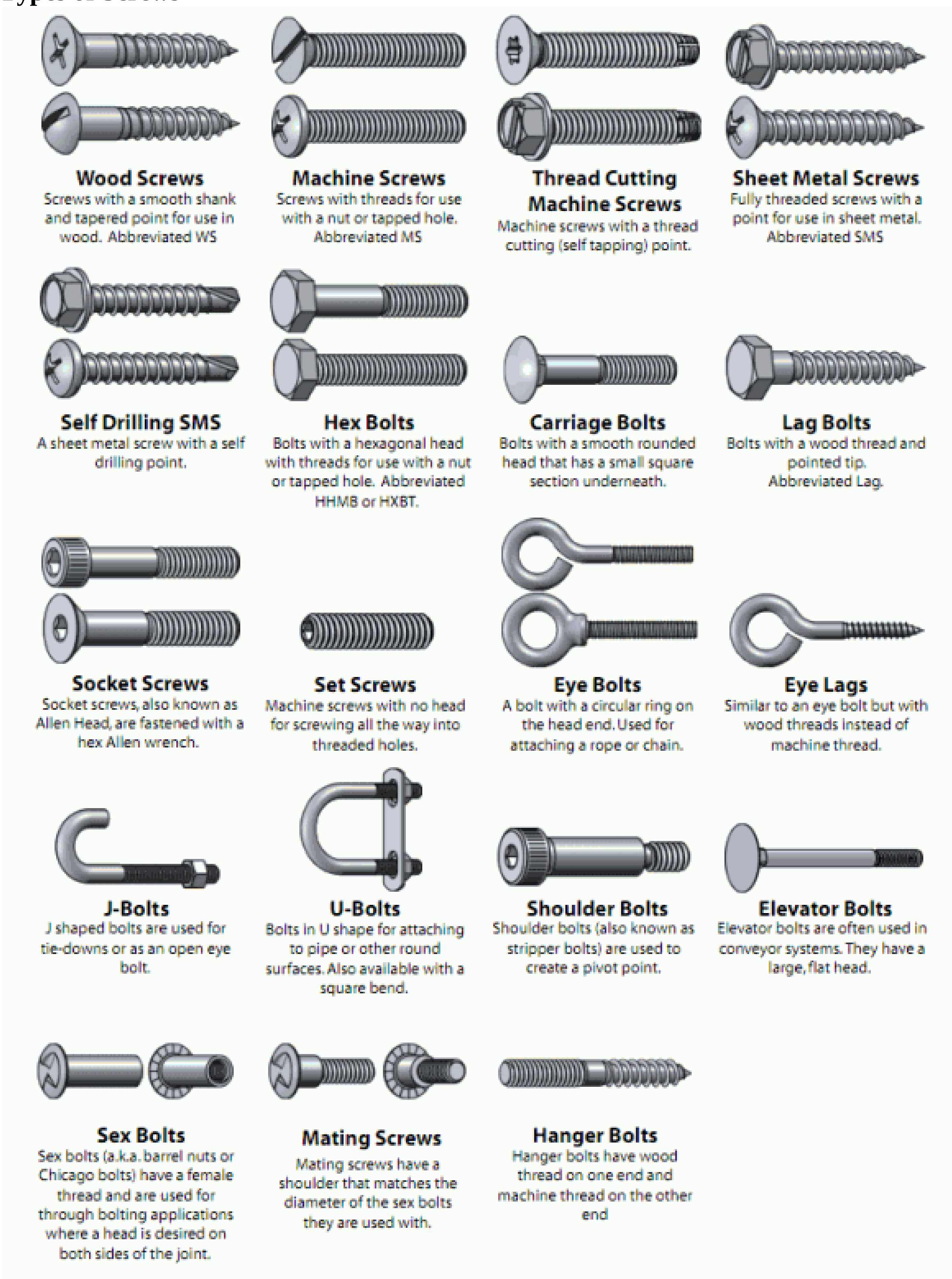
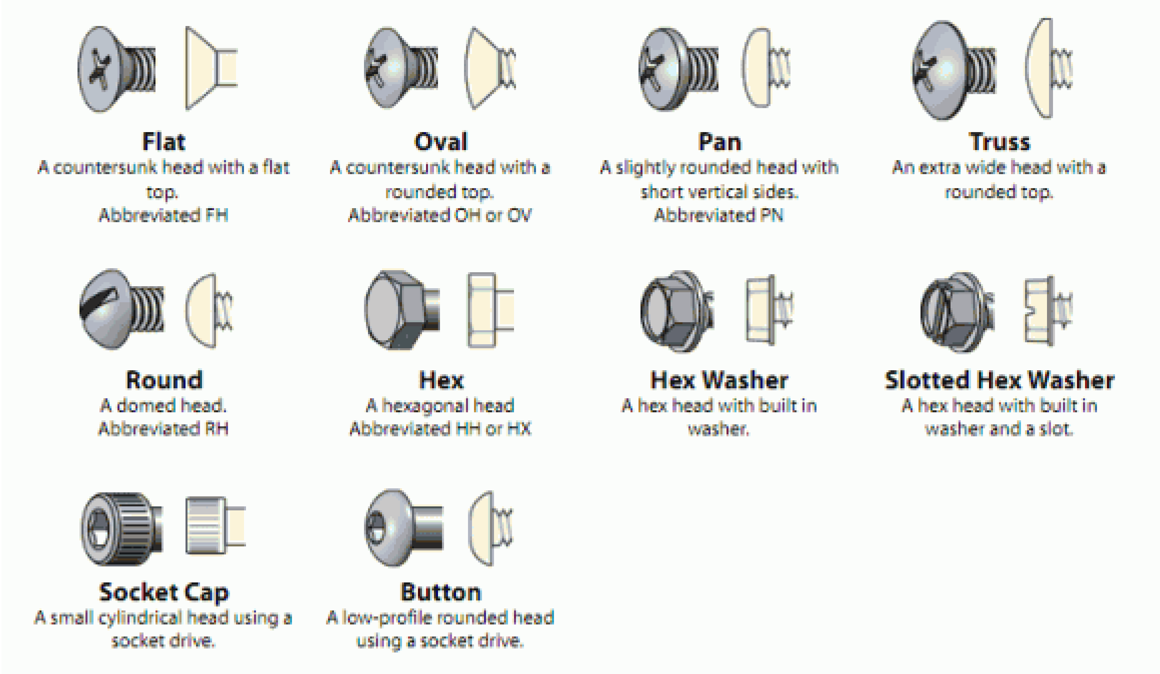
Screws, Fasteners

