Design of Machine Elements

POWER SCREWS & THREADED JOINTS

Power Screws & Threaded Joints

Power Screws

A power screw is a mechanical device used for convening rotary motion into linear motion and transmitting power. A power screw is also called a translation screw. It uses helical translatory motion of the screw thread in transmitting power rather than clamping the machine components.

The main applications of power screws are as follows:

- to raise the load. e.g.. screw-jack;

- to obtain accurate motion in machining operations, e.g., lead-screw of lathe;

- to clamp a work piece, e.g., a vice; and

- to load a specimen, e.g., universal testing machine.

There are three essential parts of the power screw, viz., screw, nut and a part to hold either the screw or the nut in its place.

Depending upon the holding arrangement, power screws operate in two different ways.

In some cases, the screw rotates in its bearing, while the nut has axial motion. The lead screw of the lathe is an example of this category.

In other applications, the nut is kept stationary and the screw moves in an axial direction. A screw jack and machine vice are the examples of this category.

ADVANTAGES OF POWER SCREW

- A power screw has large load carrying capacity.

- The overall dimensions of the power screw are small, resulting in compact construction.

- A power screw is simple to design.

- The manufacturing of a power screw is easy without requiring specialised machinery. Square threads are turned on the lathe. Trapezoidal threads are manufactured on a thread milling machine.

- A power screw provides large mechanical advantage. A load of 15 kN can be raised by applying an effort as small as 400 N. Therefore, most of the power screws used in various applications like screw-jacks. clamps, valves and vices are manually operated.

- A power screw provides precisely controlled and highly accurate linear motion required in machine tool applications.

- A power screw gives smooth and noiseless service without any maintenance.

- There are few parts in a power screw. This reduces cost and increases reliability.

- A power screw can be designed with self- locking property. In screw-jack application. self-locking characteristic is required to prevent the load from descending on its own

DISADVANTAGES OF POWER SCREWS

- A power screw has very poor efficiency, as low as 40%. Therefore, it is not used in continuous power transmission in machine tools, with the exception of the lead screw. Power screws are mainly used for intermittent motion that is occasionally required for lifting the load or actuating the mechanism,

- High friction in threads causes rapid wear of the screw or the nut. In case of square threads, the nut is usually made of soft material and replaced when worn out. In trapezoidal threads, a split-type of nut is used to compensate for the wear. Therefore, wear is a serious problem in power screws.

FORMS OF THREAD

The threads used for fastening purposes, such as **V threads** are *not suitable* for power screws. The purpose of fastening threads is to provide high frictional force, which lessens the possibility of loosening the parts assembled by threaded joint. On the other hand, the purpose of power transmission threads is to reduce friction between the screw and mil. Therefore, V threads are not suitable for power screws. Screws with smaller angle of thread, such as **trapezoidal threads**, are *preferred* for power transmission.

There are two popular types of threads used for power screws, viz., **square** and **ISO metric trapezoidal**, as shown in following figure.
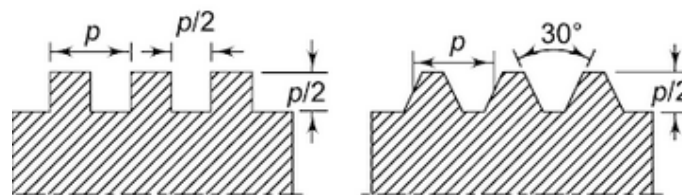
    (a) square threads   **(b) trapezoidal threads**

Advantages of square threads over trapezoidal threads

- The efficiency of square threads is more than that of trapezoidal threads,

- There is no radial pressure or side thrust on the nut. This radial pressure is called 'bursting' pressure on the nut. Since there is no side thrust, the motion of the nut is uniform. The life of the nut is also increased.

Disadvantages of square threads

- Square threads are difficult to manufacture. They are usually turned on a lathe with a single-point cutting tool. Machining with a single-point cutting tool is an expensive operation compared with machining with a multi-point cutting tool.

- The strength of a screw depends upon the thread thickness at the core diameter. As shown in given figure, square threads have less thickness at the core diameter than trapezoidal threads. This reduces the load carrying capacity of the screw,

- The wear of the thread surface becomes a serious problem in the service life of the power screw. It is not possible to compensate for wear in square threads. Therefore, when worn out, the nut or the screw requires replacement.

Advantages of trapezoidal threads over square threads

- Trapezoidal threads are manufactured on a thread milling machine. It employs a multipoint cutting tool. Machining with a multipoint cutting tool is an economic operation compared with machining with a single-point cutting tool. Therefore, trapezoidal threads are economical to manufacture.

- A trapezoidal thread has more thickness at the core diameter than a square thread. Therefore, a screw with trapezoidal threads is stronger than an equivalent screw with square threads. Such a screw has a large load carrying capacity.

Acme Thread

It is a modification of square thread. It is much stronger than square thread and can be easily produced. This is special type of *trapezoidal thread* called *acme thread*.

It is shown below.
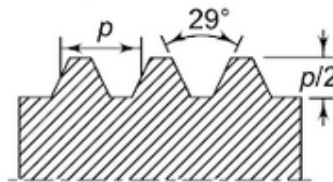
Trapezoidal and acme threads are identical in all respects except the thread angle. In an acme thread. the thread angle is 29° *instead of* 30°.

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 relative advantages and disadvantages of acme threads are same as those of trapezoidal threads.

Buttress Thread

There is another type of thread called buttress thread. It is shown below.
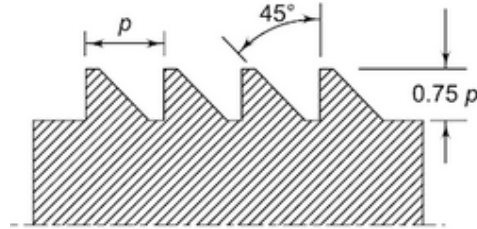
It combines the advantages of square and trapezoidal threads. Buttress threads are used where a heavy axial force acts along the screw axis in one direction only.

The *advantages* of buttress threads are as follows:

- It has higher efficiency compared with trapezoidal threads.

- It can be economically manufactured on a thread milling machine.

- The axial wear at the thread surface can be compensated by means of a split-type nut.

- A screw with buttress threads is stronger than an equivalent screw with either square threads or trapezoidal threads. This is because of greater thickness at the base of the thread.

The buttress threads have one *disadvantage*. It can transmit power and motion only in one direction. On the other hand, square and trapezoidal threads can transmit force and motion in both directions.

Uses of Various Threads

Square threads are used for screw jacks, presses and clamping devices. Trapezoidal and acme threads are used tor lead-screw and other power transmission devices in machine tools. Buttress threads are used in vices, where force is applied only in one direction. Buttress threads are ideally suited for connecting tubular components that must carry large forces such as connecting the barrel to the housing in anti-aircraft guns.

