Screwed Joints

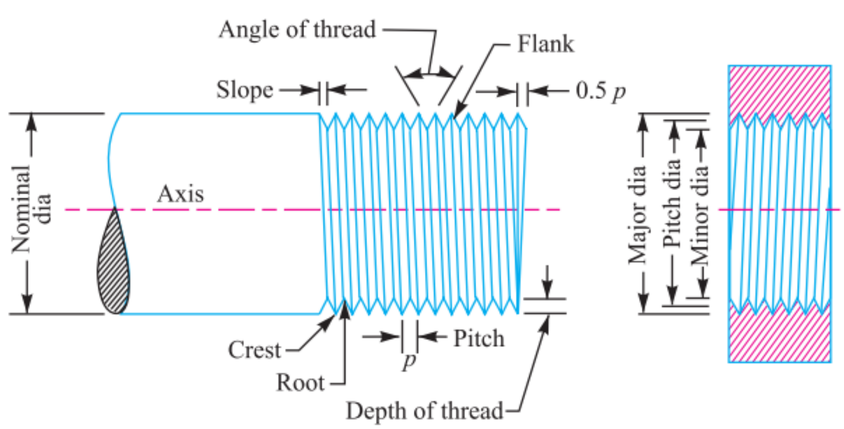
11.1 Introduction

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

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 :



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.

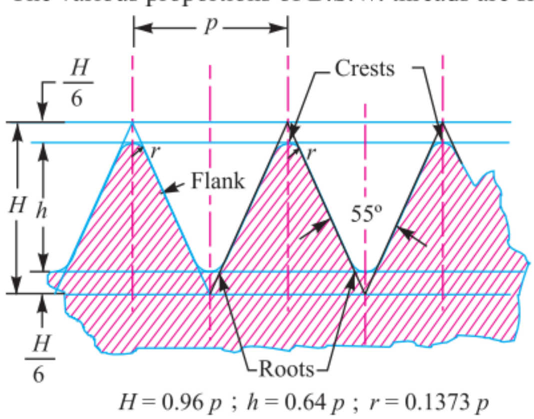


Fig. 11.2. British standard whitworth (B.S.W) thread.

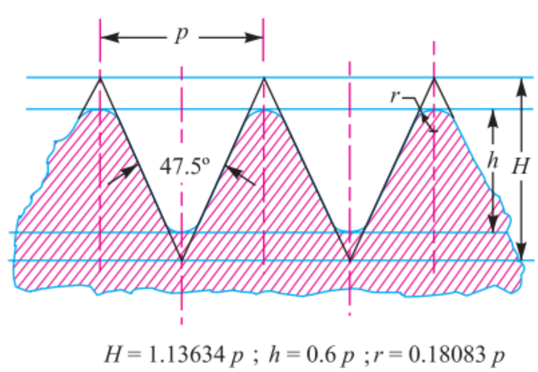


Fig. 11.3. British association (B.A.) thread.

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.

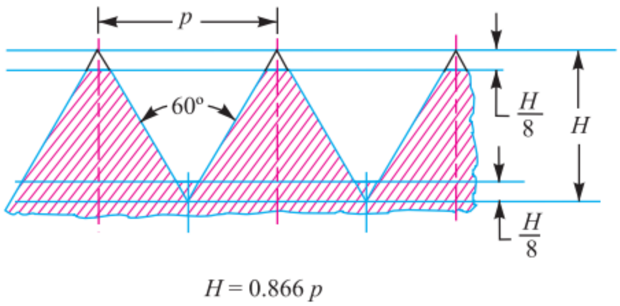


Fig. 11.4. American national standard thread.

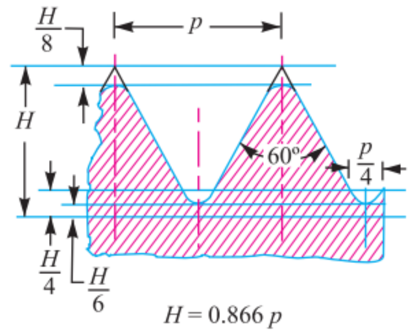


Fig. 11.5. Unified standard thread.

