of the section *X-X*,

$$R = \text{Cylinder radius} + \text{Thickness of cylinder wall} = \frac{D}{2} + t$$

Width of the section *X-X*,

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

Section modulus, $Z = \dfrac{1}{6} w (t_2)^2$

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

Example 11.6. *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/mm^2 ; $\sigma_t = 33$ MPa $= 33$ N/mm^2

Let d = Nominal diameter of studs,

d_c = Core diameter of studs, and

n = Number of studs.

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

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

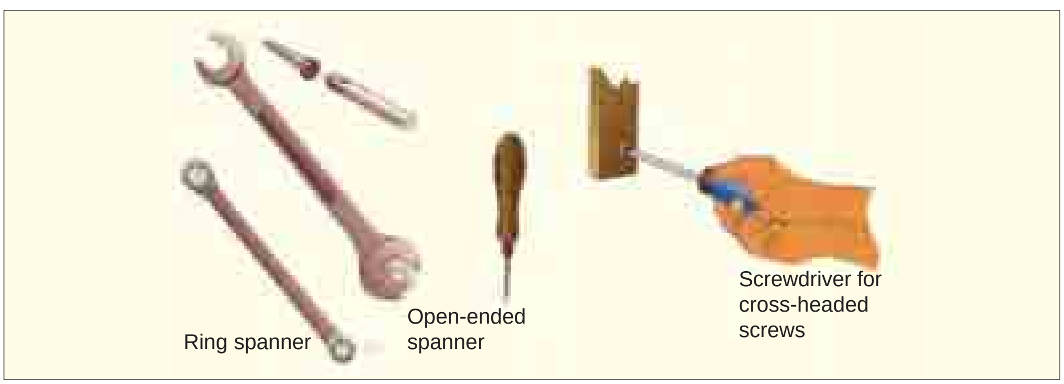
Assume that the studs of nominal diameter 24 mm are used. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the stud is 20.32 mm.

∴ 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 say 12 **Ans.**



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

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

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

...(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 stud chosen is satisfactory.

∴ Size of the stud = M 24 **Ans.**

Example 11.7. *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/mm² ; σ_t = 60 MPa = 60 N/mm² ; σ_{tb} = 40 MPa = 40 N/mm²

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),

$$P = \frac{\pi}{4}(D)^2\, p = \frac{\pi}{4}(120)^2 6 = 67\,860 \text{ N} \quad \text{...}(i)$$

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,

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

* The circumferential pitch of the studs can not be measured and marked on the cylinder cover. The centres of the holes are usually marked by angular distribution of the pitch circle into n number of equal parts. In the present case, the angular displacement of the stud hole centre will be 360°/12 = 30°.

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

$$n = 67\,860 / 12\,973 = 5.23 \text{ say } 6$$

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.

∴ 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.

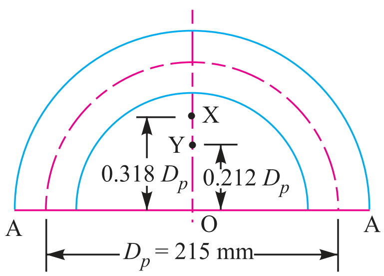


Fig. 11.27

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,

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

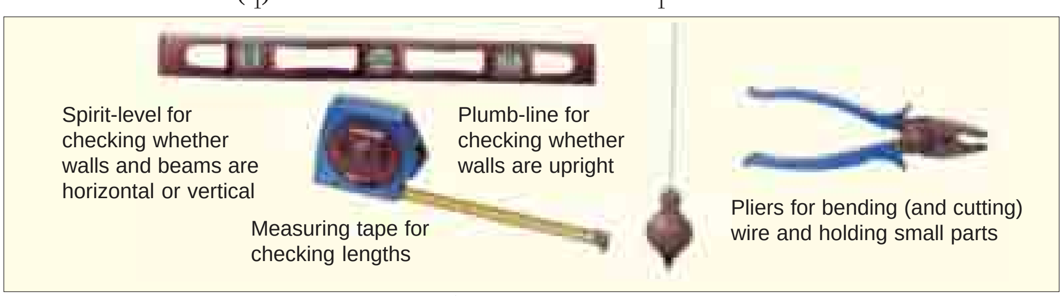
∴ Section modulus,

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

We know that bending (tensile) stress,

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

∴ $(t_1)^2 = 773\,265 / 40 \times 60 = 322$ or $t_1 = 18$ mm **Ans.**



Simple machine tools.

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

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Example 11.8. *The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm². It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa.*

Solution. Given: $p = 0.7$ N/mm² ; $n = 12$; $D = 300$ mm ; $\sigma_t = 100$ MPa $= 100$ N/mm²

We know that the total force (or the external load) acting on the cylinder head *i.e.* on 12 bolts,

$$= \frac{\pi}{4}(D)^2 \; p = \frac{\pi}{4}(300)^2 \; 0.7 = 49\,490 \text{ N}$$

∴ External load on the cylinder head per bolt,

$$P_2 = 49\,490 / 12 = 4124 \text{ N}$$

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that initial tension due to tightening of bolt,

$$P_1 = 2840\, d \text{ N} \qquad \text{... (where } d \text{ is in mm)}$$

From Table 11.2, we find that for soft copper gasket with long through bolts, the minimum value of $K = 0.5$.

∴ Resultant axial load on the bolt,

$$P = P_1 + K \,.\, P_2 = 2840\,d + 0.5 \times 4124 = (2840\,d + 2062) \text{ N}$$

We know that load on the bolt (P),

$$2840\,d + 2062 = \frac{\pi}{4}(d_c)^2 \; \sigma_t = \frac{\pi}{4}(0.84d)^2 \; 100 = 55.4\,d^2 \qquad \text{...(Taking } d_c = 0.84\,d)$$

$$\therefore \quad 55.4\,d^2 - 2840d - 2062 = 0$$

or

$$d^2 - 51.3d - 37.2 = 0$$

$$\therefore \quad d = \frac{51.3 \pm \sqrt{(51.3)^2 + 4 \times 37.2}}{2} = \frac{51.3 \pm 52.7}{2} = 52 \text{ mm}$$

..(Taking + ve sign)

Thus, we shall use a bolt of size M 52. **Ans.**

Example 11.9. *A steam engine of effective diameter 300 mm is subjected to a steam pressure of 1.5 N/mm². The cylinder head is connected by 8 bolts having yield point 330 MPa and endurance limit at 240 MPa. The bolts are tightened with an initial preload of 1.5 times the steam load. A soft copper gasket is used to make the joint leak-proof. Assuming a factor of safety 2, find the size of bolt required. The stiffness factor for copper gasket may be taken as 0.5.*

Solution. Given : $D = 300$ mm ; $p = 1.5$ N/mm² ; $n = 8$; $\sigma_y = 330$ MPa $= 330$ N/mm²; $\sigma_e = 240$ MPa $= 240$ N/mm² ; $P_1 = 1.5\, P_2$; *F.S.* = 2 ; $K = 0.5$

We know that steam load acting on the cylinder head,

$$P_2 = \frac{\pi}{4}(D)^2 \; p = \frac{\pi}{4}(300)^2 \; 1.5 = 106\,040 \text{ N}$$

∴ Initial pre-load,

$$P_1 = 1.5\, P_2 = 1.5 \times 106\,040 = 159\,060 \text{ N}$$

We know that the resultant load (or the maximum load) on the cylinder head,

$$P_{max} = P_1 + K.P_2 = 159\,060 + 0.5 \times 106\,040 = 212\,080 \text{ N}$$

This load is shared by 8 bolts, therefore maximum load on each bolt,

$$P_{max} = 212\,080 / 8 = 26\,510 \text{ N}$$

and minimum load on each bolt,

$$P_{min} = P_1 / n = 159\,060/8 = 19\,882 \text{ N}$$

We know that mean or average load on the bolt,

$$P_m = \frac{P_{max} + P_{min}}{2} = \frac{26\,510 + 19\,882}{2} = 23\,196 \text{ N}$$

and the variable load on the bolt,

$$P_v = \frac{P_{max} - P_{min}}{2} = \frac{26\,510 - 19\,882}{2} = 3314 \text{ N}$$

Let d_c = Core diameter of the bolt in mm.

∴ Stress area of the bolt,

$$A_s = \frac{\pi}{4}(d_c)^2 = 0.7854\,(d_c)^2 \text{ mm}^2$$

We know that mean or average stress on the bolt,

$$\sigma_m = \frac{P_m}{A_s} = \frac{23\,196}{0.7854\,(d_c)^2} = \frac{29\,534}{(d_c)^2} \text{ N/mm}^2$$

and variable stress on the bolt,

$$\sigma_v = \frac{P_v}{A_s} = \frac{3314}{0.7854\,(d_c)^2} = \frac{4220}{(d_c)^2} \text{ N/mm}^2$$

According to *Soderberg's formula, the variable stress,

$$\sigma_v = \sigma_e \left(\frac{1}{F.S} - \frac{\sigma_m}{\sigma_y}\right)$$

$$\frac{4220}{(d_c)^2} = 240\left(\frac{1}{2} - \frac{29\,534}{(d_c)^2\ 330}\right) = 120 - \frac{21\,480}{(d_c)^2}$$

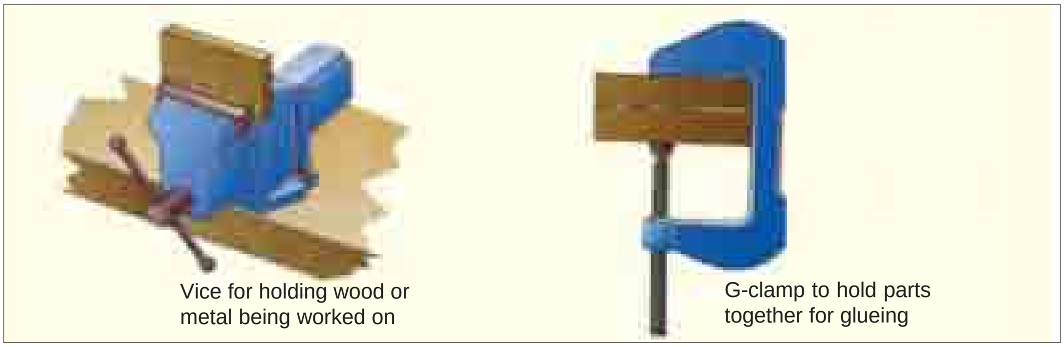
or $$\frac{4220}{(d_c)^2} + \frac{21\,480}{(d_c)^2} = 120 \quad \text{or} \quad \frac{25\,700}{(d_c)^2} = 120$$

$$\therefore \quad (d_c)^2 = 25\,700 / 120 = 214 \quad \text{or} \quad d_c = 14.6 \text{ mm}$$

From Table 11.1 (coarse series), the standard core diameter is d_c = 14.933 mm and the corresponding size of the bolt is M18. **Ans.**

11.15 Boiler Stays

In steam boilers, flat or slightly curved plates are supported by stays. The stays are used in order to increase strength and stiffness of the plate and to reduce distortion. The principal types of stays are:



Simple machine tools.

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

* See Chapter 6, Art. 6.20.

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1. ***Direct stays.*** These stays are usually screwed round bars placed at right angles to the plates supported by them.

2. ***Diagonal and gusset stays.*** These stays are used for supporting one plate by trying it to another at right angles to it.

3. ***Girder stays.*** These stays are placed edgewise on the plate to be supported and bolted to it at intervals.

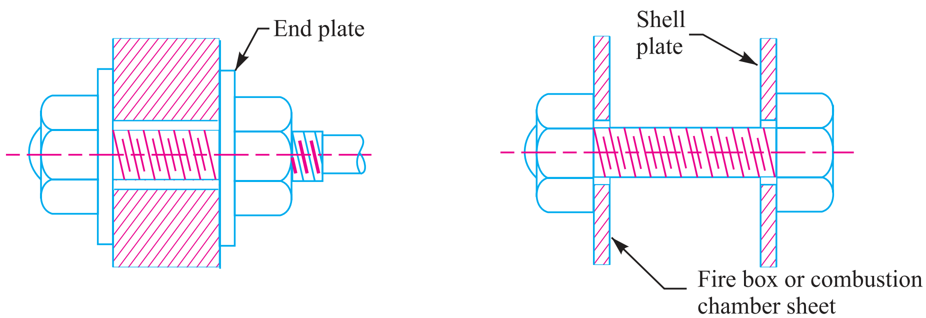

Fig. 11.28. Boiler stays.

Here we are mainly concerned with the direct stays. The direct stays may be ***bar stays*** or ***screwed stays***. A bar stay for supporting one end plate of a boiler shell from the other end plate is shown in Fig. 11.28 (*a*). The ends of the bar are screwed to receive two nuts between which the end plate is locked. The bar stays are not screwed into the plates.

The fire boxes or combustion chambers of locomotive and marine boilers are supported by screwed stays as shown in Fig. 11.28 (*b*). These stays are called screwed stays, because they are screwed into the plates which they support. The size of the bar or screwed stays may be obtained as discussed below :

Consider a short boiler having longitudinal bar stays as shown in Fig. 11.29.

Let p = Pressure of steam in a boiler,

x = Pitch of the stays,

A = Area of the plate supported by each stay = $x \times x = x^2$

d_c = Core diameter of the stays, and

σ_t = Permissible tensile stress for the material of the stays.

We know that force acting on the stay,

$$P = \text{Pressure} \times \text{Area} = p.A = p.x^2$$

Knowing the force P, we may determine the core diameter of the stays by using the following relation,

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

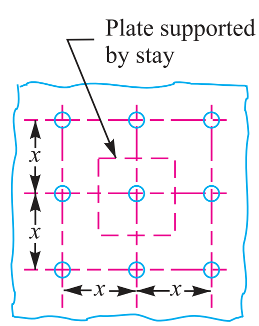

Fig. 11.29. Longitudinal bar stay.

From the core diameter, the standard size of the stay may be fixed from Table 11.1.

Example 11.10. *The longitudinal bar stays of a short boiler are pitched at 350 mm horizontally and vertically as shown in Fig. 11.29. The steam pressure is 0.84 N/mm². Find the size of mild steel bolts having tensile stress as 56 MPa.*

Solution. Given : p = 0.84 N/mm² ; σ_t = 56 MPa = 56 N/mm²

Since the pitch of the stays is 350 mm, therefore area of the plate supported by each stay,

$$A = 350 \times 350 = 122\,500 \text{ mm}^2$$

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We know that force acting on each stay,

$$P = A \times p = 122\,500 \times 0.84 = 102\,900 \text{ N}$$

Let d_c = Core diameter of the bolts.

We know that the resisting force on the bolts (P),

$$102\,900 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2\, 56 = 44\,(d_c)^2$$

$$\therefore \quad (d_c)^2 = 102\,900 / 44 = 2340 \quad \text{or} \quad d_c = 48.36 \text{ mm}$$

From Table 11.1 (coarse series), the standard core diameter is 49.177 mm. Therefore size of the bolt corresponding to 49.177 mm is M 56. **Ans.**

11.16 Bolts of Uniform Strength

When a bolt is subjected to shock loading, as in case of a cylinder head bolt of an internal combustion engine, the resilience of the bolt should be considered in order to prevent breakage at the thread. In an ordinary bolt shown in Fig. 11.30 (*a*), the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt *i.e.* the cross-sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

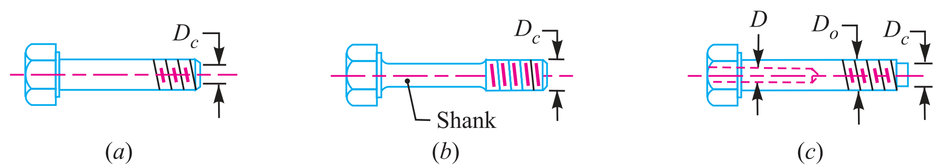

Fig. 11.30. Bolts of uniform strength.

If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (D_c) as shown in Fig. 11.30 (*b*), then shank of the bolt will undergo a higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us ***bolts of uniform strength***. The resilience of a bolt may also be increased by increasing its length.

A second alternative method of obtaining the bolts of uniform strength is shown in Fig. 11.30 (*c*). In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

Let D = Diameter of the hole.

D_o = Outer diameter of the thread, and

D_c = Root or core diameter of the thread.

$$\therefore \quad \frac{\pi}{4} D^2 = \frac{\pi}{4}\left[(D_o)^2 - (D_c)^2\right]$$

or

$$D^2 = (D_o)^2 - (D_c)^2$$

$$\therefore \quad D = \sqrt{(D_o)^2 - (D_c)^2}$$

Example 11.11. *Determine the diameter of the hole that must be drilled in a M 48 bolt such that the bolt becomes of uniform strength.*

Solution. Given : D_o = 48 mm

From Table 11.1 (coarse series), we find that the core diameter of the thread (corresponding to D_o = 48 mm) is D_c = 41.795 mm.

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We know that for bolts of uniform strength, the diameter of the hole,

$$D = \sqrt{(D_o)^2 - (D_c)^2} = \sqrt{(48)^2 - (41.795)^2} = 23.64 \text{ mm } \textbf{Ans.}$$

11.17 Design of a Nut

When a bolt and nut is made of mild steel, then the effective height of nut is made equal to the nominal diameter of the bolt. If the nut is made of weaker material than the bolt, then the height of nut should be larger, such as 1.5 *d* for gun metal, 2 *d* for cast iron and 2.5 *d* for aluminium alloys (where *d* is the nominal diameter of the bolt). In case cast iron or aluminium nut is used, then *V*-threads are permissible only for permanent fastenings, because threads in these materials are damaged due to repeated screwing and unscrewing. When these materials are to be used for parts frequently removed and fastened, a screw in steel bushing for cast iron and cast-in-bronze or monel metal insert should be used for aluminium and should be drilled and tapped in place.

11.18 Bolted Joints under Eccentric Loading

There are many applications of the bolted joints which are subjected to eccentric loading such as a wall bracket, pillar crane, etc. The eccentric load may be

1. Parallel to the axis of the bolts,

2. Perpendicular to the axis of the bolts, and

3. In the plane containing the bolts.

We shall now discuss the above cases, in detail, in the following articles.

11.19 Eccentric Load Acting Parallel to the Axis of Bolts

Consider a bracket having a rectangular base bolted to a wall by means of four bolts as shown in Fig. 11.31. A little consideration will show that each bolt is subjected to a direct tensile load of

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

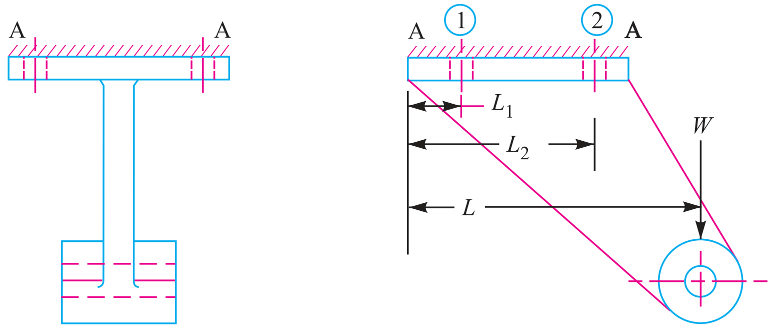

Fig. 11.31. Eccentric load acting parallel to the axis of bolts.

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.

* We know that elongation is proportional to strain which in turn is proportional to stress within elastic limits.

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∴ Load on each bolt at distance L_1,

$$W_1 = w.L_1$$

and moment of this load about the tilting edge

$$= w_1.L_1 \times L_1 = w\,(L_1)^2$$

Similarly, load on each bolt at distance L_2,

$$W_2 = w.L_2$$

and moment of this load about the tilting edge

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

∴ Total moment of the load on the bolts about the tilting edge

$$= 2w\,(L_1)^2 + 2w\,(L_2)^2 \qquad ...(i)$$

... (∵ There are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

$$= W.L \qquad ...(ii)$$

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

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

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance L_2 are heavily loaded.

∴ Tensile load on each bolt at distance L_2,

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

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

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

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

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

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

Example 11.12. *A bracket, as shown in Fig. 11.31, 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 $\quad d_c$ = Core diameter of the bolts.

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

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

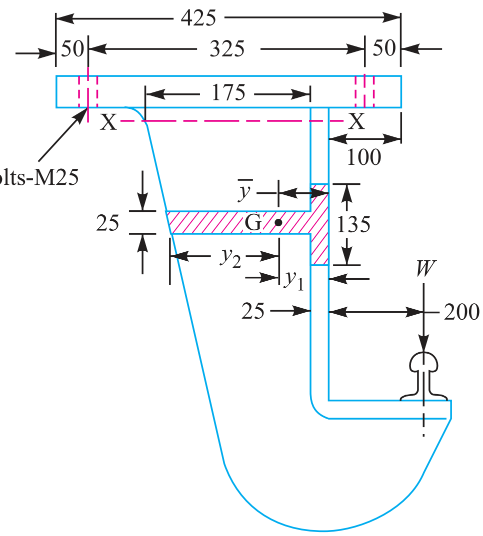
$$\therefore \quad (d_c)^2 = 34\ 750 / 47 = 740$$

or $\quad d_c$ = 27.2 mm

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 28.706 mm and the corresponding size of the bolt is M 33. **Ans.**

Example 11.13. *A crane runway bracket is shown in Fig. 11.32. Determine the tensile and compressive stresses produced in the section X-X when the magnitude of the wheel load is 15 kN.*

Also find the maximum stress produced in the bolts used for fastening the bracket to the roof truss.



All dimensions in mm

Fig. 11.32

Solution. Given : $W = 15\text{ kN} = 15 \times 10^3$ N

First of all, let us find the distance of centre of gravity of the section at X–X.

Let $\quad \bar{y}$ = Distance of centre of gravity (G) from the top of the flange.

$$\therefore \quad \bar{y} = \frac{135 \times 25 \times \dfrac{25}{2} + 175 \times 25\left(25 + \dfrac{175}{2}\right)}{135 \times 25 + 175 \times 25} = 69 \text{ mm}$$

Moment of inertia about an axis passing through the centre of gravity of the section,

$$I_{GG} = \left[\frac{135\,(25)^3}{12} + 135 \times 25\left(69 - \frac{25}{2}\right)^2\right] + \left[\frac{25\,(175)^3}{12} + 175 \times 25\left(200 - 69 - \frac{175}{2}\right)^2\right]$$

$$= 30.4 \times 10^6 \text{ mm}^4$$

Distance of C.G. from the top of the flange,

$$y_1 = \bar{y} = 69 \text{ mm}$$

and distance of C.G. from the bottom of the web,

$$y_2 = 175 + 25 - 69 = 131 \text{ mm}$$

Due to the tilting action of the load W, the cross-section of the bracket X-X will be under bending stress. The upper fibres of the top flange will be under maximum tension and the lower fibres of the web will be under maximum compression.

