Chapter Eight: Screws and Fasteners

Introduction: Thread Standards and Definitions

The terminology of screw threads, illustrated in Fig. 8–1, and the thread specification in SI and Imperial systems are given in Fig (8-2).

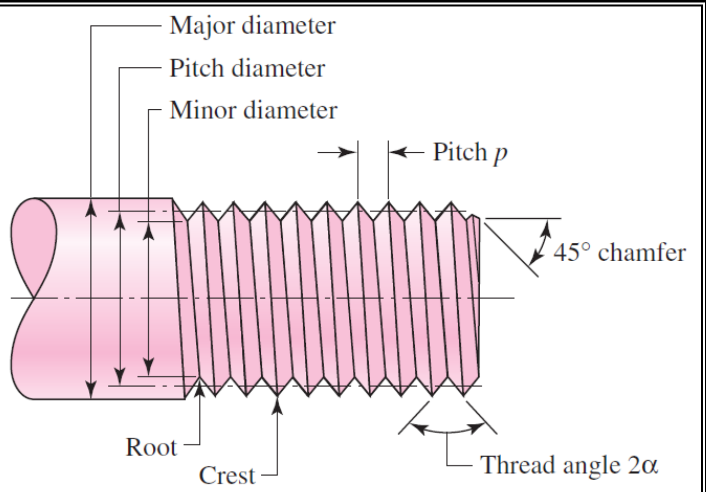
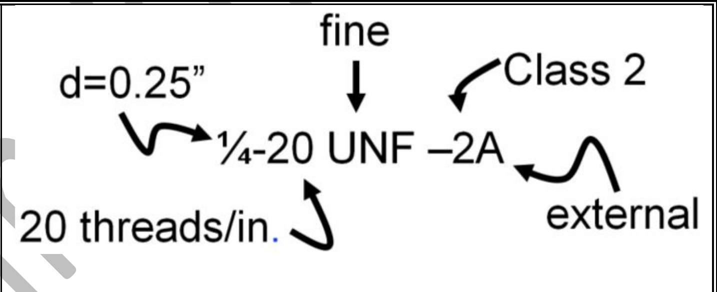


Figure 8–1
Terminology of screw threads.

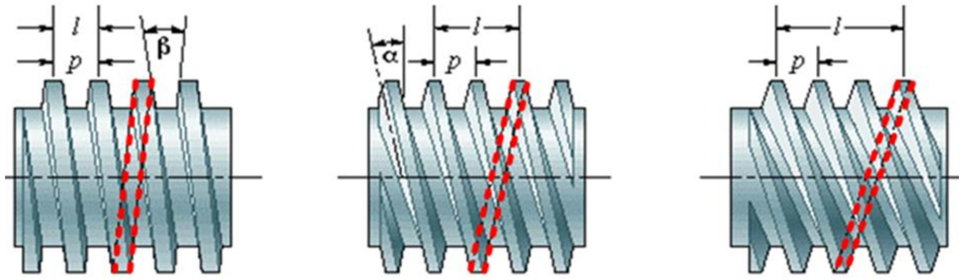
(a) metric



(b) Empirial

Fig 8-2: Thread specifications

NOTES
• The thread angle $2\alpha = 60°$ • **Pitch**: is the distance between corresponding points on adjacent threads. • **Major diameter** is the diameter over the crests of the thread • **Crest**: is the most prominent part of thread, either external or internal • **Root**: lies at the bottom of the groove between two adjacent threads • **Flank**: is the straight side of the thread between the root and the crest • **Minor diameter**: is the smallest diameter of the thread measured at right angles to the thread axis • **Effective diameter**: is the diameter on which the width of the spaces is equal to the width of the threads. • **Lead (l):** is the distance the nut moves parallel to the screw axis when the nut is given one turn. For a single thread *l=p*. For a multiple-threaded bolt *l=np*. where *n=1* for single start threads, 2 for double start , 3 for triple start etc., *see* figures *a, b* and *c* below.



(a) (b) (c)

Table 8–1 Diameters and Areas of Coarse-Pitch and Fine Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470
110				2	9180	9080

8-1- The Mechanics of Power Screws:

A power screw is a device used in machinery to change angular motion into linear motion, and usually, to transmit power.

Applications: lathes, vises, presses, jacks, see Fig. 8–4 in your textbook.

Types of threads (أنواع التسنين):

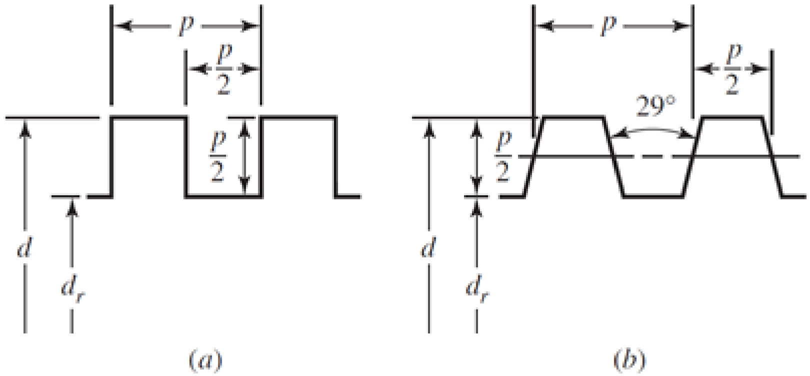


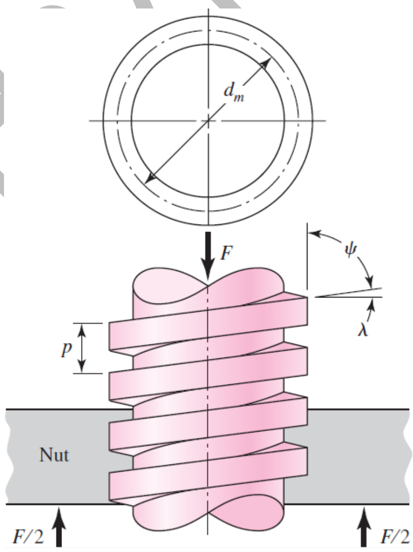
Figure 8–3 types of threads (*a*) Square thread; (*b*) Acme thread.

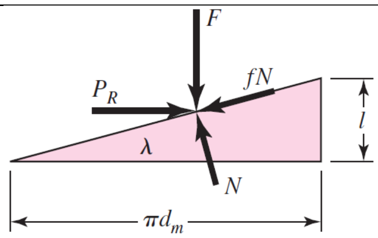
Table 8–3 Preferred Pitches (الخطوة المفضلة) for Acme Threads

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	$\frac{1}{16}$	$\frac{1}{14}$	$\frac{1}{12}$	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$

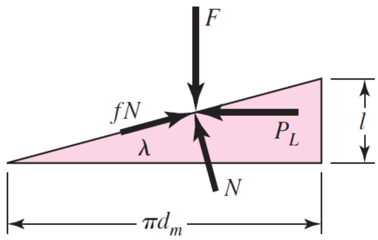
a square-threaded power screw
***n*=1**
d_m= mean diameter,
p*= pitch *p
λ = lead angle,
ψ =helix angle
***F*=axial compressive**

Figure 8–5 Portion of a power screw.





(*a*) raising the load



(*b*) lowering the load.
(c)
Figure 8–6 Force diagrams

8-1-1: The Raising Torque T_R: (عزم الرفع)

To raise the load, a force P_R acts to the right (Fig. 8–6*a*),

$$\left.\begin{aligned}\sum F_H &= P_R - N\sin\lambda - \mu\cos\lambda = 0\\ \sum F_V &= F - \mu N\sin\lambda - N\cos\lambda = 0\end{aligned}\right\} \ldots\ldots\ldots\ldots(a)$$

$$P_R = \frac{F(\sin\lambda + \mu\cos\lambda)}{\cos\lambda - \mu\sin\lambda}$$

Dividing the numerator and the denominator by $\cos\lambda$ and substituting $tan\lambda = l/\pi d_m$, gives

$$P_R = \frac{F[(l/\pi d_m) + \mu]}{1 - \mu\, l/\pi d_m}$$

The raising torque is the product of the force P_R and the mean radius $d_m/2$,

$$\boldsymbol{T_R = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m + l}{\pi d_m - \mu l}\right)} \quad \ldots\ldots\ldots\ldots\ldots(\mathbf{8-1})$$

8-1-2: The lowering torque T_L: (عزم الخفض)

to lower the load, P_L acts to the left (Fig. 8–6*b*).

$$\left.\begin{aligned}\sum F_H &= -P_L - N\sin\lambda - \mu N\cos\lambda = 0\\ \sum F_V &= F - \mu N\sin\lambda - N\cos\lambda = 0\end{aligned}\right\} \ldots\ldots\ldots\ldots(b)$$

$$P_L = \frac{F(\mu\cos\lambda - \sin\lambda)}{\cos\lambda + \mu\sin\lambda}$$

Similarly, as for raising force:

$$P_L = \frac{F[\mu - (l/\pi d_m)]}{1 + \mu\, l/\pi d_m}$$

The torque that required to lower a load in squared threaded screws is:

$$\boldsymbol{T_L = \frac{Fd_m}{2}\left(\frac{\pi\mu d_m - l}{\pi d_m + \mu l}\right)} \quad \ldots\ldots\ldots\ldots\ldots(\mathbf{8-2})$$

8-1-3: (Self – locking) (ظاهرة الاقفال الذاتي)

self-locking condition $T_L>0$. so $\pi\mu d_m > l$.