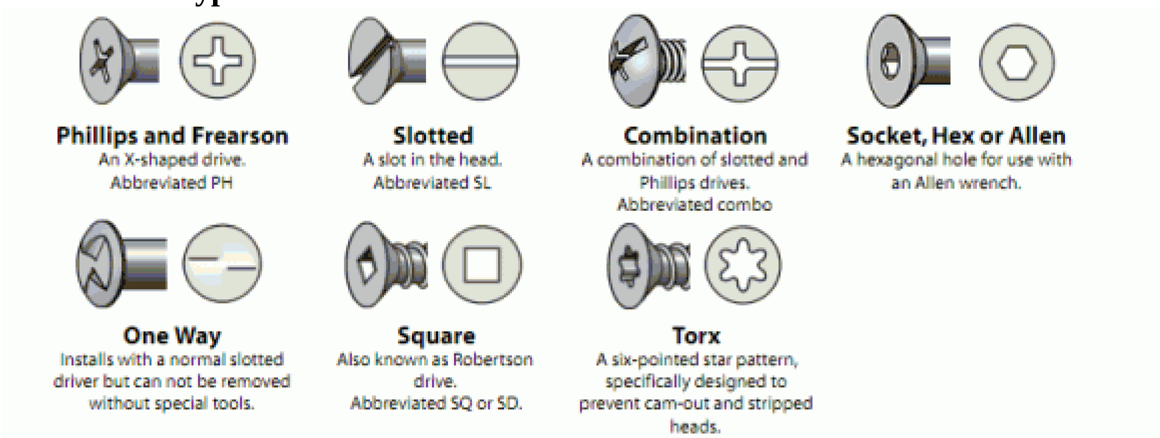
Types of Screws



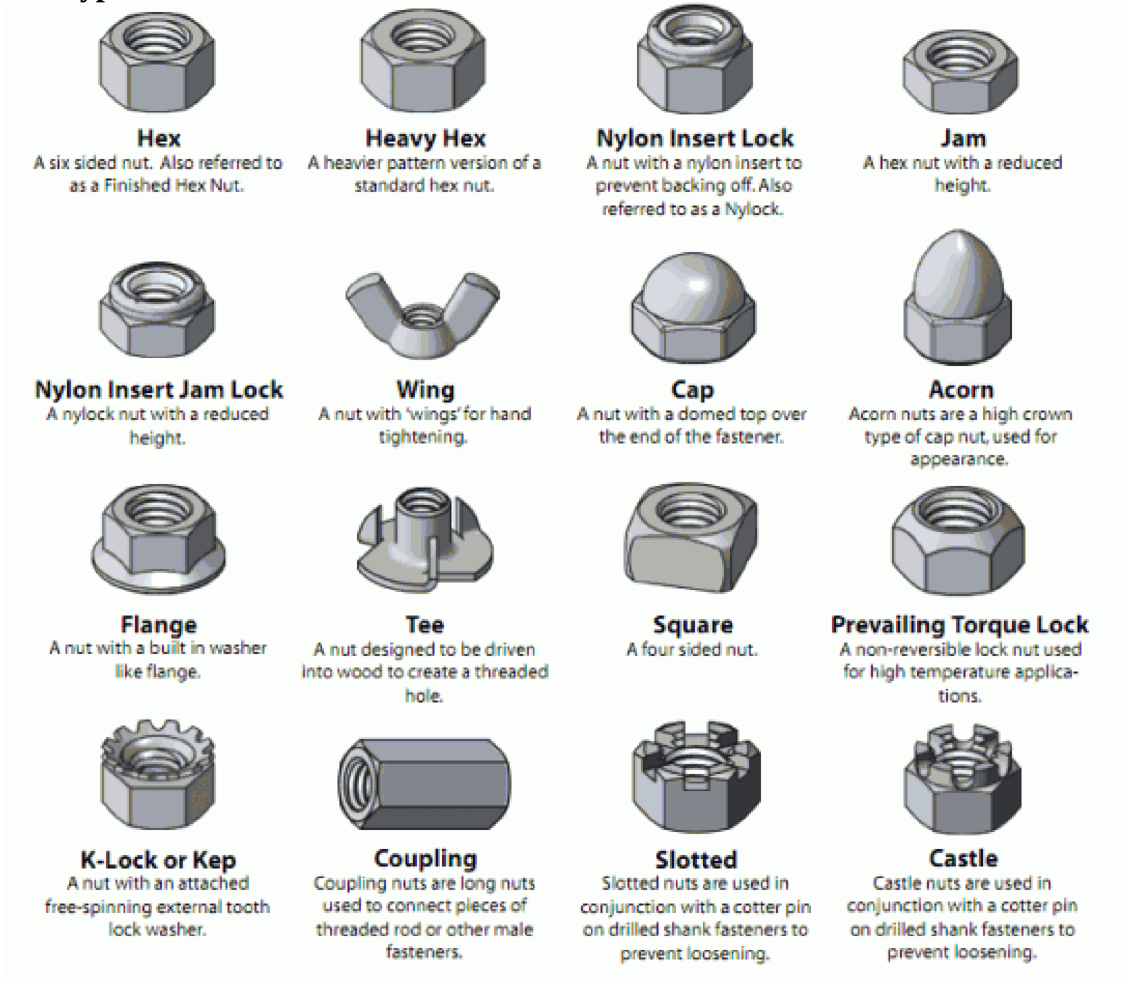
Types and Styles of Screw Heads



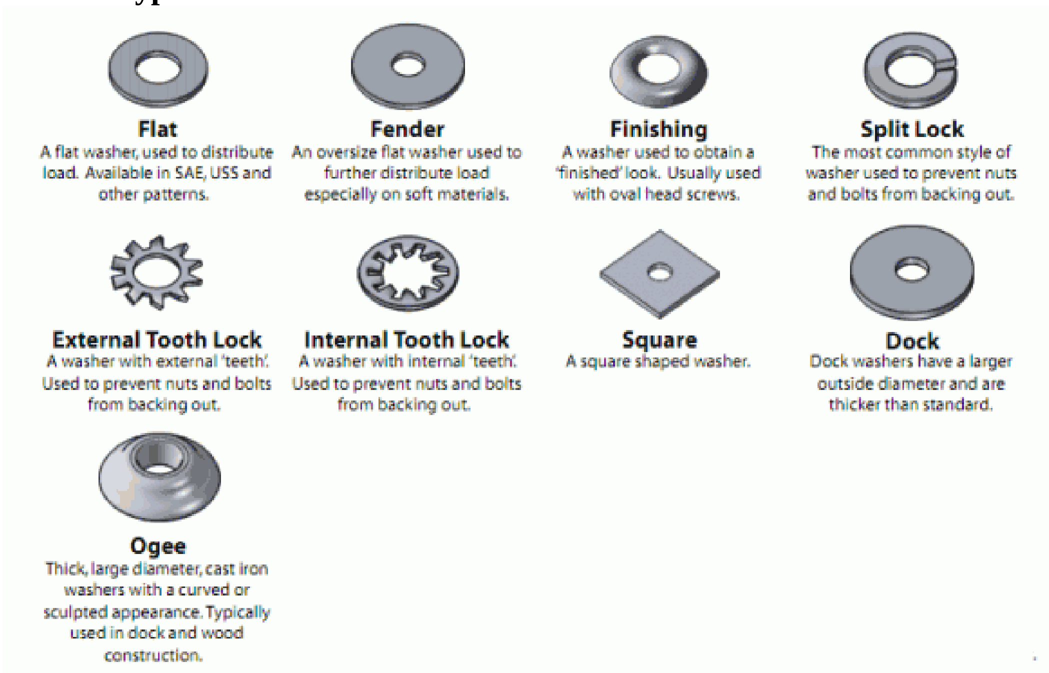
Screw Drive Types



Nut Types



Washer Type



Thread Standards and Definitions

The terminology of screw threads, illustrated in Fig. 3.1, is explained as follows:
The pitch is the distance between adjacent thread forms measured parallel to the thread axis. The pitch in U.S. units is the reciprocal of the number of thread forms per inch N.
The major diameter d is the largest diameter of a screw thread. The minor (or root) diameter d_r is the smallest diameter of a screw thread. The pitch diameter d_p is a theoretical diameter between the major and minor diameters.

