MECH 344/M

Machine Element Design

Time: M _ _ _ _14:45 - 17:30

Lecture 7

Contents of today's lecture

10

Threaded Fasteners and Power Screws

10.8 Thread Loosening and Thread Locking

- The following are among the factors influencing whether or not threads loosen.

1. The greater the helix angle (i.e., the greater the slope of the inclined plane), the greater the loosening tendency. Thus, coarse threads tend to loosen more easily than fine threads.

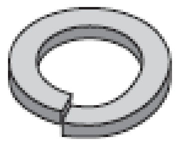
2. The greater the initial tightening, the greater the frictional force that must be overcome to initiate loosening.

3. Soft or rough clamping surfaces tend to promote slight plastic flow which decreases the initial tightening tension and thus promotes loosening.

4. Surface treatments and conditions that tend to increase the friction coefficient provide increased resistance to loosening.

- The problem of thread loosening has resulted in numerous and ingenious special designs and design modifications, and it continues to challenge the engineer to find effective and inexpensive solutions.

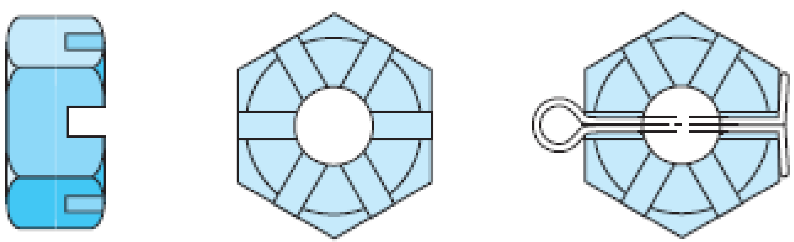
10.8 Thread Loosening and Thread Locking



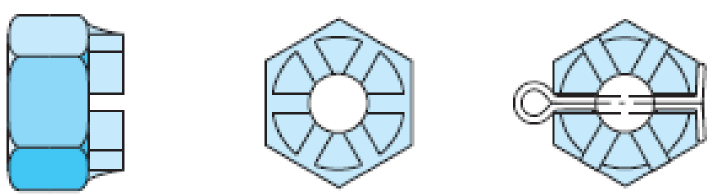
(*a*)
Helical (split) type

(*b*)
Twisted-tooth type
(Teeth may be external,
as in this illustration,
or internal.)

FIGURE 10.20
Common types of lock washers.

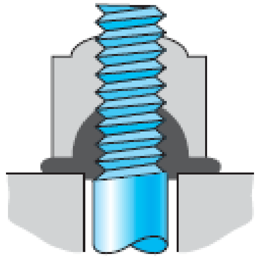


(*a*)



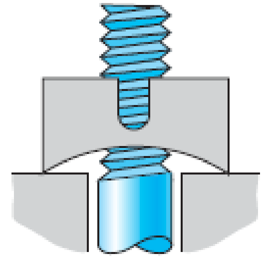
(*b*)

FIGURE 10.21
(*a*) Slotted and (*b*) castle nuts. Each is also shown with a drilled bolt and a cotter pin.



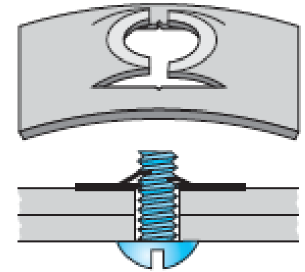
(*a*)

Insert nut (Nylon insert is compressed when nut seats to provide both locking and sealing.)



(*b*)

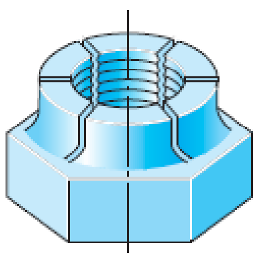
Spring nut (Top of nut pinches bolt thread when nut is tightened.)



(*c*)

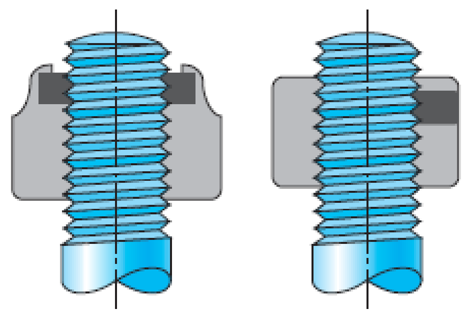
Single thread nut (Prongs pinch bolt thread when nut is tightened. This type of nut is quickly applied and used for light loads.)

Figure 10.22
Examples of free-spinning locknuts.



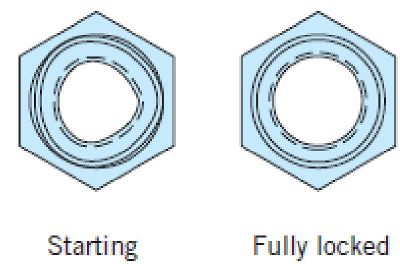
Spring- top nut
(Upper part of nut is tapered.
Segments press against bolt threads.)

(*a*)



Nylon-insert nuts
(Collar or plug of nylon exerts friction grip on bolt threads.)

(*b*)



Distorted nut (Portion of nut is distorted to provide friction grip on bolt threads.)

(*c*)

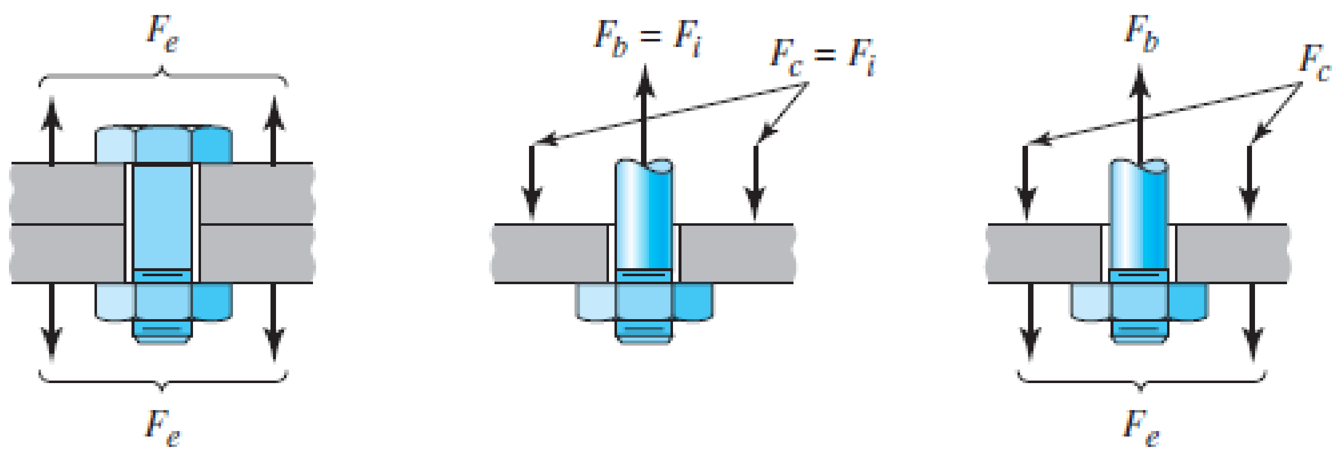
Figure 10.23
Examples of prevailing-torque locknuts. (Courtesy SPS Technologies, Inc.)

10.9 Bolt Tension with External Joint-Separating Force

- Bolts are typically used to hold parts together against to forces that pull, or slide
- Figure 10.24a shows the general case with external force F_e tending to separate
- Figure 10.24b shows a portion of this assembly as a free body. In this figure the nut has been tightened, but the external force has not yet been applied.
- The bolt axial load F_b = clamping force F_c = initial tightening force F_i .
- Figure 10.24c shows after F_e has been applied.
- Equilibrium considerations require one or both of the following:

1. an increase in F_b
2. a decrease in F_c .

- The relative magnitudes of the changes in F_b and F_c depend on the relative elasticities involved.



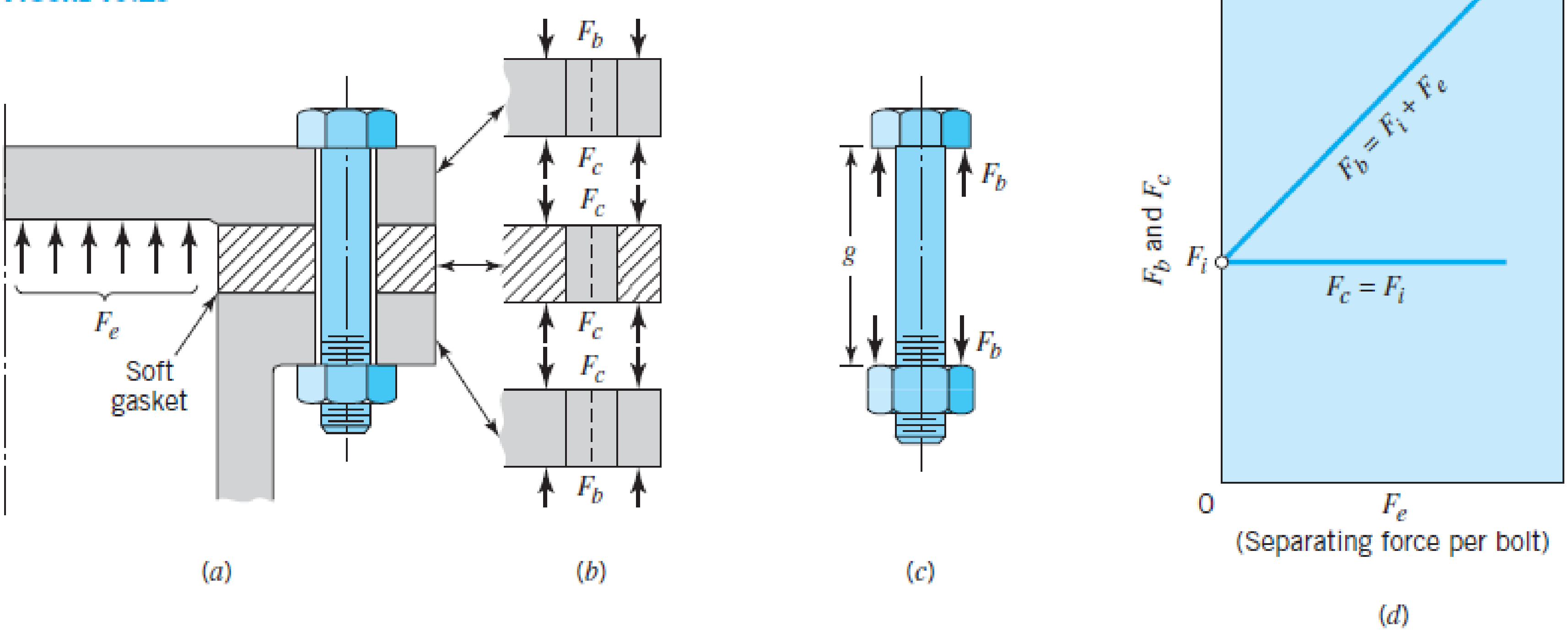
(a) Complete joint (b) Free body without external load (c) Free body with external load

FIGURE 10.24
Free-body study of bolt tensile loading.

10.9 Bolt Tension with External Joint-Separating Force

- Figure 10.25a shows a plate bolted on a pressure vessel with soft gasket so soft that the other parts can be considered infinitely rigid in comparison.
- When the nut is tightened to produce initial force F_i, the rubber gasket compresses; the bolt elongates negligibly.
- Figures 10.25b and 10.25c show details of the bolt and the clamped surfaces. Note the distance defined as the grip g. On initial tightening, $F_b = F_c = F_i$.
- Figure 10.25d shows the change in F_b and F_c as separating load F_e is applied.
- The elastic stretch of the bolt caused by F_e is so small. The clamping force F_c does not diminish and the entire load F_e goes to increasing bolt tension

FIGURE 10.25 F_b and F_c versus F_e per bolt for soft clamped members—rigid bolt.



10.9 Bolt Tension with External Joint-Separating Force

- Figure 10.26 illustrates the clamped members are "rigid" with precision-ground mating surfaces and no gasket, The bolt has a center portion made of rubber.
- Here the initial tightening stretches the bolt; it does not significantly compress the clamped members. (Sealing accomplished by a rubber O-ring).
- Figure 10.26d shows F_e is balanced by reduced F_c without increase in F_b.
- The only way the tension in the rubber bolt can be increased is to increase its length, and this cannot happen without an external force great enough to separate physically the mating clamped surfaces. (Note also that as long as the mating surfaces remain in contact, the sealing of the O-ring is undiminished.)

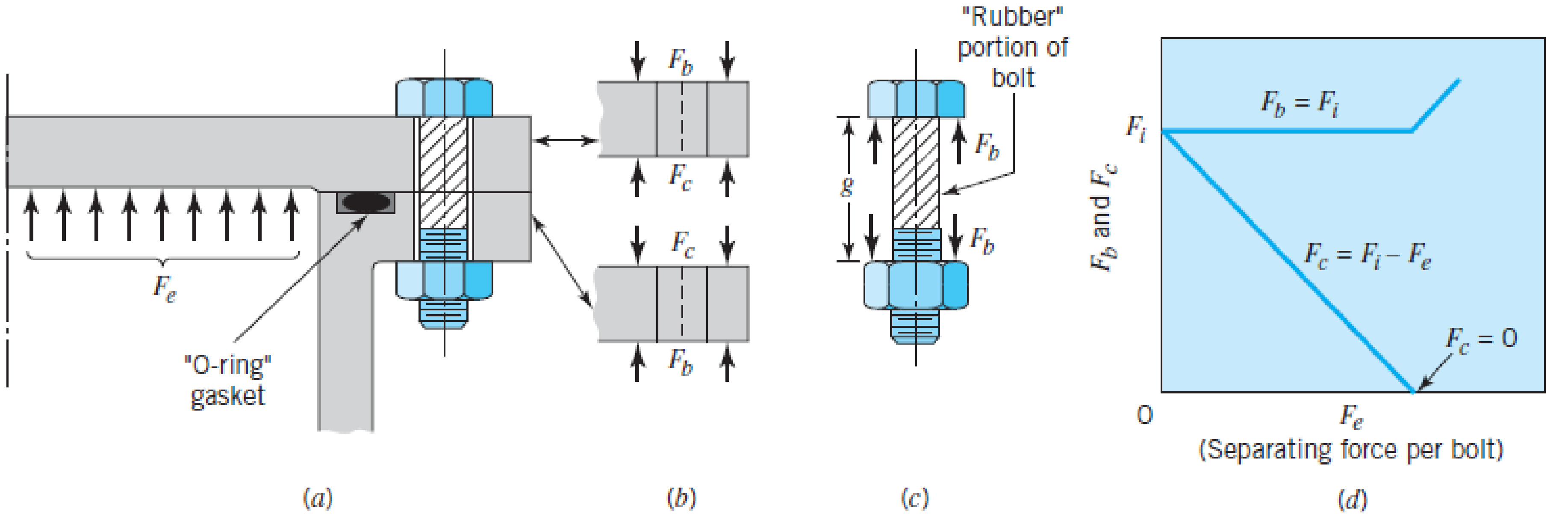


FIGURE 10.26
F_b and F_c versus F_e per bolt for rigid clamped members—soft bolt.

