Chapter 8 Threads and Bolted Connections

8-1 Introduction to threads and fasteners

Wedge and thread. From elementary mechanics we know that a wedge can help save effort. If we wrap a wedge of angle α around a cylinder of radius r, we create a thread, as shown in **Figure 8-1**. For one turn of the thread, the lift is $2\pi r \tan\alpha$, which defines the term, lead, or L_d.

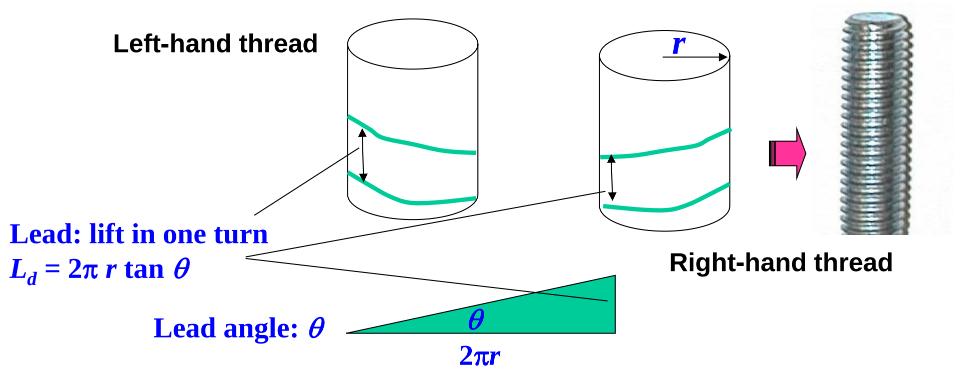


Figure 8-1 From a wedge to a thread, definition of lead, L_d.

Thread geometry. A thread may be internal, as that of a nut, and external, as that of a bolt. **Figure 8-2** shows the geometry of both internal and external threads. The nominal (or basic) diameter of a thread is the outside diameter of an external thread, or the inner diameter of an internal thread. It is called the major (or crest) diameter, which is the largest diameter of the thread. We can also define the other diameters, the root (or minor) diameter and the pitch (or mean) diameter, as shown in **Figure 8-2** and **Table 8-1**. The term, pitch, is defined as the distance between two adjacent thread teeth. The typical thread angle is $2\alpha = 60^0$. Think, if lead = pitch?

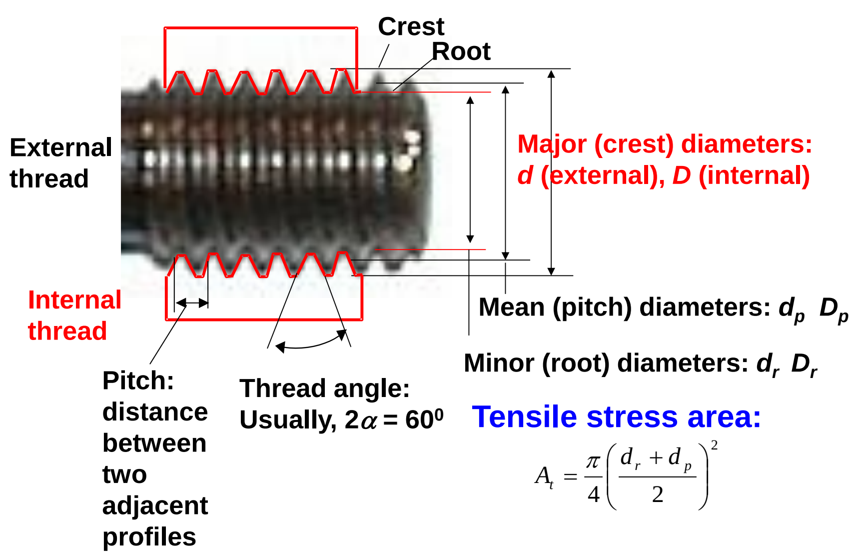


Figure 8-2 Internal and external threads, diameters, and thread angle.

Table 8-1 lists the symbols used for internal and external threads, as well as the relationship between lead and pitch. The calculation of the tensile stress area, A_t, shown in **Figure 8-2**, is also included in this table. Tensile stress area A_t is the effective area that supports a tensile load; it is the area at which the bolt stress is calculated. A_t is not determined by the pitch diameter or the internal diameter, but a diameter in between.

Table 8-1 Diameters, pitch, lead, and tensile stress area

		External thread	Internal thread
Major (Nominal) diameter (Crest diameter of a bolt)		d	D
Minor diameter		d_r $d_r = d - 1.226869p$ (Metric) $d_r = d - 1.299038p$ (US)	D_r (Root diameter of a bolt)
Mean diameter		d_p $d_p = (d_r + d_p)/2$	D_p
Pitch,	p		
Lead,	L_d	$L_d = n\,p$	
Number of thread,	n		
Tensile stress area		$A_t = \dfrac{\pi}{4}\left(\dfrac{d_r + d_p}{2}\right)^2$	

Thread length determination is largely empirical from the general engineering practice. **Table 8-2** lists several suggestions for the thread length choice based on the bolt (thread) diameter.

Table 8-2 Thread length, L_t

Metric bolts	
2d +6	L≤125
2d +12	125<L≤200
2d +25	L>200
US bolts	
2d +1/4”	L≤6”
2d +1/2”	L>6”

Thread classification. Threads may be classified based on thread geometry, thread direction, number of threads, thread series, and thread functions.

Based on thread function: A thread may be used for fastening, adjustment, or transmission of power and motion.

Based on thread geometry: There are three commonly used shapes, sharp V, square, and ACME (trapezoidal), shown in **Figure 8-3**. The thread angle for the ACME thread is $2\alpha = 29^0$.

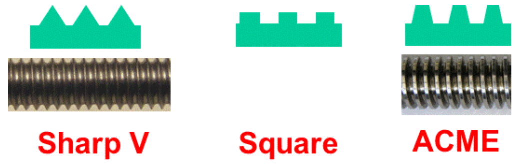


Figure 8-3 Thread geometry

Based on thread direction: There are two types of threads based on the thread direction, the right-hand thread and the left-hand thread, shown in **Figure 8-1**. A left-hand thread must be specified by LH.

Based on the number of threads. A thread may be a single-start, double-start, or multiple-start thread, as shown below. This answers the question asked before and explains why $L_d = n\,p$ in **Table 8-1**.

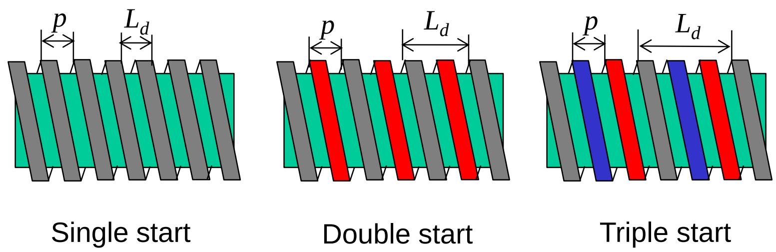


Figure 8-4 Single, double, or triple start thread, and the lead-pitch relationship

Based on thread series: UN (Unified) threads are commonly found in the US market, including the UNC (course), UNF (fine), and UNR (root modified) series. UNR means that the root radius of the thread is modified to improve the thread fatigue strength.

Metric and English threads are expressed differently, as shown in **Figure 8-5**; however, both the nominal diameter and the pitch information should be given. This figure also indicates how an internal and an external thread should be dimensioned in a drawing.

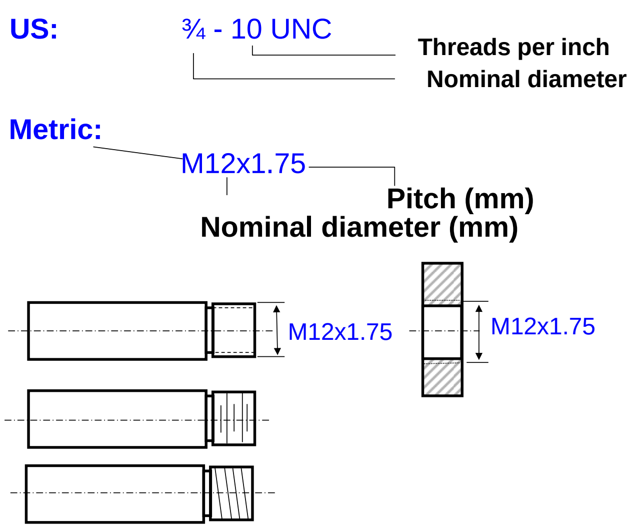


Figure 8-5 Thread expressions.

