UNIT 5 DESIGN OF SCREWS, FASTENERS AND POWER SCREWS

Structure

5.1 INTRODUCTION

Screws are used for power transmission or transmission of force. A screw is a cylinder on whose surface helical projection is created in form of thread. The thread will have specified width and depth, which bear some ratio with the diameter of the cylinder. The screw rotates in a *nut*, which has corresponding helical groove on the internal surface. Thus a nut and a screw make a connected pair in which one remains stationary while other rotates and translates axially. The helical surface of the screw thread makes surface contact with the helical groove surface of the nut. If an axial force acts on, say screw moving inside stationary nut, the point of application of the force will move as the screw advances in axial direction. This will result in work being done and hence power being transmitted. Both types – one in which screw rotates and advances in a stationary nut or one in which screw rotates between fixed support and nut is free to move axially – are used in practice. In the latter case the force acting on nut will move as nut translates. However, the friction between the surfaces of contact will require some power to be overcome. Hence the power delivered by the screw-nut pair will be less than the power supplied.

The contact surfaces of screw thread and nut groove are made perpendicular to the outside and inside cylindrical surfaces. They are sometimes given a small inclination. Such provision keeps coefficient of friction to a reasonable low level. The coefficient of friction may be further reduced by lubrication. However, by creating considerably inclined surfaces in nut and screw the effective coefficient of friction is increased. Such screw thread joint will make advancing of threaded part difficult. This combination will be used as fastening device.

Objectives

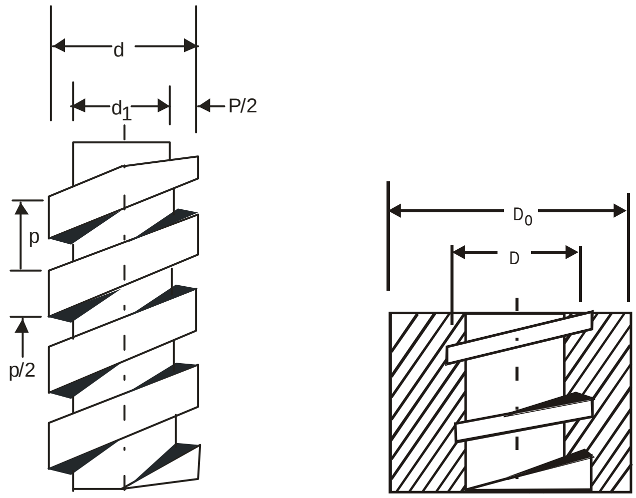
After studying this unit, you should be able to

- describe geometry of screw and nut,
- define mechanics of screw and nut,

- determine forces on screw and nut threads,
- calculate dimensions of screw and nut for transmission of force, and
- find the force on screw fastener and load transmitted to parts jointed by fasteners.

5.2 GEOMETRY OF THREAD

Look at Figures 5.1(a) and (b) and you will get a fair idea how would a screw and a nut appear. The screw will pass into nut by rotating either of them. For understanding how a helical thread can be formed on a cylinder you can take a plane sheet of paper and draw an inclined line on it. Then roll the paper to form a cylinder by bringing two opposite edges of the paper together. The line which you drew will appear like a helix on the surface of the cylinder. A line drawn as *AC*, inclined at angle α with horizontal line AA_1, will be wrapped on the cylinder to *AA*′, looking like a helix. AA_1, becomes the circumference of the cylinder and *A*′ coincides with *C*. In Figure 5.2, you can see two parallel lines *AC* and *A*′*C*′ drawn inclined at α to horizontal and then paper wrapped to form a cylinder and thus two threads are formed on cylinder. *p* is the vertical distance between *A* and *C* or between *C* and *C*′. If the paper is wrapped such that the lines drawn are on the inside surface, you can get the idea of internal thread.



(a) A Screw **(b) A Nut**

Figure 5.1 : Screw and Nut with Helical Surfaces Cut on Outside and Inside Surfaces of Cylinders, respectively

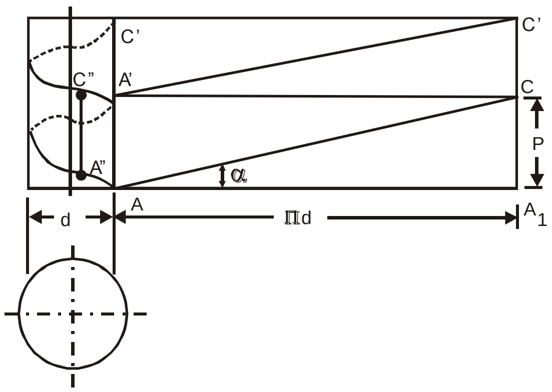


Figure 5.2 : Formation of a Helix on the Cylindrical Surface

The distance A_1C which is equal to *AA*′ and *A*″*C*″ is called the pitch of the screw. Pitch is apparently the distance between two corresponding points on two consecutive threads. The angle α between the base of the triangle and hypotenuse becomes the angle of helix. Obviously,

$$p = \pi\, d \tan \alpha$$

or $$\tan \alpha = \frac{p}{\pi d} \qquad \ldots (5.1)$$

If the helix on the outside surface ascends from right to left the thread is left hand. Such a threaded screw will have to be turned counter clockwise to engage the mating nut. On the other hand a right hand screw will be turned clockwise and its helix will appear to ascend from left to right. The thread shown in Figure 5.2 is left hand. If a plane figure, say a triangle or trapeziums placed in contact with the outer surface of the base cylinder on which the helical line was created and then is made to rotate round the cylinder along the helix then the helical surfaces will be formed on the base cylinder giving rise to thread as can be seen in Figure 5.3. If the generating plane section is a square, a square thread is created. The thread depicted in Figure 5.1(a) is a square thread. The Vee thread is created by a triangular section while trapezoidal thread has a trapezium section. This thread is also known as the Acme thread. Buttress thread has a triangular section but one side of the triangle is perpendicular to the axis. The square, the Acme and the buttress threads are used for power transmission, as they are more efficient than the Vee thread. The square thread is most efficient but difficult to produce and hence becomes costly. The adjustment for wear in square thread is very difficult but can be easily achieved in the Acme threads, by splitting the nut along the axis. The Acme threads thus can be used as power transmission element when power is to be transmitted in both the directions.

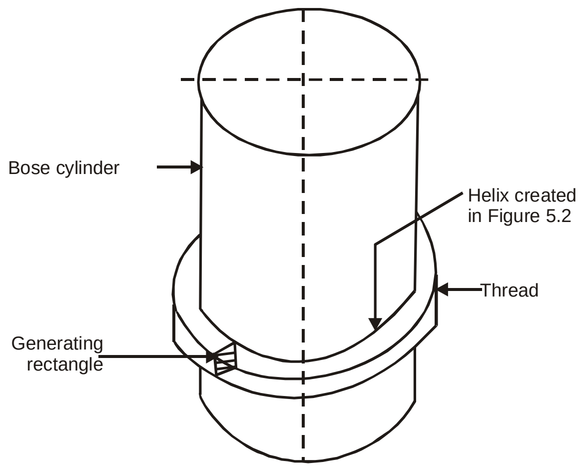


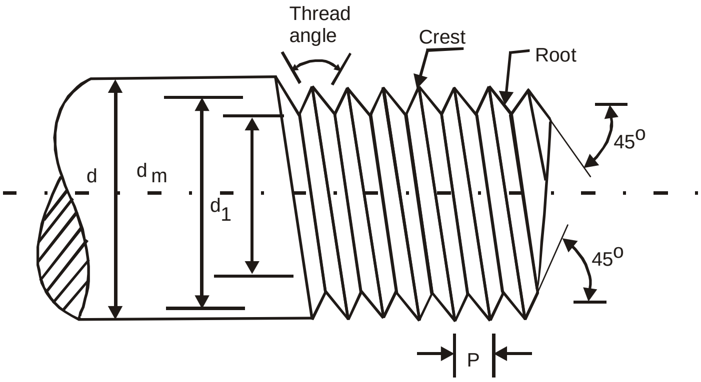
Figure 5.3 : Generating a Square Section Thread on Cylindrical Surface

There is little or no backlash in the Acme threads which are commonly used as feed and lead screws of machine tools. The buttress thread having one side flat and other sloping combines the advantage of square thread and Acme thread. The flat side provides the efficiency of power transmission while the inclined side provides the ease of adjustment.

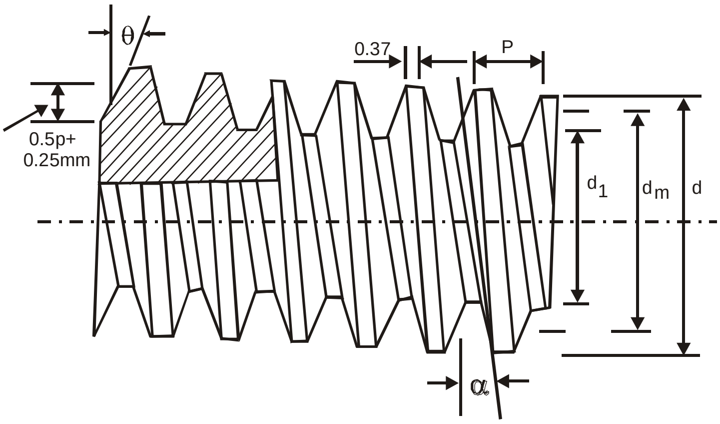
However, these advantages become possible only when the power is transmitted in one direction. Vee threads for their lower efficiency for power transmission are used as fasteners. Due to sides being inclined the effective coefficient of friction between the screw and the nut increases. Figure 5.4 shows Vee or triangular, the Acme and the buttress threads with leading nomenclature.

The major diameter is the largest diameter of the screw thread denoted by d for external thread and by D for internal thread. *Minor diameter* (d_1 or D_1) is the smallest diameter of the screw. Some times more than one thread may be cut on the screw. These multiple threads may be easily seen at the end of the screw where more than one thread will appear to start. Multiple start threads give the advantage that screw can move through a longer distance in the nut when given one rotation as compared to the screw with a single thread or start. The distance moved by a screw along its axis when given one rotation is called the *lead*. Apparently

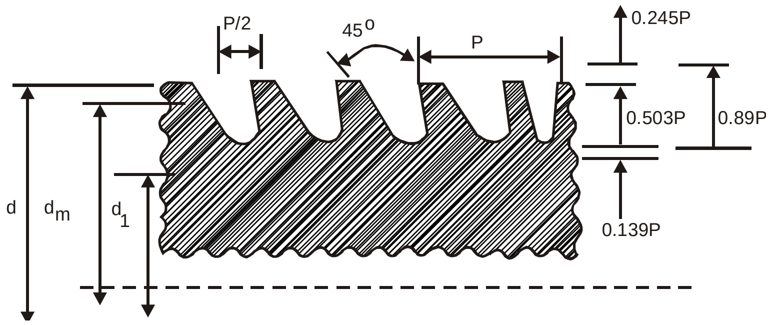
$$\text{lead} = \text{number of starts} \times \text{pitch} \qquad \ldots (5.2)$$



(a) Vee Thread



(b) Acme Thread



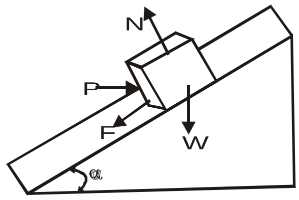
(c) Buttress Thread

Figure 5.4

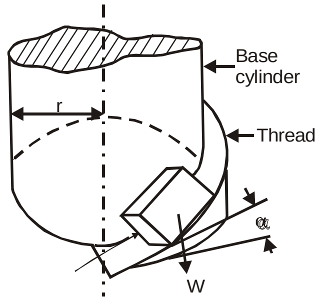
The V-threads are standardized by several organizations wherein nominal major diameter, pitch and pitch diameters are described. According to IS : 1362-1962 a screw thread (VEE type) is designated by letter *M* followed by nominal major diameter. An *M* 1.6 thread has nominal major diameter of 1.6 mm. IS : 4694-1968 describes basic dimensions of square threads.

5.3 MECHANICS OF SCREW AND NUT PAIR

It was mentioned earlier that the motion between nut and screw is like a body moving on an inclined plane. Figures 5.5(a) and (b) will explain this motion. The body of weight *W* is pushed up the inclined plane by a force *P* which acts upon the body horizontally. This inclined plane is bent round a cylinder in Figure 5.5(b) and aome body is being pushed up the plane while force *P* remains horizontal but also tangential to circular path of the body. This illustrates how the motion of the nut on the thread is similar to motion of a body on an inclined plane. The weight of the body on the inclined plane is replaced by weight carried by the nut in axial direction. The force *P* is applied by the help of a wrench and *W* may be the reaction developed between surfaces of contact.