10.9 Bolt Tension with External Joint-Separating Force

- The extreme cases can be only approximated.
- In the realistic case in which both the bolt and the clamped members have applicable stiffness. Joint tightening both elongates the bolt and compresses the clamped members.
- When F_e is applied, the bolt and clamped members elongate by δ (g + δ for both)
- From Figure 10.24 the F_e = increased F_b + the decreased F_c, or

$$F_e = \Delta F_b + \Delta F_c \qquad \textbf{(f)}$$

$$\Delta F_b = k_b\delta \quad \text{and} \quad \Delta F_c = k_c\delta \qquad \textbf{(g)}$$

- Where k_b and k_c are spring constants of bold and clamped material. So substituting

$$F_e = (k_b + k_c)\delta \quad \text{or} \quad \delta = \frac{F_e}{k_b + k_c} \qquad \textbf{(h)}$$

$$\Delta F_b = \frac{k_b}{k_b + k_c}F_e \quad \text{and} \quad \Delta F_c = \frac{k_c}{k_b + k_c}F_e \qquad \textbf{(i)}$$

- From figures 10.25 and 10.26

$$F_b = F_i + \frac{k_b}{k_b + k_c}F_e \quad \text{and} \quad F_c = F_i - \frac{k_c}{k_b + k_c}F_e \qquad \textbf{(10.13)}$$

10.9 Bolt Tension with External Joint-Separating Force

1. When the external load is sufficient to bring the F_c to zero (A), $F_b = F_e$. So figure shows $F_c = 0$ and $F_b = F_e$ for F_e in excess of A.

2. When F_e is alternately dynamic, fluctuations of F_b and F_c can be found from figure

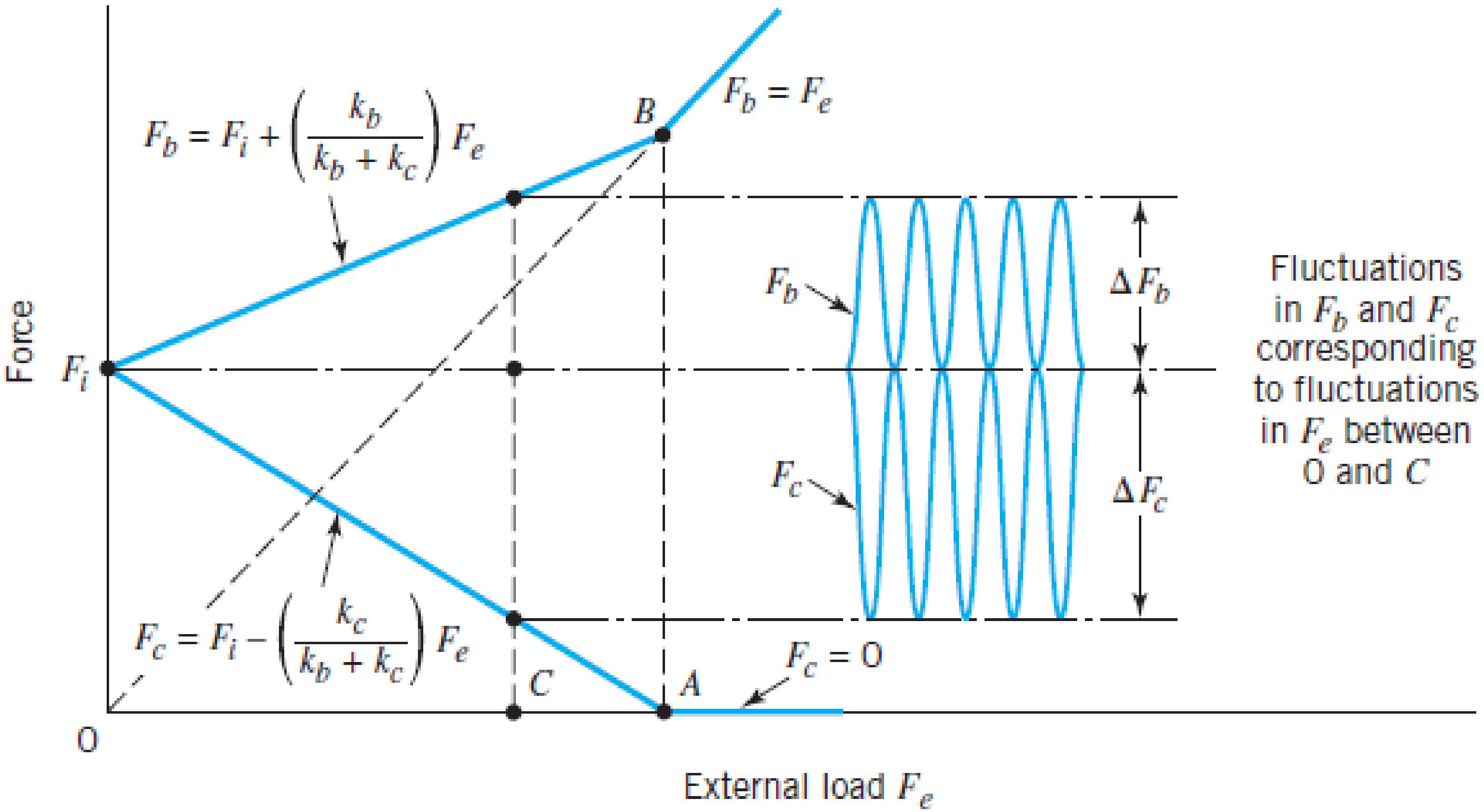


FIGURE 10.27
Force relationships for bolted connections.

$$F_b = F_i + \frac{k_b}{k_b + k_c} F_e \quad \text{and} \quad F_c = F_i - \frac{k_c}{k_b + k_c} F_e \qquad (10.13)$$

10.9 Bolt Tension with External Joint-Separating Force

- We need k_b and k_c. From the basic axial deflection ($\delta = PL/AE$) and for spring rate ($k = P/\delta$)

$$k_b = \frac{A_b E_b}{g} \quad \text{and} \quad k_c = \frac{A_c E_c}{g} \qquad \textbf{(10.14)}$$

- where the grip g represents the effective length for both. Two difficulties that commonly arise in estimating k_c are

1. The clamped members may consist of a stack of different materials, representing "springs" in series. For this case,

$$1/k = 1/k_1 + 1/k_2 + 1/k_3 + \cdots \qquad \textbf{(10.15)}$$

2. The effective CSA of the clamped members is not easy to determine. (irregular shapes, or if they extend a substantial distance from the bolt axis) An empirical procedure sometimes used to estimate A_c is illustrated in Figure.

- One method for estimating the effective area of clamped members (for calculating k_c). Effective area A_c is approximately equal to the average area of the dark grey section.

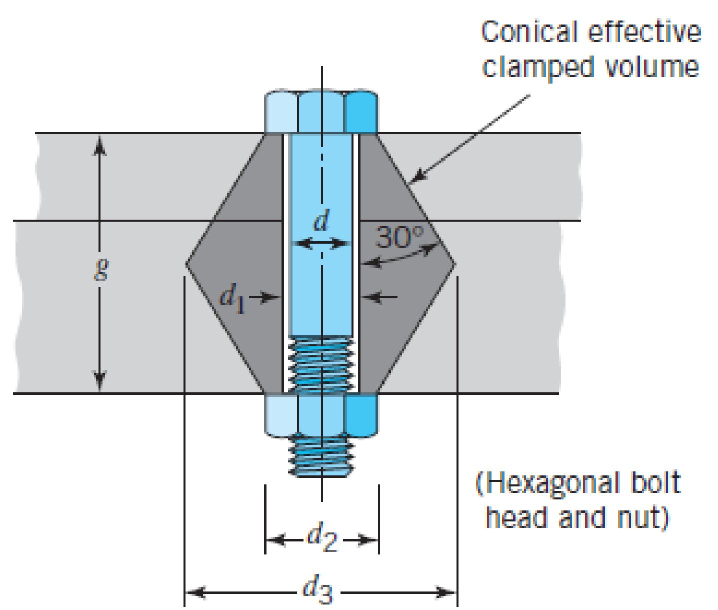


FIGURE 10.28

10.9 Bolt Tension with External Joint-Separating Force

$$A_c = \frac{\pi}{4}\left[\left(\frac{d_3 + d_2}{2}\right)^2 - d_1^2\right]$$

$d_1 \approx d$ (for small clearances)

$d_2 = 1.5d$ (for standard hexagonal-head bolts—see Figure 10.16)

$d_3 = d_2 + g \tan 30° = 1.5d + g \tan 30°$

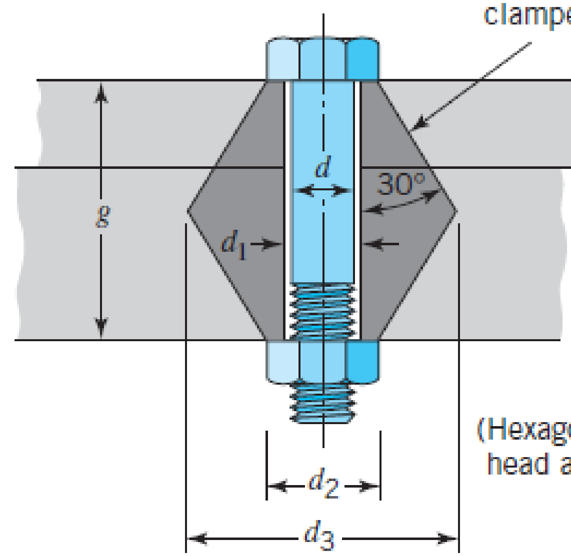


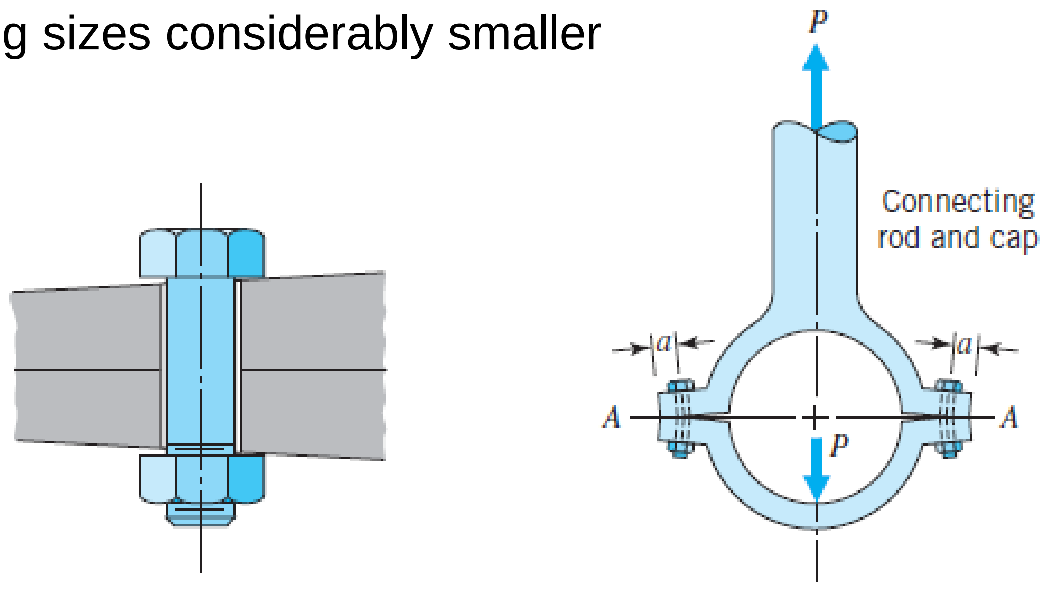
FIGURE 10.28

$$A_c = \frac{\pi}{16}(5d^2 + 6dg \tan 30° + g^2 \tan^2 30°) \approx d^2 + 0.68dg + 0.065g^2$$

- An effective experimental procedure for determining the ratio of k_b and k_c for a given joint is to use a bolt equipped with an electric-resistance strain gage or to monitor bolt length ultrasonically.
- This permits a direct measurement of F_b both before and after F_e is applied.
- Some handbooks contain rough estimates of the ratio k_c/k_b for various general types of gasketed and ungasketed joints.
- For a "typical" ungasketed joint, k_c is sometimes taken as 3 k_b, but with careful joint design $k_c = 6k_b$.

10.10 Bolt (or Screw) Selection for Static Loading

- The primary loading applied to bolts is tensile, shear, or a combination of the two.
- Some bending is usually present because the clamped surfaces are not exactly parallel to each other and perpendicular to the bolt axis (Figure 10.29a) and because the loaded members are somewhat deflected (Figure 10.29b).
- Most times screws and bolts are selected rather arbitrarily. Such is the case with noncritical applications with small loads
- Almost any size would do, including sizes considerably smaller than the ones used.
- Selection is a matter of judgment, based on factors such as appearance, ease of handling and assembly, and cost.
- Even in bolt applications with known significant loads, larger bolts than necessary are used because a smaller size “doesn’t look right,” and the cost penalty of using the larger bolts is minimal.



(a) Bolt bending caused by nonparallelism of mating surfaces. (Bolt will bend when nut is tightened.)

(b) Bolt bending caused by deflection of loaded members. (Note tendency to pivot about A; hence, bending is reduced if dimension a is increased.)

FIGURE 10.29
Examples of nonintended bolt bending.