8-2 Bolt in a tensile connection (joint)

Bolts are threaded cylindrical elements. **Figure 8-6** shows several samples of bolts and nuts. A bolt may have a threaded and an unthreaded portion, depending on the length of the bolt. A washer face may be made as if a thin washer were integrated with a bolt hexagon head or a nut.

A tensile connection is a structure where several members are clamped by a bolt and a nut (or an element that functions like a nut) to support a load. The load can be tensile or other types, like shear, bending, etc. The members may be plates, flanges, walls, and washers. A bolt in a tensile connection is preloaded and then further stretched when a tensile work load is applied. The members are pre-compressed and then released more or less under this tensile load.

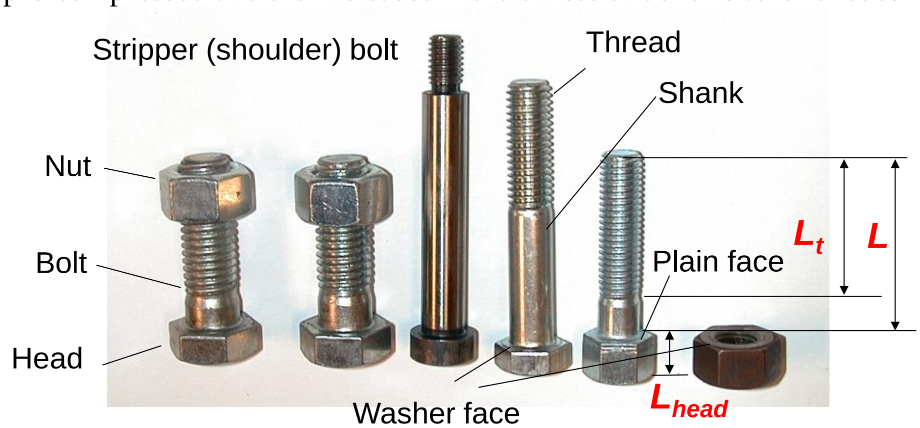


Figure 8-6 Samples of bolts and nuts; lengths of a bolt and its thread, L and L_t, and thickness of the bolt head, L_{head}, are also shown.

Our work in this section will be focused on bolted connections in tensile loading, and our major concerns are bolt strength (against plastic deformation and fatigue) and joint security (anti-loosening). The former means that a particular stress in a bolt cannot exceed the corresponding strength of the bolt in use while the latter guarantees no separation appears in the joint. We need to define several items before doing the analysis. The term, grip, is for the clamping length between the two washer faces of the bolt and the nut in a joint. Washers may still be used next to the washer face of a bolt or nut. Washers protect the surfaces of members that are usually more expensive than the bolts and washers; they increase the load application area and reduce edge stress concentrations due to tightening at the member surfaces around the nut and the bolt head. They help create flat areas for a bolt assembly if the members are cast pieces. Several lengths are defined as follows and plotted in **Figure 8-7**.

L	Bolt length: $L = l_G + l_{nut} + 2 \sim 5$ mm
L_t	Thread length: $L_t = 2d + 6$ mm (or ¼”) for $L \leq 125$ mm (6”)
l_d	Length of the unthreaded portion of the bolt in the grip
l_G	Length of the grip = summation of the member thicknesses
l_t	Length of the threaded portion of the bolt in the grip: $l_t = l_G - l_d$

In order to calculate the stress in a bolt, we need to analyze the force in the bolt. However, a bolted joint is complicated, statically indeterminate; we need to determine the bolt force from the stiffness and elastic deformation analyses.

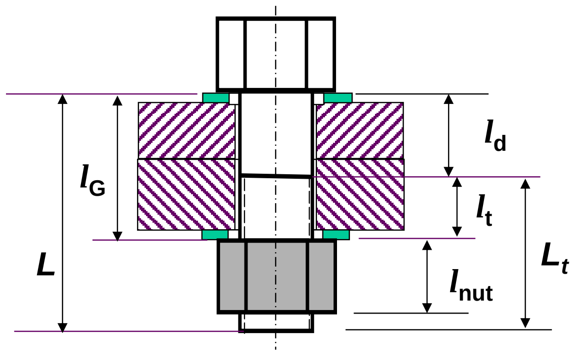
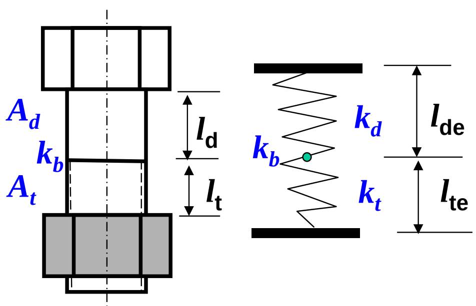


Figure 8-7 Bolted connection, the bolt, the nut, and the members, which are the two plates and two washers here. Several related dimensions are also shown.

Bolt stiffness, k_b. A bolt may have two portions: the threaded and unthreaded portions. These two different sections of the bolt should be modeled as springs of different stiffness values in series. **Figure 8-8** shows such springs and the notations used.



***b*: bolt**
***d*: unthreaded portion**
***t*: threaded portion**
***e*: equivalent**
***E*: elastic modulus**
δ: Deformation
***k*: stiffness**
***A*: area**

Figure 8-8 Bolt stiffness. A bolt in a tensile connection and the equivalent springs.

As two springs in series, the total deformation of a bolt is the sum of the deformations of both portions.

$$\delta_b = \delta_d + \delta_t \tag{8-1}$$

The deformations of the unthreaded and threaded portions of the bolt are

$$\delta_d = \frac{F l_{de}}{E A_d} \tag{8-2}$$

$$\delta_t = \frac{F l_{te}}{E A_t} \tag{8-3}$$

Equation (8-1) yields the following stiffness expression $\dfrac{1}{k_b} = \dfrac{1}{k_d} + \dfrac{1}{k_t}$ (8-4)

With

$$k_t = \frac{F}{\delta_t} = \frac{A_t E}{l_{te}} = \frac{A_t E}{l_t + 0.4 d_r} \tag{8-5}$$

$$k_d = \frac{F}{\delta_d} = \frac{A_d E}{l_{de}} = \frac{A_d E}{l_d + 0.4 d} \tag{8-6}$$

Member stiffness, k_m. Members can also be modeled as springs in series, as indicated by Equation (8-7). However, the stiffness analysis is more complicated because members are only stressed locally. The concept of frustum cones is utilized for the analysis, which assumes that only the region within the frustum cone is significantly stressed by the clamping force.

The first frustum starts at the washer face of a bolt or a nut, and lines of the cones turn at the middle of a grip. This indeed means that the washers do increase the pressurized area under the nut and the bolt head, and thus do reduce the pressure. One frustum is the cone of the same material. The frustum geometry is determined by the bolt diameter, *d*, the diameter of the smaller end of the frustum, *D*, and the thickness of the frustum, *t*. The diameter of the smaller end of the smallest frustum is $D = d_w$, at the head and the nut, no matter there is a washer face or not. How many frusta are there in the members shown in **Figure 8-9**?

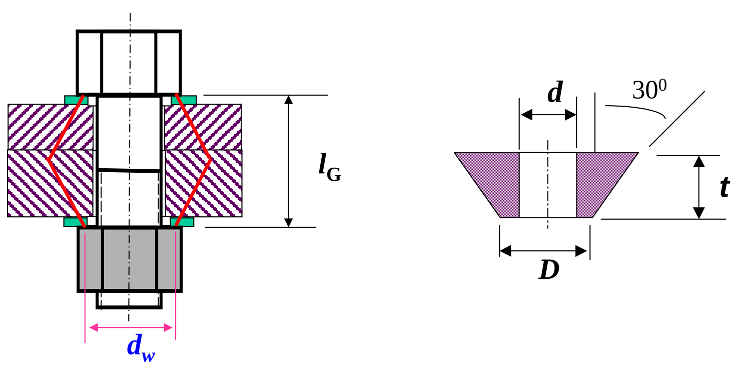


Figure 8-9 Member frusta.

$$\frac{1}{k_m} = \frac{1}{k_{m1}} + \frac{1}{k_{m2}} + \ldots + \frac{1}{k_{m\,i}} + \ldots \tag{8-7}$$

Here, k_{mi} is the stiffness of the i^{th} cone $k_{mi} = \dfrac{0.577 \pi E_i d}{\ln \dfrac{(1.15t + D - d)(D + d)}{(1.15t + D + d)(D - d)}}$ (8-8)

Note here:

(1) If one of the members is a soft gasket, then $k_m \approx k_g$, the stiffness of the gasket.
(2) The diameter of the washer face is $d_W = 1.5d$.
(3) α is in the range of 25^0~33^0. We can use 30^0. (*cf*. Shigley and Mischke, 1989)

Forces. Now we are ready to analyze forces in a bolted tension connection. The following loads are defined based on **Figure 8-10**.