TORQUE REQUIRED TO RAISE LOAD BY SQUARE THREADED SCREWS

The torque required to raise a load by means of square threaded screw may be determined by considering a screw jack as shown in following figure.
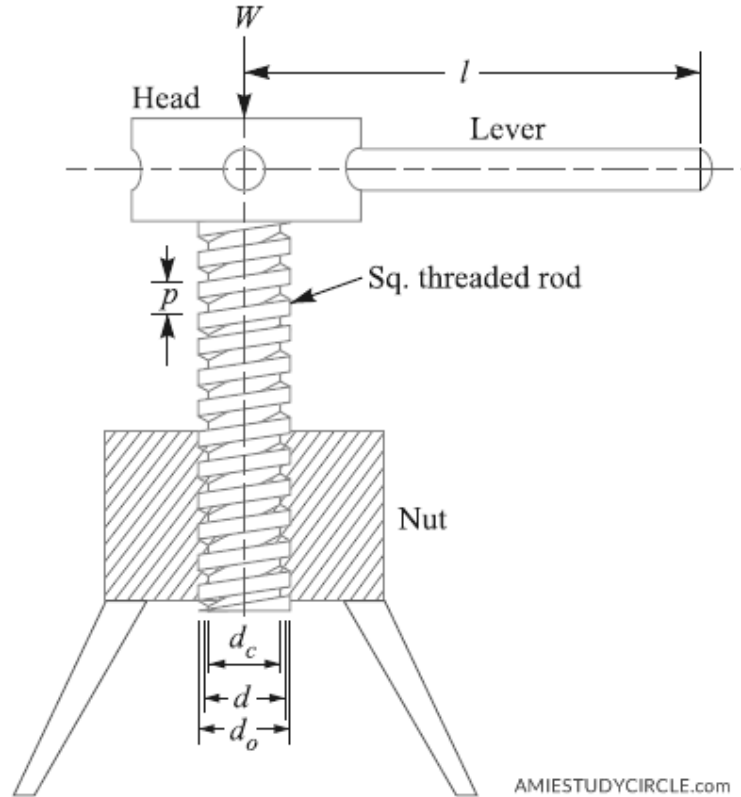
The load to be raised or lowered is placed on the head of the square threaded rod which is rotated by the application of an effort at the end of lever for lifting or lowering the load.

Development of the screw and forces acting are shown in following figure.
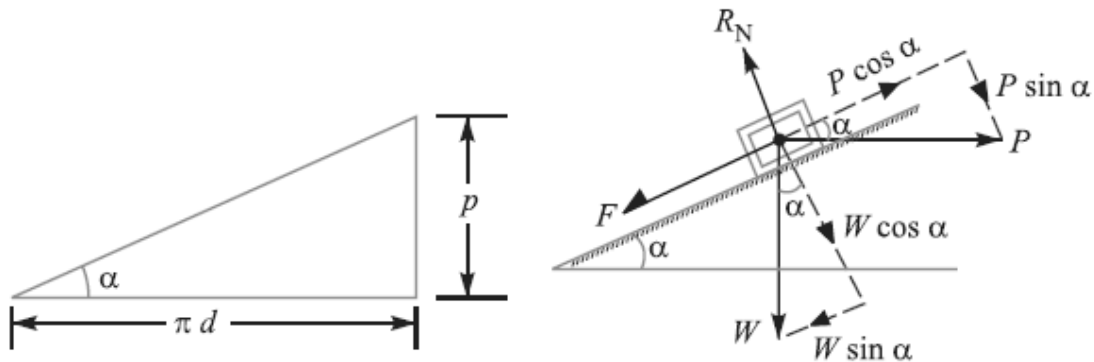
(a) Development of screw (b) Forces acting on screw

p = Pitch of the screw,

d = Mean diameter of the screw = $d_0 - p/2$ where d_0 is external diameter.

α = Helix angle.

P = Effort applied at the circumference of the screw to lift the load,

W = Load to be lifted, and

μ = Coefficient of friction, between the screw and nut

= tan ϕ, where ϕ is the friction angle.

Now $\qquad P = W \tan(\alpha + \phi)$

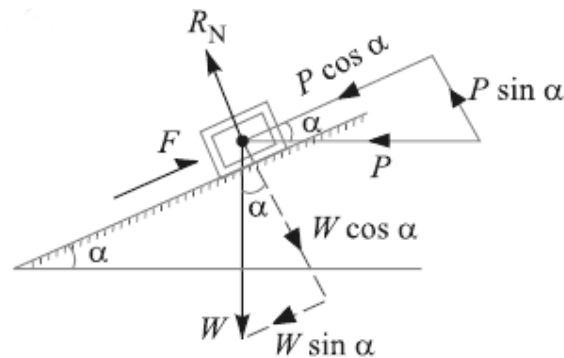From the geometry of the figure, we find that

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

Total torque required to overcome friction (i.e. to rotate the screw),

$T = P \times (d/2)$

TORQUE REQUIRED TO LOWER LOAD BY SQUARE THREADED SCREWS

A little consideration will show that when the load is being lowered, the force of friction will act upwards. All the forces acting on the body are shown in following figure.

$P = \tan(\phi - \alpha)$

Total torque required to overcome friction (i.e. to rotate the screw),

$T = P \times (d/2)$

EFFICIENCY OF SQUARE THREADED SCREWS

The efficiency of square threaded screws may be defined as the ratio between the ideal effort (i.e. the effort required to move the load, neglecting friction) to the actual effort (i.e. the effort required to move the load taking friction into account).

Efficiency $\qquad \eta$ = ideal effort/actual effort

$\qquad\qquad = W \tan\alpha / W \tan(\alpha + \phi)$

$\qquad\qquad = \tan\alpha / \tan(\alpha + \phi)$

The efficiency may also be defined as the ratio of mechanical advantage to the velocity ratio.

We know that mechanical advantage

$$MA = \frac{2l}{d \tan(\alpha + \phi)}$$

Velocity ratio

$$VR = \frac{2l}{d \tan \alpha}$$

MAXIMUM EFFICIENCY OF A SQUARE THREADED SCREW

It will be

$$\eta_{max} = \frac{1 - \sin \alpha}{1 + \sin \alpha}$$

Example

An electric motor driven power screw moves a nut in a horizontal plane against a force of 75 kN at a speed of 300 mm / min. The screw has a single square thread of 6 mm pitch on a major diameter of 40 mm. The coefficient of friction at screw threads is 0.1. Estimate power of the motor.

Solution

Given data

W = 75 kN = 75 x 10^3 N; ν = 300 mm/min; p = 6; d_o = 40 mm; μ = tanϕ = 0.1

Mean diameter of screw
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d = d_o – p/2 = 40 – 6/2 = 37 mm

and $\tan \alpha = \dfrac{p}{\pi d} = \dfrac{6}{\pi x 37} = 0.0516$

Tangential force

We know that tangential force required at the circumference of the screw

$$P = W \tan(\alpha + \phi) = W \left[\frac{\tan \alpha + \tan \phi}{1 - \tan \alpha \tan \phi} \right]$$

$$= 75 x 10^3 \left[\frac{0.0516 + 0.1}{1 - 0.0516 x 0.1} \right] = 11.43 x 10^3 N$$

Torque

$$T = P x \frac{d}{2} = 11.43 x 10^3 x \frac{37}{2} = 211.45 x 10^3 \, N - mm = 211.45 \, N - m$$

Speed of the screw

Since the screw moves in a nut at a speed of 300 mm / min and the pitch of the screw is 6 mm, therefore speed of the screw in revolutions per minute (r.p.m.),

$$N = \frac{speed(mm/\min)}{pitch(mm)} = \frac{300}{6} = 50\, rpm$$

Angular speed

$$\omega = 2\pi N/60 = 2\pi x 50/60 = 1108 W = 1.108\, kW$$

Power of motor

Power of motor = $T\omega = 211.45 x 5.24 = 1108 W = 1.108\, kW$

Example

A vertical two start square threaded screw of a 100 mm mean diameter and 20 mm pitch supports a vertical load of 18 kN. The axial thrust on the screw is taken by a collar bearing of 250 mm outside diameter and 100 mm inside diameter. Find the force required at the end of a lever which is 400 mm long in order to lift and lower the load. The coefficient of friction for the vertical screw and nut is 0.15 and that for collar bearing is 0.20.

