

Threaded Fastener Secondary Locking Requirements

D. P. Hess

Submitted: 27 June 2017 / Published online: 19 July 2017
© ASM International 2017

Abstract Secondary locking features in threaded fasteners are in widespread use in machinery, structures, and systems. In this paper, the mechanism for loosening of threaded fasteners is explained qualitatively and defined quantitatively in terms of the self-loosening moment inherent to threaded fasteners and external load-induced loosening moments. Equations for loosening moments are defined. This paper provides analysis which quantifies the locking action required to prevent loosening due to the inherent self-loosening moment and external load loosening moments in threaded fastener joints. This paper provides the basis and method for engineers to properly design or specify secondary locking features in threaded fasteners to provide sufficient locking and prevent loosening. Requirements for secondary locking feature moments are developed in terms of loosening moments and factor of safety. Test data and calculations are provided to substantiate the requirements. Examples are provided for common secondary locking features including prevailing torque and adhesive.

Keywords Fastener · Bolt · Locking · Secondary · Loosening · Requirements · Prevailing · Adhesive

Introduction and Background

Assembly of a bolted joint generally involves application of a tightening torque to threaded fasteners with resulting

relative angular motion and stretch defined by the thread helix. After assembly, this stretch and associated potential energy is often held by the inherent friction at the thread and bearing interfaces. This friction is the primary locking mechanism or feature in preloaded threaded fasteners. This friction is proportional to preload.

In many cases, a secondary locking feature is added to a threaded fastener [1]. These include mechanical locking (e.g., cotter pin or safety wire), prevailing torque locking (e.g., deformed thread or polymer patch), adhesive locking (e.g., anaerobic adhesive), and free spinning locking (e.g., serrated bearing surface or lock washer).

Use of a secondary locking feature may be required as in some aerospace applications [2] to provide redundancy and to counter possible loosening and failure in the event of unexpected loads. Often it is added to correct an observed or documented fastener loosening problem. In cases of fasteners without preload, such as found in some electrical applications, it provides the only form of locking in fasteners.

Despite the widespread availability and use of secondary locking features, very little quantitative design guidelines exist. The motivation for this paper is to quantify the locking requirement for secondary locking features in threaded fasteners. And to specifically answer the question “What moment does a secondary locking feature need to provide to prevent loosening?”

A literature search reveals very little published data on this topic. As a result, the analyses and data in this paper help fill an existing void in the literature.

Mechanism of Vibration-Induced Loosening

As a bolted joint is tightened to a desired preload or clamping force, the bolt stretches, the joining components

D. P. Hess (✉)
Department of Mechanical Engineering, University of South
Florida, 4202 E. Fowler Avenue, ENB 118, Tampa, FL 33620,
USA
e-mail: hess@usf.edu

compress, and the potential energy in the system increases. If the friction at the thread interface, and nut and bolt head interfaces can hold this energy, the preload in the joint is maintained. However, external loads, both static and dynamic, can alter the forces and moments in the system such that the holding friction forces are overcome and loosening occurs.

The simple analogy of a block on an incline plane is often used to illustrate these concepts. Figure 1a shows a block loaded by gravity on an incline. Figure 1b shows the forces acting on the block: block weight W , normal contact force N , and friction force f . The weight W can be resolved into two components: $W \cos \beta$ perpendicular to the contact plane, and $W \sin \beta$ parallel to it. The force $W \sin \beta$ acts to move the block down the plane, but is resisted by the friction force f . The friction force has an upper limit which is equal to the normal force $N = W \cos \beta$ multiplied by the coefficient of friction μ . As a result, the block will not slide down the plane when the following condition holds true