The lead λ, not shown, is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread, as in Fig. 3.1, the lead is the same as the pitch.

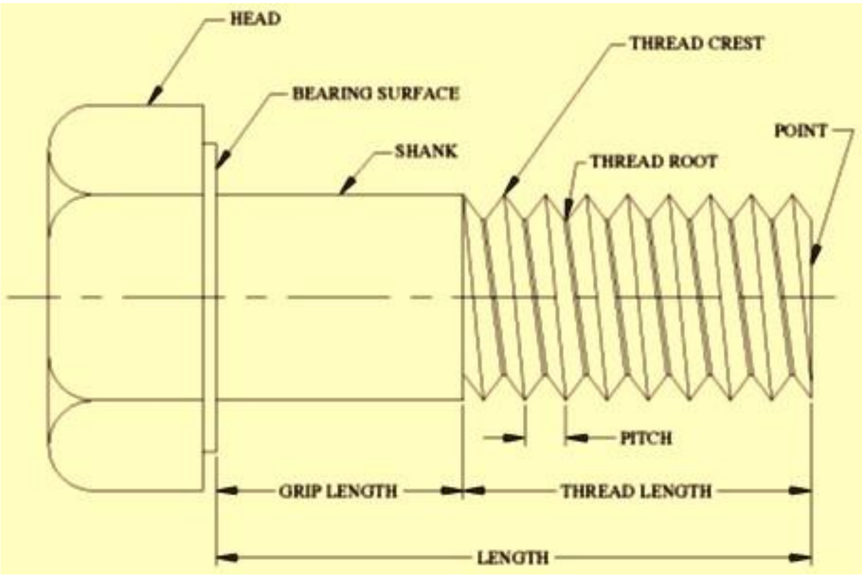
A multiple-threaded product is one having two or more threads cut beside each other (imagine two or more strings wound side by side around a pencil). Standardized products such as screws, bolts, and nuts all have single threads; a double-threaded screw has a lead equal to twice the pitch, a triple-threaded screw has a lead equal to 3 times the pitch, and so on.

All threads are made according to the right-hand rule unless otherwise noted. That is, if the bolt is turned clockwise, the bolt advances toward the nut. The American National (Unified) thread standard has been approved in this country and in Great Britain for use on all standard threaded products. The thread angle is 60° and the crests of the thread may be either flat or rounded.

Figure 3.2 shows the thread geometry of the metric M and MJ profiles. The M profile replaces the inch class and is the basic ISO 68 profile with 60° symmetric threads. The MJ profile has a rounded fillet at the root of the external thread and a larger minor diameter of both the internal and external threads. This profile is especially useful where high fatigue strength is required.

A great many tensile tests of threaded rods have shown that an unthreaded rod having a diameter equal to the mean of the pitch diameter and minor diameter will have the same tensile strength as the threaded rod. The area of this unthreaded rod is called the tensile-stress area A_t of the threaded rod.

Metric threads are specified by writing the diameter and pitch in millimeters, in that order. Thus, M12 x 1.75 is a thread having a nominal major diameter of 12 mm and a pitch of 1.75 mm. Note that the letter M, which precedes the diameter, is the clue to the metric designation.



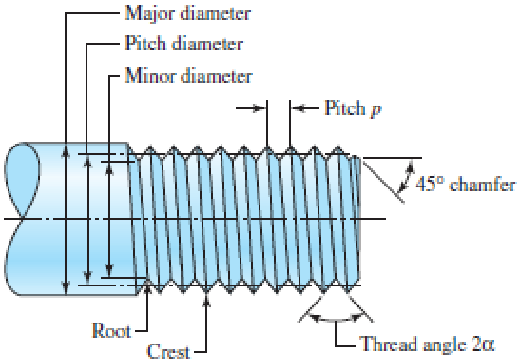


Figure 3.1

Terminology of screw threads. Sharp vee threads shown for clarity; the crests and roots are actually flattened or rounded during the forming operation.

