Tribological Aspects of the Self-Loosening of Threaded Fasteners

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ABSTRACT

Practically every engineering product with any degree of complexity uses threaded fasteners. Although threaded fasteners are generally considered a mature technology, significant problems exist with their use. This study has investigated a number of issues with the tightening and self-loosening of threaded fasteners.

- It was found that upon repeated tightening of electro-zinc plated fasteners significant wear of the contact surfaces of the bolt/nut thread and nut face occurred. This wear was accompanied by an increase in the friction coefficient causing a reduction in the clamp force provided for an assembly when tightened to a specific torque value.
- The self-loosening characteristics of prevailing torque nuts were also investigated. It was found that there was a significant loss of prevailing torque when a fastener self-loosened when compared to the prevailing torque when being deliberately disassembled. The current standard test for prevailing torque nuts on re-use does not reflect this surprising result and leads to a significant over-estimate of the capability of this class of nut to resist self-loosening. This is a contribution to knowledge on this topic.
- A further major original finding of this study has been that if an axial load is also acting on a joint which is experiencing transverse slip, prevailing torque nuts can continue to self-loosen leading to their possible detachment from bolts. A number of accidents have been caused by such detachments, but the cause was not understood partly because this detachment could not be reproduced on the standard loosening test. Work reported in this thesis has been found that if an external axial load is acting whilst the joint is experiencing transverse slip, under the appropriate conditions, the loosening process will continue until nut detachment occurs.
- A series of tests has been completed in which the forces needed to tighten and loosen threaded fasteners were measured whilst the joint was being subjected to transverse slip/vibration. Measurements were made of the frictional resistance forces in the circumferential direction and the loosening torque acting on a fastener under transverse slip conditions. It was found that the loosening torque range varied between two positive limits rather than between zero and an upper limit as anticipated by theory. It was also found that the friction coefficient in the circumferential direction in the threads is greater than that on the nut face bearing surface during conditions of transverse slip.

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NOMENCLATURE

A_{slip}	Area of the screw contact surface
D_e	The effective bearing diameter of the nut
F	Bolt preload
F_A	The axial force applied to the joint
T	Total tightening torque applied to the fastener
T_L	The torque required to loosen a bolt.
T_{loosen}	Total torque acting in the loosening direction
T_{ps}	The prevailing torque from the nut whilst transverse slip is occurring
T_R	The torque resisting loosening
T_{ss}	Loosening torque generated from the differences in the resistance offered between the ascending and descending sides of the nut thread whilst it is sliding on the bolt thread
T_{tm}	The torque reacted in the thread of the fastener
d_o	The outer bearing diameter of the nut
d_i	The inner bearing diameter of the nut face
d_2	The basic pitch diameter of the thread
n	The number of tightenings sustained by the fastener
p	Pitch of the thread
r_1	Inside diameter of the screw contact surface
r_2	Outside diameter of the screw contact surface
α	The half included angle for the thread flank (30 degrees for metric threads)
β	Lead angle of the thread
μ_n	Coefficient of friction for the nut face or bolt head (whichever is rotated during tightening)
μ_{ns}	Nut face friction coefficient in the rotational direction when transverse slip is occurring
μ_t	Coefficient of friction for the threads
μ_{ts}	Thread friction coefficient in the rotational direction when transverse slip is occurring
μ_{loosen}	The maximum friction value that would lead to self-loosening when the joint is being subjected to transverse slip
$\mu_{totalslip}$	A reference coefficient for characterising the overall friction behaviour of a bolt/nut assembly under transverse slip conditions $\mu_{totalslip} = \mu_{ts} = \mu_{ns}$

1. INTRODUCTION

1.1 LOOSENING OF THREADED FASTENERS AND THE JUSTIFICATION FOR STUDY

Practically every engineering product with any degree of complexity uses threaded fasteners. A key advantage of threaded fasteners over the majority of other joining methods is that they can readily be disassembled and re-used. This feature is often the reason why threaded fasteners are used in preference to other joining methods. Although threaded fasteners are generally considered a mature technology, significant problems exist with their use. The preload is the initial clamp force imparted into a joint by tightening a fastener. The vast majority of joints rely upon this preload for their structural integrity. The preload acts on the fastener thread creates a torque in the circumferential direction which is resisted by friction. Self-loosening of fasteners leads to a reduction and sometimes, the elimination, of this preload which frequently leads to joint failure. Such joint failures are widespread across many industries and often involve material loss and sometimes fatalities (RAIB (2008), Institute of Road Transport Engineers (2003)).

Although it is known in principle (Verein Deutscher Ingenieure, 2003) how to design bolted joints in which self-loosening does not occur, in practice loose fasteners are common. Uncertainties about the applied forces, friction and the magnitude of the preload achieved in practice can result in joints whose fasteners are prone to self-loosening. Although specific conditions have been identified that it is known will cause self-loosening, mechanisms and loading conditions by which some types of fasteners loosen is unexplained. For example, the complete detachment of prevailing torque nuts is known to occur and has been the cause of major accidents, but prior to this study has been left unexplained in the literature. Prevailing torque nuts have a locking feature in which a small torque (the “prevailing torque”) is needed to rotate the nut down the thread of an untightened bolt. One reason for this was that it was not possible to replicate such detachment under laboratory testing.

In a Ford study of mechanical devices (Munro, 1989), it was found that between 70% and 90% of all failures were directly attributed to threaded fasteners. Loosening of such fasteners is a particular problem. A survey of automobile dealer service managers in the United States by Holmes (1988) indicated that 23% of all service problems were traced to loose fasteners and even 12% of all new cars were found to have loose fasteners.

In the machine tool industry, a paper by Kaminskaya and Lipov (1990) reported that self-loosening of fasteners accounted for more than 20% of all failures of the mechanical systems of machine tools. The time taken to rectify such failures was found to represent 10% of the life to failure of the machine.

The integrity of many structures is reliant upon the clamp force generated by tightened fasteners being maintained for the life of the product. If the fastener generated preload is diminished as a result of the nut self-loosening, the joint's structural integrity is jeopardised. Self-loosening of fasteners is not an uncommon occurrence and has been found to have been the cause of several accidents. Due to the dynamic environment, self-loosening of fasteners in the transportation sector is not an uncommon problem. This is illustrated with the following three examples taken from the road, rail and aerospace sectors:

The Institute of Road Transport Engineers (1986) have reported on the loosening of nuts which secure wheels on commercial vehicles, leading to wheel detachment. A study by Knight et al. (2006) completed for the Department of Transport indicated that annually in the UK there are between 150 and 400 wheel detachments on commercial vehicles. Of the wheel detachments between 50 and 134 result in damage only accidents, 10 to 27 in injury accidents and 3 to 7 in fatal accidents. Hagelthorn (1992) indicated that such problems are not isolated to the UK.