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.

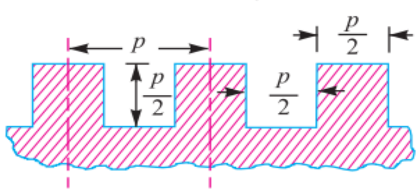


Fig. 11.6. Square thread.

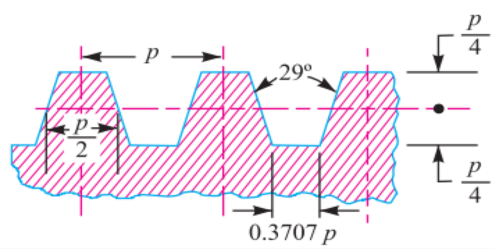


Fig. 11.7. Acme thread.

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.

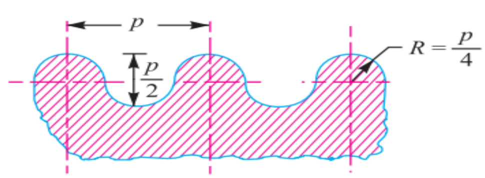
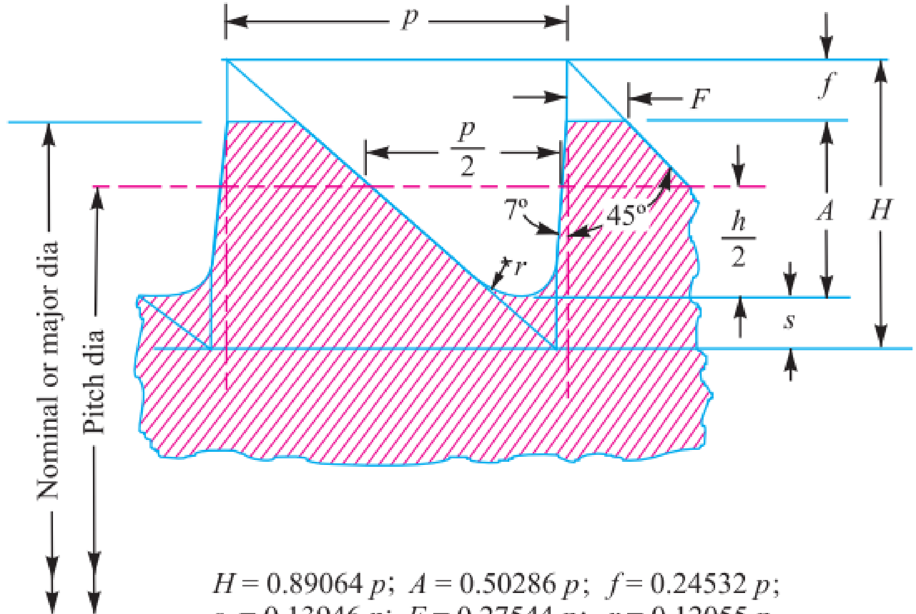


Fig. 11.8. Knuckle thread.

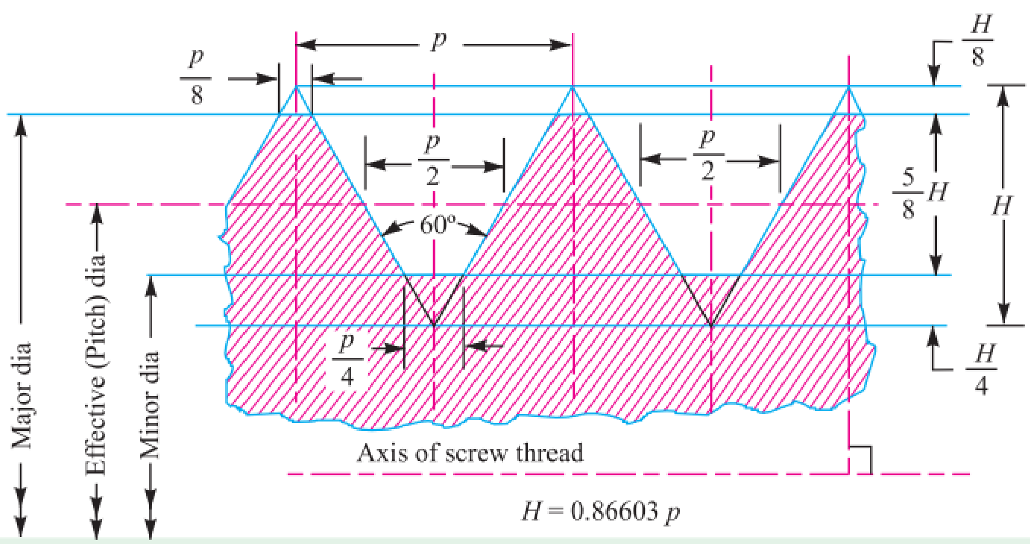
8. *Buttress thread.* It is used for transmission of power in one direction only. The force is transmitted almost parallel to the axis. This thread units the advantage of both square and V-threads. It has a low frictional resistance characteristics of the square thread and have the same strength as that 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.