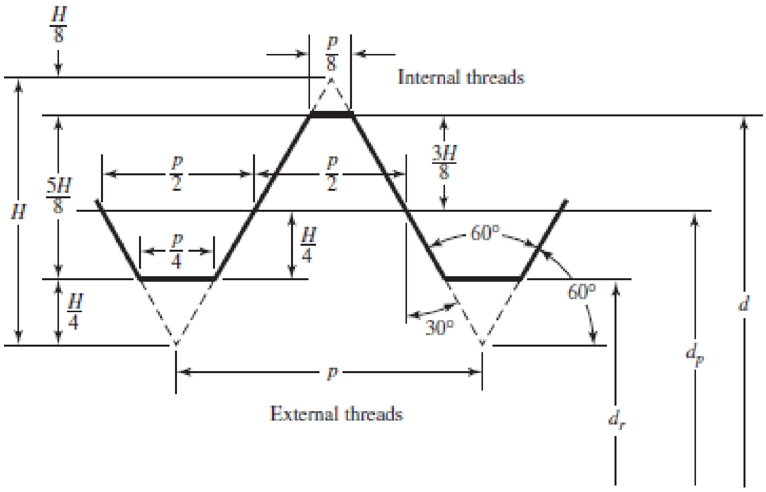


Figure 3.2

Basic profile for metric M and MJ threads.
d = major diameter
d_r = minor diameter
d_p = pitch diameter
p = pitch
$H = \frac{\sqrt{3}}{2} p$

Square and Acme threads, whose profiles are shown in Fig. 3.3a) and b), respectively, are used on screws when power is to be transmitted. However, other pitches can be and often are used, since the need for a standard for such threads is not great.

Modifications are frequently made to both Acme and square threads. For instance, the square thread is sometimes modified by cutting the space between the teeth so as to have an included thread angle of 10 to 15°. This is not difficult, since these threads are usually cut with a single-point tool anyhow; the modification retains most of the high efficiency inherent in square threads and makes the cutting simpler. Acme threads are sometimes modified to a stub form by making the teeth shorter. This results in a larger minor diameter and a somewhat stronger screw.

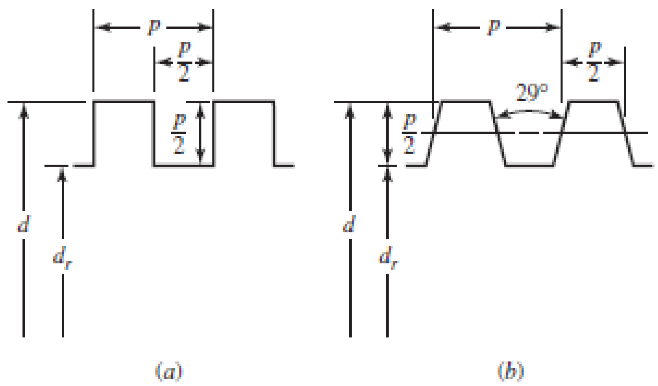


Figure 3.3

(*a*) Square thread;
(*b*) Acme thread.

The forces on the thread

In Fig.3.4 a square-threaded power screw with single thread having a mean diameter d_2, a pitch p, a lead angle γ, is loaded by the axial tension force F_Q. We wish to find an expression for the torque required to raise this load, and another expression for the torque required to lower the load.

First, imagine that a single thread of the screw is unrolled or developed (Fig. 3.5) for exactly a single turn. Then one edge of the thread will form the hypotenuse of a right triangle whose base is the circumference of the mean-thread-diameter circle and whose height is the lead. The angle γ, in Figs. 3.4 and 3.5, is the lead angle of the thread.

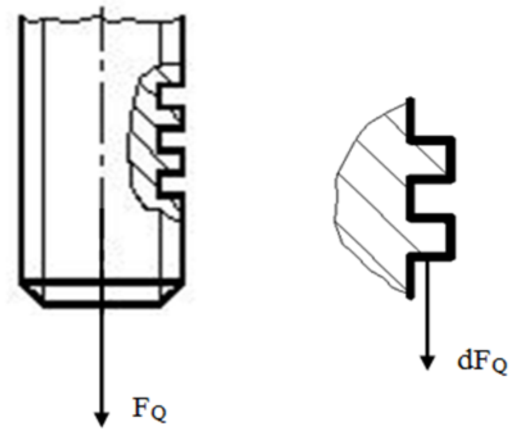


Fig. 3.4 Screw with single square-thread

F_Q – axis force

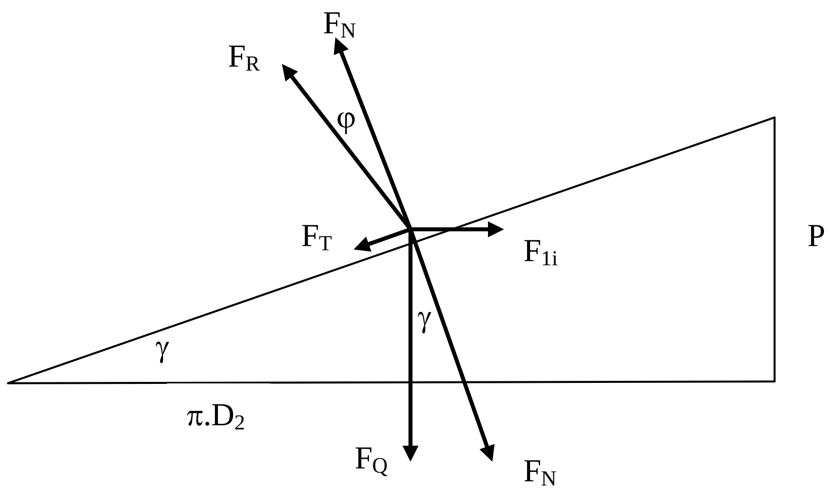


Fig.3.5 Forces diagrams

Force required to move in the thread:

F_{1i} – Ideal Force required to move in the thread

$$F_{1i} = F_Q . \tan\, \gamma$$

Angle of pitch:

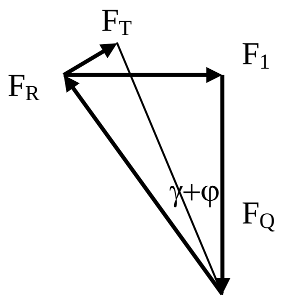
$$\tan\gamma = \frac{P}{\pi . d_2}$$

P- Pitch – see Handbook
D2 (d2) – Pitch diameter – see Handbook

Frictional force F_T:

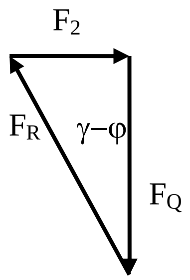
$$F_T = F_N . \mathrm{f} = F_N . \tan\varphi$$

For raising the load F_1:



$$F_1 = F_Q . \tan(\gamma + \varphi)$$

For lowering the load F_2:



$$F_2 = F_Q . \tan(\gamma - \varphi)$$

Self-locking

This is the torque required to overcome a part of the friction in lowering the load. It may turn out, in specific instances where the lead is large or the friction is low, that the load will lower itself by causing the screw to spin without any external effort. In such cases, the force F_2 from Eq. will be negative or zero. When a positive force is obtained from this equation, the screw is said to be *self-locking*. Thus the condition for self-locking is:

Limit of self-locking: $F_2 = 0$

$F_Q = 0$ - nothing

Ak $F_2 < 0$ then screw is not self-locking and $\varphi' < \gamma$

Acme threads

The preceding equations have been developed for square threads where the normal thread loads are parallel to the axis of the screw. In the case of Acme or other threads, the normal thread load is inclined to the axis because of the thread angle 2β and the lead angle γ.

Since lead angles are small, this inclination can be neglected and only the effect of the thread angle considered Fig.3.6. The effect of the angle a is to increase the frictional force by the wedging action of the threads. Therefore the frictional terms must be divided by cos β. For raising the load, or for tightening a screw or bolt, this yields.

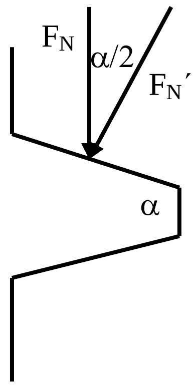


Fig.3.6

$$F_N' = \frac{F_N}{\cos\frac{\alpha}{2}}$$

$$F_T = F_N'.\mathrm{f} = \mathrm{f}.\frac{F_N}{\cos\frac{\alpha}{2}} = F_N.\mathrm{f}'$$

$$\mathrm{f}' = \frac{f}{\cos\frac{\alpha}{2}} = tan.\varphi'$$

f′ - reduced friction coefficient

φ′- reduced friction angle in thread

α – Angle of thread – see Handbook

Force for raising of acme thread F_1:

$$F_1 = F_Q . \tan(\gamma + \varphi')$$

Force for lowering of acme thread F_2:

$$F_2 = F_Q . \tan(\gamma - \varphi')$$

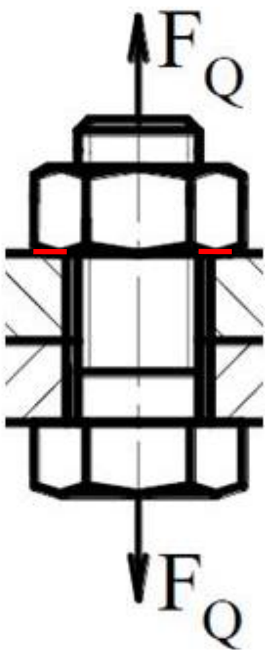
Mechanical efficiency of screws

$$\eta = \frac{F_{1i} . \Delta s}{F_1 . \Delta s} = \frac{F_Q . \tan\gamma}{F_Q . \tan(\gamma + \varphi')}$$

for substitution:

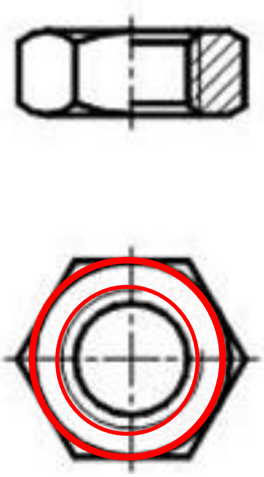
$$\eta = \frac{\tan\gamma}{\tan(\gamma + \varphi')}$$

