UNIT III TEMPORARY AND PERMANENT JOINTS

3.1 Threaded fasteners

3.1.1 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.

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.

3.1.2 Important Terms Used in Screw Threads

The following terms used in screw threads, as shown in Fig. 3.1, are important from the subject point of view :

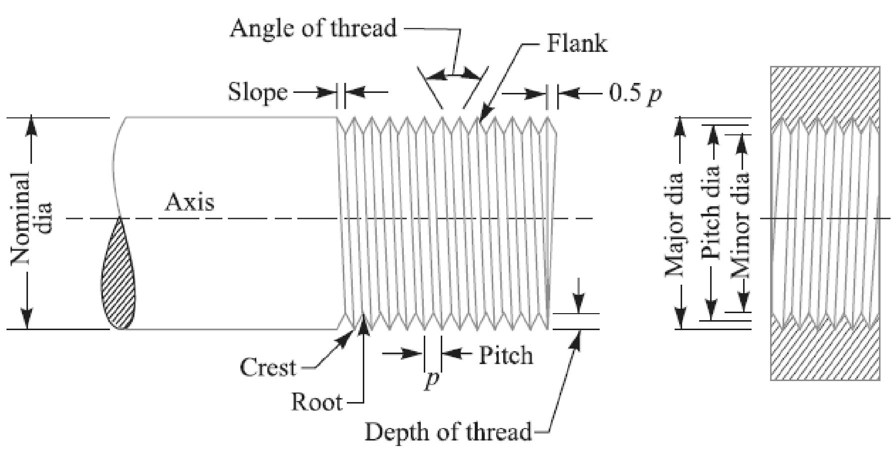


Fig. 3.1. Terms used in screw threads.

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,
Pitch =1/ No. of threads per unit length of screw
5. *Lead*. It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.
6. *Crest*. It is the top surface of the thread.
7. *Root*. It is the bottom surface created by the two adjacent flanks of the thread.
8. *Depth of thread*. It is the perpendicular distance between the crest and root.
9. *Flank*. It is the surface joining the crest and root.
10. *Angle of thread*. It is the angle included by the flanks of the thread.
11. *Slope.* It is half the pitch of the thread.

3.1.3 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. 3.2.

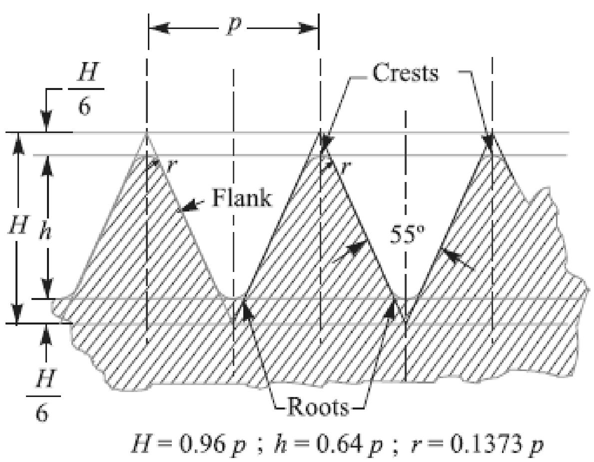


Fig. 3.2. British standard whitworth (B.S.W) thread.

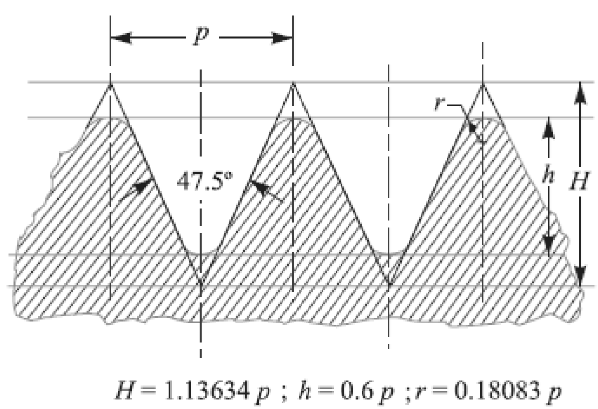


Fig. 3.3. British association (B.A.) thread.

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. 3.3. 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

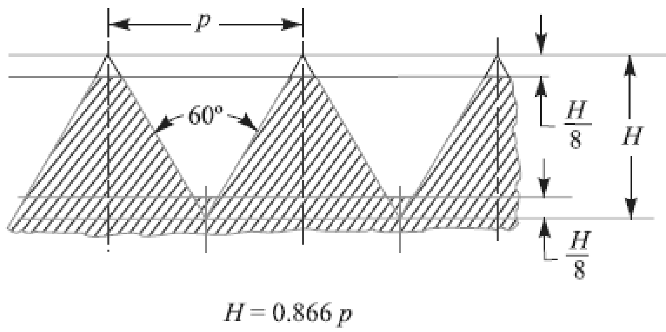


Fig. 3.4. American national standard thread.

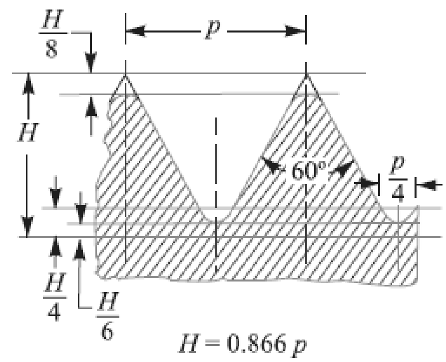


Fig. 3.5. Unified standard thread.

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. 3.5.

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. 3.6.

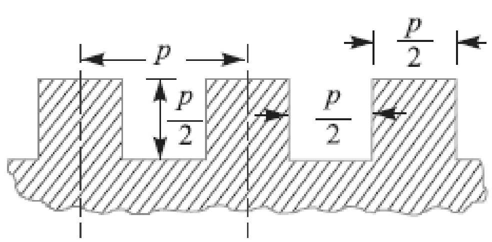


Fig. 3.6. Square thread.

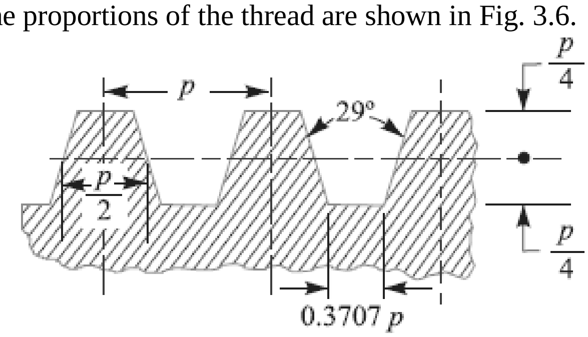


Fig. 3.7. Acme thread.

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. 3.7.

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.

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. 3.9.

