



ESDU TECHNICAL MEMORANDUM

Self-loosening of bolted joints

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SELF-LOOSENING OF BOLTED JOINTS

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SELF-LOOSENING OF BOLTED JOINTS

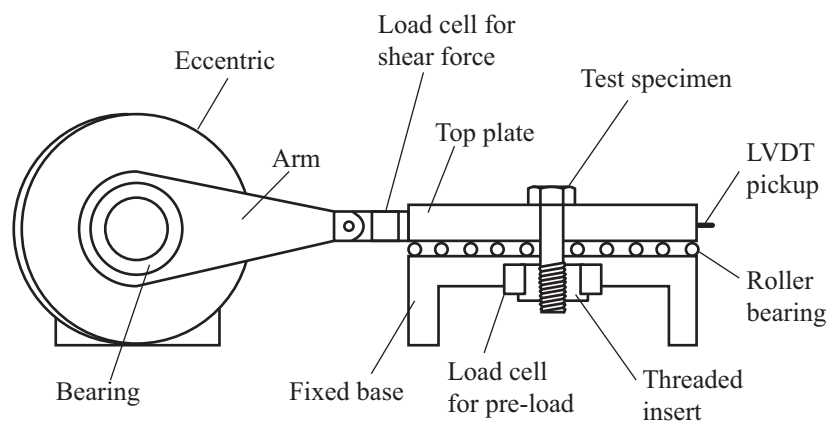
1. INTRODUCTION

The helical screw thread fastener (or bolt) is, undoubtedly, one of the most significant mechanical inventions still produced in large quantities and used in countless structures and machines. Fastener assemblies are chosen, in comparison to other types of joining processes (such as welding, bonding or riveting), because they develop favourable clamping forces and they can be dismantled and re assembled as and when required. Sometimes bolted joints are used in situations with changeable service and environmental conditions, for example when a bolted joint is exposed to dynamic shear loads causing vibration induced bolt loosening. The joint is then prone to failure due to vibration or fatigue. Fortunately, these types of failures are prevented by good joint design and by following correct guidelines during assembly. The aim of this Technical Memorandum is to understand the phenomenon of self loosening of a bolted joint during dynamic shear loading and to give guidance leading to good joint design.

There are numerous reported examples of bolted assemblies that have been subject to transverse shear loads or vibration at some point during their lifetime. These include, but are not limited to: low speed diesel engines for the propulsion of large ships (Derivation 27), aero engine rotor system (Derivation 33), missile tail wings (Derivation 23), switch gears or printing machinery (Derivation 11), engine mounting joint bolts and wheel to hub joints in cars (Derivation 31), structural steel joints using high strength bolts for construction – such as cranes, masts, smokestacks and bridges (Derivation 30), aeronautical, automotive and electronic industries (Derivation 17), aircraft (Derivation 5), points on railway tracks (Derivation 16), the International Space Station (Derivation 34) and The Space Shuttle (Derivation 12).

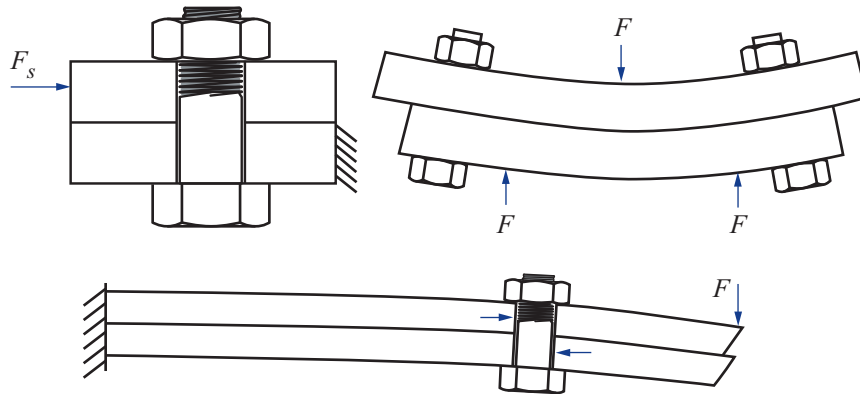
The repetitive application of shear loads in a plane perpendicular to the bolt axis causes the movement of clamped members in the joint and the relative motion of the nut and the bolt threads leading to a loss of clamping force and bolt self-loosening. The two stages that take place during bolt self loosening are slackening (relaxation of the pre-load) and unscrewing (loss of self locking ability, or friction, of the bolt), see Section 2. Generally when not enough pre load is used or if the clamped members are thin (*i.e.* the bolt has a short clamped length), bolted joints are more susceptible to self-loosening.