SAMPLE PROBLEM 10.2D

Select Screws for Pillow Block Attachment—Tensile Loading

Figure 10.30 shows a ball bearing encased in a "pillow block" and supporting one end of a rotating shaft. The shaft applies a static load of 9 kN to the pillow block, as shown. Select appropriate metric (ISO) screws for the pillow block attachment and specify an appropriate tightening torque.

SOLUTION

Known: A known static tensile load is applied to two metric (ISO) screws.

Find: Select appropriate screws and specify a tightening torque.

Schematic and Given Data:

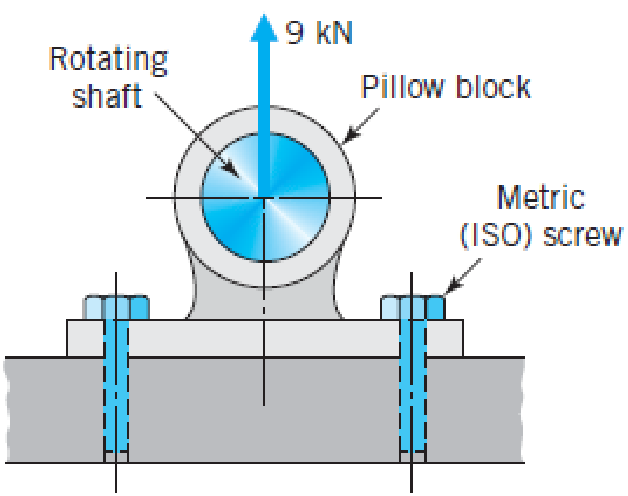


FIGURE 10.30
Pillow block attached by two machine screws.

Decisions/Assumptions:

1. A relatively inexpensive class 5.8 steel is chosen for the screw material.
2. The load of 9 kN is shared equally by each screw.
3. No bending of the machine screws (bolts) takes place; that is, the bolt load is axial tension.

Design Analysis:

1. Any class of steel could have been used, but there appears no reason to specify a costly high-strength steel. Class 5.8, with a proof strength of 380 MPa (Table 10.5), was chosen.
2. The nominal load for each of the two bolts is 4.5 kN. Reference to Section 6.12 indicates that if screw failure would not endanger human life, cause other damage, or entail costly shutdown, a safety factor of 2.5 would be reasonable. Since in this case the cost of using a larger safety factor is trivial, and since failure might prove rather costly, let us use "engineering judgment" and increase the safety factor to 4. Then, the "design overload" for each bolt is 4.5 kN $\times$ 4, or 18 kN.

10.6 Fastener Materials and Methods of Manufacture

Table 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

3. For static loading of a ductile material, stress concentration can be neglected and the simple "$\sigma = P/A$" equation used, with σ being equal to the proof strength when P is equal to the design overload:

$$380 \text{ MPa} = \frac{18{,}000\ N}{A_t} \quad \text{or} \quad A_t = 47.4 \text{ mm}^2$$

4. Reference to Table 10.2 indicates an appropriate standard size of class 5.8 screw to be M10 × 1.5 (for which $A_t = 58.0 \text{ mm}^2$).

5. Initial tightening tension might reasonably be specified (Eq. 10.11a) as

$$F_i = 0.9A_tS_p = 0.9(58.0 \text{ mm}^2)(380 \text{ MPa}) = 19{,}836 \text{ N}$$

6. This corresponds to an estimated tightening torque (Eq. 10.12) of

$$T = 0.2F_id = 0.2(19.8 \text{ kN})(10 \text{ mm}) = 39.6 \text{ N} \cdot \text{m}$$

$$F_i = K_iA_tS_p \qquad \textbf{(10.11)}$$

For ordinary applications K_i can be 0.9

$$F_i = 0.9A_tS_p \qquad \textbf{(10.11a)}$$

$$T = 0.2F_id \qquad \textbf{(10.12)}$$

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

The *shear strengths* of steel bolts of various grades was studied by Fisher and Struik [5], who concluded that a reasonable approximation is

$$S_{us} \approx 0.62 S_u \text{ For direct (not torsional) shear loading.} \tag{10.16}$$

Sample Problem 10.3 Determine Shear Load Capacity of a Bolted Joint

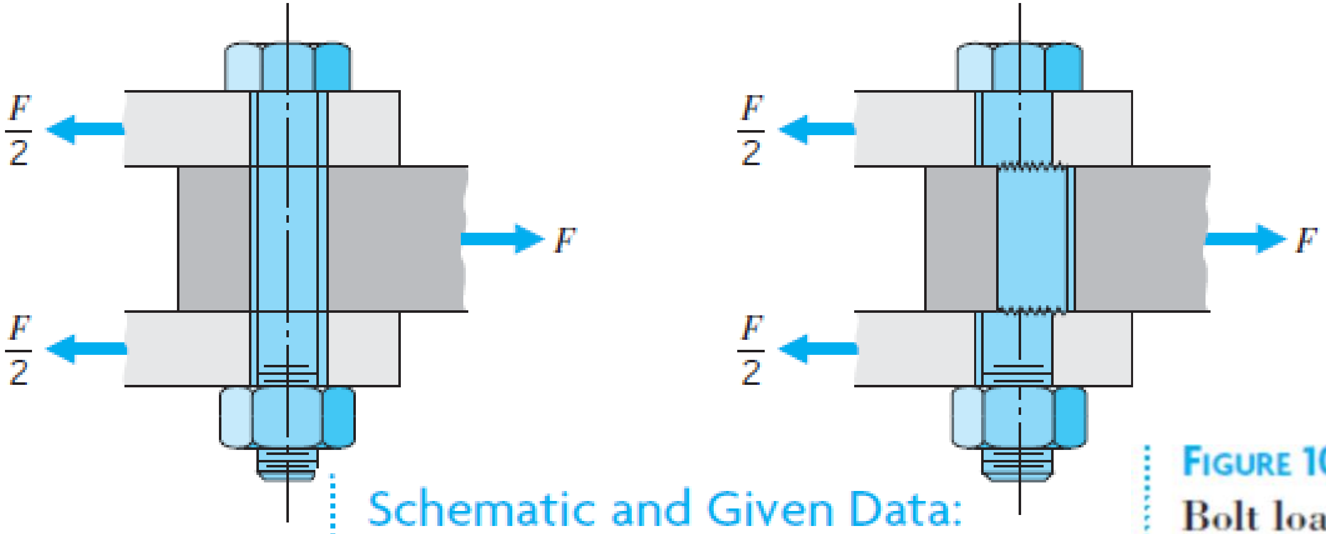
Figure 10.31 shows a $\frac{1}{2}$ in.–13UNC grade 5 steel bolt loaded in double shear (i.e., the bolt has two shear planes, as shown). The clamped plates are made of steel and have clean and dry surfaces. The bolt is to be tightened with a torque wrench to its full proof load; that is, $F_i = S_p A_t$. What force F is the joint capable of withstanding? (Note: This double shear bolt loading is the same as that on the pin in Figure 2.14. It is assumed that the bolt and plates have adequate strength to prevent the other failure modes discussed in connection with Figures 2.14 and 2.15.)

Solution

Known: A specified steel bolt clamps three steel plates and is loaded in double shear.

Find: Determine the force capacity of the joint.

Schematic and Given Data:



(a) Normal load, carried by friction forces

(b) Overload, causing shear failure

FIGURE 10.31
Bolt loaded in double shear.

Assumptions:

1. The bolt is tightened to its full proof load; that is, $F_i = S_p A_t$.
2. The bolt fails in double shear.
3. The bolt and plates have adequate strength to prevent other failure modes.
4. The wrench-torque variation is roughly ±30 percent.
5. There is a 10 percent initial loss in tension during the first few weeks of service (see Section 10.7).

Analysis:

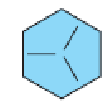
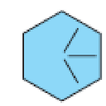
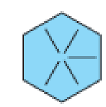
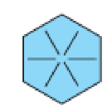
1. For the $\frac{1}{2}$ in.–13UNC grade 5 steel bolt, Table 10.1 gives $A_t = 0.1419$ in.^{2} and Table 10.4 shows that $S_p = 85$ ksi. Specified initial tension is $F_i = S_p A_t = 85{,}000 \text{ psi} \times 0.1419 \text{ in.}^2 = 12{,}060$ lb. But with a roughly estimated ±30 percent torque-wrench variation and 10 percent initial-tension loss during the first few weeks of service (see Section 10.7), a conservative assumption of working value of F_i is about 7600 lb.
2. Reference 5 gives a summary (p. 78) of friction coefficients obtained with bolted plates. The coefficient for semipolished steel is approximately 0.3, and for sand or grit-blasted steel approximately 0.5. Various paints, platings, and other surface treatments can alter the coefficient markedly, usually downward. Here a friction coefficient of 0.4 is assumed. This gives a force required to slip each of the two interfaces of 7600 lb $\times$ 0.4 $=$ 3040 lb. Thus, the value of F required to overcome friction is estimated to be in the region of 6000 lb.
3. Although it is often desirable to limit applied load F to the value that can be transmitted by friction, we should know the larger value of force that can be transmitted through the bolt itself. For the two shear planes involved, this force is equal to $2S_{sy}A$, where A is the area of the bolt *at the shear planes*—in this case, $\pi(0.5)^2/4 = 0.196$ in.^{2}. Taking advantage of the fact that the distortion energy theory gives a good estimate of shear yield strength for ductile metals, we have $S_{sy} = 0.58S_y = 0.58(92 \text{ ksi}) = 53$ ksi. Thus, for yielding of the two shear planes, $F = 2(0.196 \text{ in.}^2)(53{,}000 \text{ psi}) = 21{,}000$ lb.

TABLE 10.1 Basic Dimensions of Unified Screw Threads

		Coarse Threads—UNC			Fine Threads—UNF		
Size	Major Diameter d (in.)	Threads per Inch	Minor Diameter of External Thread d_r (in.)	Tensile Stress Area A_t (in.^{2})	Threads per Inch	Minor Diameter of External Thread d_r (in.)	Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599

10.6 Fastener Materials and Methods of Manufacture

Table 10.4 Specifications for Steel Used in Inch Series Screws and Bolts

SAE Grade	Diameter d (in.)	Proof Load (Strength)[a] S_p (ksi)	Yield Strength[b] S_y (ksi)	Tensile Strength S_u (ksi)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max	Grade Identification Marking on Bolt Head
1	$\frac{1}{4}$ thru $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
2	$\frac{1}{4}$ thru $\frac{3}{4}$	55	57	74	18	35	B80	B100	None
2	Over $\frac{3}{4}$ to $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
5	$\frac{1}{4}$ thru 1	85	92	120	14	35	C25	C34	
5	Over 1 to $1\frac{1}{2}$	74	81	105	14	35	C19	C30	
5.2	$\frac{1}{4}$ thru 1	85	92	120	14	35	C26	C36	
7	$\frac{1}{4}$ thru $1\frac{1}{2}$	105	115	133	12	35	C28	C34	
8	$\frac{1}{4}$ thru $1\frac{1}{2}$	120	130	150	12	35	C33	C39	

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J429k (1979).

4. The estimated 21,000-lb load would bring the shear stress to the yield strength over the entire cross section of the shear planes, and the very small amount of yielding would probably result in losing most or all of the clamping and friction forces. A further increase in load would cause total shear failure, as indicated in Figure 10.31*b*. This total failure load is calculated as in step 3, except for replacing S_{sy} with S_{us}. From Eq. 10.16, $S_{us} \approx 74$ ksi; the corresponding estimated load is $F = 29{,}000$ lb.

Comment: Note that in Figure 10.31 the threaded portion of the bolt does *not* extend to the shear plane. This is important for a bolt loaded in shear. Extending the thread to the shear plane is conservatively considered to reduce the shear area to a circle equal to the thread root diameter; in this case, $A = \pi(0.4056)^2/4 = 0.129$ in.^{2}, which is a reduction of 34 percent.

SAMPLE PROBLEM 10.4 Select Bolts for Bracket Attachment, Assuming Shear Carried by Friction

Figure 10.32 shows a vertically loaded bracket attached to a fixed member by three identical bolts. Although the 24-kN load is normally applied in the center, the bolts are to be selected on the basis that the load eccentricity shown could occur. Because of safety considerations, SAE class 9.8 steel bolts and a minimum safety factor of 6 (based on proof strength) are to be used. Determine an appropriate bolt size.

SOLUTION

Known: Three SAE class 9.8 steel bolts with a specified safety factor are used to attach a bracket of known geometry that supports a known vertical load.

Find: Determine an appropriate bolt size.

Schematic and Given Data:

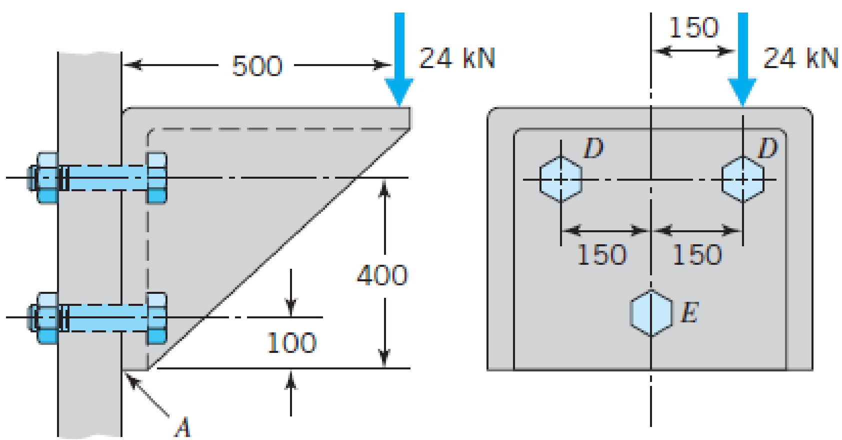


FIGURE 10.32
Vertically loaded bracket supported by three bolts.

Assumptions:

1. The clamped members are rigid and do not deflect with load.
2. The load tends to rotate the bracket about an axis through point A.
3. The shear loads are carried by friction.

Analysis:

1. With the assumptions of rigid clamped members and shear loads carried by friction, the eccentricity of the applied load has no effect on bolt loading. With the bracket tending to rotate about an axis through point A, the strain (and hence the load) imposed upon the two bolts D is four times that imposed upon bolt E. Let F_D and F_E denote the tensile loads carried by bolts D and E. Summation of moments about point A for the *design overload* of 24 kN(6) = 144 kN gives

$$500(144) = 100F_E + 400F_D + 400F_D$$
$$= 25F_D + 400F_D + 400F_D = 825F_D$$

or

$$F_D = 87.27 \text{ kN}$$

2. Class 9.8 steel has a proof strength of 650 MPa. Hence the required tensile stress area is

$$A_t = \frac{87{,}270 \text{ N}}{650 \text{ MPa}} = 134 \text{ mm}^2$$

Reference to Table 10.2 indicates the required thread size to be M16 × 2.