(a) Inclined Plane

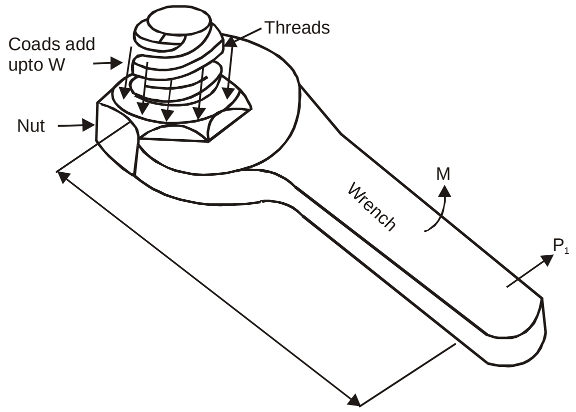


(b) Inclined Plane Wrapped Round a Base Cylinder to Form a Thread

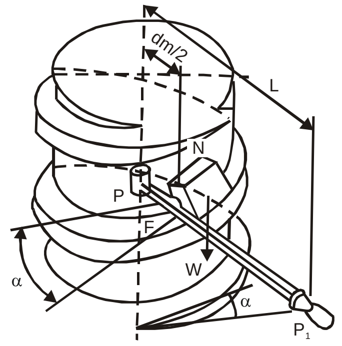
Figure 5.5

This is illustrated in Figures 5.6(a) and (b). Imagine that a lever is pivoted on the axis of the cylinder and pushes the body of weight *W* up the incline of helically wrapped plane on the cylinder. The lever touches the body at a radius *r* while a force P_1 is applied on the lever at an arm length of *L*. In the nomenclature we have already defined the outer diameter of the thread as major diameter, *d* and the smallest diameter as the diameter of the cylinder, d_1, (also called core diameter). *r* is $\frac{dm}{2}$ where d_m is the mean diameter of thread, being mean of *d* and d_1. If *P* be the force applied by the lever on the body of weight *W*, then by taking moments of forces, acting upon the lever, about the axis of the cylinder.

$$P = \frac{2P_1 L}{d_m} \qquad \ldots (5.3)$$



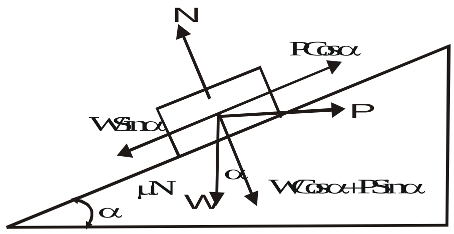
(a)



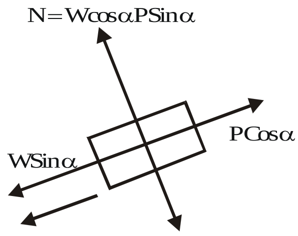
(b)

Figure 5.6

The force P has to overcome the friction as well as cause lifting of the body in vertical direction. To find the relationship between force P, called effort, and weight of the body, W, we have to consider the equilibrium of the body on an inclined plane as shown in Figures 5.7(a) and (b). The free body diagram clearly shows forces along and perpendicular to the inclined plane.



(a) A Body being Pushed Up the Inclined Plane with Angle of Inclination *a*, by a Horizontal Force *P*



(b) Free Body Diagram of Body of Weight *w*

Figure 5.7

The sum of the forces perpendicular to the plane and sum of the forces along the plane should separately be zero to satisfy the conditions of equilibrium. μ is taken as coefficient of friction between the body and the plane, which is same as coefficient of friction between the nut and screw thread surfaces. $\mu = \tan \phi$ where ϕ is the angle of friction.

Summing up the forces perpendicular to the plane.

The normal reaction,

$$N = W \cos \alpha + P \sin \alpha$$

Hence, force of friction between the surfaces of contact

$$F = \mu N = \mu W \cos \alpha + \mu P \sin \alpha$$

Summing up the forces parallel to the inclined plane

$$P \cos \alpha = W \sin \alpha + \mu W \cos \alpha + \mu P \sin \alpha \qquad \text{. . . (i)}$$

Replacing $\quad \mu$ by $\tan \phi = \dfrac{\sin \phi}{\cos \phi}$

$$P \cos \alpha - \frac{\sin \phi}{\cos \phi} P \sin \alpha = W \sin \alpha + \frac{\sin \phi}{\cos \phi} W \cos \alpha$$

$$p \cos \alpha \cos \phi - P \sin \alpha \sin \phi = W \sin \alpha \cos \phi + W \cos \alpha \sin \phi \qquad \text{. . . (ii)}$$

$$P \cos (\alpha + \phi) = W \sin (\alpha + \phi)$$

$$P = W \tan (\alpha + \phi) \qquad \text{. . . (5.4)}$$

If there is no friction, $\phi = 0$ and effort in such a case is called *ideal effort* denoted by P_i where

$$P_i = W \tan \alpha \qquad \text{. . . (5.5)}$$

Hence the efficiency of an inclined plane with inclination of α or the efficiency of a screw having helix angle α is

$$\eta = \frac{P_i}{p}$$

$$= \frac{\tan \alpha}{\tan (\alpha + \phi)} \qquad \ldots (5.6)$$

If in a situation as shown in Figure 5.7(a), P is removed, will the body slide down? Obviously it will depend upon the fact as to how large angle α is. If $W \sin \alpha > \mu W \cos \alpha$ the body will slide down under its own weight (Examine (i) and (ii) with $P = 0$). Same thing will happen in case of a nut in Figure 5.6 (a), i.e. when effort P_1 is removed from the wrench or wrench is removed, the nut will rotate back under load W. It means the nut is not self-locking. However, if α is reduced it can be seen that at $\alpha = \phi$, the downward component (along the plane) of weight W, i.e. $W \sin \alpha$ and friction force (along the plane) $\mu W \cos \alpha$ become equal and the body remains just stationary or the nut does not move down. If $\alpha < \phi$, the body will need a force to act so as to push it down. If this force is P' then

$$P' = W \tan (\phi - \alpha) \qquad \ldots (5.7)$$

Naturally screws of $\alpha > \phi$, will not be self locking or in other words they cannot act as fasteners. If, however, the angle $\alpha < \phi$, an effort P', given by Eq. (5.7) will be required to unscrew the nut, and such screws can be used as fasteners.

The Eq. (5.6) which defines the efficiency of the screw and that the condition for screw to be self locking is that $\alpha \leq \phi$ can be used to determine the maximum efficiency of a self locking screw and nut pair.

For self locking condition, the efficiency

$$\eta_s \leq \frac{\tan \phi}{\tan (\phi + \phi)}$$

$$\leq \frac{\tan \phi}{\tan 2\phi} \leq \frac{\tan \phi \, (1 - \tan^2 \phi)}{2 \tan \phi}$$

$$\leq \frac{1 - \tan^2 \phi}{2}$$

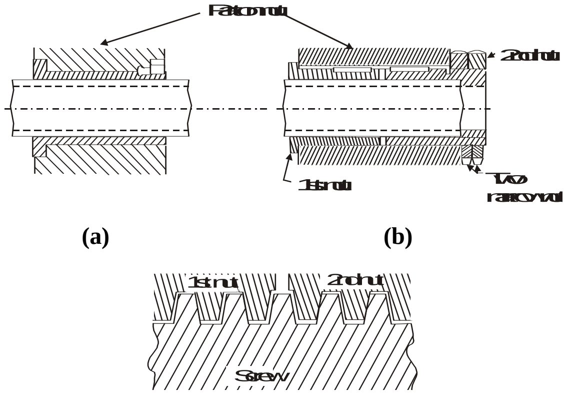
Since $\tan^2 \phi$ is always less than 1, ($\mu = \tan \phi$)

$$\eta_s < \frac{1}{2} \qquad \ldots (5.8)$$

The screw having efficiency greater than 50% is said to over haul, meaning the load W will cause the nut to roll down.

5.4 POWER SCREW MECHANICS

In the preceding section the simple case of a square thread was considered. As will be seen in the text that follows that the square thread is more efficient than the Acme thread because in the Acme thread the effective coefficient of friction increases, yet for power screws it is the Acme thread which is used more predominantly. The Acme thread can be machined more easily than the square thread and more importantly the clearance in the Acme thread can be adjusted to take care of the wear or machining inaccuracy.
Figure 5.8 shows the nuts in pair with square and the Acme threads and an adjusting mechanism for the acme thread.



(a) (b)

(c)

Figure 5.8 : (a) A Square thread and Nut (b) An Acme Thread and Two Nuts for Adjusting Clearance on Both sides of the Thread. Two Narrow nuts Threading on Outside of the nut Push the Two Nuts in Opposite Direction (c) Adjustment of Clearance on Two Sides of thread

An Acme thread has two inclination. Firstly the plane of the thread is sloping along angle of helix in the direction of the helix. The plane of the thread also slopes away from the circumference of the screw, i.e. the circumference of diameter d_1, The same is true for the V-thread. Both types of threads are as shown in Figures 5.4(a) and (b). The effect of inclination in the radial direction is to increase the normal reaction between the nut and the screw. This inclination in the radial direction of thread gives a shape of trapezium of angle 2θ as shown in Figure 5.9 and since the motion will occur perpendicular to the plane of paper; the force of friction will depend upon the normal reaction.

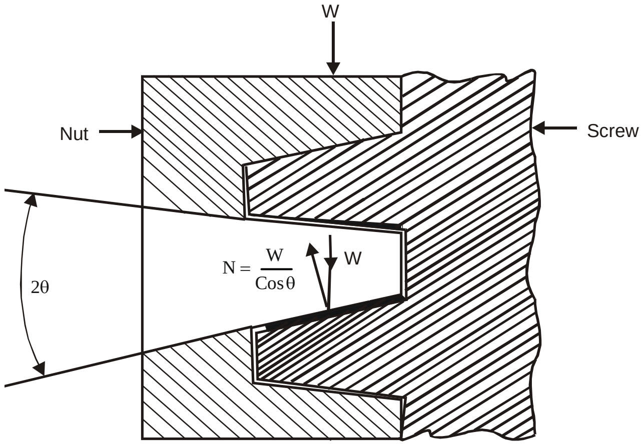


Figure 5.9 : A Nut Moving on a Screw having Acme Thread

For a vertical force W pressing the nut on the thread of screw, the normal reaction is N. Resolving N in the vertical direction and equating with W

$$N \cos \theta = W \quad \text{or} \quad N = \frac{W}{\cos \theta}$$

and hence the force of friction along the direction of helix is μN or $\frac{\mu W}{\cos \theta}$ which can also be written as $\mu' W$ and μ' can be called a modified or effective coefficient of friction. No doubt you can see that

$$\mu' = \frac{\mu}{\cos \theta} > \mu \qquad \ldots (5.9)$$

It is because cos θ is less than 1. Greater the angle θ, lesser the cos θ and hence μ′ will increase with increasing θ. This is what happens in V-thread. The force of friction between nut and thread in V-threads is greater than in Acme thread. The P-W

relationship given by Eq. (5.4) stands valid for square thread and can be modified for Acme thread by replacing ϕ by ϕ' where

$$\phi' = \tan^{-1} \mu'$$

$$P = W \tan (\alpha + \phi') \qquad \ldots (5.10)$$

Efficiency of the Acme thread will be

$$\eta' = \frac{\tan \alpha}{\tan (\alpha + \phi')} \qquad \ldots (5.11)$$

A word about the horizontal component of N, which is $N \sin \theta$ will be in order. Remember we are talking about the thread round the circumference of the screw. There is other side of the screw on right of Figure 5.9 and $N \sin \theta$ there will be acting to the right. Thus the horizontal components of N are balanced.

The force P which acts as tangent to the mean circle of diameter d, between the outer circle of diameter d and inner circle of diameter d_1, i.e. at radius $\frac{d_m}{2} = \frac{d + d_1}{4}$ will cause a moment

$$M_t = \frac{P d_m}{2}$$

or $$M_t = W \tan (\alpha + \phi') \frac{d_m}{2} \qquad \ldots (5.12)$$

Here ϕ' is the effective angle of friction which could be $\phi = \tan^{-1} \mu$ if the square thread is on the screw. Apparently the torque in Eq. (5.12) will twist the cylinder of screw and cause shearing stress in it. The cylinder is acted upon by an axial compression also. The axial compressive force causes compressive stress at any point in the section.

Example 5.1

A square threaded screw is required to work against an axial force of 6.0 kN and has following dimensions.

Major diameter d = 32 mm; pitch p = 4 mm with single start, μ = 0.08. Axial force rotates with the screw.

Calculate :

(a) Torque required when screw moves against the load.

(b) Torque required when screw moves in the same direction as the load.

(c) Efficiency of the screw.

Solution

Remember the relationship between p, d and d_1 which has been shown in Figure 5.1.

$$p = d - d_1$$

But $$d_m = \frac{d + d_1}{2} \text{ or } d_1 = 2d_m - d$$

$$p = 2 (d - d_m)$$

or $$d_m = d - \frac{p}{2}$$

Using d = 32 mm and p = 4 mm

$$d_m = 32 - 2 = 30 \text{ mm} \qquad \ldots \text{(i)}$$

The angle of helix is related to the circumference of mean circle and the pitch from description of Section 5.2.

$$\tan \alpha = \frac{p}{\pi d_m} = \frac{4}{\pi 30} = 0.042 \qquad \text{. . . (ii)}$$

$\therefore$ $$\alpha = 2.4^o \qquad \text{. . . (iii)}$$

and $$\tan \phi = \mu = 0.08 \qquad \text{. . . (iv)}$$

$$\tan \phi = 4.57^o$$

From Eq. (5.12) torque required to move screw against load

$$M_t = W \tan (\alpha + \phi) \frac{d_m}{2}$$

$$= 6 \times \tan (2.4 + 4.57) \times \frac{30}{2} = 6 \times 0.12225 \times 15 \text{ Nm}$$

$$= 11 \text{ Nm} \qquad \text{. . . (v)}$$

When screw moves in the same direction, is a case in which the body moves down the inclined plane. In this case the fore P' to push down is given by Eq. 5.7. Hence, the torque

$$M_t' = P' \frac{d_m}{2} = W \tan (\phi - \alpha) \frac{d_m}{2}$$

$$= 6 \times \tan (4.57 - 2.4) \times \frac{30}{2} = 6 \times 0.038 \times 15 \text{ Nm}$$

$$= 3.42 \text{ Nm} \qquad \text{. . . (vi)}$$

From Eq. (4.6), efficiency

$$\eta = \frac{\tan \alpha}{\tan (\alpha + \phi)} = \frac{0.042}{0.12225} = 0.344$$

or $$\eta = 34.4\% \qquad \text{. . . (vii)}$$

Example 5.2

If in the Example 5.1, the screw has the Acme thread with thread angle $2\theta = 29^o$ instead of square thread, calculate the same quantities.