Torque required to raise and lower the load



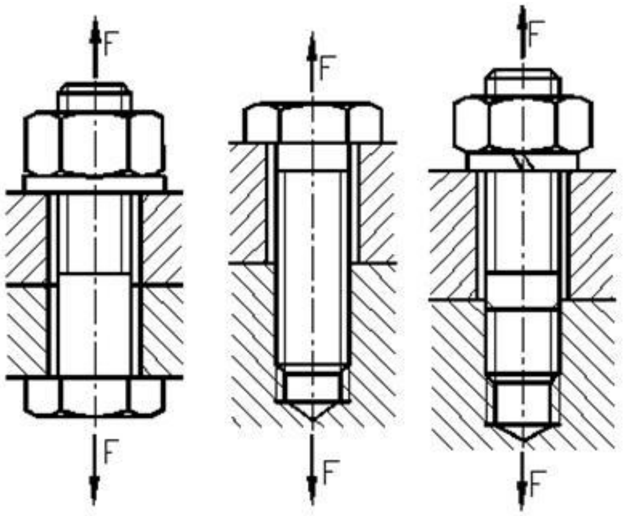
The generally Torque required to raise:

Friction torque in thread:



Finally Torque required to raise M_1:

Strain screw, fasteners



The Basic type of tensile loaded fasteners

A. Screw is torque required to raise and then screw is tensile loaded.

σ – normal stress of screw [MPa]
F_Q- axis force into screw [N]
S_3 – minimal area of thread of screw [mm^2] (the standards defined d_3 – minor diameter [mm])
R_e – yield strength [MPa]
n – safety factor

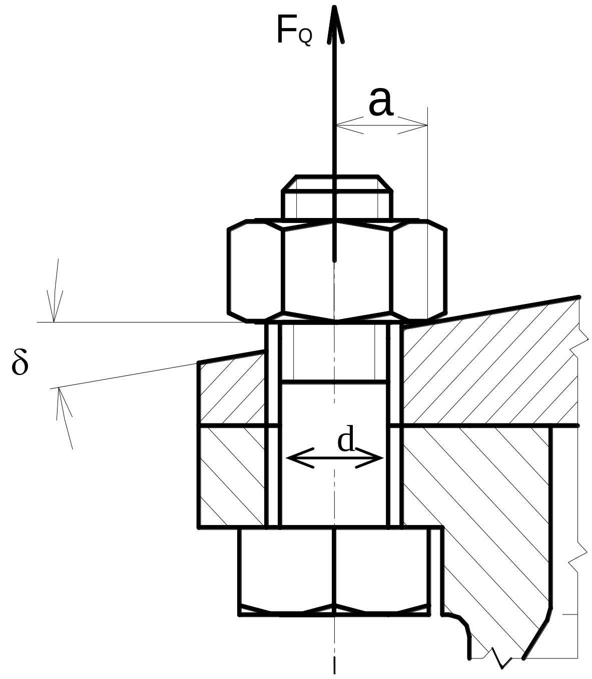
B. Tensile load screw is torque required to raise.

Tensile loaded screw:

σ – normal stress of screw [MPa]
F_Q- axis force into screw [N]
S_3 – minimal area of thread of screw [mm^2] (the standards defined d_3 – minor diameter [mm])
R_e – yield strength [MPa]
n – safety factor
M_{1z} – Friction torque in thread [Nm]
W_{k3} – modulus of torque [mm3]
τ – shear stress [MPa]
σ_{red} – finally total stress method of HMH [MPa]

C. Additional bending of fasteners.

This state of stress on the screw is undesirable and must be eliminated. Different types of elimination pads are used for elimination - see in the introduction. However, if an additional bend occurs in the screw, its calculation is performed as follows:

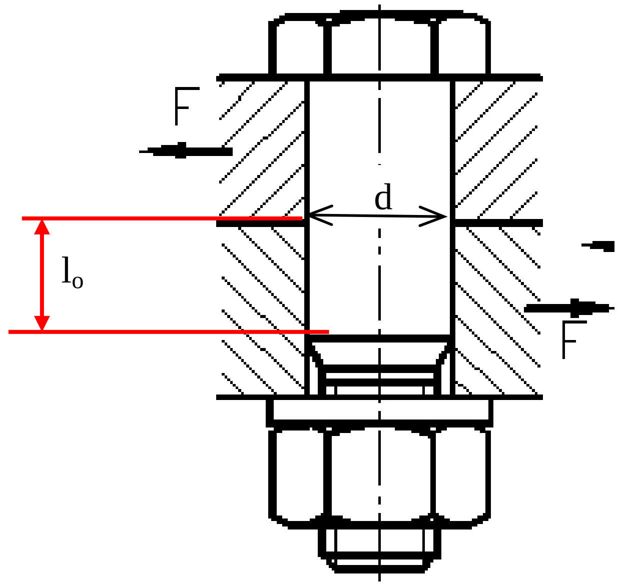


Bending stress of screw

δ- angle [rad] (in Fig.)
σ_{oADD} – additional stress [Pa]
E – Modulus of elasticity
l – Length of screw [m]
d – diameter of screw [m]

D. Shear of fasteners.

For applications of sheared fasteners must be use screws with bolt shank.