$$\mu W \cos \beta > W \sin \beta. \tag{Eq 1}$$

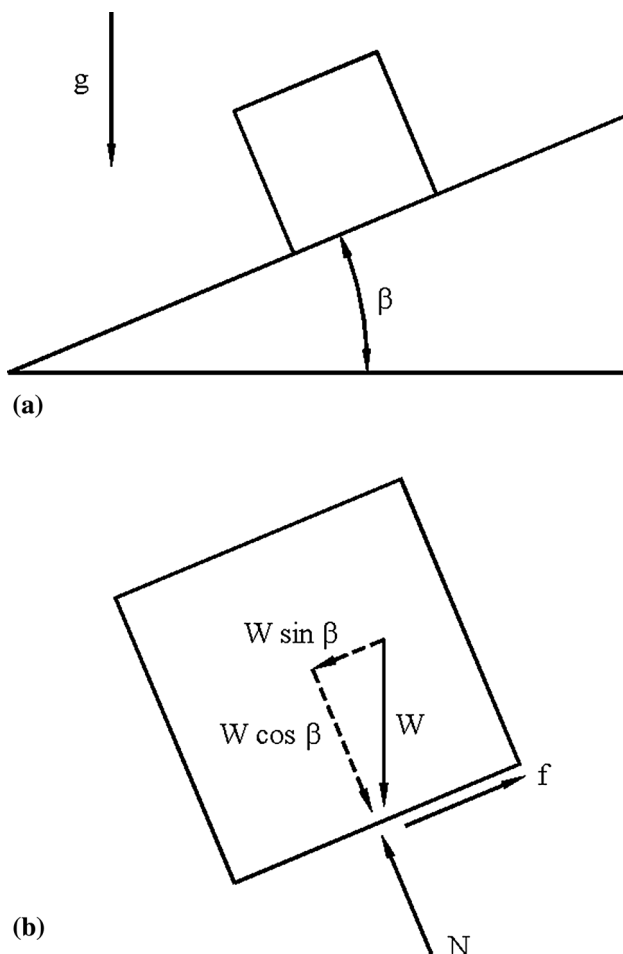


Fig. 1 Simple analogy for bolted joint: (a) block on incline loaded by gravity, (b) forces acting on block

To illustrate the effect of an external static force on the system, consider a lateral force F being applied to the block as shown in Fig. 2a. In general, the friction force will resist all forces acting on the block in the plane of contact. In this case, the friction force resists both force $W \sin \beta$ and lateral force F . The static force balance in the plane of contact is shown in Fig. 2b. Note that the direction of the friction force opposes the direction of motion which is now a combined down and lateral direction. The condition for no slip in this case becomes

$$\mu W \cos \beta > [F^2 + (W \sin \beta)^2]^{\frac{1}{2}}. \tag{Eq 2}$$

As the lateral force F is increased, this condition will eventually not be met, and slip will occur.

This analysis can be extended to explain self-loosening of bolted joints [3]. Since standard bolts and nuts are made with lateral clearance for assembly, the nut can be moved sideways by an amount equal to this lateral clearance. Consider the cross section of a bolted joint as shown in Fig. 3 with a force applied to the front of the nut. Since the right side will move with greater difficulty, it acts as a pivot point about which the nut slips and turns loose. When a force is applied to the back of the nut as shown in Fig. 4, the left side becomes the pivot point about which the nut slips and rotates loose. The net effect of a dynamic or cyclic transverse force is a ratcheting loosening motion, and the amount of slip and resulting loosening increases with lateral thread clearance.

Existing studies [1] reveal the primary mechanism of self-loosening is relative thread slip and component slip. Relative thread/component slip is caused by static and dynamic forces, moments, and reduced friction. These manifest themselves in joints through bending, pressure fluctuations, shocks, impacts, thermal expansion, and axial force fluctuations.

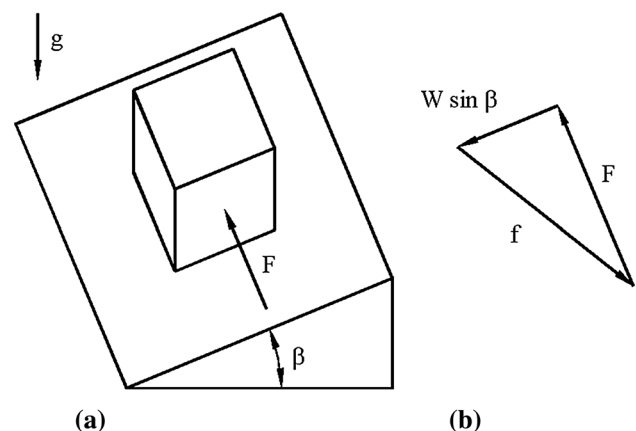


Fig. 2 (a) Block on incline with external lateral force, (b) force balance in plane of contact