Solution

There is no difference in calculation for square and the Acme thread except that in case of the Acme thread the coefficient of friction is modified and effective coefficient of friction is given by Eq. (5.9).

$$\mu' = \frac{\mu}{\cos \theta} = \frac{0.08}{\cos \frac{29}{2}} = \frac{0.08}{0.968} = 0.0826$$

$\therefore$ $$\phi' = 4.724^o$$

From Figure 5.2(b) for the Acme thread note that

$$d_m = d - \frac{p}{2} - 0.125 \text{ mm}$$

$$= 32 - 2 - 0.125$$

or $$d_m = 29.875 \text{ mm} \qquad \text{. . . (i)}$$

$\therefore$ $$\tan \alpha = \frac{p}{\pi d_m} = \frac{4}{\pi \times 29.875} = 0.0426$$

$\therefore$ $$\alpha = 2.44^o$$

$$M_t = \frac{W}{2} d_m \tan(\alpha + \phi')$$

$$= 3.0 \times 29.875 \times \tan(2.44 + 4.724) = 89.625 \times 0.126 \text{ kNmm}$$

or $$M_t = 11.265 \text{ Nm} \qquad \ldots \text{(ii)}$$

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \phi)} = \frac{\tan 2.44}{\tan(2.44 + 4.724)} = \frac{0.0426}{0.126}$$

or $$\eta = 33.8\% \qquad \ldots \text{(iii)}$$

When the screw moves in the same direction as the load, the torque

$$M_t' = \frac{W d_m}{2} \tan(\phi' - \alpha)$$

$$= \frac{6 \times 29.875}{2} \tan(4.724 - 2.44) \text{ kNmm}$$

or $$M_t' = 3.58 \text{ Nm} \qquad \ldots \text{(iv)}$$

Comparing the results of Examples 5.1 and 5.2 we can see that the screws have got same major diameter and pitch and for this reason their helix angles are different. Coefficients of friction are inherently different. But the torque on the screw increases by 2.41% and efficiency decreases by 1.744%.

SAQ 1

(a) Distinguish between square and the Acme threads, the Acme threads and the V-threads. Also mention relations for pitch, various diameters.

(b) What do you understand by multi-start thread? Define lead and the pitch and give relation between them. If two threads are having same pitch but one is single start and other is three starts, which one will advance more and how much if screw is turned through one full rotation in the nut.

(c) What reason you can put forth for preferring the Acme threads to square threads?

(d) A horizontally fixed nut carries a vertical screw of square thread whose mean diameter is 50 mm, and the pitch is10 mm. On the top of the screw a circular disc 100 N weight and 100 mm diameter is fixed and this disc has radial hole into which a rod of 1.1 m is fixed such that 1 m length is out of the disc. If at the end of this rod an effort of 280 N is required to lift a load placed on the disc, calculate the load. The coefficient of friction between the threads of the screw and nut is 0.1.

5.5 APPLICATION OF POWER SCREW

Power screws are used in machines and equipment for lifting loads, applying pull forces, translating loaded machine parts and tools and for positioning devices. It can work in two modes, either with a fixed nut and moving screw or with a fixed screw and moving nut. The rotary motion can be given to any of the nuts or screw. The simplest device one can think of is a screw jack, often used for lifting heavy loads. The load can be placed on top of a platform (like disc as described in SAQ 1(d)), and with fixed nut the screw may be rotated with the help of a lever. However, the load will rotate with the screw. The alternative method would be to rotate the screw supported in vertical direction and obstruct the screw to rotate with the nut. The two alternatives are shown in Figure 5.10.

The lead screw of a lathe machine, which moves the tool carriage, is another example of power screw in which the screw rotates in a nut and screw is supported like a shaft between two bearings. The thrust is caused on the nut, which is integral part of the tool carriage. The nut moves along the length of the screw taking the carriage. The reaction of the thrust bears on the supports of the screw. The screw can be used for accurate positioning of the carriage if it is rotated by a separate stepper motor. The screw in transferring of force can also be used in hand operated punching machines, as a lifter of dam gate or as a presser of masses.

If there is a support like a collar, shown in Figure 5.10(a) on the top of which the load is placed so that it does not rotate, then the applied torque has to be equal to the sum of the torque required to rotate the screw in the nut and the friction torque between the surfaces of the collar and load platform. The friction torque between the supporting bearing surface and stationary surface may be reduced by lubrication or by providing rolling bearing as showing in Figure 5.10(b). In any given situation the torque at bearing surface will have to be calculated.

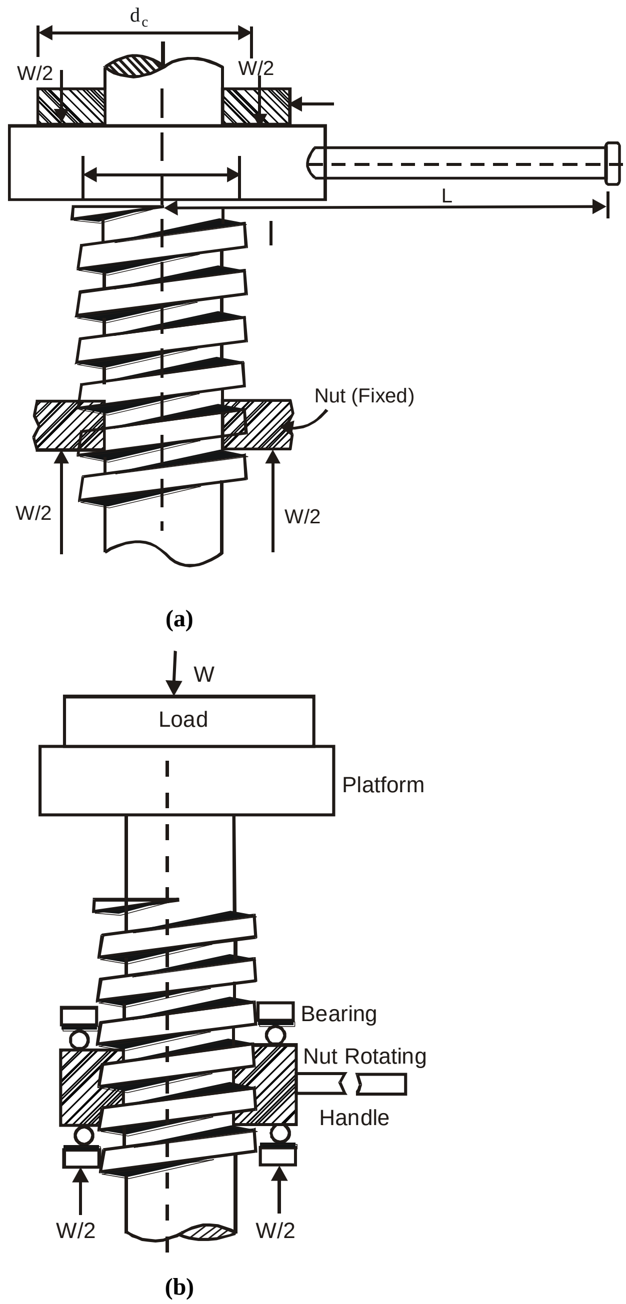


Figure 5.10

If the collar surface is like a flat disc of outer and inner diameters of d_o and d_i then the friction torque is given by

$$M_{tf} = \frac{2\mu_c W}{3} \frac{\left(\frac{d_o}{2}\right)^3 - \left(\frac{d_i}{2}\right)^3}{\left(\frac{d_o}{2}\right)^2 - \left(\frac{d_i}{2}\right)^2} = \mu_c W \frac{d_o^3 - d_i^3}{3\,(d_o^2 - d_i^2)} \qquad \ldots (5.13)$$

The Eq. (5.13) has not been derived here. In the above equation μ_c is the coefficient of friction between the collar and the platform or between bearing surfaces. Hence M_{tf} can also be writer as $\mu_c\ W_{rf}$ where r_f is the radius of an imaginary circle. Along the tangent of this circle the force of friction $\mu_e\ W$ is assumed to act. An approximate value of $r_f = \frac{d_o + d_i}{4}$ can also be used. Incidentally this value of r_f may be true for unlubricated surface which is not a reality.

5.6 STANDARD THREADS

Standards of threads describe pitch core diameter and major diameter. The standard threads can be cut in standard machine tools with standard cutters and designer can use them for calculation of sizes and ensure interchangeability. We will see in illustrated examples how the standards are used by designer. Presently we describe Indian standard IS 4694-1968 for square threads in which a thread is identified by its nominal diameter which is also the major diameter. According to standard the major diameter of nut is 0.5 mm greater then major diameter of the screw which will provide a clearance of 0.25 mm between the outer surface of screw and inner surface of nut thread. The basic dimensions of square threads are described in Table 5.1.

Table 5.1 : Basic Dimensions of Square Thread, (mm)

Pitch, p		5		
Core Dia. d_1	17	19	24	23
Major Dia. d	22	24	26	28
Pitch, p		6		
Core dia. d_1	24	26	28	30
Major dia. d	30	32	34	36
Pitch p		7		
Core dia. d_1	31	33	35	37
Major dia. d	38	40	43	44
Pitch, p		8		
Core dia. d_1	38	40	42	44
Major dia. d	46	48	50	56
Pitch, p		9		
Core dia. d_1,	46	49	51	53
Major dia. d	55	58	60	62
Pitch, p		10		
Core dia. d_1	55 58	60 62	65 68	70 72
Major dia. d	65 68	70 72	75 78	80 82

5.7 DESIGN OF SCREW AND NUT

Designing is calculating the dimensions, which can be seen in Figure 5.1. They are core diameter, d_i, major diameter d and pitch, p. the number of threads also has to be determined we have to realise that the load comes upon the screw as axial.

Compression causing compressive stress, which is uniformly distributed over circular cross section of diameter, d_1. As the screw rotates, in the nut it is subjected to a torque given by $P \frac{d_m}{2}$ or $W \tan(\alpha + \phi) \frac{d_m}{2}$. This torque will cause shearing stress, which will be maximum on the surface or at radius of $\frac{d_1}{2}$. The transfer of axial load between the screw thread and nut occurs through surface of thread. The pressure is to be kept within permissible limits, which normally is such that squeezing of oil film between contact surface should not occur. Further the thread and the cylindrical surface of the cylinder may tend to shear off under the load acting on the thread. Lasting we must realize that the axial load on screw makes the screw to act like a column. This column is not allowed to buckle. We will consider each of the above modes of failure to establish equations for calculating dimension.

Direct Stress

The axial load (force) is W, compressive in nature and the area which carries the force is the core cross section of diameter d_i. Hence, compressive stress, $\sigma = \frac{4W}{\pi d_1^2}$. We will see that this direct compressive stress combines with shearing stress to give principal and maximum shearing stresses. The resulting equations cannot be solved for d_1, hence the expression for σ is used to calculate d_1 from given permissible compressive stress. To account for other stresses, which we will see in next section, the magnitude of the compressive stress is increased by 30%.

Hence $$\sigma = \frac{1.3W}{\frac{\pi d_1^2}{4}} \quad \text{. . . (5.14)}$$

As an example if W = 50 kN and permissible compressive stress is 80 MPa, then

$$d_1^2 = \frac{4 \times 1.3 \times 50 \times 10^3}{\pi \times 80} = 1.0345 \times 10^3$$

By the helps of Table 5.2 you can see that nearest standard value value of d_1 is 33 mm with p = 7 mm and d = 40 mm. This apparently give all the information we require for a screw but we have to check for safety against other stresses.

Maximum Shearing Stress

We have already seen that torque $M_t \; W \tan(\alpha + \phi) \frac{d_m}{2}$ is required to rotate the screw to cause it to move against force W or to lift weight W. the torque will cause shearing stress in addition to direct compressive stress σ as stated earlier. The shearing stress at any point on surface of core

$$\tau = \frac{16 M_t}{\pi d_1^3}$$

The state of stress at any point on the surface of core of the screw will be compressive or a direct stress σ and a shearing stress τ as shown in Figure 5.11.

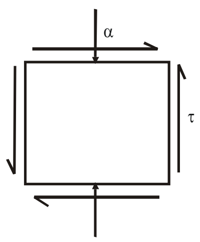


Figure 5.11 : State of Stress at any Point on Core Surface of Dia d_1

The maximum principal stress is given by

$$\sigma_{p1} = -\frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

This stress is not of much significance because it is reduced from higher magnitude σ or if it becomes tensile, to a magnitude which is still not much. However maximum shearing stress is significant.