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

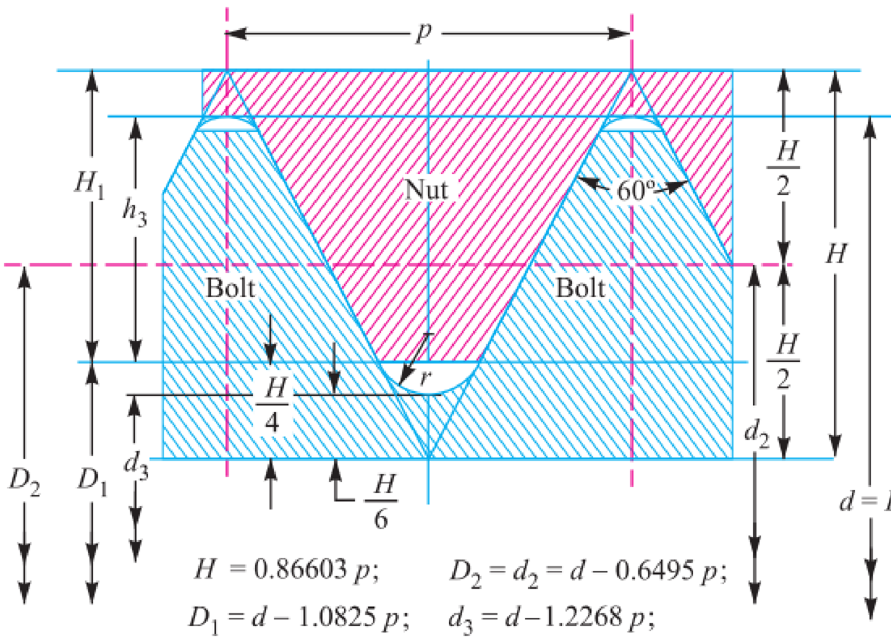
Fig. 11.9. Buttress thread.

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.



$H = 0.86603\ p$

Fig. 11.10. Basic profile of the thread.



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

$$D_1 = d - 1.0825\,p; \qquad d_3 = d - 1.2268\,p;$$

$$H_1 = \frac{5}{8}H; \qquad h_3 = \frac{17}{24}H; \qquad r = \frac{H}{6}$$

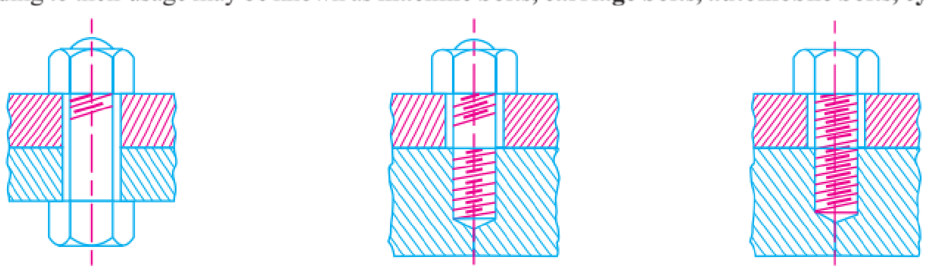
d = Diameter of nut; D = Diameter of bolt.