Solution

Given data

d = 100 mm; p = 20 mm; W = 18 kN = 18 x 10^3 N; D_1 = 250 mm or R_1 = 125 mm

D_2 = 100 mm or R_2 = 50 mm; l = 400 mm; $\mu = \tan\phi$ = 0.15; μ_1 = 0.20

Force required at the end of lever

Let P = Force required at the end of lever.

Since the screw is a two start square threaded screw, therefore lead of the screw

= 2 p = 2 × 20 = 40 mm

We know that tan α = lead/πd = 40/π x 100 = 0.127

For raising the load

Tangential force

$$P = W \tan(\alpha + \phi) = W\left[\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha \tan\phi}\right]$$

$$= 18 x 10^3 \left[\frac{0.127 + 0.15}{1 - 0.127 x 0.15}\right] = 5083\, N$$

Mean radius of the collar

$$R = \frac{R_1 + R_2}{2} = \frac{125 + 50}{2} = 87.5 \, mm$$

$\therefore$ Torque required at the end of lever

$$T = Px\frac{d}{2} + \mu_1 WR = 5083x\frac{100}{2} + 0.20x18x10^3 x87.5 = 569150 \, N - mm$$

Torque required at the end of lever (T)

$$569150 = P_1 \text{ x l} = P_1 \text{ x } 400$$

$\therefore$ $P_1 = 569150/400 = 1423 \, N$

For lowering the load

$$P = W\left[\frac{\tan\phi - \tan\alpha}{1 + \tan\phi\tan\alpha}\right] = 406.3 \text{ N after putting values}$$

Total torque required at lever

$$T = Px\frac{d}{2} + \mu_1 WR = 335315 \, N - mm \quad \text{after putting values.}$$

Torque at the end (T)

$$335315 = P_1 xl = P_1 x400$$

$\therefore$ $P_1 = 838.3 \, N$

Example

The mean diameter of the square threaded screw having pitch of 10 mm is 50 mm. A load of 20 kN is lifted through a distance of 170 mm. Find the work done in lifting the load and the efficiency of the screw, when the load rotates with the screw. The external and internal diameter of the bearing surface of the loose head are 60 mm and 10 mm respectively. The coefficient of friction for the screw and the bearing surface may be taken as 0.08.

Solution

Given data

$p = 10$ mm; $d = 50$ mm; $W = 20$ kN $= 20 \times 10^3$ N; $D_1 = 60$ mm or $R_1 = 30$ mm

$D_2 = 10$ mm or $R_2 = 5$ mm; $\mu = \tan\phi = \mu_1 = 0.08$.

Force required at the circumference of the screw

$$P = W\tan(\alpha + \phi) = W\left[\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha\tan\phi}\right]$$

= 2890 N (after putting values)

Torque to overcome friction

$$T = P \, x \, d / 2 = 2890 x 50 / 2 = 72.25 \, N - m$$

Since the load is lifted through a vertical distance of 170 mm and the distance moved by the screw in one rotation is 10 mm (equal to pitch), therefore number of rotations made by the screw,

N = 170/10 = 17

Work done in lifting the load

$$T \times 2 \, \pi \, N = 72.25 \times 2\pi \times 17 = 7718 \text{ N-m}$$

Efficiency of screw

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \phi)} = \frac{\tan \alpha (1 - \tan \alpha \tan \phi)}{\tan \alpha + \tan \phi}$$

$$= \frac{0.0637(1 - 0.0637 x 0.08)}{0.0637 + 0.08} = 0.441 \Rightarrow 44.1\%$$

Problem

In a hand vice, the screw has double start square threads of 24 mm outside diameter. If the lever is 200 mm long and the maximum force that can be applied at the end of lever is 250 N, find the force with which the job is held in the jaws of the vice. Assume a coefficient of friction of 0.12.

Answer: 17 420 N

Problem

A square threaded bolt of mean diameter 24 mm and pitch 5 mm is tightened by screwing a nut whose mean diameter of bearing surface is 50 mm. If the coefficient of friction for the nut and bolt is 0.1 and for the nut and bearing surfaces 0.16, find the force required at the end of a spanner 0.5 m long when the load on the bolt is 10 kN.

Answer: 120 N

Problem

The cross bar of a planner weighing 12 kN is raised and lowered by means of two square threaded screws of 38 mm outside diameter and 7 mm pitch. The screw is made of steel and a bronze nut of 38 mm thick. A steel collar has 75 mm outside diameter and 38 mm inside diameter. The coefficient of friction at the threads is assumed as 0.11 and at the collar 0.13. Find the force required at a radius of 100 mm to raise and lower the load.

ACME OR TRAPEZOIDAL THREADS

We know that the normal reaction in case of a square threaded screw is

$$R_N = W \cos \alpha$$

where α is the helix angle.

But in case of Acme or trapezoidal thread, the normal reaction between the screw and nut is increased because the axial component of this normal reaction must be equal to the axial load (W).

Consider an Acme or trapezoidal thread as shown in following figure.

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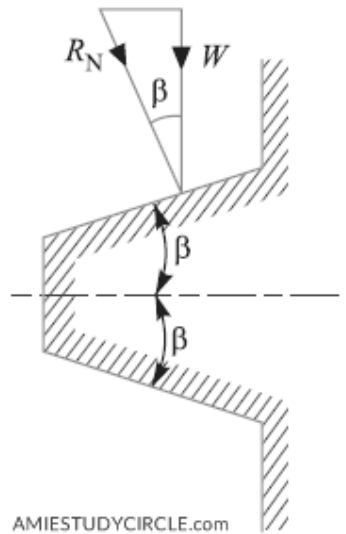
Let 2β = Angle of the Acme thread, and

β = Semi-angle of the thread.

$$\therefore \qquad R_N = \frac{W}{\cos \beta}$$

and friction force

$$F = \mu R_N = \mu x \frac{W}{\cos \beta} = \mu_1 W$$

where $\mu/\cos\beta = \mu_1$ known as virtual coefficient of friction.

When coefficient of friction, $\mu_1 = \mu/\cos\beta$ is considered, then the Acme thread is equivalent to a square thread.

All equations of square threaded screw also hold good for Acme threads. In case of Acme threads, μ_1 (i.e. tan ϕ_1) may be substituted in place of μ (i.e. tan ϕ). Thus for Acme threads,

$$P = W \tan (\alpha + \phi_1)$$

where ϕ_1 = Virtual friction angle, and $\tan\phi_1 = \mu_1$.

Example

The lead screw of a lathe has Acme threads of 50 mm outside diameter and 8 mm pitch. The screw must exert an axial pressure of 2500 N in order to drive the tool carriage. The thrust is carried on a collar 110 mm outside diameter and 55 mm inside diameter and the lead screw rotates at 30 r.p.m. Determine (a) the power required to drive the screw; and (b) the efficiency of the lead screw. Assume a coefficient of friction of 0.15 for the screw and 0.12 for the collar.