$\therefore$ Section modulus for the maximum tensile stress,

$$Z_1 = \frac{I_{GG}}{y_1} = \frac{30.4 \times 10^6}{69} = 440.6 \times 10^3 \text{ mm}^3$$

and section modulus for the maximum compressive stress,

$$Z_2 = \frac{I_{GG}}{y_2} = \frac{30.4 \times 10^6}{131} = 232 \times 10^3 \text{ mm}^3$$

We know that bending moment exerted on the section,

$$M = 15 \times 10^3 (200 + 69) = 4035 \times 10^3 \text{ N-mm}$$

∴ Maximum bending stress (tensile) in the flange,

$$\sigma_{b1} = \frac{M}{Z_1} = \frac{4035 \times 10^3}{440.6 \times 10^3} = 9.16 \text{ N/mm}^2$$

and maximum bending stress (compressive) in the web,

$$\sigma_{b2} = \frac{M}{Z_2} = \frac{4035 \times 10^3}{232 \times 10^3} = 17.4 \text{ N/mm}^2$$

The eccentric load also induces direct tensile stress in the bracket. We know that direct tensile stress,

$$\sigma_{t1} = \frac{\text{Load}}{\text{Cross-sectional area of the bracket at } X - X}$$

$$= \frac{15 \times 10^3}{135 \times 25 + 175 \times 25} = 1.94 \text{ N/mm}^2$$

∴ Maximum tensile stress produced in the section at X–X (*i.e.* in the flange),

$$\sigma_t = \sigma_{b1} + \sigma_{t1} = 9.16 + 1.94 = 11.1 \text{ N/mm}^2 = 11.1 \text{ MPa} \textbf{ Ans.}$$

and maximum compressive stress produced in the section at X–X (*i.e.* in the web),

$$\sigma_c = \sigma_{b2} - \sigma_{t1} = 17.4 - 1.94 = 15.46 \text{ N/mm}^2 = 15.46 \text{ MPa} \textbf{ Ans.}$$

Let σ_{tb} = Maximum stress produced in bolts,

n = Number of bolts = 4, and ...(Given)

d = Major diameter of the bolts = 25 mm ...(Given)

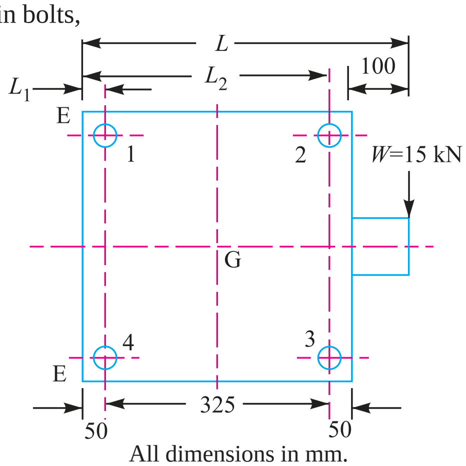


Fig. 11.33

The plan of the bracket is shown in Fig. 11.33. Due to the eccentric load W, the bracket has a tendency to tilt about the edge EE. Since the load is acting parallel to the axis of bolts, therefore direct tensile load on each bolt,

$$W_{t1} = \frac{W}{n} = \frac{15 \times 10^3}{4} = 3750 \text{ N}$$

Let w = Load in each bolt per mm distance from the edge EE due to the turning effect of the bracket,

L_1 = Distance of bolts 1 and 4 from the tilting edge EE = 50 mm, and

L_2 = Distance of bolts 2 and 3 from the tilting edge EE = 50 + 325 = 375 mm

We know that

$$w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]} = \frac{15 \times 10^3 (100 + 50 + 325 + 50)}{2[(50)^2 + (375)^2]} = 27.5 \text{ N/mm}$$

Since the heavily loaded bolts are those which lie at greater distance from the tilting edge, therefore the bolts 2 and 3 will be heavily loaded.

∴ Maximum tensile load on each of bolts 2 and 3,

$$W_{t2} = w \times L_2 = 27.5 \times 375 = 10\ 312 \text{ N}$$

and the total tensile load on each of the bolts 2 and 3,

$$W_t = W_{t1} + W_{t2} = 3750 + 10\ 312 = 14\ 062 \text{ N}$$

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

$$14\ 062 = \frac{\pi}{4}(d_c)^2 \sigma_{tb} = \frac{\pi}{4}(0.84 \times 25)^2 \sigma_{tb} = 346.4\ \sigma_{tb}$$

... (Taking, $d_c = 0.84\ d$)

∴ $$\sigma_{tb} = 14\ 062/346.4 = 40.6 \text{ N/mm}^2 = 40.6 \text{ MPa}$$ **Ans.**

11.20 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 in Fig. 11.34.

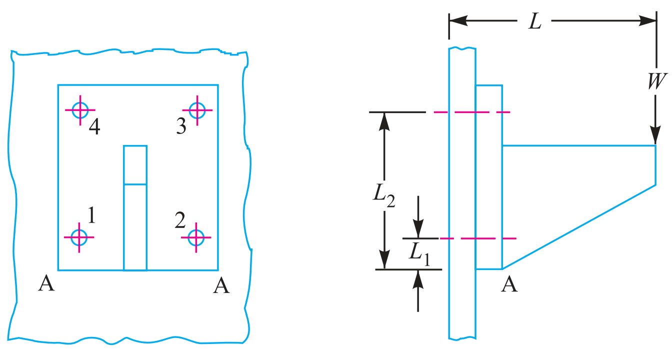


Fig. 11.34. Eccentric load perpendicular to the axis of bolts.

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,

$$W_s = W/n, \text{ where } n \text{ is number of bolts.}$$

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,

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

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.

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Example 11.14. *For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in Fig. 11.35. 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.*

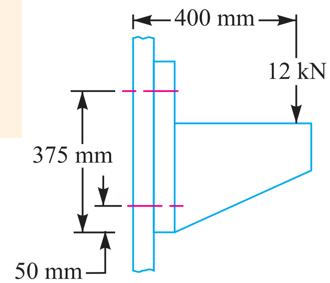

Fig. 11.35

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. 11.34), 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,

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

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

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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),

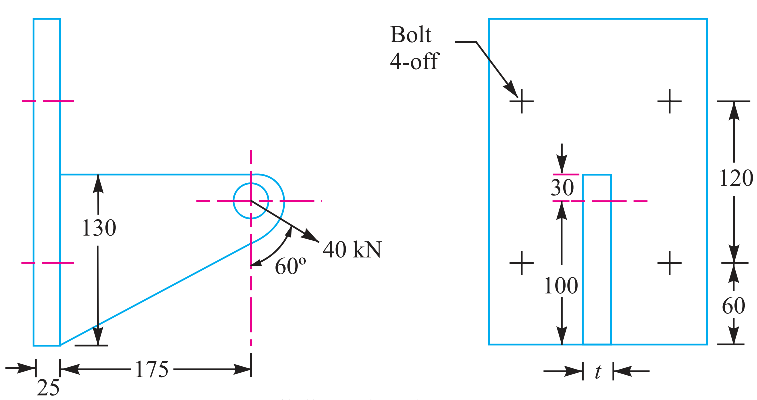
$$84 = \frac{M}{Z} = \frac{4.8 \times 10^6 \times 6}{t.b^2} = \frac{28.8 \times 10^6}{t.b^2}$$

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

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

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

Example 11.15. *Determine the size of the bolts and the thickness of the arm for the bracket as shown in Fig. 11.36, if it carries a load of 40 kN at an angle of 60° to the vertical.*



All dimensions in mm.

Fig 11.36

The material of the bracket and the bolts is same for which the safe stresses can be assumed as 70, 50 and 105 MPa in tension, shear and compression respectively.

Solution. Given : W = 40 kN = 40×10^3 N ; σ_t = 70 MPa = 70N/mm^2 ; τ = 50 MPa = 50 N/mm^2; σ_c = 105 MPa = 105 N/mm^2

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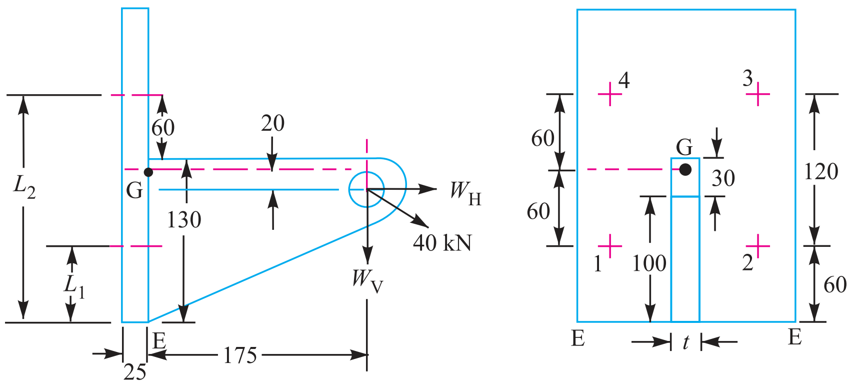
Since the load $W = 40$ kN is inclined at an angle of 60° to the vertical, therefore resolving it into horizontal and vertical components. We know that horizontal component of 40 kN,

$$W_H = 40 \times \sin 60° = 40 \times 0.866 = 34.64 \text{ kN} = 34\,640 \text{ N}$$

and vertical component of 40 kN,

$$W_V = 40 \times \cos 60° = 40 \times 0.5 = 20 \text{ kN} = 20\,000 \text{ N}$$

Due to the horizontal component (W_H), which acts parallel to the axis of the bolts as shown in Fig. 11.37, the following two effects are produced :



All dimensions in mm.

Fig. 11.37

1. A direct tensile load equally shared by all the four bolts, and
2. A turning moment about the centre of gravity of the bolts, in the anticlockwise direction.

∴ Direct tensile load on each bolt,

$$W_{t1} = \frac{W_H}{4} = \frac{34\,640}{4} = 8660 \text{ N}$$

Since the centre of gravity of all the four bolts lies in the centre at G (because of symmetrical bolts), therefore the turning moment is in the anticlockwise direction. From the geometry of the Fig. 11.37, we find that the distance of horizontal component from the centre of gravity (G) of the bolts

$$= 60 + 60 - 100 = 20 \text{ mm}$$

∴ Turning moment due to W_H about G,

$$T_H = W_H \times 20 = 34\,640 \times 20 = 692.8 \times 10^3 \text{ N-mm} \quad \text{...(Anticlockwise)}$$

Due to the vertical component W_V, which acts perpendicular to the axis of the bolts as shown in Fig. 11.37, the following two effects are produced:

1. A direct shear load equally shared by all the four bolts, and
2. A turning moment about the edge of the bracket in the clockwise direction.

∴ Direct shear load on each bolt,

$$W_s = \frac{W_V}{4} = \frac{20\,000}{4} = 5000 \text{ N}$$

Distance of vertical component from the edge E of the bracket,

$$= 175 \text{ mm}$$

∴ Turning moment due to W_V about the edge of the bracket,

$$T_V = W_V \times 175 = 20\,000 \times 175 = 3500 \times 10^3 \text{ N-mm} \quad \text{(Clockwise)}$$

From above, we see that the clockwise moment is greater than the anticlockwise moment, therefore,

Net turning moment $= 3500 \times 10^3 - 692.8 \times 10^3 = 2807.2 \times 10^3$ N-mm (Clockwise) ...(*i*)

Due to this clockwise moment, the bracket tends to tilt about the lower edge *E*.

Let w = Load on each bolt per mm distance from the edge *E* due to the turning effect of the bracket,

L_1 = Distance of bolts 1 and 2 from the tilting edge *E* = 60 mm, and

L_2 = Distance of bolts 3 and 4 from the tilting edge *E*

$= 60 + 120 = 180$ mm

∴ Total moment of the load on the bolts about the tilting edge *E*

$$= 2\,(w.L_1)\,L_1 + 2\,(w.L_2)\,L_2$$

... (∵ There are two bolts each at distance L_1 and L_2.)

$$= 2w\,(L_1)^2 + 2w(L_2)^2 = 2w\,(60)^2 + 2w(180)^2$$

$$= 72\,000\ w \text{ N-mm} \qquad \text{...(ii)}$$

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

$$w = 2807.2 \times 10^3 / 72\,000 = 39 \text{ N/mm}$$

Since the heavily loaded bolts are those which lie at a greater distance from the tilting edge, therefore the upper bolts 3 and 4 will be heavily loaded. Thus the diameter of the bolt should be based on the load on the upper bolts. We know that the maximum tensile load on each upper bolt,

$$W_{t2} = w.L_2 = 39 \times 180 = 7020 \text{ N}$$

∴ Total tensile load on each of the upper bolt,

$$W_t = W_{t1} + W_{t2} = 8660 + 7020 = 15\,680 \text{ N}$$

Since each upper bolt is subjected to a tensile load (W_t = 15 680 N) and a shear load (W_s = 5000 N), therefore equivalent tensile load,

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

$$= \frac{1}{2}\left[15\,680 + \sqrt{(15\,680)^2 + 4(5000)^2}\right] \text{N}$$

$$= \frac{1}{2}\left[15\,680 + 18\,600\right] = 17\,140 \text{ N} \qquad \text{...(iii)}$$

Size of the bolts

Let d_c = Core diameter of the bolts.

We know that tensile load on each bolt

$$= \frac{\pi}{2}(d_c)^2\,\sigma_t = \frac{\pi}{4}(d_c)^2\,70 = 55\,(d_c)^2 \text{ N} \qquad \text{...(iv)}$$

From equations (*iii*) and (*iv*), we get

$$(d_c)^2 = 17\,140 / 55 = 311.64 \quad \text{or} \quad d_c = 17.65 \text{ mm}$$

From Table 11.1 (coarse series), we find that the standard core diameter is 18.933 mm and corresponding size of the bolt is M 22. **Ans.**

Thickness of the arm of the bracket

Let t = Thickness of the arm of the bracket in mm, and

b = Depth of the arm of the bracket = 130 mm ...(Given)

We know that cross-sectional area of the arm,

$$A = b \times t = 130\, t \text{ mm}^2$$

and section modulus of the arm,

$$Z = \frac{1}{6}\, t\, (b)^2 = \frac{1}{6} \times t\, (130)^2 = 2817\, t \text{ mm}^3$$

Due to the horizontal component W_H, the following two stresses are induced in the arm :

1. Direct tensile stress,

$$\sigma_{t1} = \frac{W_H}{A} = \frac{34\,640}{130\, t} = \frac{266.5}{t} \text{ N/mm}^2$$

2. Bending stress causing tensile in the upper most fibres of the arm and compressive in the lower most fibres of the arm. We know that the bending moment of W_H about the centre of gravity of the arm,

$$M_H = W_H \left(100 - \frac{130}{2}\right) = 34\,640 \times 35 = 1212.4 \times 10^3 \text{ N-mm}$$

∴ Bending stress, $\sigma_{t2} = \dfrac{M_H}{Z} = \dfrac{1212.4 \times 10^3}{2817\, t} = \dfrac{430{\cdot}4}{t}$ N/mm²

Due to the vertical component W_V, the following two stresses are induced in the arm :

1. Direct shear stress,

$$\tau = \frac{W_V}{A} = \frac{20\,000}{130\, t} = \frac{154}{t} \text{ N/mm}^2$$

2. Bending stress causing tensile stress in the upper most fibres of the arm and compressive in the lower most fibres of the arm.

Assuming that the arm extends upto the plate used for fixing the bracket to the structure. This assumption gives stronger section for the arm of the bracket.

∴ Bending moment due to W_V,

$$M_V = W_V\, (175 + 25) = 20\,000 \times 200 = 4 \times 10^6 \text{ N-mm}$$

and bending stress, $\sigma_{t3} = \dfrac{M_V}{Z} = \dfrac{4 \times 10^6}{2817\, t} = \dfrac{1420}{t}$ N/mm²

Net tensile stress induced in the upper most fibres of the arm of the bracket,

$$\sigma_t = \sigma_{t1} + \sigma_{t2} + \sigma_{t3} = \frac{266.5}{t} + \frac{430.4}{t} + \frac{1420}{t} = \frac{2116.9}{t} \text{ N/mm}^2 \quad \text{...}(v)$$

We know that maximum tensile stress $[\sigma_{t(max)}]$,

$$70 = \frac{1}{2}\sigma_t + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2}$$

$$= \frac{1}{2} \times \frac{2116.9}{t} + \frac{1}{2}\sqrt{\left(\frac{2116.9}{t}\right)^2 + 4\left(\frac{154}{t}\right)^2}$$

$$= \frac{1058.45}{t} + \frac{1069.6}{t} = \frac{2128.05}{t}$$

∴ $t = 2128.05 / 70 = 30.4$ say 31 mm **Ans.**

Let us now check the shear stress induced in the arm. We know that maximum shear stress,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2} = \frac{1}{2}\sqrt{\left(\frac{2116.9}{t}\right)^2 + 4\left(\frac{154}{t}\right)^2}$$

$$= \frac{1069.6}{t} = \frac{1069.6}{31} = 34.5 \text{ N/mm}^2 = 34.5 \text{ MPa}$$

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Since the induced shear stress is less than the permissible stress (50 MPa), therefore the design is safe.

Notes : 1. The value of '*t*' may be obtained as discussed below :

Since the shear stress at the upper most fibres of the arm of the bracket is zero, therefore equating equation (*v*) to the given safe tensile stress (*i.e.* 70 MPa), we have

$$\frac{2116.9}{t} = 70 \quad \text{or} \quad t = 2116.9 / 70 = 30.2 \text{ say } 31 \text{ mm } \textbf{Ans.}$$

2. If the compressive stress in the lower most fibres of the arm is taken into consideration, then the net compressive stress induced in the lower most fibres of the arm,

$$\sigma_c = \sigma_{c1} + \sigma_{c2} + \sigma_{c3}$$

$$= -\sigma_{t1} + \sigma_{t2} + \sigma_{t3}$$

... ($\because$ The magnitude of tensile and compressive stresses is same.)

$$= -\frac{266.5}{t} + \frac{430.4}{t} + \frac{1420}{t} = \frac{1583.9}{t} \text{ N/mm}^2$$

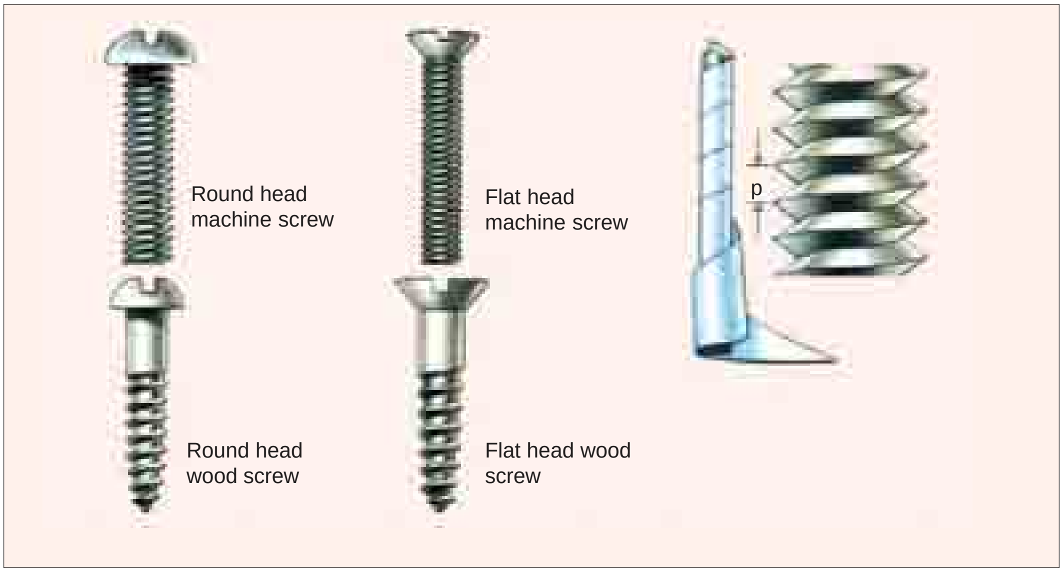
Since the safe compressive stress is 105 N/mm^2, therefore

$$105 = \frac{1583.9}{t} \quad \text{or} \quad t = 1583.9 / 105 = 15.1 \text{ mm}$$

This value of thickness is low as compared to 31 mm as calculated above. Since the higher value is taken, therefore

$$t = 31 \text{ mm } \textbf{Ans.}$$

Example 11.16. *An offset bracket, having arm of I-cross-section is fixed to a vertical steel column by means of four standard bolts as shown in Fig. 11.38. An inclined pull of 10 kN is acting on the bracket at an angle of 60° to the vertical.*



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

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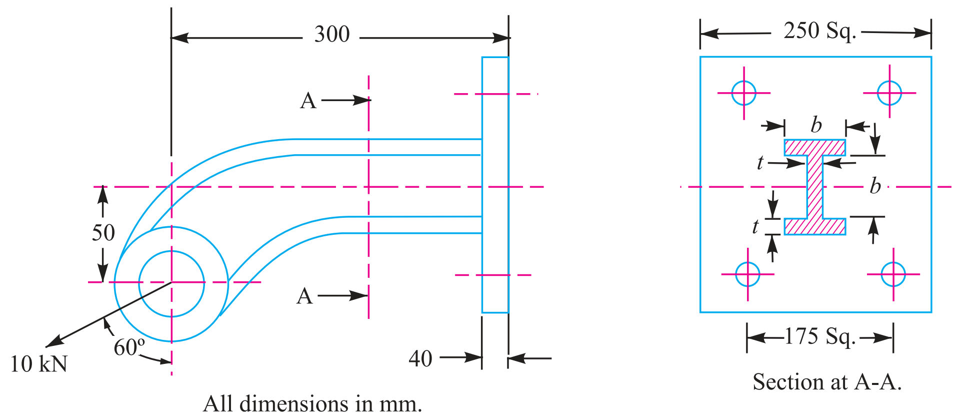

Fig. 11.38

Determine : (a) the diameter of the fixing bolts, and (b) the dimensions of the arm of the bracket if the ratio between b and t is 3 : 1.