The phenomenon of self-loosening of bolts can be simulated experimentally using a Junker's test machine, (Derivation 3 or similar, see Sketch 1.1), to repeatedly apply a transverse shear load to the joint until it fails. During these types of test, the number of cycles, the loss of clamping force and the nut or bolt rotation are recorded for analysis.



Sketch 1.1 Schematic drawing showing a simulation of dynamic shear loading applied to a bolt (Derivation 19)

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Sketch 1.2 Common causes of shear loading in bolted joints

Sketch 1.2 shows typical loads applied to bolts by direct shear loading or by bending of the clamped members (Derivations 21 and 12). Note that shear loads can also be caused by a thermally induced shear if dissimilar metals are used in bolted joints or when there are temperature variations in the assembly.

Section 3 examines the significance of the duration and amplitude of applied shear loads in bolted joints on the loss of clamping force and bolt self-loosening. Section 4 discusses the improvements that can be made to a bolted joint design to minimise bolt self loosening.

ESDU 14001^{*} considers ways of applying, measuring and maintaining the pre-load in bolted joints. If a bolted joint is subject to dynamic shear loading, then as well as a good joint design, a secondary locking device (such as a lock nut, a thread adhesive or a prevailing torque feature) may also be required to maintain the pre-load and prevent the assembly from falling apart because of bolt self loosening, see ESDU 14002[†].

A common type of lock nut is one that uses prevailing torque (a frictional torque needed to rotate the nut down the thread of an untightened bolt) to prevent self loosening (Derivation 31). Prevailing torque nuts are available in two basic varieties: one type has a nylon-insert nut and the other type comprises of an all metal nut, see ESDU 14002. In Section 6, an example of the amount of self loosening that is expected in bolts with a standard nut is compared to assemblies using prevailing torque nuts with a nylon collar insert.

The Derivation and References used in this Technical Memorandum are listed in Section 5.

^{*} ESDU 14001, "Applying, measuring and maintaining pretension in steel bolts".

[†] ESDU 14002, "Lock nuts and other thread locking devices".

1.1 Notation

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

A	deflection	m	in
a_{mod}	modified eccentricity or deflection	m	in
C_{ϕ}	torsional stiffness	Nm/rad	lbf in/rad
D	nominal diameter of bolt	m	in
d_2	basic pitch diameter of bolt thread	m	in
E	Young's modulus	N/mm ²	lbf/in ²
e	eccentricity	m	in
F	bolt axial load	N	lbf
F_s	shear load	N	lbf
F_z	loss of pre-load as a result of embedding	N	lbf
f	stress	N/mm ²	lbf/in ²
k_b	bolt stiffness	N/m	lbf/in
k_m	member stiffness	N/m	lbf/in
L	clamped length	m	in
M_A	tightening moment	N m	lbf in
$M_{H,F}$	friction under bolt head	N m	lbf in
M_L	unfastening moment	N m	lbf in
$M_{T,F}$	thread friction	N m	lbf in
$M_{H,P}$	moment from thread pitch	N m	lbf in
P_0	pre-load	N	lbf
p	thread pitch	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