Now divide both sides of this inequality by πd_m. Recognizing that

$$\frac{l}{\pi d_m} = tan\,\lambda$$

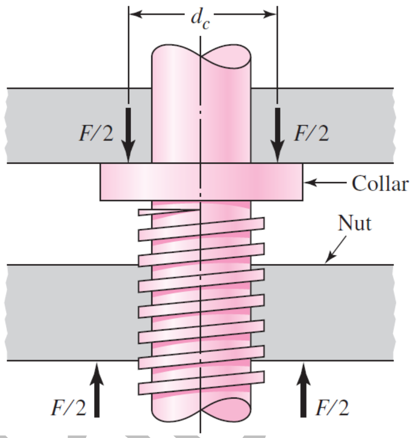
we get

$$\mu > \tan\lambda \quad \ldots\ldots\ldots\ldots\ldots.\ldots.(8-3)$$

الشرط اللازم للإقفال الذاتي هو ان تكون زاوية الاحتكاك اكبر من زاوية اللولب ($\phi > \lambda$).

8-1-4: Efficiency (كفاءة لوالب القدرة)

An expression for efficiency is also useful in the evaluation of power screws. If we let μ = 0 in Eq. (8–1), we obtain



$$T_0 = \frac{Fd_m}{2}\left(\frac{l + \pi(0)d_m}{\pi d_m - (0)l}\right) = \frac{Fd_m}{2}\left(\frac{l}{\pi d_m}\right) = \frac{Fl}{2\pi}$$

The efficiency is therefore

$$\boldsymbol{e}\% = \frac{\text{the rasing torque with no friction}}{\text{the rasing torque with friction}}$$

$$\boldsymbol{e}\ \% = \frac{\boldsymbol{T_0}}{\boldsymbol{T_R}} = \frac{\boldsymbol{Fl}}{\boldsymbol{2\pi T_R}} \times \mathbf{100}\ \% \ldots\ldots\ldots\ldots\ldots(\mathbf{8-4})$$

8-1-5: The Collar Torque, T_c (عزم حلقة الاسناد)

If d_c and f_c are the diameter and the friction coefficient of collar, the torque required is

$$\boldsymbol{T_c} = \frac{\boldsymbol{F\mu_c d_c}}{\mathbf{2}} \quad \ldots\ldots\ldots\ldots(\mathbf{8-6})$$

8-1-6: Stresses In Power Screw (الاجهادات في لولب القدرة):

Nominal body stresses in power screws can be related to thread parameters as follows.

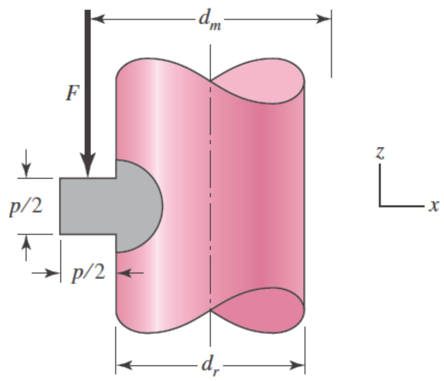
1- The shear stress τ in torsion of the screw body:

$$\boldsymbol{\tau} = \frac{\mathbf{16}\boldsymbol{T}}{\boldsymbol{\pi d_r^3}} \quad \ldots\ldots\ldots\ldots(\mathbf{8-7})$$

2- The axial stress σ in the body

$$\boldsymbol{\sigma} = \frac{\mathbf{4}\boldsymbol{F}}{\boldsymbol{\pi d_r^2}} \quad \ldots\ldots\ldots\ldots(\mathbf{8-8})$$

3- Bearing stress on threads



$$\boldsymbol{\sigma_B} = -\frac{\boldsymbol{F}}{\frac{\boldsymbol{\pi d_m n_t p}}{\mathbf{2}}} = -\frac{\mathbf{2}\boldsymbol{F}}{\boldsymbol{\pi d_m n_t p}} \quad \ldots\ldots\ldots(\mathbf{8-10})$$

4- Bending stress in threads

$$\boldsymbol{\sigma_b} = \frac{\boldsymbol{M}}{\boldsymbol{I/c}} = \frac{\boldsymbol{Fp}}{\mathbf{4}}\frac{\mathbf{24}}{\boldsymbol{\pi d_r n_t p^2}} = \frac{\mathbf{6}\boldsymbol{F}}{\boldsymbol{\pi d_r n_t p}} \quad \ldots\ldots\ldots(\mathbf{8-11})$$

EXAMPLE 8–1 A square-thread power screw has a major diameter of 32 mm and a pitch of 4 mm with three double threads. The given data include $\mu = \mu_c = 0.08, d_c = 40$ mm, and $F = 6.4$ kN per screw.

a) Find the thread depth, thread width, pitch diameter, minor diameter, and lead.

b) Find the torque required to raise and lower the load.

c) Find the efficiency during lifting the load.

d) Find the torsional and compressive body stresses.

e) Find the bearing stress.

f) Find the thread bending and shear stresses at the root.

Solution:

Given data

d=32mm , p=4 mm n_t=3, $\mu = \mu_c = 0.08$, d_c = 40 mm, $F = 6.4$ kN

Required

$h, t, d_p, d_r, l, T_R, T_L, e\%, \sigma_c, \tau, \sigma_B, \sigma_b, \tau_r$

(a)

$h = p/2 = 4/2 = 2$ mm

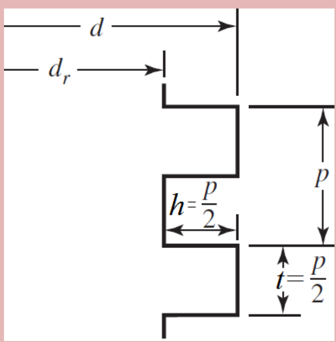
$t = h = p/2 = 4/2 = 2$ mm (squared)

$d_p = d - p/2 = 32 - 4/2 = 30$ mm

$d_r = d - p = 32 - 4 = 28$ mm

$l = np = 2p = 2(4)$

$= 8$ mm (double thread)



(b)

$$\boldsymbol{T_R = \frac{Fd_m}{2}\left(\frac{l + \pi\mu d_m}{\pi d_m - \mu l}\right) + \frac{F\mu_c d_c}{2}}$$

$$T_R = \frac{6.4(30)}{2}\left(\frac{8 + \pi(0.08)(30)}{\pi(30) - 0.08(8)}\right) + \frac{6.4(0.08)(40)}{2}$$

$$= 15.94 + 10.24 = 26.18\ \mathrm{N \cdot m}$$

$$T_L = \frac{Fd_m}{2}\left(\frac{l - \pi\mu d_m}{\pi d_m + \mu l}\right) + \frac{F\mu_c d_c}{2}$$

$$T_L = \frac{6.4(30)}{2}\left(\frac{8 - \pi(0.08)(30)}{\pi(30) + 0.08(8)}\right) + \frac{6.4(0.08)(40)}{2}$$

$$= -0.466 + 10.24 = 9.77\ \mathrm{N \cdot m}$$

(c)

$$e = \frac{Fl}{2\pi T_R} = \frac{6.4(8)}{2\pi(26.18)} \times 100\% = 31.1\%$$

(d)

$$\tau = \frac{16T_R}{\pi d_r^3} = \frac{6.4(26.18)(10^3)}{\pi(28^3)} = 6.07\ \text{MPa}$$

$$\sigma = -\frac{4F}{\pi d_r^2} = -\frac{4(6.4)(10^3)}{\pi(28^2)} = -10.4\ \text{MPa}$$

(e)

$$\sigma_B = \frac{2F}{\pi d_m n_t p} = \frac{2(6.4)(10)^3}{\pi(30)(3)(4)} = 11.3\ \text{MPa}$$

(f)

$$\sigma_b = \frac{6F}{\pi d_r n_t p} = \frac{6(6.4)(10^3)}{\pi(28)(3)(4)} = 36.4\ \text{MPa}$$

$$\tau_r = \frac{2F}{\pi d_r n_t p} = \frac{2(6.4)(10^3)}{\pi(28)(3)(4)} = 12.1\ \text{MPa}$$

Table 8–4 Screw Bearing Pressure p_b

Screw Material	Nut Material	Safe p_b, psi	Notes
Steel	Bronze	2500–3500	Low speed
Steel	Bronze	1600–2500	10 fpm
	Cast iron	1800–2500	8 fpm
Steel	Bronze	800–1400	20–40 fpm
	Cast iron	600–1000	20–40 fpm
Steel	Bronze	150–240	50 fpm

Table 8–5 Coefficients of Friction f for Threaded Pairs

Screw Material	Nut Material			
	Steel	Bronze	Brass	Cast Iron
Steel, dry	0.15–0.25	0.15–0.23	0.15–0.19	0.15–0.25
Steel, machine oil	0.11–0.17	0.10–0.16	0.10–0.15	0.11–0.17
Bronze	0.08–0.12	0.04–0.06	—	0.06–0.09