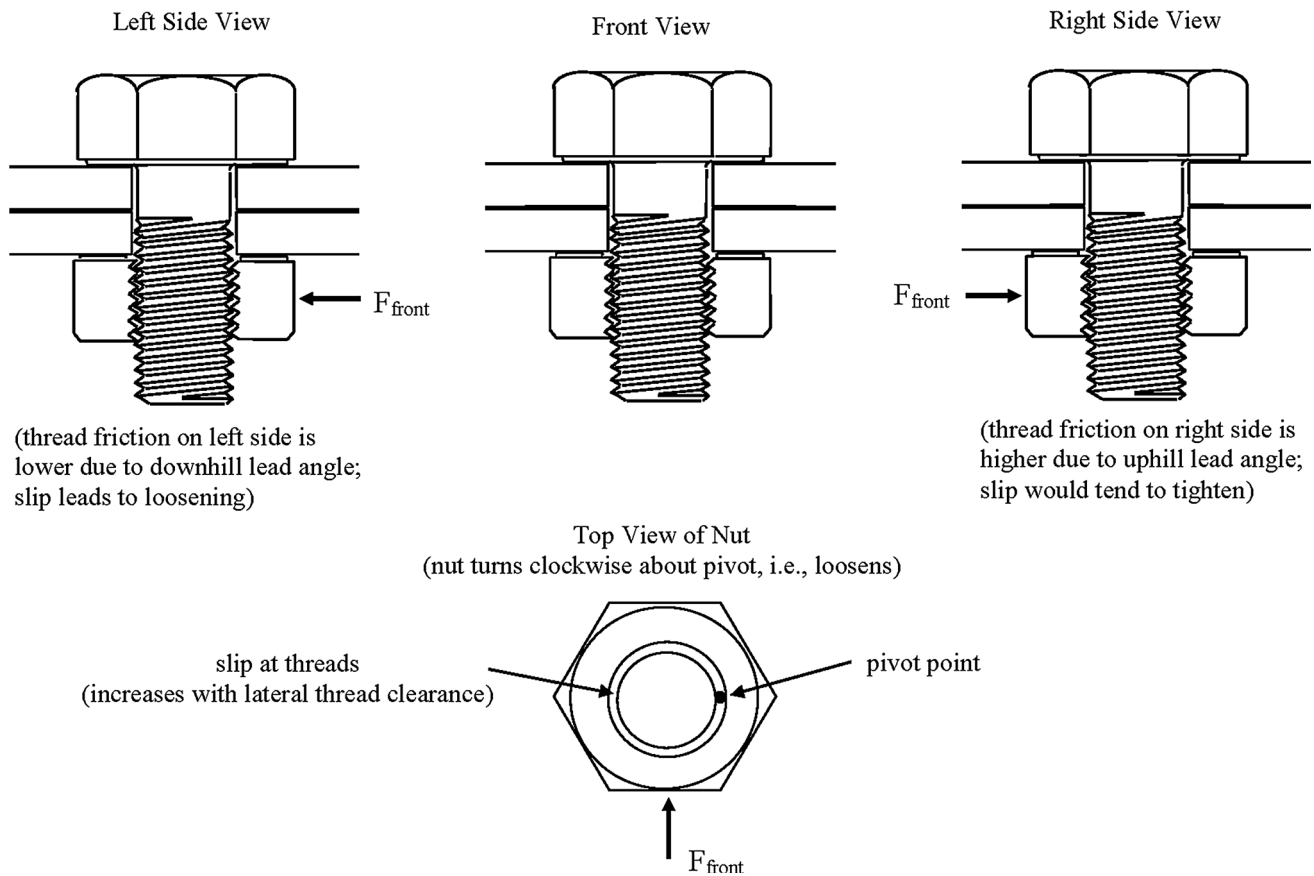


Fig. 3 Bolted joint with force applied to nut from front

The primary dynamic failure modes of bolted joints are (1) complete loosening, and (2) partial loosening followed by eventual bolt fatigue. Either can occur when predominately dynamic shear forces act on the bolted joint sufficient to cause slip. Partial loosening can occur from dynamic axial forces sufficient to cause nut dilation with slip.