On Friday 23 February 2007 a high speed train derailed on points at Lambrigg, near Grayrigg in Cumbria, UK (RAIB, 2008). This accident was as a result of both plain and prevailing torque nuts becoming detached from the bolts allowing the switch rail to be struck by the inner faces of passing train wheels. This caused subsequent failures of other parts of the switch structure and ultimately the derailment of the train leading to the injury of several people and the death of one person. Nuts coming loose are also now considered to be the cause of the Potter's Bar accident (HSE Investigation Board, 2003) in 2002 when seven people died.

In 1999, a Tupolev passenger jet crashed in China killing 61 people. A bolt that connected the pull rod and bell crank in the elevator control system became detached because a self-locking nut had self-loosened. This subsequently led to the loss of the aircraft's pitch control that resulted in the plane nose diving into the ground killing all on-board. Other accidents have also been reported on aircraft involving the self-loosening of nuts (FAA, 1982; AAIB, 2005; AAIB, 2006).

Civil engineering is not immune to the problem of self-loosening. An analytical review by Plaut and Davis (2007) has recently been published discussing the well known Tacoma Narrows Bridge accident in which it totally collapsed on the 7th November 1940. The paper indicates that the root cause of the failure was due to the loosening of bolts on a frictional grip joint that held a cable band whose failure led to the torsional motion of the deck and to the bridge's demise.

Loosening issues tend to occur where dynamic loads act on joints secured with threaded fasteners. Threaded fasteners are commonly used to secure various implants to bone within the body. Loosening of single-tooth implant screws is an ongoing problem. An American paper reports that 43% structural screws came loose in the first year (Aboyoussef et al. 2000). A Japanese paper reports that 26% of screws needed re-tightening in the first year (Khraisat et al. 2004). Researchers have investigated performance comparisons between various designs in terms of loosening resistance (Dixon et al. 1995) and how coating the dental implant can influence loosening (Elias et al. 2006). Some experimental testing of the loosening process has also been completed (Binon, 1998; Lee et al. 2002). Loosening of screws used to attach implants in joint replacements is also an ongoing issue (Möller et al. 2004; Ahn and Suh, 2009).

Although research over the last sixty years has revealed specific mechanisms that can cause fasteners to loosen, significant outstanding issues exist. The loosening in regard to threaded fasteners relates to a loss of preload. The loss of preload can either be as a result of rotation of the fastener, frequently referred to as self-loosening, or non-rotational loosening as a result of creep like processes. This study has focused on self-loosening.

1.2 ROTATIONAL AND NON-ROTATIONAL LOOSENING

Self-loosening, is when the fastener rotates under the action of external loading. Non-rotational loosening is when no relative movement occurs between the internal and external threads but a preload loss occurs. These two types of loosening process are illustrated in figure 1.1.

One characteristic of threaded fasteners is that they allow complex assemblies to be built and dismantled as necessary. Under certain conditions, a threaded fastener can self-loosen, that is, it can rotate by itself. The detailed mechanisms that result in self-loosening have been a subject of research for the last sixty years. The research presented here is one contribution to this body of knowledge. Self-loosening of threaded fasteners can lead to complete disassembly and catastrophic failure of a product.

Non-rotational loosening can occur as a result of deformation of the fastener itself, or the joint, following assembly. The preload exists as a result of the axial extension of the fastener and the compression of the joint. Normally, the fastener extension and joint compression are elastic. Subsequent to assembly, changes in the fastener extension and joint compression can lead to a preload loss occurring. The changes may be reversible, as a result of differential thermal expansion of the bolt to the joint, or permanent, as a result of plastic deformation, creep or stress relaxation. Non-rotational loosening is surveyed in more detail in chapter two.

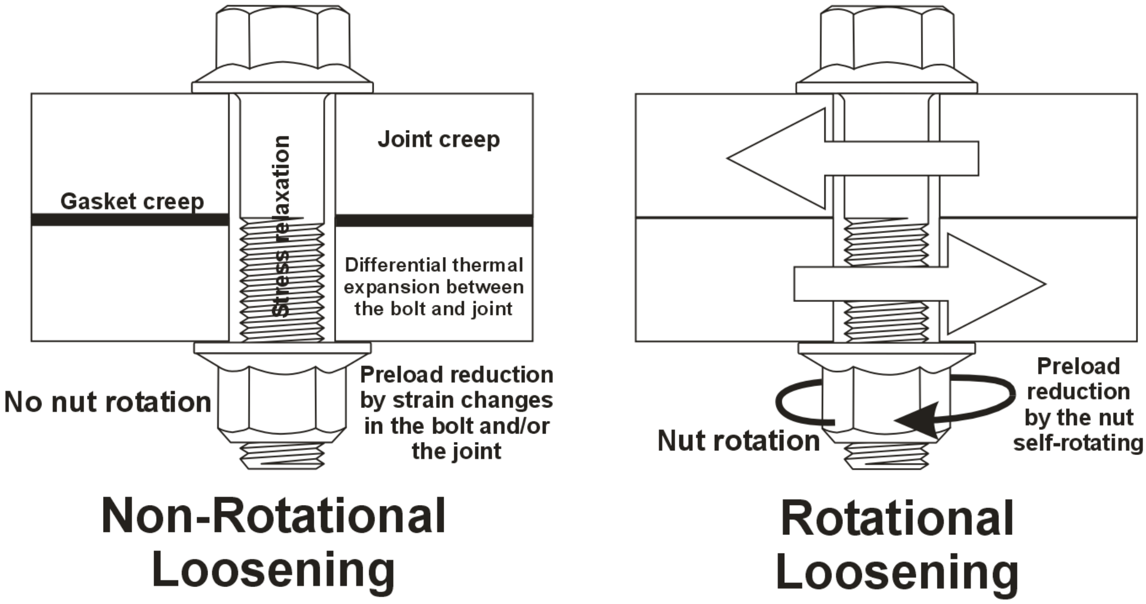

Figure 1.1 Non-Rotational and Rotational Loosening

A free body diagram for a nut being tightened onto a bolt is shown in figure 1.2. In the diagram it is assumed that the nut is tightened using a pure torque. When a nut is tightened with a spanner there will also be an additional shear force acting on the nut equal and opposite in direction to the force applied to the spanner. The diagram also assumes that the torque T_{tm} is smaller than the head friction torque otherwise a restraint would be needed on the bolt head to prevent rotation.