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

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.

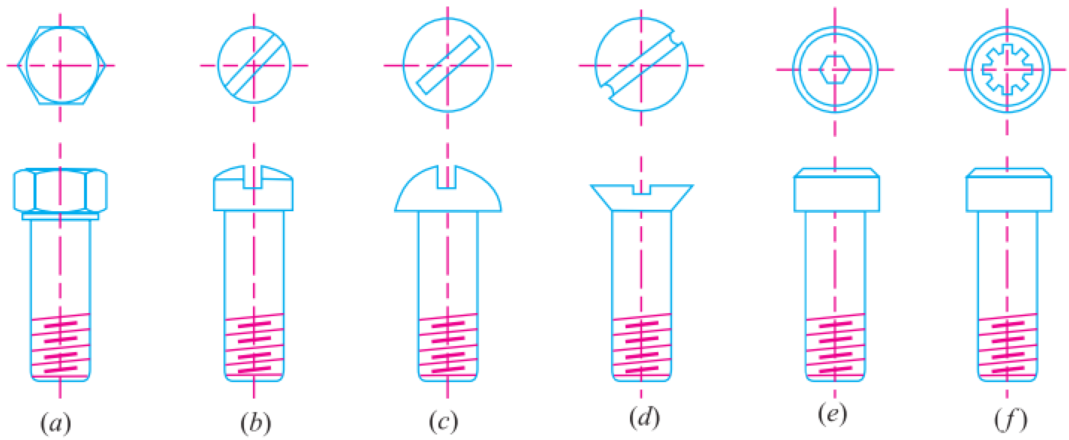
Fig. 11.12

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.

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.

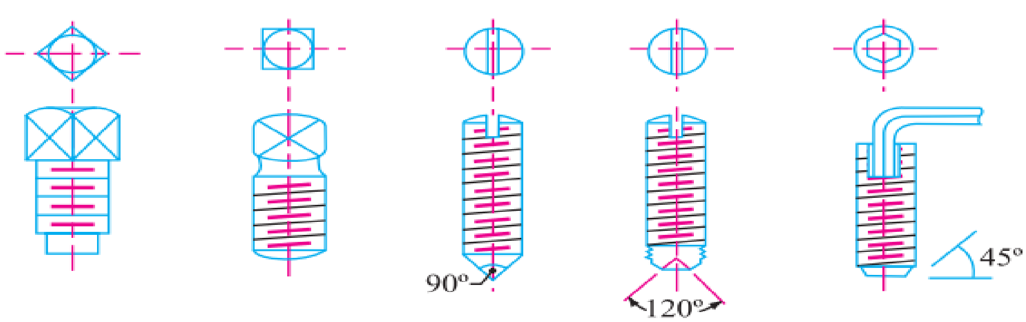


Fig. 11.14. Set screws.

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

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

∴ 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.

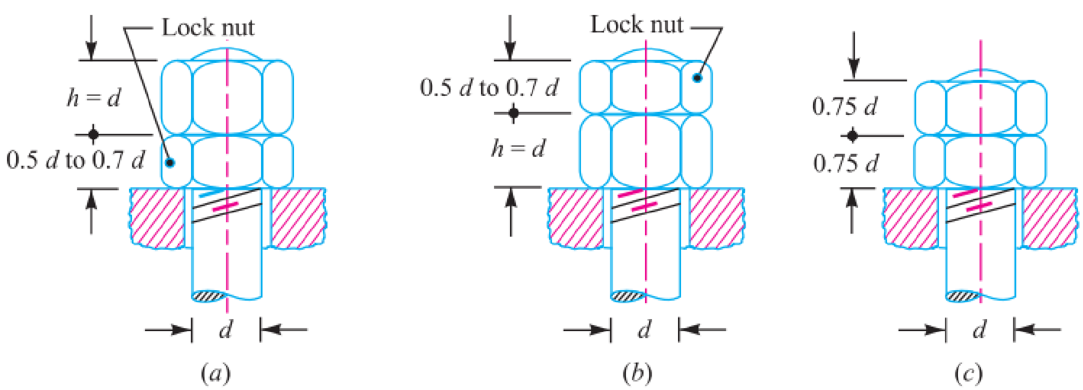


Fig. 11.15. Jam nut or lock nut.

