



ESDU 14002

Issued November 2012
Supersedes ESDU 87023

Lock nuts and other thread locking devices

Institution of
**MECHANICAL
ENGINEERS**

Endorsed by
The Institution of Mechanical Engineers

AVA Documents
Document Sharing Hub

ESDU DATA ITEMS

Data Items provide validated information in engineering design and analysis for use by, or under the supervision of, professionally qualified engineers. The data are founded on an evaluation of all the relevant information, both published and unpublished, and are invariably supported by original work of ESDU staff engineers or consultants. The whole process is subject to independent review for which crucial support is provided by industrial companies, government research laboratories, universities and others from around the world through the participation of some of their leading experts on ESDU Technical Committees. This process ensures that the results of much valuable work (theoretical, experimental and operational), which may not be widely available or in a readily usable form, can be communicated concisely and accurately to the engineering community.

We are constantly striving to develop new work and review data already issued. Any comments arising out of your use of our data, or any suggestions for new topics or information that might lead to improvements, will help us to provide a better service.

THE PREPARATION OF THIS DATA ITEM

The work on this particular Data Item was monitored and guided by the Stress Analysis and Strength of Components Committee. This Committee first met in 1964 and now has the following membership:

Chairman

Prof. T.G.F. Gray — University of Strathclyde, Glasgow

Members

Mr J. Baghurst	— Independent
Dr B. Bhakta	— Meggitt Polymers and Composites, Loughborough
Dr I.J. Bickley	— Independent
Dr M. Heller*	— Air Vehicles Division, DSTO, Victoria, Australia
Dr J.W.H. Price*	— Integrity Partners, Australia.

* Corresponding Member

The ESDU Stress and Strength Series is recognised as a valuable source of validated data and methods by the Institution of Mechanical Engineers and its Structural Technology and Materials Group is involved in continually reassessing the scope and focus of the content of this Series.

The construction and subsequent development of the Data Item were undertaken by

Dr D.L. Downs — Senior Engineer

The person with overall responsibility for the work in this subject area is Mr A.C. Quilter, Head of Strength Analysis Group and Technical Director – Aerospace.

LOCK NUTS AND OTHER THREAD LOCKING DEVICES

CONTENTS

	Page
1. INTRODUCTION	1
1.1 Notation	1
2. VIBRATION IN BOLTED ASSEMBLIES	2
3. DOUBLE PLAIN LOCK NUTS	2
3.1 General	2
3.2 Tightening Mechanism	3
3.3 The Effect of Backlash and Joint Stiffness in Lock Nuts	6
3.4 Vibration Resistance of ¼ in UNF Bolts	8
3.5 Fatigue	9
4. LOCKING BY ADHESION	9
5. MECHANICAL LOCKING	10
6. OTHER LOCKING DEVICES	10
6.1 Single Lock Nuts	10
6.2 Nut Thread Form	11
6.3 Lock Bolts	11
7. ENDURANCE CURVES	12
7.1 Combined Transverse and Axial Loading	12
8. UNSUITABLE WASHERS AND NUTS	12
9. DERIVATION AND REFERENCES	13
9.1 Derivation	13
9.2 References	14
10. TABLES AND FIGURES	15
10.1 Tables	15
10.2 Figures	28

LOCK NUTS AND OTHER THREAD LOCKING DEVICES

1. INTRODUCTION

This Data Item gives guidance on fastener locking devices which are commonly available to the designer for cases where the installation is subjected to severe vibration in service, see ESDU TM 175*. A bolted assembly is usually designed on the basis that the bolt is pretensioned (or pre-load) to a prescribed load and ESDU 14001† deals with methods of achieving the required pre load and maintaining the bolt load in service. This Data Item provides more assistance on retaining the bolt load.

If a bolted joint experiences dynamic shear loading, such as vibration (see Section 2), during its lifetime then fastener locking devices such as those on Table 10.1 can be used to reduce the loss of pre-load or even stop the joint falling apart. Fastener locking devices are chosen based on their different merits such as reusability, performance at very low or high temperatures, the other materials that make up the joint, any specific locking features or if the bolted joint needs to be disassembled and re-assembled at any time.

In Section 3, tightening of a double plain lock nut is examined and assembly guidelines are recommended for the bolted joint. The improvement of the resistance of the bolt to self-loosening and fatigue when double plain lock nuts are used is also discussed in Section 3.

Other methods of securing a bolted joint such as by adhesion, mechanical locking or with alternative bolt retention devices are considered in Sections 4, 5 and 6, respectively. In Section 7, endurance curves show the change of clamping force when dynamic shear loads are applied to different nut and bolt assemblies. Section 8 gives a list of unsuitable washers and nuts and the Derivations and References used in production of this Data Item are listed in Section 9.