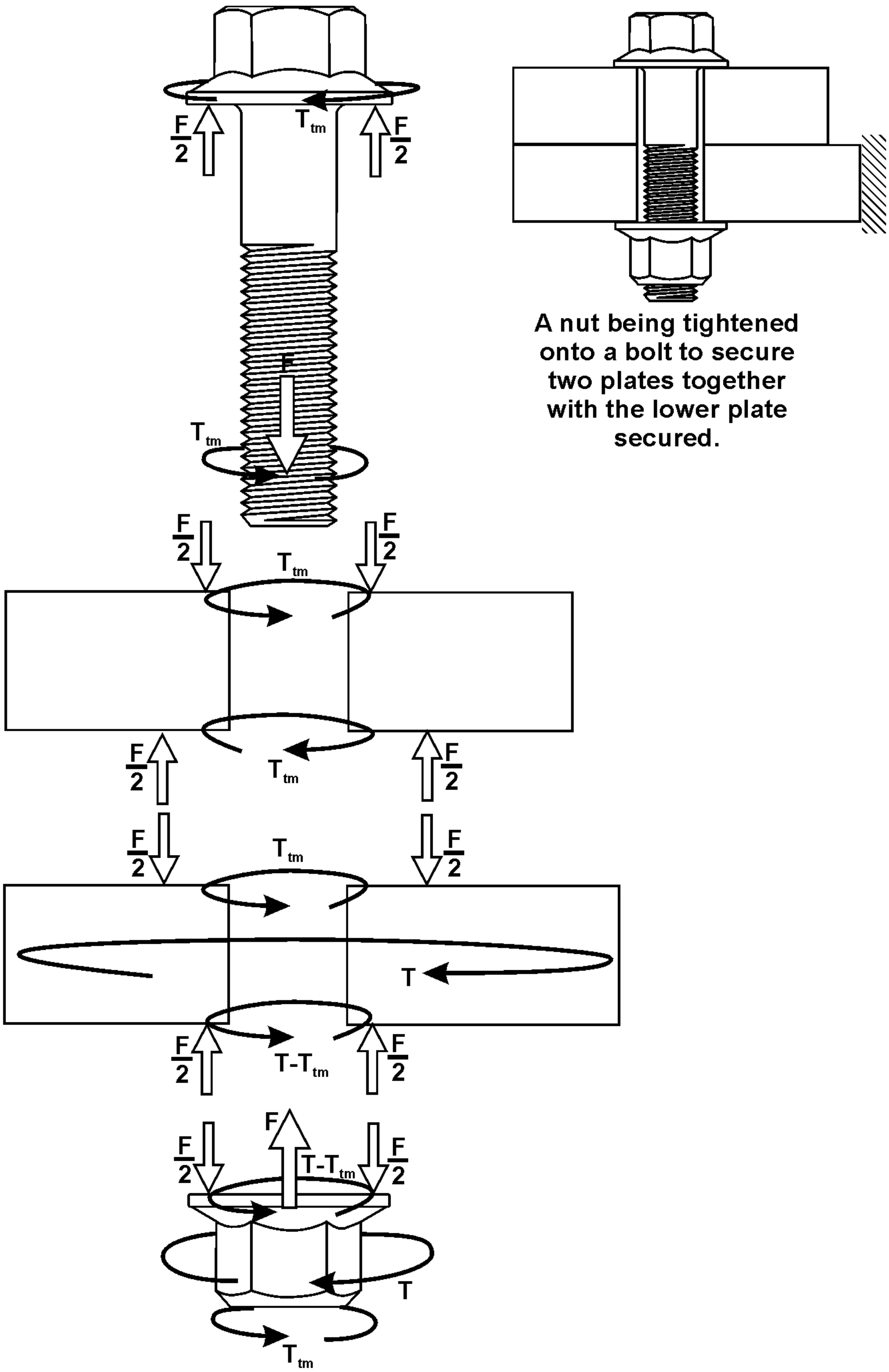

Figure 1.2 Free body diagram for a nut being tightened onto a bolt.

1.3 TYPES AND CLASSIFICATION OF FASTENER LOCKING METHODS

1.3.1 Prevention of self-loosening by design

Experience indicates that self-loosening will not occur if a bolted joint is designed so that the residual preload after any non-rotational loosening losses has occurred is sufficient to resist the applied external forces without joint movement occurring. Systematic design procedures such as VDI 2230 (2003) allow joints to be designed on this basis so that plain non-locking fasteners can be used in dynamically loaded applications.

The computational and intellectual burden associated with designing joints to VDI 2230 means that its usage is not widely understood or applied. Over the last ten years, the development of software implementing the methodology is leading to its wider adoption, although its use is still generally limited to larger organisations. Experience of the application of VDI 2230 by the large automotive companies indicates that when the procedure is systematically applied, self-loosening of fasteners does not, in general, occur. The VDI requires detailed knowledge of the forces being applied to the joint. In many applications outside the automotive and aerospace sectors such information is frequently not available. This is one reason why self-loosening of threaded fasteners is likely to remain an ongoing problem.

1.3.2 The use of fastener locking devices

Based upon the author's experience, there are several circumstances when the use of a locking device may be appropriate on a threaded fastener, specifically:

1. In many applications the forces acting on a joint are either difficult to predict or would require a large amount of research effort to establish. In such situations the engineer uses his or her judgement to estimate the likely forces acting on an appropriately sized joint. Testing can help to verify the design, however, if the application is subject to complex load pattern there will be uncertainty that the all the conditions and combinations of loading have been evaluated.
2. The bolted joint is a complex structural assembly and the knowledge to analyses effectively is not widespread in the engineering community. On low volume products the size and strength of bolts are often based upon past experience and judgement rather than a detailed analysis or test programme.
3. Friction plays a critical role in controlling the preload developed from an applied torque to a fastener as well as the slip resistance of two joint surfaces. Assumptions have to be made about the friction coefficient which develop between sliding

surface that may prove to be false on occasions. For example, contamination by grease between two joint surfaces can significantly lower the joint's slip resistance.

4. In some applications it can be difficult to define the difference between reasonable use and abuse of a product, for example, the driving of off-road vehicles. Designing for extreme loads may impose too high a cost or weight penalty. In such situations joints may be designed to slip or separate in the rare instances when extreme loading occurs.
5. Differential thermal expansion can cause joint movement that cannot be prevented without incurring structural problems. Examples of this are fishplates connecting rail sections together. The rails expand and contract depending upon the ambient temperature, the fishplates facilitate this movement whilst holding the rails together.
6. Many standard designs have had to sustain increased loads since their inception as a result of changes in practice. For example, wheel rim sizes on commercial vehicles. Improved braking efficiencies, higher engine power and increased axle loading have all contributed to increasing the loading severity on the joint.
7. The initial tightening of the fasteners of a manufactured product can be closely controlled giving some predictability in the resulting bolt preload. In service, controlled tightening can be erratic and problematic being dependent upon the tools being available.
8. The causes of loosening are not widely understood by practising engineers. In engineering magazines there is a significant amount of advertising advocating the use of various proprietry locking devices. As a result, in the authors experience, locking devices are sometimes used in applications in which their use is of questionable benefit.

For the reasons given above, locking devices are frequently applied to threaded fasteners. What is generally demanded in the above situations is that although partial loosening may occur, detachment of the nut from the bolt should not occur. Such detachment has historically been the cause of several serious accidents. In a chapter on self loosening, Hess (Bickford and Nassar 1998) classifies fastener locking devices into four groups:

- Preload independent locking methods.
- Free spinning preload dependent locking methods.
- Prevailing torque locking methods.
- Adhesive locking methods.

1.3.3 Preload independent locking methods

This category of locking devices is unique in that they allow the nut to be secured to the bolt with or without any preload being present.