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.

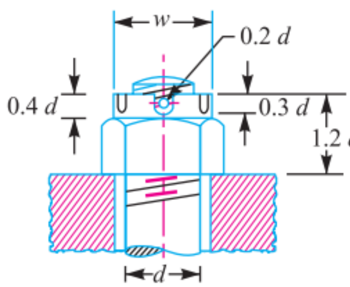


Fig. 11.16. Castle nut.

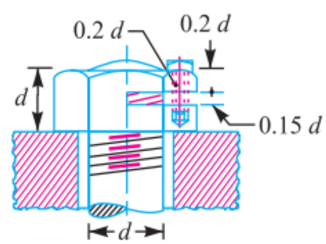


Fig. 11.17. Sawn nut.

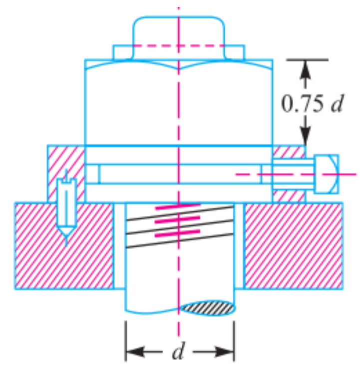


Fig. 11.18. Penn, ring or grooved nut.

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*).

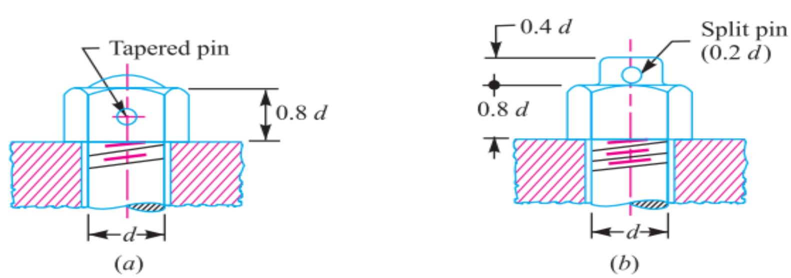


Fig. 11.19. Locking with pin.

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.

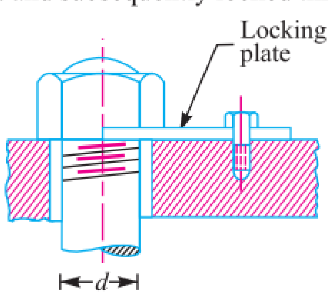


Fig. 11.20. Locking with plate.

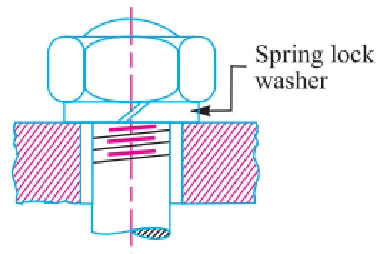


Fig. 11.21. Locking with washer.

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

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.

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

(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.

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.

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),

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

∴ σ_t = 68 160 / 324 = 210 N/mm^2 = 210 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.

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.

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 $= 60 \times 10^3$ N ; $\sigma_t = 100$ MPa $= 100$ N/mm^2

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

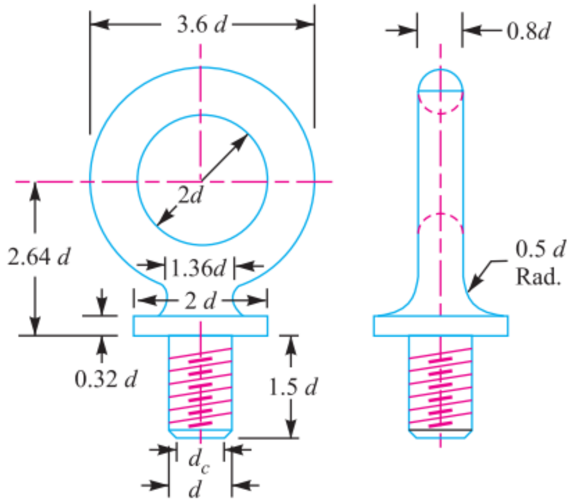


Fig. 11.22

Let d = Nominal diameter of the bolt, and

d_c = Core diameter of the bolt.