Table 8–6 Thrust-Collar Friction Coefficients

Combination	Running	Starting
Soft steel on cast iron	0.12	0.17
Hard steel on cast iron	0.09	0.15
Soft steel on bronze	0.08	0.10
Hard steel on bronze	0.06	0.08

8-1-7: The Acme Threads Screws

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 2a and the lead angle l. Since lead angles are small, this inclination can be neglected and only the effect of the thread angle (Fig. 8–7*a*) considered. The effect of the angle a is to increase the frictional force by the wedging action of the threads. Therefore the frictional terms in Eq. (8–1) must be divided by $\cos\alpha$. For raising the load, or for tightening a screw or bolt, this yields

$$\boldsymbol{T_R = \frac{F d_m}{2}\left(\frac{\pi \bar{\mu} d_m + l}{\pi d_m - \bar{\mu} l}\right)} \quad \ldots\ldots\ldots\ldots\ldots (\mathbf{8-5})$$

Where $\boldsymbol{\bar{\mu} = \mu / \cos\alpha}$ $\quad \alpha = 14.5°$ (half tread angle)

In using Eq. (8–5), remember that it is an approximation because the effect of the lead angle has been neglected. For power screws, the Acme thread is not as efficient as the square thread, because of the additional friction due to the wedging action, but it is often preferred because it is easier to machine and permits the use of a split nut, which can be adjusted to take up for wear.

8-1-8: Relation Between the Nut Speed and the Screw Speed (العلاقة بين سرعة للولب وسرعة الصامولة)

$$v_{nut} = \frac{n \cdot p \cdot N_s}{60}$$

v_{nut}: the linear speed of the nut (m/s)
n: number of thread starts (n=1,2,3,…)
p: the pitch (mm)
N_s: the rotational speed of the screw (rev/min)

8-1-9: Relation Between the Screw Speed and the motor Speed (العلاقة بين سرعة اللولب وسرعة المحرك)

$$N_s = \frac{N_m}{r}$$

N_m: is the motor speed (rev/min)
N_s: is the screw speed (rev/min)
r: is the reduction ratio

8-1-10: The Motor Power (العلاقة بين قدرة اللولب وقدرة المحرك):

the power of the motor that required to rotate the screw is:

$$P_m = \frac{P_s}{\eta_m}$$

P_m: is the motor power.
P_s: is the screw power = (lifting torque x angular velocity) = $T_R \times \omega = \frac{2\pi N_s T_R}{60}$
η_m: is the mechanical efficiency.

PROBLEMS

8–1 A power screw is 25 mm in diameter and has a thread pitch of 5 mm.

(*a*) Find the thread depth, the thread width, the mean and root diameters, and the lead, provided square threads are used.

(*b*) Repeat part (*a*) for Acme threads.

8–2 Using the information in the footnote of Table 8–1, show that the tensile-stress area is

$$A_t = \frac{\pi}{4}(d - 0.938\ 194p)^2$$

8–3 Show that for zero collar friction the efficiency of a square-thread screw is given by the equation

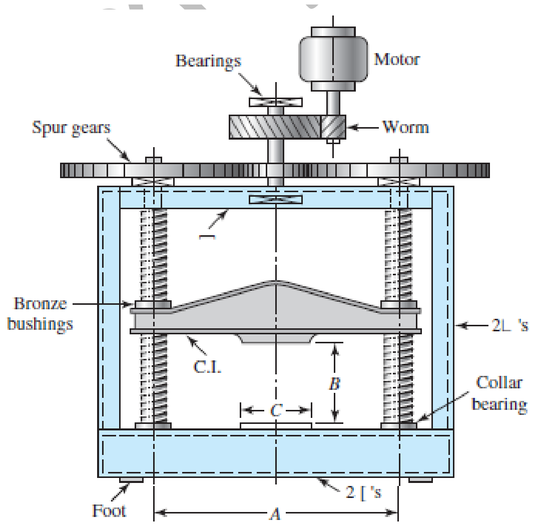
$$e = \tan\lambda \frac{1 - f\tan\lambda}{\tan\lambda + f}$$

Plot a curve of the efficiency for lead angles up to 45°. Use $f = 0.08$.

8–4 A single-threaded power screw is 25 mm in diameter with a pitch of 5 mm. A vertical load on the screw reaches a maximum of 5 kN. The coefficients of friction are 0.06 for the collar and 0.09 for the threads. The frictional diameter of the collar is 45 mm. Find the overall efficiency and the torque to "raise" and "lower" the load.

8–5 The machine shown in the figure can be used for a tension test but not for a compression test. Why? Can both screws have the same hand?

Problem 8–5



8-6: The press shown for Prob. 8–5 has a rated load of 22 kN. The twin screws have Acme threads, a diameter of 50 mm, and a pitch of 6 mm. Coefficients of friction are 0.05 for the threads and 0.08 for the collar bearings. Collar diameters are 90 mm. The gears have an efficiency of 95 percent and a speed ratio of 60:1. A slip clutch, on the motor shaft, prevents overloading. The full-load motor speed is 1720 rev/min.

(*a*) When the motor is turned on, how fast will the press head move?

(*b*) What should be the horsepower rating of the motor?

10.6 A screw jack with a 1-in. double-thread Acme screw is used to raise a load of 10,000 lb. A plain thrust collar of 2.0 in. mean diameter is used. Coefficients of running friction are estimated as 0.13 and 0.10 for f and f_c, respectively. (See Figure P10.6.)

(a) Determine the screw pitch, lead, thread depth, mean pitch diameter, and helix angle.

(b) Estimate the starting torque for raising and for lowering the load.

(c) Estimate the efficiency of the jack when raising the load.

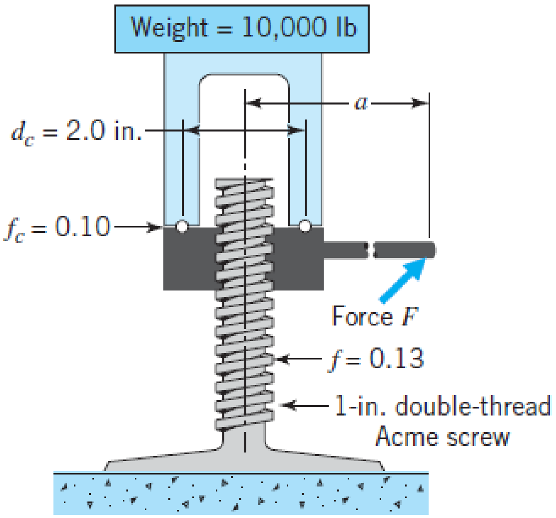


FIGURE P10.6

10.9 A jack similar to the ones shown in Figure 10.5 uses a single square-thread screw to raise a load of 50 kN. The screw has a major diameter of 36 mm and a pitch of 6 mm. The thrust collar mean diameter is 80 mm. Running friction coefficients are estimated to be 0.15 for the screw and 0.12 for the collar:

(a) Determine the thread depth and helix angle.

(b) Estimate the starting torque for raising and lowering the load.

(c) Estimate the efficiency of the jack for raising the load.

(d) Estimate the power required to drive the screw at a constant one revolution per second.

8-2: Threaded Fasteners البراغي

8-2-1 Introduction

A bolt is a device used to connect or join two or more components. Traditional forms of fastening include nuts, bolts, screws

Figure 8–9 is a drawing of a standard hexagon-head bolt. Points of stress concentration are at the fillet, at the start of the threads (run out), and at the thread-root fillet in the plane of the nut when it is present. See Table A–29 for dimensions. The diameter of the washer face is the same as the width across the flats of the hexagon. The thread length of metric-series bolts, may be found from the following relations

$$L_T = \begin{cases} 2d + 6 \quad \text{for } L \le 125 \text{ mm} \\ 2d + 12 \quad \text{for} \quad 125 < L \le 200 \text{ mm} \\ 2d + 25 \quad \text{for } L > 200 \text{ mm} \end{cases} \ldots\ldots (8-14)$$

where d is the nominal diameter of the bolt

L_T is the Length of threaded portion (طول التسنين)

$l_d = L - L_T$, Length of useful unthreaded portion (طول الجزء غير المسنن)

$l_t = l - l_d$, Length of useful threaded portion : (طول التسنين داخل العنصر)

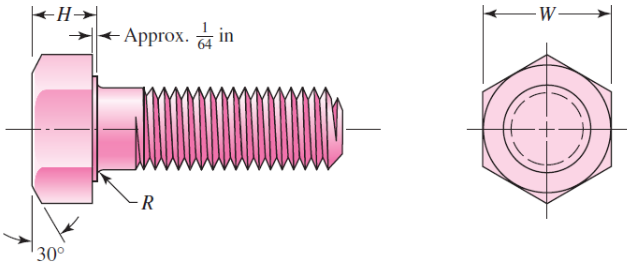
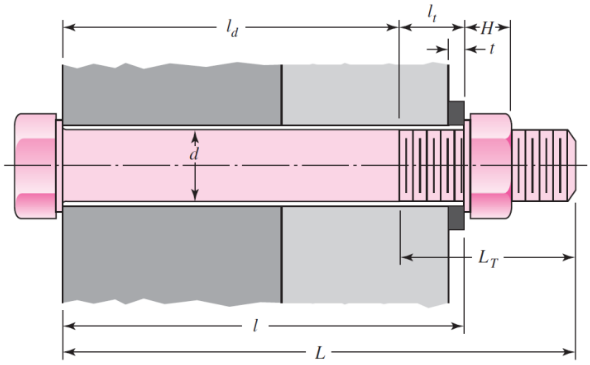