Table 1.1 Preload independent locking methods

Locking approach	**Example**
Cotter pin and castle nut	
Multiple bolts locked together	Wiring of nuts and bolts, RIC clip (figure 1.2a).
Use of two nuts	Thick nut and jam nut, Hard-Lock nuts (figure 1.2b), Symmetry bolt, PAL nut, Wheelsure device.
Restraining plates	Tabbed washer, keeper plates
One way devices	Tine Lok, Visilok, BAMAC security bolt

In general the key advantage of this category of locking methods is that they can function without preload, however, they have a cost/assembly time penalty.

1.3.4 Free spinning preload dependent locking methods

Free spinning fasteners offer the advantage that they can easily be applied without the need for a tool to run the nut down the thread prior to clamp-up.

Table 1.2 Free spinning preload dependent locking methods

Locking approach	**Example**
Face serrations	Durlock and Whizlock fasteners
Wedge-lock washers	Nord-Lock (figure 1.2c) and Disk-Lock washers (figure 1.2d)
Serrated face washers	Schnorr washer (figure 1.2e)
Lock washers	Helical spring washers (figure 1.2f), internal and external tooth washers
Special thread forms	Spiralock, Step Lock and Dardelet thread profiles

A significant disadvantage of preload dependent locking devices is that if the fasteners are not adequately tightened, or preload is lost due to non-rotational loosening, the fastener can come completely loose.

1.3.5 Prevailing torque locking methods

A large group of locking devices relies upon a prevailing torque. With this type of locking device, a torque is needed to run a nut down a bolt thread. A positive attribute of prevailing torque fasteners is that the locking feature can be verified at the time of assembly.

Table 1.3 Prevailing torque locking methods

Locking approach	Example
Feature incorporated in the thread form	Bump thread, interference threads
Polymer applied to the bolt thread	Tuflok
All-metal prevailing torque nuts	A large range of this type of nut is made, examples include the Binx (figure 1.2g), Philidas, Aerotight (figure 1.2f), Stover and Flaig and Hommel nuts.
Non-metallic insert prevailing torque nut	Nyloc and Matoplastie nuts

The prevailing torque can decrease with re-use of the nut/bolt. Prevailing torque fasteners are known to come completely loose occasionally. The cause of such loosening and detachment and is partially the subject of research published in this thesis.

1.3.6 Adhesive Locking Methods

Use of an adhesive to fill the gap between threads and prevent their relative movement was developed by Robert H. Krieble in the 1950's. He was the co-founder of the Loctite Corporation. Chemical threadlockers are anaerobic liquids that cure to a solid state when activated by a combination of contact with metal and the lack of air.

Table 1.4 Adhesive locking methods

Locking approach	Example
Adhesive applied to the thread in liquid form	Loctite, Turbo Lock, Rite-Lok
Microencapsulated adhesive	Dri-Lok, Precote

Oil and grease on fasteners can affect the adhesives adhesion to the thread surface, so degreasing the fasteners is recommended. Since many fasteners need to be disassembled, lower shear strength adhesives are used which can lead to partial loosening under severe vibration.

Examples of a selection of the fasteners mentioned are presented in figure 1.3.

Figure 1.2a RIC Clip

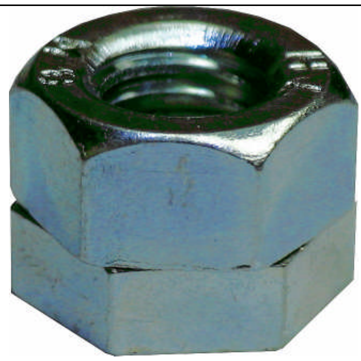

Figure 1.2b Hard Lock Nut

Figure 1.2c Disc Lock Washers

Figure 1.2d Nord Lock Washers

Figure 1.2e Schnorr Washer

Figure 1.2f Helical Spring Washer

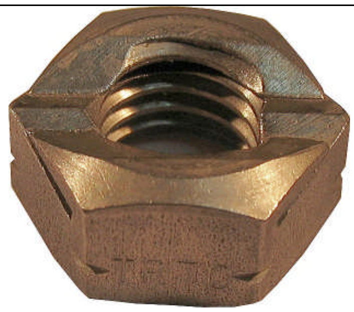

Figure 1.2g Binx Nut

Figure 1.2h Aerotight Nut

Figure 1.3 Examples of types of fastener locking devices

1.4 STRUCTURE OF THE THESIS

The subject of this thesis is the loosening of threaded fasteners and the role which friction plays in this process. The structure of the thesis is in five main sections:

1. A review of the literature on the loosening of threaded fasteners with particular focus on rotational loosening.

2. Details of measurements completed of the coefficient of friction of threaded fasteners. Particular emphasis is on the effect on the coefficient of friction of re-tightening electro-zinc plated fasteners.

3. Details of experiments completed on the self-loosening of threaded fasteners by purely transverse vibration using a standard test approach. Several types of prevailing torque nuts were tested.

4. Details of experiments completed on the self-loosening of threaded fasteners by transverse vibration with simultaneous axial loading. Modifications were made to a test machine to allow axial loading to be simultaneously applied to the joint as well as transverse vibration. A theoretical model is also presented

5. Details of experiments are described in which the reduction in the apparent friction is measured when the joint is subjected to transverse vibration. The almost total elimination of the friction force in the circumferential direction of a fastener when the joint experiences transverse vibration is an interesting phenomenon which has not been extensively studied.

2. LITERATURE SURVEY

2.1 INTRODUCTION

The aim of this literature review is to provide a critical look at the research that has been completed on the subject to date and evaluate the importance of some of the key work.

The demand for fasteners dramatically increased during the Industrial Revolution. James Watts' development of the steam engine in 1765 resulted in the need to attach large metal parts together securely so that significant load could be sustained. According to the Industrial Fastener Institute (1991) increasing mechanisation generated a strong demand for threaded fasteners that was initially met by small manufacturers producing essentially proprietary parts. The nuts and bolts could be interchanged between the same manufacturer, but not between manufacturers due to the lack of nationwide standards. On the 15 June 1841, in a paper titled "A Uniform System of Screw Threads" presented to the Institution of Civil Engineers, Joseph Whitworth proposed that threads should have identical depth, pitch and form for a given diameter (Whitworth, 1841). Prior to his work, the screw threads were made to designs and sizes determined by the individual engineer or manufacturer.

In 1864, in the United States, William Sellers proposed a thread form based upon the now standard 60 degree flank angle (Surowiecki, 2002). During the First and Second World Wars problems arose because of lack of compatibility between the Whitworth and the Sellers thread forms. This led to the adoption in 1948 of the unified thread which has a thread angle of 60 degrees like the Sellers but incorporates a rounded thread root like the Whitworth. Also in 1948 the International Organisation for Standardisation (ISO) started work on a standard screw thread system for world-wide adoption. In 1964, agreement was reached on the ISO metric thread system that is in world-wide use today.