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \sqrt{\left(\frac{4W}{2\pi d_1^2}\right)^2 + \left(\frac{16 M_t}{\pi d_1^3}\right)^2}$$

or $$\tau_{max} = \frac{16}{\pi} \sqrt{\frac{W^2}{64 d_1^4} + \frac{M_t^2}{d_1^6}} \quad \ldots (5.15)$$

The permissible shearing stress will be known but solving Eq. (5.15) will be too difficult as you can see that it contains fourth and sixth power of d_1 and M_t is also the function of d_m (or d_1 and d). Therefore it is recommended to calculate d_1 from Eq. (5.14) and using this value of d_1, calculate τ_{max}. Then you have to see that the calculated value of τ_{max} is less than permissible value of shearing stress.

Determining Number of Threads

The screw may be as long as required by consideration of geometry of machine. For example a lead screw may be as long as the length of the lathe bed. But in all cases the load transfer between the screw and nut will require total load to be shared among the threads on nut, which is smaller in length than the screw. The number of threads is decided on the basis of the load carried by thread surface perpendicular to core cylinder as shown in Figure 5.12. All threads in contact will carry axial force of the screw through uniformly distributed pressure p_b, in a square thread, the width and depth of each is equal to t. The thread section is shown on left hand side of Figure 5.12 and on right hand side wherein one thread is shown loaded by pressure. The same pressure will be acting on the thread of the nut. Area on which pressure is acting is the area between the compressive circles of diameters d and d_1 for one thread and if there are n threads in contact or n threads on the nut, then total area of contact to carry the pressure.

$$A_p = n\, \pi\, d_m\, t = n \frac{\pi}{4} (d^2 - d_1^2)$$

$\therefore$ Permissible pressure $$p_b = \frac{4W}{n\, \pi\, (d^2 - d_1^2)} = \frac{W}{n\, \pi\, d_m\, t} \quad \ldots (5.16)$$

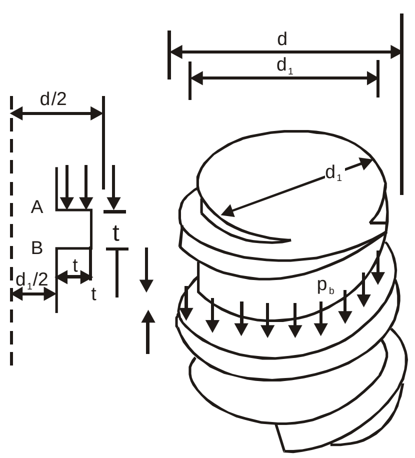


Figure 5.12 : Pressure on Contact Surface

This equation can be used for a check if the pressure between the threads of the nut and screw is within the permissible limit or it can be used to calculate the number of threads, *n*, in contact.

Shearing of Threads from the Core Cylinder

Because of the force *W*, acting as uniformly distributed pressure as shown in Figure 5.12 thread may have a tendency to shear. The area over which shear effect will occur is shown shaded in Figure 5.13. The thread is shown in broken lines. This area apparently is a strip of width, *t*, on the cylinder of diameter d_1. Hence the area

$$A_c = \pi\, d_1\, t$$

With shearing stress τ, which will be created at the bottom of *n* thread of screw,

$$\tau_{screw} = \frac{W}{n\, \pi\, d_1\, t} \qquad \ldots (5.17)$$

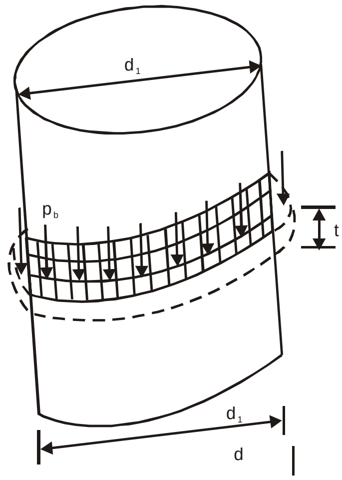


Figure 5.13 : The Area at the Bottom of a Thread in Screw

You must realise that the threads on the inside of the nut will also be similarly subjected to shearing stress at their bottom. Hence, in nut

$$\tau_{nut} = \frac{W}{n\, \pi\, d\, t} \qquad \ldots (5.18)$$

Normally, the screw and nut are not made in the same material. While screw is made in steel the preferred material for nut is either cast iron or bronze. The permissible value of shearing stress for nut material may be less and in that case Eq. (5.18) must be used to calculate *n*.

The height of the nut is simply the product if *n* and *p*, i.e.

$$h = np \qquad \ldots (5.19)$$

You must realise that the nut is threaded all along its length.

The various screw-nut material combinations are described in Table 5.2.

Table 5.2 : Screw-Nut Material Combination and Safe Bearing Pressure

Application	Material		Safe Bearing Pressure (MPa)	Rubbing Velocity at Mean Diameter m/min
	Screw	Nut		
Hand Press	Steel	Bronze	17.5-24.5	Well lubricated
	Steel	C.I	12.5-17.5	Low Velocity
Screw Jack	Steel	C.I	12.5-17.5	Velocity < 2.5
	Steel	Bronze	10.5-17.5	Velocity < 3.0
Hoisting Machine	Steel	C.I	4.0-7.0	6-12
	Steel	Bronze	35.0-100.0	6-12
Lead Screw	Steel	Bronze	10.5-17.0	> 15.0

Example 5.3

A screw press is required to exert a force of 50 kN when applied torque is 560 Nm. The unsupported length of the screw is 450 mm and a thrust bearing of hardened steel on cast iron is provided at the power end.

The permissible stresses in the steel screw are :

Tension and compression – 85 MPa, Shear – 55 MPa,

The permissible bearing pressure is 13.5 MPa for steel screw and C.I nut

The permissible shearing stress in the CI is 20 MPa

The yield strength of steel of screw, σ_Y = 260 MPa

The coefficient of friction in screw and nut is 0.15

Determine the dimensions of screw and nut, and efficiency.

Solution

Step 1

Determine diameter d_1, d, p, α to define the thread

Eq. (5.14), will be used to estimate d_1,

$$\sigma = \frac{1.3W}{\frac{\pi\, d_1^2}{4}}$$

Use W = 50,000 N, σ= 85 N/mm^2

$$\therefore \qquad d_1^2 = \frac{4 \times 1.3 \times 50{,}000}{\pi \times 85} = 973.54$$

$$\therefore \qquad d_1 = 31.2 \text{ mm}$$

Look in the Table 5.1, close to 31.2 mm the core diameter is 33 mm with pitch p = 7 mm. However, you may also use following rule : between d_1 = 30 and d_1 = 40 mm, p = 0.2 d_1.

We choose to use Table 4.1.

$$\text{So} \qquad d_1 = 33 \text{ mm},\ d = 40 \text{ mm},\ p = 7 \text{ mm} \qquad \text{. . . (i)}$$

$$d_m = \frac{d_1 + p}{2} = 33 + 3.5 = 36.5 \text{ mm}$$

$$\text{Angle of helix, } \alpha = \tan^{-1} \frac{p}{\pi d_m} = \tan^{-1} \frac{7}{\pi \times 36.5} = \tan^{-1} 0.061$$

$$\alpha = 3.5^o \qquad \text{. . . (ii)}$$

$$\tan \phi = \mu = 0.15,\ \phi = \tan^{-1} 0.15$$

$$\phi = 8.53^o \qquad \text{. . . (iii)}$$

$$\text{Efficiency of screw, } \eta = \frac{\tan \alpha}{\tan (\alpha + \phi)} = \frac{\tan 3.5}{\tan (3.5 + 8.53)} = \frac{0.061}{0.213}$$

$$\eta = 28.6\% \qquad \text{. . . (iv)}$$

Number of threads in contact, i.e. threads in nut and height of nut.

Use Eq. (5.18), put $t = \frac{p}{2} = 3.5$ mm, $\tau_{nut} = 20$ N/mm^2

$$\tau_{nut} = \frac{W}{n\,\pi d\,t}$$

$$20 = \frac{50,000}{n\,\pi \times 40 \times 3.5}$$

$$n = 5.684 \text{ say } 6$$

This has to be checked for bearing pressure, $p_b = 13.5$ N/mm^2

Use Eq. (5.16)

$$p_b = \frac{4W}{n\,\pi\,(d^2 - d_1^2)}$$

$$\therefore \quad n = \frac{4 \times 50,000}{\pi \times 13.5\,(40^2 - 33^2)} = 9.23 \text{ say } 10$$

This n is greater than the earlier calculated 6. Hence $n = 10$ is chosen.

Check for maximum shearing stress

$$M_t = W \tan(\alpha + \phi)\,\frac{d_m}{2}$$

Use $$d_m = \frac{d + d_1}{2} = \frac{40 + 33}{2} = 36.5 \text{ mm}$$

$$M_t = 50,000 \times \tan(3.5 + 8.53) \times \frac{36.5}{2} = 19.4 \times 10^4 \text{ Nmm}$$

$$\tau = \frac{16 M_t}{\pi\,d_1^3} = \frac{16 \times 19.4 \times 10^4}{\pi \times (33)^3} = \frac{98.8 \times 10^4}{3.594 \times 10^4}$$

or $$\tau = 27.4 \text{ N/mm}^2$$

Also note $$\sigma = \frac{4W}{\pi d_1^2} = \frac{4 \times 50,000}{\pi \times (33)^2} = 58.5 \text{ N/mm}^2$$

$$\therefore \quad \tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \sqrt{855.56 + 750.76} = \sqrt{1606.32}$$

$$\therefore \quad \tau_{max} = 40.1 \text{ N/mm}^2$$

The permissible shearing stress is 55 N/mm^2.

Hence screw in safe against shearing.

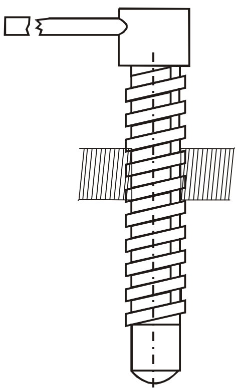


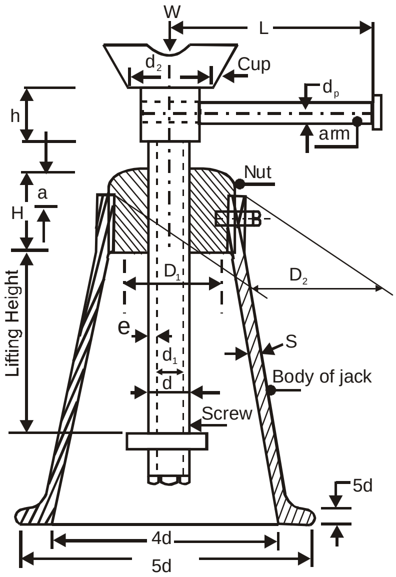
Figure 5.14

Example 5.4

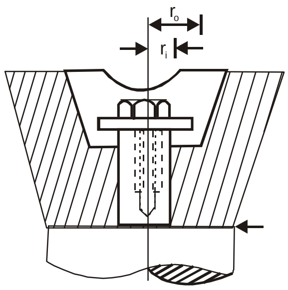
Design a screw jack to lift a load of 100 kN through a height of 300 mm. Assume σ_u = 400 MPa, τ_u = 200 MPa, σ_Y = 300 MPa, p_b = 10 MPa. The outer diameter of bearing surface is 1.6 d_1 and inner diameter of bearing surface is 0.8 d_1. Coefficient of friction between collar on screw and C.I is 0.2. Coefficient of friction between steel screw and bronze nut is 0.15. Take a factor of safety of 5 for screw and nut but take a factor of safety of 4 for operating lever.

Solution

The screw jack to be designed is shown in Figures 5.15(a) and (b) shows the details of the cup on which the load *W* is to be carried.



(a)



(b)

Figure 5.15 : (a) Screw Jack Assembled; and (b) The Load Cup

The overall design of screw jack comprises designing of

(a) Screw,

(b) Nut,

(c) Arm,

(d) Cup, and

(e) Body of the jack.

Screw

Screw becomes the central part. The other parts will be dependent upon the screw. The screw design will decide core diameter d_1, major diameter d and pitch, p. Hence screw design will consist of calculating d_1, d and p and checking for maximum shearing stress and buckling of the screw.

Assume square thread

Use Eq. (5.14) with W = 100,000 N, $\sigma = \frac{400}{5} = 80 \text{ N/mm}^2$

$$\sigma = \frac{1.3W}{\frac{\pi d_1^2}{4}}$$

$$\therefore \qquad d_1 = \left(\frac{5.2 \times 100,000}{\pi \times 80}\right)^{\frac{1}{2}} = (2069)^{\frac{1}{2}}$$

or $\qquad d_1 = 45.5$ mm

From Table 5.1, choose next higher value as $d_1 = 46$ mm with $p = 9$ mm and $d = 55$ mm.

After estimated values we go to check for maximum shearing stress and the buckling of the screw.

Maximum Shearing Stress

Compressive stress, $\sigma = \frac{4W}{\pi d_1^2} = \frac{4 \times 10^5}{\pi \times (46)^2} = 60.17 \text{ N/mm}^2$

Torque on the screw, $M_t = W \tan (\alpha + \phi) \frac{d_m}{2}$

$$d_m = \frac{d + d_1}{2} = \frac{55 + 46}{2} = 50.5 \text{ mm}$$

$$\alpha = \tan^{-1} \frac{p}{\pi d_m} = \tan^{-1} \frac{9}{\pi \times 50.5} = \tan^{-1} 0.0567 = 3.25^{\text{o}}$$

$$\phi = \tan^{-1} \mu = \tan^{-1} 0.15 = 8.53^{\text{o}}$$

$$M_t = 10^5 \tan (3.25 + 8.53) \times \frac{50.5}{2} = \frac{0.21 \times 50.5 \times 10^5}{2}$$

$$= 5.266 \times 10^5 \text{ Nmm}$$

$$\tau = \frac{16 M_t}{\pi d_1^3} = \frac{16 \times 5.266 \times 10^5}{\pi \times (46)^3} = 27.6 \text{ N/mm}^2$$

$$\tau_{\max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \sqrt{\left(\frac{60.17}{2}\right)^2 + (27.6)^2} = \sqrt{1666.87} = 40.827 \text{ N/mm}^2$$

The permissible sharing stress is $\frac{\tau_u}{5} = \frac{220}{5} = 44 \text{ N/mm}$.