Figure 8–9 is a drawing of a standard hexagon-head bolt



The total length of the bolt (L) is:

$$\boldsymbol{L = l + H + 2p}$$

where

H: is the nut height see Fig (8-9) and table A-31in the appendix

p is the thread pitch see table (8-1)

$$\boldsymbol{A_t = \frac{\pi}{16}\left(d_p + d_r\right)^2}$$

$$\boldsymbol{d_p = d - 0.649519p}$$

$$\boldsymbol{d_r = d - 1.226869p}$$

8-2-2 Bolt Stiffness

$$\boldsymbol{k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}} \qquad \dots\dots\dots\dots\dots..(\boldsymbol{8-17})$$

where A_t = tensile-stress area (Table 8–1)

A_d = major-diameter area of fastener $A_d = \pi d^2/4$

l_d = length of unthreaded portion in grip

l_t = length of threaded portion of grip

8-2-3 Joints—Member Stiffness

There may be more than two members included in the grip of the fastener. All together these act like compressive springs in series, and hence the total spring rate of the members is

$$\boldsymbol{\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \cdots + \frac{1}{k_i}} \qquad \dots\dots\dots\dots\dots.(\boldsymbol{8-18})$$

If the members of the joint have the same Young's modulus *E* with symmetrical frusta back to back, then they act as two identical springs in series. From Eq. (8–18) using

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} \qquad \text{using} \quad k_1 = k_2 \quad \therefore\ k_m = \frac{k}{2}$$

The joint stiffness is:

$$\boldsymbol{k_m = \frac{0.5774\pi E d}{2\ln\left(5\dfrac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)}} \qquad \dots\dots\dots\dots\dots\dots\dots.(\boldsymbol{8-22})$$

where

l=the joint length

E= modulus of elasticity for the member material, table (8-8)

d= bolt's diameter

It is very important to note that the *entire joint* must be made up of the *same material* to apply eq. (8-22)

EXAMPLE 8–2

Two, 12 mm-thick steel plates are clamped by a M12 bolts, with a 2.5 mm-thick washer under the nut. Find the member spring rate k_m using eq (8.22)

Solution

Data given: The grip is 12 + 12 + 2.5 = 26.5 mm=0.0265 m. E=207 GPa (table 8-8)

Using Eq. (8–22) with $l = 0.065\ m$ and $d = 0.012\ m$, we write

$$k_m = \frac{0.5774\pi E d}{2\ln\left(5\dfrac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)}$$

$$k_m = \frac{0.5774\pi(207\times10^9)(0.012)}{2\ln\left(5\dfrac{0.5774(0.0265) + 0.5(0.012)}{0.5774(0.0265) + 2.5(0.012)}\right)} = 2635\ \text{MN/m}$$

8-2-4 Bolt Strength

the minimum proof strength S_p, the minimum yield strength, S_y and the minimum tensile strength, S_u. Specifications for metric fasteners are given in Table 8–11.

Table 8–11 Metric Mechanical-Property Classes for Steel Bolts, Screws, and Studs*

Property Class	Size Range, Inclusive	Proof Strength,† MPa	Tensile Strength,† MPa	Yield Strength,† MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9

8-2-4 Tension Joints - The External Load

Let us now consider what happens when an external tensile load P, as in the figure below, is applied to a bolted connection. It is to be assumed, of course, that the clamping force, which we will call the *preload* F_i , has been correctly applied by tightening the nut *before* P is applied. The nomenclature used is:

F_i = preload

P = external tensile load

P_b = portion of P taken by bolt

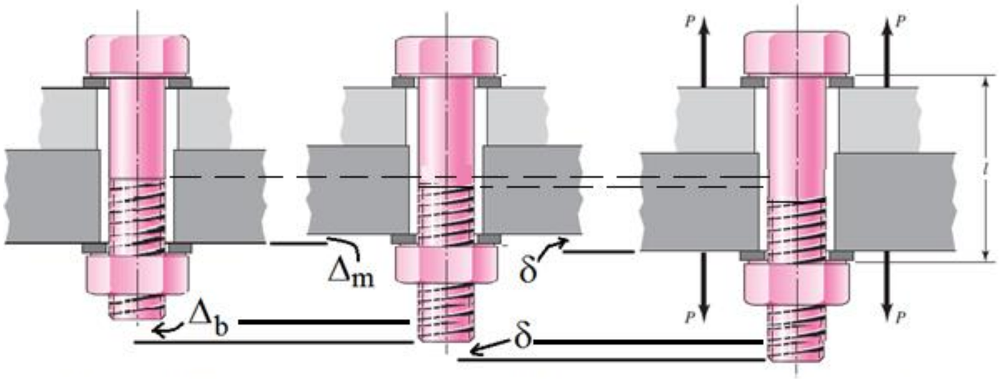
P_m = portion of P taken by members

$F_b = P_b + F_i$ = resultant bolt load

$F_m = P_m - F_i$ = resultant load on members

CP= fraction of external load P carried by bolt

$(1 - C)P$ = fraction of external load P carried by members



(a) $F_i=0$, $P=0$ (b) $F_b=F_i$, $P=0$ (c) $P=P$, $F_b=F_i+P_b$

The load P is tension, and it causes the connection to elongate, through some distance δ. We can relate this elongation to the stiffnesses by:

$$\delta = \frac{P_b}{k_b} = \frac{P_m}{k_m} \qquad \text{from compatibility}$$

$$P_m = P_b \frac{k_m}{k_b} \; \dots\dots\dots\dots (a)$$

$$P = P_b + P_m \dots\dots\dots\dots (b) \qquad \text{from equilibrium}$$

Substitute (a) in (b)

$$P = P_b + P_b \frac{k_m}{k_b} = P_b \left(1 + \frac{k_m}{k_b}\right) = P_b \left(\frac{k_b + k_m}{k_b}\right)$$

$$P_b = \frac{k_b}{k_b + k_m} \cdot P = CP$$

$$P_m = P - P_b = (1 - C)P$$

where

$$C = \frac{k_b}{k_b + k_m}$$

C is called the *stiffness constant of the joint*.

The resultant bolt load is

$$\boldsymbol{F_b = F_i + P_b = F_i + CP \qquad F_m < 0 \qquad (8-24)}$$

$$\boldsymbol{F_m = P_m - F_i = (1 - C)P - F_i \qquad (8-25)}$$

Notes:

- Table 8–12 is included to provide C
- The grip contains only two members, both of steel, and no washers.
- The ratio C describes the proportion of the external load taken by the bolt.
- The ratio $1 - C$ describes the proportion of the external load taken by the members,.
- The members take over 80 percent of the external load, this is important when fatigue loading is present
- making the grip longer causes the members to take an even greater percentage of the external load

Table 8–12: Computation of Bolt and Member Stiffness. Steel members clamped using a 1/2 in-13 NC steel bolt.

	Stiffnesses, M lbf/in			
Bolt Grip, in	k_b	k_m	C	$1-C$
2	2.57	12.69	0.168	0.832
3	1.79	11.33	0.136	0.864
4	1.37	10.63	0.114	0.886

8-2-5 Relating Bolt Torque to Bolt Tension

we can obtain a good estimate of the torque required to produce a given preload by combining Eqs. (8–5) and (8–6), see the details in your textbook: the initial tension in the bolt is related to the applied torque by:

$$\boldsymbol{T = KF_i d} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (\mathbf{8-27})$$

K is the torque coefficient

Manufacturer of fasteners recommends the values of ***K*** shown in Table 8–15.

In our lectures a value of *K* = 0.2 will be used when the bolt condition is not stated

Table 8–15 Torque Factors *K* for Use with Eq. (8–27)

Bolt Condition	K
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

8-2-6 Statically Loaded Tension Joint with Preload

The tensile stress in the bolt can be found from

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t} \qquad \ldots\ldots\ldots..(a)$$

The limiting value of σ_b is the proof strength S_p. Thus, with the introduction of a *load factor n*, Eq.(*a*) becomes

1- **safety factor for yielding ($\boldsymbol{n_p}$)**

$$\boldsymbol{n_p = \frac{S_p}{\sigma_b} = \frac{S_p A_t}{CP + F_i}} \qquad \ldots\ldots\ldots..(\mathbf{8-28})$$

2- **Safety Factor For Separation ($\boldsymbol{n_0}$)**

Separation occurs when $F_m = 0$

$$\boldsymbol{F_m = (1 - C)P_0 - F_i = 0} \qquad \textbf{(8 - 25)}$$

$$P_0 = \frac{F_i}{1 - C}$$

where P_0 is the maximum external load that causes separation

$$n_s = \frac{\text{load that causes separation}}{\text{applied load}} = \frac{P_0}{P}$$

$$\boldsymbol{n_s = \frac{F_i}{(1 - C)P}} \quad \dots\dots\dots\dots \textbf{(8 - 30)}$$

Example 8-3

A 19M × 1.6 bolt , SAE grade 8.8 bolt is subjected to a load of 26.7 kN in a tension joint. The initial bolt tension is 110 kN. The bolt and joint stiffnesses are 1.14 and 2.42 GN/m, respectively.