Interestingly, for the perspective of this thesis, patents started to appear in the middle of the nineteenth century proposing improvements in bolt and nut design to prevent them unintentionally coming loose. David Cumming in his patent of the 16th June 1868 states that his design would improve the bolt-nut assembly by *'… thereby preventing the nut from being shaken off as is frequently the case where screw-bolts are used in* *the construction of railroad-cars, carriages, bridges and other structures*' (Cumming, 1868).

2.2 NON-ROTATIONAL LOOSENING

A tightened threaded fastener can lose preload without rotating. The loss of preload can be temporary; such as can occur as a result of differential thermal expansion, or permanent, such as can result from creep. There are several causes of non-rotational loosening all of which involve either the bolt additionally elongating or the joint additionally compressing following installation. Relaxation and permanent set are sometimes used as coverall terms to include all types of plastic deformation that can occur in the bolt and the joint. This thesis covers self-loosening in which the fastener self rotates under the action of external forces. Non-rotational loosening will be first covered since this can initiate self-loosening once a reduction in the preload has occurred. The main causes of non-rotational loosening are discussed below:

2.2.1 Embedding loss

Embedding is localised plastic deformation that occurs under the nut face, in the joint faces and in the threads as a result of plastic flattening of the surface roughness. This occurs even when the loading is below the yield point of the bolt or limiting surface pressure of the joint material, and is the result of the real area of contact between surfaces being less than the apparent area. See (Rabinowicz, 1995) for a discussion on this topic. It is known that the majority of embedding losses arise when the working load is first applied to a joint, changing the contact pressures. Guideline values for embedding loss per interface for steel have been published (VDI, 2003). Research into this phenomenon (Meyer and Strelow, 1972) indicates that, once tightening has stopped, approximately 80% of the embedding loss occurs on first loading by a static or dynamic service load while the remaining 20% is caused by further loading. For typical structural joints whose joint faces are machined faces, about 10% of the preload (or thereabouts) is lost by this process. To account for this effect the VDI methodology (VDI 2003) allows the embedding loss on a particular joint to be included in the design calculations so that a reliable joint is achieved.

2.2.2 Gasket creep

Many joints contain gaskets whose purpose is to create a seal between two flanges to confine fluid or gas. A relatively soft gasket material is frequently used so that a good sealing performance is achieved. Most gasket types rely upon a certain minimum seating pressure, usually provided by bolts, to effect a seal. With such joints usually a reduction in the seating stress occurs over time as a result of creep of the gasket relieving the bolt preload. Although present design calculation procedures do not fully account for creep adequately (Nechache and Bouzid, 2006), recent developments (Zerres and Guerout, 2004) allow an improvement of the leak-tightness of the joint and the prevention of fugitive emissions. Practical steps are usually taken when tightening the bolts to ensure that preload reduction from creep does not impair the sealing performance of the joint. These steps often include re-tightening the bolts after a period of time to compensate for creep .

2.2.3 Creep from surface coatings

To protect steel fasteners from corrosion they are usually coated with a material lower than steel in the galvanic series. Zinc is a popular coating used primarily for cost reasons. Zinc is applied either via an electro plating process, giving a coating thickness of the order of 8µm, or a hot dip galvanising process giving a coating thickness of a minimum of 40µm or more. Essentially thicker coatings give greater corrosion resistance. When the joint is made up of several layers, the zinc thickness can be several times the bolt extension. This can be a problem because zinc is a relatively soft metal and creep reduces the coating thickness causing a preload loss. Research has been completed on the creep characteristics of bolted galvanised joints. Yang and DeWolf (1999) discuss this in some detail. Based upon their research, they show that the creep strain follows the relationship:

$$\varepsilon(t) = \alpha + \beta t^m$$

where $\varepsilon(t)$ is the creep strain at time t, $\alpha, \beta \; and \; m$ are constants and t is the time

Although some reports (e.g. Langill (2001)) state that the relaxation from the coating occurs over the first five days it is clear that from the work completed by Yang and DeWolf (1999) and these test results that the creep will continue for extended periods of time.

Paints of various types are commonly applied to structural members for corrosion and cosmetic reasons. Satoh, Nagatomo et al. (1997) have completed experiments to evaluate the influence of paint film upon the loosening of fasteners. Thick paint films give rise to a significant reduction in bolt preload following tightening and can also assist self-loosening. Thinner paint films based upon etch primer result in a lower amount of preload loss and a reduced tendency to self-loosen.

2.2.4 Stress Relaxation

A significant problem with bolting at high temperatures is a phenomenon known as stress relaxation (Sachs and Evans (1973)). Commonly creep occurs when a material is subjected to an elevated temperature and a constant load. Stress relaxation is a form of creep and occurs when a high stress present in a material is relieved over time; the stress is relaxed with a subsequent reduction in the bolt's preload. Previous research by Sachs and Evans (1973) provides useful information on the stress relaxation characteristics of common bolting materials. However, stress relaxation remains a significant problem when bolting at elevated temperatures. In recent years, standardised test methods published by the British Standards Institute (2006) have been developed for the determination of stress relaxation of bolts. The basis of the tests is for a bolt to be tightened to a specified initial elastic strain, subjecting the bolt to that strain for a specific temperature and time and determining the residual stress in the bolt by loosening at the end of the test.

2.2.5 Yielding from applied loading

When an external force is applied to a bolted joint, usually the majority of the force relieves the clamp force on the interface. However the bolt does sustain a proportion of the applied force. Since bolts are usually tightened to a significant percentage of their yield strength, there exists the possibility that the bolt material could yield as a result of the application of an external force. Modern design codes such as those developed by VDI (2003) allow for this possibility, but at ambient temperatures such yielding is unusual (Newnham and Curley, 1990) even when yield controlled tightening is used. (Yield controlled tightening involves measuring the applied torque and angle of rotation of the nut so that by determining the torque-angle gradient, yielding of the bolt can be detected and the tightening stopped at this point.) In bolted joints operating at elevated temperatures, significant reduction in the bolt material yield strength does occur. In such applications, the target preload is usually reduced and the joint designed accordingly. For example on flange bolting the typical criteria is a tensile stress of 50% of the minimum yield strength of the bolt (ASME (2010)) whereas it is more typical in mechanical engineering to use an equivalent (combined tension and torsion) stress of 90% of the bolt's minimum yield strength (VDI (2003)).

2.2.6 Differential thermal expansion

Bolts are typically tightened at ambient temperature even though they may operate at elevated or cryogenic temperatures. If the joint and bolt materials differ then the load sustained by the bolt may increase or decrease depending upon the expansion/contraction characteristics. Changes in Young's modulus can also lead to preload change at temperature. Transient problems can occur even when the bolt and joint materials have the same expansion coefficients. Normally the air gap in the hole insulates the bolt so that the joint heats/cools faster that the bolt leading to changes in the clamp force, which on flanged joints, can lead to fugitive emissions (Sear and King, 2004).