Thus the screw is safe against shear.

Nut

For standard square thread the depth or thickness of the thread,

$$t = \frac{p}{2} = \frac{9}{2} = 4.5 \text{ mm}$$

Use Eq. (5.16), $p_b = \frac{W}{n \pi d_m t}$ with $p_b = 10$ N/mm^2, $d_m = 50.5$ mm, $t = 4.5$ mm, $W = 10^5$ N

$$\therefore \qquad n = \frac{10^5}{10\pi \times 50.5 \times 4.5} = 14$$

Normally n >10 is not preferred. So we can go for higher d_1, d and p. Next standard values will be d_1 = 49 mm, d = 58 mm, p = 9 mm.

Hence, $d_m = \frac{58 + 49}{2} = \frac{107}{2} = 53.5$ mm, $t = \frac{p}{2} = 4.5$ mm

$$n = \frac{10^5}{10\pi \times 53.5 \times 4.5} = 13.2$$

Since this is also greater than 10, we can go for next higher value of d_1 = 51 mm with d = 60 mm and p = 9 mm.

So that $\qquad d_m = \frac{51 + 60}{2} = \frac{111}{2} = 55.5$ mm

which will also not satisfy the condition.

The last choice in the table with same p is d_1 = 53 mm and d = 62 mm.

So that $d_m = \frac{53 + 62}{2} = \frac{115}{2} = 57.5$ mm which result in n = 12.3 mm.

Still better solution is to go for next series with p = 10 mm.

With d_1 = 55, d = 65 mm, and $d_m = \frac{120}{2} = 60$; t = 5 mm

$$\therefore \qquad n = \frac{10^5}{10\pi \times 60 \times 5} = 10.6$$

Which is very close to 10, hence can be accepted.

So the solution changes to d_1 = 55 mm, d = 65 mm, p = 10 mm. You need not check these dimensions because they are larger than the safe ones. (It is important that the reader understands the reiterative nature of design and how the help from standards is derived. The iterations have been done to emphasize that the exercise in design should not be treated as a problem in strength of materials. In design the problem serves to bring practicability in focus.

The length of the nut, $H = np = 10.6 \times 10 = 106$ mm.

Outside Diameter of Nut; see Figure 5.1(b) and Figure 5.9.

The nut is in tension. The section to bear tensile stress is

$$\frac{\pi}{4} (D_0^2 - D^2)$$

$$D = d + 0.5 \text{ mm} = 65 + 0.5 = 65.5 \text{ mm}$$

Bronze is not as strong as steel. Silicon bronze (Cu = 95%, Si = 4%, Mn = 1%) is quite good for making nut. This material is available in wrought condition with σ_u = 330 MPa. If factor of safety of 5 is used, then permissible tensile stress is $\frac{330}{5} = 66$ N/mm^2.

$$\therefore \qquad \sigma_t = \frac{4W}{\pi (D_0^2 - D^2)} = 66 = \frac{4 \times 10^5}{\pi (D_0^2 - 655^2)}$$

$$\therefore \qquad D_0^2 = 1929 + 4290 = 6219$$

or $\qquad D_0$ = 78.86 say 79 mm

To fit the nut in position in the body of jack, a collar is provided at the top (see Figure 5.15(a)).

The thickness of the collar = 0.5 D = 0.5 × 65.5 = 32.72 mm

The outside diameter of the Collar D_{00} can be found by considering crushing of collar surface under compressive stress. The area under compression is $\frac{\pi}{4} (D_{00}^2 - D_0^2)$.

The compressive strength of bronze is same as tensile stress, 66 MPa

$$66 = \frac{4W}{\pi (D_{00}^2 - 79^2)} = \frac{4 \times 10^5}{\pi (D_{00}^2 - 6241)}$$

$$D_{00} = \sqrt{1929 + 6241} = \sqrt{8170} = 90.4 \text{ mm}$$

The nut is shown in Figure 5.16.

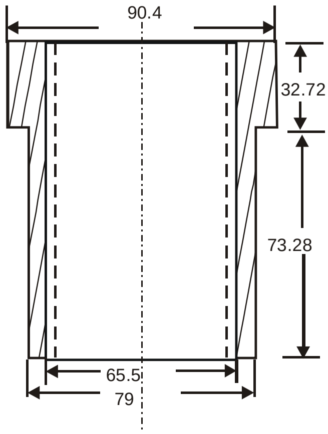


Figure 5.16

Arm

The arm is used to rotate the screw in the stationary nut. The portion of the screw at the top is enlarged to a diameter, $2r_0$
where $2r_0$ =1.6 d_1 = 1.6 × 55 = 88 mm (see Figure 5.15(b)).

$2r_0$ is the outer diameter for bearing surface of collar on which will rest the load cup. The cup will turn around a pin of dia. $2r_i$ = 0.8 d_1,
$2r_1$ = 0.8 × 55 = 44 mm.

These two diameters will be used to calculate the torque of friction at bearing surface. Call this torque M_{tf} and use Eq. (5.13)

$$M_{tf} = \frac{\mu_c W}{3} \frac{(2r_0)^3 - (2r_i)^3}{(2r_0)^2 - (2r_i)^2}$$

μ_c = 0.2 and all other values are described earlier

$$M_{tf} = \frac{0.2 \times 10^5}{3} \frac{(88)^3 - (44)^3}{(88)^2 - (44)^2} = \frac{0.2 \times 10^5 \times (44)^3}{3 \times (44)^2} \frac{8-1}{4-1}$$

$$= 6.84 \times 10^5 \text{ Nmm}$$

The arm will have to apply this torque along with the torque for lifting the load which is given as :

$$M_t = W \tan (\alpha + \phi) \frac{d_m}{2}$$

$\phi = 8.53^o$, $d_m = 60$ mm, $\alpha = \tan^{-1} \dfrac{p}{\pi d_m} = \tan^{-1} \dfrac{10}{\pi \times 60} \tan^{-1} 0.05 = 3^o$

$$M_t = 10^5 \times 30 \tan (11.53) = 6.12 \times 10^5 \text{ Nmm}$$

Total torque $M = M_{tf} + M_t = (6.84 + 6.12) \times 10^5 = 13 \times 10^5$ Nmm

A single man can apply 400 N of force but two men can apply 80 N with an efficiency of 90%. Let's assume two persons work at the end of the arm, the length of the arm

$$L = \frac{13 \times 10^5}{0.9 \times 800} = 1805.5 \text{ mm or } 1.805 \text{ m}$$

Actual length will incorporate allowance for grip and insertion in collar so arm of 2.1 m length will be appropriate.

The torque M will acts as bending moment on the arm with permissible bending stress as $\dfrac{400}{2.5} = 160$ N/mm^2. The diameter of arm

$$d_a = \left(\frac{32 M}{\pi \sigma_b}\right)^{\frac{1}{3}} = \left(\frac{32 \times 13 \times 10^5}{\pi \times 160}\right)^{\frac{1}{3}} = 43.75 \text{ mm}$$

The Cup

The shape of the cup is shown in Figure 5.15(b). It can be made in C1 and side may incline 30^o with vertical. With bottom diameter as 1.2 ($2r_i$), its height can be decided by the geometry of load to be lifted. Since no information is given we leave this design only at shape.

The Body of the Jack

Height = Lifting height +length of the nut – length of the collar on the nut + allowance for bottom plate of 2 mm thickness and head of the bolt holding plate.

= 300 + 106 – 32.72 + (2 + 4.8) (using a 6 mm bolt)

= 380.1 mm

Thickness, $\delta = 0.25\, d = 0.25 \times 65 = 16.25$ mm

Thickness of base, $\delta_1 = 0.5\, d = 0.5 \times 65 = 32.5$ mm

Diameter of base, $D_2 = 4.0\, d = 4 \times 65 = 260$ mm

Diameter of base (outside), $D_3 = 5d = 5 \times 65 = 325$ mm.

Efficiency

$$\eta = \frac{\tan \alpha}{\tan (\alpha + \phi)} = \frac{0.05}{0.204} = 24.5\%$$

Listing of Designed Dimensions.

Screw : Square Thread.

Core diameter of screw, d_1 = 55 mm

Major diameter of screw, d = 65 mm

Pitch of Thread, p = 10 mm.

Nut

Number of threads, $n = 10.6$

Height of the nut, $H = 106$ mm

Outside diameter $D_0 = 79$ mm

Diameter of Nut collar, $D_{00} = 90.4$ mm

Thickness of collar = 32.72 mm

Arm of the Jack

The length of the arm = 2.1 m

The diameter of the arm = 43.57 mm

The Body of the Jack

Thickness of the body, $\delta = 16.25$ mm

Thickness of the base, $\delta_1 = 32.5$ mm

Inside diameter of base, $D_2 = 260$ mm

Outside diameter of base, $D_3 = 325$ mm

Height of the body $H_1 = 380.1$ mm

SAQ 2

(a) Mention industrial applications of screw.

(b) Describe steps involved in designing screw.

(c) Draw the assembled view of screw jack designed in Example 5.4 of Section 5.9.

(d) For a screw of $d_1 = 17$ mm, $p = 5$ mm, $d = 22$ mm subjected to axial compression of 4000 N, calculate maximum shearing stress and bearing pressure between threads of screw and nut. Calculate factor of safety in compression of screw, shearing of screw and bearing pressure if $\sigma_u = 320$ N/mm$_2$, $\tau_u = 212$ N/mm^2, maximum pressure = 12 N/mm^2. Take $\mu = 0.12$. There are 5 threads in nut.

(e) Acme threaded screw rotating at 60 rpm pulls a broaching cutter through a job. The tensile force in the screw is supported by a collar whose internal and external diameters are respectively 60 mm and 90 mm. Coefficient of friction for all contact surfaces is 0.15 and Acme thread angle is 30^o. The requirement of the tool displacement demands that the pitch of the Acme thread should be 10 mm and corresponding major screw diameter is 55 mm. the power consumed by the machine is 0.39 kW, calculate the axial load exerted upon the tool.

5.8 THREADED FASTENER

Apart from transmitting motion and power the threaded members are also used for fastening or jointing two elements. The threads used in power screw are square or Acme while threads used in fastening screws have a vee profile as shown in Figure 5.4(a). Because of large transverse inclination the effective friction coefficient between the screw and nut increases by equation $\mu' = \frac{\mu}{\cos \theta}$ where μ is the basic coefficient of friction of the pair of screw and nut, θ is the half of thread angle and μ' is the effective coefficient of friction. The wedging effect of transverse inclination of the thread surface

was explained in Section 5.4. According to IS : 1362-1962 the metric thread has a thread angle of 60^o. The other proportions of thread profile are shown in Figure 5.17. IS : 1362 designates threads by *M* followed by a figure representing the major diameter, *d*.
For example a screw or bolt having the major diameter of 2.5 mm will be designated as *M* 2.5.

The standard describes the major (also called nominal) diameter of the bolt and nut, pitch, pitch diameter, minor or core diameter, depth of bolt thread and area resisting load (Also called stress area). Pitch diameter in case of V-threads corresponds to mean diameter in square or Acme thread. Figure 5.15 shows pitch diameter as d_p. Inequality of d_m and d_p is seen from Figure 5.17. Tables 5.3 and 5.4 describes V-thread dimensions according to IS : 1362.