10.6 Fastener Materials and Methods of Manufacture

TABLE 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

TABLE 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

Note: Metric threads are identified by diameter and pitch as "M8 × 1.25."

Comments:

1. Because of appearance, and to provide additional safety, a larger bolt size might be selected.
2. As in Sample Problem 10.2, the bolt size required is independent of k_b, k_c, and F_i, *except* for the fact that F_i must be large enough to justify the assumption that shear forces are transmitted by friction. With an assumed coefficient of friction of 0.4 and an initial tension (after considering tightening variations and initial relaxation) of at least $0.55S_pA_t$, compare the available shear friction force (using 16-mm bolts) with the applied shear overload:

$$\begin{aligned}\text{Available friction force} &= (3\text{ bolts})(0.55\,S_pA_t)F \\ &= 3(0.55)(650\text{ MPa})(157.27\text{ mm}^2)(0.4) \\ &= 67{,}500\text{ N}\end{aligned}$$

which represents a margin of safety with respect to the 24-kN applied overload, plus the rotational tendency caused by the overload eccentricity. The second effect is dealt with in Sample Problem 10.5.

SAMPLE PROBLEM 10.5

Select Bolts for Bracket Attachment, Neglecting Friction and Assuming Shear Forces Are Carried by the Bolts

Repeat Sample Problem 10.4, except neglect the frictional forces.

SOLUTION

Known: Three SAE class 9.8 steel bolts having a specified safety factor are used to attach a bracket of known geometry that supports a known vertical load.

Find: Select an appropriate bolt size.

Schematic and Given Data: See Sample Problem 10.4 and Figure 10.32.

Assumptions:

1. The shear forces caused by the eccentric vertical load are carried completely by the bolts.
2. The vertical shear load is distributed equally among the three bolts.
3. The tangential shear force carried by each bolt is proportional to its distance from the center of gravity of the group of bolts.

Analysis:

1. Neglecting friction has no effect on bolt stresses in the *threaded region*, where attention was focused in Sample Problem 10.4. For this problem attention is shifted to the *bolt shear plane* (at the interface between bracket and fixed plate). This plane experiences the tensile force of 87.27 kN calculated in Sample Problem 10.4 in addition to the shear force calculated in the following step 2.
2. The applied eccentric shear force of 24 kN(6) = 144 kN tends to displace the bracket downward and also rotate it clockwise about the center of gravity of the bolt group cross section. For three bolts of equal size, the center of gravity corresponds to the centroid of the triangular pattern, as shown in Figure 10.33.

Schematic and Given Data:

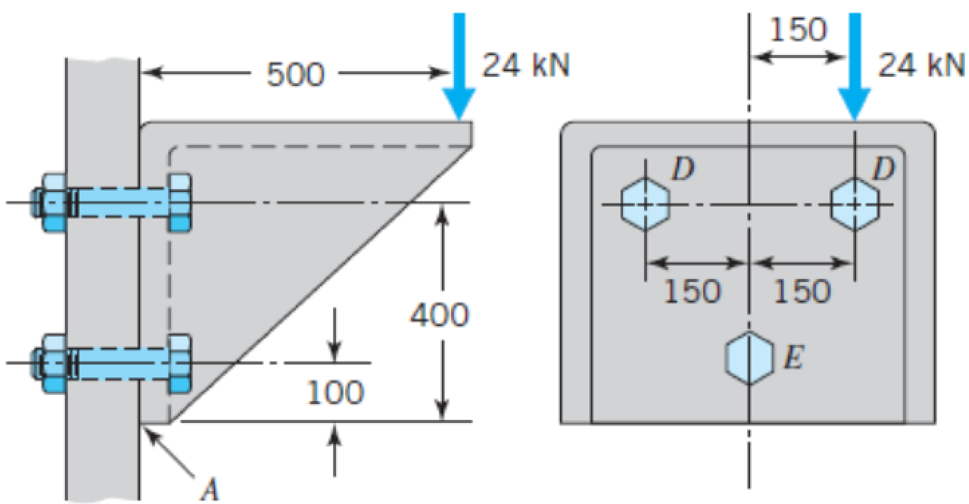
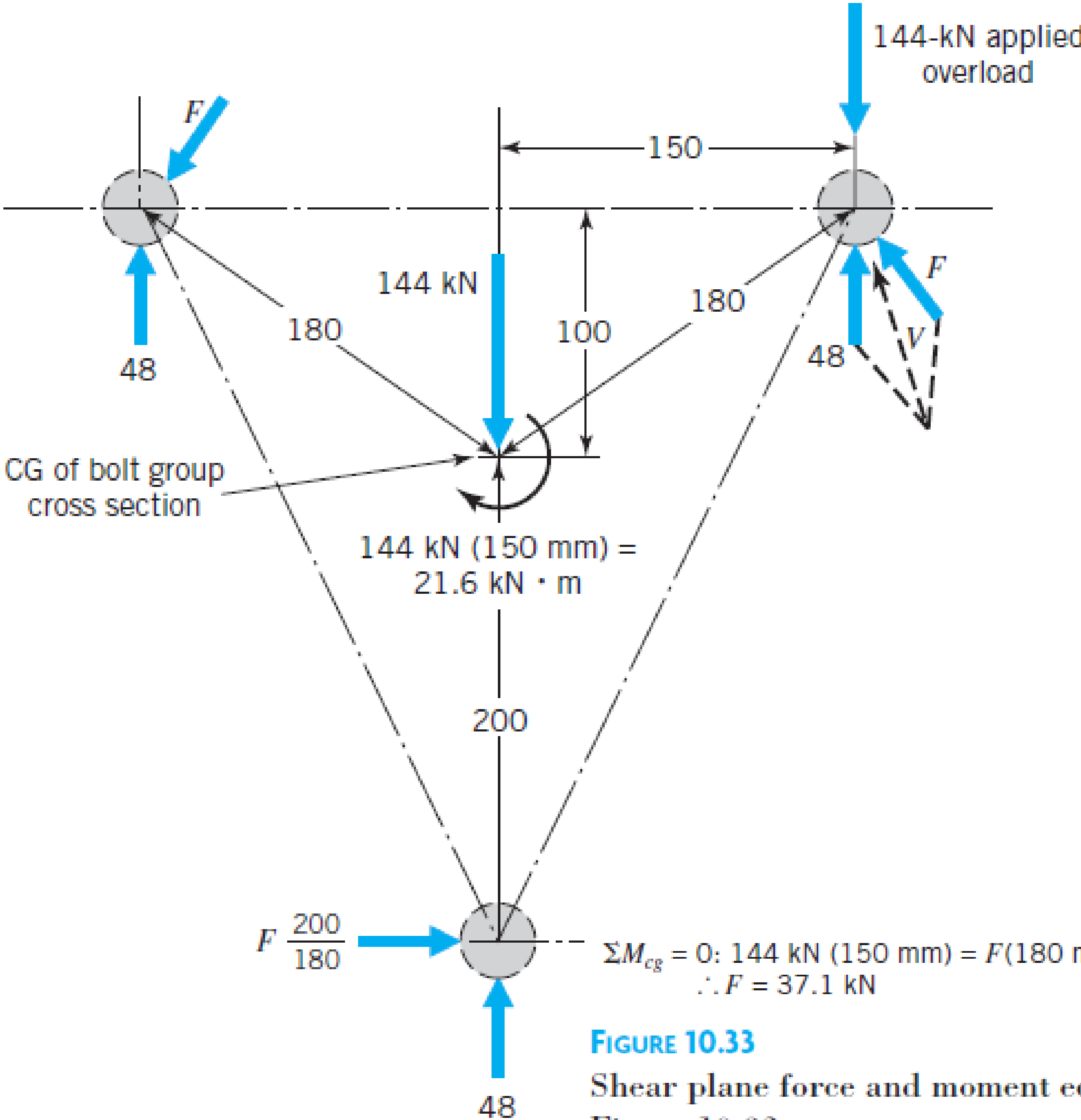


FIGURE 10.32
Vertically loaded bracket supported by three bolts.



$\Sigma M_{cg} = 0$: 144 kN (150 mm) = F(180 mm) + F(180 mm) + $F(\frac{200}{180}$ mm) (200 mm)
$\therefore F$ = 37.1 kN

FIGURE 10.33
Shear plane force and moment equilibrium for the bracket in Figure 10.32.

This figure shows the original applied load (dotted vector) replaced by an equal load applied at the centroid (solid vector) plus a torque that is equal to the product of the force and the distance it was moved. As assumed, each bolt carries one-third of the vertical shear load, plus a tangential force (with respect to rotation about the center of gravity) that is proportional to its distance from the center of gravity. Calculations on the figure show this tangential force to be 37.1 kN for each of the top bolts. The vector sum of the two shear forces is obviously greatest for the upper right bolt. Routine calculation shows $V = 81.5$ kN.

3. The critical upper right bolt is thus subjected to a tensile stress, $\sigma = 87{,}270/A$, and a shear stress, $\tau = 81{,}500/A$. Substitution in the distortion energy equation gives an equivalent tensile stress of

$$\sigma_e = \sqrt{\sigma^2 + 3\tau^2} = \frac{1}{A}\sqrt{(87{,}270)^2 + 3(81{,}500)^2} = \frac{166{,}000}{A}$$

4. Equating this to the proof stress gives

$$\frac{166{,}000}{A} = S_p = 650 \text{ MPa}$$

Therefore,

$$A = 255 \text{ mm}^2$$

10.6 Fastener Materials and Methods of Manufacture

Table 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34
9.8	1.6 thru 16	650	—	900	—	—	C27	C36
10.9	6 thru 36	830	940	1040	9	35	C33	C39
12.9	1.6 thru 36	970	1100	1220	8	35	C38	C44

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.
[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.
Source: Society of Automotive Engineers standard J1199 (1979).

5. Finally,

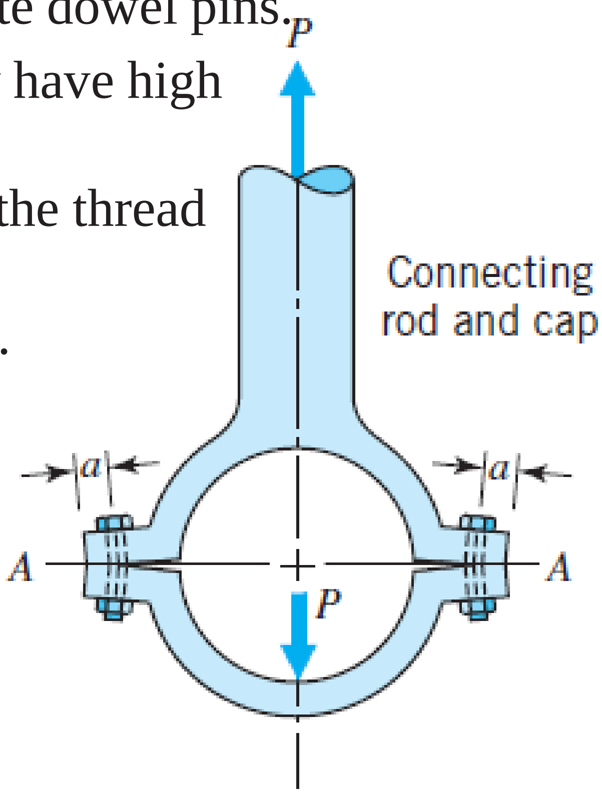
$$A = \frac{\pi d^2}{4}, \qquad \text{or} \quad d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(255)}{\pi}} = 18.03 \text{ mm}$$

Thus, a *shank* diameter of 18 mm is required.

Comment: In comparing this solution with that of Sample Problem 10.4, note that *for this particular case*, shear plus tension in the bolt shear plane proved to be more critical than tension alone in the threads.

10.11 Bolt (or Screw) Selection for Fatigue Loading: Fundamentals

- Bolt fatigue involves fluctuating *tension*, and some small alternating bending (as in Figure).
- Alternating shear loads are usually reacted by separate dowel pins.
- Because of initial tightening tension, bolts inherently have high mean stresses.
- In addition, stress concentration is always present at the thread roots.
- These two points are treated in Sections 8.9 and 8.11.
- Table 10.6 gives approximate K_f for standard screws and bolts.
- (1) rolled threads have lower K_f because of work hardening and residual stresses
- (2) hardened threads have higher K_f because of their greater notch sensitivity.
- For thread finishes of good commercial quality, these values may be used with a surface factor C_S of unity.



(b)

Bolt bending caused by deflection of loaded members. (Note tendency to pivot about A; hence, bending is reduced if dimension a is increased.)

