Instructional Objectives

At the end of this lesson, the students should have the knowledge of

- Different types of stresses developed in screw fasteners due to initial tightening and external load.
- Combined effect of initial tightening and external load on a bolted joint.
- Leak proof joints and condition for joint separation.

4.4.1 Stresses in screw fastenings

It is necessary to determine the stresses in screw fastening due to both static and dynamic loading in order to determine their dimensions. In order to design for static loading both initial tightening and external loadings need be known.

4.4.1.1 Initial tightening load

When a nut is tightened over a screw following stresses are induced:

(a) Tensile stresses due to stretching of the bolt

(b) Torsional shear stress due to frictional resistance at the threads.

(c) Shear stress across threads

(d) Compressive or crushing stress on the threads

(e) Bending stress if the surfaces under the bolt head or nut are not perfectly normal to the bolt axis.

(a) Tensile stress

Since none of the above mentioned stresses can be accurately determined bolts are usually designed on the basis of direct tensile stress with a large factor of safety. The initial tension in the bolt may be estimated by an empirical relation P_1=284 d kN, where the nominal bolt diameter d is given in mm. The relation is used for making the joint leak proof. If leak proofing is not required half of the above estimated load may be used. However, since initial stress is inversely proportional to square of the diameter $\left(\sigma = \frac{284d}{\frac{\pi}{4}d^2}\right)$, bolts of smaller diameter such as M16 or M8 may fail during initial tightening. In such cases torque wrenches must be used to apply known load.

The torque in wrenches is given by $T = C P_1 d$ where, C is a constant depending on coefficient of friction at the mating surfaces, P_1 is tightening up load and d is the bolt diameter.

(b) Torsional shear stress

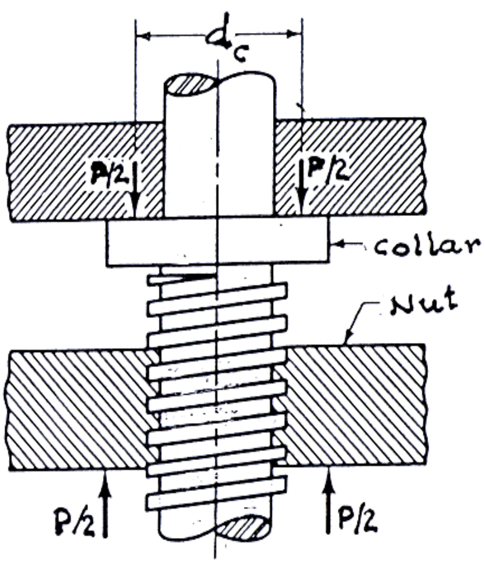
This is given by $\tau = \frac{16T}{\pi d_c^3}$ where T is the torque and d_c the core diameter. We may relate torque T to the tightening load P_1 in a power screw configuration (**figure-4.4.1.1.1**) and taking collar friction into account we may write

$$T = P_1 \frac{d_m}{2}\left(\frac{l + \mu\pi d_m \sec\alpha}{\pi d_m - \mu L \sec\alpha}\right) + \frac{P_1 \mu_c d_{cm}}{2}$$

where d_m and d_{cm} are the mean thread diameter and mean collar diameter respectively, μ and μ_c are the coefficients of thread and collar friction respectively and α is the semi thread angle. If we consider that

$$d_{cm} = \frac{(d_m + 1.5d_m)}{2}$$

then we may write $T = C P_1 d_m$ where C is a constant for a given arrangement. As discussed earlier similar equations are used to find the torque in a wrench.