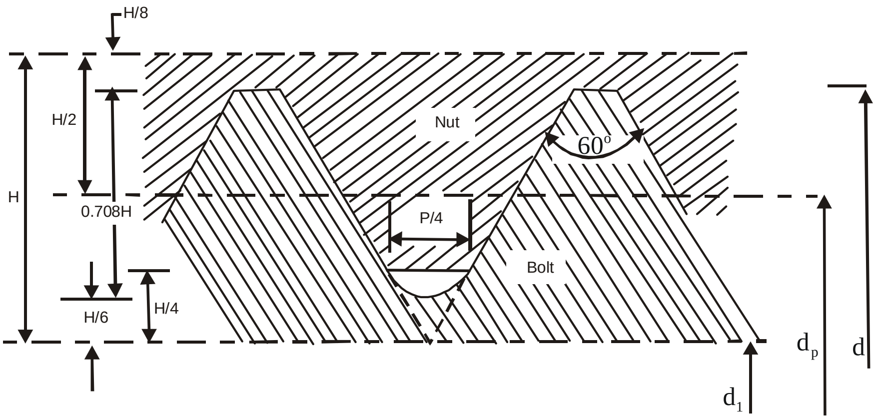


Figure 5.17 : Profiles of Fastener Threads on Screw (Bolt) and Nut

Table 5.3 : Dimensions of V-threads (Coarse)

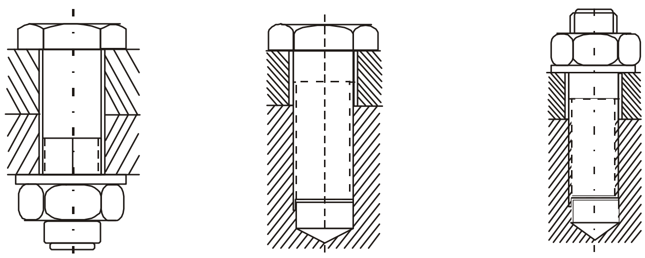
Designation	*p* (mm)	*d* or *D* (mm)	d_p (mm)	D_C (mm)		Thread Depth (mm)	Stress Area (mm^2)
				Nut	Bolt		
M 0.4	0.1	0.400	0.335	0.292	0.277	0.061	0.074
M 0.8	0.2	0.800	0.670	0.584	0.555	0.123	0.295
M 1	0.25	1.000	0.838	0.729	0.693	0.153	0.460
M 1.4	0.3	1.400	1.205	1.075	1.032	0.184	0.983
M 1.8	0.35	1.800	1.573	1.421	1.371	0.215	1.70
M 2	0.4	2.000	1.740	1.567	1.509	0.245	2.07
M 2.5	0.45	2.500	2.208	2.013	1.948	0.276	2.48
M 3	0.5	3.000	2.675	2.459	2.387	0.307	5.03
M 3.5	0.6	3.500	3.110	2.850	2.764	0.368	6.78
M 4	0.7	4.000	3.545	3.242	3.141	0.429	8.78
M 5	0.8	5.000	4.480	4.134	4.019	0.491	14.20
M6	1	6.000	5.350	4.918	4.773	0.613	20.10
M 8	1.25	8.000	7.188	6.647	6.466	0.767	36.60
M 10	1.5	10.000	9.026	8.876	8.160	0.920	58.30
M 12	1.75	12.000	10.863	10.106	9.858	1.074	84.00
M 14	2	14.000	12.701	11.835	11.564	1.227	115.00
M 16	2	16.000	14.701	13.898	13.545	1.227	157.00
M 18	2.5	18.000	16.376	15.294	14.933	1.534	192
M 20	2.5	20.000	18.376	17.294	16.933	1.534	245
M 24	3	24.000	22.051	20.752	20.320	1.840	353
M 30	3.5	30.000	27.727	26.211	25.706	2.147	561
M 36	4	36.000	33.402	31.670	31.093	2.454	976
M 45	4.5	45.000	42.077	40.129	39.416	2.760	1300
M 52	5	52.000	48.752	46.587	45.795	3.067	1755
M 60	5.5	60.000	56.428	54.046	53.177	3.374	2360

Table 5.4 : Dimensions of V-Threads (Fine)

Designation	p (mm)	d or D (mm)	d_p (mm)	D_C (mm)		Thread Depth (mm)	Stress Area (mm^2)
				Nut	Screw		
M 8 × 1	1	8.000	7.350	6.918	6.773	0.613	39.2
M 10 × 1.25	1.25	10.000	9.188	8.647	8.466	0.767	61.6
M 12 × 1.25	1.25	12.000	11.184	10.647	10.466	0.767	92.1
M 14 × 1.5	1.5	14.000	13.026	12.376	12.166	0.920	125
M 16 × 1.5	1.5	16.000	15.026	14.376	14.160	0.920	167
M 18 × 1.5	1.5	18.000	17.026	16.376	16.160	0.920	216
M 20 × 1.5	1.5	20.000	19.026	18.376	18.160	0.920	272
M 22 × 1.5	1.5	22.000	21.026	20.376	20.160	0.920	333
M 24 × 2	2	24.000	22.701	21.835	24.546	1.227	384
M 27 × 2	2	27.000	25.701	24.835	24.546	1.227	496
M 30 × 2	2	30.000	28.701	27.835	27.546	1.227	621
M 33 × 2	2	33.000	31.701	30.335	30.546	1.227	761
M 36 × 3	3	36.000	34.051	32.752	32.391	1.840	865
M 39 × 3	3	39.000	37.051	35.752	35.391	1.840	1028

Wide variety of threaded fasteners are used in engineering practice. These are cylindrical bars, which are threaded to screw into nuts or internally threaded holes. Figure 5.18 depicts three commonly used fasteners. A bolt has a head at one end of cylindrical body. The head is hexagonal in shape. The other end of the bolt is threaded. The bolt passes through slightly larger holes in two parts and is rotated into hexagonal nut, which may sit on a circular washer. The bolt is rotated into the nut by wrench on bolt head.

As shown in Figure 5.18(a) the two parts are clamped between bolt head and nut.



(a) (b) (c)

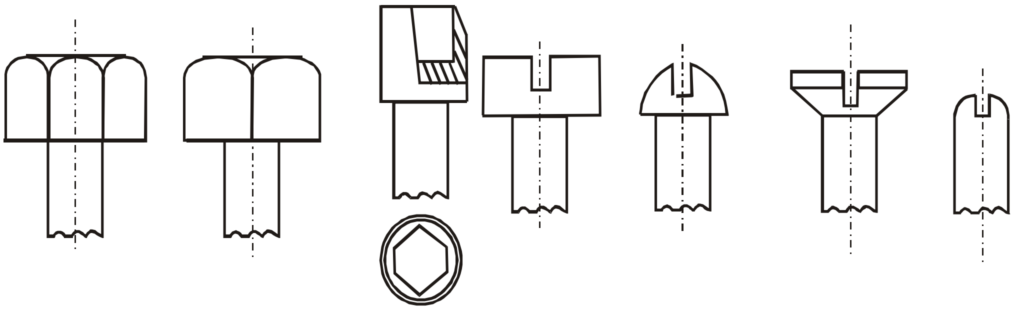
Figure 5.18 : Three Types of Threaded Fasteners

A screw is another threaded fasteners with a head and threads on part of its cylindrical body. However, the threads of the screw are threaded into an internally threaded hole as shown in Figure 5.18(b). While tightening of joint between two parts by bolt occurs by rotating either bolt or nut, the screw tightens the parts, through rotation of screw by a wrench applied at its head. In case of screw the friction occurs between bolt head bottom and surface of the part in contact, and between threads of screw and hole. In case of bolt the friction occurs either at bolt head or at the nut. The wrench has to apply torque against friction between the surface of part and bolt head or nut and in the threads of contact. Both the bolt and screw are pulled and hence carry tensile force.

A stud is another threaded fastener which is threaded at both ends and does not have a head. One of its end screws into threaded hole while the other threaded end receives nut. It is shown in Figure 5.18(c).

The bolts are available as ready to use elements in the market. Depending upon manufacturing method they are identified as black, semi finished or finished. The head in black bolt is made by hot heading. The bearing surfaces of head or shank are machine finished and threads are either cut or rolled. In semi finished bolts the head is made by cold or hot heading. The bearing surfaces of head or shank are machine finished and threads are either cut or rolled. A finished bolt is obtained by machining a bar of same section as the head. The threads are cut on a turret lathe or automatic thread cutting machine.

Besides hexagonal head the bolt or screw may have shapes as shown in Figure 5.19, except the hexagonal and square head which are common in bolts, other forms are used in machine screws. Those at (a) and (b) are tightened with wrench, the bolt or screw with internal socket is rotated with a hexagonal key, at (c) and the screws carrying slits in the head are rotated with screw driver.



(a) (b) (c) (d) (e) (f) (g)

Figure 5.19 : Heads of Threaded Fasteners; (a) Hexagonal; (b) Square; (c) Internal Socket; (d) Circular with a Slit; (e) Button with Slit; (f) Counter Sunk with a Silt; (g) Plain with a Slit

5.9 FAILURE OF BOLTS AND SCREWS

The bolts and screws may fail because of following reasons :

(a) Breaking of bolt shank

(b) Stripping of threads

(c) Crushing of threads

(d) Bending of threads

Invariably when bolt is tightened it is subjected to tensile load along its axis. There may be rare occasion, such as one shown in Figure 5.20 where bolt is pretentioned. The bolt loading situations may be identified as :

(a) No initial tension, bolt loaded during operation.

(b) Only initial tension and no loading afterwards.

(c) After initial tension bolt is further loaded in tension during operation.

(d) In addition to loading initially bolt may be subjected to bending moment and/or shearing forces.

(e) In eccentrically loaded bolted joint, the bolts are subjected to shearing stress which is dominant. Initial tension is additional. We will analyze this problem as riveted joint.

The reader must see that the core section in V-thread means the same thing as in case of square thread. It is the core section, which carries the stress and is identified by core diameter d_1. This diameter can be seen in Figure 5.3(a) and in Figure 5.17. Tables 5.3 and 5.4 also describe the area of the core section under the heading of stress area. The design equation for the bolt or screw is same as Eq. (5.14) with the difference that the fastener will always be in tension. So, if the permissible tensile stress is σ_t, then

$$\sigma_t = \frac{4 \times 1.3W}{\pi d_1^2} = \frac{5.2W}{\pi d_1^2} \qquad \ldots (5.20)$$

The wrench torque can be calculated with friction between threads and between bolt head and the washer. The former is specified as effective friction coefficient μ, given by Eq. (5.9) and latter is specified as μ_c with friction radius r_f as described after Eq. (5.13). Without going into analysis we give relationship between wrench torque, *M* and *W*.

$$M = \frac{W}{2\pi} (p + \mu' \pi\, d_p + \mu_c\, \pi\, d_c) \qquad \text{. . . (5.21)}$$

Where d_p is pitch diameter as shown in Figure 5.15 and $d_c = \frac{d_o + d_i}{2}$, the mean diameter of washer or the bolt head contact surface.

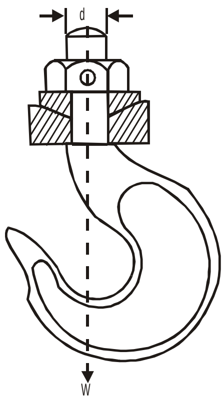
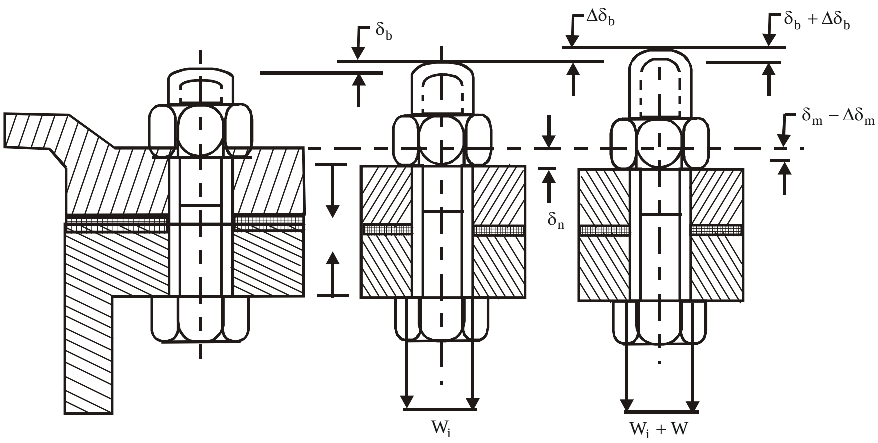


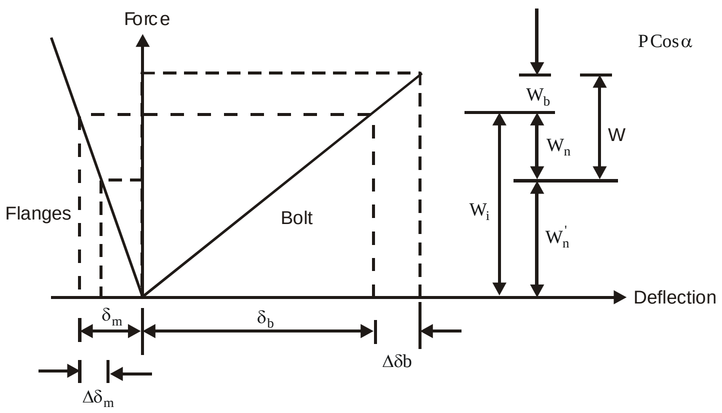
Figure 5.20 : A Bolt Carrying a Crane Hook is a Bolt without Initial Tension

5.9.1 Prestrained Bolt Subjected to External Axial Load

Depending upon type of application the tightened bolt will cause pre-compression of the members jointed. Simultaneously the bolt would have increased in length. Further application of tensile force on bolt will release compression from the parts jointed. This is explained clearly in Figures 5.21(a), (b) and (c). When untightened the length of the bolt, *l*, between the bolt head and nut is equal to the thickness of the flanges between the bolt head and the nut. When the bolt is tightened the length of the bolt changes to $l + \delta_b$ and the thickness of flanges reduces to $l + \delta_m$ where corresponding to initial load *W*, δ_b is the extension in bolt and δ_m is the compression in flanges. Figure 5.19(d) shows the load-deflection relationship for bolt and flanges. When additional tensile force W_b acts upon the bolt because the joint is subjected to external load the bolt further extends by $\Delta\, \delta_b$ making total deflection of bolt as $\delta_b + \Delta\, \delta_b$. At the same time the compression of flanges will be released, and its deformation will reduce by $\Delta\, \delta_b$ to $\delta_m - \Delta\, \delta_m$. At this stage it is essential that $\delta_m - \Delta\, \delta_m$ must be net compression of flanges, a situation, which is shown in Figure 5.21(c). If $\delta_m - \Delta\, \delta_m$ becomes positive the joint will not remain tight. When the load on the bolt has increased to $W_i + W_b$, the load on the flanges has reduced to $W_i + W_m$.



