Contents

3 Screws

Fasteners may be classified as threaded, fixed, and locking. Threaded fasteners include bolts, studs, and various forms of screws. Fixed fasteners include welds, solders, brazing, adhesives, and rivets. Locking fasteners may be used separately or in conjunction with other fasteners and include washers, keys, splines, springs, and pins.

Threaded fasteners such as screws, nuts, and bolts are important components of mechanical structures and machines. Screws may be used as removable fasteners or as devices for moving loads.

3.1 Screw Thread

A screw thread is a uniform wedge-shaped section in the form of a helix on the external or internal surface of a cylinder (straight thread) or a cone (taper thread).

The basic arrangement of a helical thread wound around a cylinder is illustrated in Fig 3.1. The terminology of an external screw threads is:

- *pitch* denoted by p is the distance, parallel to the screw axis, between corresponding points on adjacent thread forms having uniform spacing.
- *major diameter* denoted by d is the largest (outside) diameter of a screw thread.
- *minor diameter* denoted by d_r or d_1, is the smallest diameter of a screw thread.
- *pitch diameter* denoted by d_m or d_2 is the imaginary diameter for which the width of the threads and the grooves are equal.

The standard geometry of a basic profile of an external thread is shown in Fig. 3.2, and it is basically the same for both Unified (inch series) and ISO (International Standards Organization, metric) threads.

The *lead* denoted by l is the distance the nut moves parallel to the screw axis when the nut is given one turn (distance a threaded section moves axially in one revolution). A screw with two or more threads cut beside each other is called a *multiple-threaded* screw. The lead is equal to twice the pitch for a double-threaded screw, and up to 3 times the pitch for a triple-threaded screw. The pitch p, lead l, and lead angle λ are represented in Fig. 3.3.

Figure 3.3(a) shows a single thread right-hand screw and Fig. 3.3(b) shows a double-threaded left-hand screw. If a thread traverses a path in a clockwise and receding direction when viewed axially, it is a *right-hand thread*. All threads are assumed to be right-hand, unless otherwise specified.

A standard geometry of an ISO profile, M (metric) profile, with 60° symmetric threads is shown in Fig. 3.4. In Fig. 3.4 $D\,(d)$ is the basic major diameter of the internal (external) thread, $D_1\,(d_1)$ is the basic minor diameter of the internal (external) thread, $D_2\,(d_2)$ is the basic pitch diameter, and $H = 0.5\,\sqrt{3}\,p$.

Metric threads are specified by the letter M preceding the nominal major diameter in millimeters and the pitch in millimeters per thread. For example:
M 14 × 2
M is the SI thread designation, 14 mm is the outside (major) diameter, and the pitch is 2 mm per thread.

Screw size in the Unified system is designated by the size number for major diameter (in.), the number of treads per in., and the thread form and series, like this:
$\frac{5''}{8}$ − 18 UNF
$\frac{5''}{8}$ is the the outside (major) diameter where the double tick marks mean inches, and 18 threads per in. Some Unified thread series are:

UNC Unified National Coarse
UNEF Unified National Extra Fine
UNF Unified National Fine
UNS Unified National Special
UNR Unified National Round (round root)

The UNR series threads have improved fatigue strengths.

Figure 3.5(a) shows different types of thread delineation on a drawing: detailed, schematic, and simplified thread representation. The schematic representation is realistic and is used frequently in assembly drawings. The simplified thread representation is used widely because its ease of drawing. Typical screw heads are illustrated in Fig. 3.5(b).

3.2 Power Screws

For applications that require power transmission, the Acme (Fig. 3.6) and square threads (Fig. 3.7) are used.

Power screws are used to convert rotary motion to linear motion of the meeting member along the screw axis. These screws are used to lift weights (screw-type jacks) or exert large forces (presses, tensile testing machines). The power screws can also be used to obtain precise positioning of the axial movement.

A square-threaded power screw with a single thread having the pitch diameter d_m, the pitch p, and the helix angle λ is considered in Fig. 3.8. Consider that a single thread of the screw is unrolled for exactly one turn. The edge of the thread is the hypotenuse of a right triangle and the height is the lead. The base of the right triangle is the circumference of the pitch diameter circle (Fig. 3.9). The lead angle λ is the helix angle of the thread.

The screw is loaded by an axial compressive force F (Figs. 3.8 and 3.9).

The force diagram for lifting the load is shown in Fig. 3.9(a), (the force P_r is positive). The force diagram for lowering the load is shown in Fig. 3.9(b), (the force P_l is negative). The friction force is

$$F_f = \mu\, N,$$

where μ is the coefficient of dry friction and N is the normal force. The friction force is acting opposite to the motion.

The equilibrium of forces for raising the load gives

$$\sum F_x = P_r - N\ \sin\lambda - \mu\ N\ \cos\lambda = 0, \tag{3.1}$$

$$\sum F_y = F + \mu\ N \sin\lambda - N\ \cos\lambda = 0. \tag{3.2}$$

Similarly, for lowering the load one may write the equations

$$\sum F_x = -P_l - N\ \sin\lambda + \mu\ N\ \cos\lambda = 0, \tag{3.3}$$

$$\sum F_y = F - \mu\ N\ \sin\lambda - N\ \cos\lambda = 0. \tag{3.4}$$

Eliminating N and solving for P_r

$$P_r = \frac{F\ (\sin\lambda + \mu\ \cos\lambda)}{\cos\lambda - \mu\ \sin\lambda}, \tag{3.5}$$

and for lowering the load

$$P_l = \frac{F\ (\mu\ \cos\lambda - \sin\lambda)}{\cos\lambda + \mu\ \sin\lambda}. \tag{3.6}$$

Using the relation

$$\tan\lambda = l/(\pi d_m),$$

and dividing the equations by $\cos\lambda$ one may obtain

$$P_r = \frac{F\ [(l\ \pi\ d_m) + \mu]}{1 - (\mu\ l\ \pi\ d_m)}, \tag{3.7}$$

$$P_l = \frac{F\ [\mu - (l\ \pi\ d_m)]}{1 + (\mu\ l\ \pi\ d_m)}. \tag{3.8}$$

The moment required to overcome the thread friction and to raise the load is

$$M_r = P_r \frac{d_m}{2} = \frac{F\ d_m}{2} \left(\frac{l + \pi\ \mu\ d_m}{\pi\ d_m - \mu\ l} \right). \tag{3.9}$$

The moment required to lower the load (and to overcome a part of the friction) is

$$M_l = \frac{F\ d_m}{2} \left(\frac{\pi\ \mu\ d_m - l}{\pi\ d_m + \mu\ l} \right). \tag{3.10}$$

When the lead, l, is large or the friction, μ, is low the load will lower itself. In this case the screw will spin without any external effort, and the moment M_l in Eq. (3.10) will be negative or zero. When the moment is positive, $M_l > 0$, the screw is said to be *self-locking*.
The condition for self-locking is

$$\pi\ \mu\ d_m > l.$$

Dividing both sides of this inequality by $\pi\ d_m$, and using $l/(\pi\, d_m) = \tan\lambda$, yields

$$\mu > \tan\lambda. \tag{3.11}$$

The self-locking is obtained whenever the coefficient of friction is equal to or greater than the tangent of the thread lead angle.