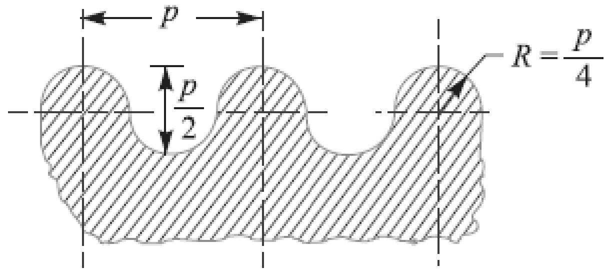


Fig. 3.8. Knuckle thread.

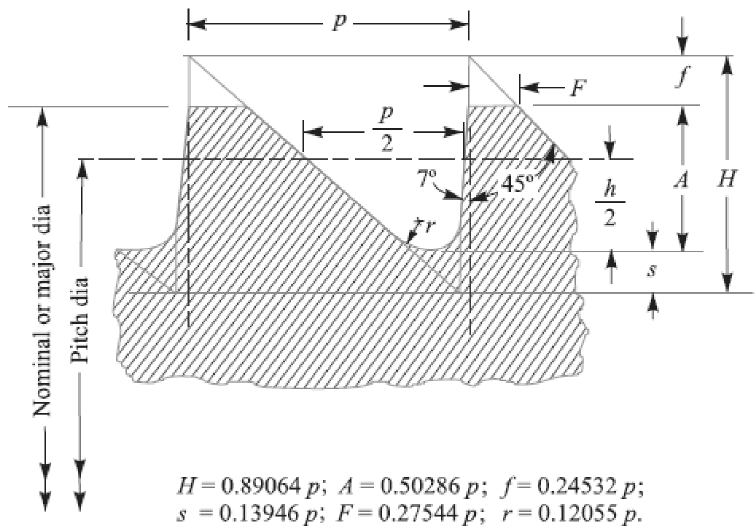


Fig. 3.9. Buttress thread.

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. 3.10 and the design profile of the nut and bolt is shown in Fig. 3.11.

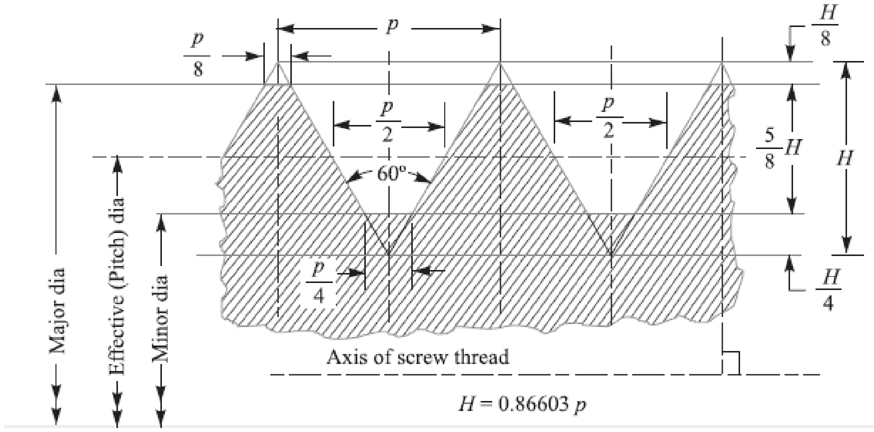


Fig. 3.10. Basic profile of the thread.

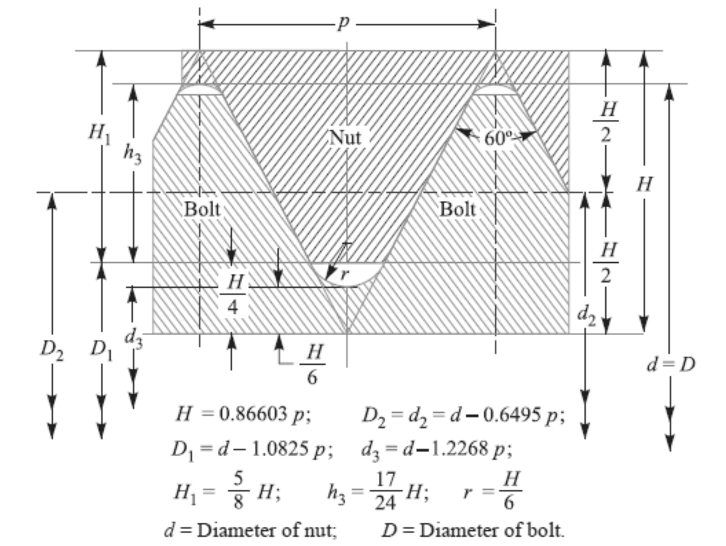


Fig. 3.11. Design profile of the nut and bolt.

3.1.4 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.

3.1.5 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. 3.12 (*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.**

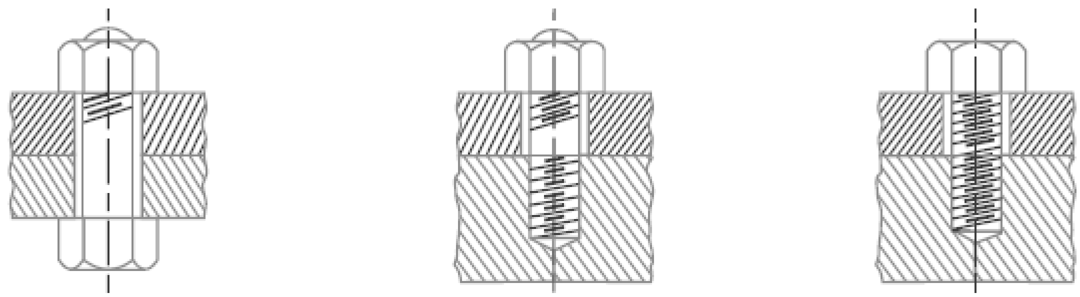


Fig. 3.12

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. 3.12 (*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. 3.12 (*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.
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. 3.13.

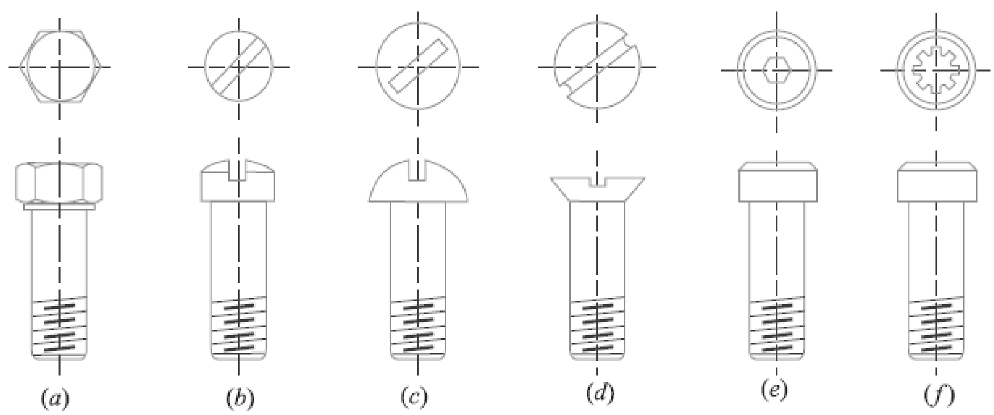


Fig. 3.13. Types of cap screws.