Solution

Given data

d_o = 50 mm; p = 8 mm; W = 2500 N; D_1 = 110 mm or R_1 = 55 mm;

D_2 = 55 mm or R_2 = 27.5 mm; N = 30 rpm; $\mu = \tan\phi$ = 0.15; μ_2 = 0.12

Virtual coefficient of friction

Diameter of screw

$$d = d_0 - p/2 = 50 - 8/2 = 46 \text{ mm}$$

$\therefore$ $$\tan\alpha = \frac{p}{\pi d} = \frac{8}{\pi x 46} = 0.055$$

Since the angle for Acme threads is $2\beta = 29^0$ or $\beta = 14.5^0$, therefore virtual coefficient of friction

$$\mu_1 = \tan\phi_1 = \frac{\mu}{\cos\beta} = \frac{0.15}{\cos 14.5} = 0.155$$

Now $$P = W\tan(\alpha + \phi_1) = W\left[\frac{\tan\alpha + \tan\phi_1}{1 - \tan\alpha\tan\phi_1}\right]$$

Putting values, we get

$$P = 530 \text{ N}$$

Torque (T_1)

Torque required to overcome friction

$$T_1 = P \times d/2 = 530 \times 46/2 = 12190 \text{ N-mm}$$

Torque required to overcome friction at collars(T_2)

Mean radius

$$R = (R_1 + R_2)/2 = (55 + 27.5)/2 = 41.25 \text{ mm}$$

Torque to overcome friction at collars

$$T_2 = \mu_2 WR = 0.12 \times 2500 \times 41.25 = 12375 \text{ N-mm}$$

Total Torque

Total torque

$$T = T_1 + T_2 = 12190 + 12375 = 24565 \text{ N-mm} = 24.565 \text{ N-m}$$

Efficiency of lead screw

Torque required to drive the screw with no friction

$$T_0 = W \tan \alpha \, x \frac{d}{2} = 2500 x 0.055 x \frac{46}{2} = 3163 \, N-mm = 3.163 \, n-m$$

∴ Efficiency of screw

$$\eta = \frac{T_o}{T} = \frac{3.163}{24.565} = 0.13 = 13\%$$

Problem

The lead screw of a lathe has Acme threads of 60 mm outside diameter and 8 mm pitch. It supplies drive to a tool carriage which needs an axial force of 2000 N. A collar bearing with inner and outer radius as 30 mm and 60 mm respectively is provided. The coefficient of friction for the screw threads is 0.12 and for the collar it is 0.10. Find the torque required to drive the screw and the efficiency of the screw.

Answer: 18.5 N-m ; 13.6%

STRESSES IN POWER SCREWS

Direct tensile or compressive stress due to an axial load

The direct stress due to the axial load may be determined by dividing the axial load (W) by the minimum cross-sectional area of the screw (Ac) i.e. area corresponding to minor or core diameter (dc).

∴ Direct stress (tensile or compressive)

$$= W/A_c$$

This is only applicable when the axial load is compressive and the unsupported length of the

screw between the load and the nut is short. But when the screw is axially loaded in compression and the unsupported length of the screw between the load and the nut is too great, then the design must be based on column theory assuming suitable end conditions. In such cases, the cross-sectional area corresponding to core diameter may be obtained by using Rankine-Gordon formula or J.B. Johnson's formula. According to this,

$$W_{cr} = A_c \sigma_y \left[1 - \frac{\sigma_y}{4C\pi^2 E} \left(\frac{L}{k} \right)^2 \right]$$

$$\therefore \qquad \sigma_c = \frac{W}{A_c} \left[\frac{1}{1 - \frac{\sigma_y}{4C\pi^2 E} \left(\frac{L}{k} \right)^2} \right]$$

where

W_{cr} = Critical load,

σ_y = Yield stress,

L = Length of screw,

k = Least radius of gyration,

C = End-fixity coefficient,

E = Modulus of elasticity, and

σ_c = Stress induced due to load W.

In actual practice, the core diameter is first obtained by considering the screw under simple compression and then checked for critical load or buckling load for stability of the screw.

Torsional shear stress

Since the screw is subjected to a twisting moment, therefore torsional shear stress is induced. This is obtained by considering the minimum cross-section of the screw. We know that torque transmitted by the screw,

$$T = \frac{\pi}{16} \tau (d_c)^3$$

or shear stress induced

$$\tau = \frac{16T}{\pi (d_c)^3}$$

When the screw is subjected to both direct stress and torsional shear stress, then the design must be based on maximum shear stress theory, according to which maximum shear stress on the minor diameter section,

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

It may be noted that when the unsupported length of the screw is short, then failure will take place when the maximum shear stress is equal to the shear yield strength of the material.

In this case, shear yield strength,

$$\tau_y = \tau_{max} \times \text{Factor of safety}$$

Shear stress due to axial load

The threads of the screw at the core or root diameter and the threads of the nut at the major diameter may shear due to the axial load.

Assuming that the load is uniformly distributed over the threads in contact, we have Shear stress for screw,

$$\tau_{screw} = \frac{W}{\pi n d_c t}$$

and shear stress for nut

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

where   W = Axial load on the screw,

     n = Number of threads in engagement,

     d_c = Core or root diameter of the screw,

     d_o = Outside or major diameter of nut or screw, and

     t = Thickness or width of thread.

Bearing pressure

In order to reduce wear of the screw and nut, the bearing pressure on the thread surfaces must be within limits. In the design of power screws, the bearing pressure depends upon the materials of the screw and nut, relative velocity between the nut and screw and the nature of lubrication. Assuming that the load is uniformly distributed over the threads in contact, the bearing pressure on the threads is given by

$$p_b = \frac{W}{\frac{\pi}{4}\left[(d_o{}^2 - d_c{}^2)\right]n} = \frac{W}{\pi d t n}$$

where   d = Mean diameter of screw,

     t = Thickness or width of screw = p / 2, and

     n = Number of threads in contact with the nut

         = height of nut/pitch of threads = h/p

Example (AMIE Winter 2012, 10 marks)

A power screw having double start square threads of 25 mm nominal diameter and 5 mm pitch is acted upon by an axial load of 10 kN. The outer and inner diameters of screw collar are 50 mm and 20 mm respectively. The coefficient of thread friction and collar friction may be assumed as 0.2 and 0.15 respectively. The screw rotates at 12 r.p.m. Assuming uniform wear condition at the collar and allowable thread bearing pressure of 5.8 N/mm², find: (i) the

torque required to rotate the screw; (ii) the stress in the screw; and (iii) the number of threads of nut in engagement with screw.