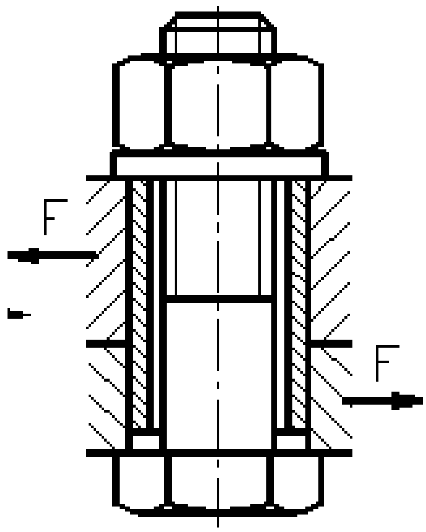
Shear stress of screw:

τ – shear stress of screw [MPa]
F- force [N]
S – shear area of screw [mm^2]
i – number of screw
k- coefficient of exploitation of screws

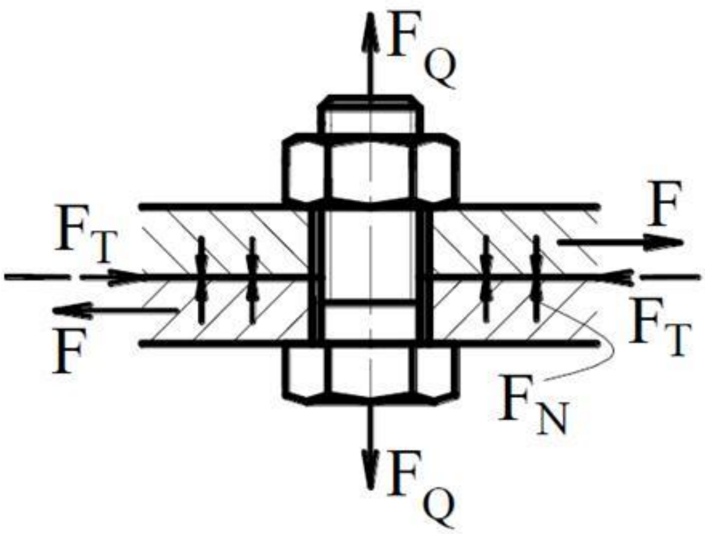
Press stress of fasteners:

E. Other design solutions of fasteners

E1: Shear stress in the screw joint can also be eliminated by using a shear insert as shown in FIG. In this case, a screw with bolt shank does not have to be used, but only a traditional screw is sufficient, the function of which is to form a screw connection and the function of the shear insert is to absorb the loading force.



E2: Another way to eliminate shear stress in a screw joint using only a traditional screw is to use frictional force between the joined materials. Such an embodiment does not require a bolt shank screw, but requires the application of a sufficiently large tensile force in the screw so that it causes a frictional force greater than the load between the bonded surfaces.



Then must be:

$$F_T \geq F$$
$$F_T = F_N.f$$
$$F_N = F_Q.i$$

Caution: increasing the number of screws i does not automatically mean increasing the load-bearing capacity of the screw joint. The reason is the uneven load on the screws. The uniformity of the load of the screws is mainly influenced by their arrangement. In the case of the arrangement of screws around the circumference (connecting flanges) and the number of screws greater than three, it is usually considered with the screw load coefficient at the factory of 75% (ie the coefficient has a value of 0.75) and used in the third equation to adjust:

$$F_N = 0{,}75.F_Q.i$$

Some authors use the safety factor k already in the first condition, which is then adjusted as follows:

$$F_T = k.F$$

And then safety coefficient of fasteners is: k = (1,3-1,5).

Gasketed Joints

If a full gasket of area A_g is present in the joint, the gasket pressure p is found by dividing the force in the member by the gasket area per bolt. Thus, for N bolts,

$$p = -F_m / (A_g/N)$$

If N bolts equally share the total external load, then $P = P_{totaly}/N$,

P_{total} - total external tensile load applied to the joint,
P - external tensile load per bolt,

With a load factor n, can be written as

$$F_m = (1 - C).n.P - F_i$$

F_m - resultant load on members,
F_i - preload.
Substituting this into Eq. gives the gasket pressure as

$$p = [F_i - n.P(1 - C)].N/A_g$$

In full-gasketed joints uniformity of pressure on the gasket is important. To maintain adequate uniformity of pressure adjacent bolts should not be placed more than six nominal diameters apart on the bolt circle. To maintain wrench clearance, bolts should be placed at least three diameters apart.

A rough rule for bolt spacing around a bolt circle is

$$3 \leq \pi.D_b/N.d \leq 6$$

where D_b is the diameter of the bolt circle and N is the number of bolts.