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Applying, measuring and maintaining pretension in steel bolts

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APPLYING, MEASURING AND MAINTAINING PRETENSION IN STEEL BOLTS

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APPLYING, MEASURING AND MAINTAINING PRETENSION IN STEEL BOLTS

1. INTRODUCTION

Designing a bolted joint assembly often requires that the bolt is pretensioned to a prescribed load. ESDU 85021* deals with calculating the desired amount of pretension.

This Item gives guidance on the methods of applying pretension in a plain nut and bolt assembly, and on ways of assessing that load. In particular, this Item presents a relationship between the tightening torque that is applied to the bolt head or to an engaged nut and the resulting tensile load induced in the bolt or the stud. Recommendations are discussed for suitable methods of maintaining the tightness of the bolt during operation under steady loads. Note that no loading beyond the yield stress of the bolt material has been considered.

1.1 Notation

Both SI and British units are quoted but any coherent system of units may be used

D	nominal diameter of bolt	m	in
d	core diameter of bolt	m	in
d_2	basic pitch diameter of the bolt thread	m	in
F	axial tensile load in bolt	N	lbf
f	average axial tensile stress in bolt core area	N/m ²	lbf/in ²
K	joint coefficient = $T/(FD)$		
p	thread pitch = $\pi d_2 \tan \alpha$	m	in
r_i	inner radius of annular bearing face of nut or bolt head	m	in
r_o	outer radius of annular bearing face of nut or bolt head	m	in
r_t	effective thread (contact) radius	m	in
T	tightening torque applied to nut or bolt head	N m	lbf in
α	thread helix angle	degree	degree
β	half thread flank angle	degree	degree
μ_a	friction coefficient between annular bearing faces		
μ_t	friction coefficient between thread mating surfaces		

* ESDU 85021, "Analysis of pretensioned bolted joints subject to tensile (separating) forces".

2. THE TORQUE/TENSION RELATIONSHIP

2.1 General

A common method of achieving pretension in a plain nut and bolt assembly is to use an empirically determined torque/tension relationship. The torque that is applied to the nut to pretension a bolt is absorbed in three main ways. Firstly, the friction between the nut and the abutment or thrust surfaces accounts for some of the applied torque. Secondly, the friction between the mating threads of the nut and the bolt absorbs a significant part of the torque. These two frictional effects may, in some cases, account for up to 90 per cent of the applied torque. Finally there is the component which drives the mating threads up the helix thus creating a tensile stress in the bolt. Therefore, the clamping force induced in a nut and bolt assembly by a given torque will be considerably greater if a lubricant is applied to the thread and thrust faces.

It follows that it is important to distinguish between lubricated and unlubricated threads. Repeated tightening of a lubricated thread assembly to the same torque will give a constant (within ± 5 per cent) pretension if new lubricant is applied between each tightening. However, it should be noted that if an initially lubricated thread is repeatedly tightened without further addition of lubricant, the pretension may be reduced considerably (to 40 per cent after 50 tightenings). Although lubrication of the threads increases the bolt tension for a given torque, it also promotes loosening of the nut in cases where proprietary or other locking devices are not used (see Section 6). Note that repeated uses of coated or plated bolts may impair the integrity of the coating and can cause the bolts to seize or lead to a false indication of the torque tension during tightening.

2.2 Mathematical Relationship

A general algebraic expression accounting for the main influencing factors in the torque/tension relationship is

$$T = F \left[\left(\frac{r_i + r_o}{2} \right) \mu_a + \frac{r_t (\tan \alpha + \mu_t \sec \beta)}{1 - \mu_t \sec \beta \tan \alpha} \right] \quad (2.1)$$

Since the friction conditions have a dominating influence, and exact values of these coefficients are not likely to be known, the usefulness of this relationship in practical applications is rather limited.

Assuming that $\mu_t \sec \beta \tan \alpha \rightarrow 0$, Equation (2.1) reduces to

$$T = F \left[\left(\frac{r_i + r_o}{2} \right) \mu_a + \frac{p}{2\pi} + \frac{d_2 \mu_t}{2 \cos \beta} \right] \quad (2.2)$$

For general use, and if only a very approximate assessment is required, a simple empirical relationship that avoids the need to quantify μ_a and μ_t , is

$$T = F d_2 / 4 \quad (2.3)$$

which gives an average value for unlubricated threads.