Solution

Given data

$d_o = 25$ mm; p = 5 mm; W = 10 kN = 10 x 10^3 N; $D_1 = 50$ mm; $D_2 = 20$ mm

$\mu = \tan\phi = 0.2$; $\mu_1 = 0.15$; N = 12 rpm; $p_b = 5.8$ N/mm^2

Tangential force

Mean diameter of screw

$d = d_o – p/2 = 25 – 5/2 = 22.5$ mm

For double start screw

lead of screw = 2p = 2 x 5 = 10 mm

$\therefore$ $\tan\alpha = \text{lead}/\pi d = 10/\pi \times 22.5 = 0.1414$

Tangential force

$$P = W \tan(\alpha + \phi) = W \left[\frac{\tan \alpha + \tan \phi}{1 - \tan \alpha \tan \phi} \right] = 3513\,N \text{ after putting values.}$$

Torque to rotate screw

Mean radius of collar

$$R = \frac{R_1 + R_2}{2} = \frac{25 + 10}{2} = 17.5$$

$\therefore$ Torque $T = Px\dfrac{d}{2} + \mu_1 WR = 3513x\dfrac{22.5}{2} + 0.15x10x10^3 x17.5\,N - mm$

$= 65771$ N-mm = 65.771 N-m

Stress in the screw

Inner (core) dia of screw

$d_c = d_o – p = 25 – 5 = 20$ mm

Cross sectional area of screw

$$A_c = \frac{\pi}{4}(d_c)^2 = \frac{\pi}{4}(20)^2 = 314.2\,mm^2$$

Direct stress

$$\sigma_c = \frac{W}{A_c} = \frac{10x10^3}{314.2} = 31.83\,N/mm^2$$

Shear stress

$$\tau = \frac{16T}{\pi(d_c)^3} = \frac{16x65771}{\pi(20)^3} = 41.86 \, N/mm^2$$

Maximum shear stress in screw

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_c^2 + 4\tau^2} = \frac{1}{2}\sqrt{31.83^2 + 4x41.86^2} = 44.8 \, N/mm^2$$

Number of threads of nut in engagement with screw

Let   n = Number of threads of nut in engagement with screw, and

t = Thickness of threads = p / 2 = 5 / 2 = 2.5 mm

We know that bearing pressure on the threads (p_b),

$$5.8 = \frac{W}{\pi dtn} = \frac{10x10^3}{\pi x22.5x2.5xn} = \frac{56.6}{n}$$

∴   n = 56.6/5.8 = 9.76 say 10.

Example

A screw press is to exert a force of 40 kN. The unsupported length of the screw is 400 mm. Nominal diameter of screw is 50 mm. The screw has square threads with pitch equal to 10 mm. The material of the screw and nut are medium carbon steel and cast iron respectively. For the steel used take ultimate crushing stress as 320 MPa, yield stress in tension or compression as 200 MPa and that in shear as 120 MPa. Allowable shear stress for cast iron is 20 MPa and allowable bearing pressure between screw and nut is 12 N/mm². Young's modulus for steel = 210 kN/mm². Determine the factor of safety of screw against failure. Find the dimensions of the nut. What is the efficiency of the arrangement? Take coefficient of friction between steel and cast iron as 0.13.

Solution

Given that

W = 40 kN; L = 400 mm; d_o = 50 mm; p = 10 mm; σ_{cu} = 320 MPa; σ_y = 200 MPa;

τ_y = 120 MPa = 120 N/mm²; τ_c = 20 MPa; p_b = 12 N/mm²; E = 210 kN/mm²;

μ = tanφ = 0.13

Compressive stress on screw

Diameter of core $d_c = d_o - p = 50 - 10 = 40$ mm

Core area $A_c = \frac{\pi}{4}d_c^2 = \frac{\pi}{4}(40)^2 = 1257 \, mm^2$

∴ Direct stress

$$\sigma_c = \frac{W}{A_c} = \frac{40x10^3}{1257} = 31.8 \, N/mm^2$$

Torque to move screw

Mean diameter of screw

$$d = \frac{d_o + d_c}{2} = \frac{50 + 40}{2} = 45 \, mm$$

and

$$\tan \alpha = \frac{p}{\pi d} = \frac{10}{\pi(45)} = 0.07$$

∴ Torque

$$T = Px\frac{d}{2} = W\tan(\alpha + \phi)\frac{d}{2} = W\left[\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha\tan\phi}\right]\frac{d}{2}$$

$$= 181.6 \times 10^3 \text{ N-mm (after putting various values)}$$

Torque transmitted by screw (T)

$$181.6x10^3 = \frac{\pi}{16}(\tau)(d_c)^3 = \frac{\pi}{16}(\tau)(40)^3 = 12568\tau$$

Solving $\tau = 14.45$ N/mm^2

Maximum shear stress

According to maximum shear stress theory

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_c^2 + 4\tau^2} = \frac{1}{2}\sqrt{31.8^2 + 4(14.45)^2} = 21.5 \, N/mm^2$$

Factor of safety

$$FS = \frac{\tau_y}{\tau_{max}} = \frac{120}{21.5} = 5.58$$

Considering the screw as a column, assuming one end fixed and other end free.

$$W_{cr} = A_c\sigma_y\left[1 - \frac{\sigma_y}{4C\pi^2 E}\left(\frac{L}{k}\right)^2\right]$$

For one end fixed and other free, C = 0.25.

$$W_{cr} = 1257x200\left[1 - \frac{200}{4x0.25x\pi^2x210x10^3}\left(\frac{400}{10}\right)^2\right]N = 212700N$$

[Here k = d$_c$/4 = 40/4 = 10 mm]

$$\therefore \qquad FS = \frac{W_{cr}}{W} = \frac{212700}{40x10^3} = 5.3$$

We shall take *larger* value of the factor of safety.

∴ Factor of safety = 5.58 say 6

Dimensions of the nut

Let   n = Number of threads in contact with nut, and

   h = Height of nut = p × n

Assume that the load is uniformly distributed over the threads in contact.

We know that the bearing pressure (p_b),

$$12 = \frac{W}{\frac{\pi}{4}\left[d_o^{\,2} - d_c^{\,2}\right]n} = \frac{40x10^3}{\frac{\pi}{4}\left[50^2 - 40^2\right]n} = \frac{56.6}{n}$$

Solving   n = 4.7 say 5 threads

and   h = p x n = 10 x 5 = 50 mm

Now let us check for the shear stress induced in the nut which is of cast iron. We know that

$$\tau_{nut} = \frac{W}{\pi n d_o t} = \frac{40x10^3}{\pi x 5 x 50 x 5} = 10.2 \, N/mm^2 = 10.2 \, MPa$$

(Here t = p / 2 = 10 / 2 = 5 mm)

This value is less than the given value of τ_c = 20 MPa, hence the nut is safe.

Efficiency of the arrangement

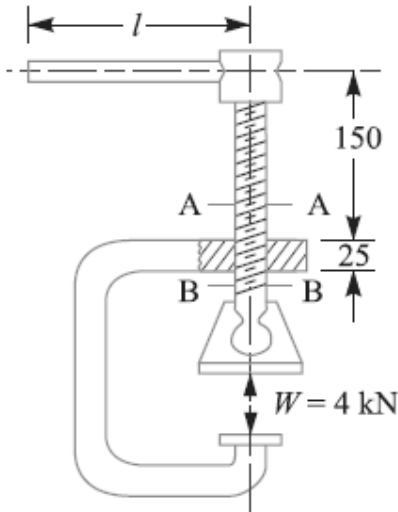We know that torque required to move the screw with no friction,

$$T_o = W \tan\alpha \, x \frac{d}{2} = 40x10^3 \, x0.07 \, x\frac{45}{2} = 63x10^3 \, N - mm$$

∴ Efficiency of the arrangement

$$\eta = \frac{T_o}{T} = \frac{63x10^3}{181.6x10^3} = 0.347 \, i.e. 34.7\%$$

Example

A C-clamp, as shown in following figure (all dimensions in mm), has trapezoidal threads of 12 mm outside diameter and 2 mm pitch.