4.4.1.1.1F- *A typical power screw configuration*

(c) Shear stress across the threads

This is given by $\tau = \frac{3P}{\pi d_c bn}$ where d_c is the core diameter and b is the base width of the thread and n is the number of threads sharing the load

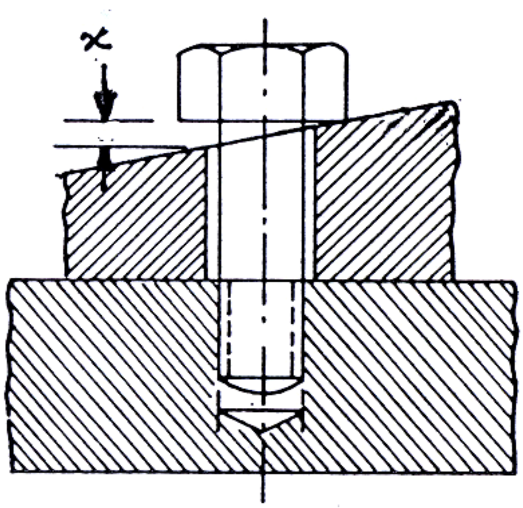
(d) Crushing stress on threads

This is given by $\sigma_c = \frac{P}{\frac{\pi}{4}\left(d_0^2 - d_c^2\right)n}$ where d_0 and d_c are the outside and core diameters as shown in **figure- 4.4.1.1.1**

(e) Bending stress

If the underside of the bolt and the bolted part are not parallel as shown in **figure- 4.4.1.1.2**, the bolt may be subjected to bending and the bending stress may be given by

$\sigma_B = \frac{xE}{2L}$ where x is the difference in height between the extreme corners of the nut or bolt head, L is length of the bolt head shank and E is the young's modulus.



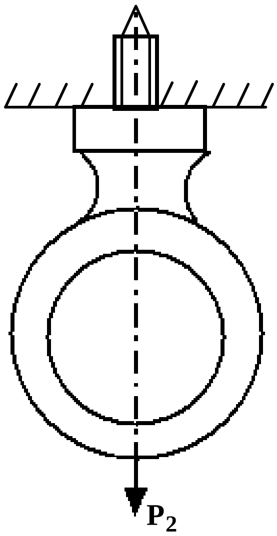
4.4.1.1.2F*- Development of bending stress in a bolt*

4.4.1.2 Stresses due to an external load

If we consider an eye hook bolt as shown in **figure- 4.4.1.2.1** where the complete machinery weight is supported by threaded portion of the bolt, then the bolt is subjected to an axial load and the weakest section will be at the root of the thread. On this basis we may write

$$P_2 = \frac{\pi}{4} d_c^{\,2}\,\sigma_t$$

where for fine threads d_c =0.88d and for coarse threads d_c =0.84d, d being the nominal diameter.

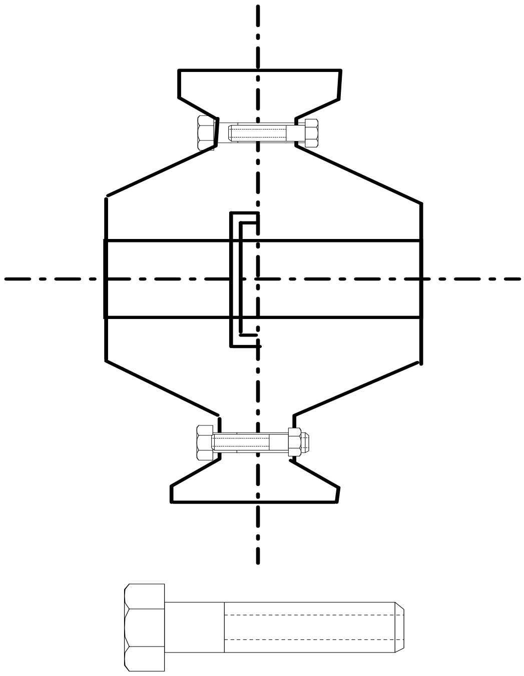


4.4.1.2.1F*- An eye hook bolt*

Bolts are occasionally subjected to shear loads also, for example bolts in a flange coupling as shown in **figure- 4.4.1.2.2**. It should be remembered in design that shear stress on the bolts must be avoided as much as possible. However if this cannot be avoided the shear plane should be on the shank of the bolt and not the threaded portion. Bolt diameter in such cases may be found from the relation