Experience and review of the past several decades of research in this area [1] reveal that for standard threaded fasteners without secondary locking features, resistance to loosening improves with: increased preload, finer thread pitch, higher thread and head friction, tighter tolerances, higher excitation frequency, and lower excitation amplitude.

Self-Loosening Moment

Threaded fasteners exhibit an inherent self-loosening moment. The basis for the self-loosening moment is the torque equation for a bolted joint. The tightening torque for a threaded fastener in a bolted joint is generally [4–6] defined as

$$T_t = F_p \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right). \quad (\text{Eq 3})$$

Here T_t is the tightening torque, F_p is the preload, p is the thread pitch, μ_t is the thread interface friction coefficient, r_t is the nominal thread interface radius, μ_n is the nut interface friction coefficient, r_n is the nominal nut interface radius, and β is the thread half angle. The first term in this equation is the torque required to stretch the bolt, and the remaining two terms are the torque required to overcome thread and nut friction.

The removal torque is

$$T_r = F_p \left(-\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right). \quad (\text{Eq 4})$$

This removal torque is the torque required to overcome thread and nut friction minus the torque from bolt stretch.

In the absence of thread and nut friction, the removal torque equation defines the self-loosening moment.

$$M_{\text{self-l}} = \frac{F_p p}{2\pi} \quad (\text{Eq 5})$$

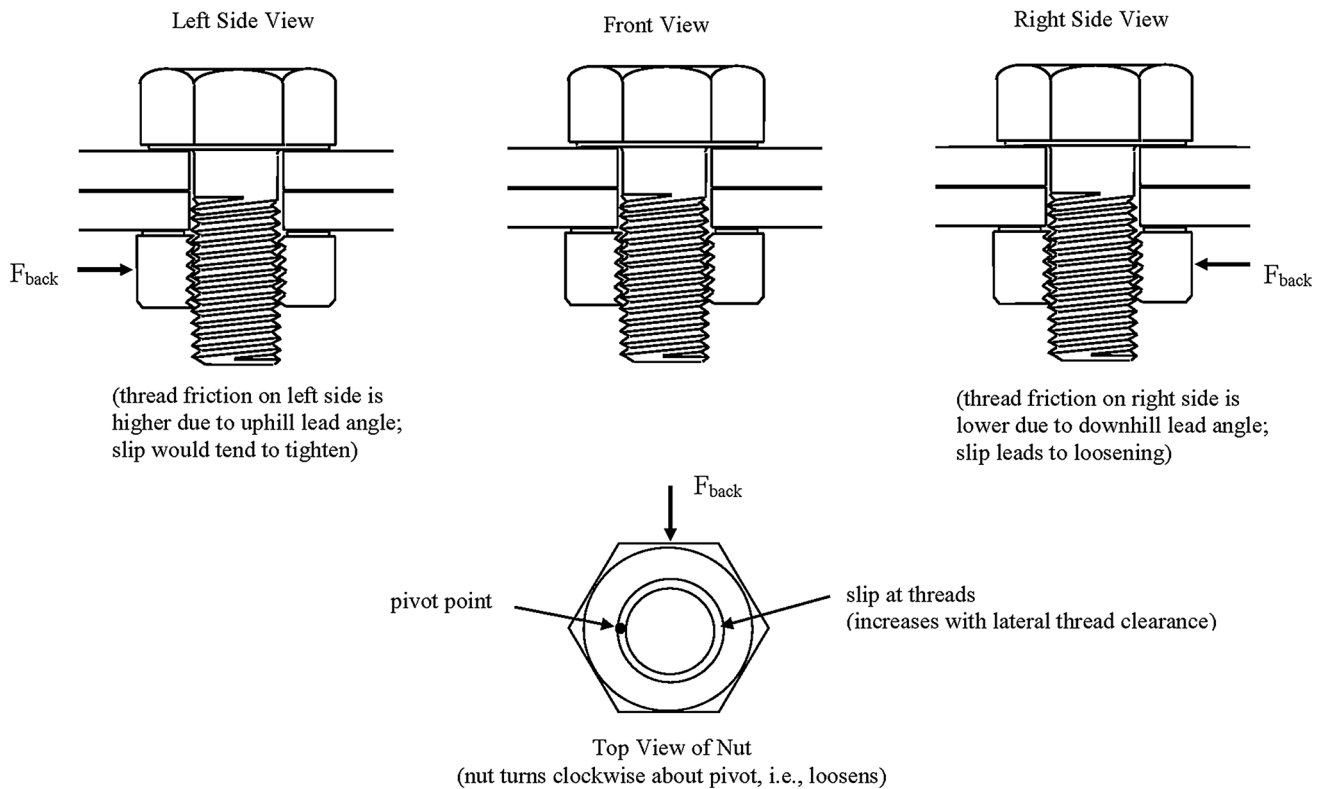


Fig. 4 Bolted joint with force applied to nut from back

This results from the bolt stretch torque and associated potential energy in the bolt. It is inherent to the threaded fastener and is proportional to preload and thread pitch.

Considering the limiting case where thread and nut friction are eliminated completely, the moment from bolt stretch emerges as a quantitative value requirement for a secondary locking feature in the absence of external loads. The secondary locking feature locking moment requirement becomes

$$M_{\text{locking}} \geq M_{\text{self-1}} \tag{Eq 6}$$

A factor of safety is defined in terms of the locking moment and self-loosening moment as

$$\text{FoS} = \frac{M_{\text{locking}}}{M_{\text{self-1}}} \tag{Eq 7}$$

As an example, consider a 0.25–28 UNJF thread fastener in a joint with a 2400 lb preload. The thread pitch is 1/28 in. The self-loosening moment is

$$M_{\text{self-1}} = \frac{F_p p}{2\pi} = \frac{2400}{2\pi(28)} = 13.6 \text{ in-lb} \tag{Eq 8}$$

If a prevailing torque locking feature is used with a prevailing torque of 20 in-lb, then the locking moment is 20 in-lb and the factor of safety against loosening is