The coefficient of friction for screw threads is 0.12 and for the collar is 0.25. The mean radius of the collar is 6 mm. If the force exerted by the operator at the end of the handle is 80 N, find: (i) The length of handle; (ii) The maximum shear stress in the body of the screw and where does this exist; and (iii) The bearing pressure on the threads.

Solution

Given data

$d_o = 12$ mm; $p = 2$ mm; $\mu = \tan\phi = 0.12$; $\mu_2 = 0.25$; $R = 6$ mm; $P_1 = 80$ N; $W = 4$ kN

Torque required to overcome friction at the screw

$$d = d_o - p/2 = 12 - 2/2 = 11 \text{ mm}$$

$\therefore$ $$\tan\alpha = \frac{p}{\pi d} = \frac{2}{\pi x 11} = 0.058$$

Angle of trapezoidal threads $= 2\beta = 30^0$ i.e. $\beta = 15^0$.

Virtual coefficient of friction

$$\mu_1 = \tan\phi_1 = \frac{\mu}{\cos\beta} = \frac{0.12}{\cos 15^0} = 0.124$$

Torque required to overcome friction at the screw

$$T_1 = Px\frac{d}{2} = W\tan(\alpha + \phi_1)\frac{d}{2} = W\left(\frac{\tan\alpha + \tan\phi_1}{1 - \tan\alpha \tan\phi_1}\right)\frac{d}{2}$$

$$= 4033 \text{ N-mm after putting various values.}$$

Torque required to overcome friction at the collar

$$T_2 = \mu_2 \, W \, R = 0.25 \times 4000 \times 6 = 6000 \text{ N-mm}$$

Total torque required at the end of handle,

$$T = T_1 + T_2 = 4033 + 6000 = 10033 \text{ N-mm}$$

Length of handle

We know that the torque required at the end of handle (T),

$$10\,033 = P_1 \times l = 80 \times l$$

or $l = 10\,033 / 80 = 125.4 \text{ mm}$

Maximum shear stress in the body of the screw

Consider two sections A-A and B-B. The section A-A just above the nut, is subjected to torque and bending. The section B-B just below the nut is subjected to collar friction torque and direct compressive load. Thus, both the sections must be checked for maximum shear stress.

Section A-A

Core (inner) diameter of screw

$$d_c = d_o - p = 12 - 2 = 10 \text{ mm}$$

Torque at A-A

$$T = \frac{\pi}{16}\tau d_c^{\ 3}$$

∴ Shear stress

$$\tau = \frac{16T}{\pi d_c^{\ 3}} = \frac{16 x 10033}{\pi x 10^3} = 51.1\,N/mm^2$$

Bending moment at A-A

$$M = P_1 \text{ x } 150 = 80 \text{ x } 150 = 12000 \text{ N-mm} = \frac{\pi}{32}\sigma_b d_c^{\ 3}$$

∴ Bending stress $\sigma_b = 122.2$ N/mm^2 after putting values.

Maximum stress

$$\tau_{\max} = \frac{1}{2}\sqrt{\sigma_b^{\ 2} + 4\tau^2} = \frac{1}{2}\sqrt{122.2^2 + 4 x 51.1^2} = 79.65\,N/mm^2$$

Section B-B

Since the section B-B is subjected to collar friction torque (T_2), therefore the shear stress

$$\tau = \frac{16T_2}{\pi d_c^{\ 3}} = \frac{16 x 6000}{\pi x 10^3} = 30.6\,N/mm^2$$

Direct compressive stress

$$\sigma_c = \frac{W}{A_c} = \frac{4W}{\pi d_c^2} = \frac{4x4000}{\pi x 10^2} = 51 N / mm^2$$

Max. shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_c^2 + 4\tau^2} = \frac{1}{2}\sqrt{51^2 + 4x30.6^2} = 39.83 N / mm^2$$

We see that maximum shear stress is 79.65 MPa which occurs at A-A.

Bearing pressure on the threads

We know that height of the nut,

h = n × p = 25 mm (given)
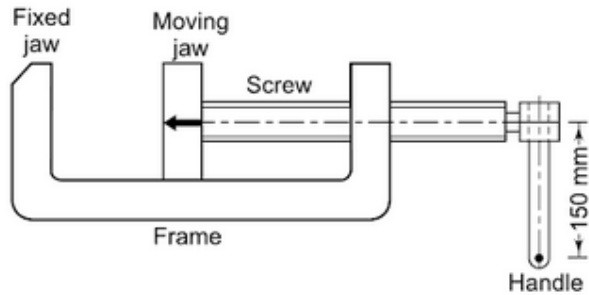
∴ Number of threads in contact,

n = h / p = 25 / 2 = 12.5

and thickness of threads, t = p / 2 = 2 / 2 = 1 mm

We know that bearing pressure on the threads,

$$p_b = \frac{W}{\pi dtn} = \frac{4000}{\pi x 11 x 1 x 12.5} = 9.26 N / mm^2$$

Problem

A machine vice, as shown in following figure, has Single-Start, square threads with 22 mm nominal diameter and 5 mm pitch.

The outer and inner diameters of the friction collar are 55 and 45 nun respectively. The coefficients of friction for thread and collar are 0.15 and 0.17 respectively. The machinist can comfortably exert a force of 125 N on the handle at a mean radius of 150 mm Assuming uniform wear for the collar calculate

    (i) the clamping force developed between the jaws; and

    (ii) the overall efficiency of the clamp.

Answer: 2868.73 N; 12.18%

A double threaded power screw is used to raise a load of 5 kN. The nominal diameter is 60 mm and the pitch is 9 mm. The threads are Acme type ($2\theta = 29^0$) and the coefficient of friction at the screw threads is 0.15. neglecting collar friction, calculate:

  (i) the torque required to raise the lead

  (ii) the torque required to lower the load

  (iii) the efficiency of the screw for lifting load.

Answer: 36.39 N-m; 7.06 N-m; 39.35%

Threaded (Bolted) Joints

Threaded joint is defined as a separable joint of two or more machine parts that are held together by means of a threaded fastening such as a bolt and a nut. The salient features of this definition are as follows:

- Threaded joints are used to hold two or more machine parts together. These parts can be dismantled, if required, without any damage to machine parts or fastening. Therefore. threaded joints are detachable joints, unlike welded joints,

- Thread is the basic clement of these joints. The thread is formed by cutting a helical groove on the surface of a cylindrical rod or cylindrical hole. The threaded element can take the shape of bolt and nut, screw or stud. Sometimes, threads are cut on the pails to be joined.

ADVANTAGES OF THREADED JOINTS

- The parts are held together by means of a large clamping force. There is wedge action at the threads, which increases the clamping force. There is no loosening of the parts. Therefore, threaded joints are 'reliable' joints.

- The parts are assembled by means of a spanner. The length of the spanner is large compared with the radius of the thread. Therefore, the mechanical advantage is more and force required to tighten the joint is small.

- Threaded joints have small overall dimensions resulting in compact construction.

- The threads are self-locking. Therefore, threaded joints can be placed in any position - vertical, horizontal or inclined. Threaded fasteners are economical to manufacture. Their manufacturing is simple. High accuracy can be maintained for the threaded components.

- The parts joined together by threaded joints can be detached as and when required. This requirement is essential in certain applications for the purpose of inspection, repair or replacement.

- Threaded fasteners are standardised and a wide variety is available for different operating conditions and applications.