T	tightening torque applied to nut or bolt head	N m	lbf in
T_L	torque required to loosen nut	N m	lbf in
T_{loosen}	total torque in loosening direction	N m	lbf in
T_{Ps}	prevailing torque under transverse slip	N m	lbf in
T_R	torque resisting loosening	N m	lbf in
T_{ss}	loosening torque due to differences in resistance between ascending and descending sides of nut threads during sliding on bolt threads	N m	lbf in
δ_z	plastic deformation as a result of embedding	m	in
β	half thread flank angle	degree	degree
β_1	lead angle	degree	degree
$\Delta\delta/2$	relative displacement between two clamped member plates	m	in
ϕ	loading direction (angle between direction of applied load and plane of clamped plates)	degree	degree
μ_a	friction coefficient between annular bearing faces		
μ_{as}	friction coefficient between annular bearing faces under transverse slip		
μ_t	friction coefficient between mating thread surfaces		
μ_{ts}	friction coefficient between mating thread surfaces under transverse slip		
θ	nut rotation	degree	degree

2. BOLT SELF-LOOSENING

The phenomenon of bolt self-loosening that takes place during the repeated application of shear loads on bolted joints can be studied experimentally or using a finite element analysis (FEA). It is important to understand which mechanisms are involved during bolt self loosening because after the joint is loose, the bolt will carry more of the load and this often leads to fatigue failure of the bolt.

2.1 Mechanism for Bolt Self-loosening

During tightening of a bolt, some of the strain energy created is stored as tension and torsion in the joint and some may be stored as bending due to misalignments of the clamped surfaces. The stored energy is retained by friction constraints at the bolt threads or at the contact surfaces between the nut, the bolt or the

other members in the assembly. When friction is overcome, the inclined plane of the bolt threads pushes the nut threads away leading to self loosening and the bolt returns to its original length before tightening, and the stored energy is released (Derivation 11).

During assembly, the tightening moment, M_A , equates to the sum of three separate moments: $M_{T,P}$, the moment from the thread pitch (which generates the pre load in the bolt), $M_{H,F}$, the friction under the bolt head or the nut, and $M_{T,F}$ the friction in the bolt threads (Derivation 10).

$$M_A = M_{T,P} + M_{H,F} + M_{T,F}. \quad (2.1)$$

The moment needed to unfasten the bolt, M_L , is:

$$M_L = -M_{T,P} + M_{H,F} + M_{T,F}. \quad (2.2)$$

In Equation (2.2), the moment from the thread pitch, $M_{T,P}$, is negative. When $M_{T,P}$ becomes larger than the sum of the moments arising from friction, self loosening begins. Theoretically, without any friction it is not possible to create pre-load in the bolt. Normally after pre-loading, $M_{T,P}$ is less than the sum of the moments arising from friction therefore the bolt retains its clamping force.

2.2 Torque/tension Relationship for Prevailing Torque Nuts

The general algebraic expression for tightening prevailing torque nuts, taking into account the main influencing factors in the torque/tension relationship is (based on Equation (3.2) in ESDU 14001):

$$T = F \left[\frac{p}{2\pi} + \left(\frac{r_i + r_o}{2} \right) \mu_a + \frac{d_2 M_t}{2 \cos \beta} \right] + T_{Ps}. \quad (2.3)$$

The product of F and the first term in the square bracket of Equation (2.3) is the torque required to stretch the bolt ($M_{T,P}$), the second term is the torque needed to overcome friction under the nut face or the bolt head ($M_{H,F}$), and the third term is the torque needed to overcome friction in the threads ($M_{T,F}$). T_{Ps} is the additional prevailing torque required to rotate the nut down the thread of the untightened bolt (T_{Ps} also resists the movement of the nut in any rotational direction).