For all parts, assume safe working stresses of 100 MPa in tension and 60 MPa in shear.

Solution. Given : $W = 10$ kN ; $\theta = 60°$; $\sigma_1 = 100$ MPa $= 100$ N/mm^2 ; $\tau = 60$ MPa $= 60$ N/mm^2

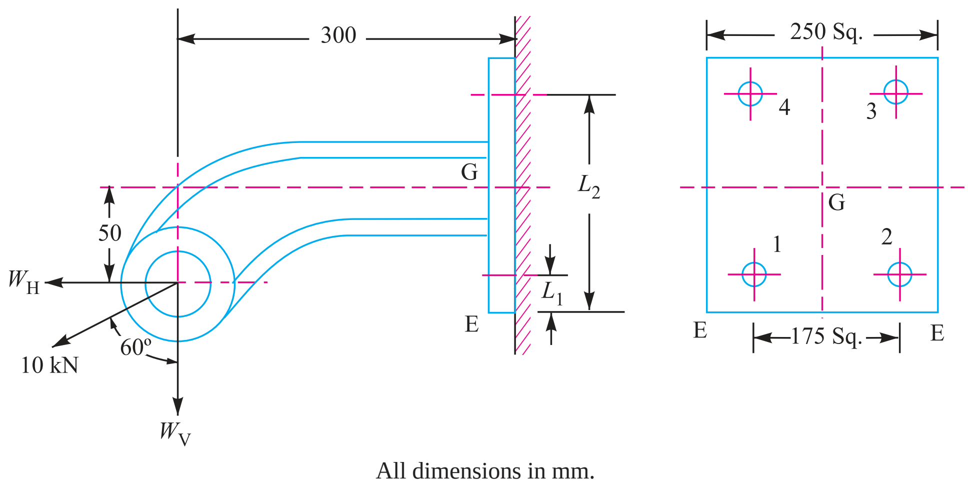

Fig. 11.39

Resolving the pull acting on the bracket (*i.e.* 10 kN) into horizontal and vertical components, we have

Horizontal component of 10 kN,

$$W_H = 10 \times \sin 60° = 10 \times 0.866 = 8.66 \text{ kN} = 8660 \text{ N}$$

and vertical component of 10 kN,

$$W_V = 10 \cos 60° = 10 \times 0.5 = 5 \text{ kN} = 5000 \text{ N}$$

Due to the horizontal component (W_H), which acts parallel to the axis of the bolts, as shown in Fig. 11.39, the following two effects are produced :

1. A direct tensile load equally shared by all the four bolts, and
2. A turning moment about the centre of gravity of the bolts. Since the centre of gravity of all the four bolts lie in the centre at *G* (because of symmetrical bolts), therefore the turning moment is in the clockwise direction.

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∴ Direct tensile load on each bolt,

$$W_{t1} = \frac{W_H}{4} = \frac{8660}{4} = 2165 \text{ N}$$

Distance of horizontal component from the centre of gravity (*G*) of the bolts

$$= 50 \text{ mm} = 0.05 \text{ m}$$

∴ Turning moment due to W_H about *G*,

$$T_H = W_H \times 0.05 = 8660 \times 0.05 = 433 \text{ N-m (Clockwise)}$$

Due to the vertical component (W_V), which acts perpendicular to the axis of the bolts, as shown in Fig. 11.39, the following two effects are produced :

1. A direct shear load equally shared by all the four bolts, and

2. A turning moment about the edge of the bracket, in the anticlockwise direction.

∴ Direct shear load on each bolt,

$$W_s = \frac{W_V}{4} = \frac{5000}{4} = 1250 \text{ N}$$

Distance of vertical component from the edge of the bracket

$$= 300 \text{ mm} = 0.3 \text{ m}$$

∴ Turning moment about the edge of the bracket,

$$T_V = W_V \times 0.3 = 5000 \times 0.3 = 1500 \text{ N-m (Anticlockwise)}$$

From above, we see that the anticlockwise moment is greater than the clockwise moment, therefore

Net turning moment

$$= 1500 - 433 = 1067 \text{ N-m (Anticlockwise)} \qquad ...(i)$$

Due to this anticlockwise moment, the bracket tends to tilt about the edge *E*.

Let w = Load in each bolt per metre distance from the edge *E*, due to the turning effect of the bracket,

L_1 = Distance of bolts 1 and 2 from the tilting edge *E*

$$= \frac{250 - 175}{2} = 37.5 \text{ mm} = 0.0375 \text{ m}$$

L_3 = Distance of bolts 3 and 4 from the tilting edge

$$= L_1 + 175 \text{ mm} = 37.5 + 175 = 212.5 \text{ mm} = 0.2125 \text{ m}$$

∴ Total moment of the load on the bolts about the tilting edge *E*

$$= 2\,(w.L_1)\,L_1 + 2\,(w.L_2)\,L_2 = 2w\,(L_1)^2 + 2w\,(L_2)^2$$

...(∵ There are two bolts each at distance L_1 and L_2.)

$$= 2w\,(0.0375)^2 + 2w\,(0.2125)^2 = 0.093\,w \text{ N-m} \qquad ...(ii)$$

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

$$w = 1067 / 0.093 = 11\,470 \text{ N/m}$$

Since the heavily loaded bolts are those which lie at a greater distance from the tilting edge, therefore the upper bolts 3 and 4 will be heavily loaded.

∴ Maximum tensile load on each upper bolt,

$$W_{t2} = w.L_2 = 11\,470 \times 0.2125 = 2435 \text{ N}$$

and total tensile load on each of the upper bolt,

$$W_t = W_{t1} + W_{t2} = 2165 + 2435 = 4600 \text{ N}$$

Since each upper bolt is subjected to a total tensile load (W_t = 4600 N) and a shear load (W_s = 1250 N), therefore equivalent tensile load,

$$W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right] = \frac{1}{2}\left[4600 + \sqrt{(4600)^2 + 4(1250)^2}\right]$$

$$= \frac{1}{2}(4600 + 5240) = 4920 \text{ N}$$

(a) Diameter of the fixing bolts

Let d_c = Core diameter of the fixing bolts.

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

$$4920 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2 100 = 78.55 (d_c)^2$$

$\therefore$ $(d_c)^2 = 4920 / 78.55 = 62.6$ or $d_c = 7.9$ mm

From Table 11.1 (coarse series), we find that standard core diameter is 8.18 mm and the corresponding size of the bolt is M 10. **Ans.**

Dimensions of the arm of the bracket

Let t = Thickness of the flanges and web in mm, and

b = Width of the flanges in mm = $3t$ ... (Given)

$\therefore$ Cross-sectional area of the *I*-section of the arms,

$$A = 3\, b.t = 3 \times 3\, t \times t = 9\, t^2 \text{ mm}^2$$

and moment of inertia of the *I*-section of the arm about an axis passing through the centre of gravity of the arm,

$$I = \frac{b\,(2t + b)^3}{12} - \frac{(b - t)\, b^3}{12}$$

$$= \frac{3t\,(2t + 3t)^3}{12} - \frac{(3t - t)\,(3t)^3}{12} = \frac{375\, t^4}{12} - \frac{54\, t^4}{12} = \frac{321\, t^4}{12}$$

$\therefore$ Section modulus of *I*-section of the arm,

$$Z = \frac{I}{t + b/2} = \frac{321\, t^4}{12\,(t + 3t/2)} = 10.7\, t^3 \text{ mm}^3$$

Due to the horizontal component W_H, the following two stresses are induced in the arm:

1. Direct tensile stress,

$$\sigma_{t1} = \frac{W_H}{A} = \frac{8660}{9t^2} = \frac{962}{t^2} \text{ N/mm}^2$$

2. Bending stress causing tensile in the lower most fibres of the bottom flange and compressive in the upper most fibres of the top flange.

We know that bending moment of W_H about the centre of gravity of the arm,

$$M_H = W_H \times 0.05 = 8660 \times 0.05 = 433 \text{ N-m} = 433 \times 10^3 \text{ N-mm}$$

$\therefore$ Bending stress,

$$\sigma_{t2} = \frac{M_H}{Z} = \frac{433 \times 10^3}{10.7\, t^3} = \frac{40.5 \times 10^3}{t^3} \text{ N/mm}^2$$

Due to the vertical component W_V, the following two stresses are induced the arm:

1. Direct shear stress,

$$\tau = \frac{W_V}{A} = \frac{5000}{9t^2} = 556 \text{ N/mm}^2$$

2. Bending stress causing tensile in the upper most fibres of the top flange and compressive in lower most fibres of the bottom flange.

Assuming that the arm extends upto the plate used for fixing the bracket to the structure.

We know that bending moment due to W_V,

$$M_V = W_V \times 0.3 = 5000 \times 0.3 = 1500 \text{ N-m} = 1500 \times 10^3 \text{ N-mm}$$

∴ Bending stress,

$$\sigma_{t3} = \frac{M_V}{Z} = \frac{1500 \times 10^3}{10.7\, t^3} = \frac{140.2 \times 10^3}{t^3} \text{ N/mm}^2$$

Considering the upper most fibres of the top flange.

Net tensile stress induced in the arm of the bracket

$$= \sigma_{t1} - \sigma_{t2} + \sigma_{t3}$$

$$= \frac{962}{t^2} - \frac{40.5 \times 10^3}{t^3} + \frac{140.2 \times 10^3}{t^3}$$

$$= \frac{962}{t^2} + \frac{99.7 \times 10^3}{t^3}$$

Since the shear stress at the top most fibres is zero, therefore equating the above expression, equal to the given safe tensile stress of 100 N/mm^2, we have

$$\frac{962}{t^2} + \frac{99.7 \times 10^3}{t^3} = 100$$

By hit and trial method, we find that

$$t = 10.4 \text{ mm } \textbf{Ans.}$$

and $$b = 3\,t = 3 \times 10.4 = 31.2 \text{ mm } \textbf{Ans.}$$

Retaining screws on a lamp.

11.21 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. 11.40.

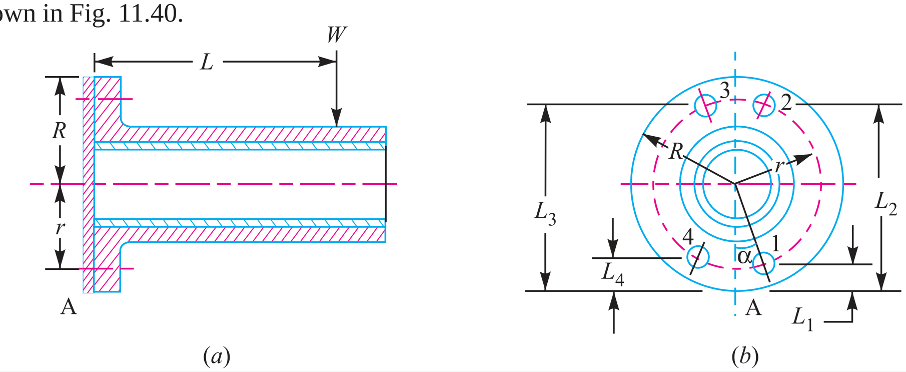

Fig. 11.40. Eccentric load on a bracket with circular base.

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 centres 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,

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

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

Now from the geometry of the Fig. 11.40 (*b*), we find that

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

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

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

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

$\therefore$ Load in the bolt situated at 1 $= w.L_1 = \dfrac{W.L.L_1}{4\,R^2 + 2\,r^2} = \dfrac{W.L\,(R - r\cos\alpha)}{4\,R^2 + 2\,r^2}$

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

$\therefore$ Maximum load in a bolt

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

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,

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

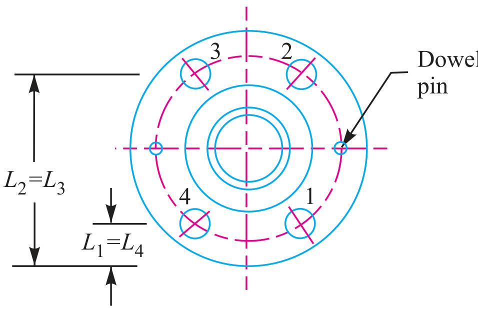

Fig. 11.41

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. 11.41. In such a case, maximum load is given by

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

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

Note : Generally, two dowel pins as shown in Fig. 11.41, 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.

Example 11.17. *The base of a pillar crane is fastened to the foundation (a level plane) by eight bolts spaced equally on a bolt circle of diameter 1.6 m. The diameter of the pillar base is 2 m. Determine the size of bolts when the crane carries a load of 100 kN at a distance of 5 m from the centre of the base. The allowable stress for the bolt material is 100 MPa. The table for metric coarse threads is given below :*

Major diameter (mm)	*20*	*24*	*30*	*36*	*42*	*48*
Pitch (mm)	*2.5*	*3.0*	*3.5*	*4.0*	*4.5*	*5.0*
Stress area (mm^2)	*245*	*353*	*561*	*817*	*1120*	*1472*

Solution. Given : $n = 8$; $d = 1.6$ m or $r = 0.8$ m ; $D = 2$m or $R = 1$m ; $W = 100$ kN $= 100 \times 10^3$ N ; $e = 5$ m; $\sigma_t = 100$ MPa $= 100$ N/mm^2

The pillar crane is shown in Fig. 11.42.

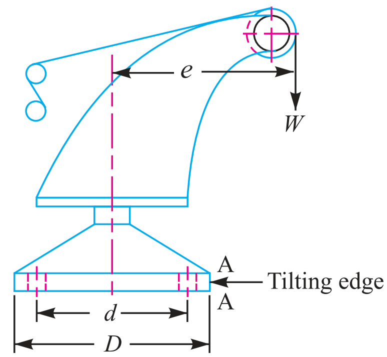

Fig. 11.42

We know that the distance of the load from the tilting edge *A-A*,

$$L = e - R = 5 - 1 = 4 \text{ m}$$

Let d_c = Core diameter of the bolts.

We know that maximum load on a bolt,

$$W_t = \frac{2\,W.L\,(R + r)}{n\,(2R^2 + r^2)} = \frac{2 \times 100 \times 10^3 \times 4\,(1 + 0.8)}{8\,[2 \times 1^2 + (0.8)^2]}$$

$$= \frac{1440 \times 10^3}{21.12} = 68.18 \times 10^3 \text{ N}$$

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

$$68.18 \times 10^3 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(d_c)^2\,100 = 78.54\,(d_c)^2$$

$\therefore \quad (d_c)^2 = 68.18 \times 10^3 / 78.54 = 868 \quad$ or $\quad d_c = 29.5$ mm

From Table 11.1 (coarse series), we find that the standard core diameter of the bolt is 31.093 mm and the corresponding size of the bolt is M 36. **Ans.**

Example 11.18. *A flanged bearing, as shown in Fig. 11.40, 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.*

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

Let d_c = Core diameter of the bolts.

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

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

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

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

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

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

From Table 11.1, we find that the standard core diameter of the bolt is 45.795 mm and corresponding size of the bolt is M 52. **Ans.**

Example 11.19. *A pillar crane having a circular base of 600 mm diameter is fixed to the foundation of concrete base by means of four bolts. The bolts are of size 30 mm and are equally spaced on a bolt circle diameter of 500 mm.*

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Determine : 1. The distance of the load from the centre of the pillar along a line X-X as shown in Fig. 11.43 (a). The load lifted by the pillar crane is 60 kN and the allowable tensile stress for the bolt material is 60 MPa.

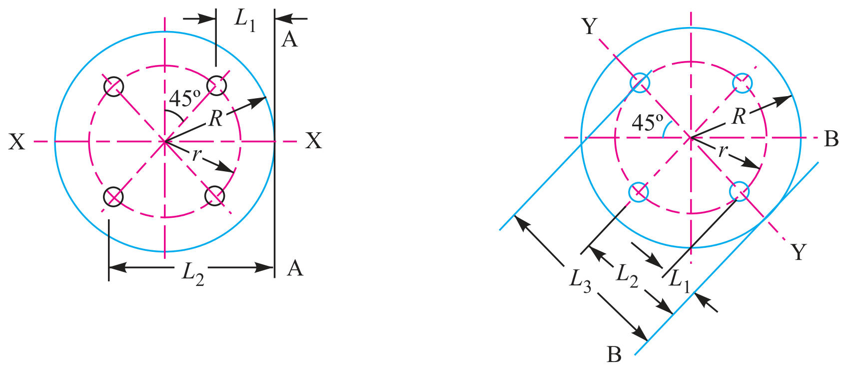

Fig. 11.43

2. The maximum stress induced in the bolts if the load is applied along a line Y-Y of the foundation as shown in Fig. 11.43 (b) at the same distance as in part (1).

Solution. Given : D = 600 mm or R = 300 mm ; n = 4; d_b = 30 mm ; d = 500 mm or r = 250 mm ; W = 60 kN ; σ_t = 60 MPa = 60 N/mm^2

Since the size of bolt (*i.e.* d_b = 30 mm), is given therefore from Table 11.1, we find that the stress area corresponding to M 30 is 561 mm^2.

We know that the maximum load carried by each bolt

$$= \text{Stress area} \times \sigma_t = 561 \times 60 = 33\ 660 \text{ N} = 33.66 \text{ kN}$$

and direct tensile load carried by each bolt

$$= \frac{W}{n} = \frac{60}{4} = 15 \text{ kN}$$

∴ Total load carried by each bolt at distance L_2 from the tilting edge A-A

$$= 33.66 + 15 = 48.66 \text{ kN} \quad ...(i)$$

From Fig. 11.43 (*a*), we find that

$$L_1 = R - r \cos 45° = 300 - 250 \times 0.707 = 123 \text{ mm} = 0.123 \text{ m}$$

and $$L_2 = R + r \cos 45° = 300 + 250 \times 0.707 = 477 \text{ mm} = 0.477 \text{ m}$$

Let w = Load (in kN) per bolt per unit distance.

∴ Total load carried by each bolt at distance L_2 from the tilting edge A-A

$$= w.L_2 = w \times 0.477 \text{ kN} \quad ...(ii)$$

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

$$w = 48.66 / 0.477 = 102 \text{ kN/m}$$

∴ Resisting moment of all the bolts about the outer (*i.e.* tilting) edge of the flange along the tangent A-A

$$= 2w\,[(L_1)^2 + (L_2)^2] = 2 \times 102\,[(0.123)^2 + (0.477)^2] = 49.4 \text{ kN-m}$$

1. Distance of the load from the centre of the pillar

Let e = Distance of the load from the centre of the pillar or eccentricity of the load, and

L = Distance of the load from the tilting edge A-A = $e - R = e - 0.3$

Top

We know that turning moment due to load W, about the tilting edge A-A of the flange

$$= W.L = 60\,(e - 0.3) \text{ kN-m}$$

Now equating the turning moment to the resisting moment of all the bolts, we have

$$60\,(e - 0.3) = 49.4$$

$$\therefore \quad e - 0.3 = 49.4 / 60 = 0.823 \quad \text{or} \quad e = 0.823 + 0.3 = 1.123 \text{ m}$$ **Ans.**

2. *Maximum stress induced in the bolt*

Since the load is applied along a line Y-Y as shown in Fig. 11.43 (*b*), and at the same distance as in part (*1*) *i.e.* at $L = e - 0.3 = 1.123 - 0.3 = 0.823$ m from the tilting edge B-B, therefore

Turning moment due to load W about the tilting edge B–B

$$= W.L = 60 \times 0.823 = 49.4 \text{ kN-m}$$

From Fig. 11.43 (*b*), we find that

$$L_1 = R - r = 300 - 250 = 50 \text{ mm} = 0.05 \text{ m}$$

$$L_2 = R = 300 \text{ mm} = 0.3 \text{ m}$$

and

$$L_3 = R + r = 300 + 250 = 550 \text{ mm} = 0.55 \text{ m}$$

$\therefore$ Resisting moment of all the bolts about B–B

$$= w\,[(L_1)^2 + 2(L_2)^2 + (L_3)^2] = w[(0.05)^2 + 2(0.3)^2 + (0.55)^2] \text{ kN-m}$$

$$= 0.485\, w \text{ kN-m}$$

Equating resisting moment of all the bolts to the turning moment, we have

$$0.485\, w = 49.4$$

or

$$w = 49.4 / 0.485 = 102 \text{ kN/m}$$

Since the bolt at a distance of L_3 is heavily loaded, therefore load carried by this bolt

$$= w.L_3 = 102 \times 0.55 = 56.1 \text{ kN}$$

Harvesting machine

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

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and net force taken by the bolt

$$= w.L_3 - \frac{W}{n} = 56.1 - \frac{60}{4} = 41.1 \text{ kN} = 41\ 100 \text{ N}$$

∴ Maximum stress induced in the bolt

$$= \frac{\text{Force}}{\text{Stress area}} = \frac{41\ 000}{516}$$

$= 79.65 \text{ N/mm}^2 = 79.65$ MPa **Ans.**

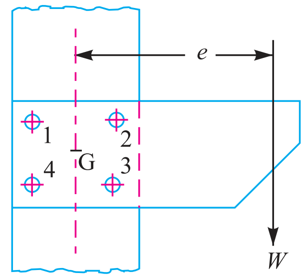


Fig. 11.44. Eccentric load in the plane containing the bolts.

11.22 Eccentric Load Acting in the Plane Containing the Bolts

When the eccentric load acts in the plane containing the bolts, as shown in Fig. 11.44, then the same procedure may be followed as discussed for eccentric loaded riveted joints.