(*a*) Determine the preload and service load stresses in the bolt. Compare these to the SAE minimum proof strength of the bolt.

(*b*) Specify the torque necessary to develop the preload, using Eq. (8–27). $\mu = \mu_c = 0.15$.

Solution:

$$A_t = \frac{\pi}{16}\left(d_p + d_r\right)^2$$

$$d_p = d - 0.649519p$$

$$d_r = d - 1.226869p$$

$$d_p = 19 - 0.649519(1.6) = 17.96\ mm$$

$$d_r = 19 - 1.226869(1.6) = 17.04\ mm$$

$$A_t = \frac{\pi}{16}\left(d_p + d_r\right)^2 = \frac{\pi}{16}(17.96 + 17.04)^2$$

$$= 240.5\ mm^2$$

the preload stress is

$$\sigma_i = \frac{F_i}{A_t} = \frac{110000}{240.5} = 457.4\ \text{MPa}$$

The stiffness constant is

$$C = \frac{k_b}{k_b + k_m} = \frac{1.14}{1.14 + 2.42} = 0.32$$

From Eq. (8–24), the stress under the service load is

$$\sigma_b = \frac{F_b}{A_t} = \frac{CP + F_i}{A_t}$$

$$= \frac{0.32(26.7) \times 10^3 + 110000}{240.5}$$

$$= 493\ \text{MPa}$$

From Table 8–9, the SAE minimum proof strength of the bolt is Sp = 85 kpsi. The preload and service load stresses are respectively 21 and 15 percent less than the proof strength.

(b) From Eq. (8–27), the torque necessary to achieve the preload is

$$T = KF_i d = 0.2(110000)(19 \times 10^{-3})$$

$$= 418\ \text{N} \cdot \text{m}$$

8-2-7 Pre-Load

It is recommended for both static and fatigue loading that the following relations to be used for preload:

$$\boldsymbol{F_i} = \begin{cases} \boldsymbol{0.75F_p} \text{ for nonpermanent connections, reused fasteners} \\ \boldsymbol{0.90F_p} \text{ for permanent connections} \end{cases} \quad \dots\dots\dots \textbf{(8 - 31)}$$

where $\boldsymbol{F_p}$ is the proof load, obtained from the equation

$$\boldsymbol{F_p = A_p S_p} \qquad \dots\dots\dots\dots \textbf{(8- 32)}$$

- S_p is the proof strength obtained from Tables 8–9 to 8–11. For other materials, an approximate value is $S_p = 0.85S_y$.
- Be very careful not to use a soft material in a threaded fastener

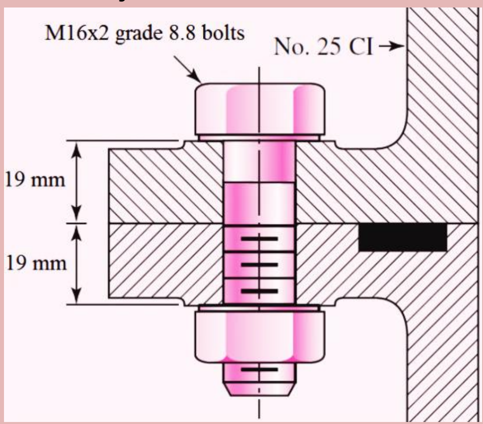
Example 8–4:

The figure is a cross section of a grade 25 cast-iron pressure vessel. A total of N, M16x2.0 grade 8.8 bolts are to be used to resist a separating force of 160 kN.

(a) Determine k_b, k_m, and C.

(b) Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart.

(c) with the number of bolts obtained in (b), determine the realized load factor for overload, the yielding factor of safety, and the separation factor of safety.



Solution:

(a) The grip length is l = 38 mm.

H_{nut}=14.8 mm (Table A–31 for d=16 mm)

Bolt length $L = l + H_{nut} + 2p$

$$L = 38 + 14.8 + 2 \times 2 = 56.8\ mm$$

L=60 mm (use preferred number table A-17)

$$L_T = 2d + 6 = 2(16) + 6 = 38\ mm$$

$$l_d = L - L_T = 60 - 38 = 22\ mm$$

$$l_t = l - l_d = 60 - 38 = 16\ mm$$

$$A_d = \frac{\pi}{4} d^2 = \frac{\pi}{4}(16)^2 = 201\ mm$$

E_{bolt}=207 GPa Table 8.8

$$k_b = \frac{A_d A_t E_{bolt}}{A_d l_t + A_t l_d} = \frac{201(157)(207000)}{201(16) + 157(22)}$$

$$= 979355 \frac{\text{N}}{\text{mm}} = 980(10)^6 \frac{\text{N}}{\text{m}}$$

E_m=E_{CI}=100 GPa=100000 N/mm^2

$$k_m = \frac{0.5774\pi E d}{2\ln\left(5\dfrac{0.5774l + 0.5d}{0.5774l + 2.5d}\right)}$$

$$= \frac{0.5774\pi(100000)(16)}{2\ln\left(5\dfrac{0.5774(38) + 0.5(16)}{0.5774(38) + 2.5(16)}\right)}$$

$$= 1644402 \frac{\text{N}}{\text{mm}} = 1644(10)^6 \frac{\text{N}}{\text{m}}$$

$$C = \frac{k_b}{k_b + k_m} = \frac{980}{980 + 1644} = 0.373$$

$$F_i = 0.75F_p = 0.75S_pA_t = 0.75(600)(157)$$

$$= 70650\ N \quad ....(8-31)$$

From eq. (8-28) the yielding factor of safety is

$$n_y = \frac{S_pA_t}{C(P_{total}/N) + F_i}$$

$$= \frac{600(157)}{0.373(160/6) \times 10^3 + 70650} = 1.17$$

From Eq. (8-30) the factor of safety against separation is

$$n_s = \frac{F_i}{(1 - C)(P_{total}/N)}$$

$$= \frac{70650}{(1 - 0.373)(\frac{160 \times 10^3}{6})}$$

$$= 4.23$$

8-2-8 : Bolted and Riveted Joints Loaded in Shear

Riveted and bolted joints loaded in shear are treated exactly alike in design and analysis. Figure 8–23a shows a riveted connection loaded in shear.

The bending moment is approximately $M = Ft/2$, where F is the shearing force and t is the grip of the rivet. The bending stress in the members or in the rivet is,

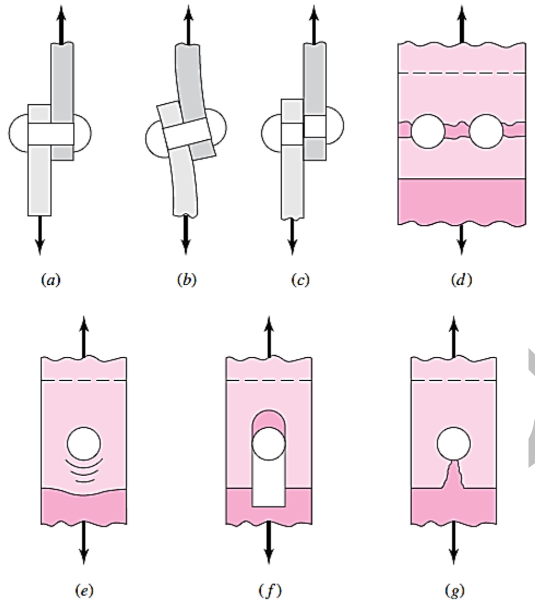
$$\sigma = \frac{M}{I/c} \qquad (8\text{–}52)$$

where I/c is the section modulus for the weakest member or for the rivet or rivets. Although this equation can be used to determine the bending stress, it is seldom used in design; instead its effect is compensated for by an increase in the factor of safety.

In Fig. 8–23*c* failure of the rivet by pure shear is shown; the stress in the rivet is

$$\tau = \frac{F}{A} \qquad (8\text{–}53)$$

Figure 8–23
Modes of failure in shear loading of a bolted or riveted connection:
(*a*) shear loading;
(*b*) bending of rivet;
(*c*) shear of rivet;
(*d*) tensile failure of members;
(*e*) bearing of rivet on members or bearing of members on rivet;
(*f*) shear tear-out;
(g) tensile tear-out



Rupture of one of the connected members or plates by pure tension is illustrated in Fig. 8–23*d*. The tensile stress is

$$\sigma = \frac{F}{A} \qquad (8\text{–}54)$$

In calculating the area for Eq. (8–54), the designer should, of course, use the combination of rivet or bolt holes that gives the smallest area.

Figure 8–23*e* illustrates a failure by crushing of the rivet or plate. Calculation of this stress, which is usually called a *bearing stress*. This gives for the stress

$$\sigma = -\frac{F}{A} \qquad (8\text{–}55)$$

where the projected area for a single rivet is $A = td$. Here, t is the thickness of the thinnest plate and d is the rivet or bolt diameter.