$$\text{FoS} = \frac{M_{\text{locking}}}{M_{\text{self-1}}} = \frac{20}{13.6} = 1.47 \tag{Eq 9}$$

This example assumes no thread or nut friction which would add to the locking moment and increase the factor of safety. The example also assumes no external loads which would add to the loosening moment and decrease the factor of safety.

Loosening Moment from External Loads

In addition to the inherent self-loosening moment in fasteners due to bolt stretch, external loads introduce loosening moments. External transverse loads on a bolted joint are illustrated in Figs. 3 and 4. Taking the pivot points on the thread pitch diameter and the external transverse load applied at the center of the fastener, the moment arm is the thread pitch radius r_t and the loosening moment from front and back external transverse loads is

$$M_{\text{ext-1}} = F_{\text{front}} r_t = F_{\text{back}} r_t \tag{Eq 10}$$

Example data [7] exist from dynamic transverse load tests with NAS1004 0.25–28 UNJF threaded fasteners. Minimum and maximum pitch diameter is 0.2243 and 0.2268 in for an average pitch radius of 0.1128 in. The

dynamic transverse force is 200 lb, and the preload is 2400 lb. Therefore, the loosening moment from this external transverse load is

$$M_{\text{ext-1}} = F_{\text{front}}r_t = F_{\text{back}}r_t = (200)(0.1128) = 22.6 \text{ in-lb} \quad (\text{Eq 11})$$

This 200 lb external transverse force is the amount that acts on the fastener. The dynamic transverse test machine [7] used is designed with bearings between the clamped components of the joint. This intentionally puts the applied external transverse force on the fastener. In practice with a typical joint, additional friction between the clamped components of the joint would counter the external transverse load. This joint friction force would need to be subtracted from the external load to determine the transverse force on the fastener that is used in the previous equations. An estimate of this joint friction force is the product of clamping force and coefficient of friction at the joint interface.

Similar loosening moments can result from other external loads such as axial loads, bending loads, and combined loads. Analyses of loosening moments from external axial, bending and combined loads are not presented in this paper.

Primary Locking

The friction terms in the tightening and removal torque Eqs 3 and 4 define the primary locking moment in a bolted joint

$$M_{\text{primary locking}} = F_p \left(\frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) \quad (\text{Eq 12})$$

This primary locking moment is dependent on preload and friction. Unfortunately, if sustained cyclic slip occurs in a bolted joint, this primary locking moment is ineffective and locking must be provided by a secondary locking feature.