The moment, M_0, required only to raise the load when the friction is zero, $\mu = 0$, is obtained from Eq. (3.9):

$$M_0 = \frac{F\ l}{2\ \pi}. \tag{3.12}$$

The screw efficiency e can be defined as

$$e = \frac{M_0}{M_r} = \frac{F\ l}{2\ \pi\ M_r}. \tag{3.13}$$

For square threads the normal thread load, F, is parallel to the axis of the screw (Figs 3.7 and 3.8). The preceding equations can be applied for square threads.

For Acme threads (Fig. 3.6) or other threads, the normal thread load is inclined to the axis due to the thread angle 2α and the lead angle λ.

The screw threads in normal and axial plane are shown in Fig. 3.10(a). The angle α_n is the thread angle measured in normal plane. The relation between the thread angle measured in axial plane, α in Fig. 3.6, and the thread angle measured in normal plane, α_n in Fig. 3.10(a), is

$$\tan\alpha_n = \frac{s}{H} = \frac{s}{H\ \cos\lambda}\cos\lambda = \tan\alpha\ \cos\lambda,$$

or

$$\tan\alpha_n = \tan\alpha\ \cos\lambda. \tag{3.14}$$

The screw thread forces in normal plane are represented in Fig. 3.10(b). The force diagram for lifting the load is shown in Fig. 3.10(b). The equilibrium of forces for raising the load gives

$$\sum F_x = P_r - N\ \cos\alpha_n\ \sin\lambda - \mu\ N\ \cos\lambda = 0, \tag{3.15}$$

$$\sum F_y = -F + N\ \cos\alpha_n\ \cos\lambda - \mu\ N\ \sin\lambda = 0. \tag{3.16}$$

Eliminating N and solving for P_r,

$$P_r = F\,\frac{\mu\ \cos\lambda + \sin\lambda\ \cos\alpha_n}{\cos\lambda\ \cos\alpha_n - \mu\ \sin\lambda} = F\,\frac{\mu + \tan\lambda\ \cos\alpha_n}{\cos\alpha_n - \mu\ \tan\lambda}. \tag{3.17}$$

The moment required to overcome the thread friction and to raise the load is

$$M_r = P_r\frac{d_m}{2} = \frac{F\ d_m}{2}\left(\frac{\mu + \tan\lambda\ \cos\alpha_n}{\cos\alpha_n - \mu\ \tan\lambda}\right). \tag{3.18}$$

Using the relation

$$\tan\lambda = l/(\pi\ d_m),$$

the following expression is obtained:

$$M_r = \frac{F\ d_m}{2}\left(\frac{\mu\ \pi\ d_m + l\ \cos\alpha_n}{\pi\ d_m\ \cos\alpha_n - \mu\ l}\right). \tag{3.19}$$

Similarly, the moment required to lower the load and to overcome a part of the friction is

$$M_l = \frac{F\ d_m}{2}\left(\frac{\mu\ \pi\ d_m - l\ \cos\alpha_n}{\pi\ d_m\ \cos\alpha_n + \mu\ l}\right). \tag{3.20}$$

For power screws the square thread ($\alpha_n = 0$) is more efficient than the Acme thread. The Acme thread adds an additional friction due to the wedging action. It is easier to machine an Acme thread than a square thread.

In general, when the screw is loaded axially, a thrust bearing or thrust collar may be used between the rotating and stationary links to carry the axial component (Fig. 3.11). The load is concentrated at the mean collar diameter d_c. The moment required is

$$M_c = \frac{F\ \mu_c\ d_c}{2}, \tag{3.21}$$

where μ_c is the coefficient of collar friction.

3.3 Force Analysis for a Square-Threaded Screw

Consider a square-threaded jack under the action of a axial load F and a moment M about the axis of the screw, Fig. 3.12(a). The screw has the mean radius r_m and the lead l. The force exerted by the frame thread on the screw thread is R. The angle θ made by R with the normal to the thread is the angle of friction [Fig. 3.12(b)]:

$$\tan\theta = \mu = \frac{F_f}{N}.$$

The unwrapped thread of the screw shown in Fig. 3.12(b) is for lifting the load. The force equilibrium equation in the axial direction is

$$F = R\ \cos(\lambda + \theta),$$

where λ is the helix angle, $\tan \lambda = l/(2\ \pi\ r_m)$. The moment of R about the vertical axis of the screw is $R\ r_m\ \sin(\lambda + \theta)$. The moment equilibrium equation for the screw becomes

$$M = R\ r_m\ \sin(\lambda + \theta).$$

Combining the expression for F and M gives

$$M = M_r = F\ r_m\ \tan(\lambda + \theta). \tag{3.22}$$

The force required to push the thread up is $P = M/r_m$.
The moment required to lower the load by unwinding the screw is obtained in a similar manner:

$$M = M_l = F\ r_m\ \tan(\theta - \lambda). \tag{3.23}$$

If $\theta < \lambda$ the screw will unwind by itself.

3.4 Threaded Fasteners

In general bolts are used to hold parts together. External forces tend to pull, or slide, the parts apart. Figure 3.13(a) shows two parts connected with a bolt. An external force, F_e, acts on the the joint and tends to separate the two parts. The free-body diagram of a portion of this joint without the external load is shown in Fig. 3.13(b). In this figure the nut has been initially tightened to a preload force F_i. The initial bolt axial load F_{b0} and the clamping force between the two plates F_{c0} are both equal to the preload force F_i: $F_{b0} = F_{c0} = F_i$. The free-body diagram of the portion with the external load F_e is shown in Fig. 3.13(c). Equilibrium requires an increase in F_b and a decrease in F_c. The separating force F_e must be equal to the sum of the increased bolt force ΔF_b plus the decreased clamping force ΔF_c:

$$F_e = \Delta F_b + \Delta F_c. \tag{3.24}$$

The bolt and the clamped members elongate the same amount:

$$\delta = \frac{\Delta F_b}{k_b} = \frac{\Delta F_c}{k_c}, \tag{3.25}$$

where k_b and k_c are the spring constants for the bolt and clamped parts, respectively. From Eqs. (3.24) and (3.25) the elongation is

$$\delta = \frac{F_e}{k_b + k_c}. \tag{3.26}$$

The bolt axial load F_b and the clamping force F_c are

$$F_b = F_i + \Delta F_b = F_i + \frac{k_b}{k_b + k_c} F_e, \tag{3.27}$$

$$F_c = F_i - \Delta F_c = F_i - \frac{k_c}{k_b + k_c} F_e. \tag{3.28}$$

The *joint constant* is defined as a dimensionless stiffness parameter given by

$$C = \frac{k_b}{k_b + k_c}. \tag{3.29}$$

Equations (3.27) and (3.28) will become

$$F_b = F_i + C\, F_e, \tag{3.30}$$

$$F_c = F_i - (1 - C)\, F_e. \tag{3.31}$$

The general axial deflection equation

$$\delta = \frac{F\ l}{A\ E},$$

gives the spring constant (stiffness)

$$k = \frac{F}{\delta} = \frac{A\ E}{l},$$

where A is the cross-sectional area, E is the modulus of elasticity, and l is the length.

Bolt Stiffness

A bolt with thread is considered as a shaft with a variable section. The minor diameter (root diameter), d_r, is used for the threaded section of the bolt, and the major diameter (crest diameter), d, is used for the unthreaded section of the bolt (shank). The stiffness of the bolt is

$$\frac{1}{k_b} = \frac{1}{k_{\text{thread}}} + \frac{1}{k_{\text{shank}}}.$$

For a bolt with a shank having a constant major diameter the spring constant is [6]:

$$\frac{1}{k_b} = \frac{4}{\pi\ E} \left(\frac{l_{se}}{d^2} + \frac{l_{te}}{d_r^2} \right) = \frac{4}{\pi\ E} \left(\frac{l_s + 0.4d}{d^2} + \frac{l_t + 0.4d_r}{d_r^2} \right), \tag{3.32}$$

where l_s is the length of the unthreaded section and l_t is the length of the threaded section (Fig. 3.14). The effective lengths of the unthreaded and threaded sections are l_{se} and l_{te}, respectively. The modulus of elasticity of the bolt is E.

Shigley and Mischke [20] proposed the following expressions for the stiffness of the unthreaded section of the bolt, k_s:

$$k_s = \frac{A_s\, E}{l_s} = \frac{\pi\, d^2\, E}{4 l_s}, \tag{3.33}$$

and for the stiffness of the threaded section of the bolt, k_t

$$k_t = \frac{A_t\, E}{l_t}. \tag{3.34}$$

The *tensile strength area*, A_t is defined as [6]:

$$A_t = 0.7854\, (d - 0.9743/n)^2 \quad \text{in.}^2 \tag{3.35}$$

for UN thread profiles where d is in inches and n is the number of threads per in.

$$A_t = 0.7854(d - 0.9382p)^2 \quad \text{mm}^2, \tag{3.36}$$

for M thread profiles with the major diameter d and the pitch p in millimeters. The tensile strength area is also given in Tables 3.1 and 3.2 [6].

Thus, the bolt stiffness is

$$k_b = \frac{A_s\, A_t\, E}{A_s\, l_t + A_t\, l_s}. \tag{3.37}$$

Stiffness of the clamped parts

Difficulties commonly arise in estimating the stiffness of the clamped parts or the joint stiffness. The clamped parts may consist of a combination of different materials. The parts may represent "springs" in series, as shown in Fig. 3.14, and the stiffness of the clamped parts is

$$\frac{1}{k_c} = \frac{1}{k_1} + \frac{1}{k_2} + ... + \frac{1}{k_i} + ...$$

Shigley and Mischke [20] proposed the following expression for the joint stiffness:

$$k_i = \frac{0.577\ \pi\ E_i\ d}{2 \ln\left(5 \dfrac{0.577\ l_i + 0.5\ d}{0.577\ l_i + 2.5\ d}\right)} \tag{3.38}$$

Wileman et al. [25] obtained an exponential expression using finite element analysis:

$$k_i = E_i \; d \; A \; e^{B \; d/l_i}, \tag{3.39}$$

the numerical constants are:
$A = 0.78715$, $B = 0.62873$ for steel;
$A = 0.79670$, $B = 0.63816$ for aluminun;
$A = 0.79568$, $B = 0.63553$ for copper, and
$A = 0.77871$, $B = 0.61616$ for gray cast iron.

Bolt Preload

The initial tensile force F_i is defined as [7]:

$$F_i = K \; A_t \; S_p, \tag{3.40}$$

where A_t is the tensile stress area of the thread and S_p is the *proof strength* of the material [7, 20]. The proof strength of steel bolts is given in Tables 3.3 and 3.4 for various sizes [6]. The International Organization for Standardization (ISO) defines a metric grade number as a range of 4.6 to 12.9 (Table 3.3) and the Society of Automotive Engineers (SAE) specifies grade number from 1 to 8. The higher grade numbers represent greater strength. The bolt grades are numbered according to the tensile strength. The constant K is 0.75 for reused connections and 0.90 for permanent connections. The *proof load* is defined as $F_p = A_t \, S_p$ and is the maximum load that a bolt can withstand without acquiring a permanent set.

Static Loading of the Joint

The bolt stress can be calculated from Eq. (3.30):

$$\sigma_b = \frac{F_b}{A_t} = \frac{F_i}{A_t} + C \frac{F_e}{A_t}, \tag{3.41}$$

where A_t is the tensile stress. The limiting value for the bolt stress, σ_b, represents the proof strength, S_p. A safety factor n_b is introduced for the bolt stress and Eq. (3.41) becomes

$$S_p = \frac{F_i}{A_t} + C \frac{F_{max} \, n_b}{A_t}. \tag{3.42}$$

The safety factor is not applied to the preload stress F_i/A_t. The bolt failure safety factor is

$$n_{bf} = \frac{S_p\, A_t - F_i}{C\, F_{max,b}}, \tag{3.43}$$

where $F_{max,b}$ is the maximum external load applied to the bolt.

Separation occurs when the clamping force is zero, $F_c = 0$. The safety factor against separation of the parts of the joint is obtained from Eq. (3.31) with $F_c = 0$ and has the expression

$$n_s = \frac{F_i}{F_{max}(1 - C)}, \tag{3.44}$$

where F_{max} is the maximum external load applied to joint.