Preload, or the clamping force:	P_i
Applied load supported by the bolt:	P_b
Applied load supported by the member:	P_m
Applied tensile load:	$P = P_b + P_m$
Resultant bolt load:	$F_b = P_i + P_b$

Let's analyze the resultant load in a bolt, F_b. In a bolted joint, the magnitude of bolt further elongation due to the applied load taken by the bolt, P_b, equals that of the restoration of member compression due to the applied load taken by the member, P_m (Equation (8-9)).

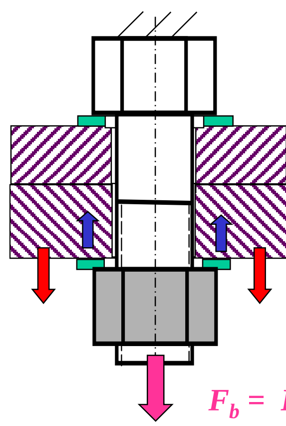


Figure 8-10 Forces in a bolted connection.

$$\frac{P_b}{k_b} = \frac{P_m}{k_m} \tag{8-9}$$

The applied load is

$$P = P_b + P_m \tag{8-10}$$

with $$P_b = \frac{k_b P_m}{k_m} \tag{8-11}$$

Or $$P_m = \frac{k_m P_b}{k_b} \tag{8-12}$$

From Equation (8-10), we get the applied load taken by the bolt

$$P_b = (P - P_m) = P - \frac{k_m}{k_b} P_b \tag{8-13}$$

which is $$P_b = \frac{k_b}{k_m + k_b} P \tag{8-14}$$

and the applied load supported by the member $$P_m = (P - P_b) = \frac{k_m}{k_m + k_b} P \tag{8-15}$$

The resultant bolt load (force) is $$F_b = (P_b + P_i) = \frac{k_b}{k_m + k_b} P + P_i \tag{8-16}$$

The resultant member force is $$F_m = (P_m - F_i) = \frac{k_m}{k_m + k_b} P - P_i \tag{8-17}$$

Equation (8-17) defines a joint constant, $C = k_b/(k_m + k_b)$ (8-18)

Therefore, Equation (8-16) may be re-written as

$$F_b = (P_b + P_i) = \frac{k_b}{k_m + k_b} P + P_i = CP + P_i \tag{8-19}$$

The stress in the bolt, $$\sigma_b = \frac{F_b}{A_t} = \frac{CP}{A_t} + \frac{P_i}{A_t} \tag{8-20}$$

We need to determine the preload in Equation (8-21), which is actually a matter of selection

$P_i = 0.75 A_t S_p$ for reused connection (8-21)
$P_i = 0.90 A_t S_p$ for permanent connection (8-22)

Bolt strength. Bolt strength is expressed by the proof strength defined based on the concept of proof load, which is the maximum load that a bolt can withstand without a permanent set (plastic deformation). Corresponding to the proof load, we can define the proof strength as follows, as if we were taking away the plastic deformation allowed in the yield strength.

$$S_p = \frac{\text{Proof load}}{\text{Tensile strength area:}A_T} \approx 0.9 S_y \tag{8-23}$$

Other strengths, such as the yield strength, which is the minimum yield strength for bolts, and the tensile strength, which is the minimum ultimate strength for bolts, can also be found in design handbooks. **Table 8-3** lists the properties of a number of metric bolts and screws. The numbers shown in the heads, the last column, are the average tensile strength of the bolts.

Table 8-3 Mechanical properties of metric bolts and screws

Property class	Size range	Min. proof strength MPa	Min. tensile strength MPa	Min. 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

(Shigley and Mischke 1989)

Factors of safety

Anti-over-proof. The bolt strength analysis is to determine a load factor, *n*, which is the factor of safety for anti-over-proof failure (or anti-plastic deformation). In a bolted connection, the stress control is achieved through controlling the applied load by load factor *n*. When the load, *P*, becomes *nP*, the bolt stress reaches it maximum, S_p, the proof strength. Here, *nP* is the maximum load that can be applied without failure, and *n* is also called the anti-over-proof factor of safety. The load factor is the margin to the largest applied load that can lead to the bolt failure due to plastic deformation; it is only applied to the applied load, not the preload, P_i, because P_i is an assembly requirement and cannot be reduced.

$$\sigma_b = \frac{F_b}{A_t} = \frac{CnP}{A_t} + \frac{p_i}{A_t} = S_p$$

Therefore, $n = \dfrac{S_p A_t - P_i}{CP}$ (8-24)

Anti separation. The resultant member load should remain compressive:

$$-F_m = P_i - P_m$$

or $F_m = P_m - P_i = (1-C)P - P_i$ (8-25)

The external load to cause separation, P_o, can be determined by

$$(1\text{-}C)P_o - P_i = 0 \qquad (8\text{-}26)$$

or $$P_o = P_i/(1\text{-}C) \qquad (8\text{-}27)$$

We can now define the factor of safety, n_{sp}, guarding against separation, as follows.

$$n_{sp} = \frac{P_0}{P} = \frac{P_i}{P(1-C)} \qquad (8\text{-}28)$$

Figure 8-11 summarizes the load-deformation relationships for a bolt and a set of members in preloading and work loading. Keep in mind that the application of tensile load P further increases the bolt deformation but releases the member compression.

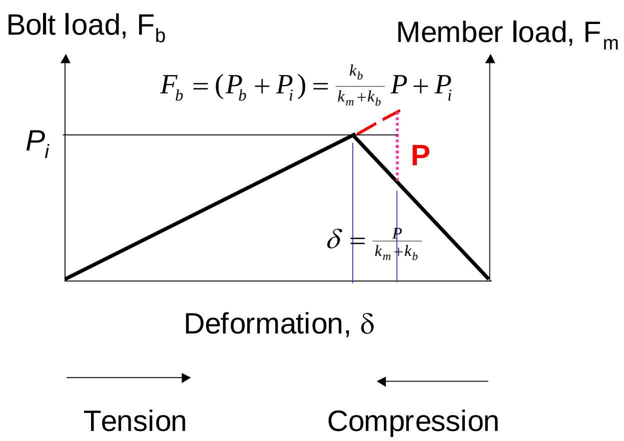


Figure 8-11 Loading lines for a bolt and a set of members in a bolted tensile connection.

Example 8-1 Eight M10x1.5 bolts are used for the enclosure of a pressure vessel, and each supports an axial load of 15 KN. If the grip length is 24 mm and the bolt length is 35 mm, calculate:

a. The stiffness of the bolt
b. The stiffness of the members
c. Joint constant
d. Preload (reused)
e. Factors of safety

Problem understanding:

I. Eight bolts are used to support a tensile load. The bolt load: P = 15 KN, and the total load supported by these eight bolts is W = 8 x 15 KN.
II. Pressure in the pressure vessel should be, p = W/surface area
III. Each bolt-nut set forms a tension connection.
IV. The bolt is under a tensile loading condition.

Solution

Bolt geometry:

L_t = 2d +6 =2(10)+6 = 26 mm

$l_d = L - L_t$ = 35-26 = 9 mm $\qquad$ $l_t = l_G - l_d$ = 24 – 9 = 15 mm

$A_t = (\pi/4)\{(d_r+d_p)/2\}^2$=58 mm^2 $\qquad$ $A_d = (1/4)\,\pi\, d^2 = 0.25\ p\ 10^2$ = 78.5 mm^2

a. Stiffness of the bolt

$$\frac{1}{k_b} = \frac{1}{k_d} + \frac{1}{k_t}\ , \qquad k_d = \frac{A_d E}{l_d + 0.4d}\ , \quad k_t = \frac{A_t E}{l_t + 0.4d_r}$$

$$k_b = \frac{A_d A_t E}{A_d(l_t + 0.4d_r) + A_t(l_d + 0.4d)} = \frac{(58)(78.5)(10^{-12})(200)(10^9)}{(78.5)(15 + 0.4(8.16))10^{-9} + 58(9 + 0.4(10))10^{-9}} = 4.16(10^8)(N/m)$$