Example 11.20. *Fig. 11.45 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 tension 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

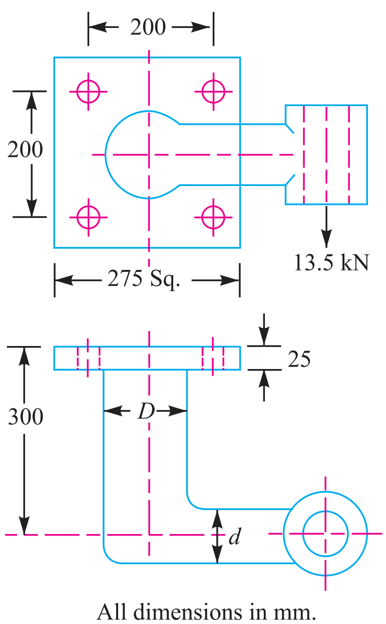


Fig. 11.45

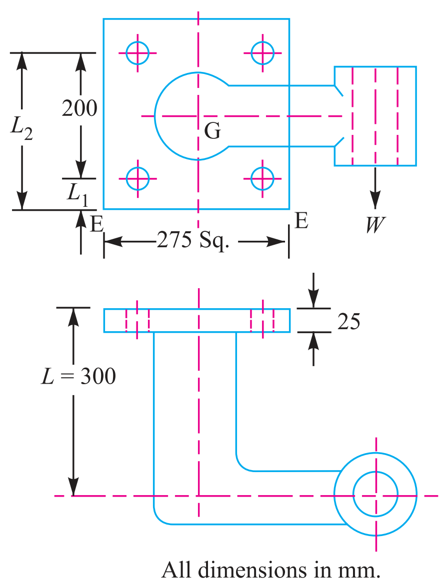


Fig. 11.46

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,

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

Top

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

∴ Equivalent twisting moment,

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

We know that equivalent twisting moment (T_e),

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

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

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

Diameter (d) for the arm of the bracket

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

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

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

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

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

∴ $d^3 = 29.2 \times 10^6 / 110 = 265.5 \times 10^3$ 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 $E–E$

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

...(∵ $L_1 = 37.5$ mm and $L_2 = 237.5$ mm)

and turning moment of the load about the tilting edge

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

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

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

∴ Tensile load on each top bolt

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

Maximum shearing force on each bolt

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

$$W_{s1} = \frac{W}{n} = \frac{13\,500}{4} = 3375 \text{ N}$$ ...(∵ 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,

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

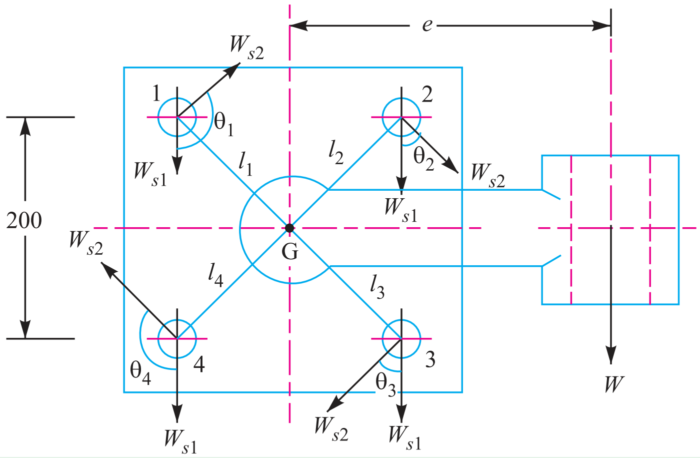


Fig. 11.47

∴ 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. 11.47, 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. 11.47, 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

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

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

EXERCISES

1. Determine the safe tensile load for bolts of M 20 and M 36. Assume that the bolts are not initially stressed and take the safe tensile stress as 200 MPa. **[Ans. 49 kN; 16.43 kN]**

2. An eye bolt carries a tensile load of 20 kN. Find the size of the bolt, if the tensile stress is not to exceed 100 MPa. Draw a neat proportioned figure for the bolt. **[Ans. M 20]**

3. An engine cylinder is 300 mm in diameter and the steam pressure is 0.7 N/mm^2. If the cylinder head is held by 12 studs, find the size. Assume safe tensile stress as 28 MPa. **[Ans. M 24]**

4. Find the size of 14 bolts required for a C.I. steam engine cylinder head. The diameter of the cylinder is 400 mm and the steam pressure is 0.12 N/mm^2. Take the permissible tensile stress as 35 MPa. **[Ans. M 24]**

5. The cylinder head of a steam engine is subjected to a pressure of 1 N/mm^2. It is held in position by means of 12 bolts. The effective diameter of the cylinder is 300 mm. A soft copper gasket is used to make the joint leak proof. Determine the size of the bolts so that the stress in the bolts does not exceed 100 MPa. **[Ans. M 36]**

6. A steam engine cylinder of 300 mm diameter is supplied with steam at 1.5 N/mm^2. The cylinder cover is fastened by means of 8 bolts of size M 20. The joint is made leak proof by means of suitable gaskets. Find the stress produced in the bolts. **[Ans. 249 MPa]**

7. The effective diameter of the cylinder is 400 mm. The maximum pressure of steam acting on the cylinder cover is 1.12 N/mm^2. Find the number and size of studs required to fix the cover. Draw a neat proportioned sketch for the elevation of the cylinder cover. **[Ans. 14; M 24]**

8. Specify the size and number of studs required to fasten the head of a 400 mm diameter cylinder containing steam at 2 N/mm^2. A hard gasket (gasket constant = 0.3) is used in making the joint. Draw a neat sketch of the joint also. Other data may be assumed. **[Ans. M 30; 12]**

9. A steam engine cylinder has an effective diameter of 200 mm. It is subjected to a maximum steam pressure of 1.75 N/mm^2. Calculate the number and size of studs required to fix the cylinder cover onto the cylinder flange assuming the permissible stress in the studs as 30 MPa. Take the pitch circle diameter of the studs as 320 mm and the total load on the studs as 20% higher than the external load on the joint. Also check the circumferential pitch of the studs so as to give a leak proof joint.

[Ans. 16; M 16]

10. A steam engine cylinder of size 300 mm × 400 mm operates at 1.5 N/mm^2 pressure. The cylinder head is connected by means of 8 bolts having yield point stress of 350 MPa and endurance limit of 240 MPa. The bolts are tightened with an initial preload of 1.8 times the steam lead. The joint is made leak-proof by using soft copper gasket which renderes the effect of external load to be half. Determine the size of bolts, if factor of safety is 2 and stress concentration factor is 3. **[Ans. M 20]**

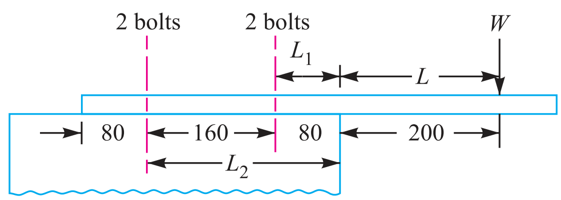
11. The cylinder head of a 200 mm × 350 mm compressor is secured by means of 12 studs of rolled mild steel. The gas pressure is 1.5 N/mm^2 gauge. The initial tension in the bolts, assumed to be equally loaded such that a cylinder pressure of 3 N/mm^2 gauge is required for the joint to be on the point of opening. Suggest the suitable size of the studs in accordance with Soderberg's equation assuming the equivalent diameter of the compressed parts to be twice the bolt size and factor of safety 2. The stress concentration factor may be taken as 2.8 and the value of endurance strength for reversed axial loading is half the value of ultimate strength. **[Ans. M 12]**

12. Find the diameter of screwed boiler stays, each stay supports an area equal to 200 mm × 150 mm. The steam pressure is 1 N/mm^2. The permissible tensile stress for the stay material is 34 MPa. **[Ans. M 36]**

13. What size of hole must be drilled in a M 42 bolt so as to make the bolt of uniform strength?

[Ans. 18.4 mm]

14. A mounting plate for a drive unit is fixed to the support by means of four M 12 bolts as shown in Fig. 11.48. The core diameter of the bolts can be considered as 9.858 mm. Determine the maximum value of 'W' if the allowable tensile stress in bolt material is 60 MPa. **[Ans. 12.212 kN]**

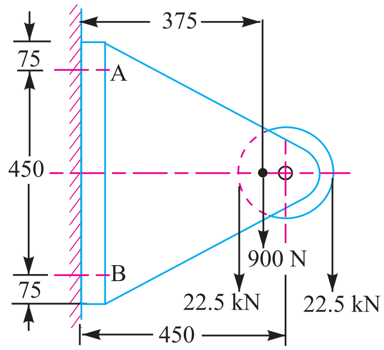


All dimensions in mm.

Fig. 11.48

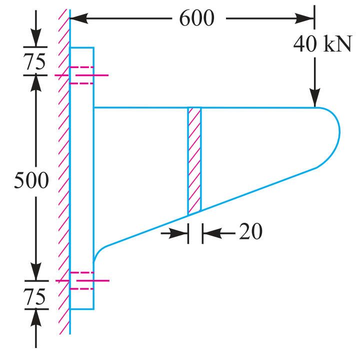
15. A pulley bracket, as shown in Fig. 11.49, is supported by 4 bolts, two at *A-A* and two at *B-B*. Determine the size of bolts using an allowable shear stress of 25 MPa for the material of the bolts. **[Ans. M 27]**

16. A wall bracket, as shown in Fig. 11.50, is fixed to a wall by means of four bolts. Find the size of the bolts and the width of bracket. The safe stress in tension for the bolt and bracket may be assumed as 70 MPa. **[Ans. M 30 ; 320 mm]**



All dimensions in mm.

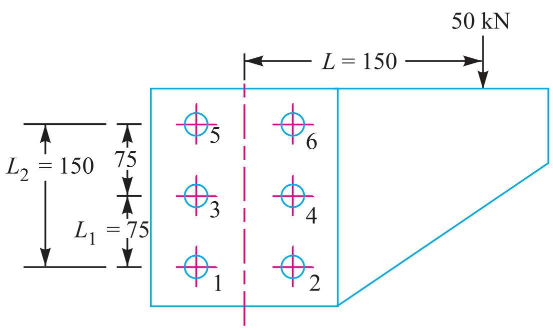
Fig. 11.49

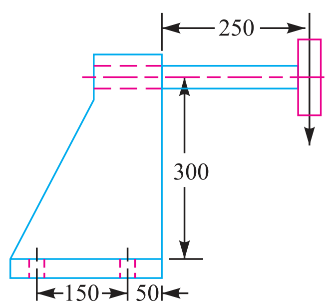


All dimensions in mm.

Fig. 11.50

17. A bracket is bolted to a column by 6 bolts of equal size as shown in Fig. 11.51. It carries a load of 50 kN at a distance of 150 mm from the centre of column. If the maximum stress in the bolts is to be limited to 150 MPa, determine the diameter of bolt. **[Ans. 14 mm]**





All dimensions in mm.

Fig. 11.51 **Fig. 11.52**

18. A cast iron bracket to carry a shaft and a belt pulley is shown in Fig. 11.52. The bracket is fixed to the main body by means of four standard bolts. The tensions in the slack and tight sides of the belt are 2.2 kN and 4.25 kN respectively. Find the size of the bolts, if the safe tensile stress for bolts is 50 MPa. **[Ans. M 16]**

19. Determine the size of the foundation bolts for a 60 kN pillar crane as shown in Fig. 11.42 (page 421) from the following data :

Distance of the load from the centre of the pillar	= 1.25 m
Diameter of pillar flange	= 600 mm
Diameter of bolt circle	= 500 mm
Number of bolts, equally spaced	= 4
Allowable tensile stress for bolts	= 600 MPa

[Ans. M 33]

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20. A bracket, as shown in Fig. 11.53, is fixed to a vertical steel column by means of five standard bolts. Determine : (*a*) The diameter of the fixing bolts, and (*b*) The thickness of the arm of the bracket. Assume safe working stresses of 70 MPa in tension and 50 MPa in shear.

[Ans. M 18; 50 mm]

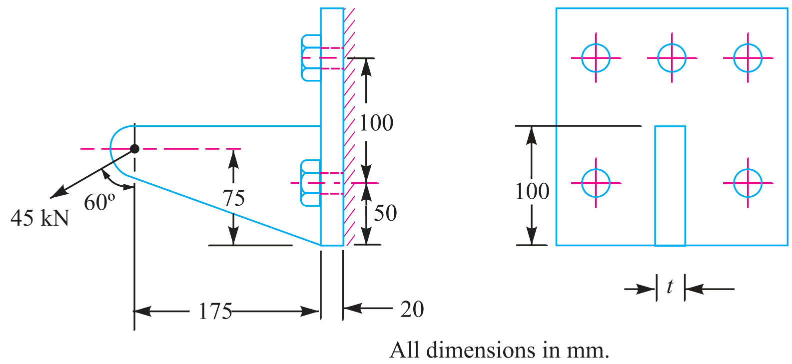

Fig. 11.53

QUESTIONS

1. What do you understand by the single start and double start threads ?
2. Define the following terms :
   (*a*) Major diameter, (*b*) Minor diameter, (*c*) Pitch, and (*d*) Lead.
3. Write short note on nut locking devices covering the necessity and various types. Your answer should be illustrated with neat sketches.
4. Discuss the significance of the initial tightening load and the applied load so far as bolts are concerned. Explain which of the above loads must be greater for a properly designed bolted joint and show how each affects the total load on the bolt.
5. Discuss on bolts of uniform strength giving examples of practical applications of such bolts.
6. Bolts less than M 16 should normally be used in pre loaded joints. Comment.
7. How the core diameter of the bolt is determined when a bracket having a rectangular base is bolted to a wall by four bolts and carries an eccentric load parallel to the axis of the bolt?
8. Derive an expression for the maximum load in a bolt when a bracket with circular base is bolted to a wall by means of four bolts.
9. Explain the method of determining the size of the bolt when the bracket carries an eccentric load perpendicular to the axis of the bolt.

OBJECTIVE TYPE QUESTIONS

1. The largest diameter of an external or internal screw thread is known as
   (*a*) minor diameter (*b*) major diameter
   (*c*) pitch diameter (*d*) none of these
2. The pitch diameter is the .... diameter of an external or internal screw thread.
   (*a*) effective (*b*) smallest (*c*) largest
3. A screw is specified by its
   (*a*) major diameter (*b*) minor diameter

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(c) pitch diameter (d) pitch

4. The railway carriage coupling have
   (a) square threads (b) acme threads
   (c) knuckle threads (d) buttress threads
5. The square threads are usually found on
   (a) spindles of bench vices (b) railway carriage couplings
   (c) feed mechanism of machine tools (d) screw cutting lathes
6. A locking device in which the bottom cylindrical portion is recessed to receive the tip of the locking set screw, is called
   (a) castle nut (b) jam nut
   (c) ring nut (d) screw nut
7. Which one is not a positive locking device ?
   (a) Spring washer (b) Cotter pin
   (c) Tongued washer (d) Spring wire lock
8. The washer is generally specified by its
   (a) outer diameter (b) hole diameter
   (c) thickness (d) mean diameter
9. A locking device extensively used in automobile industry is a
   (a) jam nut (b) castle nut
   (c) screw nut (d) ring nut
10. A bolt of M 24 × 2 means that
   (a) the pitch of the thread is 24 mm and depth is 2 mm
   (b) the cross-sectional area of the threads is 24 mm^2
   (c) the nominal diameter of bolt is 24 mm and the pitch is 2 mm
   (d) the effective diameter of the bolt is 24 mm and there are two threads per cm
11. When a nut is tightened by placing a washer below it, the bolt will be subjected to
   (a) tensile stress (b) compressive stress
   (c) shear stress (d) none of these
12. The eye bolts are used for
   (a) transmission of power (b) locking devices
   (c) lifting and transporting heavy machines (d) absorbing shocks and vibrations
13. The shock absorbing capacity of a bolt may be increased by
   (a) increasing its shank diameter
   (b) decreasing its shank diameter
   (c) tightening the bolt properly
   (d) making the shank diameter equal to the core diameter of the thread.
14. The resilience of a bolt may be increased by
   (a) increasing its shank diameter (b) increasing its length
   (c) decreasing its shank diameter (d) decreasing its length
15. A bolt of uniform strength can be developed by
   (a) keeping the core diameter of threads equal to the diameter of unthreaded portion of the bolt
   (b) keeping the core diameter of threads smaller than the diameter of unthreaded portion of the bolt
   (c) keeping the nominal diameter of threads equal to the diameter of unthreaded portion of bolt
   (d) none of the above

ANSWERS

1. (b)	2. (a)	3. (a)	4. (d)	5. (c)
6. (c)	7. (a)	8. (b)	9. (b)	10. (c)
11. (a)	12. (c)	13. (b)	14. (b)	15. (a)

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

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$$\frac{\text{Load}}{\text{Cross-sectional area of the bracket at } X - X}$$

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$\therefore$ $(d_c)^2$ = 4920 / 78.55 = 62.6 or d_c = 7.9 mm

W_H

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Solution. Given : n = 8 ; d = 1.6 m or r = 0.8 m ; D = 2m or R = 1m ; W = 100 kN = 100 × 10^3 N ; e = 5 m; σ_t = 100 MPa = 100 N/mm^2

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

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Solution. Given : D = 600 mm or R = 300 mm ; n = 4; d_b = 30 mm ; d = 500 mm or r = 250 mm ; W = 60 kN ; σ_t = 60 MPa = 60 N/mm^2

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$\therefore \qquad d^2 = 30 \times 10^3 / 39.3 = 763 \quad \text{or} \quad d = 27.6 \text{ say } 28 \text{ mm}$ **Ans.**

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Width of gib, $\quad b_1 = 0.55\, B$; and width of cotter, $b = 0.45\, B$

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find the value of d_3 by substituting σ_c = 84 N/mm^2 in the above expression, *i.e.*

$$70\ 695 = (d_3 - 55)\ 16.5 \times 84 = (d_3 - 55)\ 1386$$

$$\therefore \quad d_3 - 55 = 70\ 695 / 1386 = 51$$

or $\qquad d_3 = 55 + 51 = 106$ mm **Ans.**

We know the tapered length of the piston rod,

$$L = 2.2\, d_2 = 2.2 \times 55 = 121 \text{ mm}$$ **Ans.**

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm}$$ **Ans.**

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm}$$ **Ans.**

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

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We shall now discuss the above types of keys, in detail, in the following pages.

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Shaft couplings are divided into two main groups as follows :

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

Sleeve or muff coupling.

Clamp or split-muff or compression coupling, and

Flange coupling.

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

Bushed pin type coupling,

Universal coupling, and

Oldham coupling.

We shall now discuss the above types of couplings, in detail, in the following pages.

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Solution. Given : P = 40 kW = 40 × 10^3 W; N = 350 r.p.m.; τ_s = 40 MPa = 40 N/mm^2; σ_{cs} = 80 MPa = 80 N/mm^2; τ_c = 15 MPa = 15 N/mm^2

The muff coupling is shown in Fig. 13.10. It is designed as discussed below :

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$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$\therefore \quad d^3 = 1100 \times 10^3 / 7.86 = 140 \times 10^3 \quad \text{or} \quad d = 52 \text{ say } 55 \text{ mm } \textbf{Ans.}$$

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$$1100 \times 10^3 = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right) = \frac{\pi}{16} \times \tau_c \left[\frac{(125)^4 - (55)^4}{125} \right]$$

$$= 370 \times 103\ \tau_c$$

$$\therefore \quad \tau_c = 1100 \times 10^3 / 370 \times 10^3 = 2.97 \text{ N/mm}^2$$

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Solution. Given : P = 30 kW = 30 × 10^3 W ; N = 100 r.p.m. ; τ = 40 MPa = 40 N/mm^2 ; n = 6 ; σ_t = 70 MPa = 70 N/mm$_2$; μ = 0.3

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$$\therefore \qquad (d_b)^2 = 2865 \times 10^3 / 5830 = 492 \quad \text{or} \quad d_b = 22.2 \text{ mm}$$

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Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 900 r.p.m. ; Service factor = 1.35 ; $\tau_s = \tau_b = \tau_k$ = 40 MPa = 40 N/mm^2 ; $\sigma_{cb} = \sigma_{ck}$ = 80 MPa = 80 N/mm^2 ; τ_c = 8 MPa = 8 N/mm^2

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Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 200 r.p.m. ; τ_s = 40 MPa = 40 N/mm^2 ; τ_b = 30 MPa = 30 N/mm^2 ; $\sigma_{ck} = 2t_k$; $T_{max} = 1.25\ T_{mean}$; τ_c = 14 MPa = 14 N/mm^2

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$$\ldots \left(\because J = \frac{\pi}{32} \times d^4\right)$$

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C H A P T E R

12

Cotter and Knuckle Joints

12.1 Introduction

A cotter is a flat wedge shaped piece of rectangular cross-section and its width is tapered (either on one side or both sides) from one end to another for an easy adjustment. The taper varies from 1 in 48 to 1 in 24 and it may be increased up to 1 in 8, if a locking device is provided. The locking device may be a taper pin or a set screw used on the lower end of the cotter. The cotter is usually made of mild steel or wrought iron. A cotter joint is a temporary fastening and is used to connect rigidly two co-axial rods or bars which are subjected to axial tensile or compressive forces. It is usually used in connecting a piston rod to the cross-head of a reciprocating steam engine, a piston rod and its extension as a tail or pump rod, strap end of connecting rod etc.

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12.2 Types of Cotter Joints

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

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

The design of these types of joints are discussed, in detail, in the following pages.

12.3 Socket and Spigot Cotter Joint

In a socket and spigot cotter joint, one end of the rods (say *A*) is provided with a socket type of end as shown in Fig. 12.1 and the other end of the other rod (say *B*) is inserted into a socket. The end of the rod which goes into a socket is also called ***spigot.*** *A* rectangular hole is made in the socket and spigot. *A* cotter is then driven tightly through a hole in order to make the temporary connection between the two rods. The load is usually acting axially, but it changes its direction and hence the cotter joint must be designed to carry both the tensile and compressive loads. The compressive load is taken up by the collar on the spigot.

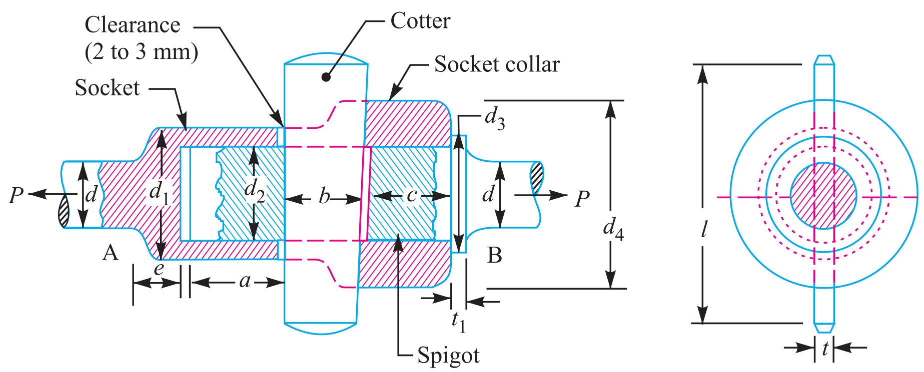

Fig. 12.1. Socket and spigot cotter joint.