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. 3.14. 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.

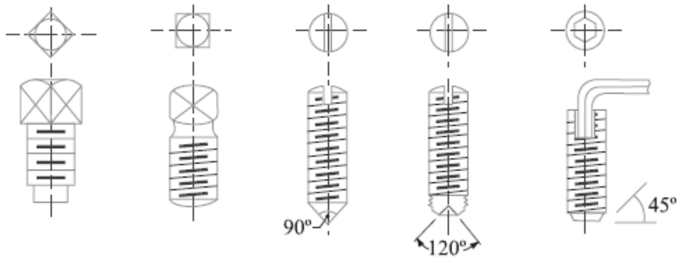


Fig. 3.14. Set screws.

The diameter of the set screw (d) may be obtained from the following expression:
$d = 0.125\,D + 8$ 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

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

∴ Torque transmitted by a set screw,

$$T = F \times \frac{D}{2} \text{ N-m}$$

and power transmitted (in watts)

$$P = \frac{2\pi N.T}{60}$$

where N is the speed in r.p.m.

3.1.6 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.3.15 (*a*). The upper nut is then held tightly while the lower one is slackened back against it.

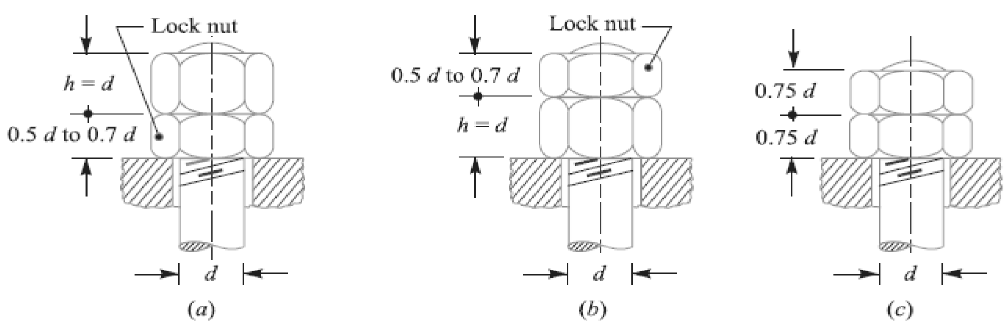


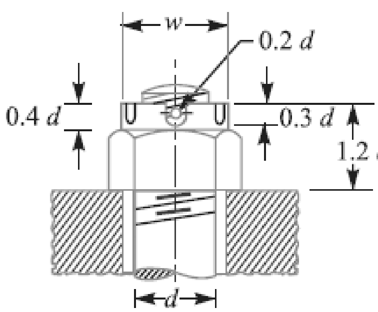
Fig. 11.15. Jam nut or lock nut.

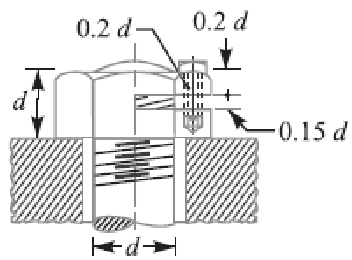
In slackening back the lock nut, a thin spanner is required which is difficult to find in many shops. Therefore to overcome this difficulty, a thin nut is placed on the top as shown in Fig. 3.15 (*b*). If the nuts are really tightened down as they should be, the upper nut carries a greater tensile load 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. 3.15 (*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. 3.16. 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. 3.17. 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. 3.18. 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.





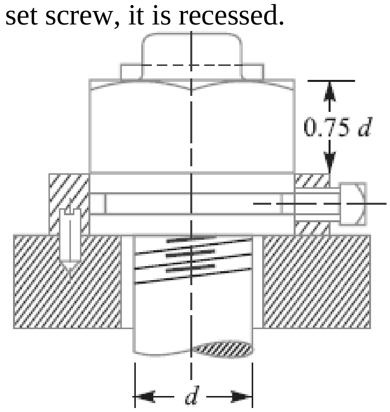


Fig. 3.16. Castle nut. **Fig. 3.17.** Sawn nut. **Fig. 3.18.** Penn, ring or grooved nut.

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. 3.19 (*a*). But a split pin is often driven through the bolt above the nut, as shown in Fig. 3.19 (*b*).

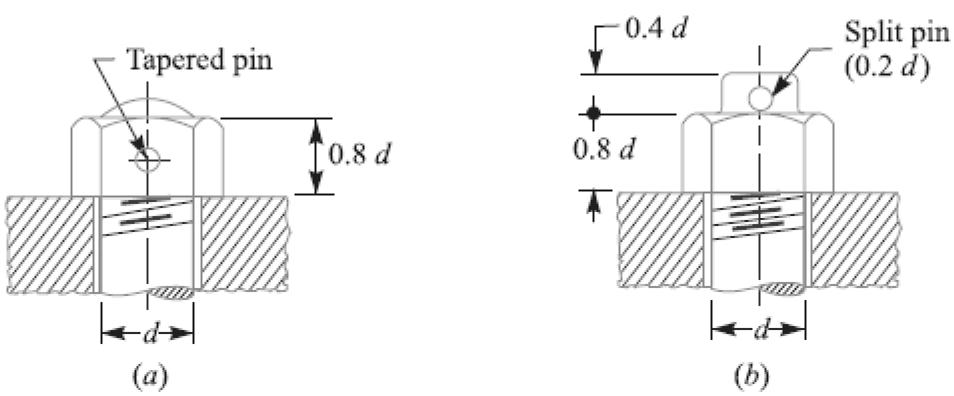


Fig. 3.19. Locking with pin.

6. *Locking with plate*.A form of stop plate or locking plate is shown in Fig. 3.20. The nut can be adjusted and subsequently locked through angular intervals of 30° by using these plates.

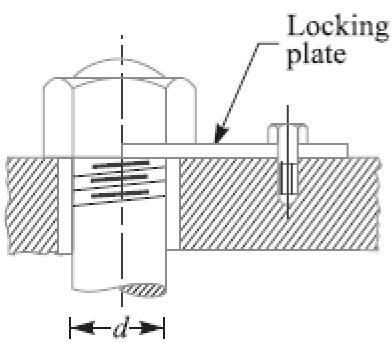


Fig. 3.20. Locking with plate.

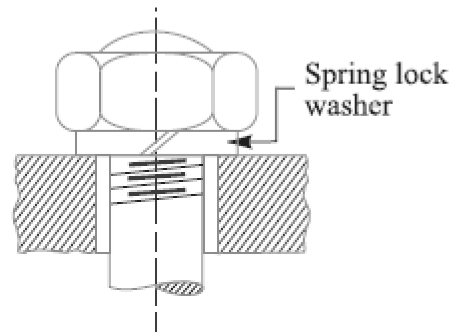


Fig. 3.21. Locking with washer.

***Spring lock washer*.** A spring lock washer is shown in Fig. 3.21. 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.