2.3 Other Influencing Factors

A detailed evaluation has been carried out on a number of factors that are thought to have a significant effect on the torque/tension relationship.

2.3.1 Torque wrench assessment

The accuracy with which a wrench can be set to apply the required torque at the nut is important. The evaluation here considers the basic types of wrench that are likely to be used.

- (a) Torque meters that incorporate a dial indicator with a slave pointer to record the maximum torque applied.
- (b) Break-back wrenches, in which the drive is disengaged at a pre-set torque value thereby limiting the maximum torque that can be applied.
- (c) Pneumatic nut runners, in which the torque setting is achieved by a screw adjustment that controls the spring in the clutch mechanism.

The measured tightening of lubricated and unlubricated nuts of various sizes with the types of wrench listed above show that, while there is no significant difference in the joint coefficient (K) value obtained with carefully set hand operated wrenches, the values that were achieved using the pneumatic nut runner differ considerably. Using pneumatic wrenches, the unlubricated threads give lower K values (that is, they require less torque to achieve a given bolt load) and the lubricated threads give bigger values of K .

2.3.2 Bolt head tightening

It is sometimes more convenient to tighten the fastener from the bolt head end, rather than at the nut. The different methods used to manufacture the nut and the bolt cause different surface geometries and friction characteristics. Experiments carried out on 50 specimens using hand torque tightening at a controlled tightening rate at the bolt head and at the nut produced a wide scatter of results. The K values for bolt head tightening can be up to 50 per cent greater than those found for nut tightening.

2.3.3 Rate of tightening

The effect of the tightening speed is neither consistent nor repeatable. With both lubricated and unlubricated threads a scatter of ± 5 per cent in the K value is possible for a wide range of rates of tightening.

2.3.4 Miscellaneous effects

Any variation in the bolt loaded length produces no discernible effect to the joint coefficient. Similarly, an intermittent or a stop-start tightening has been discounted as having any significant effect. Although between batches of bolts there are differences in geometry and surface finish that occur within acceptable tolerances, these are also discounted as factors that could have influence the tightening performance.

3. MEASUREMENT OF BOLT LOADS

Although the applied torque is the most commonly used measure of bolt tension, it must be regarded as only an approximate indication of true bolt tightness. Exact repeatability, even within an allegedly identical batch, cannot be guaranteed. A constant torque applied to a batch of nuts and bolts in an as received unlubricated condition may give maximum bolt loads that are 50 per cent higher than the minimum. For installations where greater precision is needed, other methods to determine the bolt load should be used. However, as Table 3.1 shows, a greater accuracy can be achieved only at extra cost.

Two-part fastening systems, such as Hi-Lok (Derivation 26), are interference-fit fasteners that apply a controlled amount of torque during assembly. Such fasteners consistently achieve the design value of pretension and eliminate the need for any later torque inspection.

TABLE 3.1 Relative cost and accuracy from a particular tightening evaluation

<i>Tightening method*</i>	<i>Relative cost</i>	<i>Accuracy (per cent)</i>
Tightening by “feel”	1	± 35
Torque wrench	1.5 to 2	± 25 to 30
Nut rotation (Section 3.2)	3	± 15 to 20
Load indicating devices, <i>e.g.</i> calibrated washers (Section 3.3)	7 to 8	± 10 to 15
Bolt heating	8	± 15
Mechanical tension indication	10	± 10
Ultrasonic (Section 3.5)	12	± 10
Bolt extension (Section 3.1)	15	± 3 to 5
Hydraulic bolt tensioning	15	± 12
Strain gauge measurement (Section 3.4)	20	± 1
SmartBolt system – a visual indication of tension	5	± 1

* Factory assembly conditions.

3.1 Bolt Extension

This method is normally based on the measurement of the overall length of a bolt using vernier callipers or micrometers. Unless the bolts are calibrated experimentally and a true load/extension curve is obtained, allowance must be made for different stiffnesses in the threaded part of the bolt, the shank and other reduced sections otherwise the calculated load using the total strain will be approximate. A greater accuracy of calculated strain can be obtained using longer bolts.

3.2 Nut Rotation

This method of achieving a specified bolt tension should be regarded as inaccurate. The tension is calculated on the basis that the bolt is extended by one thread pitch length for every 360 degrees of nut rotation. Even after the initial “bedding down” stage of tightening a bolted assembly, the assumption that nut rotation is directly related to bolt strain is inaccurate since no allowance has been made for the compressive flexibility of the clamped components.