1.1 Notation

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

A	effective cross-sectional area of bolt	m^2	in^2
a	diametral tolerance of effective diameters of fastener assembly	m	in
D	effective diameter of bolt	m	in
d	effective diameter of nut	m	in
E	modulus of elasticity of bolt material	N/m^2	lbf/in^2
F_1	initial bolt pre-load induced by first nut in a plain lock nut system	N	lbf
F_2	second stage bolt pre-load carried by second nut in a plain lock nut system	N	lbf
F'_2	third stage bolt pre-load carried by second nut in a plain lock nut system	N	lbf

* ESDU TM 175, "Self-loosening of bolted joints".

† ESDU 14001, "Applying, measuring and maintaining pretension in steel bolts".

F_f	final total bolt pre-load carried by second nut in a plain lock nut system	N	lbf
k	stiffness of load path from second nut through bolt and back through joint material and first nut to interface between nuts	N/m	lbf/in
L	joint clamping length (measured from bolt head)	m	in
δ_1	elongation of bolt induced by F_1	m	in
δ_2	elongation of bolt during third stage tightening of second nut (can be evaluated from thread tolerances between bolt and first nut)	m	in
θ	thread form included angle	degree	degree

Suffixes

min, max minimum and maximum values, respectively

2. VIBRATION IN BOLTED ASSEMBLIES

Generally, an adequately tightened nut and bolt assembly is not likely to loosen in service under steady operating conditions. Usually after initial tightening, some pre-load is lost when the surfaces of the mating components bed-in and/or fret. The amount of loss depends on factors such as the bolt length and the hardness of the mating surfaces (clamped members made from softer steels may cause the bolt to lose up to 20 per cent pre-load, see ESDU TM 175). Other and more serious losses of clamping force can occur when bolted assemblies are subjected to vibration in service.

Vibratory motions cause a bolt to lose its pre-load due to minute slippages at the contact surfaces in the bolt and the assembly. If the vibration is sufficiently intense then there may be a condition of zero friction at the contact surfaces for short intervals during each cycle. Under this condition, a net loosening torque developed by the wedge effect of the axially loaded nut and bolt threads initiates bolt self loosening as vibration continues. The most significant loss of pre-load due to nut and/or bolt rotation is due to dynamic loading applied transverse to the bolt axis. Transverse slippage is transmitted through the bolt contact surfaces at the thread faces and at other surfaces in the assembly.

There are many ways of reducing the effects of vibration in a bolted joint. Before resorting to any of the widely available locking methods and without forfeiting the integrity of the assembly, the designer should always consider eliminating the cause of bolt self loosening. For example, if it is possible to use a larger pre-load on the bolt then the friction forces in the assembly increase and improve the resistance to vibratory motion and prevent bolt self loosening. The theoretical internal loosening torque is directly proportional to the helix angle of the bolt thread therefore using a bolt with a coarse thread pitch (a large helix angle) would generate a greater loosening torque and is less vibration resistant than fine pitch thread bolts.

3. DOUBLE PLAIN LOCK NUTS

3.1 General

The double plain nut locking system is an effective method to avoid the need for specialist or proprietary items in bolted joints. Furthermore this method only requires the minimum of skill in assembly, provided certain basic rules are understood and observed. Note that various names are used for the double plain lock

nuts such as "Joint nuts" and "Free spinning nuts". Effective locking is achieved by tightening the first nut to assemble the joint followed by tightening of the second nut. Historically nuts of different thickness are used; one of standard thickness and the other being half of the standard thickness. As discussed later in this section, it is preferable to assemble the thinner nut first because the threads of the second nut are required to carry a greater load. However, this is not essential. Double plain lock nuts can also be used in assemblies where the members are not clamped but are allowed to slide or rotate relative to each other, for example when linkages are used. If such lock nuts are subject to shear vibration loads, then they might be treated as in ESDU TM 175.

3.2 Tightening Mechanism

Note that the following explanation is for situations where members are clamped.

The initial stage of tightening requires a simple assembly and tightening of the first nut to induce a pre-load, F_1 , less than the final bolt tightening load. Throughout the entire tightening procedure it is important to prevent rotation of the first nut relative to the bolt otherwise this may result in a loss of pre-load. This load is carried totally by the threads of the first nut and any "locking" is provided by friction alone (see Sketch 3.1). The initial pre-load from the first nut is calculated from

$$F_1 = (AE/L)\delta_1. \quad (3.1)$$

For the second stage of tightening, the second nut is assembled and tightened on to the first nut and the pre-load is supported by the threads of the second nut. At this point the threads of the first nut and the bolt disengage. The assembly in this condition is no different from that with a bolt and a single nut except that marginally more of the bolt is tensioned and $F_2 = F_1$ (see Sketch 3.2).