3.1.7 Designation of Screw Threads

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

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*.

3.1.8 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 3.1.

Table 3.1. Design dimensions of screw threads, bolts and nuts according to IS : 4218 (Part III) 1976 (Reaffirmed 1996)

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 mm^2*
				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

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

3.1.9 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.

3.1.10 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

$P_i = 2840\, d$ 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

$P_i = 1420\, d$ 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 3.1 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.

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

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

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

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 he 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 3.1. *Determine the safe tensile load for a bolt of M 30, assuming a safe tensile stress of 42 MPa.*

Given :

d = 30 mm

σ_t = 42 MPa = 42 N/mm^2

Solution:

From Table 3.1 (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 mm2.

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

Example 3.2. *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.*

Given : d = 24 mm

Solution.

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 × 24 = 68 160 N

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

$$68\ 160 = \frac{\pi}{4}(d_c)^2\,\sigma_t = \frac{\pi}{4}(20.30)^2\,\sigma_t = 324\,\sigma_t$$
$$\sigma_t = 68\ 160 / 324 = 210 \text{ N/mm}^2 = 210 \text{ MPa } \textbf{Ans.}$$

3.1.11 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. 3.22 (*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.

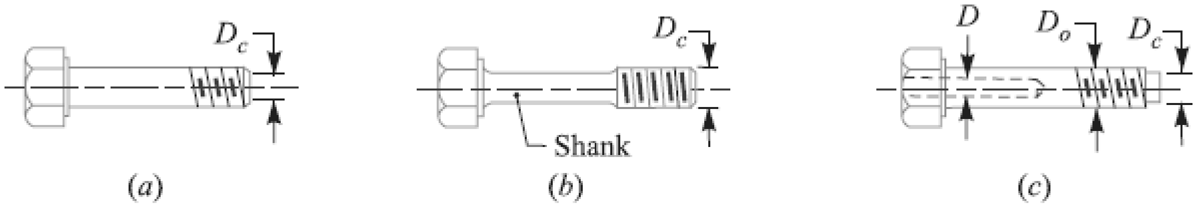


Fig. 3.22. Bolts of uniform strength.

If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (Dc) as shown in Fig. 3.22 (*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. 3.22 (*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.
Do = Outer diameter of the thread, and
Dc = 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 3.3. *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 : Do = 48 mm
From Table 11.1 (coarse series), we find that the core diameter of the thread (corresponding to Do = 48 mm) is Dc = 41.795 mm.

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

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

3.1.12 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.

3.1.13 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.

3.1.14 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. 3.23. A little consideration will show that each bolt is subjected to a direct tensile load of

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

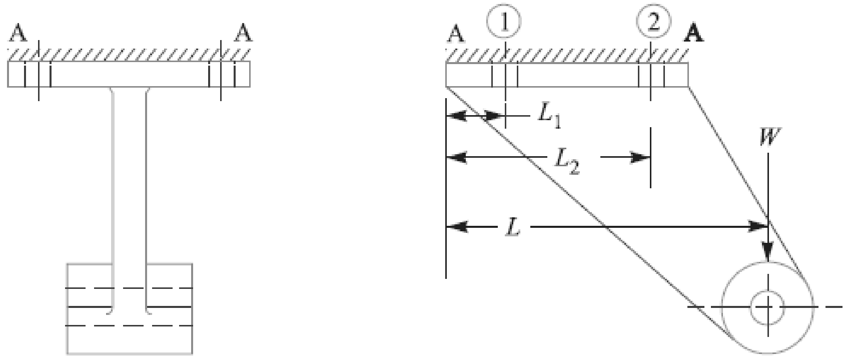


Fig. 3.23. Eccentric load acting parallel to the axis of bolts.

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

$\therefore$ 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

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

$\therefore$ Total moment of the load on the bolts about the tilting edge

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

... ($\because$ There are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

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

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

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

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

$\therefore$ Tensile load on each bolt at distance L_2,

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

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

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

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

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

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

Example 3.4. *A bracket, as shown in Fig. 3.23, supports a load of 30 kN. Determine the size of bolts, if the maximum allowable tensile stress in the bolt material is 60 MPa. The distances are : L_1 = 80 mm, L_2= 250 mm, and L = 500 mm.*

Solution.

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

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

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

and load in a bolt per unit distance,

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

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

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

∴ Maximum tensile load on the heavily loaded bolt,

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

Let d_c = Core diameter of the bolts.
We know that the maximum tensile load on the bolt (W_t),

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

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

d_c = 27.2 mm

From Table 3.1 (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**

3.1.15 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. 3.24.

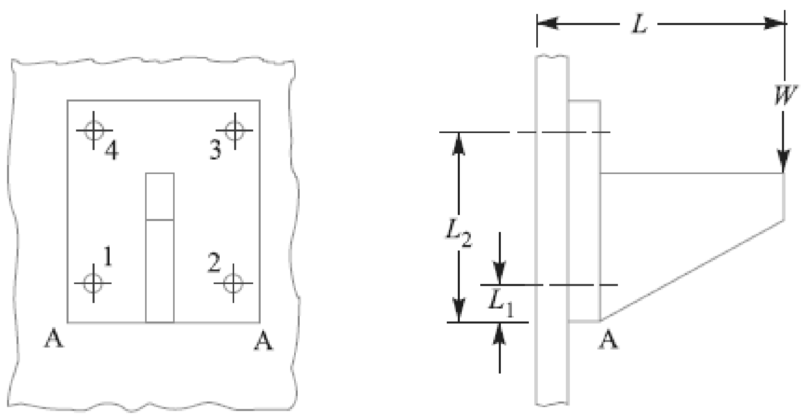


Fig.3.24. Eccentric load perpendicular to the axis of bolts.

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

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

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

∴ Maximum tensile load on bolt 3 or 4,

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

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.

Example 3.5. *For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in Fig. 3.24. The maximum load that comes on the bracket is 12 kN acting vertically at a distance of 400 mm from the face of the column. The vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at a distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for the bolt material is 84 MPa. Also find the cross-section of the arm of the bracket which is rectangular.*

Solution.

Given : W = 12 kN = 12 × 10^3 N

L = 400 mm

L_1 = 50 mm

L_2 = 375 mm

σ_t = 84 MPa = 84 N/mm2 ;

n = 4

We know that direct shear load on each bolt,

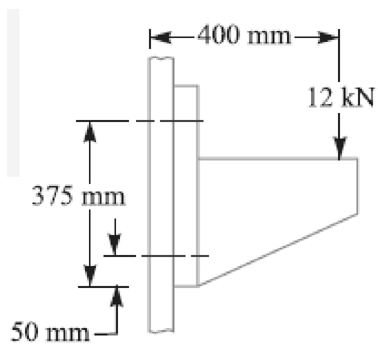


Fig. 3.24

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. 3.24), because they lie at the greatest distance from the tilting edge A–A (*i.e.* lower edge).

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

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

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

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

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

Size of the bolt

Let d_c = Core diameter of the bolt.