3.5 Examples

Example 3.1: Double square-thread power screw.
A double square-thread power screw (Fig. 3.15) has the major diameter $d = 40$ mm and the pitch $p = 6$ mm. The coefficient of friction of the thread is $\mu = 0.08$ and the coefficient of collar friction is $\mu_c = 0.1$. The mean collar diameter is $d_c = 45$ mm. The external load on the screw is $F = 8$ kN.

Find:
a) the lead, the pitch (mean) diameter and the minor diameter;
b) the moment required to raise the load;
c) the moment required to lower the load;
d) the efficiency of the device.

Solution

a) From Fig. 3.7:
the minor diameter is

$$d_r = d - p = 40 - 6 = 34 \text{ mm},$$

the pitch (mean) diameter is

$$d_m = d - p/2 = 40 - 3 = 37 \text{ mm}.$$

The lead is

$$l = 2\,p = 2\,(6) = 12 \text{ mm}.$$

b) The moment required to raise the load is [Eqs. (3.9) and (3.21]

$$
\begin{aligned}
M_r &= \frac{Fd_m}{2}\left(\frac{l+\pi\mu d_m}{\pi d_m-\mu l}\right)+\frac{F\mu_c d_c}{2}\\
&= \frac{8\,(10^3)\,(37)\,(10^{-3})}{2}\left[\frac{12+0.08(37)\pi}{37\pi-0.08\,(12)}\right]+\frac{8\,(10^3)\,(0.1)\,(45)\,(10^{-3})}{2}\\
&= 45.344 \text{ N m.}
\end{aligned}
$$

c) The moment required to lower the load is [Eqs. (3.10) and (3.21)]:

$$
\begin{aligned}
M_l &= \frac{Fd_m}{2}\left(\frac{\pi\mu d_m-l}{\pi d_m+\mu l}\right)+\frac{F\mu_c d_c}{2}\\
&= \frac{8\,(10^3)\,(37)\,(10^{-3})}{2}\left[\frac{0.08(37)\pi-12}{37\pi+0.08\,(12)}\right]+\frac{8\,(10^3)\,(0.1)\,(45)\,(10^{-3})}{2}\\
&= 14.589 \text{ N m.}
\end{aligned}
$$

The screw is not self-locking: $\pi\mu d_m - l = 0.08(37)\pi - 12 = -2.700 \, < \, 0.$
d) The overall efficiency is [Eq. (3.13)]:

$$
e = \frac{Fl}{2\pi M_r} = \frac{8\,(10^3)\,(12)(10^{-3})}{2(45.344)\pi} = 0.336.
$$

The *Mathematica*™ program for this example is given in Program 3.1.

Example 3.2. Acme-thread power screw.
A double-thread Acme screw is used in a jack to raise a load of 2000 lb (Fig. 3.16). The major diameter of the screw is $d = 2$ in. A plain thrust collar is used. The mean diameter of the collar is $d_c = 3$ in. The coefficient of friction of the thread is $\mu = 0.12$ and the coefficient of collar friction is $\mu_c = 0.09$.

Determine:

a) the screw pitch, lead, thread depth, mean pitch diameter, and helix angle;

b) the starting moment for raising and for lowering the load;

c) the efficiency of the jack.

Solution.

a) The preferred pitches for Acme threads are [20]:

d [in.]	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3
p [in.]	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

For the major diameter $d = 2$ in. the preferred screw pitch is $p = 0.25$ in. Because of the double thread the lead is

$$l = 2\,p = 2\,(0.25) = 0.5\,\text{in.}$$

The pitch (mean) diameter is (Fig. 3.6):

$$d_m = d - p/2 = 2 - 0.25/2 = 1.875\,\text{in.}$$

The helix angle is

$$\lambda = \tan^{-1}\frac{l}{\pi d_m} = \tan^{-1}\frac{0.5}{1.875\pi} = 4.851^\circ.$$

b) The starting friction is about one-third higher than running friction and the coefficients of starting friction are [7]

$$\mu_s = \frac{4}{3}\mu = \frac{4}{3}\,(0.12) = 0.16 \quad \text{and} \quad \mu_{sc} = \frac{4}{3}\mu_c = \frac{4}{3}\,(0.09) = 0.12.$$

The angle α_n is calculated with the formula

$$\alpha_n = \tan^{-1}(\tan\alpha\cos\lambda) = \tan^{-1}(\tan 14.5^\circ \cos 4.851^\circ) = 14.450^\circ,$$

where $\alpha = 14.5^\circ$ (Fig. 3.6). The moment for lifting the load is

$$M_{rs} = \frac{F d_m}{2}\left(\frac{\mu_s \pi d_m + l\cos\alpha_n}{\pi d_m \cos\alpha_n - \mu_s l}\right) + \frac{F\mu_{cs} d_c}{2} =$$
$$\frac{2000(1.875)}{2}\left(\frac{0.16\pi(1.875) + 0.5\cos 14.450^\circ}{\pi(1.875)\cos 14.450^\circ - (0.16)(0.5)}\right) + \frac{2000(0.12)(3)}{2}$$
$$= 835.626 \text{ lb in.}$$

Similarly, the moment required to lower the load and to overcome a part of the friction is

$$M_{ls} = \frac{F d_m}{2}\left(\frac{\mu_s \pi d_m - l\cos\alpha_n}{\pi d_m \cos\alpha_n + \mu_s l}\right) + \frac{F\mu_{cs} d_c}{2} =$$
$$\frac{2000(1.875)}{2}\left(\frac{0.16\pi(1.875) - 0.5\cos 14.450^\circ}{\pi(1.875)\cos 14.450^\circ + (0.16)(0.5)}\right) + \frac{2000(0.12)(3)}{2}$$
$$= 508.562 \text{ lb in.}$$

c) Changing the coefficient of friction to the running values of μ and μ_c, the moment for lifting the load is

$$M_r = \frac{Fd_m}{2}\left(\frac{\mu\pi d_m + l\cos\alpha_n}{\pi d_m\cos\alpha_n - \mu l}\right) + \frac{F\mu_c d_c}{2} =$$
$$\frac{2000(1.875)}{2}\left(\frac{0.12\pi(1.875) + 0.5\cos 14.450^\circ}{\pi(1.875)\cos 14.450^\circ - (0.12)(0.5)}\right) + \frac{2000(0.09)(3)}{2}$$
$$= 665.667 \text{ lb in.}$$

With both friction coefficients zero, the moment to raise the load is

$$M_0 = \frac{Fd_m}{2}\left(\frac{l\cos\alpha_n}{\pi d_m\cos\alpha_n}\right) = \frac{Fl}{2\pi} = \frac{2000(0.5)}{2\pi} = 159.155 \text{ lb in.}$$

d) The efficiency is the ratio of friction-free moment to actual moment, or

$$e = \frac{Fl}{2\pi M_r} = \frac{M_0}{M_r} = \frac{159.155}{665.667} = 23.909\,\%.$$

If the collar friction is neglected ($\mu_c = 0$), the efficiency

$$e = \frac{\cos\alpha_n - \mu\tan\lambda}{\cos\alpha_n + \mu\cot\lambda},$$

function of λ is plotted in Fig. 3.17 where $\alpha_n = \tan^{-1}(\tan 14.5^\circ\cos\lambda)$ and $\mu = \{0.05;\ 0.12;\ 0.15\}$.
The *Mathematica*$^{\text{TM}}$ program for this example is given in Program 3.2.