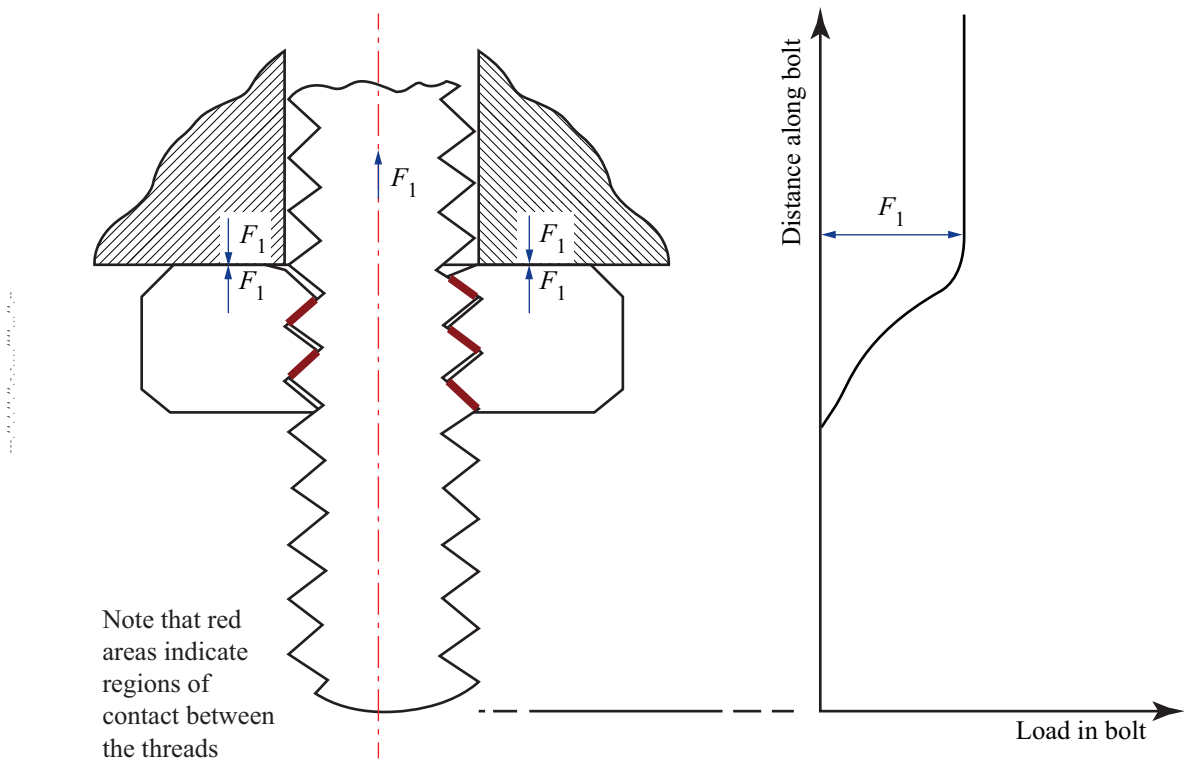
During the third stage, the second nut is tightened further to elongate the bolt within the first nut until the axial thread clearance within the first nut is taken up. At this point the bolt thread outer faces (on the side away from the bolt head) begin to bear on the inner thread flanks of the first nut (see Sketch 3.3) and the final pre-load, F'_2 , is carried by the threads of the second (outer) nut. The final pre-load is the sum of the initial pre-load F_1 in the bolt by the first nut and the additional pre-load produced by the second nut.

The magnitude of F'_2 depends on the class of fit between the bolt threads and the threads of the first nut only (that is, the amount of axial backlash). At this stage