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

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

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

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

Cross-section of the arm of the bracket

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

$\therefore$ Section modulus,

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

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

$\therefore$ Maximum bending moment on the bracket,

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

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

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

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

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

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

3.1.16 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. 3.25.

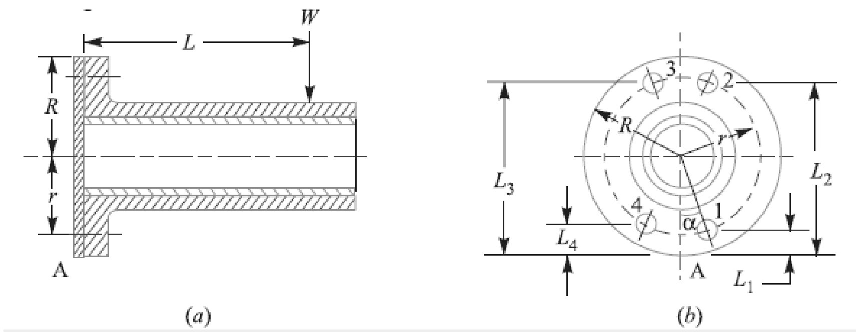


Fig. 3.25. Eccentric load on a bracket with circular base.

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
$L1$, $L2$, $L3$, and $L4$ = 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,

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

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

Now from the geometry of the Fig. 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

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

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

This load will be maximum when cos α is minimum *i.e*. when cos α = – 1 or α= 180^0
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,

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

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

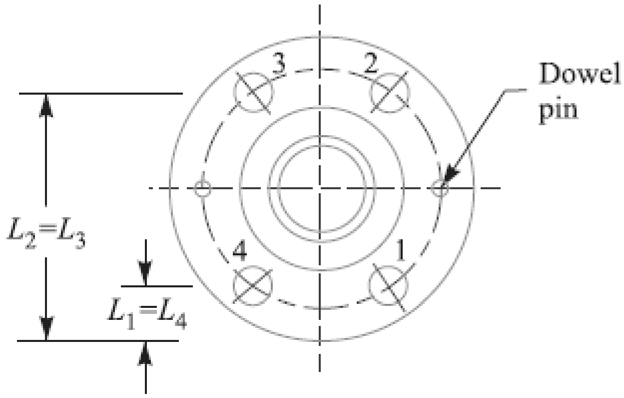


Fig. 3.26

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

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

Example 3.6. *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.*

Determine : 1. The distance of the load from the centre of the pillar along a line X-X as shown in Fig. 3.27 (a). The load lifted by the pillar crane is 60 kN and the allowable tensile stress for the bolt material is 60 MPa.

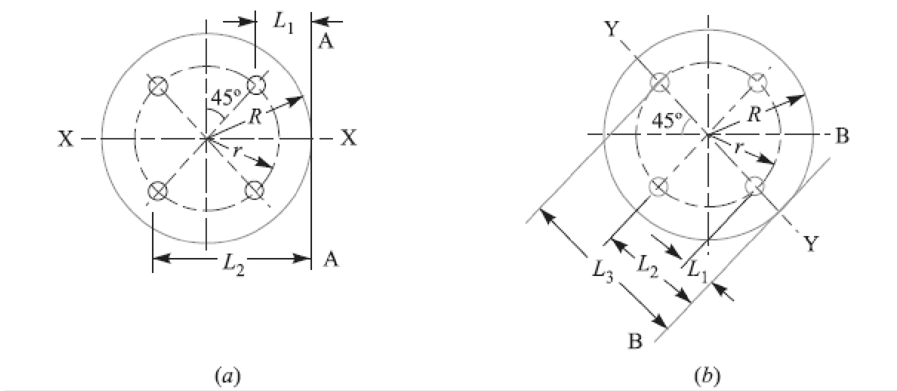


Fig. 3.27

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. 3.27 (b) at the same distance as in part (1).

Solution. Given :

D = 600 mm or R = 300 mm

n = 4

db = 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 3.1, we find that the stress area corresponding to M 30 is 561 mm^2.
We know that the maximum load carried by each bolt
= Stress area × σ_t = 561 × 60 = 33 660 N = 33.66 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 $L2$ from the tilting edge *A-A*
= 33.66 + 15 = 48.66 kN ...***(i)***
From Fig. 11.43 (*a*), we find that
$L_1 = R - r \cos 45° = 300 - 250 \times 0.707 = 123$ mm = 0.123 m
and $L_2 = R + r \cos 45° = 300 + 250 \times 0.707 = 477$ mm = 0.477 m
Let w = Load (in kN) per bolt per unit distance.
∴ Total load carried by each bolt at distance $L2$from the tilting edge *A-A*
= $w.L2 = w \times 0.477$ kN ...***(ii)***
From equations ***(i)*** and ***(ii)***, we have
w = 48.66 / 0.477 = 102 kN/m
∴ Resisting moment of all the bolts about the outer (*i.e.* tilting) edge of the flange along the tangent *A-A*
= $2w\,[(L1)2 + (L2)2] = 2 \times 102\,[(0.123)2 + (0.477)2]$ = 49.4 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* = $e - R = e - 0.3$

2. *Maximum stress induced in the bolt*
Since the load is applied along a line *Y-Y* as shown in Fig. 3.27 (*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 kN-m
From Fig. 11.43 (*b*), we find that
$L1 = R - r = 300 - 250 = 50$ mm = 0.05 m
$L2 = R = 300$ mm = 0.3 m
and $L3 = R + r = 300 + 250 = 550$ mm = 0.55 m
∴ Resisting moment of all the bolts about *B–B*
= $w\,[(L1)2 + 2(L2)2 + (L3)2] = w[(0.05)2 + 2(0.3)2 + (0.55)2]$ kN-m
= 0.485 w 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 kN/m
Since the bolt at a distance of $L3$ is heavily loaded, therefore load carried by this bolt
= $w.L3 = 102 \times 0.55$ = 56.1 kN

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

$$= \frac{\text{Force}}{\text{Stress area}} = \frac{41\ 000}{516}$$
$$= 79.65 \text{ N/mm}^2 = 79.65 \text{ MPa Ans.}$$

3.1.17 Eccentric Load Acting in the Plane Containing the Bolts

When the eccentric load acts in the plane containing the bolts, as shown in Fig. 3.28, then the same procedure may be followed as discussed for eccentric loaded riveted joints.