Secondary Locking Requirements

In a joint subjected to only an external transverse load, the total loosening moment is

$$M_1 = M_{\text{self-1}} + M_{\text{ext-1}} = \frac{F_p p}{2\pi} + F_{\text{ext-trans}} r_t \quad (\text{Eq 13})$$

In the limiting case where thread and nut friction are eliminated, this total loosening moment from combined self-loosening moment and external load loosening moment defines the requirement for a secondary locking feature. The secondary locking feature locking moment requirement becomes

$$M_{\text{locking}} \geq M_1 = M_{\text{self-1}} + M_{\text{ext-1}} \quad (\text{Eq 14})$$

A factor of safety is defined in terms of the locking moment and total loosening moment as

$$\text{FoS} = \frac{M_{\text{locking}}}{M_1} = \frac{M_{\text{locking}}}{M_{\text{self-1}} + M_{\text{ext-1}}} \quad (\text{Eq 15})$$

For the example of a 0.25–28 UNJF threaded fastener with 2400 lb preload and subjected to dynamic external transverse load of 200 lb, the total loosening moment is

$$M_1 = M_{\text{self-1}} + M_{\text{ext-1}} = 13.6 + 22.6 = 36.2 \text{ in-lb} \quad (\text{Eq 16})$$

If a prevailing torque locking feature is used with a prevailing torque of 20 in-lb, then the locking moment is 20 in-lb and the factor of safety against loosening is

$$\text{FoS} = \frac{M_{\text{locking}}}{M_1} = \frac{20}{36.2} = 0.55. \quad (\text{Eq 17})$$

This example assumes no thread or nut friction which would add to the locking moment and increase the factor of safety.

Prevailing Torque Locking

The tightening torque equation with secondary locking prevailing torque feature added is

$$T_t = F_p \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) + T_{\text{pv}}. \quad (\text{Eq 18})$$

Here T_{pv} is the prevailing torque. It is independent of preload F_p .

The removal torque equation with secondary locking prevailing torque feature added is

$$T_r = F_p \left(-\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) + T_{\text{pv}}. \quad (\text{Eq 19})$$

In the absence of preload, a locking moment equal to the prevailing torque T_{pv} remains. Prevailing torque locking features provide locking even with the loss of all preload. As a result, prevailing torque locking is often used in applications where complete disassembly and loss of components must not occur.

Well-defined standards exist for fasteners with prevailing torque locking [8–13]. These provide a range of allowable prevailing torque for a given thread size. For example, for a 0.25–28 UNJF thread fastener, the standards list a 3.5–30 in-lb prevailing torque requirement for this thread. Since the self-loosening moment for the thread is 13.6 in-lb, meeting the minimum 3.5 in-lb requirement is not enough to counter the self-loosening moment.

Anaerobic Adhesive Locking

Tightening torque and breakaway torque measurements can be used to quantify the locking moment provided by adhesive locking. Breakaway torque is the initial value of the removal torque.

To illustrate this approach, the results of measurements from using anaerobic adhesive Loctite 242 with 0.25–28 UNJF thread A286 fasteners are presented. A tightening torque of approximately 100 in-lb provides a preload of 2400 lb. Uncured anaerobic adhesive adds no additional moment to the tightening torque equation; therefore,

$$T_t = F_p \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) = 100 \text{ in-lb} \quad (\text{Eq 20})$$

Since the first term is 13.6 in-lb (for $F_p = 2400$ lb and $p = 1/28$), the combined friction torque, or primary locking moment, is approximately 86.4 in-lb.

After allowing 24 h for the adhesive to cure, the measured breakaway torque from several test samples ranged between 100 and 120 in-lb. Cured adhesive adds an adhesive locking torque to the removal torque equation. Breakaway torque, the initial removal torque, is therefore defined as

$$T_{br} = F_p \left(-\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) + T_{ad}. \quad (\text{Eq 21})$$

The adhesive locking torque T_{ad} at breakaway is computed to be between 27 and 47 in-lb. This locking exceeds the self-loosening torque of 13.6 in-lb for a 0.25–28 UNJF thread fastener.

Test Results

A set of test data [7] exists for 0.25–28 UNJF fasteners with no secondary locking, prevailing torque locking, and adhesive locking. These tests were performed with a dynamic transverse loading apparatus setup to cause cyclic slip and comparative loosening for the test fasteners.