(a) **(b)** **(c)**



(d)

Figure 5.21

(a) bolt not tightened

(b) bolt tightened to initial load W_i

(c) external load W applied upon the joint

(d) deflection in bolt and joint

Following notations are used in the analysis :

W_i = Initial load on the bolt,

W = External load applied upon the joint,

W_b = Part of the load W taken by bolt,

W_m = Part of the load W taken by jointed members,

W_o = Resultant load on bolt,

W'_o = Resultant load on jointed members,

k_b = Stiffness of the bolt (N/mm), and

k_m = Stiffness of jointed members (N/mm).

From above description and Figure 5.21(d)

$$W_0 = W_i + W_b \qquad \text{. . . (i)}$$

but $$W_b = k_b \, \Delta \, \delta_b$$

and $$W_m = k_m \, \Delta \, \delta_m$$

$$\therefore \quad W = W_b + W_m = k_b \, \Delta \, \delta_b + k_m \, \Delta \, \delta_m$$

Since $$\Delta \, \delta_b = \Delta \, \delta_m$$

$$W = \Delta \, \delta_b \, (k_b + k_m)$$

but $$\Delta \, \delta_b = \frac{W_b}{k_b}$$

$$\therefore \quad W = W_b \left(\frac{k_b + k_m}{k_b} \right)$$

or $$W_b = W \, \frac{k_b}{k_b + k_m} \qquad \text{. . . (ii)}$$

Using Eqs. (ii) and (i)

$$W_0 = W_i + W \, \frac{k_b}{k_b + k_m} \qquad \text{. . . (5.22)}$$

Many times where calculation of k_b and k_m is not possible as first estimate following can be used

$$W_0 = 2W \qquad \ldots (5.23)$$

The calculation of the minor or core diameter of the thread then can be based upon the equation

$$\sigma_t = \frac{4W_0}{\pi d_1^2} \qquad \ldots \text{(i)}$$

Apparently the load on the jointed members

$$W_0 = W_i - W \frac{k_m}{k_b + k_m} \qquad \ldots \text{(ii)}$$

The joint will not remain leak proof when $W_0 = 0$ or

$$W_i = W \frac{k_m}{k_b + k_m} \qquad \ldots \text{(iii)}$$

Using Eq. (iii) in Eq. (5.22) the condition for break down of leak proof joint is

$$W_0 = W$$

Substituting this relation in Eq. (5.22), the condition of initial force in the bolt for a leak proof joint can be obtained as

$$W_i = W \frac{k_m}{k_m + k_b} \qquad \ldots (5.24)$$

5.9.2 Bolted Joint Subjected to Transverse Load

Figure 5.22 shows bolts used to connect the bearing to the foundation. The load *P* coming upon the shaft is transmitted to the bearing as horizontal force and will have a tendency to displace the bearing over the foundation. However, such a tendency is opposed by the friction force *F*. The relative displacement between the bearing and foundation will be eliminated if

$$F > P$$

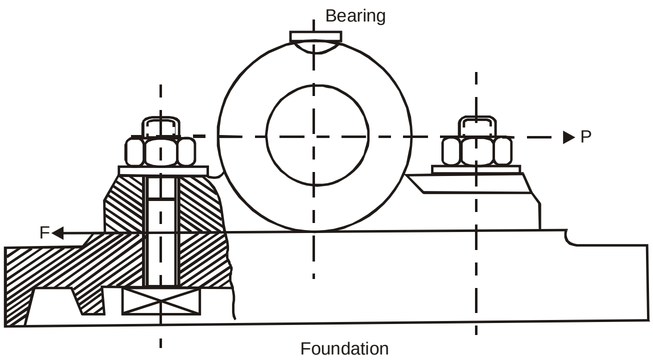


Figure 5.22 : Transverse Loading on Bolts

If *W* is the force which exists in the bolt when it is tightened to make the joint and μ is the coefficient of friction between two surfaces then,

$$F = \mu W$$

$$\therefore \qquad W > \frac{P}{\mu} \approx 10P \approx \frac{1.2}{\mu} P$$

If *n* number of bolts are used in the foundation to make joint with the bearing, then

$$n W_1 = \frac{1.2}{\mu} P$$

where W_1 is the initial tension in one bolt. Each bolt will have to be designed for a force W_1 so that

$$\sigma_t = \frac{4W_1}{\pi d_1^2} = \frac{4 \times 1.2 P}{n \pi d_1^2 \mu}$$

i.e.

$$\sigma_t = \frac{4.8 P}{\mu n \pi d_1^2} \quad \ldots (5.25)$$

5.9.3 Eccentric Load on Bolt

Depending upon the design requirement some times bolt with a jib head has to be employed. One such example is shown in Figure 5.23 where the load W on the jib head acts at A whose line of action is at a distance a from the bolt centre line. Two forces, each equal to W, acting opposite to each other may be assumed to be acting along the axis of the bolt. This results into a direct tension W and a clockwise moment on the bolt. The bolt is thus subjected to direct stress σ_t (tension) and bending stress σ_b (tension or compression). It was explained in the beginning of Section 5.11 that the effective tensile stress in the bolt will be 30% in excess of that calculated for tensile load W. This magnitude is given by Eq. (5.21). The resultant tensile stress on bolt due to tensile load W and bending moment W_a is calculated as

$$\sigma_\gamma = \sigma_t + \sigma_b$$

$$= \frac{5.2 W_1}{\pi d_1^2} + \frac{32 W_a}{\pi d_1^3}$$

$$\sigma_\gamma = \frac{4W_1}{\pi d_1^2}\left[1.3 + \frac{8a}{d_1}\right] \quad \ldots (5.26)$$

If $a = d_1$ then the resultant tensile stress in the bolt is 9 times nominal stress of $\frac{4w}{\pi d_1^2}$.

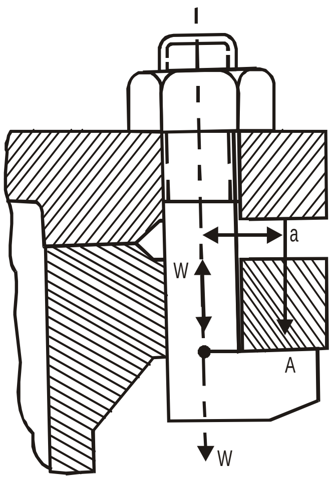


Figure 5.23

5.10 PERMISSIBLE STRESSES IN BOLTS

Bolts are often made in steel having carbon percentage varying between 0.08 to 0.25. However, high quality bolts and particularly those of smaller diameter are made in alloy steel and given treatment of quenching followed by tempering. Medium carbon steels may also be improved in tensile strength by similar heat treatment. Since it is not always possible to determine the wrench torque when bolts are fitted on shop floor, the initial tightening torque often tends to be higher than necessary. This will obviously induce higher stress in the bolt even without external load. Such stresses are particularly high in case of smaller diameter bolt and reduce as the bolt diameter increases. This kind of tightening stresses call for varying permissible stresses in case of bolt which are small when bolt diameter is small and high when bolt diameter is large. This is unlike other machine parts where trend of permissible stress is just the reverse. The correlation of permissible stress and bolt diameter requires that the process of selection of diameter will be reiterative. An empirical formula that correlates the permissible tensile stress, σ_t, and bolt diameter at the stress section d_1, is given below

$$\sigma_t = 5.375\,(d_1)^{0.84} \quad \ldots (5.27)$$

Eq. (5.27) is plotted in Figure 5.24 which can be used as an alternative to Eq. (5.27). Both the Eq. (5.27) and Figure 5.24 are applicable to medium carbon steel bolts only.

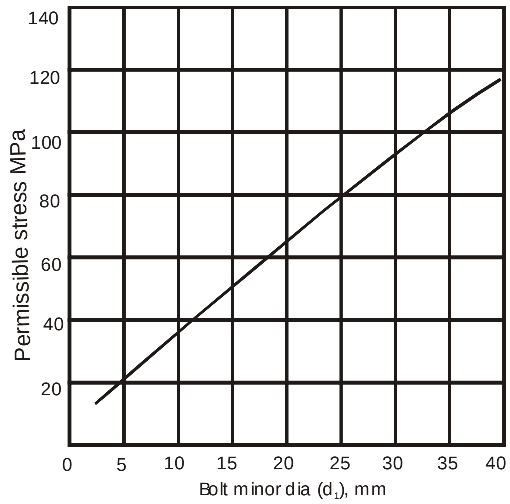


Figure 5.24 : Permissible Bolt Stress as Function of Minor (Core) Diameter for Medium Carbon Steel Bolts

Example 5.5

A 100 KN cover of gear reducer is to be lifted by two eye bolts as shown in Figure 5.25. Each bolt is equidistant from the centre of gravity and lies in the central plane. The bolts are made in steel for which permissible tensile stress is 85 MPa. Find the nominal bolt diameter.

Solution

Figure 5.25(a) shows two eye bolts in the cover of a gear reducer. An eye bolt is shown in Figure 5.25(b).

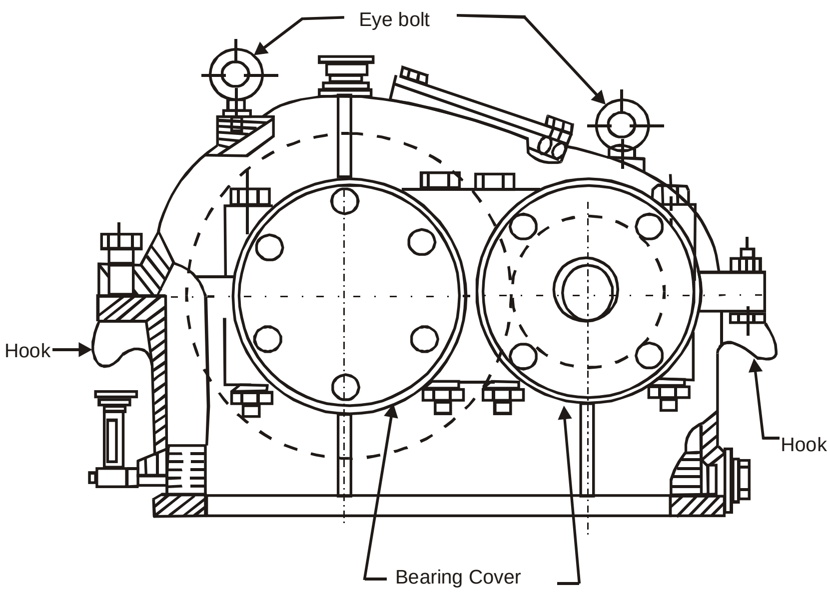


Figure 5.25(a) : A Gear Reducer with Two Eye Bolts in the Cover (the Upper Half of the Body)

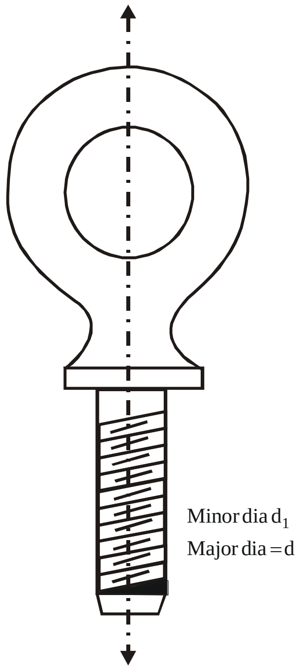


Figure 5.25(b) : The Geometry of the Eye Bolt

The eye bolt is another example of a bolt in which there is no initial tension. Hence, Eq. (5.20) is used. The weight of the cover will be equally divided between two bolts since they are in central plane and equidistant from the centre of gravity. Thus W = 50 kN. Use this value and σ_t = 85 MPa in Eq. (5.20) to obtain core diameter, d_1

$$d_1 = \left(\frac{4W}{\pi\sigma_t}\right)^{\frac{1}{2}} = \left(\frac{4 \times 50,000}{\pi \times 85}\right)^{\frac{1}{2}} = (749)^{\frac{1}{2}}$$

or $\quad d_1 = 27.37$ mm

From Table 5.4 the nearest higher diameter of thread core d_c = 27.546 mm. The bolt in fine series is designated as M 30 × 2 for which d = 30 mm, p = 2 mm, d_1 or d_c = 27.546, stress area = 621 mm^2.

$$\sigma = \frac{4W}{\pi d_1^2} = \frac{W}{\text{Stress area}} = \frac{50,000}{621} = 80.5 \text{ MPa}$$

Example 5.6

A pressure vessel used for storing gas at a pressure of 1.2 MPa is closed by a cover tightened by a number of bolts. The diameter of the bolt circle is 480 mm over a tank diameter of 400 mm. Calculate the diameter of each bolt and number of bolts. Use relationship for permissible stress, $\sigma_t = 5.375\ (d_1)^{0.84}$.