Edge shearing, or tearing, of the margin is shown in Fig. 8–23*f* and *g*, respectively. In structural practice this failure is avoided by spacing the rivets at least $1\frac{1}{2}$ diameters away from the edge, and hence this type of failure may usually be neglected.

EXAMPLE 8–6

Two 25- by 102-mm 1018 cold-rolled steel bars are butt-spliced with two 12.5- by 102-mm 1018 cold-rolled splice plates using four M20x1.5 grade 8.8 bolts as depicted in Fig. 8–24. For a design factor of n_d = 1.5 estimate the static load F that can be carried if the bolts lose preload.

Solution

From Table A–20, minimum strengths of S_y = 370 MPa and S_{ut} = 440 MPa are found for the members.
From Table 8–9 minimum strengths of S_p = 600 MPa and S_u = 830 MPa Sy=660 MPa for the bolts.

Bearing in bolts, all bolts loaded:

$$\sigma = \frac{F}{2td} = \frac{S_p}{n_d}$$

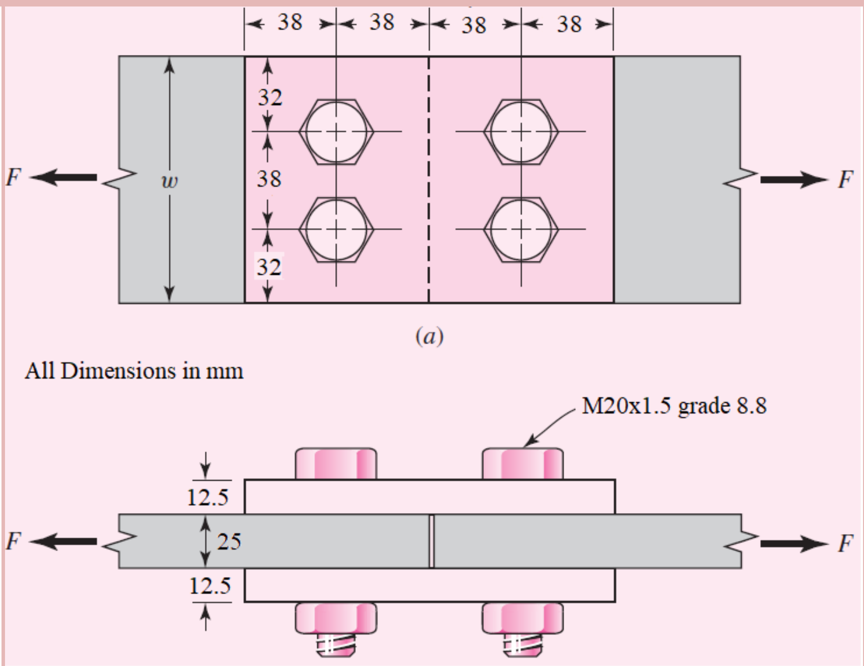


Fig 8-24

$$F = \frac{2tdS_p}{n_d} = \frac{2(25)(20)(600)}{1.5} = 400 \text{ kN}$$

Bearing in members, all bolts active:

$$\sigma = \frac{F}{2td} = \frac{(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{2td(S_y)_{mem}}{n_d} = \frac{2(25)(20)(370)}{1.5} = 247 \text{ kN}$$

Shear of bolt, all bolts active: If the bolt threads do not extend into the shear planes for four shanks:

$$\tau = \frac{F}{4\pi d^2/4} = 0.5\frac{S_p}{n_d}$$

$$F = 0.5\left(\pi d^2 \frac{S_p}{n_d}\right) = 0.5\pi(20)^2\left(\frac{600}{1.5}\right) = 251.3 \text{ kN}$$

If the bolt threads extend into a shear plane:

$$\tau = \frac{F}{4A_r} = 0.5\left(\frac{S_p}{n_d}\right)$$

$$F = \frac{0.5(4)A_rS_p}{n_d} = \frac{0.5(4)(259)(600)}{1.5} = 207.2 \text{ kN}$$

Edge shearing of member at two margin bolts: From Fig. 8–25,

$$\tau = \frac{F}{4at} = \frac{0.5(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{4at(0.5)(S_y)_{mem}}{n_d} = \frac{4(28)(25)(0.5)(370)}{1.5} = 345.3 \text{ kN}$$

Tensile yielding of members across bolt holes:

$$\sigma = \frac{F}{[102\text{-}2(20)]t} = \frac{(S_y)_{\text{mem}}}{n_d}$$

$$F = \frac{[102 - 2(20)]t(S_y)_{mem}}{n_d} = \frac{[102 - 2(20)](25)(370)_{mem}}{1.5} = 382.3 \text{ kN}$$

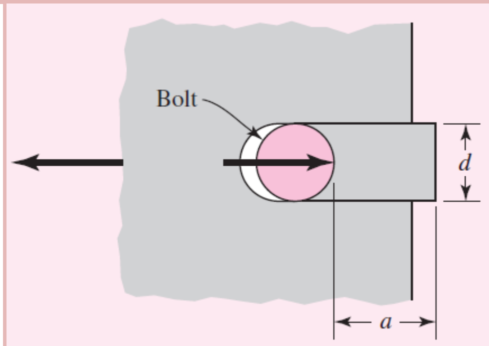


Fig 8-25

On the basis of bolt shear, the limiting value of the force is 207.2 kN, assuming the threads extend into a shear plane. However, it would be poor design to allow the threads to extend into a shear plane. So, assuming a *good* design based on bolt shear, the limiting value of the force is 251.3 kN. For the members, the bearing stress limits the load to 247 kN.

8-2-9 Shear Joints With Eccentric Loading (Bolt Groups)

The analysis of a shear joint undergoing eccentric loading requires locating the center of relative motion between the two members. In Fig. 8–26 let A_1 to A_5 be the respective cross-sectional areas of a group of five pins, or hot-driven rivets, or tight-fitting shoulder bolts

$$\bar{x} = \frac{A_1x_1 + A_2x_2 + A_3x_3 + A_4x_4 + A_5x_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i x_i}{\sum_1^n A_i}$$

$$\bar{y} = \frac{A_1y_1 + A_2y_2 + A_3y_3 + A_4y_4 + A_5y_5}{A_1 + A_2 + A_3 + A_4 + A_5} = \frac{\sum_1^n A_i y_i}{\sum_1^n A_i} \qquad (8\text{–}56)$$

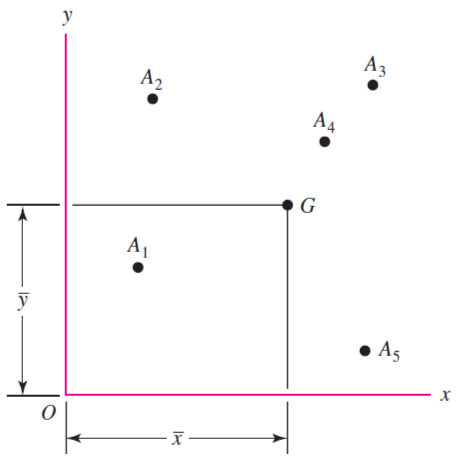


Fig 8-26 Centroid of pins, rivets, or bolts.

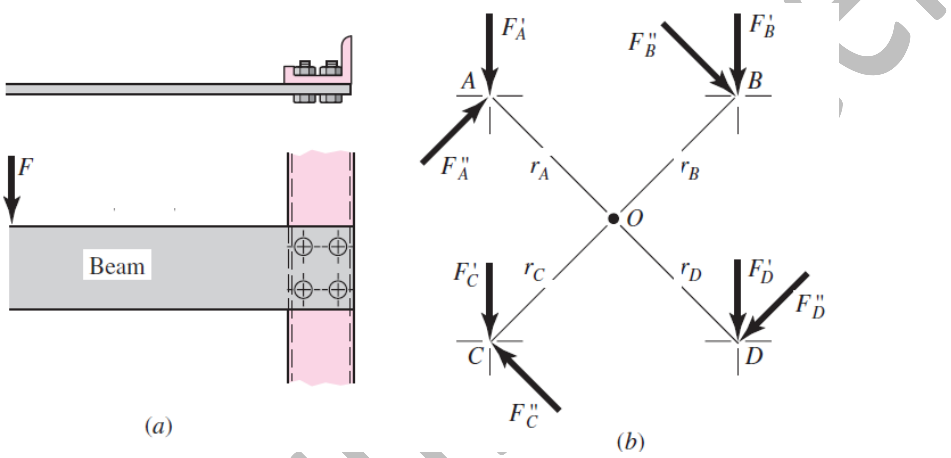


Fig 8-27

The total load taken by each bolt will be calculated in three steps.

1- In the first step the shear V_1 is divided equally among the bolts so that each bolt takes

$$\boldsymbol{F' = \frac{V_1}{n}}$$

where n : number of bolts

$\boldsymbol{F'}$ is called the *direct load*, or *primary shear*.

The direct loads F'_n are shown as vectors on the loading diagram (Fig. 8–27*b*).