In general when using prevailing torque nuts, and if only a very approximate assessment of the average value for unlubricated threads is required, the following simple empirical relationship (avoiding quantifying μ_a and μ_t) can be used (based on Equation (3.3) in ESDU 14001):

$$T = F d_2 / 4 + T_{Ps}. \quad (2.4)$$

2.2.1 Calculating torque ratios

The relative magnitude of the three torque values in the square bracket of Equation (2.3) can be illustrated in an example using a M12 $\times$ 1.75 – 6g/6H coarse thread bolt with a basic pitch diameter of $d_2 = 10.863$ mm, an outer radius of annular bearing face of bolt head (across flats) of $r_o = 9.5$ mm, an inner radius of annular bearing face of bolt head (bolt radius) of $r_i = 6$ mm, and $\mu_a = \mu_t = 0.15$:

$$\frac{p}{2\pi} = \frac{1.75}{2\pi} = 0.28, \quad \left(\frac{r_i + r_o}{2} \right) \mu_a = \left(\frac{6 + 9.5}{2} \right) \times 0.15 = 1.16, \quad \frac{d_2 \mu_t}{2 \cos \beta} = \frac{10.863 \times 0.15}{2 \times \cos 30} = 0.94.$$

These values show a typical ratio of around 10 per cent of the torque is used to stretch the bolt and compress the clamped members, 50 per cent of the torque is needed to overcome the friction at the nut face or under the bolt head and the remaining 40 per cent of the torque overcomes friction in the bolt threads.

2.3 Self-loosening of Prevailing Torque Nuts

In threaded fasteners, the loosening direction is evidently opposite to the tightening direction. The torque required to loosen a prevailing torque nut, T_L , is:

$$T_L = F \left[-\frac{p}{2\pi} + \left(\frac{r_i + r_o}{2} \right) \mu_a + \frac{d_2 \mu_t}{2 \cos \beta} \right] + T_{Ps}. \quad (2.5)$$

Equation (2.5) applies to static situations and due to the negative term, the loosening torque is typically 15 to 20 per cent less than the tightening torque.

The torque in the loosening direction when transverse slip in the threads takes place, T_{ss} , can be calculated using the method discussed in Appendix A:

$$T_{ss} = -\frac{Fp}{4\pi} \quad (2.6)$$

When transverse slip does take place, the total torque in the loosening direction, T_{loosen} , becomes the sum of the torque in the loosening direction (Equation (2.6)) added to the torque needed to stretch the bolt (the first term in the square brackets of Equation (2.5)):

$$T_{loosen} = -\left(\frac{Fp}{4\pi} + \frac{Fp}{2\pi} \right) = -\frac{3Fp}{4\pi}. \quad (2.7)$$

Once friction is overcome in one shear direction, the resistance to any movement in the other directions is reduced dramatically. To differentiate between the friction coefficients under static conditions and those that happen while transverse slip is taking place, the terms μ_a and μ_{ts} denote the friction coefficients at the nut face and thread in the rotational direction during transverse slip, respectively. During the course of transverse slip, the torque resisting loosening, T_R , is:

$$T_R = F \left[\left(\frac{r_i + r_o}{2} \right) \mu_{as} + \frac{d_2 \mu_{ts}}{2 \cos \beta} \right] + T_{Ps}. \quad (2.8)$$

To create rotation in the loosening direction, the torque in the loosening direction, T_{loosen} , must be greater than the torque resisting it, T_R . Since the values of μ_{as} and μ_{ts} are close to zero (Derivation 31), the approximate condition for self loosening of a prevailing torque nut can be written as:

$$T_{loosen} > T_{Ps}. \quad (2.9)$$

2.4 Relative Motion

There are many reasons (see Derivation 35) why relative motion can happen in the bolt threads, such as the following.

- Component bending in the bolted assembly resulting in forces at the friction surfaces.
- Differential thermal effects due to temperature differences or when dissimilar metals are used in the assembly.
- Dynamic loads applied to the bolted joint causing a shearing movement of the contact surfaces.

Sometimes only a small relative motion of the threads is enough to overcome friction and create a loosening torque that is proportional to the thread pitch and the pre-load. Note that when there is no movement in the threads, regardless of any shear loads applied to the assembly, there will be no bolt self loosening.