Example 3.3. A hexagonal bolt made of steel with the modulus of elasticity $E_b = 206.8$ GPa is used to join two parts made of cast iron. The modulus of elasticity for gray cast iron is $E_c = 100.0$ GPa. The thread major diameter is $d = 14$ mm, the root (minor) diameter is $d_r = 12$ mm, and the pitch is $p = 2$ mm. The bolt has the axial length $l = 50$ mm and the threaded portion of the bolt is $l/2$. The cast iron parts have the same axial length, $l/2$. Determine:
a) the stiffness of the bolt using Eq. (3.32);
b) the stiffness of the clamped parts using Eq. (3.38);
c) the stiffness of the clamped parts using Eq. (3.39).
Solution.

a) The lengths of the threaded section and unthreaded section of the bolt are $l_t = l_s = l/2 = 25$ mm. The effective length of the threaded bolt is

$$l_{te} = l_t + 0.4\ d_r = 25 + 0.4\ (12) = 29.8 \text{ mm} = 0.0298 \text{ m}.$$

The effective length of the unthreaded bolt is

$$l_{se} = l_s + 0.4\ d = 25 + 0.4\ (14) = 30.6 \text{ mm} = 0.0306 \text{ m}.$$

The minor diameter area is

$$A_r = \pi\ d_r^2/4 = \pi(0.012)^2/4 = 113.097 \times 10^{-6} \text{ m}^2.$$

The major diameter area is

$$A_s = \pi\ d^2/4 = \pi(0.014)^2/4 = 153.938 \times 10^{6} \text{ m}^2.$$

The stiffness of the threaded portion is

$$k_t = \frac{A_r E_b}{l_{te}} = \frac{(113.097 \times 10^{-6})\ (206.8 \times 10^9)}{0.0298} = 7.8485 \times 10^8 \text{ N/m}.$$

The stiffness of the unthreaded portion is

$$k_s = \frac{A_s E_b}{l_{se}} = \frac{(153.938 \times 10^{-6})\ (206.8 \times 10^9)}{0.0306} = 1.04034 \times 10^9 \text{ N/m}.$$

The bolt stiffness is

$$k_b = \frac{k_t k_s}{k_t + k_s} = \frac{(7.8485 \times 10^8)\ (1.04034 \times 10^9)}{7.8485 \times 10^8 + 1.04034 \times 10^9} = 4.47357 \times 10^8 \text{ N/m}.$$

b) Shigley and Mischke [20] proposed the following expression for the stiffness of the clamped parts [see Eq. (3.38)]:

$$\begin{aligned} k_c &= \frac{0.577\ \pi\ E_c\ d}{2\ln\left(5\dfrac{0.577\ l + 0.5\ d}{0.577\ l + 2.5\ d}\right)} = \frac{0.577\ \pi\ (100.0 \times 10^9)\ (0.014)}{2\ln\left(5\dfrac{0.577\ (0.05) + 0.5\ (0.014)}{0.577\ (0.05) + 2.5\ (0.014)}\right)} \\ &= 1.22925 \times 10^9 \text{ N/m}. \end{aligned}$$

c) Wileman et al. [25] proposed the expression for the stiffness of the clamped parts [Eq. (3.39)] with the numerical constants $A = 0.77871$ and $B = 0.61616$ for gray cast iron:

$$\begin{aligned} k_c &= E_c\ d\ A\ e^{Bd/l} = (100.0 \times 10^9)\ (0.014)\ (0.77871)\ e^{0.61616\ (0.014)/0.05} \\ &= 1.29548 \times 10^9 \text{ N/m}. \end{aligned}$$

The *Mathematica*™ program for this example is given in Program 3.3.

Example 3.4. A hexagonal bolt and nut assembly is used to join two parts. The bolt and nut are made of steel (modulus of elasticity $E_b = 30$ Mpsi for steel). One part is made of steel and the other part is made of cast iron (modulus of elasticity $E_c = 12$ Mpsi for cast iron). The thread major diameter is $d = 5/8$ in., the root (minor) diameter is $d_r = 0.5135$ in. There are 11 threads per inch, hence the pitch is $p = 1/11$ in. The assembly and the bolt have the axial length $l = 1.5$ in. The length of the threaded portion of the bolt is $l/2$. The axial length of the cast iron part is $l/2$.

Determine:

a) the stiffness of the bolt using Eq. (3.37);

b) the stiffness of the clamped parts using Eq. (3.38);

c) the stiffness of the clamped parts using Eq. (3.39).

Solution.

a) The lengths of the threaded section and unthreaded section of the bolt are $l_t = l_s = l/2 = 0.75$ in. The major diameter area is

$$A_s = \pi\ d^2/4 = \pi(5/8)^2/4 = 0.306 \text{ in.}^2$$

The tensile strength area is given by Eq. (3.36):

$$A_t = 0.7854\ (d - 0.9743\ p)^2 = 0.7854\ \left(\frac{5}{8} - 0.9743\ \frac{1}{11}\right)^2 = 0.226 \text{ in.}^2$$

The stiffness of the unthreaded portion is

$$k_s = A_s\ E_b/l_s = 0.306\ (30)/0.75 = 12.271 \text{ Mlb/in.}$$

The stiffness of the threaded portion is

$$k_t = A_t\ E_b/l_t = 0.226\ (30)/0.75 = 9.040 \text{ Mlb/in.}$$

The bolt stiffness is

$$k_b = \frac{k_s\ k_t}{k_s + k_t} = \frac{2.271\ (9.040)}{2.271 + 9.040} = 5.205 \text{ Mlb/in.}$$

b) The axial length of the cast iron clamped part and steel clamped part are $l_1 = l_2 = l/2 = 0.75$ in. Using Eq. (3.38), the stiffness of the cast iron clamped part is

$$\begin{aligned} k_1 &= \frac{0.577\ \pi\ E_c\ d}{2 \ln \left(5 \dfrac{0.577\ l_1 + 0.5\ d}{0.577\ l_1 + 2.5\ d} \right)} = \frac{0.577\ \pi\ (12)\ (5/8)}{2 \ln \left[5 \dfrac{0.577\ (0.75) + 0.5\ (5/8)}{0.577\ (0.75) + 2.5\ (5/8)} \right]} \\ &= 10.882 \text{ Mlb/in.} \end{aligned}$$

and the stiffness of the steel clamped part is

$$\begin{aligned} k_2 &= \frac{0.577\ \pi\ E_b\ d}{2 \ln \left(5 \dfrac{0.577\ l_2 + 0.5\ d}{0.577\ l_2 + 2.5\ d} \right)} = \frac{0.577\ \pi\ (30)\ (5/8)}{2 \ln \left[5 \dfrac{0.577\ (0.75) + 0.5\ (5/8)}{0.577\ (0.75) + 2.5\ (5/8)} \right]} \\ &= 27.206 \text{ Mlb/in.} \end{aligned}$$