2- The *moment load*, or *secondary shear*, is the additional load on each bolt due to the moment M_1.

$$M_1 = F''_A r_A + F''_B r_B + F''_C r_C + \cdots F''_n r_n \qquad (a)$$

where r_A, r_B , r_C , ... etc, are the radial distances from the centroid to the center of each bolt, The force taken by each bolt depends upon its radial distance from the centroid

$$\frac{F''_A}{r_A} = \frac{F''_B}{r_B} = \frac{F''_C}{r_C} \qquad (b)$$

Solving Eqs. (*a*) and (*b*) simultaneously, we obtain

$$\boldsymbol{F''_A = \frac{M_1 r_n}{r_A^2 + r_B^2 + r_C^2 + \cdots}} \qquad (\mathbf{8-57})$$

3- the third step, the direct and moment loads are added vectorially to obtain the resultant load on each bolt.

EXAMPLE 8–7

Shown in Fig. 8–28 is a 15- by 200-mm rectangular steel bar cantilevered to a 250-mm steel channel using four tightly fitted bolts located at A, B, C, and D.

For a $F = 16$ kN load find

(a) The resultant load on each bolt

(b) The maximum shear stress in each bolt

(c) The maximum bearing stress

(d) The critical bending stress in the bar

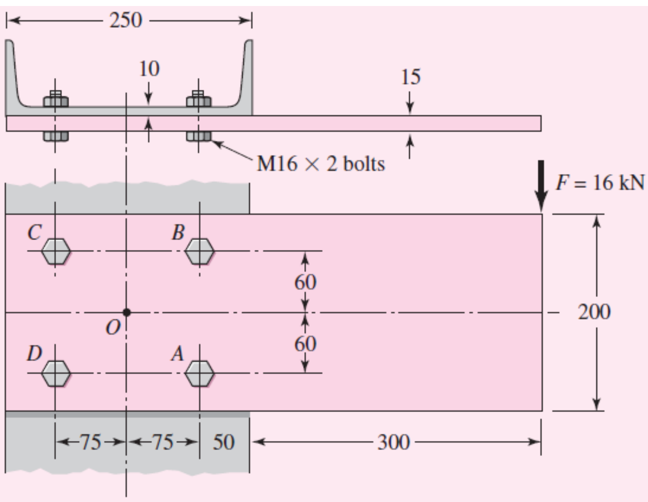
Solution

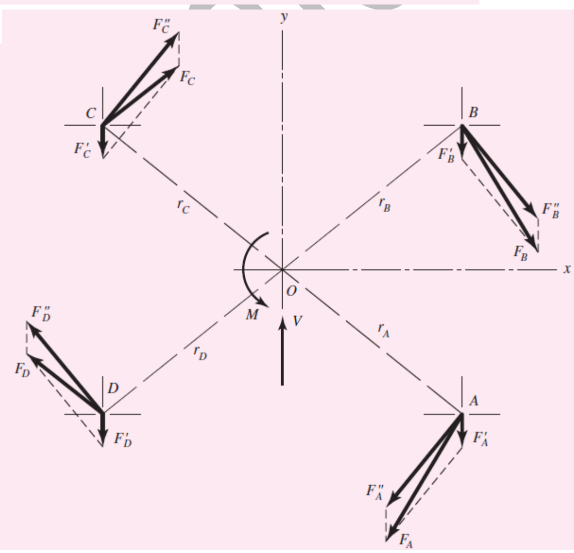
(a) Point O, the centroid of the bolt group in Fig. 8–28, is found by symmetry. If a free-body diagram of the beam were constructed, the shear reaction V would pass through O and the moment reactions M would be about O. These reactions are

$$V = 16\ \text{kN} \qquad M = 16(425) = 6800\ \text{N} \cdot \text{m}$$

In Fig. 8–29, the bolt group has been drawn to a larger scale and the reactions are shown. The distance from the centroid to the center of each bolt is

$$r = \sqrt{(60)^2 + (75)^2} = 96.0\ \text{mm}$$





The primary shear load per bolt is

$$F' = \frac{V}{n} = \frac{16}{4} = 4\ \text{kN}$$

Since the secondary shear forces are equal, Eq. (8–57) becomes

$$F'' = \frac{Mr}{4r^2} = \frac{M}{4r} = \frac{6800}{4(96.0)} = 17.7\ \text{kN}$$

The primary and secondary shear forces are plotted to scale in Fig. 8–29 and the resultants obtained by using the parallelogram rule. The magnitudes are found by measurement (or analysis) to be

$$F_A = F_B = 21.0\ \text{kN} \qquad \text{Answer}$$

$$F_C = F_D = 14.8\ \text{kN} \qquad \text{Answer}$$

(*b*) Bolts *A* and *B* are critical because they carry the largest shear load. Does this shear act on the threaded portion of the bolt, or on the unthreaded portion? The bolt length will be 25 mm plus the height of the nut plus about 2 mm for a washer. Table A–31 gives the nut height as 14.8 mm. Including two threads beyond the nut, this adds up to a length of 43.8 mm, and so a bolt 46 mm long will be needed. From Eq. (8–14) we compute the thread length as $L_T = 38$ mm. Thus the unthreaded portion of the bolt is 46 − 38 = 8 mm long. This is less than the 15 mm for the plate in Fig. 8–28, and so the bolt will tend to shear across its minor diameter. Therefore the shear-stress area is $A_s = 144$ mm^2, and so the shear stress is

$$\tau = \frac{F}{A_s} = -\frac{21.0(10)^3}{144} = 146 \text{ MPa} \qquad \text{Answer}$$

(*c*) The channel is thinner than the bar, and so the largest bearing stress is due to the pressing of the bolt against the channel web. The bearing area is $A_b = td = 10(16) = 160$ mm^2. Thus the bearing stress is

$$\sigma = -\frac{F}{A_b} = -\frac{21.0(10)^3}{160} = -131 \text{ MPa} \qquad \text{Answer}$$

(*d*) The critical bending stress in the bar is assumed to occur in a section parallel to the *y* axis and through bolts *A* and *B*. At this section the bending moment is

$$M = 16(300 + 50) = 5600 \text{ N} \cdot \text{m}$$

The second moment of area through this section is obtained by the use of the transfer formula, as follows:

$$I = I_{\text{bar}} - 2(I_{\text{holes}} + \bar{d}^2 A)$$
$$= \frac{15(200)^3}{12} - 2\left[\frac{15(16)^3}{12} + (60)^2(15)(16)\right] = 8.26(10)^6 \text{ mm}^4$$

Then

$$\sigma = \frac{Mc}{I} = \frac{5600(100)}{8.26(10)^6}(10)^3 = 67.8 \text{ MPa} \qquad \text{Answer}$$

SELECTED PROBLEMS

8–19 A 30-mm thick AISI 1020 steel plate is sandwiched between two 10-mm thick 2024-T3 aluminum plates and compressed with a bolt and nut with no washers. The bolt is M10 × 1.5, property class 5.8.

(*a*) Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

(*b*) Determine the bolt stiffness.

(*c*) Determine the stiffness of the members.

8–20 Repeat Prob. 8–19 with the bottom aluminum plate replaced by one that is 20 mm thick.

8–21 Repeat Prob. 8–19 with the bottom aluminum plate having a threaded hole to eliminate the nut.

8–22 Two 20-mm steel plates are to be clamped together with a bolt and nut. Specify a bolt to provide a joint constant C between 0.2 and 0.3.

8–24 An aluminum bracket with a $\frac{1}{2}$-in thick flange is to be clamped to a steel column with a $\frac{3}{4}$-in wall thickness. A cap screw passes through a hole in the bracket flange, and threads into a tapped hole through the column wall. Specify a cap screw to provide a joint constant C between 0.2 and 0.3.

8–25 An M14 × 2 hex-head bolt with a nut is used to clamp together two 20-mm steel plates. Compare the results of finding the overall member stiffness by use of Eqs. (8–20), (8–22), and (8–23).

8–31 For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 1.0$ MN/mm and the stiffness of the members is $k_m = 2.6$ MN/mm per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use M6 × 1 class 5.8 bolts with rolled threads.

(*a*) Determine the maximum external load P_{max} that can be applied to the entire joint without exceeding the proof strength of the bolts.

(*b*) Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

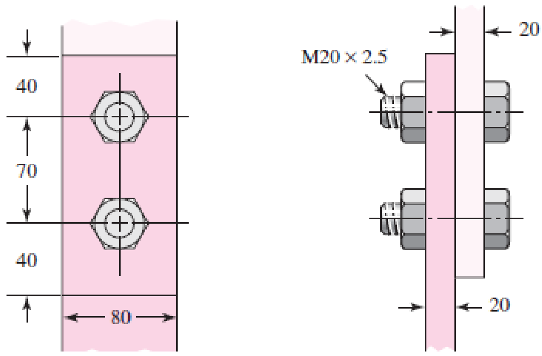
8–32 For a bolted assembly, the stiffness of each bolt is $k_b = 4$ Mlbf/in and the stiffness of the members is $k_m = 12$ Mlbf/in per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\text{max}} = 80$ kips and $P_{\text{min}} = 20$ kips. Assume the load is equally distributed to all the bolts. It has been determined to use $\frac{1}{2}$ in-13 UNC grade 8 bolts with rolled threads.

(*a*) Determine the minimum number of bolts necessary to avoid yielding of the bolts.

(*b*) Determine the minimum number of bolts necessary to avoid joint separation.