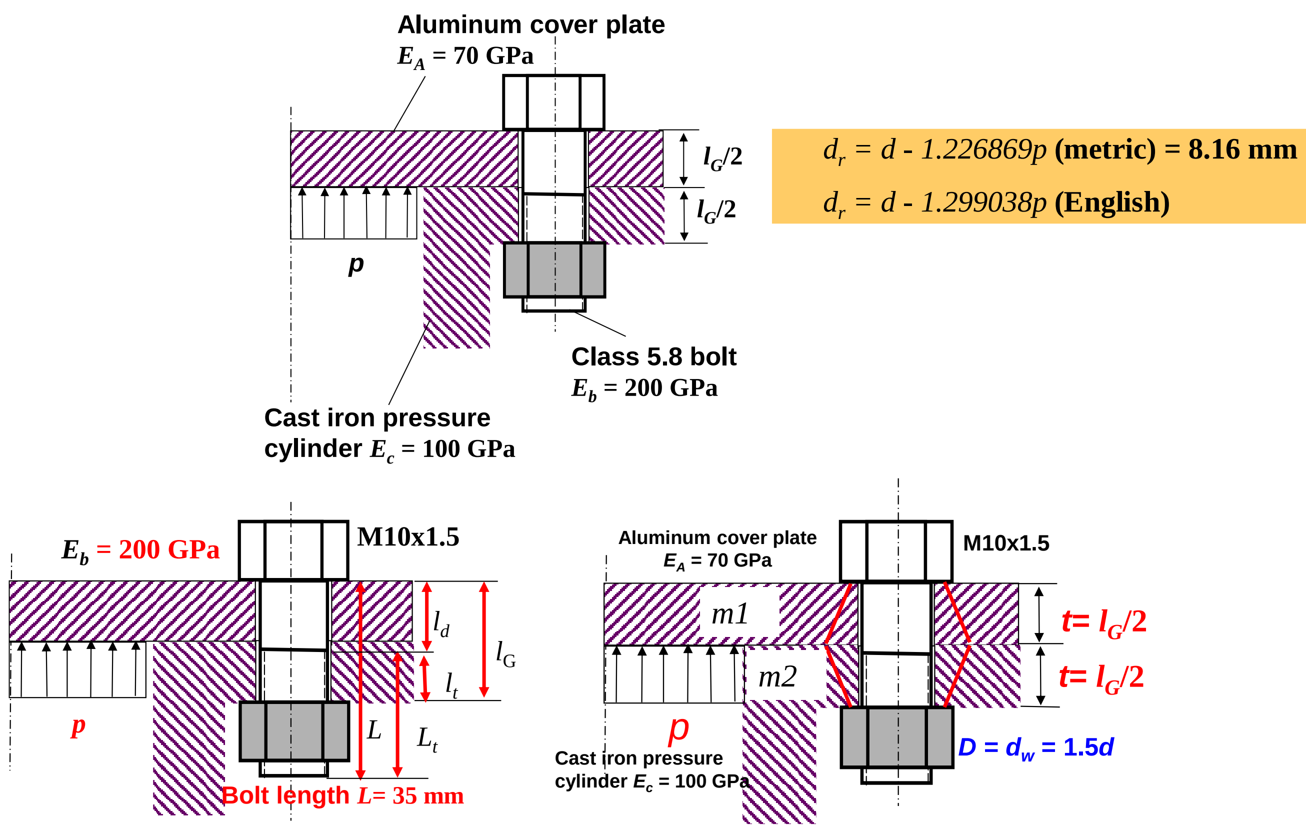


Figure 8-12. Illustration of **Example 8-1** (top), a bolt in a bolt group for the pressure vessel assembly. Here, the calculation of the root diameter, d_r, of the thread of the bolt is also given. Bottom left shows the bolt analysis, and bottom right the member analysis, where members 1, 2 are marked as m1 and m2.

b. Stiffness of the members

$$\frac{1}{k_m} = \frac{1}{k_{m1}} + \frac{1}{k_{m2}}$$

$$k_{m1} = \frac{0.577\pi E_A d}{\ln\left(\frac{(1.15t+D-d)(D+d)}{(1.15t+D+d)(D-d)}\right)} = \frac{0.577\pi(70)(10^9)(10)(10^{-3})}{\ln\left(\frac{(1.15(24/2)+1.5(10)-10)(1.5(10)+10)}{(1.15(24/2)+1.5(10)+10)(1.5(10)-10)}\right)} = 1442(10^6)N/m$$

$$k_{m2} = \frac{E_C}{E_A}k_{m1} = \frac{100}{70}(1442) \ = 2060(10^6) \qquad N/m$$

$$\frac{1}{k_m} = \frac{1}{k_{m1}} + \frac{1}{k_{m2}} = \frac{1}{1442(10^6)} + \frac{1}{2060(10^6)}$$, and therefore k_m = 848 (10^6) *N/m*

c. Joint constant

k_b = 413 (10^6) N/m, and k_m = 848 (10^6) N/m

Therefore, $C = \frac{k_b}{k_m + k_b} = \frac{416}{848+416} = 0.329$

d. Bolt preload

The proof strength of the bolt is Sp = 380 MPa (class 5.8).

Preload for reused connections is P_i = 0.75 *Sp* A_T = 0.75(380 x10^6) (58x10^{-6}) = 16.5 KN.

e. Factors of safety

Anti separation $\quad n_{sp} = \frac{P_i}{P(1-C)} = \frac{16500}{15000(1-0.328)} = 1.64$

Anti over proof $\quad n = \frac{S_p A_T - P_i}{CP} = \frac{380(10^6)(58)(10^{-6}) - 16500}{0.329(15000)} = 1.12$. Safe

Why is the anti over proof factor of safety so small?

8-3 Bolt under variable tensile loading

Most structures are under variable loading. Even for a so called static-loading case, the loading and unloading processes cause the force in a bolt to fluctuate from preload to its working force. Failure due to variable loading is fatigue. Therefore, we should extend our stress analysis to the stress and strength of the bolt under variable loading. The Goodman line will be used again here. Recall the analysis of variable loading, and details are reviewed in **Figure 8-13**. Note that the changing portion is the applied load, not the preload. The amplitude portion, P_a, and the midrange portion, $P_{m'}$, of the applied load are

$$P_a = \frac{P_{\max} - P_{\min}}{2} \tag{8-29}$$

$$P_{m'} = \frac{P_{\max} + P_{\min}}{2} \tag{8-30}$$

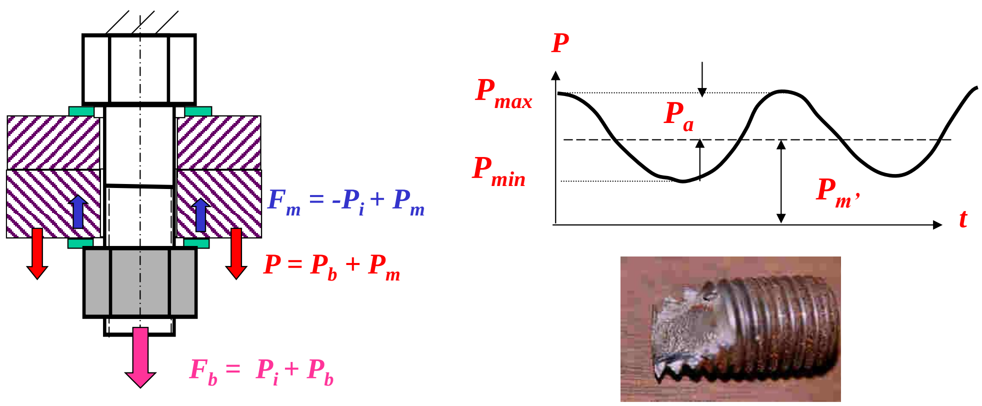


Figure 8-13. Variable loading, the applied tensile load is varying. Bottom right, an example of bolt fatigue fracture.

We know that in a bolted joint under tension, the bolt force is

$$F_b = \frac{k_b}{k_m + k_b} P + P_i = CP + P_i \tag{8-31}$$

Therefore, the variation of the force in a bolt is characterized by

$$F_{b\max} = CP_{\max} + P_i \tag{8-32}$$

$$F_{b\min} = CP_{\min} + P_i \tag{8-33}$$

The amplitude, F_{ba}, and the midrange, F_{bm}, of the force in the bolt are

$$F_{ba} = \frac{F_{b\max} - F_{b\min}}{2} = \frac{\left(CP_{\max} + P_i\right) - \left(CP_{\min} + P_i\right)}{2} = C\,\frac{P_{\max} - P_{\min}}{2} = CP_a \tag{8-34}$$

$$F_{bm} = \frac{F_{b\max} + F_{b\min}}{2} = \frac{\left(CP_{\max} + P_i\right) + \left(CP_{\min} + P_i\right)}{2} = P_i + C\,\frac{P_{\max} + P_{\min}}{2} = P_i + CP_{m'} \tag{8-35}$$

We consider that a bolt will reach fatigue failure if the applied load reaches $n_{ft}P$, where n_{ft} is the factor of safety against fatigue failure, and the corresponding amplitude and the midrange of the tensile stress, σ_{ba} and σ_{bm}, in a bolt are

$$\sigma_{ba} = \frac{CP_a n_{ft}}{A_t} \tag{8-36}$$

$$\sigma_{bm} = \frac{P_i + CP_{m'} n_{ft}}{A_t} \tag{8-37}$$

Goodman's line should be applicable for the factor of safety against fatigue failure. Here, fatigue-stress concentration factor, K_f, should be taken into account.