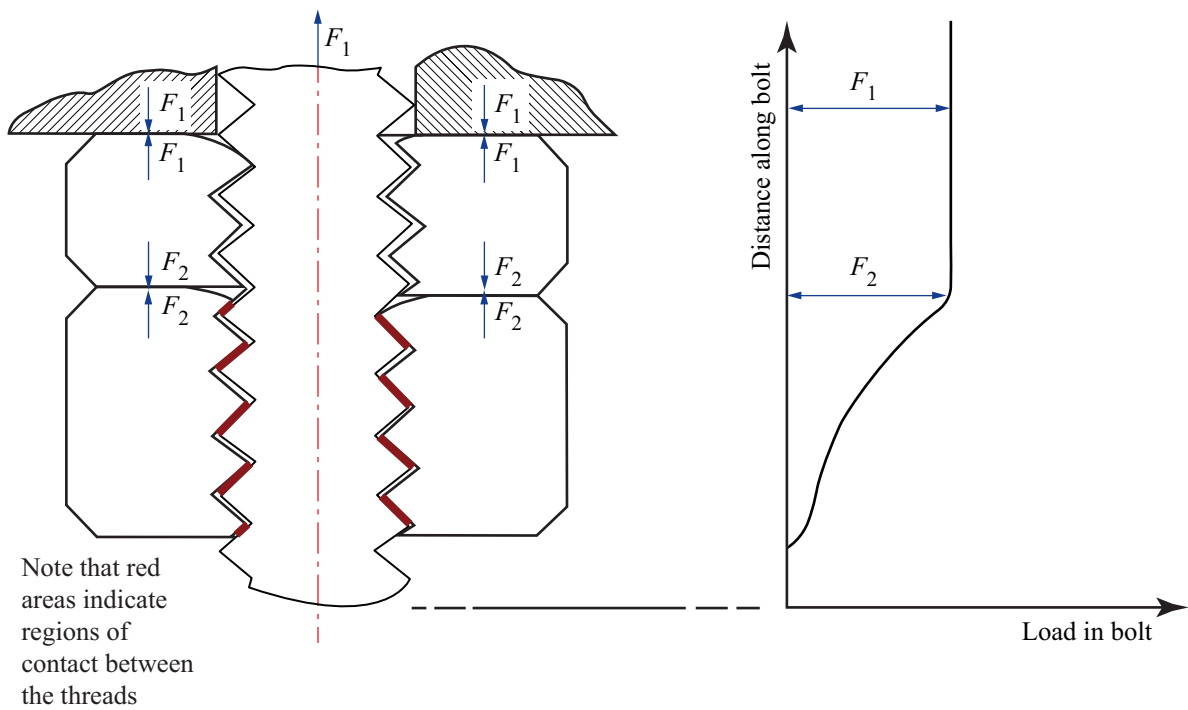
$$F'_2 = F_1 + k\delta_2 \quad (3.2)$$

An example of the estimation of the backlash and the corresponding values of F_1 for various clamped lengths is given in Section 3.3.

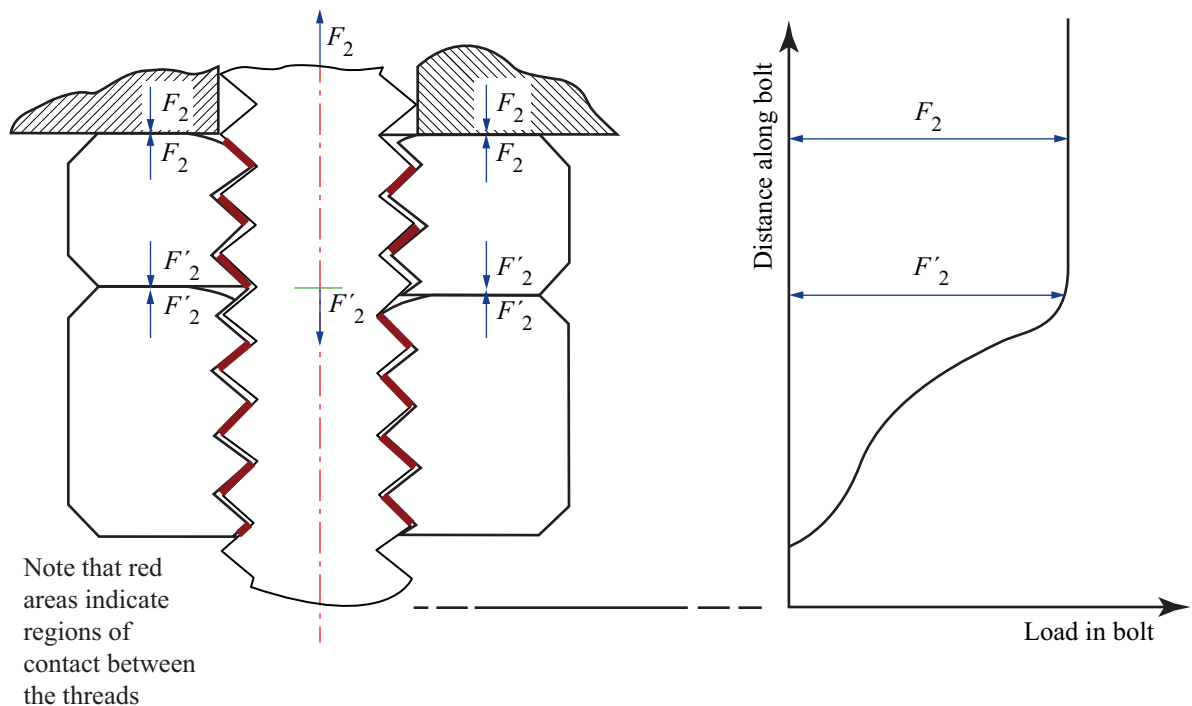
Further tightening of the second nut does not induce larger clamping loads in the bolt but will increase the locking between the two nuts, Sketch 3.4. The system locks because, in addition to the friction force generated by the desired pre-load (which always exists even if only a single nut is used), the pair of nuts are locked against each other by the force $F_f - F_2$ which generates additional frictional resistance at the mating thread faces. The value of F_f is indeterminate since its final value only depends on further tightening of the outer nut against the inner nut. There is a danger that overtightening can strip the threads or break the bolt between the nuts.