2.5 Slackening

When loads are applied to bolted joints, the stresses that are produced may exceed the compressive strength of the materials and cause embedding (a plastic flattening of the surface roughness at the contact surfaces, loaded bolt threads or any other interfaces in the assembly). This takes the form of either *settling*; a pressure induced smoothing of surface roughness, or *creep*; a time dependent yielding. Embedding reduces the clamped length of the bolt which in turn lowers the clamping force. This is often referred to as bolt slackening (Derivation 36). Any more embedding taking place during assembly does not contribute to further losses of clamping force once the bolted joint has been pre loaded.

The relationship between the loss of pre load, F_z , and the amount of plastic deformation, δ_z , is:

$$F_z = \frac{\delta_z}{\left(\frac{1}{k_m} + \frac{1}{k_b}\right)}, \quad (2.10)$$

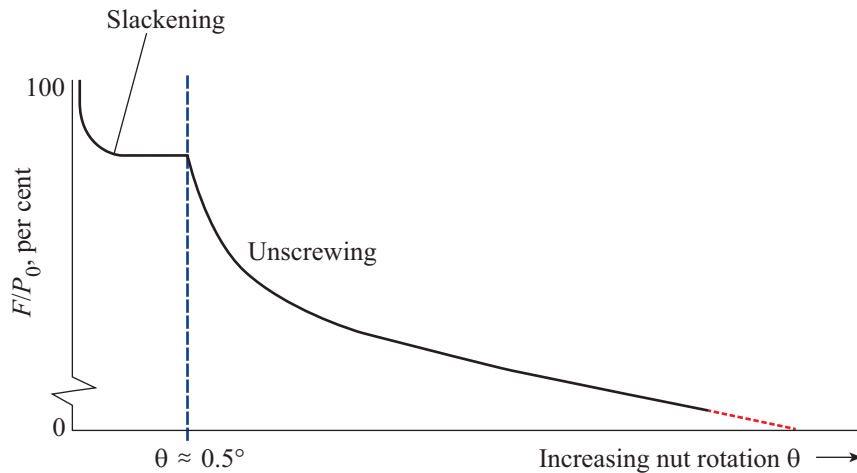
where k_m is the member stiffness and k_b is the bolt stiffness, see ESDU 13015*.

The amount of embedding depends on the applied load, the surface roughness and the number of contact surfaces in the assembly, also see Section 2.6. Embedding is naturally more for softer materials, such as aluminium alloys, compared to harder materials like steel (Derivation 41). For steel bolted assemblies only, more clamping force is lost due to embedding if the working temperature is greater than 50 per cent of the recrystallization temperature of the steel (Derivation 41).

Bolt slackening can be minimised by optimising the joint design, for example by increasing the length to diameter ratio of the bolt or reducing the bolt stiffness. Joints with bolts that have a large head diameter have less contact pressure so the tendency for settling and creep at the contact surfaces is smaller. Also, using a compressible spring element such as a stiff conical spring washer or a disc spring could decrease the loss of clamping force due to embedding. Note that although hard spring washers reduce the amount of embedding that takes place after the assembly has been pre loaded, conical spring washers or disc springs and other similar devices should only be used with care if the bolted assembly is likely to be subject to vibration, see ESDU 14002. During bolt slackening, it is assumed that there is insignificant relative motion at the nut and bolt threads meaning that it is difficult to designate the point when unscrewing begins.

* ESDU 13015, "An introduction to some methods to calculate loads and stiffnesses in bolted assemblies".

Sketch 2.1 shows the change in the ratio of bolt load to pre load due to nut rotation. The point at which unscrewing (see Section 2.8) begins is not clear however the onset of unscrewing is often taken to occur at a nut rotation of about 0.5° and unscrewing continues until the clamping force is lost (Derivation 22).