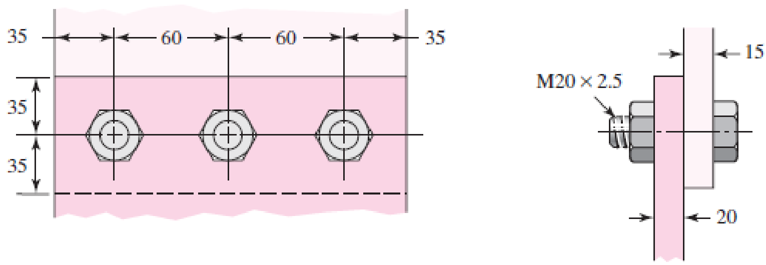
8–68 A bolted lap joint using ISO class 5.8 bolts and members made of cold-drawn SAE 1040 steel is shown in the figure. Find the tensile shear load F that can be applied to this connection to provide a minimum factor of safety of 2.5 for the following failure modes: shear of bolts, bearing on bolts, bearing on members, and tension of members.

Problem 8–68
Dimensions in millimeters.



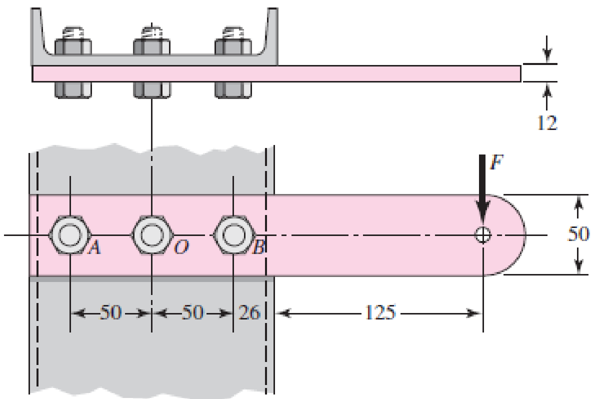
8–69 The bolted connection shown in the figure is subjected to a tensile shear load of 90 kN. The bolts are ISO class 5.8 and the material is cold-drawn AISI 1015 steel. Find the factor of safety of the connection for all possible modes of failure.

Problem 8–69
Dimensions in millimeters.



8–75 A vertical channel 152 × 76 (see Table A–7) has a cantilever beam bolted to it as shown. The channel is hot-rolled AISI 1006 steel. The bar is of hot-rolled AISI 1015 steel. The shoulder bolts are M10 × 1.5 ISO 5.8. For a design factor of 2.0, find the safe force F that can be applied to the cantilever.

Problem 8–75
Dimensions in millimeters.



8–76 The cantilever bracket is bolted to a column with three M12 × 1.75 ISO 5.8 bolts. The bracket is made from AISI 1020 hot-rolled steel. Find the factors of safety for the following failure modes: shear of bolts, bearing of bolts, bearing of bracket, and bending of bracket.

Problem 8–76

Dimensions in millimeters.

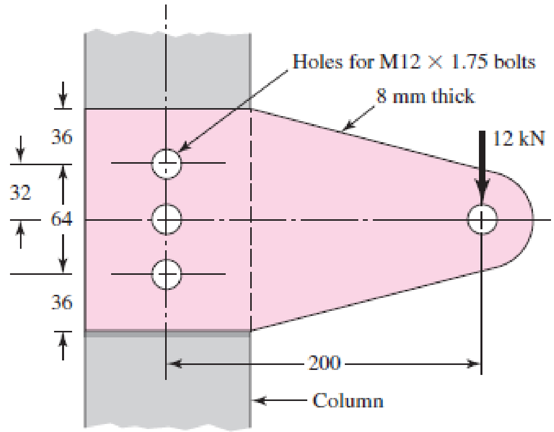


Table A–17 Preferred Sizes (When a choice can be made, use one of these sizes; however, not all parts or items are available in all the sizes shown in the table.)

Decimal Inches

0.010, 0.012, 0.016, 0.020, 0.025, 0.032, 0.040, 0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.24, 0.30, 0.40, 0.50, 0.60, 0.80, 1.00, 1.20, 1.40, 1.60, 1.80, 2.0, 2.4, 2.6, 2.8, 3.0, 3.2, 3.4, 3.6, 3.8, 4.0, 4.2, 4.4, 4.6, 4.8, 5.0, 5.2, 5.4, 5.6, 5.8, 6.0, 7.0, 7.5, 8.5, 9.0, 9.5, 10.0, 10.5, 11.0, 11.5, 12.0, 12.5, 13.0, 13.5, 14.0, 14.5, 15.0, 15.5, 16.0, 16.5, 17.0, 17.5, 18.0, 18.5, 19.0, 19.5, 20

Millimeters

0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.25, 0.30, 0.40, 0.50, 0.60, 0.70, 0.80, 0.90, 1.0, 1.1, 1.2, 1.4, 1.5, 1.6, 1.8, 2.0, 2.2, 2.5, 2.8, 3.0, 3.5, 4.0, 4.5, 5.0, 5.5, 6.0, 6.5, 7.0, 8.0, 9.0, 10, 11, 12, 14, 16, 18, 20, 22, 25, 28, 30, 32, 35, 40, 45, 50, 60, 80, 100, 120, 140, 160, 180, 200, 250, 300

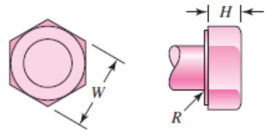
Table A–20

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolledand Cold-Drawn Steels

1 UNS No.	2 SAE and/or AISI No.	3 Processing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170
G10450	1045	HR	570 (82)	310 (45)	16	40	163
		CD	630 (91)	530 (77)	12	35	179
G10500	1050	HR	620 (90)	340 (49.5)	15	35	179
		CD	690 (100)	580 (84)	10	30	197
G10600	1060	HR	680 (98)	370 (54)	12	30	201
G10800	1080	HR	770 (112)	420 (61.5)	10	25	229

Table A–29

Dimensions of Square and Hexagonal Bolts



Nominal Size, mm											
M5	8	3.58	8	3.58	0.2						
M6			10	4.38	0.3						
M8			13	5.68	0.4						
M10			16	6.85	0.4						
M12			18	7.95	0.6	21	7.95	0.6			
M14			21	9.25	0.6	24	9.25	0.6			
M16			24	10.75	0.6	27	10.75	0.6	27	10.75	0.6
M20			30	13.40	0.8	34	13.40	0.8	34	13.40	0.8
M24			36	15.90	0.8	41	15.90	0.8	41	15.90	1.0
M30			46	19.75	1.0	50	19.75	1.0	50	19.75	1.2
M36			55	23.55	1.0	60	23.55	1.0	60	23.55	1.5

Table A–31 Dimensions of Hexagonal Nuts

Nominal Size, mm	Width W	Height H: Regular Hexagonal	Height H: Thick or Slotted	Height H: JAM
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

Table A–33

Dimensions of Metric Plain Washers (All Dimensions in Millimeters)

Washer Size*	Minimum ID	Maximum OD	Maximum Thickness	Washer Size*	Minimum ID	Maximum OD	Maximum Thickness
1.6 N	1.95	4.00	0.70	10 N	10.85	20.00	2.30
1.6 R	1.95	5.00	0.70	10 R	10.85	28.00	2.80
1.6 W	1.95	6.00	0.90	10 W	10.85	39.00	3.50
2 N	2.50	5.00	0.90	12 N	13.30	25.40	2.80
2 R	2.50	6.00	0.90	12 R	13.30	34.00	3.50
2 W	2.50	8.00	0.90	12 W	13.30	44.00	3.50
2.5 N	3.00	6.00	0.90	14 N	15.25	28.00	2.80
2.5 R	3.00	8.00	0.90	14 R	15.25	39.00	3.50
2.5 W	3.00	10.00	1.20	14 W	15.25	50.00	4.00
3 N	3.50	7.00	0.90	16 N	17.25	32.00	3.50
3 R	3.50	10.00	1.20	16 R	17.25	44.00	4.00
3 W	3.50	12.00	1.40	16 W	17.25	56.00	4.60
3.5 N	4.00	9.00	1.20	20 N	21.80	39.00	4.00
3.5 R	4.00	10.00	1.40	20 R	21.80	50.00	4.60
3.5 W	4.00	15.00	1.75	20 W	21.80	66.00	5.10
4 N	4.70	10.00	1.20	24 N	25.60	44.00	4.60
4 R	4.70	12.00	1.40	24 R	25.60	56.00	5.10
4 W	4.70	16.00	2.30	24 W	25.60	72.00	5.60
5 N	5.50	11.00	1.40	30 N	32.40	56.00	5.10
5 R	5.50	15.00	1.75	30 R	32.40	72.00	5.60
5 W	5.50	20.00	2.30	30 W	32.40	90.00	6.40
6 N	6.65	13.00	1.75	36 N	38.30	66.00	5.60
6 R	6.65	18.80	1.75	36 R	38.30	90.00	6.40
6 W	6.65	25.40	2.30	36 W	38.30	110.00	8.50
8 N	8.90	18.80	2.30				
8 R	8.90	25.40	2.30				
8 W	8.90	32.00	2.80				

N = narrow; R = regular; W = wide.
*Same as screw or bolt size.