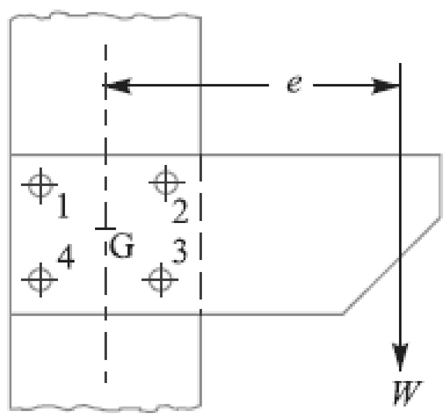
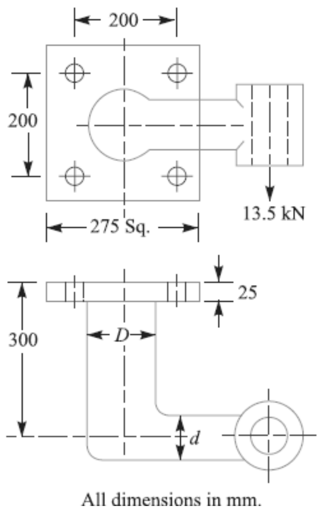


Fig. 3.28. Eccentric load in the plane containing the bolts.

Example 3.7. *Fig. 3.29 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 ;
σ_t= 110 MPa = 110 N/mm2 ;
τ= 65 MPa = 65 N/mm2



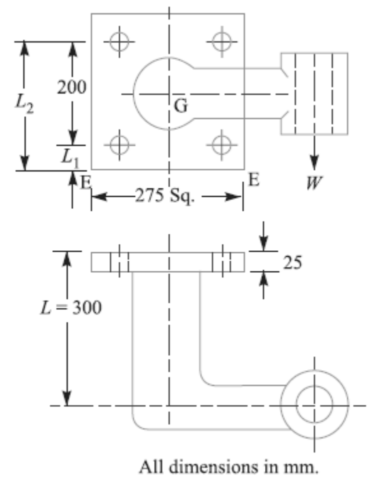


Fig. 3.29 **Fig. 3.30**

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 × (300 – 25) = 3712.5 × 103 N-mm

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

∴ Equivalent twisting moment,

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

$$= 5017 \times 10^3 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

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

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

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

Diameter (d) for the arm of the bracket

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

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

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

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

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

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

Tensile load on each top bolt

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

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

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

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

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

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

and turning moment of the load about the tilting edge

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

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

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

∴ Tensile load on each top bolt

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

Maximum shearing force on each bolt

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

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

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

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

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

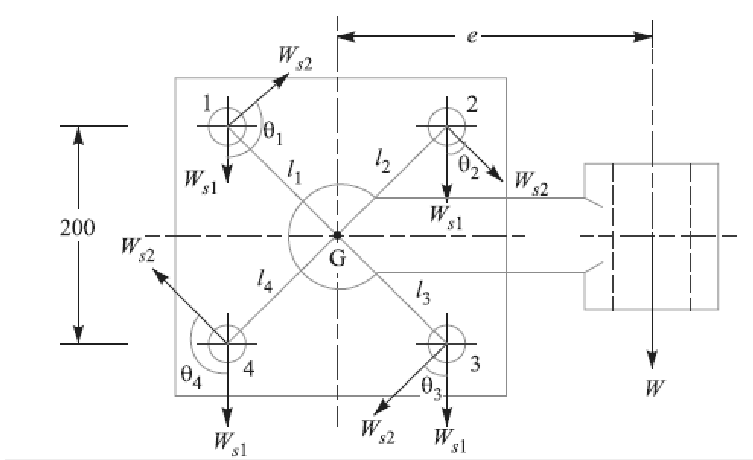


Fig. 3.31

∴ 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. 3.31, 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

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

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

3.2 Cotter and Knuckle Joints

3.2.1 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 crosshead of a reciprocating steam engine, a piston rod and its extension as a tail or pump rod, strap end of connecting rod etc.

3.2.2 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.

3.2.3 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. 3.32 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.

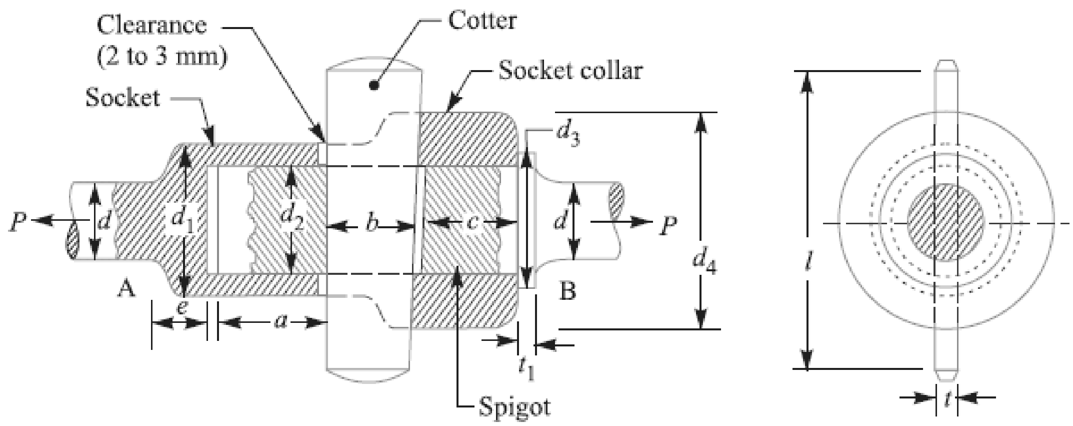


Fig. 3.32. Socket and spigot cotter joint.

3.2.4 Design of Socket and Spigot Cotter Joint

The socket and spigot cotter joint is shown in Fig. 3.32.

Let P = Load carried by the rods,

d = Diameter of the rods,

$d1$ = Outside diameter of socket,

$d2$ = Diameter of spigot or inside diameter of socket,

$d3$ = Outside diameter of spigot collar,

$t1$ = Thickness of spigot collar,

$d4$ = 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 :

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. 3.33, therefore 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

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

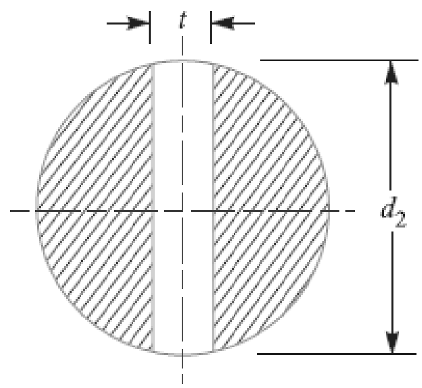


Fig. 3.33.

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

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

$= d2 \times t$

∴ Crushing strength = $d2 \times t \times \sigma_c$

Equating this to load (P), we have

$P = d2 \times t \times \sigma_c$

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

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. 3.34

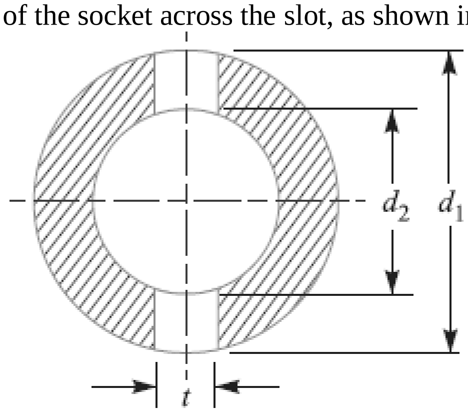


Fig. 3.34

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

$\therefore$ Tearing strength of the socket across the slot

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

Equating this to load (P), we have

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

From this equation, outside diameter of socket ($d1$) may be determined.