10.11 Bolt (or Screw) Selection for Fatigue Loading: Fundamentals

TABLE 10.6 Fatigue Stress Concentration Factors K_f for Steel Threaded Members (Approximate Values for Unified and ISO Threads)

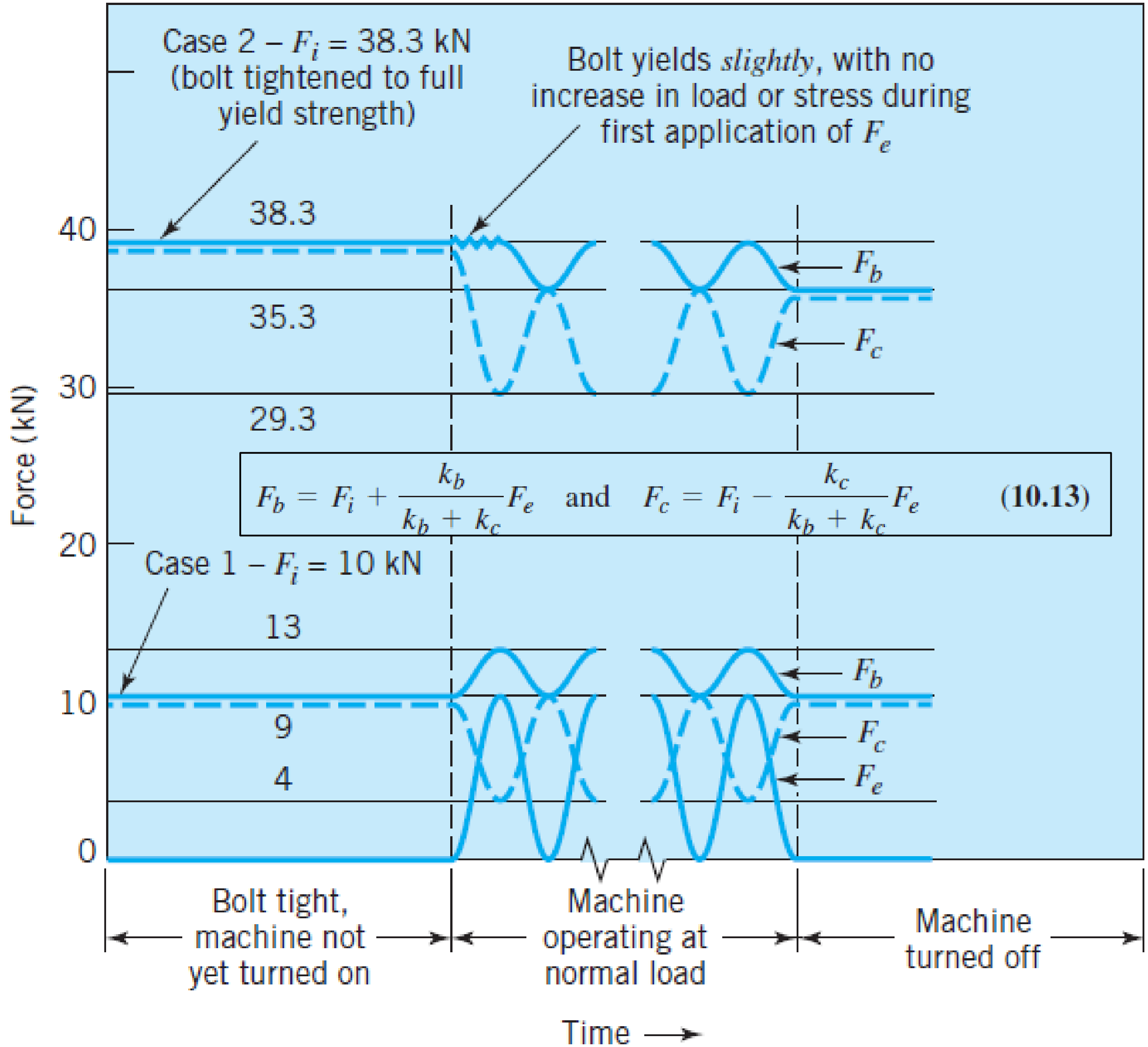
Hardness	SAE Grade (Unified Threads)	SAE Class (ISO Threads)	K_f[a] Rolled Threads	K_f[a] Cut Threads
Below 200 Bhn (annealed)	2 and below	5.8 and below	2.2	2.8
Above 200 Bhn (hardened)	4 and above	8.8 and above	3.0	3.8

[a] With good commercial surfaces, use $C_s = 1$ (rather than a value from Fig. 8.13) when using these values of K_f.

- We have seen the tendency for most of the bolt load to be carried by the threads nearest to the loaded face of the nut, and the degree to which the stresses were concentrated in this region was influenced by the nut design.
- This is one reason why actual values of K_f may differ from those given in Table 10.6.

10.11.1 Analysis of Bolt Fatigue Strength at High and Low Initial Tightening Tension

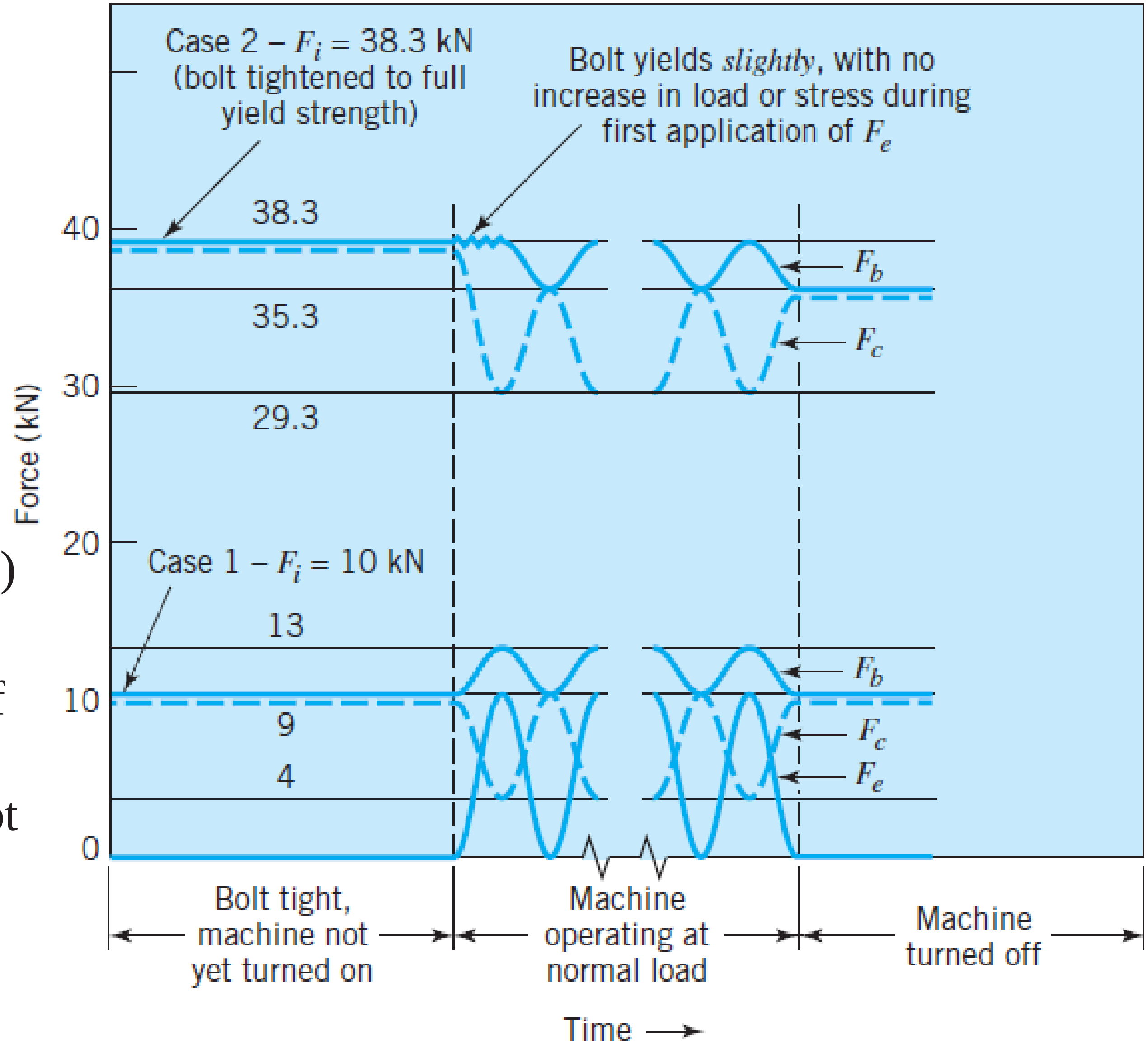
- Fatigue strength of a steel M10 * 1.5 bolt class 8.8 installed in a joint with $k_c = 2k_b$, and subjected to an external load which fluctuates between 0 and 9 kN.
- Curves correspond to F_i 10 kN.
- Each time F_e is applied, F_b increases and F_c decreases, with the sum of the two effects being equal to F_e of 9 kN.
- When F_e is removed, F_b and F_c revert to their initial value of 10 kN.



(*a*) Fluctuation in F_b and F_c caused by fluctuations in F_e

10.11.1 Analysis of Bolt Fatigue Strength at High and Low Initial Tightening Tension

- Fatigue strength of a steel M10 * 1.5 bolt class 8.8 installed in a joint with $k_c = 2k_b$, and subjected to an external load which fluctuates between 0 and 9 kN.
- In case 2, curves are for $F_i = S_y A_t = 660 * 58.0 = 38.3$ kN) (table 10.2 & 5)
- With the F_i = yield strength, the bolt CSA of A_t is stressed to S_y, first application of F_e does not increase F_b as no elongation possible.
- So all of it is reacted by decrease in F_c



(a) Fluctuation in F_b and F_c caused by fluctuations in F_e

10.11.1 Analysis of Bolt Fatigue Strength at High and Low Initial Tightening Tension

Table 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03
3.5	0.6	2.76	6.78
4	0.7	3.14	8.78
5	0.8	4.02	14.2
6	1	4.77	20.1
7	1	5.77	28.9
8	1.25	6.47	36.6
10	1.5	8.16	58.0

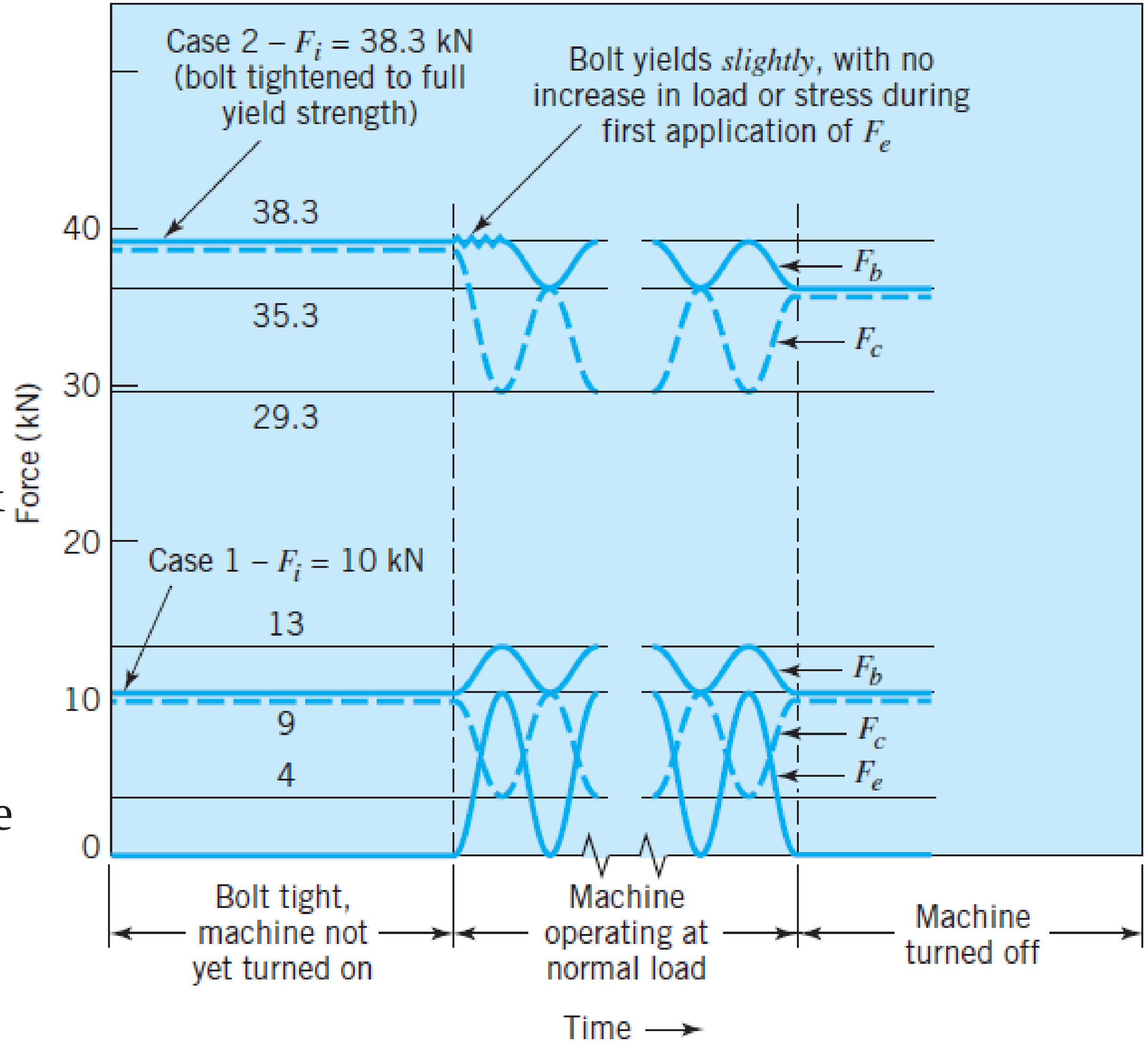
Table 10.5 Specifications for Steel Used in Millimeter Series Screws and Bolts

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)	Tensile Strength S_u (MPa)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max
4.6	5 thru 36	225	240	400	22	35	B67	B87
4.8	1.6 thru 16	310	—	420	—	—	B71	B87
5.8	5 thru 24	380	—	520	—	—	B82	B95
8.8	17 thru 36	600	660	830	12	35	C23	C34