12.4 Design of Socket and Spigot Cotter Joint

The socket and spigot cotter joint is shown in Fig. 12.1.

Let P = Load carried by the rods,

d = Diameter of the rods,

d_1 = Outside diameter of socket,

d_2 = Diameter of spigot or inside diameter of socket,

d_3 = Outside diameter of spigot collar,

t_1 = Thickness of spigot collar,

d_4 = Diameter of socket collar,

c = Thickness of socket collar,

b = Mean width of cotter,

t = Thickness of cotter,

l = Length of cotter,

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

σ_t = Permissible tensile stress for the rods material,

τ = Permissible shear stress for the cotter material, and

σ_c = Permissible crushing stress for the cotter material.

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

Fork lift is used to move goods from one place to the other within the factory.

1. *Failure of the rods in tension*

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

Area resisting tearing

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

∴ Tearing strength of the rods,

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

Equating this to load (P), we have

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

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

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

Since the weakest section of the spigot is that section which has a slot in it for the cotter, as shown in Fig. 12.2, therefore

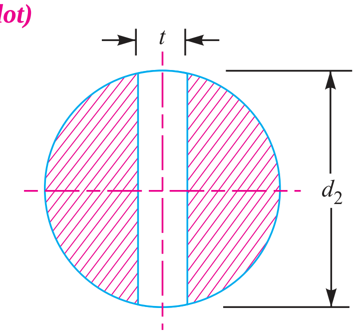

Fig. 12.2

Area resisting tearing of the spigot across the slot

$$= \frac{\pi}{4}(d_2)^2 - d_2 \times t$$

and tearing strength of the spigot across the slot

$$= \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$$

Equating this to load (P), we have

$$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$$

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

Note : In actual practice, the thickness of cotter is usually taken as $d_2 / 4$.

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

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

$$= d_2 \times t$$

∴ Crushing strength $= d_2 \times t \times \sigma_c$

Equating this to load (P), we have

$$P = d_2 \times t \times \sigma_c$$

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

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4. Failure of the socket in tension across the slot

We know that the resisting area of the socket across the slot, as shown in Fig. 12.3

$$= \frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)\, t$$

∴ Tearing strength of the socket across the slot

$$= \left\{\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\, t\right\} \sigma_t$$

Equating this to load (P), we have

$$P = \left\{\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\, t\right\} \sigma_t$$

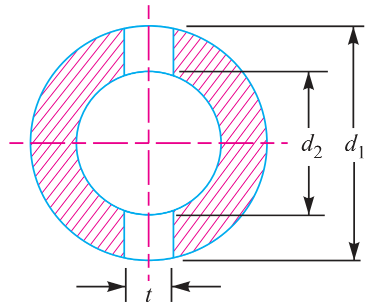

Fig. 12.3

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

5. Failure of cotter in shear

Considering the failure of cotter in shear as shown in Fig. 12.4. Since the cotter is in double shear, therefore shearing area of the cotter

$$= 2\, b \times t$$

and shearing strength of the cotter

$$= 2\, b \times t \times \tau$$

Equating this to load (P), we have

$$P = 2\, b \times t \times \tau$$

From this equation, width of cotter (b) is determined.

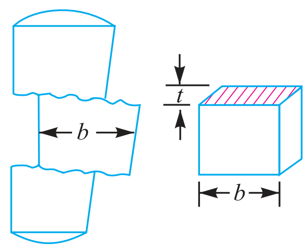

Fig. 12.4

6. Failure of the socket collar in crushing

Considering the failure of socket collar in crushing as shown in Fig. 12.5.

We know that area that resists crushing of socket collar

$$= (d_4 - d_2)\, t$$

and crushing strength $= (d_4 - d_2)\, t \times \sigma_c$

Equating this to load (P), we have

$$P = (d_4 - d_2)\, t \times \sigma_c$$

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

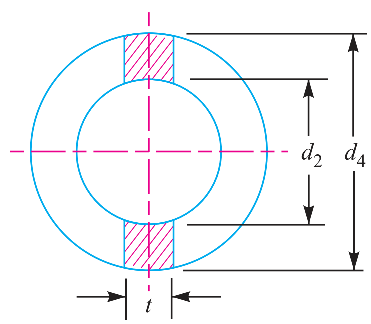

Fig. 12.5

7. Failure of socket end in shearing

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

$$= 2\, (d_4 - d_2)\, c$$

and shearing strength of socket collar

$$= 2\, (d_4 - d_2)\, c \times \tau$$

Equating this to load (P), we have

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

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

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8. *Failure of rod end in shear*

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

$$= 2\,a \times d_2$$

and shear strength of the rod end

$$= 2\,a \times d_2 \times \tau$$

Equating this to load (P), we have

$$P = 2\,a \times d_2 \times \tau$$

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

9. *Failure of spigot collar in crushing*

Considering the failure of the spigot collar in crushing as shown in Fig. 12.6. We know that area that resists crushing of the collar

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]$$

and crushing strength of the collar

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

Equating this to load (P), we have

$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$$

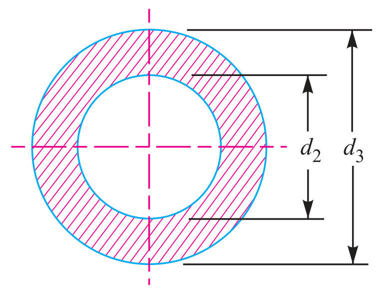

Fig. 12.6

From this equation, the diameter of the spigot collar (d_3) may be obtained.

10. *Failure of the spigot collar in shearing*

Considering the failure of the spigot collar in shearing as shown in Fig. 12.7. We know that area that resists shearing of the collar

$$= \pi\, d_2 \times t_1$$

and shearing strength of the collar,

$$= \pi\, d_2 \times t_1 \times \tau$$

Equating this to load (P) we have

$$P = \pi\, d_2 \times t_1 \times \tau$$

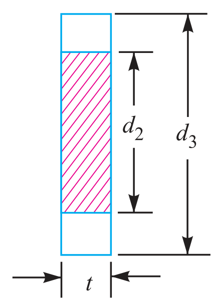

Fig. 12.7

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

11. *Failure of cotter in bending*

In all the above relations, it is assumed that the load is uniformly distributed over the various cross-sections of the joint. But in actual practice, this does not happen and the cotter is subjected to bending. In order to find out the bending stress induced, it is assumed that the load on the cotter in the rod end is uniformly distributed while in the socket end it varies from zero at the outer diameter (d_4) and maximum at the inner diameter (d_2), as shown in Fig. 12.8.

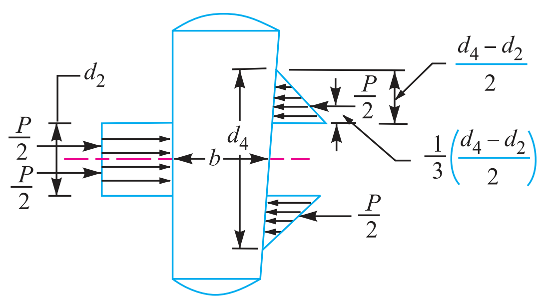

Fig. 12.8

The maximum bending moment occurs at the centre of the cotter and is given by

$$M_{max} = \frac{P}{2}\left(\frac{1}{3} \times \frac{d_4 - d_2}{2} + \frac{d_2}{2}\right) - \frac{P}{2} \times \frac{d_2}{4}$$

$$= \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{2} - \frac{d_2}{4}\right) = \frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)$$

We know that section modulus of the cotter,

$$Z = t \times b^2 / 6$$

∴ Bending stress induced in the cotter,

$$\sigma_b = \frac{M_{max}}{Z} = \frac{\frac{P}{2}\left(\frac{d_4 - d_2}{6} + \frac{d_2}{4}\right)}{t \times b^2 / 6} = \frac{P\,(d_4 + 0.5\,d_2)}{2\,t \times b^2}$$

This bending stress induced in the cotter should be less than the allowable bending stress of the cotter.

12. The length of cotter (l) is taken as 4 d.

13. The taper in cotter should not exceed 1 in 24. In case the greater taper is required, then a locking device must be provided.

14. The draw of cotter is generally taken as 2 to 3 mm.

Notes: 1. When all the parts of the joint are made of steel, the following proportions in terms of diameter of the rod (d) are generally adopted :

$d_1 = 1.75\,d$, $d_2 = 1.21\,d$, $d_3 = 1.5\,d$, $d_4 = 2.4\,d$, $a = c = 0.75\,d$, $b = 1.3\,d$, $l = 4\,d$, $t = 0.31\,d$, $t_1 = 0.45\,d$, $e = 1.2\,d$.

Taper of cotter = 1 in 25, and draw of cotter = 2 to 3 mm.

2. If the rod and cotter are made of steel or wrought iron, then $\tau = 0.8\,\sigma_t$ and $\sigma_c = 2\,\sigma_t$ may be taken.

Example 12.1. *Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically.*

Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress = 90 MPa.

Solution. Given : $P = 30$ kN $= 30 \times 10^3$ N ; $\sigma_t = 50$ MPa $= 50$ N / mm^2 ; $\tau = 35$ MPa $= 35$ N / mm^2 ; $\sigma_c = 90$ MPa $= 90$ N/mm^2

Accessories for hand operated sockets.

The cotter joint is shown in Fig. 12.1. The joint is designed as discussed below :

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rod in tension. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\, d^2$$

$$\therefore \quad d^2 = 30 \times 10^3 / 39.3 = 763 \quad \text{or} \quad d = 27.6 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

2. *Diameter of spigot and thickness of cotter*

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]50 = 26.8\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4 \quad \text{or} \quad d_2 = 33.4 \text{ say } 34 \text{ mm}$$

and thickness of cotter, $t = \dfrac{d_2}{4} = \dfrac{34}{4} = 8.5 \text{ mm}$

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\, \sigma_c$$

$$\therefore \quad \sigma_c = 30 \times 10^3 / 289 = 103.8 \text{ N/mm}^2$$

Since this value of σ_c is more than the given value of $\sigma_c = 90$ N/mm^2, therefore the dimensions $d_2 = 34$ mm and $t = 8.5$ mm are not safe. Now let us find the values of d_2 and t by substituting the value of $\sigma_c = 90$ N/mm^2 in the above expression, *i.e.*

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 30 \times 10^3 / 22.5 = 1333 \quad \text{or} \quad d_2 = 36.5 \text{ say } 40 \text{ mm} \quad \textbf{Ans.}$$

and $t = d_2 / 4 = 40 / 4 = 10$ mm **Ans.**

3. *Outside diameter of socket*

Let d_1 = Outside diameter of socket.

Considering the failure of the socket in tension across the slot. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\, t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}\left\{(d_1)^2 - (40)^2\right\} - (d_1 - 40)\, 10\right]50$$

$$30 \times 10^3 / 50 = 0.7854\,(d_1)^2 - 1256.6 - 10\, d_1 + 400$$

$$\text{or} \quad (d_1)^2 - 12.7\, d_1 - 1854.6 = 0$$

$$\therefore \quad d_1 = \frac{12.7 \pm \sqrt{(12.7)^2 + 4 \times 1854.6}}{2} = \frac{12.7 \pm 87.1}{2}$$

$$= 49.9 \text{ say } 50 \text{ mm} \quad \textbf{Ans.} \qquad \text{...(Taking +ve sign)}$$

4. *Width of cotter*

Let b = Width of cotter.

Considering the failure of the cotter in shear. Since the cotter is in double shear, therefore load (P),

$$30 \times 10^3 = 2\, b \times t \times \tau = 2\, b \times 10 \times 35 = 700\, b$$

$$\therefore \quad b = 30 \times 10^3 / 700 = 43 \text{ mm } \textbf{Ans.}$$

5. *Diameter of socket collar*

Let d_4 = Diameter of socket collar.

Considering the failure of the socket collar and cotter in crushing. We know that load (P),

$$30 \times 10^3 = (d_4 - d_2)\, t \times \sigma_c = (d_4 - 40) 10 \times 90 = (d_4 - 40)\, 900$$

$$\therefore \quad d_4 - 40 = 30 \times 10^3 / 900 = 33.3 \quad \text{or} \quad d_4 = 33.3 + 40 = 73.3 \text{ say } 75 \text{ mm } \textbf{Ans.}$$

6. *Thickness of socket collar*

Let c = Thickness of socket collar.

Considering the failure of the socket end in shearing. Since the socket end is in double shear, therefore load (P),

$$30 \times 10^3 = 2(d_4 - d_2)\, c \times \tau = 2\,(75 - 40)\, c \times 35 = 2450\, c$$

$$\therefore \quad c = 30 \times 10^3 / 2450 = 12 \text{ mm } \textbf{Ans.}$$

7. *Distance from the end of the slot to the end of the rod*

Let a = Distance from the end of slot to the end of the rod.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$30 \times 10^3 = 2\, a \times d_2 \times \tau = 2a \times 40 \times 35 = 2800\, a$$

$$\therefore \quad a = 30 \times 10^3 / 2800 = 10.7 \text{ say } 11 \text{ mm } \textbf{Ans.}$$

8. *Diameter of spigot collar*

Let d_3 = Diameter of spigot collar.

Considering the failure of spigot collar in crushing. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \sigma_c = \frac{\pi}{4}\left[(d_3)^2 - (40)^2\right] 90$$

or
$$(d_3)^2 - (40)^2 = \frac{30 \times 10^3 \times 4}{90 \times \pi} = 424$$

$$\therefore \quad (d_3)^2 = 424 + (40)^2 = 2024 \quad \text{or} \quad d_3 = 45 \text{ mm } \textbf{Ans.}$$

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9. *Thickness of spigot collar*

Let t_1 = Thickness of spigot collar.

Considering the failure of spigot collar in shearing. We know that load (P),

$$30 \times 10^3 = \pi\, d_2 \times t_1 \times \tau = \pi \times 40 \times t_1 \times 35 = 4400\, t_1$$

$\therefore$ $t_1 = 30 \times 10^3 / 4400 = 6.8$ say 8 mm **Ans.**

10. The length of cotter (l) is taken as 4 d.

$\therefore$ $l = 4\,d = 4 \times 28 = 112$ mm **Ans.**

11. The dimension e is taken as 1.2 d.

$\therefore$ $e = 1.2 \times 28 = 33.6$ say 34 mm **Ans.**

12.5 Sleeve and Cotter Joint

Sometimes, a sleeve and cotter joint as shown in Fig. 12.9, is used to connect two round rods or bars. In this type of joint, a sleeve or muff is used over the two rods and then two cotters (one on each rod end) are inserted in the holes provided for them in the sleeve and rods. The taper of cotter is usually 1 in 24. It may be noted that the taper sides of the two cotters should face each other as shown in Fig. 12.9. The clearance is so adjusted that when the cotters are driven in, the two rods come closer to each other thus making the joint tight.

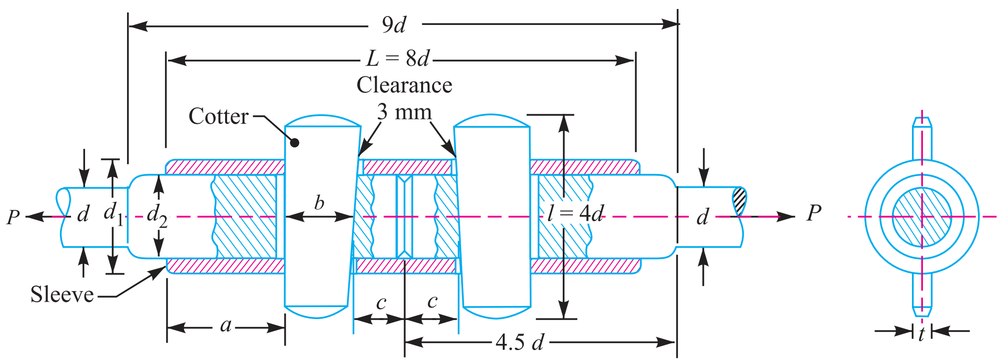

Fig. 12.9. Sleeve and cotter joint.

The various proportions for the sleeve and cotter joint in terms of the diameter of rod (d) are as follows :

Outside diameter of sleeve,

$$d_1 = 2.5\, d$$

Diameter of enlarged end of rod,

$$d_2 = \text{Inside diameter of sleeve} = 1.25\, d$$

Length of sleeve, $L = 8\, d$

Thickness of cotter, $t = d_2/4$ or $0.31\, d$

Width of cotter, $b = 1.25\, d$

Length of cotter, $l = 4\, d$

Distance of the rod end (a) from the beginning to the cotter hole (inside the sleeve end)

= Distance of the rod end (c) from its end to the cotter hole

$= 1.25\, d$

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12.6 Design of Sleeve and Cotter Joint

The sleeve and cotter joint is shown in Fig. 12.9.

Let P = Load carried by the rods,

d = Diameter of the rods,

d_1 = Outside diameter of sleeve,

d_2 = Diameter of the enlarged end of rod,

t = Thickness of cotter,

l = Length of cotter,

b = Width of cotter,

a = Distance of the rod end from the beginning to the cotter hole (inside the sleeve end),

c = Distance of the rod end from its end to the cotter hole,

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses respectively for the material of the rods and cotter.

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

1. *Failure of the rods in tension*

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

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

∴ Tearing strength of the rods

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

Equating this to load (P), we have

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

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

2. *Failure of the rod in tension across the weakest section (i.e. slot)*

Since the weakest section is that section of the rod which has a slot in it for the cotter, therefore area resisting tearing of the rod across the slot

$$= \frac{\pi}{4} (d_2)^2 - d_2 \times t$$

and tearing strength of the rod across the slot

$$= \left[\frac{\pi}{4} (d_2)^2 - d_2 \times t\right] \sigma_t$$

Equating this to load (P), we have

$$P = \left[\frac{\pi}{4} (d_2)^2 - d_2 \times t\right] \sigma_t$$

From this equation, the diameter of enlarged end of the rod (d_2) may be obtained.

Note: The thickness of cotter is usually taken as $d_2 / 4$.

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

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

$$= d_2 \times t$$

$\therefore$ Crushing strength $= d_2 \times t \times \sigma_c$

Equating this to load (P), we have

$$P = d_2 \times t \times \sigma_c$$

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

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

We know that the resisting area of sleeve across the slot

$$= \frac{\pi}{4}\left[(d_1)^2 - (d_2)^2 \right] - (d_1 - d_2)\, t$$

$\therefore$ Tearing strength of the sleeve across the slot

$$= \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\, t \right] \sigma_t$$

Equating this to load (P), we have

$$P = \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\, t \right] \sigma_t$$

From this equation, the outside diameter of sleeve (d_1) may be obtained.

5. *Failure of cotter in shear*

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

$$= 2b \times t$$

and shear strength of the cotter

$$= 2b \times t \times \tau$$

Equating this to load (P), we have

$$P = 2b \times t \times \tau$$

From this equation, width of cotter (b) may be determined.

6. *Failure of rod end in shear*

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

$$= 2\, a \times d_2$$

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and shear strength of the rod end

$$= 2\,a \times d_2 \times \tau$$

Equating this to load (P), we have

$$P = 2\,a \times d_2 \times \tau$$

From this equation, distance (a) may be determined.

7. *Failure of sleeve end in shear*

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

$$= 2\,(d_1 - d_2)\,c$$

and shear strength of the sleeve end

$$= 2\,(d_1 - d_2)\,c \times \tau$$

Equating this to load (P), we have

$$P = 2\,(d_1 - d_2)\,c \times \tau$$

From this equation, distance (c) may be determined.

Example 12.2. *Design a sleeve and cotter joint to resist a tensile load of 60 kN. All parts of the joint are made of the same material with the following allowable stresses :*

σ_t = 60 MPa ; τ = 70 MPa ; and σ_c = 125 MPa.

Solution. Given : $P = 60$ kN $= 60 \times 10^3$ N ; $\sigma_t = 60$ MPa $= 60$ N/mm^2 ; $\tau = 70$ MPa $= 70$ N/mm^2 ; $\sigma_c = 125$ MPa $= 125$ N/mm^2

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rods in tension. We know that load (P),

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 60 = 47.13\,d^2$$

$$\therefore \quad d^2 = 60 \times 10^3 / 47.13 = 1273 \quad \text{or} \quad d = 35.7 \text{ say } 36 \text{ mm}$$ **Ans.**

2. *Diameter of enlarged end of rod and thickness of cotter*

Let d_2 = Diameter of enlarged end of rod, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of the rod in tension across the weakest section (*i.e.* slot). We know that load (P),

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]60 = 32.13\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 60 \times 10^3 / 32.13 = 1867 \quad \text{or} \quad d_2 = 43.2 \text{ say } 44 \text{ mm}$$ **Ans.**

and thickness of cotter,

$$t = \frac{d_2}{4} = \frac{44}{4} = 11 \text{ mm}$$ **Ans.**

Let us now check the induced crushing stress in the rod or cotter. We know that load (P),

$$60 \times 10^3 = d_2 \times t \times \sigma_c = 44 \times 11 \times \sigma_c = 484\,\sigma_c$$

$$\therefore \quad \sigma_c = 60 \times 10^3 / 484 = 124 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given value of 125 N/mm^2, therefore the dimensions d_2 and t are within safe limits.

3. *Outside diameter of sleeve*

Let d_1 = Outside diameter of sleeve.

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Considering the failure of sleeve in tension across the slot. We know that load (P)

$$60 \times 10^3 = \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\,t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}[(d_1)^2 - (44)^2] - (d_1 - 44)\,11\right]60$$

$$\therefore \quad 60 \times 10^3 / 60 = 0.7854\,(d_1)^2 - 1520.7 - 11\,d_1 + 484$$

or $\quad (d_1)^2 - 14\,d_1 - 2593 = 0$

$$\therefore \quad d_1 = \frac{14 \pm \sqrt{(14)^2 + 4 \times 2593}}{2} = \frac{14 \pm 102.8}{2}$$

$= 58.4$ say 60 mm **Ans.** ...(Taking +ve sign)

4. *Width of cotter*

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$60 \times 10^3 = 2\,b \times t \times \tau = 2 \times b \times 11 \times 70 = 1540\,b$$

$$\therefore \quad b = 60 \times 10^3 / 1540 = 38.96 \text{ say } 40 \text{ mm}$$ **Ans.**

5. *Distance of the rod from the beginning to the cotter hole (inside the sleeve end)*

Let a = Required distance.