$$T = n\frac{\pi}{4} d_c^{\,2}\,\tau\,\frac{PCD}{2}$$

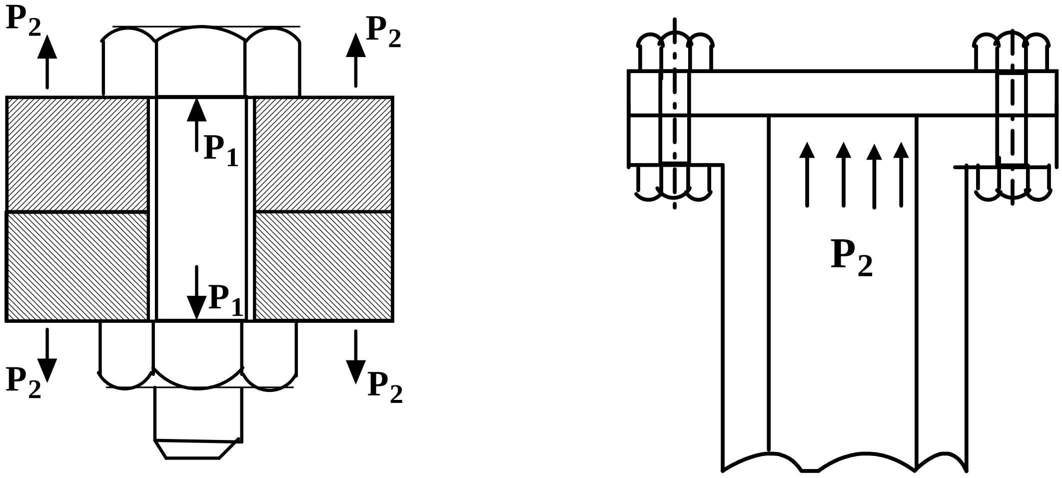
where n is the number of bolts sharing the load, τ is the shear yield stress of the bolt material. If the bolt is subjected to both tensile and shear loads, the shank should be designed for shear and the threaded portion for tension. A diameter slightly larger than that required for both the cases should be used and it should be checked for failure using a suitable failure theory.



__4.4.1.2.2F__- A typical rigid flange coupling

4.4.1.3 Combined effect of initial tightening load and external load

When a bolt is subjected to both initial tightening and external loads i.e. when a preloaded bolt is in tension or compression the resultant load on the bolt will depend on the relative elastic yielding of the bolt and the connected members. This situation may occur in steam engine cylinder cover joint for example. In this case the bolts are initially tightened and then the steam pressure applies a tensile load on the bolts. This is shown in **figure-4.4.1.3.1 (a)** and **4.4.1.3.1 (b).**



(a) **(b)**

__4.4.1.3.1F__- A bolted joint subjected to both initial tightening and external load

Initially due to preloading the bolt is elongated and the connected members are compressed. When the external load P is applied, the bolt deformation increases and the compression of the connected members decreases. Here P_1 and P_2 in **figure 4.4.1.3.1 (a)** are the tensile loads on the bolt due to initial tightening and external load respectively.

The increase in bolt deformation is given by $\delta_B = \frac{P_b}{K_b}$ and decrease in member compression is $\delta_C = \frac{P_C}{K_C}$ where, P_b is the share of P_2 in bolt, P_C is the share of P_2 in members, K_b and K_c are the stiffnesses of bolt and members. If the parts are not separated then $\delta_b = \delta_c$ and this gives

$$\frac{P_b}{K_b} = \frac{P_C}{K_C}$$

Therefore, the total applied load P_2 due to steam pressure is given by

$$P_2 = P_b + P_C$$

This gives $P_b = P_2 K$, where $K = \frac{K_b}{(K_b + K_c)}$. Therefore the resultant load on bolt is $P_1 + KP_2$. Sometimes connected members may be more yielding than the bolt and this may occurs when a soft gasket is placed between the surfaces. Under these circumstances

$K_b >> K_c$ or $\frac{K_c}{K_b} << 1$ and this gives $K \approx 1$. Therefore the total load $P = P_1 + P_2$

Normally K has a value around 0.25 or 0.5 for a hard copper gasket with long through bolts. On the other hand if $K_C >> K_b$, K approaches zero and the total load P equals the initial tightening load. This may occur when there is no soft gasket and metal to metal contact occurs. This is not desirable. Some typical values of the constant K are given in **table 4.4.1.3.1**.