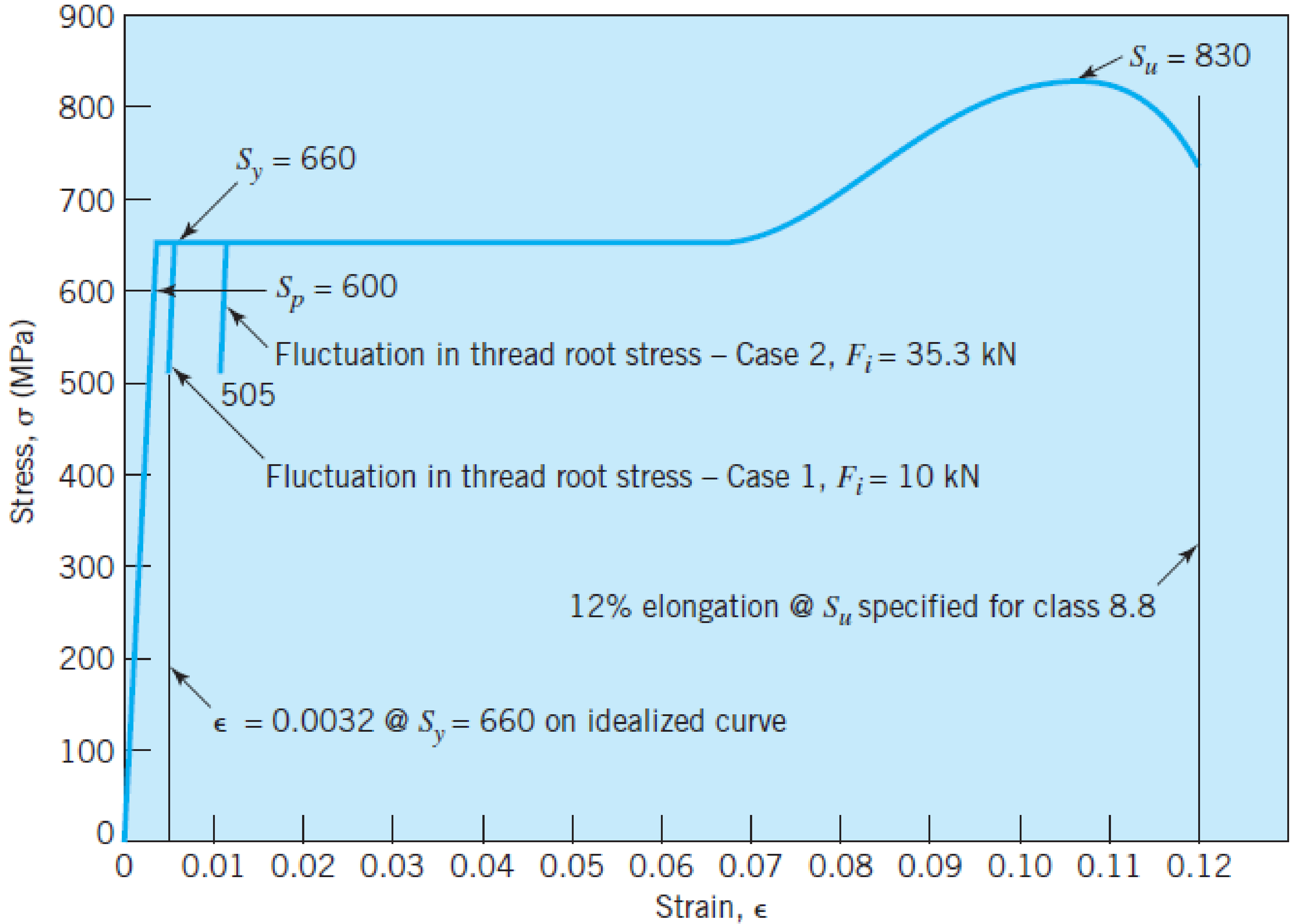
10.11.1 Analysis of Bolt Fatigu and Low Initial Tighter

$$\sigma = \frac{F_i}{A_t} K_f = \frac{10{,}000\ \text{N}}{58.0\ \text{mm}^2}(3) = 517\ \text{MPa}$$

- When F_e is released, bolt relaxes slightly, and this relaxation is elastic.
- Hence, changes in F_b & F_c are controlled by k_b & k_c .
- Elastic bolt relaxation is reversible, and the load can be reapplied without yielding. As F_e cycles, F_b & F_c fluctuate.
- Stress on the case 1 is 517MPa for 10KN and 672MPa for 13KN. Since yielding happens at 660, the max stress stays at 660.
- When the F_b comes back to 10KN the elastic difference is 155KN (672-517)



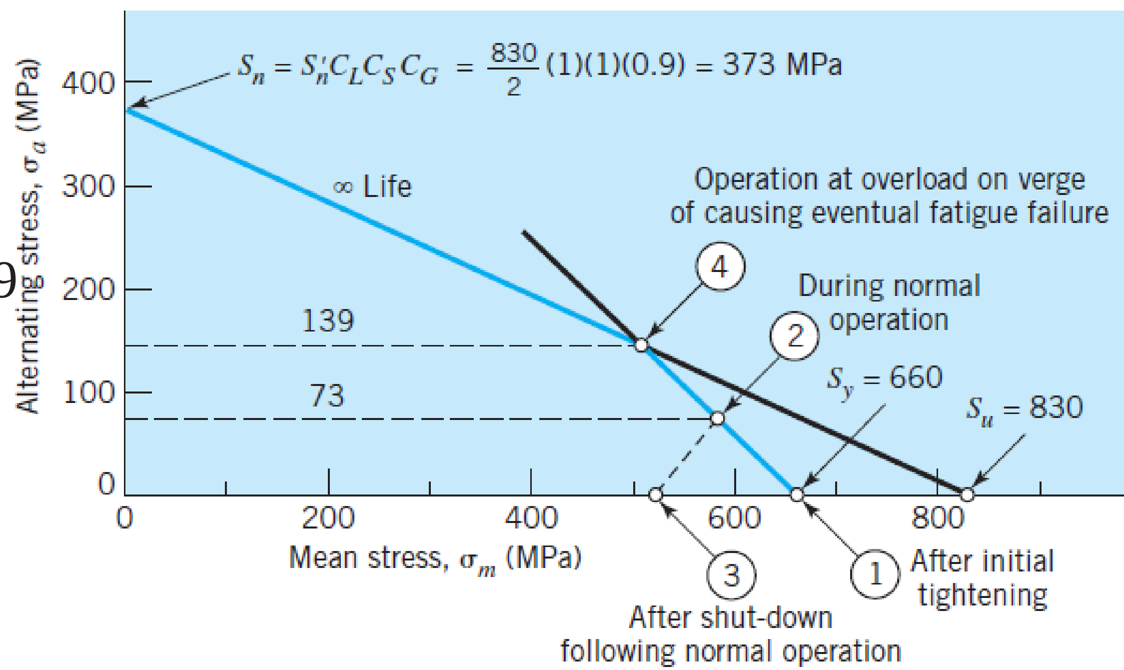
(*a*) Fluctuation in F_b and F_c caused by fluctuations in F_e



(*b*) Idealized (*not* actual) stress–strain curve for class 8.8 bolt steel

10.11.1 Analysis of Bolt Fatigue Strength at High and Low Initial Tightening Tension

- In both cases the σ_{max} corresponds to S_y and the σ_{min} results from an elastic relaxation during the removal of load, represented by the same point 2 on Fig
- The difference in case 2 is greater yielding due to higher F_i, Point 2 applies to any F_i between 10 and 38.3 kN. If F_i>12.8 kN, the thread root stress reaches point 1 upon initial tightening.
- Point 3 shows the thread root stress after F_e removal. The difference between points 1 and 3 is caused by bolt yielding during the initial application of F_e.
- When F_e added the point moves from 3 back to 2.
- The SF wrt fatigue failure is 139/73 = 1.9, as overload to failure would be point 4 or 139 MPa (σ_{alt})
- It is an F_e fluctuation between zero and 1.9 times 9 kN, or 17.1 kN.



(c) Mean stress-alternating stress diagram for plotting thread root stresses

10.11.2 Advantages of High Initial Bolt Tension

- Tightening bolts to full yield strength should not be specified, it is desirable to specify tightening to the full proof strength (i.e., $F_i = S_p A_t$)
- The advantages of bolt tightening this tightly are

1. The dynamic load on the bolt is reduced because the effective area of the clamped members is larger. (The greater the initial tightening, the more intimately in contact the clamped surfaces remain during load cycling, particularly when considering the effect of load eccentricity
2. There is maximum protection against overloads which cause joint separation
3. There is maximum protection against thread loosening

- It is important to recognize that the small amount of thread root yielding that occurs when bolts are tightened to the full proof load is not harmful to any bolt material of acceptable ductility.
- Note, for example, that all the steels listed in Tables 10.4 and 10.5 have an area reduction of about 35 %.

SAMPLE PROBLEM 10.6 Importance of Initial Tension on Bolt Fatigue Load Capacity

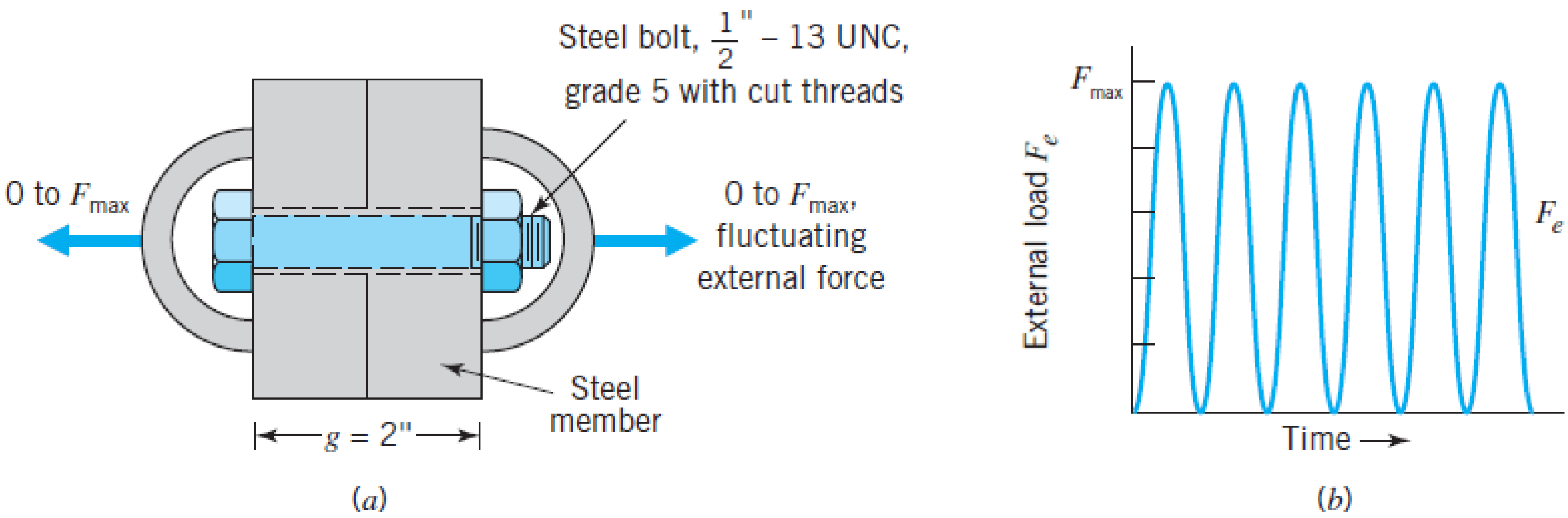
Figure 10.35*a* presents a model of two steel machine parts clamped together with a single $\frac{1}{2}$ in.–13 UNC grade 5 bolt with cut threads and subjected to a separating force that fluctuates between zero and F_{max}. What is the greatest value of F_{max} that would give infinite bolt fatigue life (a) if the bolt has *no* initial tension and (b) if the bolt is initially tightened to its full proof load?

SOLUTION

Known: Two plates of specified thickness are clamped together with a given bolt, and the assembly is subjected to a zero to F_{max} fluctuating force of separation. The assembly is to have infinite life (a) if the bolt has *no* initial tension and (b) if the bolt is preloaded to its full proof load.

Find: Determine F_{max} for cases a and b.

Schematic and Given Data:

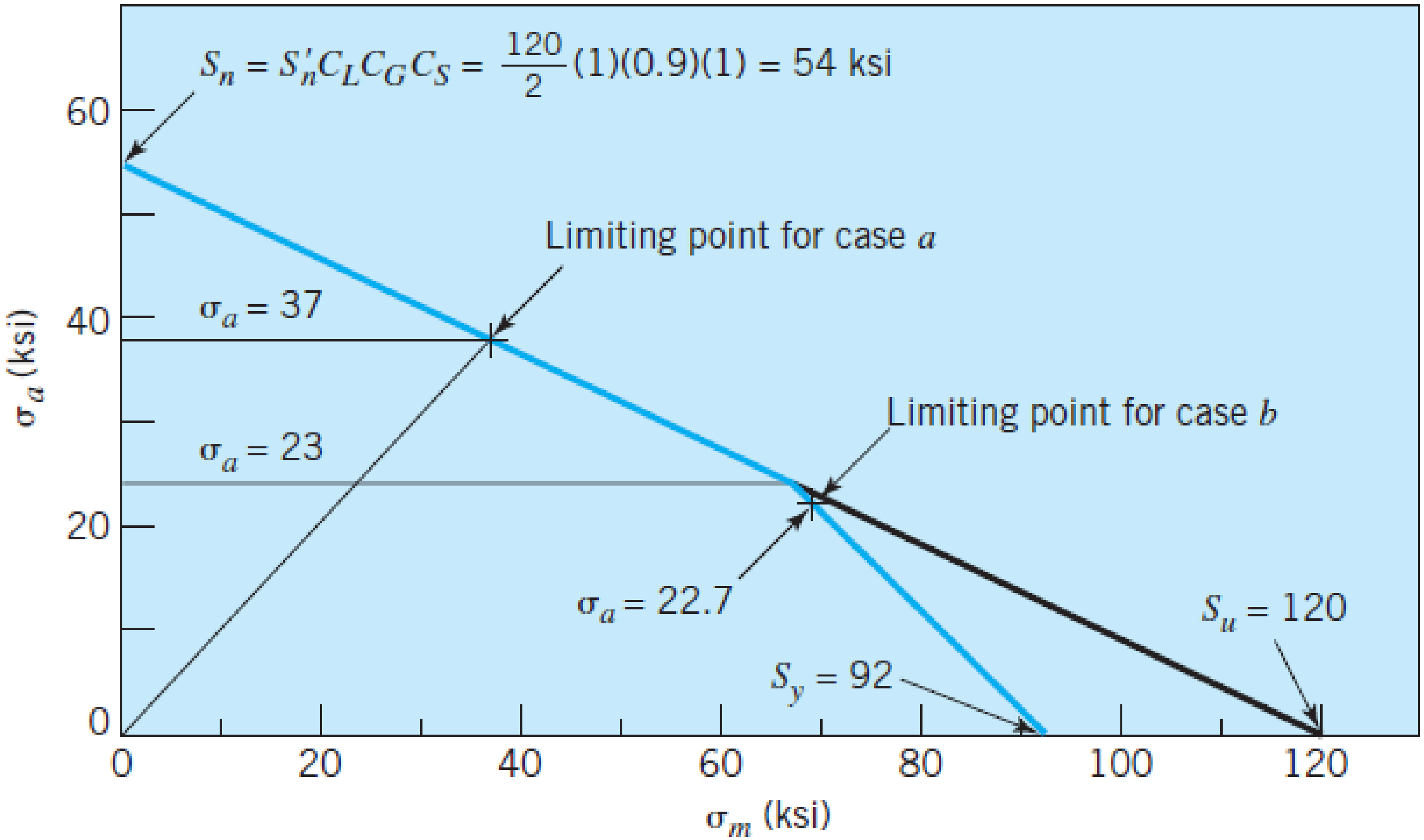


Simplified model of machine members bolted together

Fluctuating separating force versus time

Assumptions:

1. The bolt threads extend only slightly above the nut, and the bolt shank has a $\frac{1}{2}$ in. diameter over its entire length.
2. The two steel machined plates have smooth flat surfaces, and there is no gasket between them.
3. The effective area of the clamped members can be approximated as per Figure 10.28.