The resulting stiffness of the clamped parts is

$$k_c = \frac{k_1\ k_2}{k_1 + k_2} = \frac{10.882\ (27.206)}{10.882 + 27.206} = 7.773 \text{ Mlb/in.}$$

c) Using Eq. (3.39), the stiffness of the cast iron clamped part is ($A = 0.77871$ and $B = 0.61616$ for gray cast iron)

$$\begin{aligned} k_1 &= E_c\ d\ A\ e^{Bd/l_1} = 12\ (5/8)\ (0.77871)\ e^{0.61616\ (5/8)/0.75} \\ &= 9.759 \text{ Mlb/in.} \end{aligned}$$

and the stiffness of the steel clamped part is ($A = 0.78715$ and $B = 0.62873$ for steel)

$$\begin{aligned} k_2 &= E_b\ d\ A\ e^{Bd/l_2} = 30\ (5/8)\ (0.78715)\ e^{0.62873\ (5/8)/0.75} \\ &= 24.923 \text{ Mlb/in.} \end{aligned}$$

The resulting stiffness of the clamped parts is

$$k_c = \frac{k_1\ k_2}{k_1 + k_2} = \frac{9.759\ (24.923)}{9.759 + 24.923} = 7.013 \text{ Mlb/in.}$$

The *Mathematica*™ program for this example is given in Program 3.4.

Example 3.5. A bolt made from cold-drawn steel with the stiffness k_b is used to clamp two steel plates with the stiffness k_c. The elasticities are such that $k_c = 5\,k_b$. The plates and the bolt have the same length. The external joint separating force fluctuates continuously between 0 and 6 000 lb. Determine: a) the minimum required value of initial preload to prevent loss of compression of the plates. b) the minimum force in the plates for fluctuating load, if the preload is 6 500 lb.

Solution.

a) Compression of the plates is lost when $F_c = 0$ when maximum load is applied. Equation (3.28) becomes

$$F_i = F_c + F_e \frac{k_c}{k_b + k_c} = 0 + 6\,000\ \frac{5}{1+5} = 5\,000 \text{ lb.}$$

b) Minimum force in plates occurs when fluctuating load is maximum. From Eq. (3.28) with $F_i = 6\,500$ lb, it results

$$F_c = F_i - F_e \frac{k_c}{k_b + k_c} = 6\,500 - 6\,000\ \frac{5}{1+5} = 1\,500 \text{ lb.}$$

Example 3.6. A bolt and nut assembly is used to join two parts made of cast iron. The bolt has the thread major diameter $d = 10$ mm, the pitch $p = 1.5$ mm, and a 4.8 grade. The clamped plates have a stiffness k_c six times the bolt stiffness k_b. The assembly and the bolt have the axial length and the cast iron parts have the same axial length. Determine the maximum load for bolt and joint failure assuming a reused connection and a static safety factor of $n_f = 2$.

Solution.

The joint constant is

$$C = \frac{k_b}{k_b + k_c} = \frac{k_b}{k_b + 6\ k_b} = \frac{1}{7}.$$

The tensile strength area is

$$A_t = 0.7854\ (d - 0.9382\ p)^2 = 0.7854\ [10 - 0.9382\ (1.5)]^2 = 57.989 \text{ mm}^2.$$

From Table 3.3, for a 4.8 grade the proof strength is $S_p = 310$ MPa. The proof load is

$$F_p = A_t\ S_p = (57.989 \times 10^{-6})(310 \times 10^6) = 17\,976.8 \text{ N.}$$

The preload for reused connections is

$$F_i = 0.75\ F_p = 0.75\ (17976.8) = 13\,482.6\ \text{N}.$$

The maximum external load applied to the bolt is

$$F_{max,b} = \frac{S_p\,A_t - F_i}{n_f\,C} = \frac{(310 \times 10^6)(57.989 \times 10^{-6}) - 13\,482.6}{2\,(1/7)} = 15\,729.7\ \text{N}.$$

Equation (3.44) gives the maximum external load applied to the joint before separation as

$$F_{max} = \frac{F_i}{n_f\,(1 - C)} = \frac{13\,482.6}{2\,[1 - (1/7)]} = 7\,864.84\ \text{N}.$$

Example 3.7. A number of N identical bolts, 1 in. $-$ 8 UNC grade 5, are used to join two members. The joint constant is $C = 0.5$ and the separating force is 60 kip. Assume that the bolts may be reused when the joint is taken apart. Find the number of bolts (N) for a design safety factor of 2.

Solution.

From Table 3.2 for $d = 1$ in. and $n = 8$, the tensile strength area is $A_t = 0.606$ in.^{2} From Table 3.4 for grade 5, the proof strength is $S_p = 85$ kpsi. The recommended preload for reused connections is

$$F_i = 0.75\ A_t\ S_p = 0.75\ (0.606)\ (85) = 38.632\ \text{kip}.$$

For N bolts Eq. (3.43) can be written as

$$n_f = \frac{S_p\,A_t - F_i}{C\,(F_{max,b}/N)}, \tag{3.45}$$

or

$$N = \frac{C\,n_f\,F_{max,b}}{S_p\,A_t - F_i} = \frac{0.5\,(2)\,(60)}{(85)\,(0.606) - 38.632} = 4.659.$$

Five bolts are selected. Using Eq. (3.45) with $N = 5$ the safety factor is

$$n_{fc} = \frac{S_p\,A_t - F_i}{C\,(F_{max,b}/N)} = \frac{(85)\,(0.606) - 38.632}{0.5\,(60/5)} = 2.146,$$

which is greater than the required safety factor of 2; therefore five bolts will be used for the recommended preload in tightening.