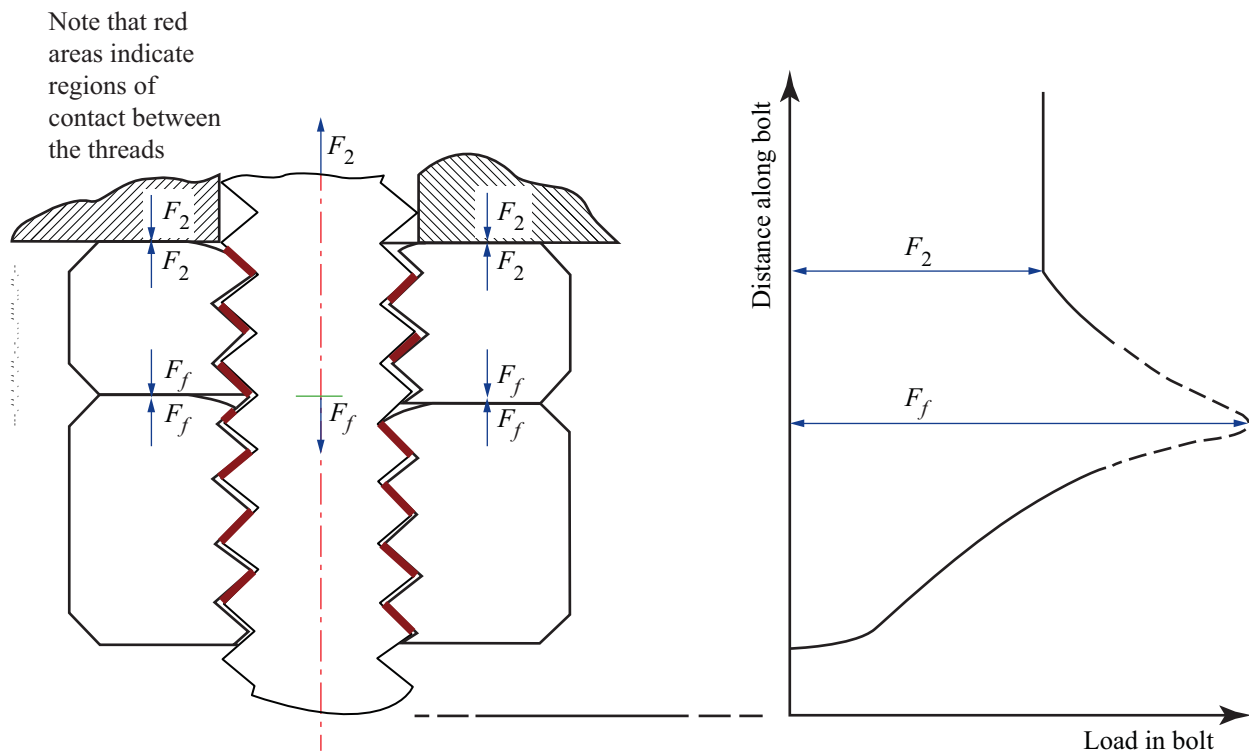
Sketch 3.1 Lock nut tightening – Stage 1 Initial pre-load induced in the fastener



Sketch 3.2 Lock nut tightening – Stage 2 Threads of thinner nut unloaded



Sketch 3.3 Lock nut tightening – Stage 3



Sketch 3.4 Lock nut tightening – final locking

3.3 The Effect of Backlash and Joint Stiffness in Lock Nuts

This example examines joints made using 1/4 in steel bolts with 28 UNF threads. BS 1580* is referenced for the dimensional data. Note that similar data is available for all standard thread forms.

Consider nuts with class 2B fit being assembled with bolts with class 2A fit from BS 1580.

For the bolt

$$D_{\max} = 0.2258 \text{ in} \quad \text{and} \quad D_{\min} = 0.2225 \text{ in.}$$

For the nut

$$d_{\max} = 0.2311 \text{ in} \quad \text{and} \quad d_{\min} = 0.2268 \text{ in.}$$

The maximum and minimum axial backlash values are given by

$$\delta_{2\max} = 2a_{\max} \tan(\theta/2) \quad (3.3)$$

$$\text{and} \quad \delta_{2\min} = 2a_{\min} \tan(\theta/2), \quad (3.4)$$

in which $a_{\max}$ and $a_{\min}$ are

$$a_{\max} = (d_{\max} - D_{\min})/2 = (0.2311 - 0.2225)/2 = 0.0043 \text{ in}$$

$$\text{and} \quad a_{\min} = (d_{\min} - D_{\max})/2 = (0.2268 - 0.2258)/2 = 0.0005 \text{ in.}$$

Noting that $\theta = 60^\circ$ for UNF threads, and substituting the numerical values, gives

$$\delta_{2\max} = 0.0086 \times \tan(60/2) = 4.9652 \times 10^{-3} \text{ in} \quad \text{maximum backlash}$$

$$\text{and} \quad \delta_{2\min} = 0.0010 \times \tan(60/2) = 5.7735 \times 10^{-4} \text{ in} \quad \text{minimum backlash.}$$

The value of k is required for use with δ_2 in Equation (3.2). The total stiffness is the sum of the stiffness of the threaded and un-threaded parts of the bolt, the clamped members and the first nut. These stiffnesses are combined into one stiffness value using the following relationship:

$$k^{-1} = k_1^{-1} + k_2^{-1} + k_3^{-1} + \dots + k_n^{-1},$$

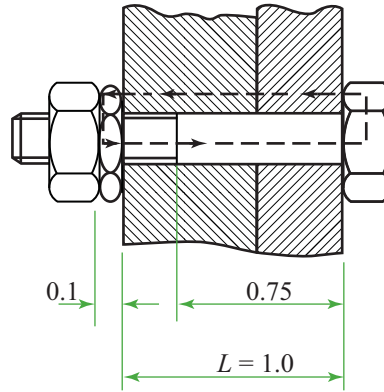
where $k_1, k_2, \dots, k_n$ are the stiffnesses of each component in the load path.