$$\frac{K_f \sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = 1 \tag{8-38}$$

Why not this one?

$$\frac{K_f \sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = 1/n \tag{8-39}$$

Again, the safety factor does not apply to the preload. Therefore, the factor of safety against fatigue failure is obtained by extending the load factor concept. With the assistance of Equations (8-36) and (8-37), the Goodman line expressed by Equation (8-38) becomes

$$\frac{K_f CP_a n_{ft}}{S_e A_t} + \frac{P_i + CP_{m'} n_{ft}}{S_{ut} A_t} = 1 \tag{8-40}$$

The factor of safety against fatigue failure is

$$n_{ft} = {\left(1 - \frac{P_i}{S_{ut} A_t}\right)} \Big/ {\left(\frac{K_f CP_a}{S_e A_t} + \frac{CP_{m'}}{S_{ut} A_t}\right)} = \frac{S_e\left(S_{ut} A_t - P_i\right)}{C\left(K_f S_{ut} P_a + S_e P_{m'}\right)} \tag{8-41}$$

Remember how the endurance limit of a part should be obtained from the endurance limit of a material? The same concept applies here, but the ultimate strength is that for the bolt material. For bolts and screws, factors k_s, k_m and k_t may be ignored because the properties in Table 8-3 are given for bolts of particular sizes. Only the reliability factor (Table 3-2) should be included here in the following equation.

$$S_e = k_s k_r k_t k_m S_e^{'} \approx k_r S_e^{'} \tag{8-42}$$

For a bolt in a tensile connection, we use $S_e^{'} = 0.45 S_{ut}$, which is exactly for axial-loading cases. The fatigue stress concentration factor, K_f, in Equation (8-41) is given in **Table 8-4**.

Table 8-4 Fatigue stress concentration factor, K_f

Metric grade	Rolled threads	Cut threads
3.6-5.8	2.2	2.8
6.6-10.9	3.0	3.8

Example 8-2 Eight M10x1.5 bolts are used and the axial load on each bolt changes from 0 to 15 KN. If the grip length is 24 mm and the bolt length is 35 mm, please calculate the factor of safety against fatigue failure for a 90% reliability.

This is a continuation of **Example 8-1**.

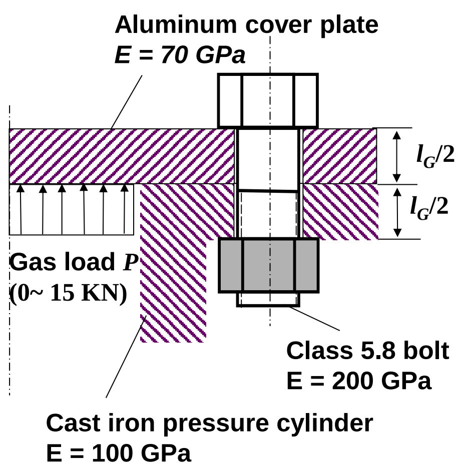


Figure 8-14 Bolted joint analyzed in **Example 8-1**.

Solution. The applied load changes from 0 to a maximum. Therefore, the stress cycle is a released-tension cycle.

From **Example 8-1**, we have

$$C = \frac{k_b}{k_m + k_b} = \frac{413}{848 + 413} = 0.329, \; A_t = 58 \text{ mm}^2, \text{ and } P_i = 0.75 \, S_p A_T = 16.5 \text{ KN}$$

Let's get the tensile strength, fatigue stress-concentration factor, and the endurance limit for the bolts. From **Table 8-3**, we have S_{ut} = 520MPa for the class 5.8 bolts.

The fatigue stress concentration factor, K_f, for a metric rolled thread is K_f = 2.2 from **Table 8-4**.

Therefore, for axial loading, $S_e^{'} = 0.45 S_{ut} = 0.45(520) = 234 MPa$

$k_r = 0.9$ for 90% reliability

$$S_e = k_f k_s k_r k_t k_m S_e^{'} \approx k_r S_e^{'} = 0.9(234) = 211 MPa$$

The amplitude and mean of the applied load

$$P_a = \frac{P_{\max} - P_{\min}}{2} = \frac{15 - 0}{2} = 7.5 KN \qquad P_{m'} = \frac{P_{\max} + P_{\min}}{2} = \frac{15 + 0}{2} = 7.5 KN$$

$$n_{ft} = \frac{S_e \left(S_{ut} A_t - P_i\right)}{C\left(K_f S_{ut} P_a + S_e P_{m'}\right)} = \frac{211(10^6)\left(520(10^6)(58)(10^{-6})_t - 16500\right)}{0.329\left[2.2(520)(10^6)7500 + 211(10^6)7500\right]} = 0.86$$

Here is a summary of the results for the design calculations for this bolted joint.

For static loading of 15KN, and the peak load of 15KN, the factors of safety are

Anti-separation $\quad n_{sp} = \dfrac{P_i}{P(1-C)} = 1.64$

Anti-over-proof $\quad n = \dfrac{S_p A_T - P_i}{CP} = 1.12$

Anti-fatigue failure $\quad n_{ft} = \dfrac{S_e \left(S_{ut} A_t - P_i\right)}{C\left(K_f S_{ut} P_a + S_e P_{m'}\right)} == 0.86$

Insufficient! There are several ways to improve the design, such as using a stronger material, but this is not a simple issue; using larger bolts, but this also increase the required preload; or using more bolts, which means the reduction of the peak load. We need to reduce the peak load value to 11,720N, or $P_a = P_{m'}$ =5860N, in order to increase the anti-fatigue failure factor of safety to about 1.1.

8-4 Bolt groups

Although bolts are under tensile loading, the bolt groups can take any loading modes: tension, bending, shear, and torsion. We know tensile loading already, **Figure 8-15** shows bolt shear under different external loads, F_1 and F_3. F_2 is balanced by bolt tension in B, C and compression in A, D. We will discuss bolt shear in this section.

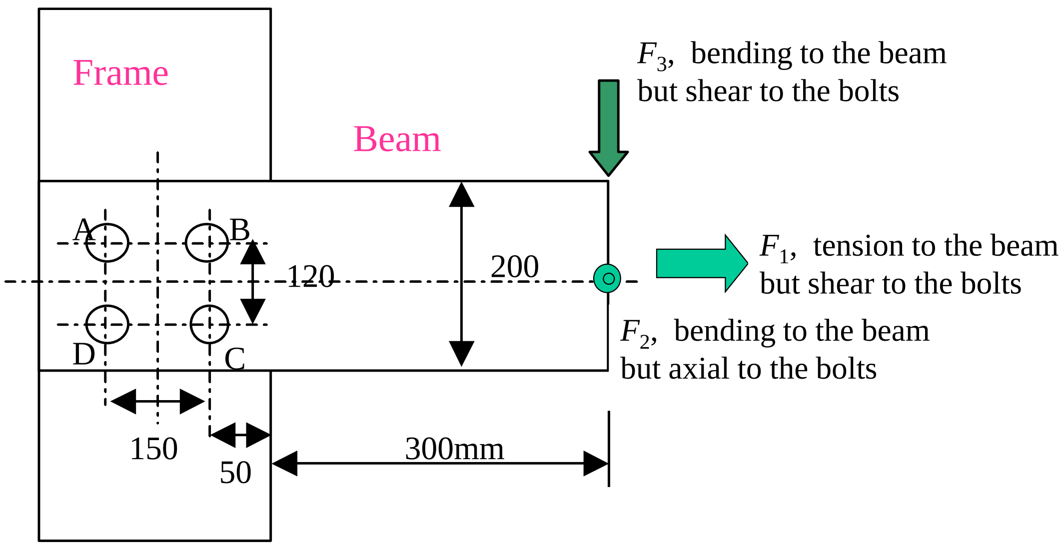


Figure 8-15 A bolt group under combined loading modes.

The structure shown in **Figure 8-16** illustrates the loading of a pair of shear forces. Here, the shear load applied to the plates is taken by the friction at the interface formed by the plates. The maximum shear load that the interfacial friction can sustain without slip is $F = fP_i$, or the static friction coefficient, f, by the preload, P_i. The static friction coefficient is $f = 0.5$.