The test fasteners with prevailing torque locking had a measured prevailing torque of 15–20 in-lb. The test fasteners with adhesive locking had an adhesive locking torque of 27–47 in-lb. Therefore, in the absence of primary locking, the locking moment for fasteners with no secondary locking is zero, for fasteners with prevailing torque locking is 15–20 in-lb, and for fasteners with adhesive locking is 27–47 in-lb.

At a preload of 2400 lb, the test fasteners were subjected to a dynamic external transverse load of 200 lb, and a total loosening moment of

$$M_l = M_{self-l} + M_{ext-l} = 13.6 + 22.6 = 36.2 \text{ in-lb} \quad (\text{Eq 22})$$

This total loosening moment of 36.2 in-lb exceeds the locking moment of 15–20 in-lb for fasteners with prevailing torque locking and is within the locking moment range of 27–47 in-lb for the adhesive locking. Therefore, some loosening is expected in both cases. However, since the adhesive locking has a higher locking moment, less loosening is expected for fasteners with adhesive locking compared to those with prevailing torque locking.

Recall that the self-loosening moment decreases with preload. In these tests, the dynamic external transverse load of 200 lb at 2400 lb preload decreases with preload to 120 lb at 250 lb preload and to 70 lb at 50 lb preload. Therefore, the loosening moment from the external transverse load also decreases with preload.

Junker test result data [7] from the three fastener configurations are shown in Figs. 5, 6 and 7. Figure 5 shows complete loosening for the fasteners with no secondary locking feature since the preload reduces to zero between 500 and 1500 cycles of dynamic transverse load.

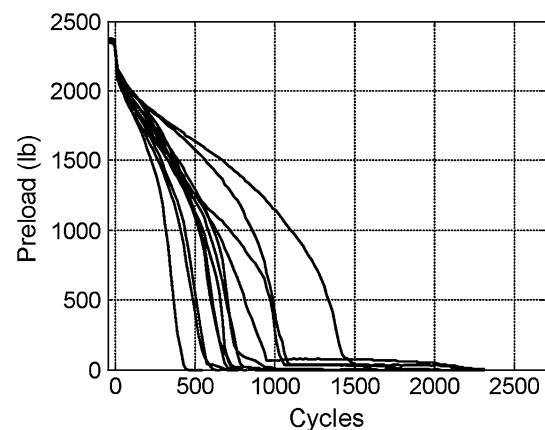


Fig. 5 Preload with transverse load cycles for fasteners with no secondary locking

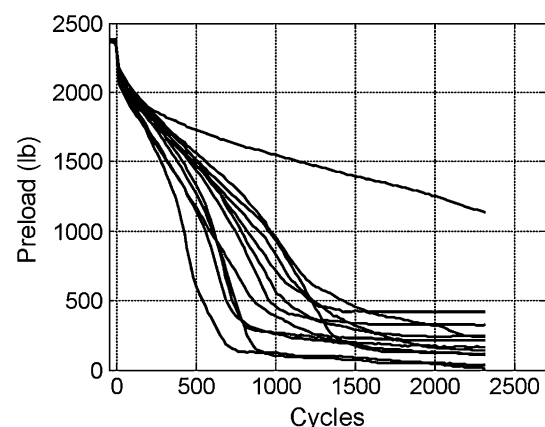


Fig. 6 Preload with transverse load cycles for fasteners with prevailing torque secondary locking

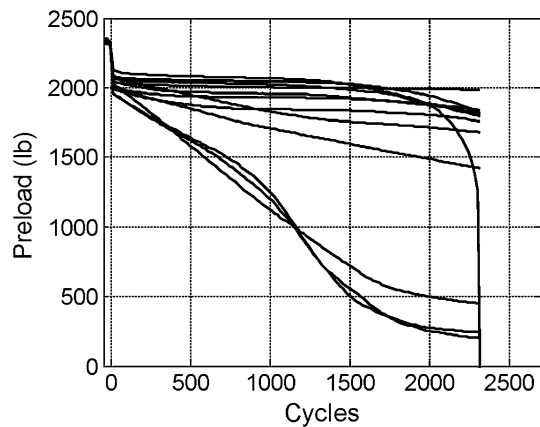


Fig. 7 Preload with transverse load cycles for fasteners with adhesive secondary locking

Figure 6 shows significant loosening for the fasteners with prevailing torque locking feature. There is 50–450 lb steady-state preload remaining after 2300 cycles of dynamic transverse load. This reveals the loosening stops once the loosening moment (which decreases with preload) balances with the 15–20 in-lb of prevailing torque locking plus any friction locking.