It is assumed that the thinner, first nut and the clamped material have an effective outer diameter of material resisting compression in the bolt that is equal to twice the bolt diameter. Therefore, for this example, the effective outer diameter is 0.5 in. A more accurate assessment of this diameter can be made using Figure 1 of ESDU 85021†.

* BS 1580-1: 2007 "Unified screw threads. Part 1: Screw threads with diameters 1/4 in and larger – Requirements".

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

In order to calculate the stiffnesses, it is assumed that all the components in this joint have the same modulus of elasticity as the bolt material, which is $28.5 \times 10^6 \text{ lbf/in}^2$. Sketch 3.5 shows the values of length assumed for calculating stiffnesses. The root area of threaded portion of bolt, A , is 0.0364 in^2 .



Sketch 3.5
(All dimensions in inches)

The stiffness of threaded part of bolt is

$$\frac{0.0364 \times 28.5 \times 10^6}{1.0 + 0.1 - 0.75} = 2.96 \times 10^6 \text{ lbf/in.}$$

The stiffness of the un-threaded part of bolt is

$$\frac{\pi \times 0.25^2 \times 28.5 \times 10^6}{4 \times 0.75} = 1.86 \times 10^6 \text{ lbf/in.}$$

The stiffness of the clamped members is

$$\frac{\pi \times [(2.0 \times 0.25)^2 - 0.25^2] \times 28.5 \times 10^6}{4 \times 1.0} = 4.20 \times 10^6 \text{ lbf/in.}$$

The stiffness of the thinner, first nut is

$$\frac{\pi \times [(2.0 \times 0.25)^2 - 0.25^2] \times 28.5 \times 10^6}{4 \times 0.1} = 42.0 \times 10^6 \text{ lbf/in.}$$

Therefore the total stiffness of the assembly is

$$k^{-1} = (2.96 \times 10^6)^{-1} + (1.86 \times 10^6)^{-1} + (4.20 \times 10^6)^{-1} + (42.0 \times 10^6)^{-1},$$

so

$$k = 0.879 \times 10^6 \text{ lbf/in.}$$

Given a third stage pre-load of $F'_2 = 2660$ lbf, the value of F_1 can be estimated by re-arranging Equation (3.2):

$$F_1 = F'_2 - k\delta_2. \quad (3.5)$$

Thus, for maximum backlash,

$$F_1 = 2660 - 0.879 \times 10^6 \times 4.9652 \times 10^{-3} = -1702 \text{ lbf}.$$

(Note that the negative sign indicates that it is not possible to take up the entire backlash without exceeding the prescribed bolt load of 2660 lbf. See ESDU 14001 for notes on maintaining the pre-load.)

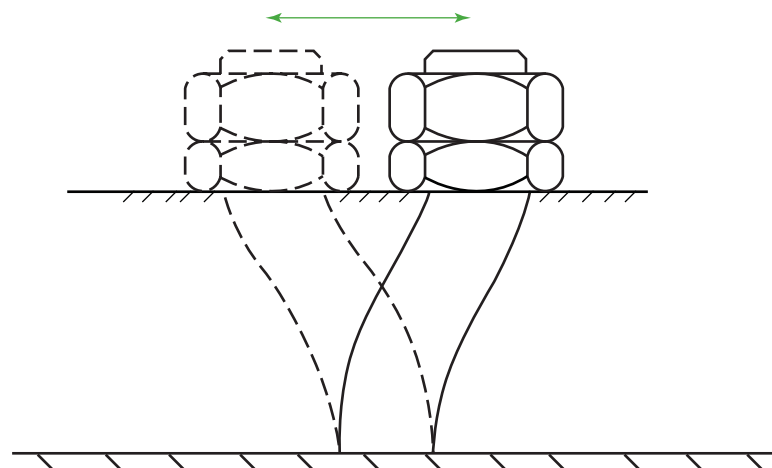
For minimum backlash,

$$F_1 = 2660 - 0.879 \times 10^6 \times 5.7735 \times 10^{-4} = 2152 \text{ lbf}.$$

Table 10.2 and Figure 1 show the calculation of F_1 for other clamping lengths between 1.0 in and 2.0 in. The large variation in F_1 is because of backlash in the first thinner nut.