Sketch 2.1 Two stages of self-loosening

2.6 Influence of Coatings on Loss of Pre-load

The loss of pre-load during shear loading of coated bolts has been investigated using a series of experiments carried out on M20 10.9 bolts with hard or soft coatings (see Table 7.1 for coating details) that have been assembled using 160 kN pre load* (Derivation 32). On Figure 1, all of the bolts show an initial drop of between 7 and 81 per cent of pre load, probably due to embedding (see Section 2.5). Usually the pre load reaches a constant level after about a day when embedding is expected to have finished. Derivation 32 reported that the softer coatings were 'squeezed' from between the contact surfaces in the joint after pre loading. Thinner, harder coatings – or rather no coatings at all – are preferable as they show the smallest loss of pre load.

2.7 Lubricants and Coefficients of Friction

Many types of thread lubricants are available such as: oil, grease or wax, graphite and molybdenum disulphide, see Table 7.2. Some thread locking compounds, particularly the types that allow the removal of bolts, e.g. Loctite 222, 242 and 262, can also be used as lubricants (Derivation 8 and 42). Lubrication under the bolt head or between the clamped members in the joint can decrease the friction coefficient at the contact surfaces and reduces any resistance to self-loosening (Derivation 12).

The contact surfaces in bolted joints may be rough, smooth, oxidised, chemically treated or lubricated. This means that the friction coefficient at the contact surfaces varies greatly (Derivation 6). More friction means a bigger resistance during bolt tightening so that additional torque is needed to attain the required pre-load. About 90 per cent of the applied torque is used to overcome friction leaving the rest to contribute to the pre load (see Section 2.2). Care must be taken not to over-tighten a lubricated bolt.

Table 7.3 shows the range of friction coefficients for bolted assemblies using different metals, surface finishes and types of lubrication. The aim of a good bolted joint design is to achieve a friction coefficient somewhere in Class B of Table 7.3 so as to apply as much pre-load as possible with the least scatter. This does not necessarily mean that the lowest friction coefficient value must be used or that the scatter of the friction coefficients corresponds to the class spread. Table 7.4 shows approximate values of static friction coefficients at the member contact surfaces in typical bolted joints.

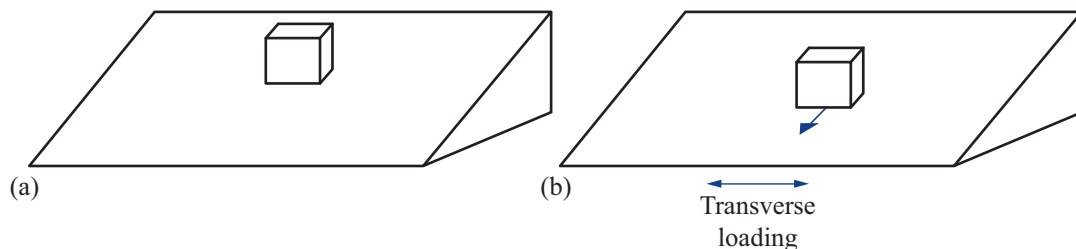
* Note that 160 kN is about 70 per cent of the yield strength.

Other variables that affect the friction coefficient in bolted joints are the metal's hardness, the speed of nut tightening, the thread fit, hole clearance surface pressures and the presence or absence of washers. Bolts with rolled threads can exert greater pre-loads for a given torque with less scatter in the pre load values compared to bolts that have cut threads. Since it is likely that less than average pre loads and large degrees of scatter produce bigger friction coefficients in the threads, the method of thread manufacture is important to the joint design (Derivation 28).

2.8 Unscrewing

When a bolt is subject to dynamic shear loading in the plane perpendicular to the bolt axis, the contact forces in the threads and at the bolt head change from the circumferential direction to the applied shear load direction, and circumferential friction is reduced. As the dynamic shear loads and frictional stresses alternate, the wedge shape parts of the bolt and nut thread begin to slip. If this slip is not recovered when the bolt returns to its normal position then the nut starts to rotate (Derivation 25). Note that in one load cycle the rotation of the nut may be relatively small but it is the gradual rotation of the nut over many load cycles that leads to the loss of clamping force.