DISADVANTAGES OF THREADED JOINTS

- Threaded joints require holes in the machine parts that are to be clamped. This results in stress concentration near the threaded portion of the parts. Such areas are vulnerable to fatigue failure.

- Threaded joints loosen when subjected to vibrations.

- Threaded fasteners are considered as a major obstacle for efficient assembly.

- Manual assembly, the cost of tightening a screw can be six to ten times the cost of the screw itself Therefore.

BOLTS VS. SCREW

There are three parts of a threaded fastening, viz., a bolt or screw, a nut and a washer. There is a basic difference between the bolt and the screw. A boll is a fastener with a head and straight threaded shank and intended to be used with a nut to clamp two or more parts. The same bolt can be called screw when it is threaded into a tapped hole in one of the parts and not into the nut. Although boll and screw are similar, there is a fundamental difference in their assembly. A bolt is held stationary, while torque is applied to the nut to make threaded joint, whereas the torque is applied to the screw to turn it into matching threads in one of the parts. A nut is a small symmetrical part, usually having hexagonal or square shape, containing matching internal threads.

IMPORTANT TERMS USED IN SCREW THREADS

The following terms used in screw threads, as shown in following figure, are important from the subject point of view:
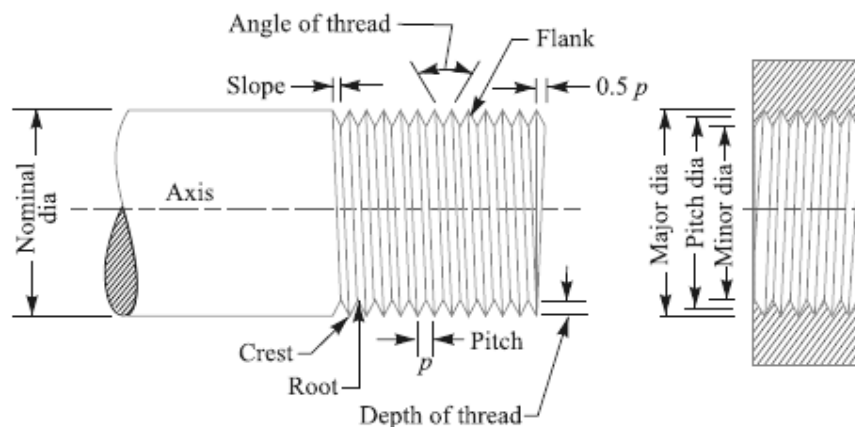
Major diameter

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.

Minor diameter

It is the smallest diameter of an external or internal screw thread. It is also known as core or root diameter.

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.

Pitch

It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane.

Mathematically,

Pitch = 1/No. of threads per unit length of screw

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.

Crest

 It is the top surface of the thread.

Root

It is the bottom surface created by the two adjacent flanks of the thread.

Depth of thread

It is the perpendicular distance between the crest and root.

Flank

It is the surface joining the crest and root.

Angle of thread

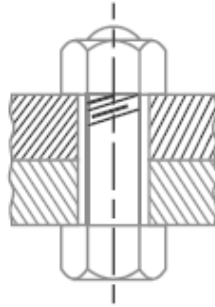It is the angle included by the flanks of the thread.

Slope

It is half the pitch of the thread.

TYPES OF SCREW FASTENINGS

Following are the common types of screw fastenings :

Through bolts

A through bolt (or simply a bolt) is shown in following figure. 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.
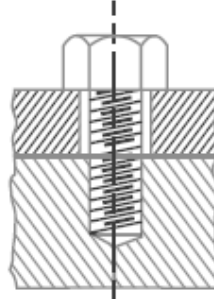
Tap bolts

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 figure.
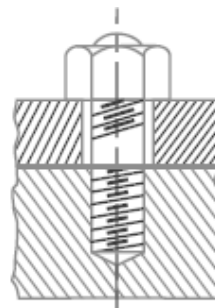
Studs

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 given figure.

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.

INITIAL STRESSES DUE TO SCREWING UP FORCES

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

Tensile stress due to stretching of bolt

Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$P_i = 2840\ d$ N

where P_i = Initial tension in a bolt, and

   d = Nominal diameter of bolt, in mm.

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint, then the initial tension in a bolt may be reduced to half of the above value. In such cases

$P_i = 1420\ d$ N

The small diameter bolts may fail during tightening, therefore bolts of smaller diameter (less than M 16 or M 18) are not permitted in making fluid tight joints.

If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it, is given by

   P = Permissible stress × Cross-sectional area at bottom of the thread

      (i.e. stress area)

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

Stress area is given by

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

where d_p = Pitch diameter, and

d_c = Core or minor diameter.

Torsional shear stress

The torsional shear stress caused by the frictional resistance of the threads during its tightening may be obtained by using the torsion equation.

$$\tau = \frac{16T}{\pi d_c^{\;3}}$$

where τ = Torsional shear stress,

T = Torque applied, and

d_c = Minor or core diameter of the thread.

Shear stress across the threads

The average thread shearing stress for the screw (τ_s) is obtained by using the relation :

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

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

The average thread shearing stress for the nut is

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

where d is major diameter.

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.

Bending stress

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

$$\sigma_b = \frac{xE}{2l}$$

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

Two machine parts are fastened together tightly by means of a 24 mm tap bolt (d_c = 20.32 mm). 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

Let σ_t = Stress set up in the bolt.

We know that initial tension in the bolt,

$$P = 2840 \ d = 2840 \times 24 = 68 \ 160 \ N$$

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

$$68160 = \frac{\pi}{4} d_c^{\ 2} \sigma_t = \frac{\pi}{4}(20.30)^2 \sigma_t = 324\sigma_t$$

$\therefore$ σ_t = 68 160 / 324 = 210 N/mm^2 = 210 MPa

STRESSES DUE TO EXTERNAL FORCES

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,

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

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,

$$P_s = \frac{\pi}{4} d^2 \tau n$$

Combined tension and shear stress

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,

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

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

Example

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

Solution

Given data

      T = 25 N-m; n = 4; R_p = 30 mm; τ = 30 MPa

Load carried by flange coupling

$$P_s = \frac{T}{R_p} = \frac{25x10^3}{30} = 833.3N$$

Resisting load on bolts

It will be $\frac{\pi}{4} d_c^2 \tau n = \frac{\pi}{4} d_c^2 (30)n = 94.26 d_c^2$

Core diameter of bolt

Solving above two equations

$$d_c{}^2 = 833.3 / 94.26 = 8.84 \Rightarrow d_c = 2.97 \, mm$$

From table/code (coarse series), we find that the standard core diameter of the bolt is 3.141 mm and the corresponding size of the bolt is M 4.

STRESS DUE TO COMBINED FORCES

It is calculated from following formula

$$P = P_1 + \frac{a}{1+a} x P_2 = P_1 + K P_2$$

where

P_1 = Initial tension due to tightening of the bolt.

P_2 = External load on the bolt, and

a = Ratio of elasticity of connected parts to the elasticity of bolt.

Example

The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. Take K = 0.5.

Solution

Given that

$p = 0.7$ N/mm^2; n = 12; D = 300 mm; σ_t = 100 MP

External load

It will be $\dfrac{\pi}{4} D^2 p = \dfrac{\pi}{4} (300)^2 (0.7) = 49490 N$

$\therefore$ External load on the cylinder head per bolt

$P_2 = 49490/12 = 4124$ N

Design of bolts

Let   d = nominal diameter of the bolt

d_c = core diameter of the bolt

n = Number of bolts.