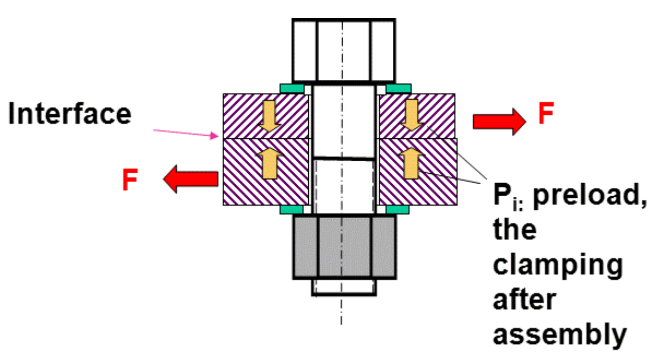


Figure 8-16 Tension connection for shear loading taken by interfacial friction.

What would happen if the shear load is $F > fP_i$?

An alternative design is to use the shank of a stripper bolt for the shear force. **Figure 8-17** is the drawing for such a structure. Here, the shear load is taken by both the shank of the bolt and the interfacial friction.

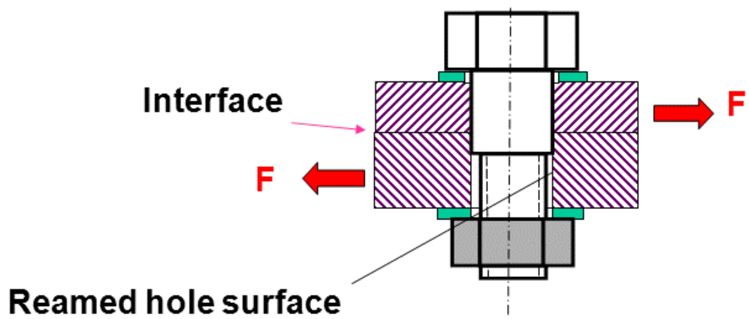


Figure 8-17. Tensile connection for shear loading taken by the shank of the stripper bolt fit to the reamed holes in the members, together with the interfacial friction.

Example 8-3 A cantilever structure is formed by a group of bolted tension connections, **Figure 8-18**, where the bolts there are under tension due to assembly and shear due to loading. Four bolts are used with a certain amount of preload. Which bolt is under the maximum shear load? How much? Are class 5.8 M10x1.5 bolts proper selections?

Solution

Both the force, F= 16KN, and the moment, which is the toque to the bolt group, T=16KNx(300+50+150/2)mm, are shear loads to the bolts. We need to determine which bolt is under the maximum shear load and how much this load is with reference to **Figure 8-19**.

Forces on each bolt due to F is F_F = 4KN, the same force is applied to each of the bolts, in the same direction;

Forces on each bolt due to T is F_T = 17.7 KN, the same magnitude, but different directions from bolt to bolt, obtained from

$$16\text{KN}(300+50+150/2)\text{mm} = 4\,(F_T)\,(OB)$$

$$F_T = 16(10^3)\,(0.3+0.05+0.15/2)/\{4[((0.12/2)^2+0.15/2)^2)^{1/2}]\} = 17.7\text{ KN}$$

Combining F_F and F_T by means of the law of cosine, we get F_B=F_C = 20.7 KN, which is the shear force on bolt C, the largest. F_B on bolt B should have the same magnitude.

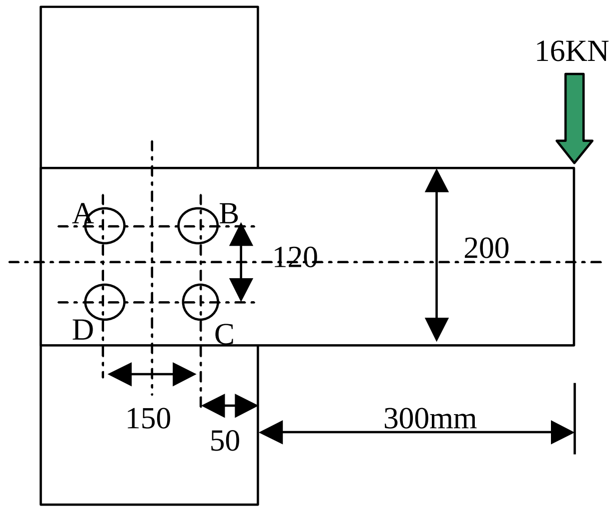


Figure 8-18 A cantilever beam under bending, but its bolts are under shearing.

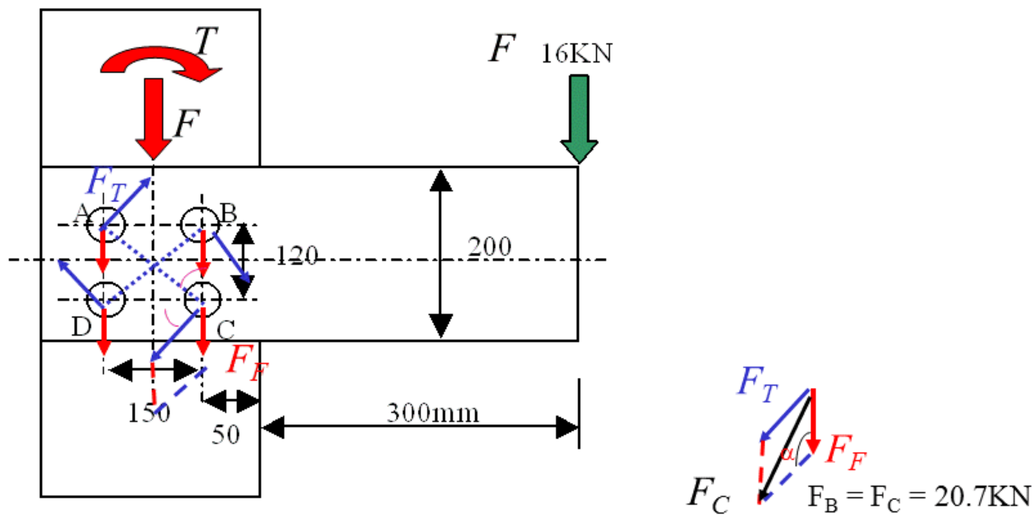


Figure 8-19 Equivalent shear forces at the bolts and the resultant shear force on Bolt C.

If friction is used, and $f = 0.5$, the maximum friction is fP_i.

For a reused connection $P_i = 0.75A_t\,S_p$

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

Note: for metric bolts: $\mathbf{d_r = \mathit{d} - 1.226869\mathit{p}}$ Here, p is the pitch.
for English bolts: $\mathbf{d_r = \mathit{d} - 1.299038\mathit{p}}$

For a class 5.8 M10x1.5 bolt,

d_r = 10-1.226869(1.5)=8.16mm,

$$d_p = \frac{d_r + d}{2} = \frac{8.16 + 10}{2} = 9.08mm$$

$$A_t = \frac{\pi}{4}\left(\frac{d_r + d_p}{2}\right)^2 = \frac{\pi}{4}\left(\frac{8.16 + 9.08}{2}\right)^2 \approx 58mm^2$$

S_p = 380 MPa

P_i = 0.75(58)(380)=16.53KN

Thus, the maximum friction is only 0.5(16.53) =8.265KN. Not enough!

Therefore, we need the design given in **Figure 8-17** with a stripper bolt for this loading, where the shear force is taken by the shank of the bolt and the interfacial friction.

8-5 Design of bolted connections

Recall the bolts we saw before, which is reprinted below in **Figure 8-20**. We seem to have a variety of bolts (and/or screws). We will see how they are used in structure designs.

Tensile connections are widely used in part assembly. Such a connection may take tensile loading, shear loading, and bending loading, etc, as we discussed in the previous section. We will talk about some of these here. More will be given in the class.

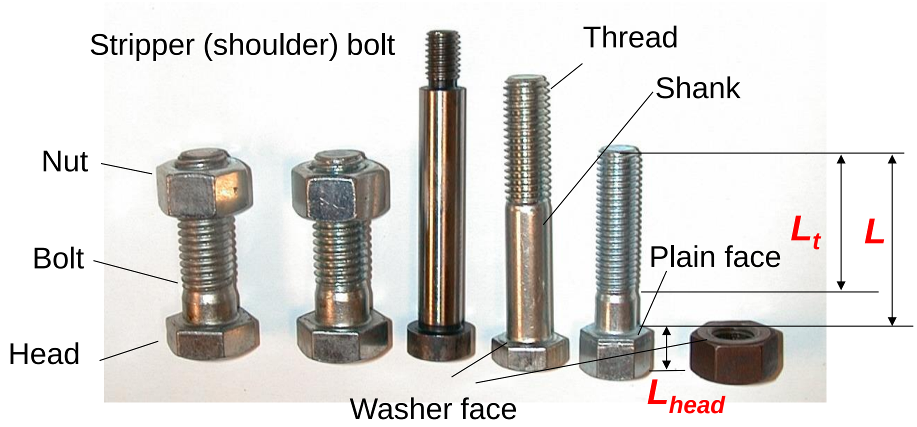


Figure 8-20 Bolt samples again.