2.2.7 Loosening through wear

Bolted joints can sustain fretting wear resulting in a dark brown powder that is called "cocoa" (Sakai, 2008). One cause of this fretting wear is micro-slip which is the result of regions away from the bolt hole experiencing slip while those close to the hole do not slip (Ibrahim and Pettit, 2005). Micro-slip in joints assists in damping vibrations (Padmanabhan and Murty, 1990), but quantifying the details can be difficult to investigate and compute (Ouyang, Oldfield et al. 2006). Investigations on the transition from micro-slip to gross-slip conditions (Olofsson, 1995) found that the situation is complex and can occur in a number of different ways. Progress has been made in explaining the physical processes involved in micro-slip and in the development of the governing laws (Hagman and Olofsson, 1998; Sellgren and Olofsson, 1999). Pai and Hess (2003) conducted detailed finite element modelling which led them to conclude that localised slip can accumulate and lead to complete slip and self-loosening of a fastener.

On some joints, movement of the plates comprising the joint occurs by design. Fishplates for example, which secure rail sections together, sustain movement as a result of the rails expanding and contracting. Under such circumstances loosening of the fasteners by wear processes can occur. Higher rates of wear are associated with high friction coefficients (Fouvry, Kapsa et al. 2001) and, when joint movement cannot be prevented by increasing the clamp force from the bolt, Sakai (2008) suggests lubricating the joint surface to minimise wear effects.

2.2.8 Plastic deformation of the fastener and the joint

High surface pressures are created under a bolt head and nut face. On soft materials this can lead to indentation by the nut or bolt head into the joint. When applied loads are sustained by the joint further collapse of the surface asperities can occur resulting in a loss of bolt extension and consequent preload loss. An example of this can occur when large clearance or slotted holes are used to reduce manufacturing costs. When standard washers are used, which are usually mild steel, the washer can collapse to a concave shape. When the joint is loaded, further deformation of the washer occurs resulting in preload loss. To assist in overcoming this problem flange headed nuts and bolts are often used with soft joint materials or when large clearance holes are used.

Jiang et al. (2001) found that when a joint is subjected to transverse cyclic loading and the nut is locked onto the bolt using a high strength adhesive, significant preload loss can still occur. Depending upon the loading magnitude, they found that preload reductions between 10% and 41% could occur. They considered that this loss, based upon a finite element study they had completed, was due to local cyclic plasticity occurring at the roots of the threads resulting in cyclic strain ratchetting. This cyclic plastic deformation led to a redistribution of the stresses in the bolt and a gradual loss of clamping force. Leonavicius et al. (2006) investigated crack propagation and shakedown in threads. They found that a 25% loss of initial preload could occur prior to complete fracture as a result of crack propagation. The author of this thesis has noted similar losses on a locked fastener on a Junker test as a fatigue crack propagates across the thread.

2.3 INTRODUCTION TO THE SELF-LOOSENING OF THREADED FASTENERS

As previously noted, self-loosening of threaded fasteners was observed in the middle of the nineteenth century. The focus was on practical measures to enhance fastener design to resist loosening. Many of the locking methods in use today can be traced to patents lodged over 100 years ago. Some early tests were conducted at the National Physical Laboratory to assess the resistance of fasteners to vibration. One article (Engineering Magazine, 1930) reports on how the performance of nuts and bolts made to the Dardelet thread form were assessed. The test consisted of the bolts being used to attach 12 lb masses to a vibration rig. The frame was oscillated at 1230 rpm with amplitude from 1/8" to 3/16". The test was continued for over 70 hours during which the frame received 5250000 cycles. At the end of the test no movement of the nuts on the bolts could be detected. In an article, Boomsma reviews the work completed on bolt loosening up to that date (Boomsma, 1955). He reports that researchers in Germany in the 1930s attributed loosening to plastic deformation of the fastener and joint surfaces.

It was not clear initially what process was the cause of self-loosening of threaded fasteners. Researchers studied axial loading and impact but it was Junker (Junker 1969) with his discovery that transverse joint movement could completely loosen fasteners who changed the focus of subsequent research.

2.4 LOOSENING FROM AXIAL LOADING AND VIBRATION

Goodier and Sweeney (1945) were the first to investigate, the self loosening of threaded fasteners in detail. In many cases, loosening was not attributable to any obvious cause and so they considered that the process was a natural consequence of the bolt being axially loaded. Their theory of loosening was based upon small movements occurring in the bolt/nut threads as a consequence of being loaded. The elastic radial expansion of the nut and the elastic contraction of the bolt diameter resulted in minute movements, which they concluded, cause the loosening process. They referred to the process as that of a "frictional ratchet".

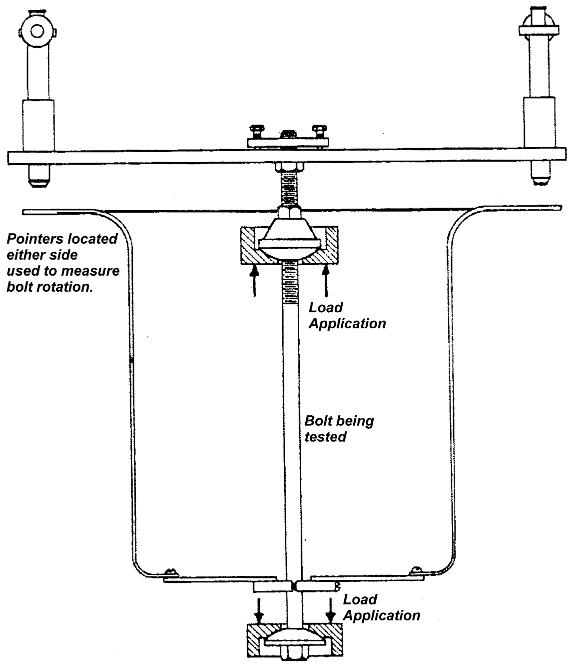

Figure 2.1 Test apparatus used by Goodier and Sweeney (1945)

To verify their theory they constructed a test apparatus that is shown in figure 2.1. They used ¾ inch diameter bolts that were loaded repeatedly from 500 lb to 6000 lb in tensile test machines. Two pointers which were attached to the nut, allowed any rotation to be measured with the aid of a measuring microscope focused on the pointers. It was found that different testing machines gave differing results; of three machines involved in the testing, the maximum nut rotation was 5.5×10^{-3} rotations (2 degrees), the minimum 2×10^{-3} rotations (0.7 degrees) in 500 cycles. The cause of this was uncertain, but the authors considered that the machines applied small twists, as well as tensions arising from the screw threads through which the loads are transmitted or slight imperfections in the machines themselves. The rate of loosening was found to fluctuate cyclically i.e. be larger at the start of the test and then diminishing, then relatively large and then diminishing again and so on. If there was any pause in the test then there was almost always a larger rate of loosening initially when the test was resumed.