Considering the failure of the rod end in shear. Since the rod end is in double shear, therefore load (P),

$$60 \times 10^3 = 2\,a \times d_2 \times \tau = 2\,a \times 44 \times 70 = 6160\,a$$

$$\therefore \quad a = 60 \times 10^3 / 6160 = 9.74 \text{ say } 10 \text{ mm}$$ **Ans.**

6. *Distance of the rod end from its end to the cotter hole*

Let c = Required distance.

Considering the failure of the sleeve end in shear. Since the sleeve end is in double shear, therefore load (P),

$$60 \times 10^3 = 2\,(d_1 - d_2)\,c \times \tau = 2\,(60 - 44)\,c \times 70 = 2240\,c$$

$$\therefore \quad c = 60 \times 10^3 / 2240 = 26.78 \text{ say } 28 \text{ mm}$$ **Ans.**

12.7 Gib and Cotter Joint

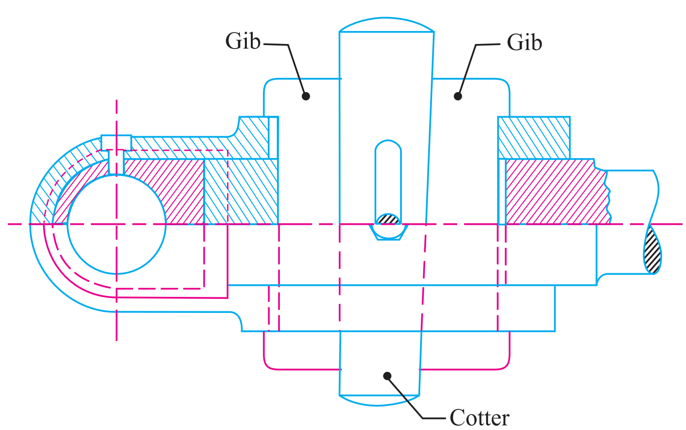

Fig. 12.10. Gib and cotter joint for strap end of a connecting rod.

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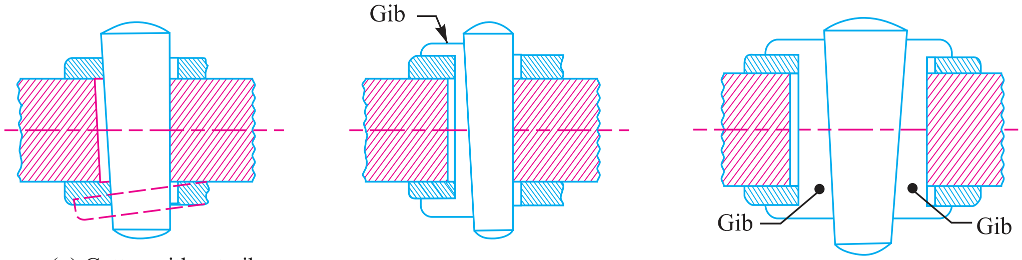
A *gib and cotter joint is usually used in strap end (or big end) of a connecting rod as shown in Fig. 12.10. In such cases, when the cotter alone (*i.e.* without gib) is driven, the friction between its ends and the inside of the slots in the strap tends to cause the sides of the strap to spring open (or spread) outwards as shown dotted in Fig. 12.11 (*a*). In order to prevent this, gibs as shown in Fig. 12.11 (*b*) and (*c*), are used which hold together the ends of the strap. Moreover, gibs provide a larger bearing surface for the cotter to slide on, due to the increased holding power. Thus, the tendency of cotter to slacken back owing to friction is considerably decreased. The jib, also, enables parallel holes to be used.



(*a*) Cotter without gib. (*b*) Cotter with one gib. (*c*) Cotter with double gib.

Fig. 12.11. Gib and cotter Joints.

Notes : 1. When one gib is used, the cotter with one side tapered is provided and the gib is always on the outside as shown in Fig. 12.11 (*b*).

2. When two jibs are used, the cotter with both sides tapered is provided.

3. Sometimes to prevent loosening of cotter, a small set screw is used through the rod jamming against the cotter.

12.8 Design of a Gib and Cotter Joint for Strap End of a Connecting Rod

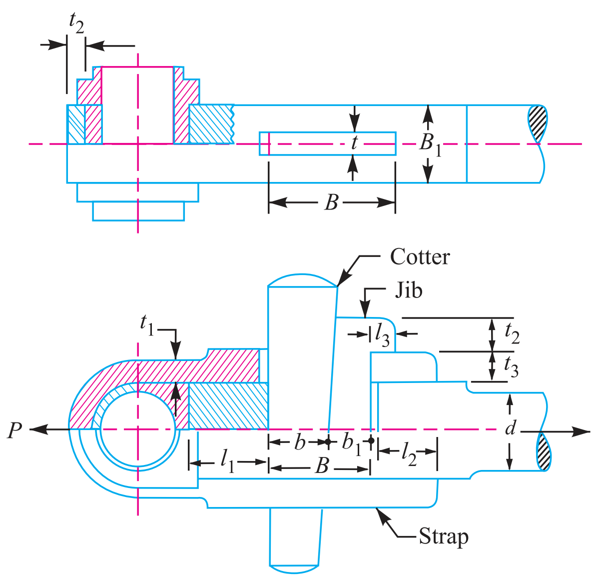


Fig. 12.12. Gib and cotter joint for strap end of a connecting rod.

Consider a gib and cotter joint for strap end (or big end) of a connecting rod as shown in Fig. 12.12. The connecting rod is subjected to tensile and compressive loads.

* A gib is a piece of mild steel having the same thickness and taper as the cotter.

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Let P = Maximum thrust or pull in the connecting rod,

d = Diameter of the adjacent end of the round part of the rod,

B_1 = Width of the strap,

B = Total width of gib and cotter,

t = Thickness of cotter,

t_1 = Thickness of the strap at the thinnest part,

σ_t = Permissible tensile stress for the material of the strap, and

τ = Permissible shear stress for the material of the cotter and gib.

The width of strap (B_1) is generally taken equal to the diameter of the adjacent end of the round part of the rod (d). The other dimensions may be fixed as follows :

Thickness of cotter,

$$t = \frac{\text{Width of strap}}{4} = \frac{B_1}{4}$$

Thickness of gib = Thickness of cotter (t)

Height (t_2) and length of gib head (l_3)

= Thickness of cotter (t)

In designing the gib and cotter joint for strap end of a connecting rod, the following modes of failure are considered.

1. *Failure of the strap in tension*

Assuming that no hole is provided for lubrication, the area that resists the failure of the strap due to tearing $= 2\, B_1 \times t_1$

∴ Tearing strength of the strap

$= 2\, B_1 \times t_1 \times \sigma_t$

Equating this to the load (P), we get

$$P = 2\, B_1 \times t_1 \times \sigma_t$$

From this equation, the thickness of the strap at the thinnest part (t_1) may be obtained. When an oil hole is provided in the strap, then its weakening effect should be considered.

The thickness of the strap at the cotter (t_3) is increased such that the area of cross-section of the strap at the cotter hole is not less than the area of the strap at the thinnest part. In other words

$$2\, t_3\, (B_1 - t) = 2\, t_1 \times B_1$$

From this expression, the value of t_3 may be obtained.

(a) Hand operated sqaure drive sockets (b) Machine operated sockets.

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

2. Failure of the gib and cotter in shearing

Since the gib and cotter are in double shear, therefore area resisting failure

$$= 2\,B \times t$$

and resisting strength $= 2\,B \times t \times \tau$

Equating this to the load (P), we get

$$P = 2\,B \times t \times \tau$$

From this equation, the total width of gib and cotter (B) may be obtained. In the joint, as shown in Fig. 12.12, one gib is used, the proportions of which are

Width of gib, $b_1 = 0.55\,B$; and width of cotter, $b = 0.45\,B$

The other dimensions may be fixed as follows :

Thickness of the strap at the crown,

$$t_4 = 1.15\,t_1 \text{ to } 1.5\,t_1$$

$$l_1 = 2\,t_1;\ \text{and}\ l_2 = 2.5\,t_1$$

Example 12.3. *The big end of a connecting rod, as shown in Fig. 12.12, is subjected to a maximum load of 50 kN. The diameter of the circular part of the rod adjacent to the strap end is 75 mm. Design the joint, assuming permissible tensile stress for the material of the strap as 25 MPa and permissible shear stress for the material of cotter and gib as 20 MPa.*

Solution. Given : $P = 50$ kN $= 50 \times 10^3$ N ; $d = 75$ mm ; $\sigma_t = 25$ MPa $= 25$ N/mm^2 ; $\tau = 20$ MPa $= 20$ N/mm^2

1. Width of the strap

Let B_1 = Width of the strap.

The width of the strap is generally made equal to the diameter of the adjacent end of the round part of the rod (d).

$\therefore$ $B_1 = d = 75$ mm **Ans.**

Other dimensions are fixed as follows :

Thickness of the cotter

$$t = \frac{B_1}{4} = \frac{75}{4} = 18.75 \text{ say } 20 \text{ mm} \quad \textbf{Ans.}$$

Thickness of gib = Thickness of cotter = 20 mm **Ans.**

Height (t_2) and length of gib head (l_3)

= Thickness of cotter = 20 mm **Ans.**

2. Thickness of the strap at the thinnest part

Let t_1 = Thickness of the strap at the thinnest part.

Considering the failure of the strap in tension. We know that load (P),

$$50 \times 10^3 = 2\,B_1 \times t_1 \times \sigma_t = 2 \times 75 \times t_1 \times 25 = 3750\,t_1$$

$$\therefore \quad t_1 = 50 \times 10^3 / 3750 = 13.3 \text{ say } 15 \text{ mm} \quad \textbf{Ans.}$$

3. Thickness of the strap at the cotter

Let t_3 = Thickness of the strap at the cotter.

The thickness of the strap at the cotter is increased such that the area of the cross-section of the strap at the cotter hole is not less than the area of the strap at the thinnest part. In other words,

$$2\,t_3\,(B_1 - t) = 2\,t_1 \times B_1$$

$$2\,t_3\,(75 - 20) = 2 \times 15 \times 75 \quad \text{or} \quad 110\,t_3 = 2250$$

$$\therefore \quad t_3 = 2250 / 110 = 20.45 \text{ say } 21 \text{ mm} \quad \textbf{Ans.}$$

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4. Total width of gib and cotter

Let B = Total width of gib and cotter.

Considering the failure of gib and cotter in double shear. We know that load (P),

$$50 \times 10^3 = 2\,B \times t \times \tau = 2\,B \times 20 \times 20 = 800\,B$$

$$\therefore \quad B = 50 \times 10^3 / 800 = 62.5 \text{ say } 65 \text{ mm} \quad \textbf{Ans.}$$

Since one gib is used, therefore width of gib,

$$b_1 = 0.55\,B = 0.55 \times 65 = 35.75 \text{ say } 36 \text{ mm} \quad \textbf{Ans.}$$

and width of cotter, $b = 0.45\,B = 0.45 \times 65 = 29.25$ say 30 mm **Ans.**

The other dimensions are fixed as follows :

$$t_4 = 1.25\,t_1 = 1.25 \times 15 = 18.75 \text{ say } 20 \text{ mm} \quad \textbf{Ans.}$$

$$l_1 = 2\,t_1 = 2 \times 15 = 30 \text{ mm} \quad \textbf{Ans.}$$

and $l_2 = 2.5\,t_1 = 2.5 \times 15 = 37.5$ say 40 mm **Ans.**

12.9 Design of Gib and Cotter Joint for Square Rods

Consider a gib and cotter joint for square rods as shown in Fig. 12.13. The rods may be subjected to a tensile or compressive load. All components of the joint are assumed to be of the same material.

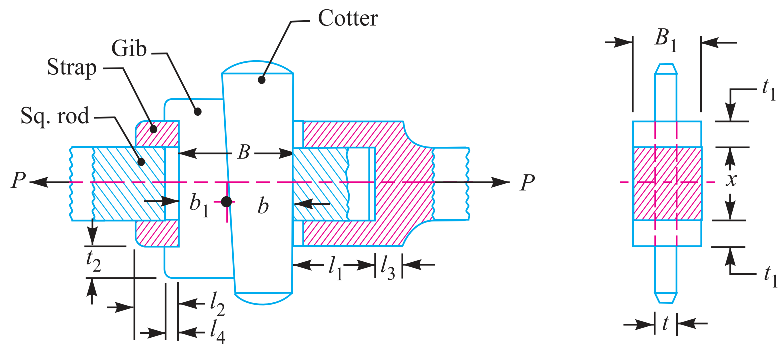

Fig. 12.13. Gib and cotter joint for square rods.

Let P = Load carried by the rods,

x = Each side of the rod,

B = Total width of gib and cotter,

B_1 = Width of the strap,

t = Thickness of cotter,

t_1 = Thickness of the strap, and

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses.

In designing a gib and cotter joint, the following modes of failure are considered.

1. Failure of the rod in tension

The rod may fail in tension due to the tensile load P. We know that

Area resisting tearing $= x \times x = x^2$

$\therefore$ Tearing strength of the rod

$$= x^2 \times \sigma_t$$

Equating this to the load (P), we have

$$P = x^2 \times \sigma_t$$

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From this equation, the side of the square rod (x) may be determined. The other dimensions are fixed as under :

Width of strap, B_1 = Side of the square rod = x

Thickness of cotter, $t = \frac{1}{4}$ width of strap $= \frac{B_1}{4}$

Thickness of gib = Thickness of cotter (t)

Height (t_2) and length of gib head (l_4)

= Thickness of cotter (t)

2. Failure of the gib and cotter in shearing

Since the gib and cotter are in double shear, therefore,

Area resisting failure $= 2\,B \times t$

and resisting strength $= 2\,B \times t \times \tau$

Equating this to the load (P), we have

$$P = 2B \times t \times \tau$$

From this equation, the width of gib and cotter (B) may be obtained. In the joint, as shown in Fig. 12.13, one gib is used, the proportions of which are

Width of gib, $b_1 = 0.55\,B$; and width of cotter, $b = 0.45\,B$

In case two gibs are used, then

Width of each gib $= 0.3\,B$; and width of cotter $= 0.4\,B$

3. Failure of the strap end in tension at the location of gib and cotter

Area resisting failure $= 2\,[B_1 \times t_1 - t_1 \times t] = 2\,[x \times t_1 - t_1 \times t]$ ... ($\because B_1 = x$)

$\therefore$ Resisting strength $= 2\,[\,x \times t_1 - t_1 \times t]\,\sigma_t$

Equating this to the load (P), we have

$$P = 2\,[x \times t_1 - t_1 \times t]\,\sigma_t$$

From this equation, the thickness of strap (t_1) may be determined.

4. Failure of the strap or gib in crushing

The strap or gib (at the strap hole) may fail due to crushing.

Area resisting failure $= 2\,t_1 \times t$

$\therefore$ Resisting strength $= 2\,t_1 \times t \times \sigma_c$

Equating this to the load (P), we have

$$P = 2\,t_1 \times t \times \sigma_c$$

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

5. Failure of the rod end in shearing

Since the rod is in double shear, therefore

Area resisting failure $= 2\,l_1 \times x$

$\therefore$ Resisting strength $= 2\,l_1 \times x \times \tau$

Equating this to the load (P), we have

$$P = 2\,l_1 \times x \times \tau$$

From this equation, the dimension l_1 may be determined.

6. Failure of the strap end in shearing

Since the length of rod (l_2) is in double shearing, therefore

Area resisting failure $= 2 \times 2\,l_2 \times t_1$

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∴ Resisting strength $= 2 \times 2\, l_2 \times t_1 \times \tau$

Equating this to the load (P), we have

$$P = 2 \times 2\, l_2 \times t_1 \times \tau$$

From this equation, the length of rod (l_2) may be determined. The length l_3 of the strap end is proportioned as $\frac{2}{3}$ rd of side of the rod. The clearance is usually kept 3 mm. The length of cotter is generally taken as 4 times the side of the rod.

Example 12.4. *Design a gib and cottor joint as shown in Fig. 12.13, to carry a maximum load of 35 kN. Assuming that the gib, cotter and rod are of same material and have the following allowable stresses :*

$$\sigma_t = 20 \text{ MPa} \; ; \; \tau = 15 \text{ MPa} \; ; \text{ and } \sigma_c = 50 \text{ MPa}$$

Solution. Given : P = 35 kN = 35 000 N ; σ_t = 20 MPa = 20 N/mm^2 ; τ = 15 MPa = 15 N/mm^2 ; σ_c = 50 MPa = 50 N/mm^2

1. Side of the square rod

Let x = Each side of the square rod.

Considering the failure of the rod in tension. We know that load (P),

$$35\,000 = x^2 \times \sigma_t = x^2 \times 20 = 20\, x^2$$

∴ $x^2 = 35\,000 / 20 = 1750$ or x = 41.8 say 42 mm **Ans.**

Other dimensions are fixed as follows :

Width of strap, $B_1 = x =$ 42 mm **Ans.**

Thickness of cotter, $t = \frac{B_1}{4} = \frac{42}{4}$ = 10.5 say 12 mm **Ans.**

Thickness of gib = Thickness of cotter = 12 mm **Ans.**

Height (t_2) and length of gib head (l_4)

= Thickness of cotter = 12 mm **Ans.**

2. Width of gib and cotter

Let B = Width of gib and cotter.

Considering the failure of the gib and cotter in double shear. We know that load (P),

$$35\,000 = 2\, B \times t \times \tau = 2\, B \times 12 \times 15 = 360\, B$$

∴ $B = 35\,000 / 360 = 97.2$ say 100 mm **Ans.**

Since one gib is used, therefore

Width of gib, $b_1 = 0.55\, B = 0.55 \times 100 = 55$ mm **Ans.**

and width of cotter, $b = 0.45\, B = 0.45 \times 100 = 45$ mm **Ans.**

3. Thickness of strap

Let t_1 = Thickness of strap.

Considering the failure of the strap end in tension at the location of the gib and cotter. We know that load (P),

$$35\,000 = 2\,(x \times t_1 - t_1 \times t)\, \sigma_t = 2\,(42 \times t_1 - t_1 \times 12)\, 20 = 1200\, t_1$$

∴ $t_1 = 35\,000 / 1200 = 29.1$ say 30 mm **Ans.**

Now the induced crushing stress may be checked by considering the failure of the strap or gib in crushing. We know that load (P),

$$35\,000 = 2\, t_1 \times t \times \sigma_c = 2 \times 30 \times 12 \times \sigma_c = 720\, \sigma_c$$

∴ $\sigma_c = 35\,000 / 720 = 48.6$ N/mm^2

Since the induced crushing stress is less than the given crushing stress, therefore the joint is safe.

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4. Length (l_1) of the rod

Considering the failure of the rod end in shearing. Since the rod is in double shear, therefore load (P),

$$35\,000 = 2\,l_1 \times x \times \tau = 2\,l_1 \times 42 \times 15 = 1260\,l_1$$

$$\therefore \quad l_1 = 35\,000 / 1260 = 27.7 \text{ say } 28 \text{ mm } \textbf{Ans.}$$

5. Length (l_2) of the rod

Considering the failure of the strap end in shearing. Since the length of the rod (l_2) is in double shear, therefore load (P),

$$35\,000 = 2 \times 2\,l_2 \times t_1 \times \tau = 2 \times 2\,l_2 \times 30 \times 15 = 1800\,l_2$$

$$\therefore \quad l_2 = 35\,000 / 1800 = 19.4 \text{ say } 20 \text{ mm } \textbf{Ans.}$$

Length (l_3) of the strap end

$$= \frac{2}{3} \times x = \frac{2}{3} \times 42 = 28 \text{ mm } \textbf{Ans.}$$

and length of cotter $= 4\,x = 4 \times 42 = 168$ mm **Ans.**

12.10 Design of Cotter Joint to Connect Piston Rod and Crosshead

The cotter joint to connect piston rod and crosshead is shown in Fig. 12.14. In such a type of joint, the piston rod is tapered in order to resist the thrust instead of being provided with a collar for the purpose. The taper may be from 1 in 24 to 1 in 12.

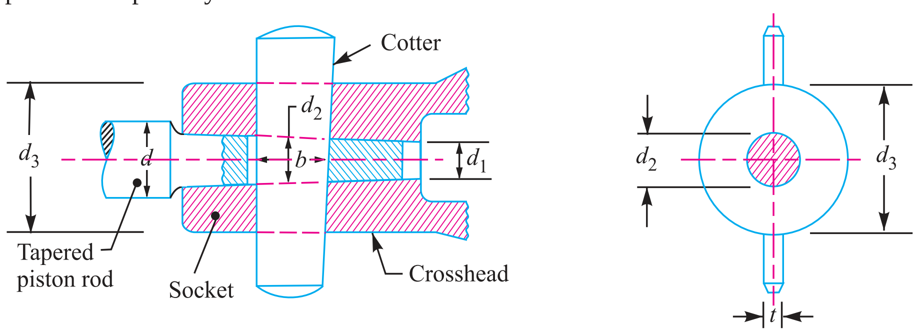

Fig. 12.14. Cotter joint to connect piston rod and crosshead.

Let
d = Diameter of parallel part of the piston rod,
d_1 = Diameter at tapered end of the piston,
d_2 = Diameter of piston rod at the cotter,
d_3 = Diameter of socket through the cotter hole,
b = Width of cotter at the centre,
t = Thickness of cotter,
σ_t, τ and σ_c = Permissible stresses in tension, shear and crushing respectively.

We know that maximum load on the piston,

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

where
D = Diameter of the piston, and
p = Effective steam pressure on the piston.

Let us now consider the various failures of the joint as discussed below :

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1. *Failure of piston rod in tension at cotter*

The piston rod may fail in tension at cotter due to the maximum load on the piston. We know that area resisting tearing at the cotter

$$= \frac{\pi}{4}(d_2)^2 - d_2 \times t$$

$\therefore$ Tearing strength of the piston rod at the cotter

$$= \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$$

Equating this to maximum load (P), we have

$$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$$

From this equation, the diameter of piston rod at the cotter (d_2) may be determined.

Note: The thickness of cotter (t) is taken as 0.3 d_2.

2. *Failure of cotter in shear*

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

$$= 2\,b \times t$$

and shearing strength of the cotter

$$= 2\,b \times t \times \tau$$

Equating this to maximum load (P), we have

$$P = 2\,b \times t \times \tau$$

From this equation, width of cotter (b) is obtained.