Type of joint	K
Metal to metal contact with through bolt Hard copper gasket with long through bolt Soft copper gasket with through bolts Soft packing with through bolts Soft packing with studs	0-0.1 0.25-0.5 0.50- 0.75 0.75- 1.00 1.00

4.4.1.3.1T

4.4.2 Leak proof joint

The above analysis is true as long as some initial compression exists. If the external load is large enough the compression will be completely removed and the whole external load will be carried by the bolt and the members may bodily separate leading to leakage.

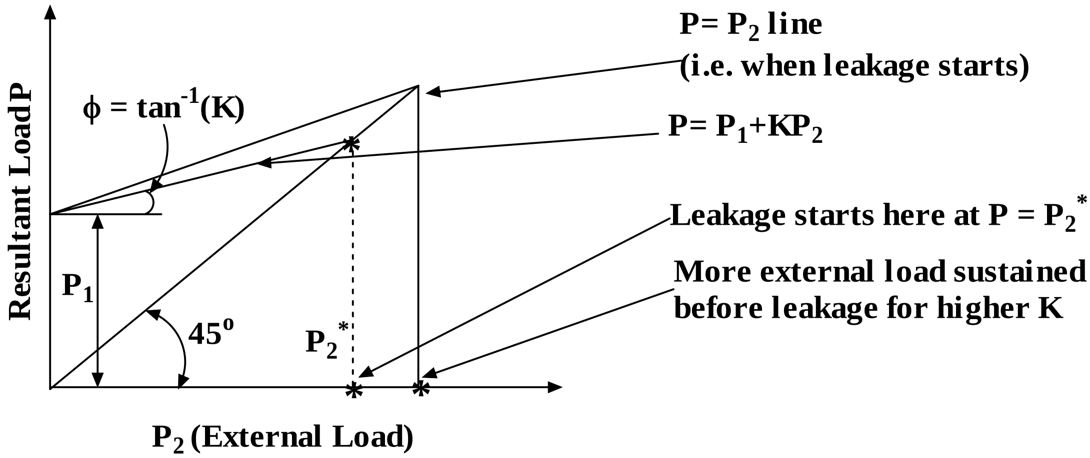
Therefore, the condition for leak proof joint is $\frac{P_1}{K_c} > \frac{P_b}{K_b}$. Substituting $P_b = P_2 K$ and $\frac{K_c}{K_b} = \frac{1-K}{K}$ the condition for a leak proof joint reduces to $P_1 > P_2(1-K)$. It is therefore necessary to maintain a minimum level of initial tightening to avoid leakage.

4.4.3 Joint separation

Clearly if the resultant load on a bolt vanishes a joint would separate and the condition for joint separating may be written as $P_1 + KP_2 = 0$

Therefore if $P_1 > KP_2$ and $P_1 < A_b\ \sigma_{tyb}$, there will be no joint separation. Here A_b and σ_{tyb} are the bolt contact area and tensile yield stress of the bolt material respectively and condition ensures that there would be no yielding of the bolt due to initial tightening load.

The requirement for higher initial tension and higher gasket factor (K) for a better joint may be explained by the simple diagram as in **figure- 4.4.3.1**.



***4.4.3.1F** – Force diagram for joint separation*

4.4.4 Problems with Answers

Q.1: 12 M20 x 2.5C bolts are used to hold the cylinder head of a reciprocating air compressor in position. The air pressure is 7 MPa and the cylinder bore diameter is 100 mm. A soft copper gasket with long bolts is used for sealing. If the tensile yield stress of the bolt material is 500 MPa find the suitability of the bolt for the purpose. Check if the joint is leak proof and also if any joint separation may occur.