Sauer et al. (1950) re-examined the work of Goodier and Sweeney, but instead of using the quasi-static load application that a tensile testing machine produces, they subjected bolts to vibratory conditions more representative of practice. In a series of tests, 5/16 – 18 screws (5/16" nominal thread diameter having 18 threads per inch) were subjected to various dynamic to static load ratios at a frequency of 30 Hz in a Sonntag fatigue testing machine. After a relatively rapid initial loosening at 5000 cycles and below, they found that subsequently only a small amount of loosening was occurring at up to 20000 cycles. The amount of loosening was, although greater than that observed by Goodier and Sweeney, still small – the maximum being less than 6 degrees in any of the experiments. In one of their typical tests they statically loaded a fastener to 500 lbf and then superimposed a dynamic load of 400 lbf onto this. They found that this load ratio affected the amount of loosening experienced. Higher load ratios gave a larger amount of loosening. They also found that nuts that had been previously used exhibited a significantly smaller tendency to loosen than virgin (unused) nuts. Typical sets of results are shown in figure 2.2.

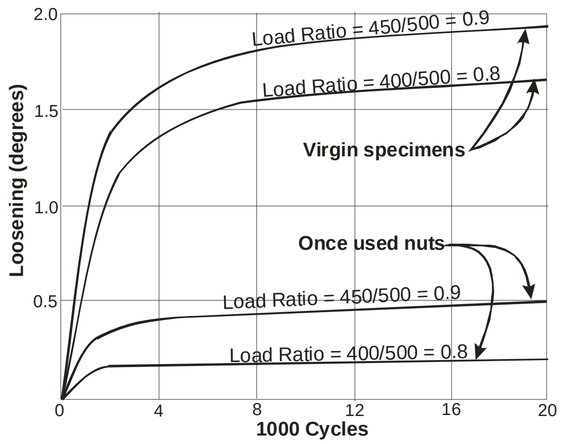

Figure 2.2 Average loosening curves from Sauer et al. (1950)

Clark and Cook (1966) conducted a series of tests to reproduce the work of Goodier and Sweeney, but instead of using nuts they used tapped holes in plates. They found that they could not observe loosening by the process of fluctuating tension alone.

Gambrell (1968) conducted tests to compare the performance of coarse and fine threaded bolts when subjected to similar direct tensile loading and lubrication conditions. His test apparatus consisted of a single bolt joint that was loaded, via a load cell, in tension by being connected to a lever that was driven by a cam connected to a variable speed motor. The apparatus allowed the dynamic to static ratio to be controlled as well as the initial preload, the frequency of loading, the lubrication condition and the number of load cycles. Gambrell found that for a dynamic to static ratio (DSR) less than unity, there was no significant difference between fine and coarse threads. For DSR values greater than unity, fine threads loosened less than coarse threads. The loosening of fine threads could not be attributed to the lubrication condition. However, for coarse threads, lubrication was found to be a significant factor (as illustrated in figure 2.3). He also noted that as the number of load cycles increased the rate of loosening decreased. He found that in the frequency range tested (3.3 Hz to 20 Hz), frequency had no effect on loosening. It should be noted that in practical bolted joints, the dynamic to static ratio is always less than unity, for ratios greater than unity the joint plates would separate leading to fretting and bolt fatigue. It is not clear from his report whether Gambrell eliminated the possibility of fretting of the joint surfaces leading to preload loss and affecting the results he reported.

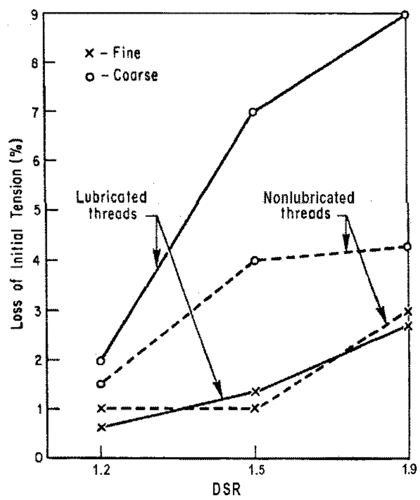

Figure 2.3 Effect of loading ratio on the loss of tension in coarse and fine threaded bolts – from Gambrell (1968)

Kumakura et al. (1995) investigated the performance of various locking devices under repeated tensile loads. They used a tensile testing machine to apply repeated tensile loading to a joint comprising an M10 nut and bolt at a frequency of between 5 to 20 cycles per minute. The loosening that they observed was small but the double nut method was found to be superior in terms of resistance to self-loosening to various types of washers tested. They state that the bolt twisting under an incremental axial load is the loosening mechanism. However, it was unclear from the paper what the mechanism was that caused such a twist.

Hess and Davis (1996) investigated the response of threaded fasteners to axial harmonic vibration experimentally. They found that the direction of rotation of a fastener, either in the loosening or tightening direction, depended upon the frequency and amplitude of the vibratory input. The tests were performed on ¼ - 28 UNF threads that were loose, i.e. no preload present. Hess (1996) subsequently completed a kinematic analysis of the twisting of threaded components loaded by gravity and subjected to axial harmonic vibration. He further developed a theoretical model for the loosening process with co-workers (Hess, Basava et al. 1996; Hess and Sudhirkashyap, 1997; Rashquinha and Hess, 1997; Basava and Hess, 1998). This work presents a mechanism by which a nut can become detached from a bolt once it is completely loose.

2.5 LOOSENING FROM TORSIONAL LOADING

There are many applications in which a single bolt secures two or more parts together on a shaft and are subjected to a fluctuating torque. An example is a pulley secured by a central bolt to a shaft.

Clark and Cook (1966) investigated the effects that fluctuating torque has on bolt. They studied the effect by conducting a series of tests on a bolt tightened into a tapped hole in a bar. A cyclic angular displacement could then be applied to the bar. By the use of strain gauges they could measure the bolt preload as the test was being conducted. The arrangement is illustrated in figure 2.4.

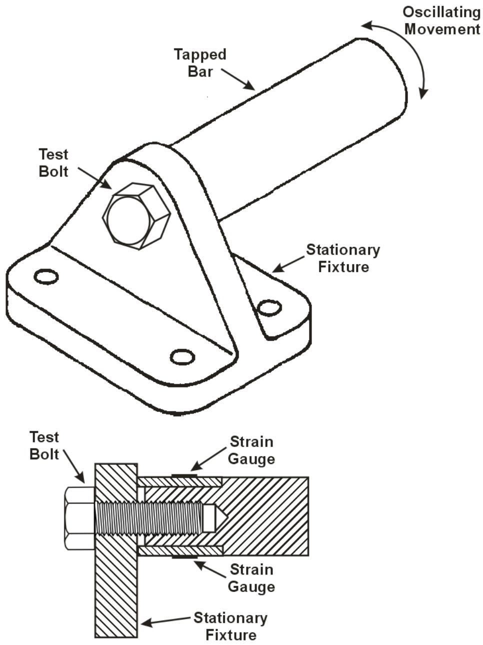

Figure 2.4 Clark and Cook Test Apparatus

They found that there was a limiting value of angular displacement below which no amount of cyclic oscillation would result in bolt loosening. They found that there was a definite relationship between the bolt preload, the oscillating torque that was applied and the number of cycles that could be applied until the bolt became loose. They found that the higher the pre-stress in the bolt (which would produce a higher preload), the greater was the resistance of the bolt to loosening. This is illustrated in figure 2.5. (Note, in this paper, the torque was measured in lbf-ft and the stresses, such as 30 K, refer to the stress in 1000 lbf/in^2.)