We know that load on the bolt (P),

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

$$\therefore \quad (d_c)^2 = 600 \times 10^3 / 78.55 = 764 \quad \text{or} \quad d_c = 27.6 \text{ mm}$$

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.**

Case 2

When a welded joint is loaded eccentrically as shown in Fig. 10.23, the following two types of the stresses are induced:

1. Direct or primary shear stress, and
2. Shear stress due to turning moment.

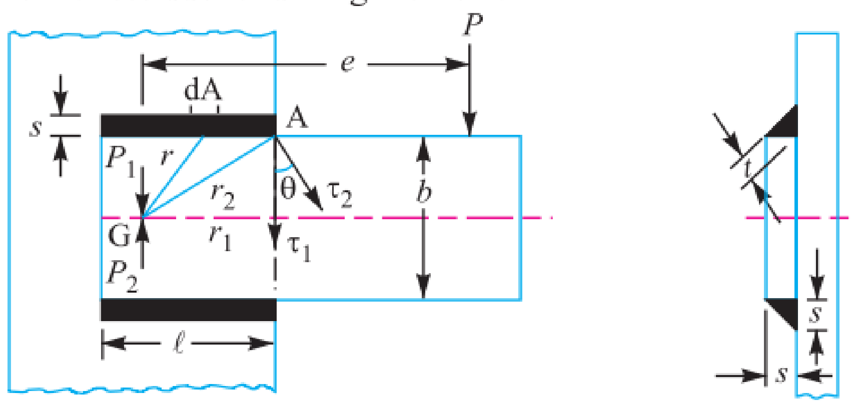


Fig. 10.23. Eccentrically loaded welded joint.

Let P = Eccentric load,

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

l = Length of single weld,

s = Size or leg of weld, and

t = Throat thickness.

Let two loads P_1 and P_2 (each equal to P) are introduced at the centre of gravity 'G' of the weld system. The effect of load $P_1 = P$ is to produce direct shear stress which is assumed to be uniform over the entire weld length. The effect of load $P_2 = P$ is to produce a turning moment of magnitude $P \times e$ which tends of rotate the joint about the centre of gravity 'G' of the weld system. Due to the turning moment, secondary shear stress is induced.

We know that the direct or primary shear stress,

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

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

... ($\because$ Throat area for single fillet weld = $t \times l = 0.707\ s \times$ l)

Since the shear stress produced due to the turning moment ($T = P \times e$) at any section is proportional to its radial distance from G, therefore stress due to $P \times e$ at the point A is proportional to AG (r_2) and is in a direction at right angles to AG. In other words,

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

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

where τ_2 is the shear stress at the maximum distance (r_2) and τ is the shear stress at any distance r.

Consider a small section of the weld having area dA at a distance r from G.

$\therefore$ Shear force on this small section

$= \tau \times dA$

and turning moment of this shear force about G,

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

$\therefore$ Total turning moment over the whole weld area,

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

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

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

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

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

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

$\therefore$ Resultant shear stress at A,

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

where θ = Angle between τ_1 and τ_2, and

$\cos\theta = r_1 / r_2$

Note: The polar moment of inertia of the throat area (A) about the centre of gravity (G) is obtained by the parallel axis theorem, *i.e.*

$$J = 2\,[I_{xx} + A \times x^2] \qquad \text{... (}\because \text{ of double fillet weld)}$$

$$= 2\left[\frac{A \times l^2}{12} + A \times x^2\right] = 2A\left(\frac{l_2}{12} + x^2\right)$$

where A = Throat area = $t \times l = 0.707\, s \times l$,

l = Length of weld, and

x = Perpendicular distance between the two parallel axes.