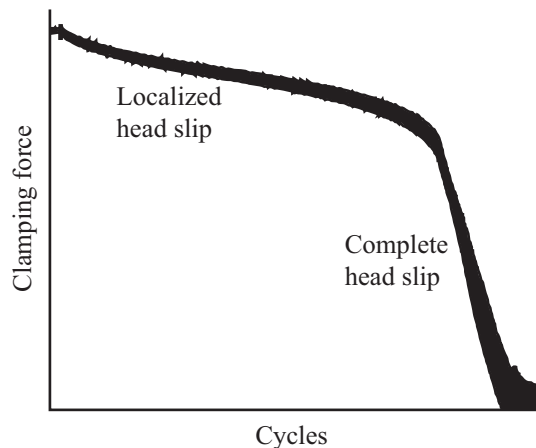
Unscrewing taking place during transverse loading of bolted joints can be illustrated using the example of a block on a wedge. Without any external loads, the friction between the block and the wedge is sufficient to prevent the block from sliding down the incline, Sketch 2.2a. If an applied transverse shear load is large enough to overcome friction between the block and the wedge, the block not only slips in the loading direction but also down the incline, see Sketch 2.2b. In bolted joints, as soon as the applied transverse shear load is enough to overcome friction in the transverse direction, the joint is free of circumferential friction and the slip limit is exceeded. The moment developed due to pre load around the thread helix will then cause the bolt to loosen.



Sketch 2.2 Movement of a block on a wedge during transverse shear loading (Derivation 19)

The process of 'localized slip' develops when the friction in the parts in the bolted assembly that are in contact is overcome. 'Complete slip' starts when the loads in the joint are large enough to overcome friction across the whole contact areas including the bolt head, the threads and the clamped members. During unscrewing, the contact surfaces below the bolt head and at the nut threads undergo both localized and complete slip which accumulates over many loading cycles (Derivation 20).

Sketch 2.3 shows a typical slip response below the bolt head during transverse shear loading of a bolted joint (Derivation 20). Self-loosening (loss of clamping force during loading) increases dramatically when localized slip changes to complete slip with unscrewing of the nut. A finite element analysis (FEA) shows that localized head slip causes bolt self-loosening to take place at significantly lower shear loads than those that are required for complete slip. Derivation 20 reported that loosening by localized slip happens when the shear force is only 9 per cent of the pre-load whilst loosening by complete slip occurs when the shear force is around 16 per cent of the pre-load. Self loosening due to localized slip is a critical process with respect to joint design and failure can be avoided by ensuring that the dynamic loads on the bolt are less than the load required to cause localised slip.



Sketch 2.3 Behaviour of bolt head slip during transverse shear loading (Derivation 20)

The interaction of friction at the contacting surfaces during pre-loading and shear loading may be non linear (Derivation 17). A FEA of sliding taking place between the contacting surfaces in bolted joints during shear loading shows that the surfaces separate and come together many times and the contact surfaces initially slide a small way up the thread incline but, on the whole, there is sliding down the incline. At faster loading frequencies, the separation and sliding of the contact surfaces increases as does the tendency for the surfaces to slide further up the plane resulting in slight tightening. FEA confirms the overall loss of friction that leads to self loosening.

3. PRESENTATION OF BOLT SELF-LOOSENING DATA

3.1 Shear Displacement Failure Curves

Shear displacement failure curves show the number of cycles to failure for given shear displacements applied to bolted joints, see Derivation 26. Table 7.5 tabulates the data from a shear displacement curve showing the self-loosening of M12 bolts with a 25 kN pre-load (about 32 per cent of yield strength). This data is re-plotted on Figure 2 and shows that larger applied shear load displacement amplitudes reduce the number of cycles until self loosening begins. In most cases, if the shear load displacement amplitude is below a threshold value, approximately $\Delta\delta/2 = 0.22$ for Figure 2, the intermediate stages of a 25 per cent or a 75 pre load drop do not take place and only an initial nut rotation is seen followed by no further loss of clamping force.