We know that the total upward force acting on the cover plate (or on the bolts),

We know that initial tension due to tightening of bolt,

$P_1 = 2840\ d$ N (where d is in mm)

∴ Resultant axial load on the bolt,

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

We know that load on the bolt (P),

$$2840d + 2062 = \frac{\pi}{4} d_c^2 \sigma_t = \frac{\pi}{4}(0.84d)^2 100 = 55.4 d^2$$

∴ $55.4d^2 - 2840d - 2062 = 0$

∴ $d = \dfrac{51.3 \pm \sqrt{(51.3)^2 + 4 x 37.2}}{2} = 52\,mm$

Thus, we shall use a bolt of size M52.

Problem

The cylinder head of a steam engine is subjected to a pressure of 1 N/mm². It is held in position by means of 12 bolts. The effective diameter of the cylinder is 300 mm. A soft copper gasket is used to make the joint leak proof. Determine the size of the bolts so that the stress in the bolts does not exceed 100 MPa.

Answer: M 36

BOLT OF UNIFORM STRENGTH

Bolts are subjected to shock and impact loads in certain applications. The bolts of cylinder head of an internal combustion engine or the bolts of connecting rod are the examples of such applications. In such cases, *resilience of the bolt* is important design consideration to prevent breakage at the threads. Resilience is defined as the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. A resilient bolt absorbs energy within elastic range without any permanent deformation and releases this energy when unloaded. It can be called spring property of the bolt. A resilient bolt absorbs shocks and vibrations like leaf springs of the vehicle. In other words, the bolt acts like a spring.

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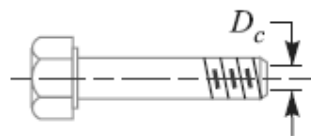
In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

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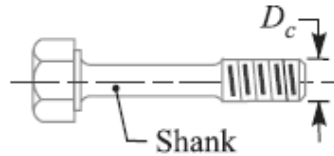
This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us bolts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

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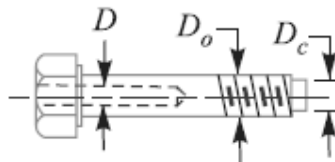
In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the thread.

Let   D = Diameter of the hole.

         D_o = Outer diameter of the thread, and

         D_c = Root or core diameter of the thread.

∴ $$\frac{\pi}{4}D^2 = \frac{\pi}{4}\left[D_o^2 - D_c^2\right]$$

∴ $$D = \sqrt{D_o^2 - D_c^2}$$

ASSIGNMENT

Q.1. (AMIE S17, 6 marks): What are the advantages of square threads over trapezoidal threads and what are the disadvantages of square threads with trapezoidal threads?

Q.2. (AMIE S12, 9 marks): A broaching cutter is pulled through a job by an Acme threaded screw which rotates at 600 rpm. A collar of internal and external diameters respectively 60 mm and 90 mm supports the tensile force in the screw. The coefficient of friction for all contact surfaces is 0.15 and the Acme thread angle is 30°. The pitch of the Acme thread is 10 mm and the screw diameter is 55 mm. The machine consumes 0.37 kW of power. Calculate the axial force upon the tool.

Answer: 1072 N

Q.3. (AMIE S14, 10 marks): A screw press is to exert a force of 60 kN with an applied torsional moment of 550 Nm. The unsupported length of the screw is 0.45 m and a thrust bearing of hardened steel on CI is provided at the power end. The screw is to be made of steel having an ultimate strength of 530 MN/m^2 and a yield stress of 265 MN/m^2. The design stresses are to be 87.5 MN/m^2 in tension and compression 52.5 MN/m^2 in shear and 14 MN/m^2 in thread bearing. The nut is of Cl and the permissible shear is 21 MN /m^2. Determine the dimension of the screw and nut.

Answer: Main dia of screw = 45 mm; height of nut = 30 mm

Q.4. (AMIE W14, 12 marks): A square threaded screw is required to work against an axial force of 8.0 kN and has the following dimensions:

> Major diameter, d = 32 mm
>
> Pitch, p = 4 mm with single start
>
> Coefficient of friction, μ = 0.08

Axial force rotates with screw.

Calculate the (i) torque required when screw moves against the load. (ii) torque required when screw moves in the same direction an the load, (iii) efficiency of the screw and (iv) if instead of the square thread, the screw has the Acme threads with thread angle $2\theta = 29^0$, calculate the same quantities.

Answer: For acme thread, T = 16441.07 N-mm, 5613.83 N-mm; For square thread, T = 15279.67 N-mm, T = 5231.68 N-mm, 32.67%

Q.5. (AMIE W12, S15, 10 marks): A power screw, having double start square thread of 25 mm nominal diameter, 21 mm core diameter and 5 mm pitch, acted upon by an axial thrust of 10 kN. The outer diameter of collar is 50 mm and inner diameter of collar is 20 mm. The coefficient of friction at the screw is 0.2. The screw rotates at 12 rpm. Calculate the (i) stresses in screws (ii) torque required to rotate the screw and (iii) power required to drive it. Take μ = 0.15.

Q.6. (AMIE S17, 10 marks): A double-threaded power screw, with ISO metric trapezoidal threads is used to raise a load of 300 kN. The nominal diameter is 100 mm and the pitch is 12 mm. The coefficient of friction at the screw threads is 0.15. Neglecting collar friction, calculate (i) torque required to raise the load (ii) torque required to lower the load; and (iii) efficiency of the screw.

Answer: (i) 3378.72 N-m (ii) 1030.39 N-m (iii) 33.93%

Q.7. (AMIE W17, 6 marks): Sketch the shape of Vee thread, Acme thread and Buttress thread.

Q.8. (AMIE W17, 4 marks): Derive the expression for efficiency of the screw.

Q.9. (AMIE W17, 10 marks): A 25 mm single start square threaded screw with pitch 5 mm is 400 mm long between the nut and collar. The axial load is 25 kN and coefficient of friction between the screw and nut is 0.12. Determine the factor of safety if yield point stress is 350 MPa.

Q.10. (AMIE W16, 4 marks): Mention the aspects by which a hexagonal head differs from square in bolted joints.

Q.11. (AMIE S14, 18, 5 marks): Discuss bolts of uniform strength in detail.

Q.12. (AMIE S15, 16, 6 marks): What are different stresses set up in a bolt due to initial tightening? Explain these stresses in detail.

Q.13. (AMIE W11, S13, 7 marks): The cylinder head of a steam engine is subjected to a steam pressure of 0.7 N/mm^2. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of the cylinder is 300 mm. Find the size of the bolts so that the stress in the bolts is not to exceed 100 MPa. Take core diameter $d_c = 0.84d$, where d is the nominal diameter (effective).

Answer: 52 mm, M52

Q.14. (AMIE W16, 10 marks): A machine vice has single start square threads with 20 mm nominal diameter and 4 mm pitch. The inner and outer diameters of the friction collar are 12 mm and 25 mm respectively. The coefficient of friction for threads and collar are 0.15 and 0.20 respectively. If the person operating the vice can exert a force of 100 N on the handle, at a radius of 180 mm, determine

       (i)   the clamping force developed between the jaws, and

       (ii) the efficiency of the clamp

Assume uniform pressure at the contacting collar surface.

Answer: 4675.32 N; 16.54%

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