5. *Failure of cotter in shear*
Considering the failure of cotter in shear as shown in Fig. 3.35. 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

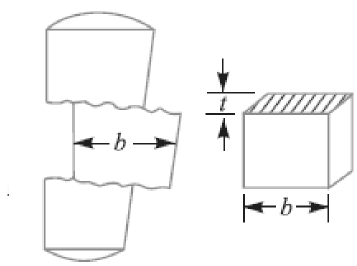


Fig. 3.35

6. *Failure of the socket collar in crushing*
Considering the failure of socket collar in crushing as shown in Fig. 3.36.

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.

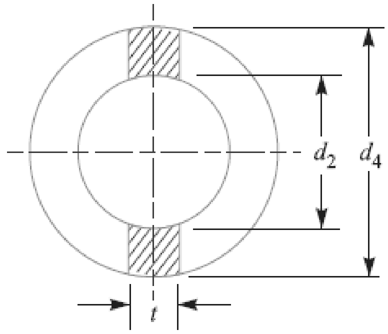


Fig. 3.36

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.

9. *Failure of spigot collar in crushing*
Considering the failure of the spigot collar in crushing as shown in Fig. 3.37. We know that area that resists crushing of the collar

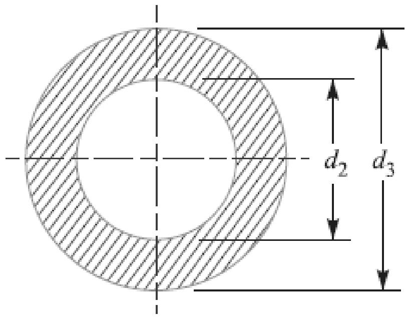


Fig. 3.37

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

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

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

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. 3.38. 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

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

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

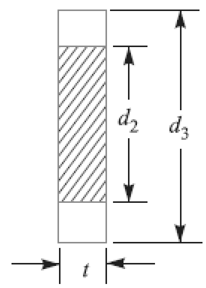


Fig. 3.38

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 ($d4$) and maximum at the inner diameter (d_2), as shown in Fig. 3.39.

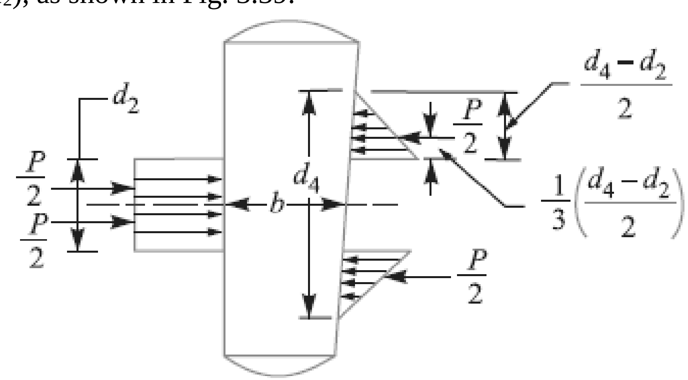


Fig. 3.39

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.

Example 3.8. *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 × 103 N
σ_t = 50 MPa = 50 N / mm^2
τ= 35 MPa = 35 N / mm ^{2}
σ_c = 90 MPa = 90 N/mm^2

The cotter joint is shown in Fig. 3.32. 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.*

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

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

and $\quad t = d_2 / 4 = 40 / 4 = 10 \text{ mm } \textbf{Ans.}$

3. *Outside diameter of socket*

Let $d1$ = Outside diameter of socket.

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

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

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

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

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

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

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

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),

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

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

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

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

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),

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

$$\therefore \quad t_1 = 30 \times 10^3 / 4400 = 6.8 \text{ say } 8 \text{ mm } \textbf{Ans.}$$

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

$$\therefore \quad l = 4\,d = 4 \times 28 = 112 \text{ mm } \textbf{Ans.}$$

11. The dimension e is taken as $1.2\,d$.

$$\therefore \quad e = 1.2 \times 28 = 33.6 \text{ say } 34 \text{ mm } \textbf{Ans.}$$

3.2.5 Sleeve and Cotter Joint

Sometimes, a sleeve and cotter joint as shown in Fig. 3.40, 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. 3.40. 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.

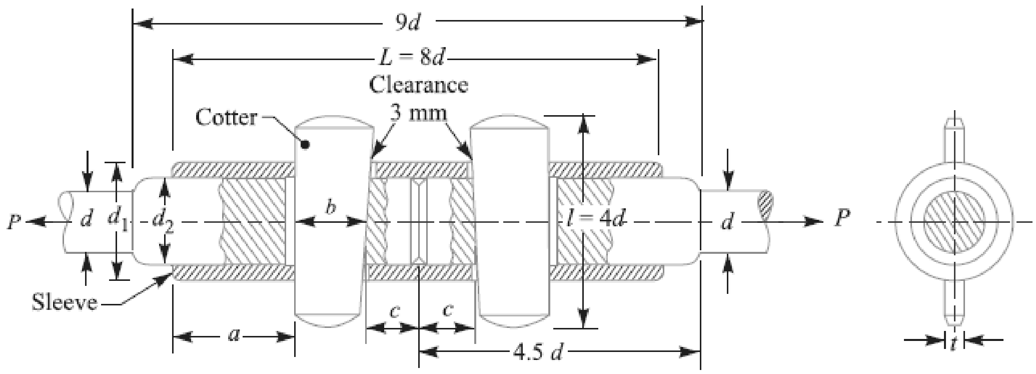


Fig. 3.40. Sleeve and cotter joint.

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 = Inside diameter of sleeve = 1.25 *d*

Length of sleeve, $L = 8\ d$

Thickness of cotter, $t = d2/4$ 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*

3.2.6 Design of Sleeve and Cotter Joint

The sleeve and cotter joint is shown in Fig. 3.40.

Let P = Load carried by the rods,
d = Diameter of the rods,
$d1$ = Outside diameter of sleeve,
$d2$ = 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 ,$\tau$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

1. *Failure of the rods in tension*

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

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

$\therefore$ 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.

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

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

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 3.9.

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 = 60×10^3 N

σ_t = 60 MPa = 60 N/mm^2

τ = 70 MPa = 70 N/mm^2

σ_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),

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 60 = 47.13\, d^2$$

$\therefore$ $d^2 = 60 \times 10^3 / 47.13 = 1273$ or $d = 35.7$ say 36 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),

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]60 = 32.13\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 60 \times 10^3 / 32.13 = 1867$ or $d_2 = 43.2$ say 44 mm **Ans.**

and thickness of cotter,

$$t = \frac{d_2}{4} = \frac{44}{4} = 11 \text{ mm } \textbf{Ans.}$$

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

$$60 \times 10^3 = d_2 \times t \times \sigma_c = 44 \times 11 \times \sigma_c = 484\,\sigma_c$$

$\therefore$ $\sigma_c = 60 \times 10^3 / 484 = 124 \text{ N/mm}^2$

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.