Figure 7 shows some loosening for the fasteners with adhesive locking. As expected, since the adhesive locking has a higher locking moment, less loosening occurs for fasteners with adhesive locking compared to those with prevailing torque locking over the same number of loading cycles. Excluding the one test specimen which failed in fracture, there is 250–2000 lb steady-state preload remaining after 2300 cycles of dynamic transverse load.

In these tests, a tightening torque of approximately 100 in-lb (excluding any prevailing torque) provides the 2400 lb preload. Therefore, the combined friction torque or primary locking moment is approximately 86.4 in-lb. Even though this exceeds the total loosening moment of 36.2 in-lb by more than a factor of two, the test results reveal this primary locking moment cannot be counted on in the presence of cyclic slip.

Figure 5 shows complete loosening for fasteners with no secondary locking feature. Figures 6 and 7 show less loosening for fasteners with secondary locking of 15–20 in-lb for prevailing torque locking and 27–47 in-lb for adhesive locking, respectively. This data reveals that comparing the secondary locking moment to the total loosening moment provides a quantitative prediction of locking. Specifically, the secondary locking moment should be greater than or equal to the total loosening moment. This data supports the proposed secondary locking requirement stated in Eq 14.

Other Secondary Locking

Similar calculations for locking moment can be performed for other secondary locking features. For example, locking

moments provided by mechanical locking features such as lock wire and cotter pins are computed from basic mechanics. Free spinning locking devices such as serrated washers and split washers are not recommended for secondary locking since their locking is dependent on preload.

Conclusions

This paper provides the basis and method for engineers to properly design or specify secondary locking features in threaded fasteners to provide sufficient locking and prevent loosening.

The mechanism for loosening of threaded fasteners is explained qualitatively and defined quantitatively in terms of the self-loosening moment inherent to threaded fasteners and an external load loosening moment.

Equations for loosening moments are defined. Requirements for secondary locking feature moments are defined in terms of loosening moments and factor of safety.

Test data and calculations are provided to substantiate the requirements. Examples are provided for common secondary locking features including prevailing torque and adhesive.

References

1. D. Hess, Vibration and shock induced loosening, in *Handbook of Bolts and Bolted Joints*, Chap. 40, ed. by J. Bickford, S. Nassar (Marcel Dekker, New York, 1998), pp. 757–824
2. NASA-STD-5020, Requirements for threaded fastening systems in spaceflight hardware (2012)
3. G. Haviland, Designing with threaded fasteners. *Mech. Eng.* **105**, 17–31 (1983)
4. J. Bickford, *An Introduction to the Design and Behavior of Bolted Joints* (CRC Press, Boca Raton, 1995), pp. 140–142
5. VDI 2230, *Systematic Calculation of High Duty Bolted Joints—Joints with One Cylindrical Bolt* (Verein Deutscher Ingenieure, Düsseldorf, 2003)
6. E. Oberg, F. Jones, H. Horton, H. Ryffel, *Machinery's Handbook*, 25th edn. (Industrial Press, New York, 1996), p. 1408
7. C. Cheatham, C. Acosta, D. Hess, Tests and analysis of secondary locking features in threaded inserts. *Eng. Fail. Anal.* **16**, 39–57 (2009)
8. NASM25027, Nut, self-locking, 250°F, 450°F and 800°F, National Aerospace Standard (1999)
9. NASM8846, Inserts, screw-thread, helical coil, National Aerospace Standard (1998)
10. MIL-I-45914A, Insert, screw thread—locked in, key locked, Department of Defense Military Specification (1991)
11. MIL-I-45932/1C, Insert, screw thread—thin wall, locked in, Department of Defense Military Specification (1994)
12. NAS3350, Nuts, Self-locking, 450°F and 800°F, high quality, National Aerospace Standard (2006)
13. MIL-DTL-18240F, Fastener element, self-locking, threaded fastener—250°F maximum, Department of Defense Detail Specification (1997)