(*c*)

Fatigue diagram for thread root

Analysis:

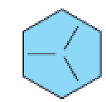
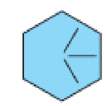
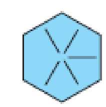
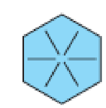
1. A σ_m–σ_a diagram is routinely constructed in Figure 10.35*c*. For *case a* the only stresses are due to the fluctuating load, with

$$\sigma_m = \sigma_a = \frac{F_{\max}}{2A_t}K_f = \frac{F_{\max}}{2(0.1419)}(3.8) = 13.39F_{\max}$$

(Units are pounds and inches.)

10.6 Fastener Materials and Methods of Manufacture

Table 10.4 Specifications for Steel Used in Inch Series Screws and Bolts

SAE Grade	Diameter d (in.)	Proof Load (Strength)[a] S_p (ksi)	Yield Strength[b] S_y (ksi)	Tensile Strength S_u (ksi)	Elongation, Minimum (%)	Reduction of Area, Minimum (%)	Core Hardness, Rockwell Min	Core Hardness, Rockwell Max	Grade Identification Marking on Bolt Head
1	$\frac{1}{4}$ thru $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
2	$\frac{1}{4}$ thru $\frac{3}{4}$	55	57	74	18	35	B80	B100	None
2	Over $\frac{3}{4}$ to $1\frac{1}{2}$	33	36	60	18	35	B70	B100	None
5	$\frac{1}{4}$ thru 1	85	92	120	14	35	C25	C34	
5	Over 1 to $1\frac{1}{2}$	74	81	105	14	35	C19	C30	
5.2	$\frac{1}{4}$ thru 1	85	92	120	14	35	C26	C36	
7	$\frac{1}{4}$ thru $1\frac{1}{2}$	105	115	133	12	35	C28	C34	
8	$\frac{1}{4}$ thru $1\frac{1}{2}$	120	130	150	12	35	C33	C39	

[a] Proof load (strength) corresponds to the axially applied load that the screw or bolt must withstand without permanent set.

[b] Yield strength corresponds to 0.2 percent offset measured on machine test specimens.

Source: Society of Automotive Engineers standard J429k (1979).

Table 10.1 Basic Dimensions of Unified Screw Threads

Size	Major Diameter d (in.)	Coarse Threads—UNC: Threads per Inch	Coarse Threads—UNC: Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC: Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF: Threads per Inch	Fine Threads—UNF: Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF: Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
3(.099)	0.0990	48	0.0734	0.00487	56	0.0771	0.00523
4(.112)	0.1120	40	0.0813	0.00604	48	0.0864	0.00661
5(.125)	0.1250	40	0.0943	0.00796	44	0.0971	0.00830
6(.138)	0.1380	32	0.0997	0.00909	40	0.1073	0.01015
8(.164)	0.1640	32	0.1257	0.0140	36	0.1299	0.01474
10(.190)	0.1900	24	0.1389	0.0175	32	0.1517	0.0200
12(.216)	0.2160	24	0.1649	0.0242	28	0.1722	0.0258
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599

TABLE 10.6 **Fatigue Stress Concentration Factors K_f for Steel Threaded Members (Approximate Values for Unified and ISO Threads)**

Hardness	SAE Grade (Unified Threads)	SAE Class (ISO Threads)	K_f[a] Rolled Threads	K_f[a] Cut Threads
Below 200 Bhn (annealed)	2 and below	5.8 and below	2.2	2.8
Above 200 Bhn (hardened)	4 and above	8.8 and above	3.0	3.8

[a] With good commercial surfaces, use $C_s = 1$ (rather than a value from Fig. 8.13) when using these values of K_f.

2. For borderline infinite fatigue life, Figure 10.35*c* shows

$$\sigma_m = \sigma_a = 37{,}000 \text{ psi}$$

Therefore, $13.39F_{\text{max}} = 37{,}000$, or (after rounding off)

$$F_{\text{max}} = 2760 \text{ lb}$$

3. For *case b*, the initial tension is

$$k_b = \frac{A_b E_b}{g} \quad \text{and} \quad k_c = \frac{A_c E_c}{g} \qquad \textbf{(10.14)}$$

$$F_i = S_p A_t = (85{,}000)(0.1419) = 12{,}060 \text{ lb}$$

4. If the steel plates as assumed have smooth, flat surfaces, and there is no gasket between them, k_b and k_c are simply proportional to A_b and A_c (see Eq. 10.14). With the assumptions that the bolt threads extend only slightly above the nut, and that the bolt shank is $\frac{1}{2}$ in. in diameter over its full length,

$$A_b = \frac{\pi}{4}d^2 = \frac{\pi}{4}\left(\tfrac{1}{2}\text{ in.}\right)^2 = 0.196 \text{ in.}^2$$

Using Figure 10.28 to estimate A_c, we have

$$A_c = \frac{\pi}{16}(5d^2 + 6dg \tan 30^\circ + g^2 \tan^2 30^\circ)$$

$$= \frac{\pi}{16}\left[5\left(\tfrac{1}{2}\right)^2 + 6\left(\tfrac{1}{2}\right)(2)(0.577) + (2)^2(0.333)\right] = 1.19 \text{ in.}^2$$

Thus,

$$\frac{k_b}{k_b + k_c} = \frac{A_b}{A_b + A_c} = \frac{0.196}{0.196 + 1.19} = 0.14$$

which means that only 14 percent of the external load fluctuation is felt by the bolt (86 percent goes to decreasing clamping pressure).

$$k_b = \frac{A_b E_b}{g} \quad \text{and} \quad k_c = \frac{A_c E_c}{g} \qquad \textbf{(10.14)}$$

$$A_c = \frac{\pi}{16}(5d^2 + 6dg \tan 30^\circ + g^2 \tan^2 30^\circ) \approx d^2 + 0.68dg + 0.065g^2$$

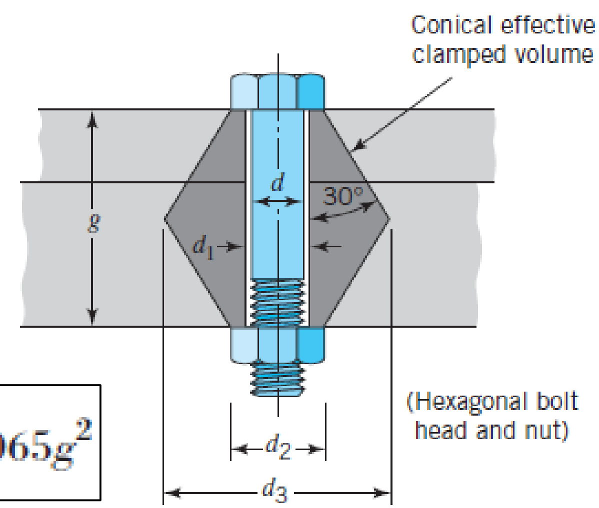


FIGURE 10.28

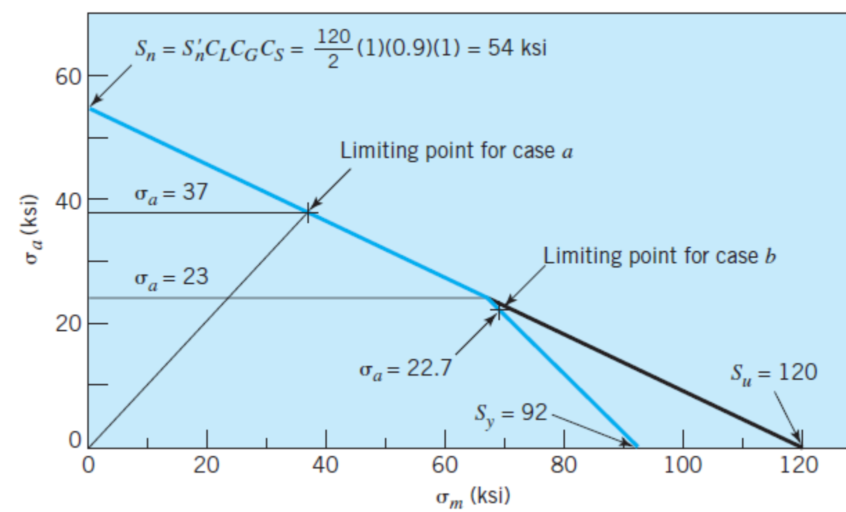
5. The alternating bolt load is half of the peak-to-peak load fluctuation, or $0.07F_{\text{max}}$. Thus the alternating bolt stress is

$$\sigma_a = \frac{F_a}{A_t}K_f = \frac{0.07F_{\text{max}}}{0.1419}(3.8) = 1.88F_{\text{max}}$$

6. With $F_i = S_p A_t = 12{,}060$ lb, external loads up to a little over 12,060 lb will not cause joint separation. Hence, $F_{\text{max}} = 12{,}060$ lb is just satisfactory *if* the alternating bolt stress does not cause fatigue failure. For $F_{\text{max}} = 12{,}060$ lb,

$$\sigma_a = 1.88F_{\text{max}} = 1.88(12{,}060) = 22{,}670 \text{ psi}$$

Figure 10.35*c* shows that this point is just below the infinite-life Goodman line (σ_a could go as high as 23 ksi). Hence, the answer for *case b* is (rounded off): $F_{\text{max}} = 12{,}000$ lb, or $4\frac{1}{2}$ *times* the value for *case a*.



(*c*)
Fatigue diagram for thread root

10.12 Bolt Selection for Fatigue Loading: Using Special Test Data

Table 10.7 Fatigue Strength of Tightened Bolts, S_a

Material	Thread Rolling	Finish	Thread ISO	Alternating Nominal Stress[a] S_a	
				ksi	MPa
Steel, S_u = 120–260 ksi	Before H.T.	Phosphate and oil	Standard	10	69
Steel, S_u = 120–260 ksi	After H.T.	Phosphate and oil	Standard	21	145
Steel, S_u = 120–260 ksi	After H.T.	Cadmium plate	Standard	19	131
Steel, S_u = 120–260 ksi	After H.T.	Phosphate and oil	Special[b]	26	179
Steel, S_u = 120–260 ksi	After H.T.	Cadmium plate	Special[b]	23	158
Titanium, S_u = 160 ksi			Standard	10	69
Titanium, S_u = 160 ksi			Special[b]	14	96

[a] Alternating nominal stress is defined as alternating bolt force/A_t, 50 percent probability of failure, bolt sizes to 1 in. or 25 mm [3,4,11].

[b] SPS Technologies, Inc. "Asymmetric" thread (incorporates large root radius). (The fillet under the bolt head must be rolled to make this region as strong in fatigue as the thread.)

SAMPLE PROBLEM 10.7 Selection of Pressure Vessel Flange Bolts

The flanged joint in Figure 10.36 involves a cylinder internal diameter of 250 mm, a bolt circle diameter of 350 mm, and an internal gage pressure that fluctuates rapidly between zero and 2.5 MPa. Twelve conventional class 8.8 steel bolts with threads rolled before heat treatment are to be used. The cylinder is made of cast iron ($E = 100$ GPa) and the cover plate of aluminum ($E = 70$ GPa). Construction details are such that the effective clamped area A_c can conservatively be assumed equal to $5A_b$. The clamped thicknesses of the cast iron and aluminum members are the same. For infinite fatigue life with a safety factor of 2, determine an appropriate bolt size. Assume that after a period of operation, the initial tension may be as low as $0.55S_pA_t$.

SOLUTION

Known: An aluminum cover plate of specified bolt circle diameter is bolted to a cast-iron cylinder of given internal diameter wherein the internal gage pressure fluctuates between known pressures. Twelve class 8.8 steel bolts with rolled threads clamp an area of five times the bolt cross-sectional area. An infinite fatigue life with a safety factor of 2 is desired.

Find: Select an appropriate bolt size.

Schematic and Given Data:

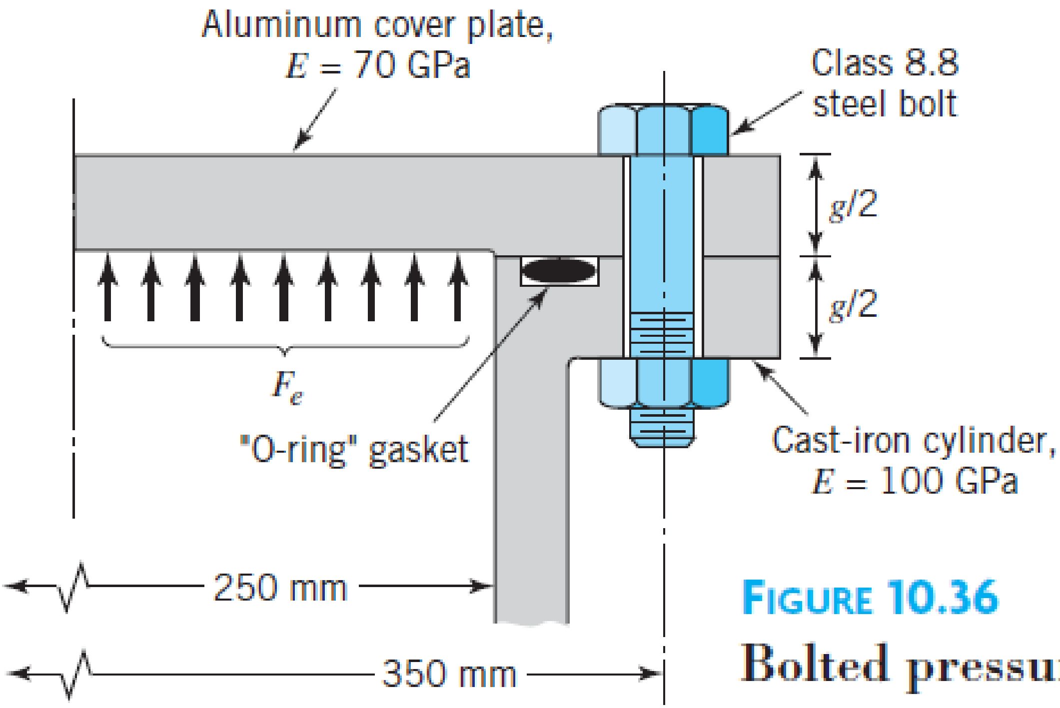


FIGURE 10.36

Bolted pressure vessel flange.