Connecting two plates. A bolt and a nut are used to connect two plates in the structure shown by the left drawing of **Figure 8-21**, where the plates are not very thick. The plates and washes are all members. No thread is made in either member. Remember that a clearance is needed between the bolt and the bores in members, which is exaggerated in **Figure 8-21**. Such a structure requires a wrench space on each side of the connection.

A thread can be made in one of the members if it is sufficiently thick, as shown in the right drawing of **Figure 8-21**. A cap screw (bolt only, no nut) can be used but one of the members acts as the “nut” needed. This structure requires a wrench space only on one side of the connection. **Figure 8-22** shows the drawings for these bolted connections.

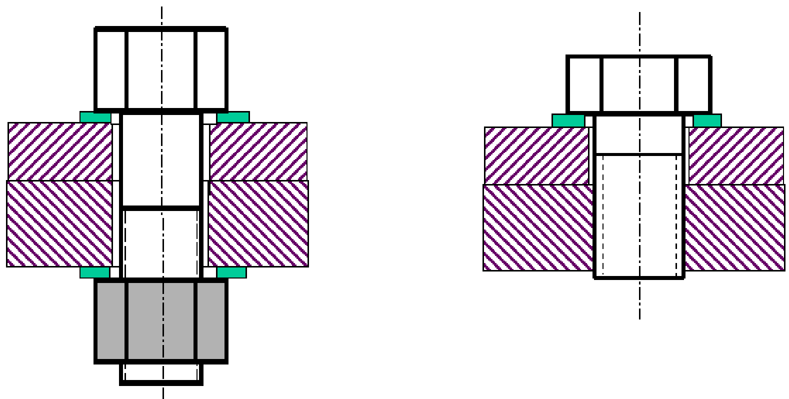


Figure 8-21 Tension connections by a bolt and nuts (left) and by a cap screw (right).

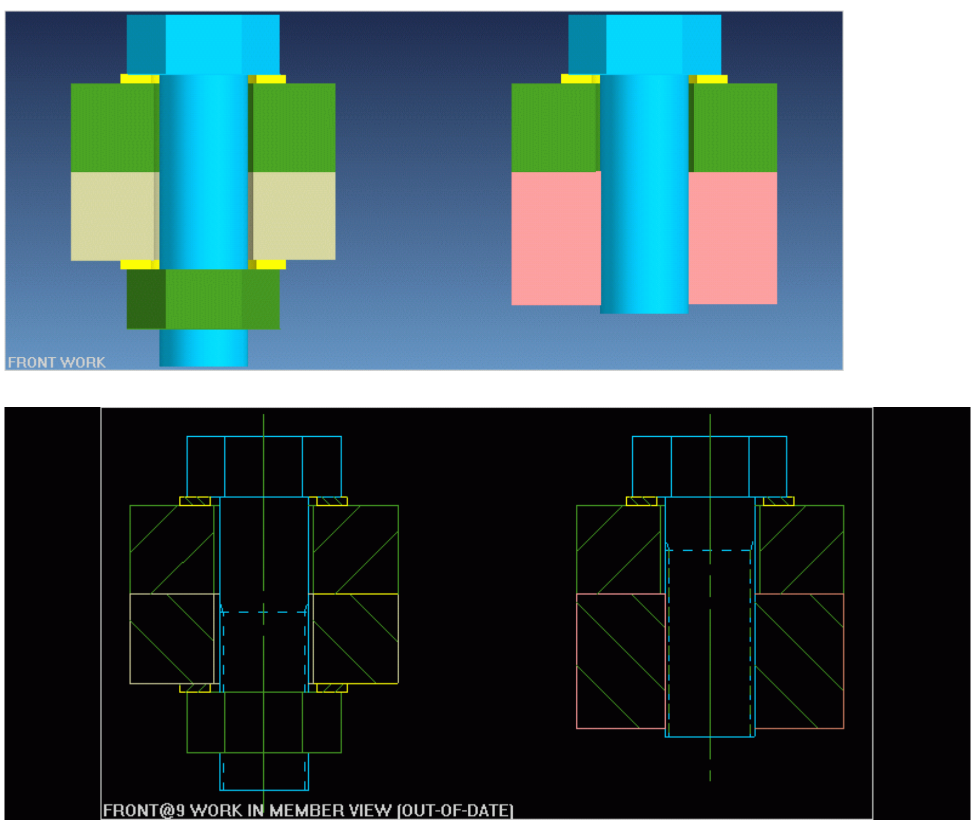
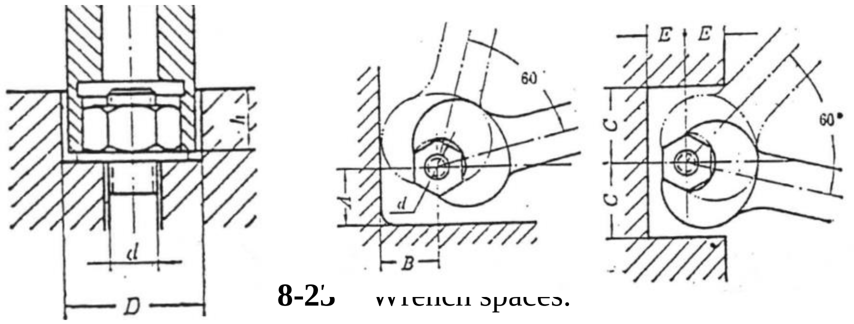


Figure 8-22 Drawing of bolted connections.

Assembly space. Do not forget the wrench space for assembly and disassembly. **Figure 8-23** shows several possible designs for the use of a pipe wrench (left) and a regular wrench (middle and right). Obviously, the pipe wrench requires less space.



8-23 Wrench spaces.

Blind threaded hole in a thick member. If one of the members has a large thickness, we need to make a blind threaded hole in the member. **Figure 8-24** shows the drilling, tapping, and assembly of the thick member in a bolted connection. The assembled structure is shown in **Figure 8-25**. Pay a close attention to the ends of the hole, the thread, and the screw; they should be correctly expressed in the drawing.

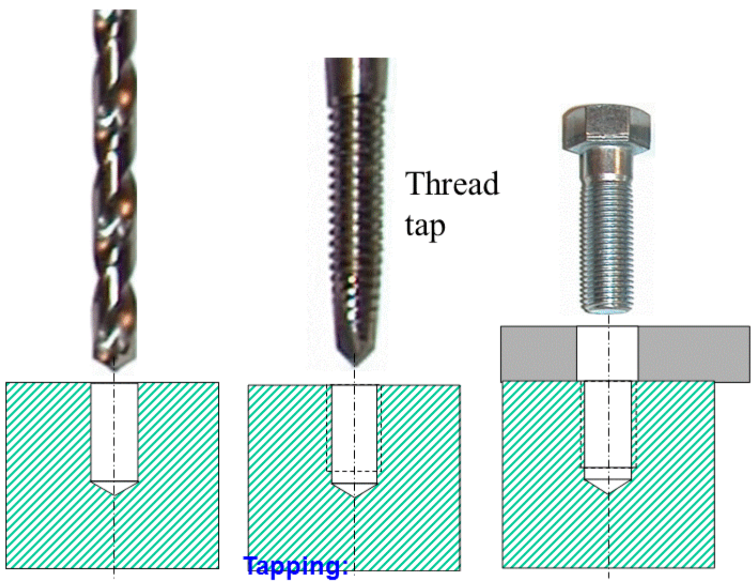


Figure 8-24 Drilling, tapping, and assembly of a thick member.

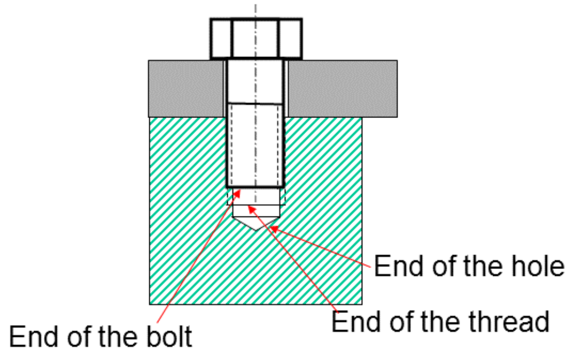


Figure 8-25. Parts shown in **Figure 8-24** after assembly.