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

$$60 \times 10^3 = \left[\frac{\pi}{4}[(d_1)^2 - (d_2)^2] - (d_1 - d_2)\,t\right]\sigma_t$$

$$= \left[\frac{\pi}{4}[(d_1)^2 - (44)^2] - (d_1 - 44)\,11\right]60$$

$\therefore$ $60 \times 10^3 / 60 = 0.7854\,(d_1)^2 - 1520.7 - 11\,d_1 + 484$

or $(d_1)^2 - 14\,d_1 - 2593 = 0$

$$\therefore \quad d_1 = \frac{14 \pm \sqrt{(14)^2 + 4 \times 2593}}{2} = \frac{14 \pm 102.8}{2}$$

$= 58.4$ 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),

$$60 \times 10^3 = 2\,b \times t \times \tau = 2 \times b \times 11 \times 70 = 1540\,b$$

$\therefore$ $b = 60 \times 10^3 / 1540 = 38.96$ say 40 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),

$$60 \times 10^3 = 2\,a \times d_2 \times \tau = 2\,a \times 44 \times 70 = 6160\,a$$

$\therefore$ $a = 60 \times 10^3 / 6160 = 9.74$ say 10 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),

$$60 \times 10^3 = 2\,(d_1 - d_2)\,c \times \tau = 2\,(60 - 44)\,c \times 70 = 2240\,c$$

$$\therefore \quad c = 60 \times 10^3 / 2240 = 26.78 \text{ say } 28 \text{ mm} \quad \textbf{Ans.}$$

3.2.7 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.

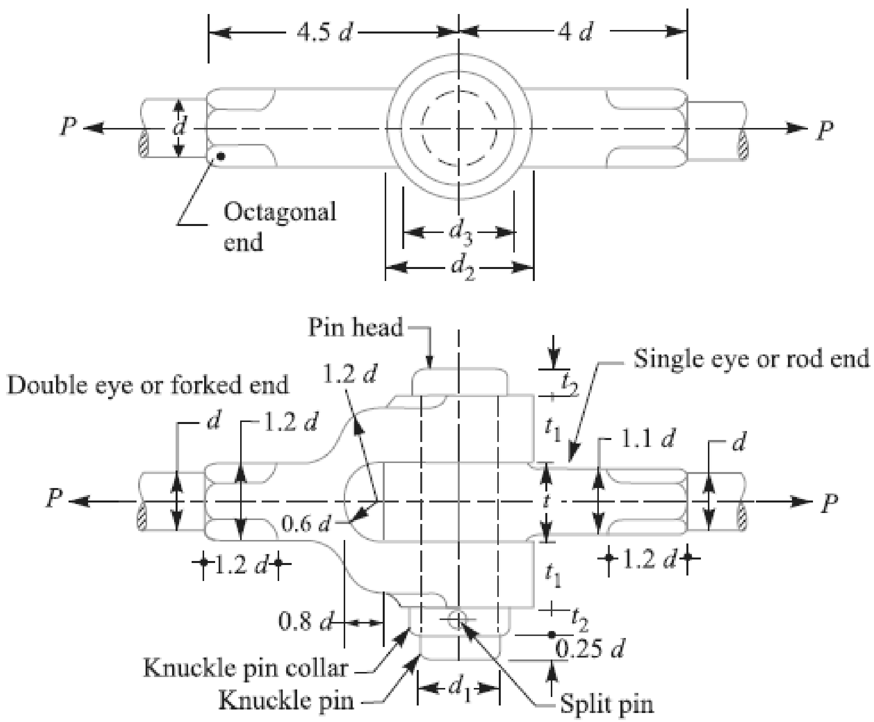


Fig. 3.41. Kunckle joint.

In knuckle joint (the two views of which are shown in Fig. 3.41), 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.

3.2.8 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,

$d_2 = 2\,d$

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, $t1 = 0.75\,d$

Thickness of pin head, $t2 = 0.5\,d$

Other dimensions of the joint are shown in Fig. 3.41.

3.2.9 Methods of Failure of Knuckle Joint

Consider a knuckle joint as shown in Fig. 3.41.

Let P = Tensile load acting on the rod,

d = Diameter of the rod,

$d1$ = Diameter of the pin,

$d2$ = Outer diameter of eye,

t = Thickness of single eye,

$t1$ = Thickness of fork.

σ_t ,$\tau$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.*, $d1 = d$), 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. 3.42. Thus in the forks, a load $P/2$ acts through a distance of $t1 / 3$ 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

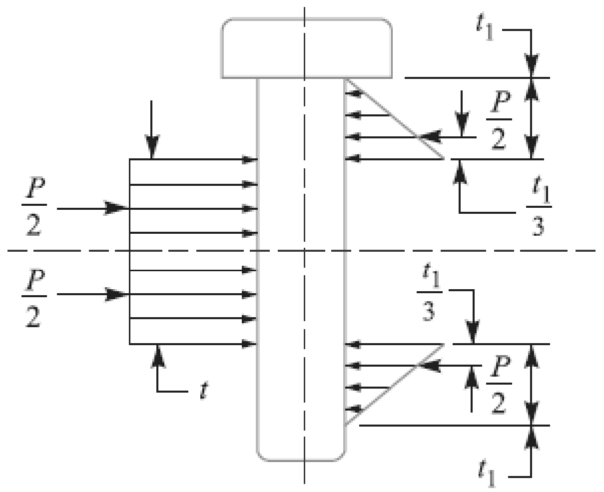


Fig. 3.42. Distribution of load on the pin.

$$M = \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2}\right) - \frac{P}{2} \times \frac{t}{4}$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{2} - \frac{t}{4}\right)$$

$$= \frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)$$

and section modulus, $Z = \frac{\pi}{32}(d_1)^3$

$\therefore$ Maximum bending (tensile) stress,

$$\sigma_t = \frac{M}{Z} = \frac{\frac{P}{2}\left(\frac{t_1}{3} + \frac{t}{4}\right)}{\frac{\pi}{32}(d_1)^3}$$

From this expression, the value of $d1$ 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 $= (d_2 - d_1)\ t$

$\therefore$ 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 $= (d_2 - d_1)\ t$

$\therefore$ 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 $= d_1 \times t$

$\therefore$ 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

$$= (d_2 - d_1) \times 2\ t_1$$

$\therefore$ 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.

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.

3.2.10 Design Procedure of 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 ($d1$) 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 $d1$ 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. 3.2.8
4. The induced stresses are obtained by substituting the empirical dimensions in the relations as discussed in Art. 3.2.9
In case the induced stress is more than the allowable stress, then the corresponding dimension may be increased.

Example 3.10. *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. 3.41. 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),

$$150 \times 10^3 = 2 \times \frac{\pi}{4} \times (d_1)^2\, \tau = 2 \times \frac{\pi}{4} \times (52)^2\, \tau = 4248\, \tau$$

$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$

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.