3. *Failure of the socket in tension at cotter*

We know that area that resists tearing of socket at cotter

$$= \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] - (d_3 - d_2)\,t$$

and tearing strength of socket at cotter

$$= \left[\frac{\pi}{4}\left\{(d_3)^2 - (d_2)^2\right\} - (d_3 - d_2)\,t\right]\sigma_t$$

Equating this to maximum load (P), we have

$$P = \left[\frac{\pi}{4}\left\{(d_3)^2 - (d_2)^2\right\} - (d_3 - d_2)\,t\right]\sigma_t$$

From this equation, diameter of socket (d_3) is obtained.

4. *Failure of socket in crushing*

We know that area that resists crushing of socket

$$= (d_3 - d_2)\,t$$

and crushing strength of socket

$$= (d_3 - d_2)\,t \times \sigma_c$$

Equating this to maximum load (P), we have

$$P = (d_3 - d_2)\,t \times \sigma_c$$

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

The length of the tapered portion of the piston rod (L) is taken as 2.2 d_2. The diameter of the parallel part of the piston rod (d) and diameter of the piston rod at the tapered end (d_1) may be obtained as follows :

$$d = d_2 + \frac{L}{2} \times \text{taper} \text{ ; and } d_1 = d_2 - \frac{L}{2} \times \text{taper}$$

Note: The taper on the piston rod is usually taken as 1 in 20.

Example 12.5. *Design a cotter joint to connect piston rod to the crosshead of a double acting steam engine. The diameter of the cylinder is 300 mm and the steam pressure is 1 N/mm². The allowable stresses for the material of cotter and piston rod are as follows :*

$$\sigma_t = 50 \text{ MPa} \text{ ; } \tau = 40 \text{ MPa} \text{ ; and } \sigma_c = 84 \text{ MPa}$$

Solution. Given : $D = 300$ mm ; $p = 1$ N/mm² ; $\sigma_t = 50$ MPa $= 50$ N/mm² ; $\tau = 40$ MPa $= 40$ N/mm² ; $\sigma_c = 84$ MPa $= 84$ N/mm²

We know that maximum load on the piston rod,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} (300)^2 \, 1 = 70\ 695 \text{ N}$$

The various dimensions for the cotter joint are obtained by considering the different modes of failure as discussed below :

1. *Diameter of piston rod at cotter*

Let d_2 = Diameter of piston rod at cotter, and

t = Thickness of cotter. It may be taken as $0.3\, d_2$.

Considering the failure of piston rod in tension at cotter. We know that load (P),

$$70\ 695 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right] \sigma_t = \left[\frac{\pi}{4}(d_2)^2 - 0.3\,(d_2)^2\right] 50 = 24.27\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 70\,695 / 24.27 = 2913$ or $d_2 = 53.97$ say 55 mm **Ans.**

and $t = 0.3\, d_2 = 0.3 \times 55 = 16.5$ mm **Ans.**

2. *Width of cotter*

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$70\ 695 = 2\, b \times t \times \tau = 2\, b \times 16.5 \times 40 = 1320\, b$$

$\therefore$ $b = 70\,695 / 1320 = 53.5$ say 54 mm **Ans.**

3. *Diameter of socket*

Let d_3 = Diameter of socket.

Considering the failure of socket in tension at cotter. We know that load (P),

$$70\ 695 = \left\{\frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] - (d_3 - d_2)\, t\right\} \sigma_t$$

$$= \left\{\frac{\pi}{4}\left[(d_3)^2 - (55)^2\right] - (d_3 - 55)\, 16.5\right\} 50$$

$$= 39.27\,(d_3)^2 - 118\,792 - 825\, d_3 + 45\,375$$

or $(d_3)^2 - 21\, d_3 - 3670 = 0$

$$\therefore \quad d_3 = \frac{21 \pm \sqrt{(21)^2 + 4 \times 3670}}{2} = \frac{21 \pm 123}{2} = 72 \text{ mm} \quad \text{...(Taking + ve sign)}$$

Let us now check the induced crushing stress in the socket. We know that load (P),

$$70\ 695 = (d_3 - d_2)\, t \times \sigma_c = (72 - 55)\, 16.5 \times \sigma_c = 280.5\, \sigma_c$$

$\therefore$ $\sigma_c = 70\,695 / 280.5 = 252$ N/mm²

Since the induced crushing is greater than the permissible value of 84 N/mm², therefore let us find the value of d_3 by substituting σ_c = 84 N/mm^2 in the above expression, *i.e.*

$$70\ 695 = (d_3 - 55)\ 16.5 \times 84 = (d_3 - 55)\ 1386$$

$$\therefore \quad d_3 - 55 = 70\ 695 / 1386 = 51$$

or $$d_3 = 55 + 51 = 106 \text{ mm } \textbf{Ans.}$$

We know the tapered length of the piston rod,

$$L = 2.2\ d_2 = 2.2 \times 55 = 121 \text{ mm } \textbf{Ans.}$$

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm } \textbf{Ans.}$$

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm } \textbf{Ans.}$$

Variable speed Knee-type milling machine.

12.11 Design of Cotter Foundation Bolt

The cotter foundation bolt is mostly used in conjunction with foundation and holding down bolts to fasten heavy machinery to foundations. It is generally used where an ordinary bolt or stud cannot be conveniently used. Fig. 12.15 shows the two views of the application of such a cotter foundation bolt. In this case, the bolt is dropped down from above and the cotter is driven in from the side. Now this assembly is tightened by screwing down the nut. It may be noted that two base plates (one under the nut and the other under the cotter) are used to provide more bearing area in order to take up the tightening load on the bolt as well as to distribute the same uniformly over the large surface.

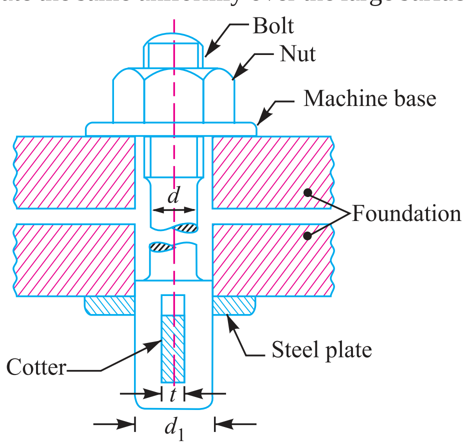

Fig. 12.15. Cotter foundation bolt.

Let d = Diameter of bolt,

d_1 = Diameter of the enlarged end of bolt,

t = Thickness of cotter, and

b = Width of cotter.

The various modes of failure of the cotter foundation bolt are discussed as below:

Contents

1. *Failure of bolt in tension*

The bolt may fail in tension due to the load (P). We know that area resisting tearing

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

∴ Tearing strength of the bolt

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

Equating this to the load (P), we have

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

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

2. *Failure of the enlarged end of the bolt in tension at the cotter*

We know that area resisting tearing

$$= \left[\frac{\pi}{4}(d_1)^2 - d_1 \times t\right]$$

∴ Tearing strength of the enlarged end of the bolt

$$= \left[\frac{\pi}{4}(d_1)^2 - d_1 \times t\right]\sigma_t$$

Equating this to the load (P), we have

$$P = \left[\frac{\pi}{4}(d_1)^2 - d_1 \times t\right]\sigma_t$$

From this equation, the diameter of the enlarged end of the bolt (d_1) may be determined.

Note: The thickness of cotter is usually taken as $d_1 / 4$.

3. *Failure of cotter in shear*

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

$$= 2\, b \times t$$

∴ Shearing strength of cotter

$$= 2\, b \times t \times \tau$$

Equating this to the load (P), we have

$$P = 2\, b \times t \times \tau$$

From this equation, the width of cotter (b) may be determined.

4. *Failure of cotter in crushing*

We know that area resisting crushing

$$= b \times t$$

∴ Crushing strength of cotter

$$= b \times t \times \sigma_c$$

Equating this to the load (P), we have

$$P = b \times t \times \sigma_c$$

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

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Example 12.6. *Design and draw a cottered foundation bolt which is subjected to a maximum pull of 50 kN. The allowable stresses are :*

σ_t = 80 MPa ; τ = 50 MPa ; and σ_c = 100 MPa

Solution. Given: $P = 50$ kN $= 50 \times 10^3$ N ; $\sigma_t = 80$ MPa $= 80$ N/mm^2 ; $\tau = 50$ MPa $= 50$ N/mm^2 ; $\sigma_c = 100$ MPa $= 100$ N/mm^2

1. Diameter of bolt

Let d = Diameter of bolt.

Considering the failure of the bolt in tension. We know that load (P),

$$50 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 80 = 62.84\, d^2$$

$$\therefore \quad d^2 = 50 \times 10^3 / 62.84 = 795.7 \text{ or } d = 28.2 \text{ say } 30 \text{ mm} \quad \textbf{Ans.}$$

2. Diameter of enlarged end of the bolt and thickness of cotter

Let d_1 = Diameter of enlarged end of the bolt, and

t = Thickness of cotter. It may be taken as $d_1 / 4$.

Considering the failure of the enlarged end of the bolt in tension at the cotter. We know that load (P),

$$50 \times 10^3 = \left[\frac{\pi}{4}(d_1)^2 - d_1 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_1)^2 - d_1 \times \frac{d_1}{4}\right] 80 = 42.84\,(d_1)^2$$

$$\therefore \quad (d_1)^2 = 50 \times 10^3 / 42.84 = 1167 \quad \text{or} \quad d_1 = 34 \text{ say } 36 \text{ mm} \quad \textbf{Ans.}$$

and
$$t = \frac{d_1}{4} = \frac{36}{4} = 9 \text{ mm} \quad \textbf{Ans.}$$

3. Width of cotter

Let b = Width of cotter.

Considering the failure of cotter in shear. Since the cotter is in double shear, therefore load (P),

$$50 \times 10^3 = 2\, b \times t \times \tau = 2\, b \times 9 \times 50 = 900\, b$$

$$\therefore \quad b = 50 \times 10^3 / 900 = 55.5 \text{ mm say } 60 \text{ mm} \quad \textbf{Ans.}$$

Let us now check the crushing stress induced in the cotter. Considering the failure of cotter in crushing. We know that load (P),

$$50 \times 10^3 = b \times t \times \sigma_c = 60 \times 9 \times \sigma_c = 540\, \sigma_c$$

$$\therefore \quad \sigma_c = 50 \times 10^3 / 540 = 92.5 \text{ N/mm}^2$$

Since the induced crushing stress is less than the permissible value of 100 N/mm^2, therefore the design is safe.

12.12 Knuckle Joint

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

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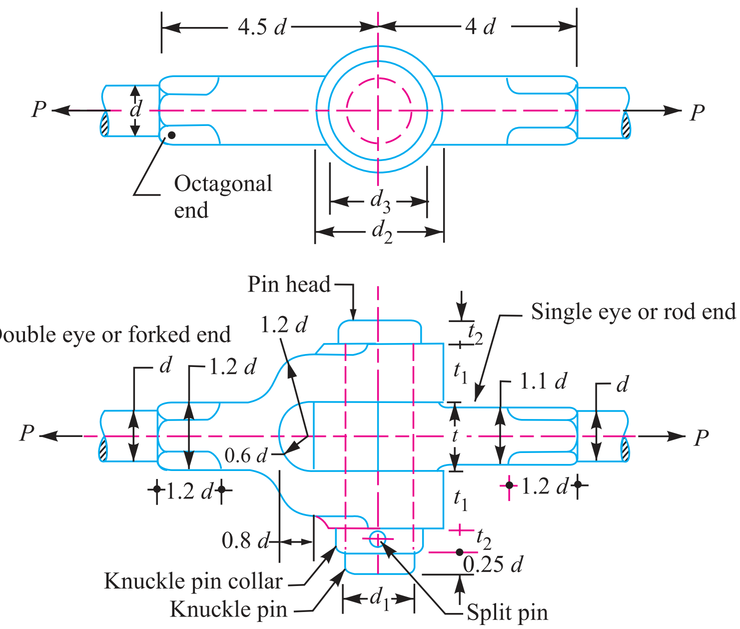
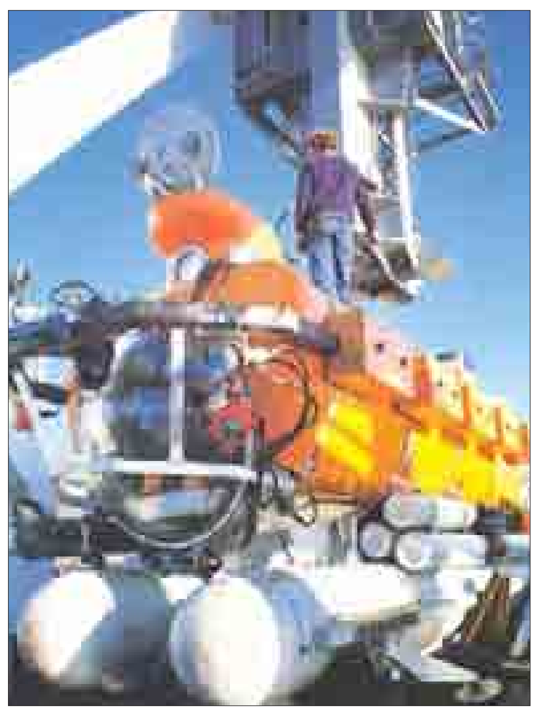


Fig. 12.16. Kunckle joint.

In knuckle joint (the two views of which are shown in Fig. 12.16), one end of one of the rods is made into an eye and the end of the other rod is formed into a fork with an eye in each of the fork leg. The knuckle pin passes through both the eye hole and the fork holes and may be secured by means of a collar and taper pin or spilt pin. The knuckle pin may be prevented from rotating in the fork by means of a small stop, pin, peg or snug. In order to get a better quality of joint, the sides of the fork and eye are machined, the hole is accurately drilled and pin turned. The material used for the joint may be steel or wrought iron.



Submersibles like this can work at much greater ocean depths and high pressures where divers cannot reach.

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

12.13 Dimensions of Various Parts of the Knuckle Joint

The dimensions of various parts of the knuckle joint are fixed by empirical relations as given below. It may be noted that all the parts should be made of the same material *i.e.* mild steel or wrought iron.

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

$$d_1 = d$$

Outer diameter of eye,

$$d_2 = 2\,d$$

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Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d$$

Thickness of single eye or rod end,

$$t = 1.25\,d$$

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

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

Other dimensions of the joint are shown in Fig. 12.16.

12.14 Methods of Failure of Knuckle Joint

Consider a knuckle joint as shown in Fig. 12.16.

Let P = Tensile load acting on the rod,

d = Diameter of the rod,

d_1 = Diameter of the pin,

d_2 = Outer diameter of eye,

t = Thickness of single eye,

t_1 = Thickness of fork.

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

In determining the strength of the joint for the various methods of failure, it is assumed that

1. There is no stress concentration, and
2. The load is uniformly distributed over each part of the joint.

Due to these assumptions, the strengths are approximate, however they serve to indicate a well proportioned joint. Following are the various methods of failure of the joint :

1. Failure of the solid rod in tension

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

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

Equating this to the load (P) acting on the rod, we have

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

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

2. Failure of the knuckle pin in shear

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

$$= 2 \times \frac{\pi}{4}\,(d_1)^2$$

and the shear strength of the pin

$$= 2 \times \frac{\pi}{4}\,(d_1)^2\ \tau$$

Equating this to the load (P) acting on the rod, we have

$$P = 2 \times \frac{\pi}{4}\,(d_1)^2\ \tau$$

From this equation, diameter of the knuckle pin (d_1) is obtained. This assumes that there is no slack and clearance between the pin and the fork and hence there is no bending of the pin. But, in actual practice, the knuckle pin is loose in forks in order to permit angular movement of one with respect to the other, therefore the pin is subjected to bending in addition to shearing. By making the diameter of knuckle pin equal to the diameter of the rod (*i.e.*, $d_1 = d$), a margin of strength is provided to allow for the bending of the pin.

In case, the stress due to bending is taken into account, it is assumed that the load on the pin is uniformly distributed along the middle portion (*i.e.* the eye end) and varies uniformly over the forks as shown in Fig. 12.17. Thus in the forks, a load $P/2$ acts through a distance of $t_1 / 3$ from the inner edge and the bending moment will be maximum at the centre of the pin. The value of maximum bending moment is given by

$$M = \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2}\right) - \frac{P}{2} \times \frac{t}{4}$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2} - \frac{t}{4}\right)$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)$$

and section modulus, $Z = \dfrac{\pi}{32}(d_1)^3$

∴ Maximum bending (tensile) stress,

$$\sigma_t = \frac{M}{Z} = \frac{\frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)}{\frac{\pi}{32}(d_1)^3}$$

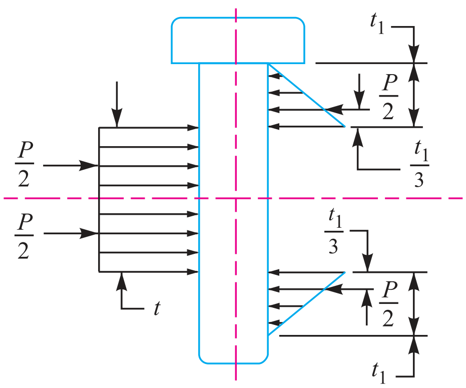

Fig. 12.17. Distribution of load on the pin.

From this expression, the value of d_1 may be obtained.

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

The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing $= (d_2 - d_1)\, t$

∴ Tearing strength of single eye or rod end

$$= (d_2 - d_1)\, t \times \sigma_t$$

Equating this to the load (P) we have

$$P = (d_2 - d_1)\, t \times \sigma_t$$

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

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

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing $= (d_2 - d_1)\, t$

∴ Shearing strength of single eye or rod end

$$= (d_2 - d_1)\, t \times \tau$$

Equating this to the load (P), we have

$$P = (d_2 - d_1)\, t \times \tau$$

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

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

The single eye or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times t$

∴ Crushing strength of single eye or rod end

$$= d_1 \times t \times \sigma_c$$

Equating this to the load (P), we have

$$\therefore \quad P = d_1 \times t \times \sigma_c$$

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

6. *Failure of the forked end in tension*

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

$$= (d_2 - d_1) \times 2\, t_1$$

∴ Tearing strength of the forked end

$$= (d_2 - d_1) \times 2\, t_1 \times \sigma_t$$

Equating this to the load (P), we have

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

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

7. *Failure of the forked end in shear*

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

$$= (d_2 - d_1) \times 2t_1$$

∴ Shearing strength of the forked end

$$= (d_2 - d_1) \times 2t_1 \times \tau$$

Equating this to the load (P), we have

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

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

8. *Failure of the forked end in crushing*

The forked end or pin may fail in crushing due to the tensile load. We know that area resisting crushing $= d_1 \times 2\, t_1$

∴ Crushing strength of the forked end

$$= d_1 \times 2\, t_1 \times \sigma_c$$

Equating this to the load (P), we have

$$P = d_1 \times 2\, t_1 \times \sigma_c$$

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

Note: From the above failures of the joint, we see that the thickness of fork (t_1) should be equal to half the thickness of single eye ($t / 2$). But, in actual practice $t_1 > t / 2$ in order to prevent deflection or spreading of the forks which would introduce excessive bending of pin.

12.15 Design Procedure of Knuckle Joint

The empirical dimensions as discussed in Art. 12.13 have been formulated after wide experience on a particular service. These dimensions are of more practical value than the theoretical analysis. Thus, a designer should consider the empirical relations in designing a knuckle joint. The following procedure may be adopted :

1. First of all, find the diameter of the rod by considering the failure of the rod in tension. We know that tensile load acting on the rod,

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

where d = Diameter of the rod, and

σ_t = Permissible tensile stress for the material of the rod.

2. After determining the diameter of the rod, the diameter of pin (d_1) may be determined by considering the failure of the pin in shear. We know that load,

$$P = 2 \times \frac{\pi}{4} (d_1)^2 \ \tau$$

A little consideration will show that the value of d_1 as obtained by the above relation is less than the specified value (*i.e.* the diameter of rod). So fix the diameter of the pin equal to the diameter of the rod.

3. Other dimensions of the joint are fixed by empirical relations as discussed in Art. 12.13.

4. The induced stresses are obtained by substituting the empirical dimensions in the relations as discussed in Art. 12.14.

In case the induced stress is more than the allowable stress, then the corresponding dimension may be increased.

Example 12.7. *Design a knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression.*

Solution. Given : P = 150 kN = 150 × 10^3 N ; σ_t = 75 MPa = 75 N/mm^2 ; τ = 60 MPa = 60 N/mm^2 ; σ_c = 150 MPa = 150 N/mm^2

The knuckle joint is shown in Fig. 12.16. The joint is designed by considering the various methods of failure as discussed below :

1. Failure of the solid rod in tension

Let d = Diameter of the rod.

We know that the load transmitted (P),

$$150 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 75 = 59\, d^2$$

$\therefore$ $d^2 = 150 \times 10^3 / 59 = 2540$ or $d = 50.4$ say 52 mm **Ans.**

Now the various dimensions are fixed as follows :

Diameter of knuckle pin,

$$d_1 = d = 52 \text{ mm}$$

Outer diameter of eye, $d_2 = 2\,d = 2 \times 52 = 104$ mm

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\,d = 1.5 \times 52 = 78 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\,d = 1.25 \times 52 = 65 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\,d = 0.75 \times 52 = 39$ say 40 mm

Thickness of pin head, $t_2 = 0.5\,d = 0.5 \times 52 = 26$ mm

2. Failure of the knuckle pin in shear

Since the knuckle pin is in double shear, therefore load (P),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\, \tau = 2 \times \frac{\pi}{4} \times (52)^2\, \tau = 4248\, \tau$$

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$$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$$

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

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

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

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

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

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (104 - 52)\, 2 \times 40 \times \sigma_t = 4160\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. *Failure of the forked end in shear*

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \tau = (104 - 52)\, 2 \times 40 \times \tau = 4160\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. *Failure of the forked end in crushing*

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\, t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

Example 12.8. *Design a knuckle joint for a tie rod of a circular section to sustain a maximum pull of 70 kN. The ultimate strength of the material of the rod against tearing is 420 MPa. The ultimate tensile and shearing strength of the pin material are 510 MPa and 396 MPa respectively. Determine the tie rod section and pin section. Take factor of safety = 6.*

Solution. Given : $P = 70$ kN $= 70\,000$ N ; σ_{tu} for rod $= 420$ MPa ; *σ_{tu} for pin $= 510$ MPa ; $\tau_u = 396$ MPa ; $F.S. = 6$

We know that the permissible tensile stress for the rod material,

$$\sigma_t = \frac{\sigma_{tu} \text{ for rod}}{F.S.} = \frac{420}{6} = 70 \text{ MPa} = 70 \text{ N/mm}^2$$

and permissible shear stress for the pin material,

$$\tau = \frac{\tau_u}{F.S.} = \frac{396}{6} = 66 \text{ MPa} = 66 \text{ N/mm}^2$$

* Superfluous data.