3.4 Vibration Resistance of 1/4 in UNF Bolts

There is no simple method to calculate the resistance to bolt self loosening using double plain lock nuts as many factors influence the joint. The extensive testing of double plain lock nuts described in Derivations 7, 9 and 10 provides a basis for comparison to bolts with a single nut. The tests on double plain lock nuts were carried out using 1/4 in UNF steel nuts and bolts, uniformly tightened to a pre load of 2300 lbf then laterally vibrated as shown in Sketch 3.6. The amount of bolt self loosening due to the vibration was measured by the change in the clamping force.



Sketch 3.6 Motion of double nut specimen when vibrated

It was found that small variations in the nut configuration would significantly influence the resistance of the bolt to self loosening. For example, if both of the nuts had a "lead on" thread chamfer on one side only, each of the eight possible orders and directions of assembly gave their own characteristic resistance to

loosening. Tests showed conclusively that the joint was most resistant to bolt self loosening if both of the nuts were installed with the chamfered thread face towards the bolt head. This remained valid whether or not the thick or the thin nut was assembled first. From the data for a typical class 2 fit, it was clearly established that the optimum resistance to loosening was achieved using the following procedure.

- (1) Assemble the thin nut first then tighten to 30 per cent of full pre-load.
- (2) Hold the thin nut and the bolt stationary as the thick nut is tightened to the full pre-load.

Figure 2 shows that the resistance to bolt self loosening depends on the severity of the vibration. Larger vibration amplitudes leads to a rapid drop of bolt load when a single nut is used. If double plain lock nuts are used, and the amplitude of vibration is small, then the joint is capable of withstanding a very large number of cycles with no significant reduction of the bolt load. Larger vibration amplitudes result in some loss of bolt load but the improvement over using single nuts is still substantial.

3.5 Fatigue

Most fatigue failures in a single nut and bolt assemblies occur at the fillets of the peak-loaded threads immediately inside the contacting face of the nut. Generally about 75 per cent of the total applied load is carried by the first one or two turns of the bolt thread.

When a second nut is used to improve the resistance to bolt self loosening it also has the advantage of increasing the fatigue life of the assembly if it is subject to tensile separating loading. The locking effect of the second nut redistributes the peak load from the first threads of the first nut and ensures a more uniform load distribution through the mating threads. Figure 3 shows the improved fatigue strength of double nuts compared to single nuts in tests with $\frac{3}{4}$ in BSF steel bolts. The bolts had been tightened to a mean load of 11 200 lbf and the tests were carried out at various alternating loads up to ± 6720 lbf. For endurances in excess of 10^7 cycles, bolted joints using double nuts can carry 50 per cent more alternating load than bolts with single nuts can.

4. LOCKING BY ADHESION

Special adhesives developed specifically for the purpose of maintaining bolt tightness are available commercially but a detailed assessment of their characteristics lies outside the scope of this Data Item. In general, there is always a conflict between the two major requirements of locking adhesives.

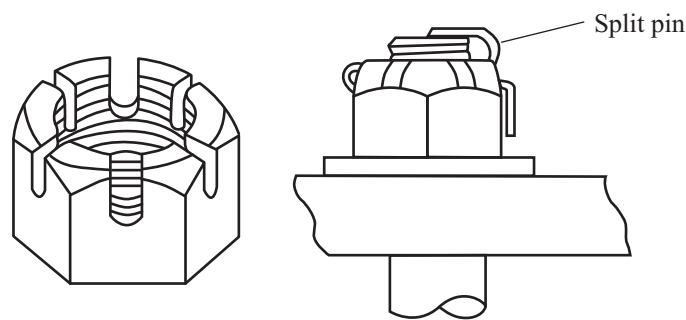
- (1) The adhesives must not provide such a rigid bond that the joints cannot be dismantled.
- (2) The adhesive must provide a rigid bond to resist the loosening torque caused by vibration.

Providing the adhesives manufacturers keep these requirements in acceptable balance, using adhesives to secure bolts has the considerable advantage of simplicity of installation.

It is important that the designer informs the adhesive manufacturer about the precise details of the bolted assembly and expected service conditions, particularly the vibration pattern and the operating temperature. The adhesives manufacturer should be able to provide experimental evidence of suitable products for the size and material of the components used. Having gained such assurances, the user must ensure that all the recommendations such as thread cleanness, re-use of components and storage life of adhesive are observed meticulously. A good adhesive, correctly used, is expected to provide resistance to loosening which is at least comparable to that achieved using double plain lock nuts, and for cases of vibration it can be somewhat better.

5. MECHANICAL LOCKING

Relative movement between the nut and the bolt in service can be prevented entirely by positive or mechanical means. An additional component can be used to secure the nut, such as a split pin (or more rarely in modern engineering a cotter pin) which passes diametrically through both nut and bolt, Sketch 5.1. If a standard nut is used, a hole drilled at its mid-thickness must align with a hole in the bolt when the latter is extended by the pre-load. Using castellated or slotted nuts, that is nuts with a regular pattern of radial grooves in the outer face of the nut, removes the need for drilling holes in the nut and the fitting process is marginally easier, Sketch 5.1. The nut is torqued to a required value then the nut is rotated either forward or backward (depending on the application) to the nearest slot. Accurate fitting is required to achieve this, but the final assembly remains tight without the loss of pre-load, even if the installation is subjected to severe vibrations. Mechanical locking of this type is suitable for use when high service temperatures apply. Note that split pins occasionally shear due to the loosening torque. Effectively, split pins provide enough strength to resist the loosening torque, see ESDU TM 175.