Assumptions:

1. The load is shared equally by each of the 12 bolts.
2. Table 10.7 lists the fatigue strength for the bolt material.
3. The bolt tensile stress is computed using the stress area, which is based on the average of the pitch and root diameters.
4. The initial bolt tension may be as low as $0.55S_pA_t$ after a period of operation.

Analysis:

1. The total value of F_c at the design overload (normal load times safety factor) is

$$\frac{\pi}{4}d^2 p_{\max} = \frac{\pi}{4}(250 \text{ mm})^2(5.0 \text{ MPa}) = 245.4 \text{ kN}$$

which, divided among 12 bolts, gives 20.5 kN per bolt.

2. Stiffness k_c is the resultant of two "springs" in series (let the cast iron be "spring" 1 and the aluminum "spring" 2), for which Eq. 10.15 applies:

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

Here

$$k_1 = \frac{A_1 E_1}{L_1} = \frac{5A_b(100)}{g/2}$$

and

$$k_2 = \frac{A_2 E_2}{L_2} = \frac{5A_b(70)}{g/2}$$

Substituting gives

$$k_c = \frac{k_1 k_2}{k_1 + k_2} = \frac{412A_b}{g}$$

From Eq. 10.14, we have

$$k_b = \frac{A_b E_b}{g} = \frac{A_b(200)}{g}$$

which leads to

$$k_c/k_b = 2.06$$

From Eq. i, the increased bolt force is

$$\Delta F_b = \frac{k_b}{k_b + k_c} F_e = \left(\frac{1}{1 + 2.06}\right)(20{,}500) = 6766 \text{ N}$$

The alternating force is $F_a = \Delta F_b/2 = 3383$ N.

3. Let us use the fatigue strength data in Table 10.7. For this bolt material, the table lists 69 MPa as the fatigue-limiting value of alternating *nominal* stress. The actual value of alternating nominal stress is

$$\sigma_a = \frac{F_a}{A_t} = \frac{3383}{A_t}$$

Hence, the required value of A_t is 3383/69 = 49 mm^2.

4. From Table 10.2, select the next larger standard size: M10 × 1.5 with A_t = 58.0 mm^2.

5. The minimum initial clamping force is given as $0.55S_pA_t$ = (0.55)(0.600 GPa)(58.0 mm^2) = 19.2 kN. Since 33 percent of the applied 20.5-kN load contributed to bolt tension, the remaining 67 percent (i.e., 13.7 kN) will decrease clamping force, thereby leaving a minimum clamping force of 5.5 kN.

Comment: The ratio of bolt spacing to bolt diameter in this problem came out to $350\pi/(12 \times 10)$ or 9.16. Empirical guidelines sometimes used are that this ratio should be (a) less than 10 to maintain good flange pressure between bolts and (b) greater than 5 to provide convenient clearance for standard wrenches.

10.12 Bolt Selection for Fatigue Loading: Using Special Test Data

Table 10.7 Fatigue Strength of Tightened Bolts, S_a

Material	Thread Rolling	Finish	Thread ISO	Alternating Nominal Stress[a] S_a ksi	MPa
Steel, $S_u = 120–260$ ksi	Before H.T.	Phosphate and oil	Standard	10	69
Steel, $S_u = 120–260$ ksi	After H.T.	Phosphate and oil	Standard	21	145

Table 10.2 Basic Dimensions of ISO Metric Screw Threads

	Coarse Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03
3.5	0.6	2.76	6.78
4	0.7	3.14	8.78
5	0.8	4.02	14.2
6	1	4.77	20.1
7	1	5.77	28.9
8	1.25	6.47	36.6
10	1.5	8.16	58.0

Table 10.5 Specifications for Steel Used in Millimeter Series

SAE Class	Diameter d (mm)	Proof Load (Strength)[a] S_p (MPa)	Yield Strength[b] S_y (MPa)
4.6	5 thru 36	225	240
4.8	1.6 thru 16	310	—
5.8	5 thru 24	380	—
8.8	17 thru 36	600	660

10.13 Increasing Bolted-Joint Fatigue Strength

1. Modify stiffnesses to decrease the portion of the F_e that increases F_b .
   a. Increase K_c by using higher E materials, flat and smooth mating surfaces (without gaskets), & greater area and thickness of plates in compression.
   b. Decrease K_b by securing the desired clamping force with smaller bolts of greater strength and by fully utilizing the material strength through more precise control of initial tensioning.
2. Modify the nut (female threaded member) to equalize the load carried by several contact threads, and make sure # of threads in contact is adequate.
3. Reduce the thread root stress concentration by using a larger root radius.
   a. MIL-B-7838, calls for modifying the basic profile of the external thread by using a 0.144p thread root fillet radius for tension bolts up to 180-ksi tensile strength.
   b. Standard MIL-S-8879 specifies a fillet radius of 0.180p, for bolts of 180-ksi tensile strength and higher.
   c. Exotic aerospace bolts of columbium, tantalum, beryllium, and other highly notch-sensitive materials sometimes use fillet radii of 0.224p

10.13 Increasing Bolted-Joint Fatigue Strength

4. Use a material of highest practical proof strength in order to obtain maximum initial tension.
5. Use tightening procedures that ensure values of F_i as close as possible to $A_t S_p$.
6. Be sure that the threads are rolled rather than cut and that threads are rolled *after* heat treatment. The greater the strength, the more important it is to roll *after* hardening. This has been experimentally verified for tensile strengths as high as 300 ksi.
7. After reducing stress concentration and strengthening the thread as much as possible, be sure that the fillet radius under the bolt head is sufficient to avoid failures at this point. *Cold-roll* this fillet if necessary.
8. Minimize bolt bending.
9. Guard against partial loss of initial tension in service because threads loosen or materials take a permanent set. Retighten bolts as necessary. Also take steps to ensure proper tightening when bolts are replaced after being removed for servicing, and to replace bolts before they yield heavily due to repeated retightening.

12 Springs

12.1 Introduction

- Springs are elastic members that exert forces, or torques, and absorb energy, which is usually stored and later released.
- Mostly made of metal. Plastics, and rubber are used when loads are light
- For applications requiring compact springs providing very large forces with small deflections, hydraulic springs have proved effective.
- If energy absorption with maximum efficiency (minimum spring mass) is the objective, the ideal solution is an unnotched tensile bar,
- Unfortunately, tensile bars of any reasonable length are too stiff for most spring applications; hence it is necessary to form the spring material so that it can be loaded in torsion or bending.

12.2 Torsion Bar Springs

- Simplest spring is the torsion bar spring
- Used in automotive applications
- Stress, angular deflection and spring rate

$$\tau = \frac{Tr}{J} \qquad \theta = \frac{TL}{JG} \qquad K = \frac{JG}{L}$$

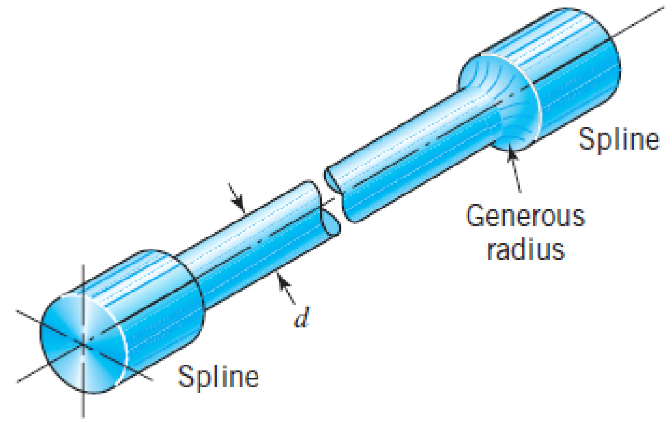
(see Table 5.1)

- For a solid bar of diameter ‘d’

$$\tau = \frac{16T}{\pi d^3} \qquad \theta = \frac{32TL}{\pi d^4 G} \qquad K = \frac{\pi d^4 G}{32L}$$

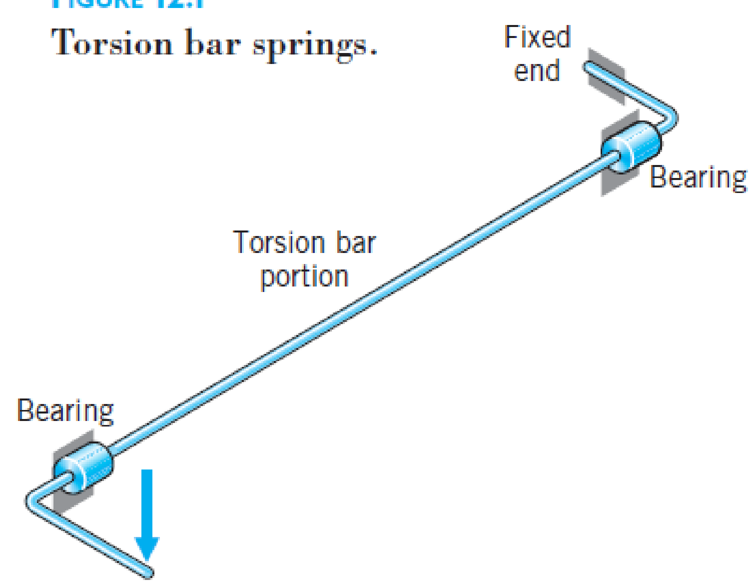
- Shear modulus G is

$$G = \frac{E}{2(1 + \nu)}$$



(a)
Torsion bar with splined ends
(type used in auto suspensions, etc.)

FIGURE 12.1
Torsion bar springs.



(b)
Rod with bent ends serving as torsion bar spring
(type used for auto hood and trunk counterbalancing, etc.)

12.3 Coil Spring Stress and Deflection Equations

- Figure shows compression and extension springs of small helix angle λ
- Force F applied along helix axis, and on the whole length the wire experiences F (transverse force) and FD/2 (torsion force)
- For spring of solid wire with dia ‘d’

$$\tau = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{8FD}{\pi d^3}$$

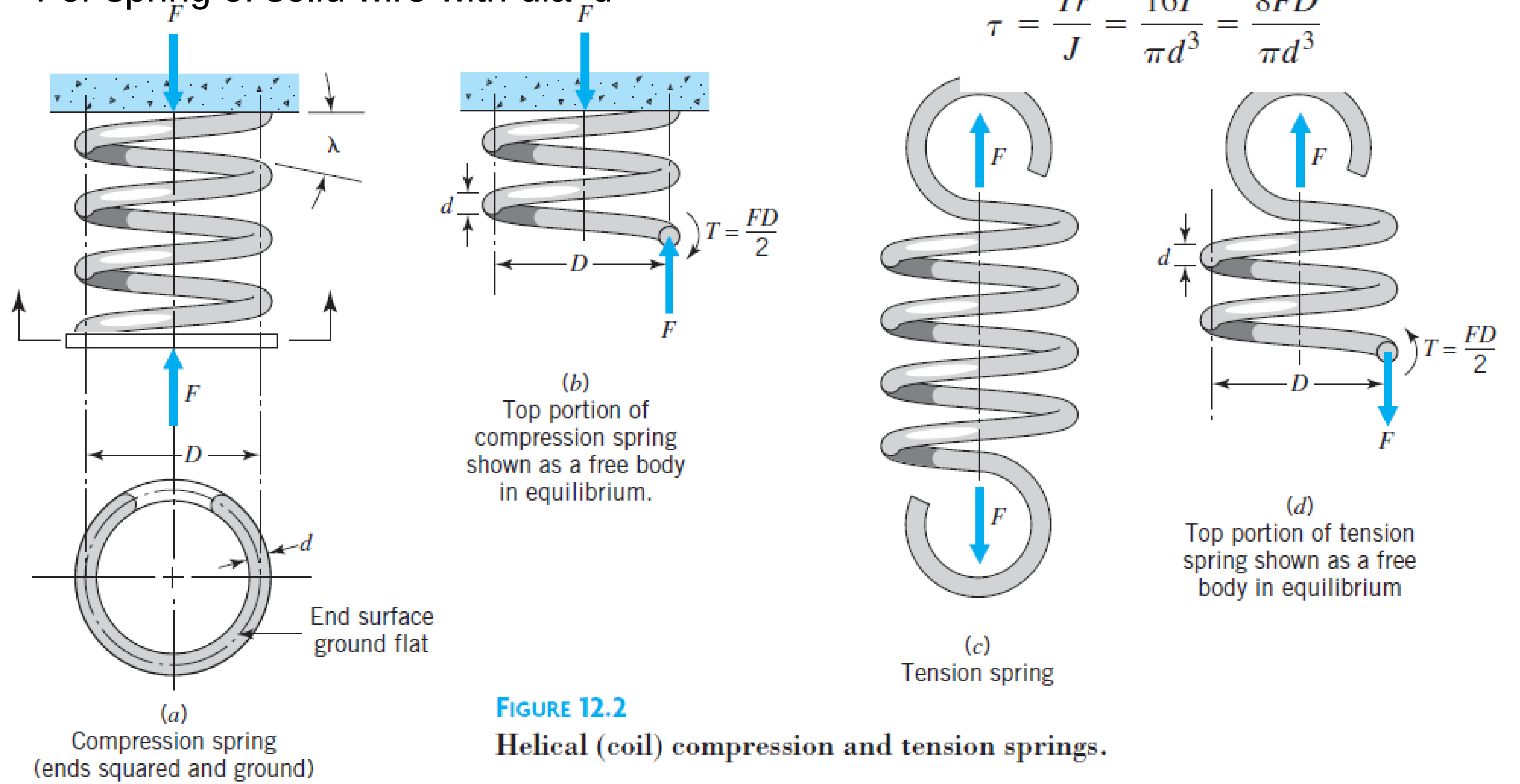


FIGURE 12.2
Helical (coil) compression and tension springs.