Example 3.8. The support block of a machine is attached to the ground with two screws. The machine applies a tensile static load of 10 kN to the block. a) Select appropriate metric screws of class 5.8 for the block attachment. b) Find appropriate tightening moment. Use a safety factor of 4 based on proof strength.

Solution.

a) The load of 10 kN is applied equally by each screw and the bolt load is axial tension. The nominal load for each of the two bolts is $F = 5$ kN. With a safety factor of $n_b = 4$, the design overload for each bolt is $n_b F = 4\,(5) = 20$ kN. For static loading of a ductile material the stress equation is $\sigma = P/A$. When P is equal to the design overload, σ is equal to the proof strength and

$$S_p = \frac{n_b F}{A_t}. \tag{3.46}$$

For class 5.8 a proof strength of $S_p = 380$ MPa is selected from Table 3.3. Equation (3.46) gives the tensile stress area

$$A_t = \frac{n_b F}{S_p} = \frac{20\,000}{380 \times 10^6} = 52.631 \times 10^{-6}\ \text{m}^2 = 52.631\ \text{mm}^2.$$

From Table 3.1 an appropriate standard size is M 10×1.5 ($A_t = 58.0$ mm^2).

b) The initial tightening tension F_i is defined as

$$F_i = K\,A_t\,S_p = 0.9\,(58.01 \times 10^{-6})\,(380 \times 10^6) = 19\,836\ \text{N},$$

where the constant K is 0.90 for permanent connections. Juvinall and Marshek give an estimated tightening moment for standard screw threads [7] as

$$T = 0.2\,F_i\,d = 0.2\,(19\,836)\,(10 \times 10^{-3}) = 39.672\ \text{Nm}. \tag{3.47}$$

3.6 Problems

3.1 The double square-threaded screw has the major diameter $d = 1$ in. and the pitch $p = 0.2$ in. The coefficient of friction in the threads is 0.15. A moment $M = 60$ lb-in. is applied about the axis of the screw, Fig. 3.18. Find the axial force required to advance the screw: a) to the right, and b) to the left.

3.2 A double square-thread power screw has a pitch (mean) diameter of 30 mm and a pitch of 4 mm (Fig. 3.15). The coefficient of friction of the thread is 0.08 and the coefficient of collar friction is also 0.08. The mean collar diameter is 40 mm. The external load on the screw is 6.4 kN. Determine the moment required to lower the load and the overall efficiency.

3.3 A power screw has a double square thread with a mean diameter of 40 mm and a pitch of 12 mm. The coefficient of friction in the thread is 0.15. Determine if the the screw is self-locking.

3.4 The single-threaded screw of a vise has a mean diameter of 1 in. and has 5 square threads per in. The coefficient of static friction in the threads is 0.20. Determine the helix angle and the fiction angle for the thread.

3.5 A triple-thread Acme screw is used in a jack (as shown in Fig. 3.16) to raise a load of 4000 lb. The major diameter of the screw is 3 in. A plain thrust collar is used. The mean diameter of the collar is 4 in. The coefficient of friction of the thread is 0.08 and the coefficient of collar friction is 0.1. Determine: a) the screw pitch, lead, thread depth, mean pitch diameter, and helix angle; b) the starting moment for raising and for lowering the load; c) the efficiency of the jack.

3.6 A C-clamp develops a 250 lb clamping force (Fig. 3.19). The clamp uses a 1/2 in. Acme single thread. The collar of the clamp has a mean diameter of 5/8 in. The coefficients of running friction are estimated as 0.1 for both the collar and the screw. Estimate the force required at the end of a 6 in. handle.

3.7 A bolt made of steel is used to join two parts made of cast iron. The thread major diameter is $d = 14$ mm, the root (minor) diameter is

$d_r = 12$ mm, and the pitch is $p = 2$ mm. The bolt has the axial length $l = 60$ mm and the threaded portion of the bolt is $l/3$. The cast iron parts have the same axial length $l/2$. Determine: a) the stiffness of the bolt using Eq. (3.32); b) the stiffness of the clamped parts using Eq. (3.38); c) the stiffness of the clamped parts using Eq. (3.39).

3.8 A hexagonal bolt and nut assembly is used to join two parts, as illustrated in Fig. 3.20. The bolt and nut are made of steel. One part is made of steel and the other part is made of cast iron. The thread major diameter is $d = 5/8$ in., the root (minor) diameter is $d_r = 0.5135$ in., and there are $n = 11$ threads per in. The assembly and the bolt have the axial length $l = 1.8$ in. The length of the threaded portion of the bolt is $l/3$. The axial length of the cast iron part is $l/3$. Determine: a) the stiffness of the bolt using Eq. (3.37); b) the stiffness of the clamped parts using Eq. (3.38); c) the stiffness of the clamped parts using Eq. (3.39).

3.9 A bolt M 12×1.25 ISO grade 5.8 made of steel is used to join two parts made of cast iron. The assembly and the bolt have the axial length $l = 80$ mm and the threaded portion of the bolt is $l/4$. The cast iron parts have the same axial length $l/2$. The external joint separating force fluctuates continuously between 0 and 20 kN. Determine the minimum required value of initial preload to prevent loss of compression of the parts.

3.10 A bolt M 10×1 ISO grade 4.8 made of steel is used to join two plates made of cast iron and steel. The bolt and the assembly have the axial length $l = 60$ mm and the threaded portion of the bolt is $l/2$. The cast iron plate has the axial length $l/4$. The external joint separating force fluctuates continuously between 0 and 20 kN. The bolt is tightened to an initial tension of 5 kN. Determine the minimum force in the plates.

3.11 A bolt made from steel has the stiffness k_b. Two steel plates are held together by the bolt and have a stiffness k_c. The elasticities are such that $k_c = 7\,k_b$. The plates and the bolt have the same length. The external joint separating force fluctuates continuously between 0 and 2500 lb. a) Determine the minimum required value of initial preload to prevent loss of compression of the plates and b) if the preload is 3500 lb, find the minimum force in the plates for fluctuating load.

3.12 Repeat the previous problem, except that the external joint separating force varies between 0 and 8500 lb.

3.13 A bolt and nut assembly is used to join two parts made of cast iron. The bolt has the thread (3/4)”-16 UNF, SAE grade 5. The clamped plates have a stiffness k_c six times the bolt stiffness k_b. The assembly and the bolt have the axial length and the cast iron parts have the same axial length. Determine the maximum load for bolt and joint failure assuming a reused connection and a static safety factor of $n_f = 2$.

3.14 A number of N identical bolts, M 10×1.5, ISO grade 4.6, are used to join two members. The joint constant is $C = 0.45$ and the separating force is 6 kN. Assume that the bolts may be reused when the joint is taken apart. Find the number of bolts (N) for a design safety factor of 2.