A.1:

According to Indian Standard Thread designation M20 x 2.5C indicates a metric bolt of nominal diameter 20 mm and a course pitch of 2.5 mm. Some typical bolt dimensions are quoted in **table-4.4.4.1** as recommended by I.S. 4218-1978 (Part VI) :

Designation	Pitch (mm)	Minor Diameter		Stress area (mm^2)
		Bolt (mm)	Nut (mm)	
M2	0.40	1.509	1.567	207
M5	0.8	4.019	4.134	14.2
M10	1.25	8.466	8.647	61.6
M16	1.5	14.160	14.376	167
M20	1.5	18.160	18.376	272
M24	2	21.546	21.835	384

4.4.4.1T-

Based on this for M20 x 2.5C bolt the initial tightening load is given by P_1=284 d which is 56.8 kN.

External load on each bolt $P_2 = \dfrac{\frac{\pi}{4} x \left(0.1\right)^2 x 7x10^6}{12}$ i.e. 4.58 kN.

From section 4.4.1.3 the constant K = 0.5-0.75. Taking an average value of K=0.625 the total resultant load P is given by P=56.8+0.625x4.58 = 59.66 kN.

From the table above, the stress area for M20 x 2.5C bolt is 245 mm^2. The stress produced in the bolt = $\dfrac{59.66x10^3}{245x10^{-6}} = 243\,MPa$.

The stress is within the yield stress of the material and gives a factor of safety of 500/243 ≈ 2.

<u>Test for leak proof joint</u>

Refer to section 4.4.2. The condition for leak proofing is $P_1 > P_2$ (1-K).

P_2 (1-K) = 1.717 kN which is much less than P_1= 56.8 kN. Therefore the joint is leak proof.

<u>Test for joint separation</u>

Two conditions are $P_1 > KP_2$ and $P_1 < A\sigma_{ty}$. KP_2 =2.86 kN which is much less than P_1 = 56.8 kN and $A_b\sigma_{ty}$= 245x10^{-6} = 122.5 kN which is much higher than P_1 = 56.8 kN. Therefore the joint separation will not take place.

Q.2: In a steam engine the steam pressure is 2 MPa and the cylinder diameter is 250 mm. The contact surfaces of the head and cylinder are ground and no packing is required. Choose a suitable bolt so that the joint is leak proof. Assume number of bolts to be used is 12.

A.2:

Let the nominal diameter of the bolt to be chosen is d mm. The initial tightening load = 248d kg i.e. 2.48d kN.

The external load per bolt = $\frac{\pi}{4} x (0.25)^2 x 2x10^6 \big/ 12 = 8.18\,kN$. Now the condition for leak proofing is $P_1 > P_2 (1-K)$. Here for ground surfaces K=0.1. Therefore

2.48d = 8.18 x 0.9. This gives d = 2.97 mm. This is the minimum requirement and we take d = 10 mm. We also check for yielding $(P_1 + K P_2)/ A_b < \sigma_{ty}$.

Here, A_b from the **table-4.4.4.1** is 58 mm^2 and therefore $(P_1 + K P_2)/ A_b$=

$\frac{(2.48x10 + 0.1x8.18)x10^3}{58}$ = 442 MPa which is well within the range. It therefore seems that from strength point of view a smaller diameter bolt will suffice. However, the choice of M10 x 1.5C would provide a good safety margin and rigidity.

4.4.5 Summary of this Lesson

In this lesson stresses developed in screw fastenings due to initial tightening load and external load have been discussed along with relevant examples. Following this combined effect of initial tightening and external load on bolts is discussed and the condition for the bolted parts not to separate is derived. Condition for leak proof joints and joint separation have also been discussed.

4.4.6 Reference for Module-4

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4) Mechanical engineering design by Joseph E. Shigley, McGraw Hill, 1986.
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