3.1.1 Effect of clamped length and loading direction on bolt self-loosening

Figures 3 and 4 show how different clamped lengths, L , and applied load directions, ϕ , respectively, effect the shear load displacement limit (the threshold below which self loosening life is essentially infinite) in bolted joints (Derivation 26). The shear load was applied at a frequency of 0.5 Hz on M12 $\times$ 1.75 class 10.9 bolts with a 25 kN pre load (about 32 per cent of yield strength). Note that $\phi = 0^\circ$ for shear loading and $\phi = 90^\circ$ for axial loading.

On Figure 3, longer clamped lengths improve the shear load displacement limit and hence the bolt self loosening resistance. The load carrying capability of the joint is not affected significantly by the clamped length, unless self loosening begins. The ability of longer bolts to maintain clamping force during repetitive shear loading is discussed in Derivation 30. Figure 4 shows the shear load displacement limit is slightly higher when there is a larger angle between the applied external load and the shear load direction.

3.1.2 Effect of amount of pre-load on bolt self-loosening

Figure 5 shows that increasing the pre load from 25 to 52.5 kN (about 32 to 66 per cent of yield strength) raises the shear load displacement limit for M12 $\times$ 1.75 bolts cycled at a frequency of 0.5 Hz (Derivation 22). It is recommended that care should be exercised if large pre-loads are used (for example 52.5 kN on Figure 5) as they can cause the bolt to fail due to fatigue.

In Figure 5, all three of the bolted joints had the same thread and bearing friction coefficients however the assembly using the zinc-plated washer with a regular nut has a slightly longer clamped length which raises the shear load displacement limit by a factor of two compared to using a flange hex nut. When flange hex nuts are used instead of regular nuts, there is a larger contact area between the nut and the bearing surface leading to larger bending moments on the bolt and more embedding at the contact surfaces meaning that the resistance to bolt self loosening is poor (Derivation 22).

3.1.3 Effect of pulse duration, friction coefficient and lead angle on bolt self-loosening

Derivation 11 uses a mathematical model to examine the effect of the duration of the applied shear load (8, 10 and 12 microseconds), coefficient of friction in the threads ($\mu_t = 0.1, 0.15$ and 0.2) and lead angle ($\beta_1 = 5, 7$ and 9°) on reducing the clamping force in bolted joints experiencing intermittent vibrations. The resistance to bolt self-loosening can be improved by the following.

- Applying the load for long(er) durations. Shorter durations of applied external loads cause high frequency excitations, creating large internal forces that accelerate self-loosening.
- Increasing the coefficient of friction to reduce any loss of clamping force. Friction prevents the transverse motion of slippage at the contact surfaces therefore stopping thread loosening. The best way to maintain high friction forces in joint is to use a high pre-load.
- Selecting bolts with fine-pitch threads that have smaller lead angles. A coarse pitch thread produces greater internal loosening torques and will be less vibration resistant. The theoretical loosening torque is directly proportional to the lead angle of the thread.

3.2 Reduction in Clamping Force During Shear Loading

At the start of dynamic shear loading tests, the small reduction in pre load observed in most bolts is probably due to plastic deformation in the bolt and embedding at the contact surfaces, *i.e.* slackening (Section 2.5). On Sketch 3.1, the characteristic loss of clamping force during dynamic shear loading of bolted joints with a regular nut is shown (Derivation 29). The change from a steady, initial loss of clamping force to a secondary, faster loss happens when localized slip gives way to complete slip in the joint. If a lock nut or another thread locking devices is used then it is possible that the initial lower, steady loss of clamping force would continue until the final value of clamping force is reached at the end of the test.

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