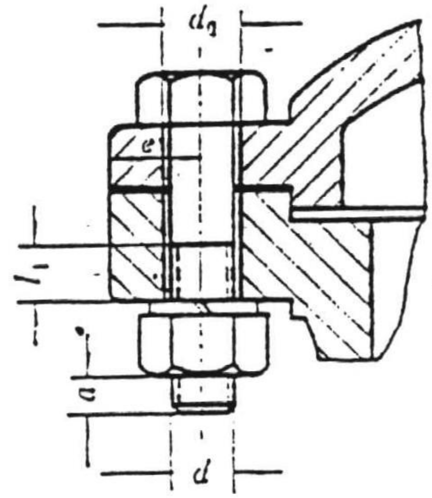
There is no thread in the top member! Why? Think about the manufacturing, assembly, and performance of the parts in the connection, and also think about the thread of the screw (bolt).

Comparing tension connections. Figure 8-26 shows three typical structures of a bolted tension connection. Let's understand and compare them. In this figure, several bolts (different bolts are used in different designs) are utilized to connect the flanges of a pressure vessel. The flanges shown in **Figure 8-26** (a) are through bolted, where the bolt holes in the members are free of thread. The structure shown in **Figure 8-26** (b) and (c) are different. Stud bolts are used in the structure shown in **Figure 8-26** (b). **Figure 8-26** (d) is a simplified drawing of a stud bolt. Cap screws are used in the structure shown in **Figure 8-26** (c). In each of these two structures, the holes in the top member of the pressure vessel are free of thread, and the threaded member is very thick. In the assembly of structure (b), the stud bolts are "planted" into the threaded member first, and then the top cover is placed. Removing the cover does not require removal of the bolts. The assembly of structure (c) is opposite; here, the screws are assembled last.

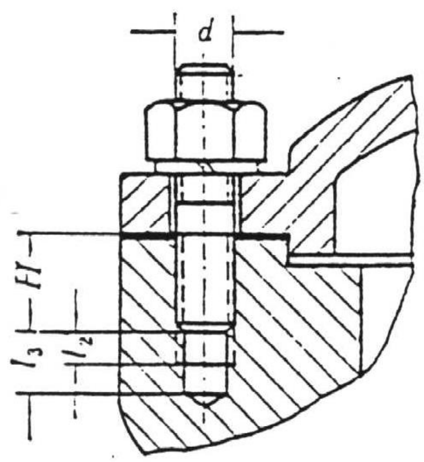
Spring washers are used in each of the structures for the anti-loosening purpose.

Note that both the bodies and the covers of these pressure vessels are cast pieces.

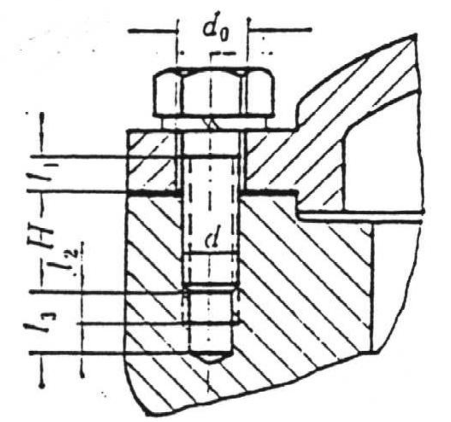
Can we compare these three structures?



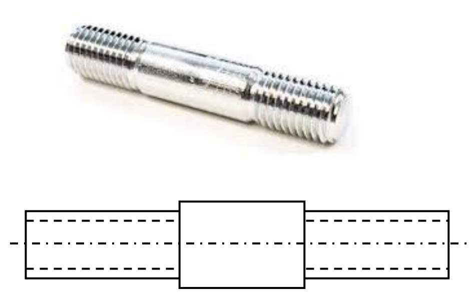
(a) Through bolted



(b) Stud bolted



(c) Tap bolted by a cap screw



(d) A stud bolt

Figure 8-26 Three typical structures for a bolted tensile connection.

Preventing loosening

There are a number of designs for loosening prevention, which are 1) double nuts, 2) spring washer of many types, 3) spring cotter pin, 4) self-locking nut, 5) wiring, 6) thread damage, 7) superbolt, 8) wedge-lock washers, 9) lock bolt, and many other innovative structures. The first four are shown in **Figure 8-27**, which are for re-usable bolts. Items 5-6 are shown in **Figure 8-28**, which use permanent measures to lock mating threads. The last three will be shown in YouTube movies.

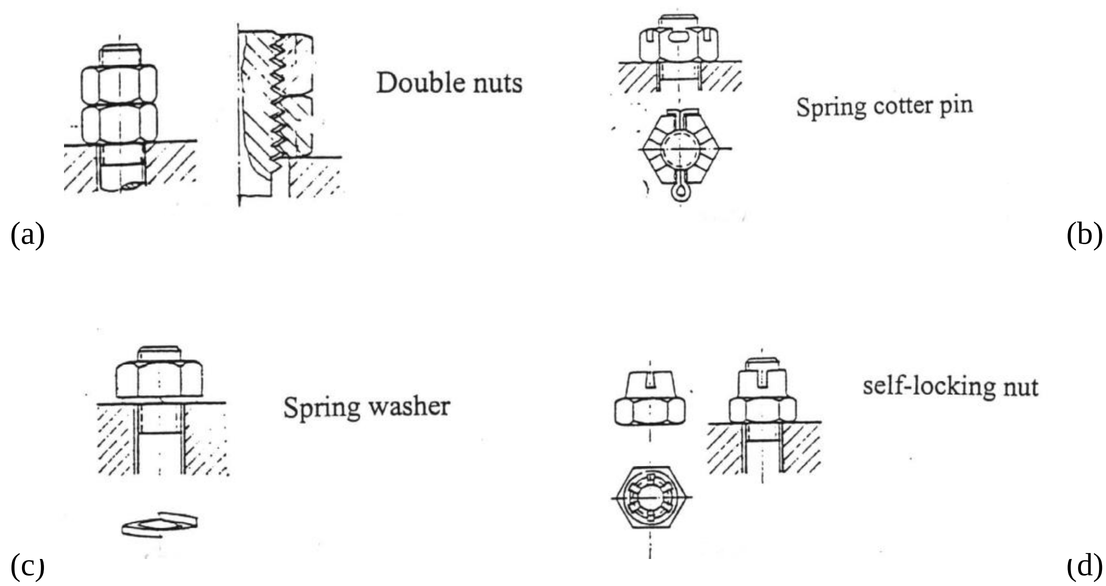


Figure 8-27 Preventing loosening by reusable parts.

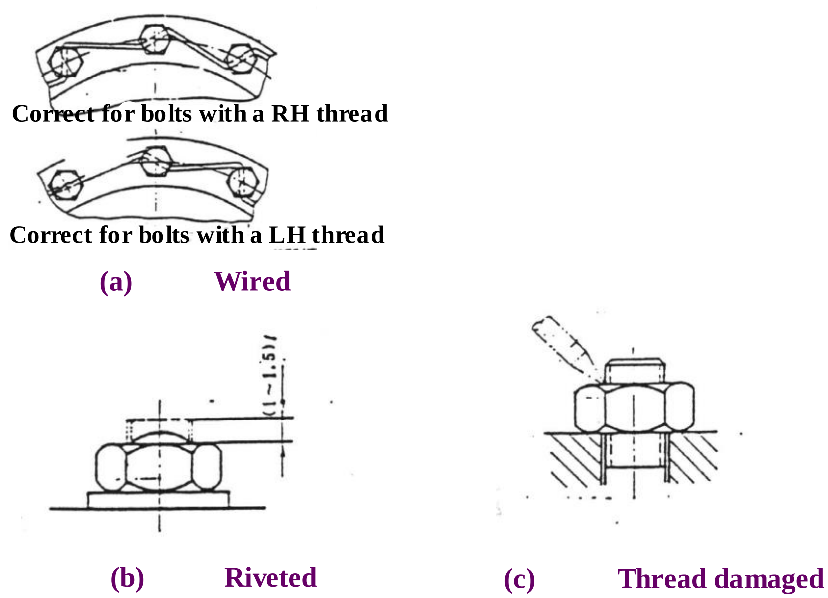


Figure 8-28 Preventing loosening by permanent means.

8-6. Bolt manufacturing

Bolt manufacturing has be highly specialized, and two very important processes are involved, cold heading and cold thread rolling. The shapes are formed through compression, and the process produces zero waste. More will be discussed in class.

Chapter summary

This chapter studies bolted connections and how a tensile connection is made and used to support tensile, shear, bending and torsional loading. The concepts of bolt and member stiffness, joint constant, preload, proof strength are introduced. Factors of safety against plastic deformation, joint separation, and bolt fatigue are derived. Issues related to bolted-joint design are explained.

References

Shigley, J. and Mischke, C., 1989, 2001, *Mechanical Engineering Design*, McGraw Hills.
Pu, L., Chen, D., and Wu, L., 1982, 2013, *Machine Design*, China Higher Education Press.

Media: Several U-Tube movies are on line. 315 students should watch the short U-Tube movies before and after the classes.