3.15 Figure 3.21 shows the connection of a cylinder head to a pressure vessel using 10 identical steel bolts, M 16×2, ISO grade 8.8. The parts are made of steel. All the dimensions are illustrated in Fig. 3.21 and are all in mm. The static pressure inside the pressure vessel is 5.5 MPa. Determine the load safety factor.

3.16 Repeat the previous problem for the pressure vessel made of cast iron and the cover plate made of aluminum (for aluminum, $E = 70$ GPa).

3.17 A rotating shaft applies a load of 20 kN on a block. Select the appropriate size for the two screws of class 4.8 for the block attachment and find the tightening moment.

3.18 Figure 3.22 shows an M 8×1.25, ISO grade 4.6, steel bolt tightened to its full proof load. The bolt is loaded in double shear (the bolt has two shear planes). The clamped plates are made of steel with the coefficient of friction approximately 0.3. Determine the force F the joint is capable to withstand.

3.7 Programs

Program 3.1.

Program 3.2.

Program 3.3.

Program 3.4.

References

[1] M. Atanasiu, *Mechanics* [Mecanica], EDP, Bucharest, 1973.

[2] A. Bedford, and W. Fowler, *Dynamics*, Addison Wesley, Menlo Park, CA, 1999.

[3] A. Bedford, and W. Fowler, *Statics*, Addison Wesley, Menlo Park, CA, 1999.

[4] A. Ertas, J. C. Jones, *The Engineering Design Process*, John Wiley & Sons, New York, 1996.

[5] A. S. Hall, A. R. Holowenko, and H. G. Laughlin, *Theory and Problems of Machine Design*, Schaum's Outline Series, McGraw-Hill, New York, 1961.

[6] B. G. Hamrock, B. Jacobson, and S. R. Schmid, *Fundamentals of Machine Elements*, McGraw-Hill, New York, 1999.

[7] R. C. Juvinall and K. M. Marshek, *Fundamentals of Machine Component Design*, 3rd ed., John Wiley & Sons, New York, 2000.

[8] D. B. Marghitu, *Mechanical Engineer's Handbook*, Academic Press, San Diego, CA, 2001.

[9] D. B. Marghitu, M. J. Crocker, *Analytical Elements of Mechanisms*, Cambridge University Press, Cambridge, 2001.

[10] D. B. Marghitu and E. D. Stoenescu, *Kinematics and Dynamics of Machines and Machine Design*, class notes, available at www.eng.auburn.edu/users/marghitu/, 2004.

[11] C. R. Mischke, "Prediction of Stochastic Endurance Strength," *Transaction of ASME, Journal Vibration, Acoustics, Stress, and Reliability in Design*, Vol. 109 (1), pp. 113-122, 1987.

[12] R. L. Mott, *Machine Elements in Mechanical Design*, Prentice Hall, Upper Saddle River, NJ, 1999.

[13] W. A. Nash, *Strength of Materials*, Schaum's Outline Series, McGraw-Hill, New York, 1972.

[14] R. L. Norton, *Machine Design*, Prentice-Hall, Upper Saddle River, NJ, 1996.

[15] R. L. Norton, *Design of Machinery*, McGraw-Hill, New York, 1999.

[16] W. C. Orthwein, *Machine Component Design*, West Publishing Company, St. Paul, 1990.

[17] I. Popescu, *Mechanisms*, University of Craiova Press, Craiova, Romania, 1990.

[18] C. A. Rubin, *The Student Edition of Working Model*, Addison–Wesley Publishing Company, Reading, MA, 1995.

[19] I. H. Shames, *Engineering Mechanics - Statics and Dynamics*, Prentice-Hall, Upper Saddle River, NJ, 1997.

[20] J. E. Shigley and C. R. Mischke, *Mechanical Engineering Design*, McGraw-Hill, New York, 1989.

[21] J. E. Shigley, C. R. Mischke, and R. G. Budynas, *Mechanical Engineering Design*, 7th ed., McGraw-Hill, New York, 2004.

[22] J. E. Shigley and J. J. Uicker, *Theory of Machines and Mechanisms*, McGraw-Hill, New York, 1995.

[23] A. C. Ugural, *Mechanical Design*, McGraw-Hill, New York, 2004.

[24] R. Voinea, D. Voiculescu, and V. Ceausu, *Mechanics* [Mecanica], EDP, Bucharest, 1983.

[25] J. Wileman, M. Choudhury, and I. Green, "Computation of Member Stiffness in Bolted Connections," *Journal of Machine Design*, Vol. 193, pp. 432-437, 1991.

[26] S. Wolfram, *Mathematica*, Wolfram Media/Cambridge University Press, Cambridge, 1999.

[27] National Council of Examiners for Engineering and Surveying (NCEES), *Fundamentals of Engineering. Supplied-Reference Handbook*, Clemson, SC, 2001.

[28] * * * ,*The Theory of Mechanisms and Machines* [Teoria mehanizmov i masin], Vassaia scola, Minsk, Russia, 1970.

[29] * * * ,*Working Model 2D, Users Manual*, Knowledge Revolution, San Mateo, CA, 1996.

Figure captions

Fig. 3.1. Terminology of an external screw thread.

Fig. 3.2. Geometry of an external thread.

Fig. 3.3. (a) Single threaded right-hand screw and (b) double-threaded left-hand screw.

Fig. 3.4. Standard geometry of an ISO profile.

Fig. 3.5. (a) Thread representation and (b) typical screw heads.

Fig. 3.6 Acme threads.

Fig. 3.7 Square threads.

Fig. 3.8 Power screw.

Fig. 3.9 Force diagrams for (a) lifting the load and (b) lowering the load

Fig. 3.10. (a) Acme screw threads in normal and axial planes and (b) force diagram.

Fig. 3.11. Thrust collar.

Fig. 3.12. (a) Square-threaded jack and (b) force diagram.

Fig. 3.13. (a) Force diagrams for two parts connected with a bolt; (b) free-body diagram of a portion without the external load; and (c) free-body diagram of a portion with the external load.

Fig. 3.14. Geometry of a bolt and nut assembly.

Fig. 3.15. Screw jack used in Example 3.1.

Fig. 3.16. Acme screw jack used in Example 3.2.

Fig. 3.17. Efficiency of Acme screw thread (the collar friction is neglected).

Fig. 3.18. Double square-threaded screw.

Fig. 3.19. C-clamp.

Fig. 3.20. Bolt and nut assembly.

Fig. 3.21. Bolted pressure vessel.

Fig. 3.22. Bolt with two shear planes.