Solution

This problem represents the example of pre-stressed bolt which will further be subjected to tension when vessel is pressurised. Eq. (5.23) describes the force acting axially on the bolt as

$$W_0 = 2W$$

where W is the force acting upon the pre-stressed bolt. This force is because of the gas pressure. If the diameter of the pressure vessel is D_o then total force that pushes the cover out is

$$P = p\,\frac{\pi}{4}\,D_0^2 = 1.2 \times \frac{\pi}{4} \times 400^2 = 15.1 \times 10^4 \text{ N}$$

The joint between the cover plate and the pressure vessel is shown in Figure 5.26. Let there be *n* bolts. So that the force acting on one bolt due to pressure

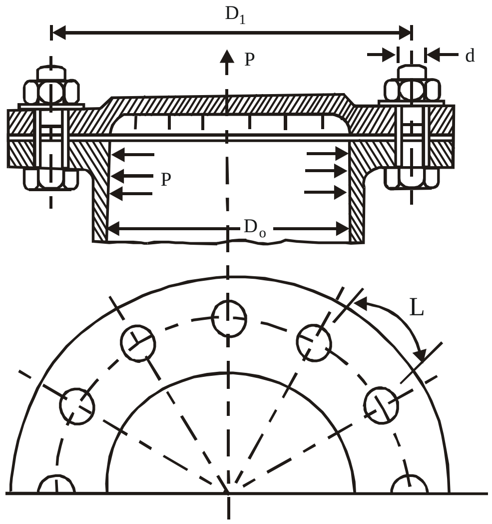


Figure 5.26 : Pressure Vessel Cover and Fastening Bolts

$$W = \frac{P}{n} = \frac{15.1 \times 10^4}{n}$$

We make a small assumption which is a common practice that distance between the centres of two adjacent bolts should be $4d_1$.

$$n \times 4d_1 = \pi D_1 = \text{Circumference of the bolt circle}$$

$$W = 15.1 \times 10^4 \; \frac{4d_1}{\pi D_1}$$

and since D_1 = 480 mm

$$W = \frac{60.4\, d_1}{\pi \times 480} \times 10^4 = 400.5\, d_1 \text{ N}$$

$\therefore$ $$W_0 = 2W = 801\ d_1 \text{ N}$$

$\therefore$ $$\sigma_t = \frac{4W_0}{\pi d_1^2} = \frac{3204\, d_1}{\pi d_1^2} = \frac{1020}{d_1} \text{ N}$$

But the permissible stress in the bolt is 5.375 $(d_1)^{0.84}$

$\therefore$ $$5.375\,(d_1)^{0.84} = \frac{1020}{d_1}$$

or $$d_1 = \left(\frac{1020}{5.375}\right)^{\frac{1}{1.84}} = (189.77)^{0.5435} = 17.31 \text{ mm} \qquad \ldots \text{(i)}$$

From Table 4.4, the nearest core diameter d_c or d_1 is 18.16 mm for M 20 × 1.5 bolt with stress area of 272 mm^2.

Now permissible stress = 5.375 $(18.16)^{0.84}$ = 61.4 MPa. . . . (ii)

$$\text{Working stress, } \sigma_t = \frac{W_0}{\text{Stress area}} = \frac{801 \times 18.16}{272}$$

or $$\sigma_t = 53.5 \text{ MPa} \qquad \ldots \text{(iii)}$$

$$n = \frac{\pi D_1}{4d_1} = \frac{\pi \times 480}{4 \times 18.16} = 20.76 \text{ say } 21 \qquad \ldots \text{(iv)}$$

Thus 21 bolts – M 20 × 1.5 distributed uniformly over circle of 480 mm dia.
M 20 × 1.5 bolt dimensions are, d = 20 mm, d_1 = 18.16 mm, P = 1.5 mm.

SAQ 3

(a) Sketch standard V-thread on screw and mention different dimensions.

(b) Mention three types of threaded fasteners, and different head shapes that are used on threaded fasteners.

(c) In how many ways a bolt is loaded in practice? Explain by the help of diagram how do the strains in the bolt and joint flanges vary in a pre-strained bolted joint.

(d) An M20 jib headed bolt has a jib, which extends to 19.75 mm from the bolt axis but the centre of pressure where jib reaction acts is at a distance of 15mm from the bolt axis. To what axial load the bolt must be tightened so that stress does not exceed $5.375\ (d_1)^{0.84}$.

(e) The cover of a cast iron pressure vessel shown in Figure 5.28 is held in place by 10 M 12 steel bolts having initial tightening load of 22 kN when the vessel is at room temperature and atmospheric pressure. If the pressure is increased to 1.4 MPa, calculate force in each bolt. Modulus of elasticity, steel = 207 GPa, C.I = 83 GPa, Zinc = 28 GPa. Zinc gasket is placed between C.I flanges.

5.11 SUMMARY

A screw and a nut form a pair in which one moves relative to other and combination can transmit force from one point to other. This is the cause of transmission of power. The screw translates by rotating through stationary nut, the nut translates on the length of a screw rotating between two fixed supports or the screw can translate without rotation if the nut rotates in its position supported by bearings on to sides. In each case a force acts axially on translating element and thus the power is transmitted.

The relationship between force required to move nut on the threads of the screw and the axial force is derived from similar situation of a mass moving up an inclined plane. The screw-nut combination may be used for lifting a load or moving a body like tool through a job or applying force on a stationary object. For force transmission the threads in screw and nut are square or trapezoidal in section in which friction is relatively less than in case of V-threads in which case the two sides of thread are inclined at a higher angle than even the trapezium. The V-threads provide jamming force where by they can be used for tightening two flanges. The V-threads screw is used as fastener.

For designing the screw the compressive stress caused by axial compression is used for determining the core diameter, which is the diameter of cylinder on which thread is present as projection. This cylinder is also subjected to a torque due to friction between threads of screw and nut. The direct compressive and torsional shearing stress combine to give maximum shearing stress in the core cylinder. The diameter which is obtained from consideration of compressive stress is checked for maximum shearing stress. If maximum shearing stress is less than permissible shearing stress then the threaded screw and nut are safe. The pitch of thread and the outside diameter which is core diameter plus the pitch are decided from core diameter or standard may be consulted to find major diameters and pitch. The number of threads on the nut which is essential number of threads to make contact between screw and nut is determined from pressure between threads in contact. Apparently the axial force on the screw is sum of the bearing force on all the threads in contact.

The fastening screws are required to move in a nut or threaded hole pressing two parts between the under side of the head on one side and a nut or the threads on the hole on the other side. The bolts have nuts to rotate on threaded part whereas screws rotate in the threads made in the part of joining flanges. The fastening bolt may be prestrained in the joint or left without any pretension where bolts do not make a joint like carrying a crane hook. In case of prestrained bolts the tension in bolt will increase to as much as twice the initial tension before the contact surfaces of the joint may begin to separate. Determining the torque required to tighten the screw or bolt in the joint may be achieved by establishing the initial tension in the bolt screw.

The bolts and screws are available in the market as ready to use element. They are made in steel with varying carbon content heads made with cold or hot heading and threads cut on machines or rolled. The head is hexagonal square or with a socket in the round head. A finished bolt is machined from hexagonal, square or circular bar leaving a head of same shape as original section.

5.12 KEY WORDS

Screw : A screw is a cylinder on whose surface helical projection is created in form of thread.

Square Thread : A square thread is formed if the generating plane section is square.

Acme Thread : The vee thread is created by a triangular section while trapezoidal thread has a trapezium section. This thread is also known as the Acme thread.

Pitch : The distance measured axially, between corresponding points on the consecutive thread forms in the same axial plane and on the same side of axis is known as pitch length.

Lead : It is axial distance a screw thread advances in one revolution.

Overhauling of Screw : If the coefficient of friction between the nut and screw is small or the lead is large, the axial load may be sufficient to turn the nut, and the load will start moving downward without the application of any torque. Such a condition is known as overhauling of screws.

5.13 ANSWER TO SAQs

SAQ 1

(d) Screw-square threads, d_m = 50 mm, p =10 mm, μ = 0.1.

$$W = (W_1 + 100) \text{ N}$$

Torque about the axis is

$$M_t = 280\ (1000 + 50) = 2.8 \times 1.05 \times 10^5 \text{ Nmm}$$

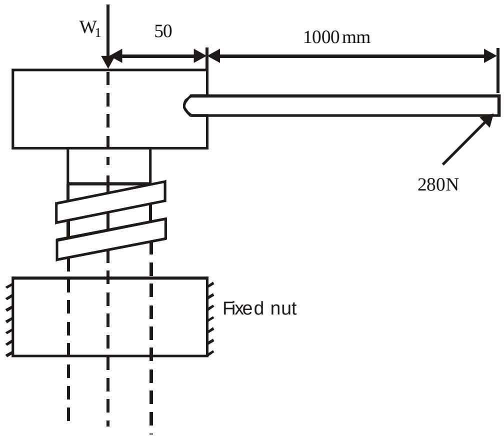


Figure 5.27

This torque is equal to friction torque in the threads which is given by

$$P \times \frac{d_m}{2}$$

where $P = W \tan (\alpha + \phi)$

$$\alpha = \tan^{-1} \frac{p}{\pi d_m} = \tan^{-1} \frac{10}{\pi \times 50} = 3.64^o$$

$$\phi = \tan^{-1} \mu = \tan^{-1} 0.1 = 5.71^o$$

$$\tan (\alpha + \phi) = \tan (3.64 + 5.71) = \tan 9.35 = 0.165$$

$\therefore$ $$M_t = P \frac{d_m}{2} = W \tan (\alpha + \phi) \frac{d_m}{2}$$

$$= W \times 0.165 \times \frac{50}{2} = 4.125\, W \text{ Nmm}$$

$\therefore$ $$4.125\, W = 2.8 \times 1.05 \times 10^5$$

or $$W = \frac{2.8 \times 1.05}{4.125} \times 10^5 = 71273 \text{ N}$$

$\therefore$ $$W_1 = 71273 - 100 = 71173 \text{ N}$$

SAQ 2

(d) Screw with d_1 = 17 mm, p = 5 mm, d = 22 mm, μ = 0.12, W = 4000 N

$$M_t = W \tan (\alpha + \phi) \frac{d_m}{2}$$

$$d_m = \frac{d_1 + d}{2} = \frac{17 + 22}{2} = \frac{39}{2} = 19.5 \text{ mm}$$

$$\alpha = \tan^{-1} \frac{p}{\pi d_m} = \tan^{-1} \frac{5}{\pi \times 19.5} = \tan^{-1} 0.0816 = 4.666^o$$

$$\phi = \tan^{-1} 0.12 = 6.843^o$$

$\therefore$ $$M_t = 4000 \tan (4.666 + 6.843) \times \frac{19.5}{2} = 4000 \times 0.2036 \times 9.75$$

$$= 7940.4 \text{ Nmm}$$

$$\therefore \qquad \tau = \frac{16 M_t}{\pi d_1^3} = \frac{16 \times 7940.4}{\pi \times (17)^3} = 8.23 \text{ N/mm}^2$$

$$\sigma = \frac{4W}{\pi d_1^2} = \frac{4 \times 4000}{\pi \times (17)^2} = 17.62 \text{ N/mm}^2$$

$$\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} = \sqrt{\left(\frac{17.62}{2}\right)^2 + (8.23)^2} = 12.05 \text{ N/mm}^2$$

$$\text{Bearing area} = n \pi d_m t = \frac{n}{2} \pi d_m p = \frac{5}{2} \times \pi \times 19.5 \times 5 = 765.76$$

$$\therefore \qquad \text{Bearing pressure} = \frac{4000}{765.76} = 5.22 \text{ N/mm}^2$$

So Factors of safety are calculated for direct stress, shearing stress and bearing pressure.

$$FS \text{ in direct stress} = \frac{\sigma_u}{\sigma} = \frac{320}{17.62} = 18.16$$

$$FS \text{ in shear} = \frac{\tau_u}{\tau_{max}} = \frac{212}{12.05} = 17.6$$

$$FS \text{ in bearing pressure} = \frac{12}{5.22} = 2.3$$

(e) $\mu = 0.15$, $2\theta = 30$, $p = 10$ mm, $d = 55$ mm

$d_m = d - p = 55 - 10 = 45$ mm

$$\mu = 0.15, \ \mu' = \frac{0.15}{\cos \theta} = \frac{0.15}{\cos 15} = 0.1533$$

$$\text{Collar friction surface } d_c = \frac{d_i + d_o}{2} = \frac{60 + 90}{2} = 75 \text{ mm}$$

$$\mu_c = 0.15$$

The torque due to collar friction,

$$M_{tc} = \mu_c W \frac{d_c}{2} = 0.15 W \ 37.5 = 5.625 W \text{ Nmm}$$

$$\alpha = \tan^{-1} \frac{p}{\pi d_m} = \tan^{-1} \frac{5}{\pi \times 45} = \tan^{-1} 0.0354 = 2.03^{o}$$

$$\phi' = \tan^{-1} \mu' = \tan^{-1} 0.1553 = 8.83^{o}$$

$$\tan(\alpha + \phi') = \tan(2.03 + 8.83) = \tan 10.86 = 0.192$$

Torque to create a pull of W in screw,

$$M_t = W \tan(\alpha + \phi) \times \frac{d_m}{2} = W \times 0.192 \times 22.5 = 4.32 W \text{ Nmm}$$

Total torque to be applied on screw

$$M = M_t + M_{tc} = (5.625 + 4.32) W = 9.945 W \text{ Nmm}$$

$$\text{The power, } H = M \omega = M \times \frac{2\pi N}{60} = 9.945 W \frac{2\pi \times 600}{60} \times 10^{-3} \text{ Watt}$$

$$= 624.863\, W \times 10^{-3} \text{ Watt}$$

$$= 0.37 \times 1000$$

$$\therefore \qquad W = \frac{370 \times 10^3}{624.863} = 592 \text{ N}$$

The axial pull exerted by the screw is 592 N.

SAQ 3

Please refer the preceding text.