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We shall now consider the various methods of failure of the joint as discussed below:

1. *Failure of the rod in tension*

Let d = Diameter of the rod.

We know that the load (P),

$$70\,000 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 70 = 55\, d^2$$

$$\therefore \quad d^2 = 70\,000 / 55 = 1273 \quad \text{or} \quad d = 35.7 \text{ say } 36 \text{ mm} \quad \textbf{Ans.}$$

The other dimensions of the joint are fixed as given below :

Diameter of the knuckle pin,

$$d_1 = d = 36 \text{ mm}$$

Outer diameter of the eye,

$$d_2 = 2\, d = 2 \times 36 = 72 \text{ mm}$$

Diameter of knuckle pin head and collar,

$$d_3 = 1.5\, d = 1.5 \times 36 = 54 \text{ mm}$$

Thickness of single eye or rod end,

$$t = 1.25\, d = 1.25 \times 36 = 45 \text{ mm}$$

Thickness of fork, $t_1 = 0.75\, d = 0.75 \times 36 = 27$ mm

Now we shall check for the induced streses as discussed below :

2. *Failure of the knuckle pin in shear*

Since the knuckle pin is in double shear, therefore load (P),

$$70\,000 = 2 \times \frac{\pi}{4}\, (d_1)^2\, \tau = 2 \times \frac{\pi}{4}\, (36)^2\, \tau = 2036\, \tau$$

$$\therefore \quad \tau = 70\,000 / 2036 = 34.4 \text{ N/mm}^2$$

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

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$70\,000 = (d_2 - d_1)\, t \times \sigma_t = (72 - 36)\, 45\, \sigma_t = 1620\, \sigma_t$$

$$\therefore \quad \sigma_t = 70\,000 / 1620 = 43.2 \text{ N/mm}^2$$

4. *Failure of the forked end in tension*

The forked end may fail in tension due to the load. We know that load (P),

$$70\,000 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (72 - 36) \times 2 \times 27 \times \sigma_t = 1944\, \sigma_t$$

$$\therefore \quad \sigma_t = 70\,000 / 1944 = 36 \text{ N/mm}^2$$

From above we see that the induced stresses are less than given permissible stresses, therefore the joint is safe.

12.16 Adjustable Screwed Joint for Round Rods (Turnbuckle)

Turnbuckle.

Sometimes, two round tie rods, as shown in Fig. 12.18, are connected by means of a coupling known as a ***turnbuckle***. In this type of joint, one of the rods has right hand threads and the other rod has left hand threads. The rods are screwed to a coupler which has a threaded hole. The coupler is of hexagonal or rectangular shape in the centre and round at both the ends in order to facilitate the rods to tighten or loosen with the help of a spanner when required. Sometimes

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instead of a spanner, a round iron rod may be used. The iron rod is inserted in a hole in the coupler as shown dotted in Fig. 12.18.

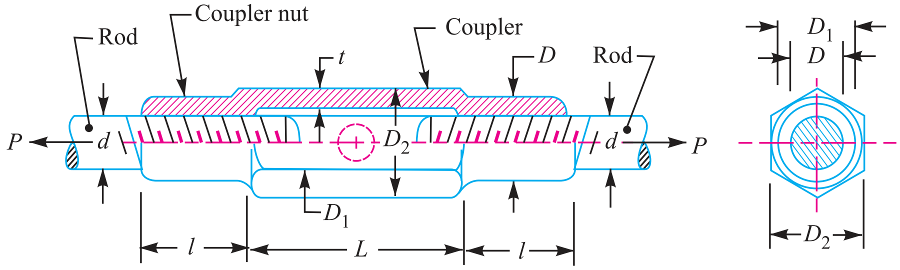

Fig. 12.18. Turnbuckle.

A turnbuckle commonly used in engineering practice (mostly in aeroplanes) is shown in Fig. 12.19. This type of turnbuckle is made hollow in the middle to reduce its weight. In this case, the two ends of the rods may also be seen. It is not necessary that the material of the rods and the turnbuckle may be same or different. It depends upon the pull acting on the joint.

12.17 Design of Turnbuckle

Consider a turnbuckle, subjected to an axial load P, as shown in Fig. 12.19. Due to this load, the threaded rod will be subjected to tensile stress whose magnitude is given by

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

where d_c = Core diameter of the threaded rod.

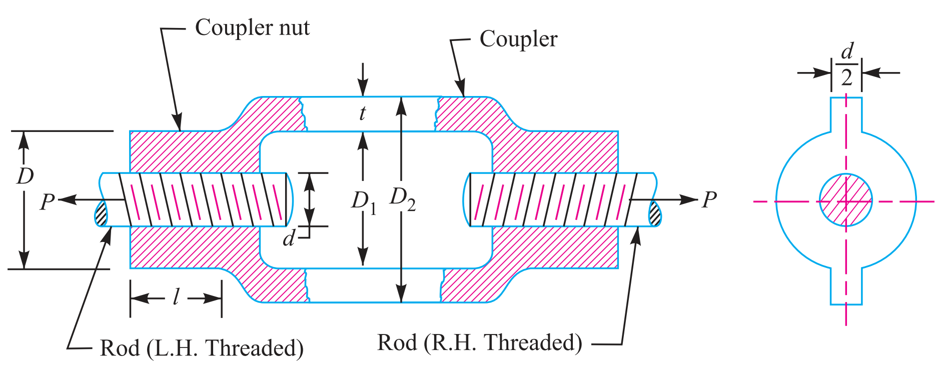

Fig. 12.19. Turnbuckle.

In order to drive the rods, the torque required is given by

$$T = P \tan(\alpha + \phi)\frac{d_p}{2}$$

where α = Helix angle,

$\tan\phi$ = Coefficient of friction between the threaded rod and the coupler nut, and

d_p = Pitch diameter or mean diameter of the threaded rod.

∴ Shear stress produced by the torque,

$$\tau = \frac{T}{J} \times \frac{d_p}{2} = \frac{P \tan(\alpha + \phi) \frac{d_p}{2}}{\frac{\pi}{32}(d_p)^4} \times \frac{d_p}{2} = P \tan(\alpha + \phi) \times \frac{8}{\pi (d_p)^2}$$

$$= \frac{8P}{\pi (d_p)^2}\left(\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha \times \tan\phi}\right)$$

The usual values of tan α, tan φ and d_p are as follows :

$\tan\alpha = 0.03$, $\tan\phi = 0.2$, and $d_p = 1.08\, d_c$

Substituting these values in the above expression, we get

$$\tau = \frac{8P}{\pi(1.08\, d_c)^2}\left[\frac{0.03 + 0.2}{1 - 0.03 \times 0.2}\right] = \frac{8P}{4\pi (d_c)^2} = \frac{P}{2A} = \frac{\sigma_t}{2}$$

$$\ldots\left[\because A = \frac{\pi}{4}(d_c)^2\right]$$

Since the threaded rod is subjected to tensile stress as well as shear stress, therefore maximum principal stress,

$$\sigma_{t\,(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + 4\tau^2} = \frac{\sigma_t}{2} + \frac{1}{2}\sqrt{(\sigma_t)^2 + (\sigma_t)^2} \qquad \ldots\left(\because \tau = \frac{\sigma_t}{2}\right)$$

$$= 0.5\,\sigma_t + 0.707\,\sigma_t = 1.207\,\sigma_t = 1.207\,P/A$$

Giving a margin for higher coefficient of friction, the maximum principal stress may be taken as 1.3 times the normal stress. Therefore for designing a threaded section, we shall take the design load as 1.3 times the normal load, *i.e.*

Design load, $P_d = 1.3\,P$

The following procedure may be adopted in designing a turn-buckle :

1. *Diameter of the rods*

The diameter of the rods (*d*) may be obtained by considering the tearing of the threads of the rods at their roots. We know that

Tearing resistance of the threads of the rod

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

Equating the design load (P_d) to the tearing resistance of the threads, we have

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

where d_c = Core diameter of the threads of the rod, and

σ_t = Permissible tensile stress for the material of the rod.

From the above expression, the core diameter of the threads may be obtained. The nominal diameter of the threads (or diameter of the rod) may be found from Table 11.1, corresponding to the core diameter, assuming coarse threads.

2. *Length of the coupler nut*

The length of the coupler nut (*l*) is obtained by considering the shearing of the threads at their roots in the coupler nut. We know that

Shearing resistance of the threads of the coupler nut

$$= (\pi\, d_c \times l)\,\tau$$

where τ = Shear stress for the material of the coupler nut.

Equating the design load to the shearing resistance of the threads in the coupler nut, we have

$$P_d = (\pi\, d_c \times l\,)\,\tau$$

From this expression, the value of l may be calculated. In actual practice, the length of coupler nut (l) is taken d to 1.25 d for steel nuts and 1.5 d to 2 d for cast iron and softer material nut. The length of the coupler nut may also be checked for crushing of threads. We know that

Crushing resistance of the threads in the coupler nut

$$= \frac{\pi}{4}\left[(d)^2 - (d_c)^2\right] n \times l \times \sigma_c$$

where σ_c = Crushing stress induced in the coupler nut, and

n = Number of threads per mm length.

Equating the design load to the crushing resistance of the threads, we have

$$P_d = \frac{\pi}{4}\left[(d)^2 - (d_c)^2\right] n \times l \times \sigma_c$$

From this expression, the induced σ_c may be checked.

3. *Outside diameter of the coupler nut*

The outside diameter of the coupler nut (D) may be obtained by considering the tearing at the coupler nut. We know that

Tearing resistance at the coupler nut

$$= \frac{\pi}{4}(D^2 - d^2)\,\sigma_t$$

where σ_t = Permissible tensile stress for the material of the coupler nut.

Equating the axial load to the tearing resistance at the coupler nut, we have

$$P = \frac{\pi}{4}(D^2 - d^2)\,\sigma_t$$

From this expression, the value of D may be calculated. In actual practice, the diameter of the coupler nut (D) is taken from 1.25 d to 1.5 d.

4. *Outside diameter of the coupler*

The outside diameter of the coupler (D_2) may be obtained by considering the tearing of the coupler. We know that

Tearing resistance of the coupler

$$= \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_t$$

where D_1 = Inside diameter of the coupler. It is generally taken as (d + 6 mm), and

σ_t = Permissible tensile stress for the material of the coupler.

Equating the axial load to the tearing resistance of the coupler, we have

$$P = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_t$$

From this expression, the value of D_2 may be calculated. In actual practice, the outside diameter of the coupler (D_2) is taken as 1.5 d to 1.7 d. If the section of the coupler is to be made hexagonal or rectangular to fit the spanner, it may be circumscribed over the circle of outside diameter D_2.

5. The length of the coupler between the nuts (L) depends upon the amount of adjustment required. It is usually taken as 6 d.

6. The thickness of the coupler is usually taken as t = 0.75 d, and thickness of the coupler nut, t_1 = 0.5 d.

Example 12.9. *The pull in the tie rod of an iron roof truss is 50 kN. Design a suitable adjustable screwed joint. The permissible stresses are 75 MPa in tension, 37.5 MPa in shear and 90 MPa in crushing.*

Solution. Given : $P = 50$ kN $= 50 \times 10^3$ N ; $\sigma_t = 75$ MPa $= 75$ N/mm^2 ; $\tau = 37.5$ MPa $= 37.5$ N/mm^2

We know that the design load for the threaded section,

$$P_d = 1.3\,P = 1.3 \times 50 \times 10^3 = 65 \times 10^3 \text{ N}$$

An adjustable screwed joint, as shown in Fig. 12.19, is suitable for the given purpose. The various dimensions for the joint are determined as discussed below :

1. *Diameter of the tie rod*

Let d = Diameter of the tie rod, and

d_c = Core diameter of threads on the tie rod.

Considering tearing of the threads on the tie rod at their roots.

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(d_c)^2\,75 = 59\,(d_c)^2$$

$$\therefore \quad (d_c)^2 = 65 \times 10^3 / 59 = 1100 \quad \text{or} \quad d_c = 33.2 \text{ mm}$$

From Table 11.1 for coarse series, we find that the standard core diameter is 34.093 mm and the corresponding nominal diameter of the threads or diameter of tie rod,

$$d = 39 \text{ mm}$$ **Ans.**

2. *Length of the coupler nut*

Let l = Length of the coupler nut.

Considering the shearing of threads at their roots in the coupler nut. We know that design load (P_d),

$$65 \times 10^3 = (\pi\, d_c.l)\,\tau = \pi \times 34.093 \times l \times 37.5 = 4107\,l$$

$$\therefore \quad l = 65 \times 10^3 / 4017 = 16.2 \text{ mm}$$

Since the length of the coupler nut is taken from d to 1.25 d, therefore we shall take

$$l = d = 39 \text{ mm}$$ **Ans.**

We shall now check the length of the coupler nut for crushing of threads.

From Table 11.1 for coarse series, we find that the pitch of the threads is 4 mm. Therefore the number of threads per mm length,

$$n = 1/4 = 0.25$$

We know that design load (P_d),

$$65 \times 10^3 = \frac{\pi}{4}\left[(d)^2 - (d_c)^2\right] n \times l \times \sigma_c$$

$$= \frac{\pi}{4}\left[(39)^2 - (34.093)^2\right] 0.25 \times 39 \times \sigma_c = 2750\,\sigma_c$$

$$\therefore \quad \sigma_c = 65 \times 10^3 / 2750 = 23.6 \text{ N/mm}^2 = 23.6 \text{ MPa}$$

Since the induced crushing stress in the threads of the coupler nut is less than the permissible stress, therefore the design is satisfactory.

3. *Outside diameter of the coupler nut*

Let D = Outside diameter of the coupler nut

Considering tearing of the coupler nut. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}(D^2 - d^2)\sigma_t$$

$$= \frac{\pi}{4}\left[D^2 - (39)^2\right] 75 = 59\left[D^2 - (39)^2\right]$$

or $\qquad D^2 - (39)^2 = 50 \times 10^3 / 59 = 848$

$\therefore \qquad D^2 = 848 + (39)^2 = 2369 \quad$ or $\quad D = 48.7$ say 50 mm **Ans.**

Since the minimum outside diameter of coupler nut is taken as 1.25 d (*i.e.* 1.25 × 39 = 48.75 mm), therefore the above value of D is satisfactory.

4. *Outside diameter of the coupler*

Let $\qquad D_2$ = Outside diameter of the coupler, and

D_1 = Inside diameter of the coupler = d + 6 mm = 39 + 6 = 45 mm

Considering tearing of the coupler. We know that axial load (P),

$$50 \times 10^3 = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2\right]\sigma_t = \frac{\pi}{4}\left[(D_2)^2 - (45)^2\right] 75 = 59\left[(D_2)^2 - (45)^2\right]$$

$\therefore \qquad (D_2)^2 = 50 \times 10^3 / 59 + (45)^2 = 2873$ or $D_2 = 53.6$ mm

Since the minimum outside diameter of the coupler is taken as 1.5 d (*i.e.* 1.5 × 39 = 58.5 say 60 mm), therefore we shall take

D_2 = 60 mm **Ans.**

5. Length of the coupler between nuts,

$L = 6\,d = 6 \times 39 = 234$ mm **Ans.**

6. Thickness of the coupler,

$t_1 = 0.75\,d = 0.75 \times 39 = 29.25$ say 30 mm **Ans.**

and thickness of the coupler nut,

$t = 0.5\,d = 0.5 \times 39 = 19.5$ say 20 mm **Ans.**

EXERCISES

1. Design a cotter joint to connect two mild steel rods for a pull of 30 kN. The maximum permissible stresses are 55 MPa in tension ; 40 MPa in shear and 70 MPa in crushing. Draw a neat sketch of the joint designed.

[Ans. d = 22 mm; d_2 = 32 mm ; t = 14 mm ; d_1 = 44 mm ; b = 30 mm ; a = 12 mm ; d_4 = 65 mm ; c = 12 mm ; d_3 = 40 mm ; t_1 = 8 mm]

2. Two rod ends of a pump are joined by means of a cotter and spigot and socket at the ends. Design the joint for an axial load of 100 kN which alternately changes from tensile to compressive. The allowable stresses for the material used are 50 MPa in tension, 40 MPa in shear and 100 MPa in crushing.

[Ans. d = 51 mm ; d_2 = 62 mm ; t = 16 mm ; d_1 = 72 mm ; b = 78 mm ; a = 20 mm ; d_3 = 83 mm ; d_4 = 125 mm ; c = 16 mm ; t_1 = 13 mm]

3. Two mild steel rods 40 mm diameter are to be connected by a cotter joint. The thickness of the cotter is 12 mm. Calculate the dimensions of the joint, if the maximum permissible stresses are: 46 MPa in tension ; 35 MPa in shear and 70 MPa in crushing.

[Ans. d_2 = 30 mm ; d_1 = 48 mm ; b = 70 mm ; a = 27.5 mm ; d_4 = 100 mm ; c = 12 mm ; d_3 = 44.2 mm ; t = 35 mm ; t_1 = 13.5 mm]

4. The big end of a connecting rod is subjected to a load of 40 kN. The diameter of the circular part adjacent to the strap is 50 mm.

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Design the joint assuming the permissible tensile stress in the strap as 30 MPa and permissible shear stress in the cotter and gib as 20 MPa.

[Ans. B_1 = 50 mm ; t = 15 mm ; t_1 = 15 mm ; t_3 = 22 mm ; B = 70 mm]

5. Design a cotter joint to connect a piston rod to the crosshead. The maximum steam pressure on the piston rod is 35 kN. Assuming that all the parts are made of the same material having the following permissible stresses :

σ_1 = 50 MPa ; τ = 60 MPa and σ_c = 90 MPa.

[Ans. d_2 = 40 mm ; t = 12 mm ; d_3 = 75 mm ; L = 88 mm ; d = 44 mm ; d_1 = 38 mm]

6. Design and draw a cotter foundation bolt to take a load of 90 kN. Assume the permissible stresses as follows :

σ_t = 50 MPa, τ = 60 MPa and σ_c = 100 MPa.

[Ans. d = 50 mm ; d_1 = 60 mm ; t = 15 mm ; b = 60 mm]

7. Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.

[Ans. $d = d_1$ = 23 mm ; d_2 = 46 mm ; d_3 = 35 mm ; t = 29 mm ; t_1 = 18 mm]

8. A knuckle joint is required to withstand a tensile load of 25 kN. Design the joint if the permissible stresses are :

σ_t = 56 MPa ; τ = 40 MPa and σ_c = 70 MPa.

[Ans. $d = d_1$ = 28 mm ; d_2 = 56 mm ; d_3 = 42 mm ; t_1 = 21 mm]

9. The pull in the tie rod of a roof truss is 44 kN. Design a suitable adjustable screw joint. The permissible tensile and shear stresses are 75 MPa and 37.5 MPa respectively. Draw full size two suitable views of the joint. **[Ans. d = 36 mm ; l = 11 mm ; D = 45 mm ; D_2 = 58 mm]**

QUESTIONS

1. What is a cotter joint? Explain with the help of a neat sketch, how a cotter joint is made ?
2. What are the applications of a cottered joint ?
3. Discuss the design procedure of spigot and socket cotter joint.
4. Why gibs are used in a cotter joint? Explain with the help of a neat sketch the use of single and double gib.
5. Describe the design procedure of a gib and cotter joint.
6. Distinguish between cotter joint and knuckle joint.
7. Sketch two views of a knuckle joint and write the equations showing the strength of joint for the most probable modes of failure.
8. Explain the purpose of a turn buckle. Describe its design procedure.

OBJECTIVE TYPE QUESTIONS

1. A cotter joint is used to transmit

(*a*) axial tensile load only (*b*) axial compressive load only

(*c*) combined axial and twisting loads (*d*) axial tensile or compressive loads

2. The taper on cotter varies from

(*a*) 1 in 15 to 1 in 10 (*b*) 1 in 24 to 1 in 20

(*c*) 1 in 32 to 1 in 24 (*d*) 1 in 48 to 1 in 24

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3. Which of the following cotter joint is used to connect strap end of a connecting rod ?

(*a*) Socket and spigot cotter joint (*b*) Sleeve and cotter joint

(*c*) Gib and cotter joint (*d*) none of these

4. In designing a sleeve and cotter joint, the outside diameter of the sleeve is taken as

(*a*) 1.5 d (*b*) 2.5 d

(*c*) 3 d (*d*) 4 d

where d = Diameter of the rod.

5. The length of cotter, in a sleeve and cotter joint, is taken as

(*a*) 1.5 d (*b*) 2.5 d

(*c*) 3 d (*d*) 4 d

6. In a gib and cotter joint, the thickness of gib is ....thickness of cotter.

(*a*) more than (*b*) less than (*c*) equal to

7. When one gib is used in a gib and cotter joint, then the width of gib should be taken as

(*a*) 0.45 B (*b*) 0.55 B

(*c*) 0.65 B (*d*) 0.75 B

where B = Total width of gib and cotter.

8. In a steam engine, the piston rod is usually connected to the crosshead by means of a

(*a*) knuckle joint (*b*) universal joint

(*c*) flange coupling (*d*) cotter joint

9. In a steam engine, the valve rod is connected to an eccentric by means of a

(*a*) knuckle joint (*b*) universal joint

(*c*) flange coupling (*d*) cotter joint

10. In a turn buckle, if one of the rods has left hand threads, then the other rod will have

(*a*) right hand threads (*b*) left hand threads

(*c*) pointed threads (*d*) multiple threads

ANSWERS

1. (*d*)	**2.** (*d*)	**3.** (*c*)	**4.** (*b*)	**5.** (*d*)
6. (*c*)	**7.** (*b*)	**8.** (*d*)	**9.** (*a*)	**10.** (*a*)

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