Sketch 5.1 Castellated nut (Derivation 11)

For larger nuts, locking plates (passing over the nut and engaging on its corners) can prevent nut rotation. The plates are fixed to the clamped members by set screws which must be large enough to withstand the shear load produced by the loosening torque.

6. OTHER LOCKING DEVICES

6.1 Single Lock Nuts

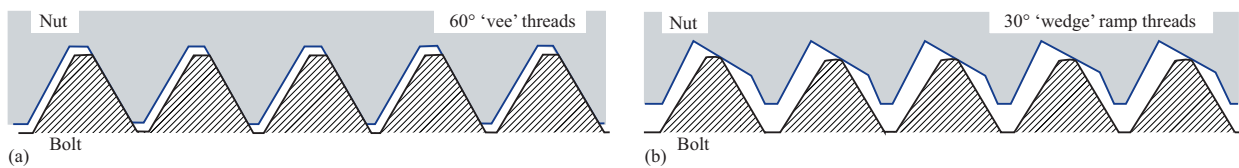
Using commercially available lock nuts is a way to reduce bolt self loosening due to vibration. Each type of lock nut is specialised and, though a full analysis of each one is not given here, Table 10.3 briefly describes common lock nuts and other locking devices as well as a Hi-Lok lock bolt, a step lock bolt and a Heli-Coil threaded insert for comparison.

The lock nuts that are commercially available are generally divided into two categories, friction locking and deformation locking. By suitable modification of the nut geometry, friction forces at the contact surfaces can be increased so that more loosening torque is required to overcome these forces. Other modifications make the nut more liable to deform and provide an interference-fit between the nut and the bolt. Since a mathematical evaluation of the effectiveness of either of these locking methods is not possible it is recommended that each method should be subjected to full experimental examination. Testing should always relate to the specific materials and operating conditions.

6.2 Nut Thread Form

Nuts with the traditional 60° 'vee' thread angle (Sketch 6.1(a)) are prone to the relative motion that causes bolts to self-loosening particularly during shear loading. Also, as around 80 per cent of the pre-load acts upon the first and the second engaged threads this may lead to thread stripping or shearing.

Nuts with a Spirallock thread form are designed with the internal thread including a 30° 'wedge' ramp cut into the root of the female thread, Sketch 6.1(b) (Derivation 22). When the pre-load is applied, the crests of the threads of a bolt are drawn tightly against the 'wedge' ramp in the nut producing locking and redistributing the load throughout all the engaged threads in the nut (Derivation 11).



Sketch 6.1 (a) Traditional and (b) Spirallock thread form

Nuts with the Spirallock thread form are used by diesel engine and vehicle manufactures because they allow for both thermal expansion and contraction without slippage, and they stop the sideways motion that causes vibration loosening. These nuts spin freely during assembly until they reach their final torque retaining position thus eliminating galling. Note that nuts with the Spirallock thread form require much larger torques to develop a specified tension compared to nuts with standard thread forms.

6.3 Lock Bolts

In general, lock bolts are non-expanding high strength fasteners with either a swaged collar or a type of threaded collar to lock them in place (Derivation 11). They are installed in standard drilled holes with a snug fit. Lock bolts are similar to rivets because the locking collar or nut is weak in tension loading and difficult to remove after installation. Also, like rivets, lock bolts have either countersunk or protruding heads and can be installed from one or both sides of the joint. Lock bolts are much faster to install compared to standard nuts and bolts. Lock bolts should only be used when it is not possible to install a comparable conventional nut and bolt since it is difficult to determine if a lock bolt is installed properly. See Derivation 11 for more information on widely available and proprietary lock bolts such as Hi-Lok lock bolts, Jo-bolts, Taper-Lok bolts, Rivnuts, Hi-Shear Rivets and Lightweight Groove Proportioned lock bolts.

Self-loosening happens when there is relative slip between the nut and the bolt threads causing torsion in the bolt. There is also relative slip between the contact surfaces of the bolt or nut and the clamped material leading to a slackening of the torsion. Derivation 13 describes an 8 mm step-lock bolt with eight horizontal steps (*i.e.* no lead angle) in the circumference of the helix of the screw which prevents torsion developing and improves the bolt resistance to self loosening during dynamic shear loading. This bolt can be used with a conventional nut to prevent slip between the bolt and the nut threads.

Heli-Coil[®] inserts made from high tensile strength stainless steel wire are formed into a diamond section to provide a permanent, conventional, 60° internal screw thread insert that protects and strengthens the tapped threads whilst accommodating any standard bolt or screw. Using Heli-Coil thread inserts dramatically increases the thread life and the joints may be disassembled and reassembled easily.