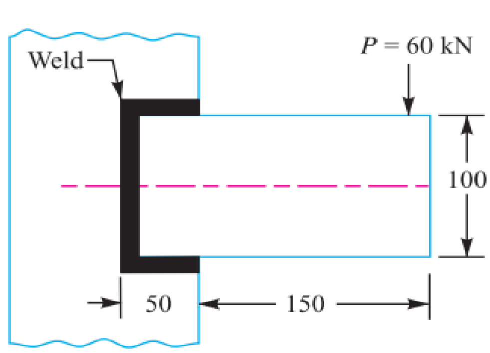
Example 10.14. *A rectangular steel plate is welded as a cantilever to a vertical column and supports a single concentrated load P, as shown in Fig. 10.30.*

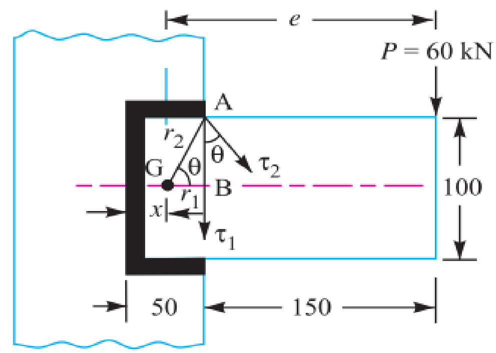
Determine the weld size if shear stress in the same is not to exceed 140 MPa.

Solution. Given : $P = 60$ kN $= 60 \times 10^3$ N ; $b = 100$ mm ; $l = 50$ mm ; $\tau = 140$ MPa $= 140$ N/mm^2

Let s = Weld size, and

t = Throat thickness.





All dimensions in mm.

Fig. 10.30 **Fig. 10.31**

First of all, let us find the centre of gravity (G) of the weld system, as shown in Fig. 10.31.

Let x be the distance of centre of gravity (G) from the left hand edge of the weld system. From Table 10.7, we find that for a section as shown in Fig. 10.31,

$$x = \frac{l^2}{2l + b} = \frac{(50)^2}{2 \times 50 + 100} = 12.5 \text{ mm}$$

and polar moment of inertia of the throat area of the weld system about G,

$$J = t\left[\frac{(b + 2l)^3}{12} - \frac{l^2\,(b + l)^2}{b + 2l}\right]$$

$$= 0.707\,s\left[\frac{(100 + 2 \times 50)^3}{12} - \frac{(50)^2\,(100 + 50)^2}{100 + 2 \times 50}\right] \quad \ldots (\because\ t = 0.707\,s)$$

$$= 0.707s\,[670 \times 10^3 - 281 \times 10^3] = 275 \times 10^3\,s \text{ mm}^4$$

Distance of load from the centre of gravity (G) *i.e.* eccentricity,

$$e = 150 + 50 - 12.5 = 187.5 \text{ mm}$$

$$r_1 = BG = 50 - x = 50 - 12.5 = 37.5 \text{ mm}$$

$$AB = 100 / 2 = 50 \text{ mm}$$

We know that maximum radius of the weld,

$$r_2 = \sqrt{(AB)^2 + (BG)^2} = \sqrt{(50)^2 + (37.5)^2} = 62.5 \text{ mm}$$

$$\therefore \quad \cos\theta = \frac{r_1}{r_2} = \frac{37.5}{62.5} = 0.6$$

We know that throat area of the weld system,

$$A = 2 \times 0.707s \times l + 0.707s \times b = 0.707\,s\,(2l + b)$$

$$= 0.707s\,(2 \times 50 + 100) = 141.4\,s \text{ mm}^2$$

$\therefore$ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{60 \times 10^3}{141.4\,s} = \frac{424}{s} \text{ N/mm}^2$$

and shear stress due to the turning moment or secondary shear stress,

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{60 \times 10^3 \times 187.5 \times 62.5}{275 \times 10^3 \; s} = \frac{2557}{s} \text{ N/mm}^2$$

We know that the resultant shear stress,

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

$$140 = \sqrt{\left(\frac{424}{s}\right)^2 + \left(\frac{2557}{s}\right)^2 + 2 \times \frac{424}{s} \times \frac{2557}{s} \times 0.6} = \frac{2832}{s}$$

$\therefore$ $s = 2832 / 140 = 20.23$ mm **Ans.**