Sakai (1978) in an important paper on this aspect of loosening presented experimental and theoretical work completed at the Toyota Motor company. From this work he derived the condition for bolts to self-loosen. He showed there is a minimum relative rotation angle between the clamped parts that is necessary for self-loosening to occur. If movement occurs below this critical angle then loosening will occur without bolt rotation as a result of fretting wear. He also noted that even though a key or pin is commonly inserted between the pulley and the shaft, small relative movement could occur as a result of clearance and elastic/plastic deformation. This generally leads to fretting wear and subsequent loosening over a period of time (greater than 10^6 cycles). Modern practice on joints subjected to rapidly changing dynamic loads, such as the joint between the crankshaft and the damper on an engine, is to transmit the load via

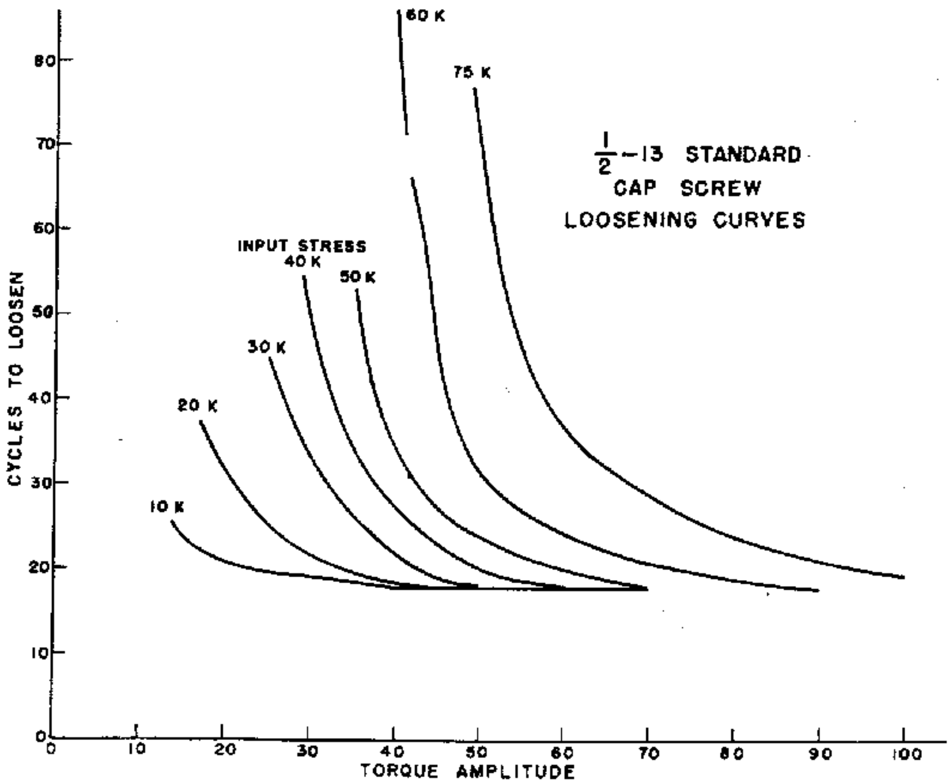

Figure 2.5 Loosening curve for ½-13 cap screw from Clark and Cook (1966)

friction grip. A central bolt provides a clamp force that allows torque to be transmitted by friction between the joint surfaces. Friction enhancing shims can be inserted between the joint surfaces to further increase torque transmission.

2.6 LOOSENING BY IMPACT

During the mid 1960s investigative tests into the cause of fastener loosening were conducted at the Union, New Jersey Plant of the Elastic Stop Nut Company of America

(Baubles et al. 1966). They developed a theory proposing that structural vibrations resulted in a fastener vibrating at its resonant frequency which led to self-loosening. Tests were conducted using a Sonntag universal fatigue testing machine and an electro magnetic vibrator designed in-house. The test they developed became the NASM1312-7 test (National Aerospace Standard, 1997). The work they completed investigated the effect of vibration amplitude on loosening characteristics of the fastener. Although a significant amount of testing was completed, there was little theoretical development to explain when fastener resonance should result in self-loosening.

The NASM1312-7 test involves securing the test fastener in spool like arbours reciprocating within a slotted fixture (as shown in figures 2.6 and 2.7). The fixture is vibrated at 30 Hz at a maximum amplitude peak to peak of 0.45 inches either by using an electromagnetic shaker table or by an electric motor driving an eccentric.

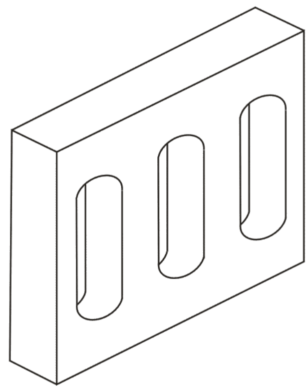

Figure 2.6 NASM-1312-7 Test Fixture

The fastener to be tested is first assembled onto a hollow cylinder that is longer than the width of the test fixture. Washers placed on the cylinder constrain the assembly to act within slotted holes in the test fixture. The assembly is free to move around in the slot. The impact forces acting on the fasteners are perpendicular to their axis. The size of the slots and their clearance in the test fixture is specified in the standard for a range of fastener sizes. Problems with using this test to assess the self loosening characteristics of fasteners include the fact that it is not possible to determine the displacement between the test cylinder and the fastener or the fall off rate of the preload. Presently, the test is largely limited to the aerospace sector. The majority of nuts used on aircraft world-wide have been validated and approved using this test.

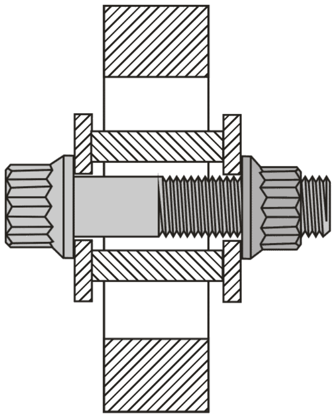

Figure 2.7 Bolt fitted in arbour and located in the NASM1312-7 test fixture.

Koga (1970) developed a theory based upon loosening as a result of impact loading. In his theory an impact to a joint results in compressive waves being transmitted from the pressure flank of the thread. These are reflected from the free end of the bolt and change to tensile stress waves and returned to the pressure flank on the other side of the bolt. If the tensile stress waves are of sufficient magnitude, the clamp force between the threads is overcome, resulting in a loosening action. Koga (1973) followed up on his original work by a theoretical paper that sought to establish the optimum thread angle to prevent loosening. He claimed that the optimum angle was between 62 and 63 degrees depending upon the thread pitch. In his paper he also asserted that thread angles of 60 (metric and Unified threads) and 55 degrees (Whitworth thread) resulted in a comparatively strong loosening action. He subsequently further developed his theory with a co-worker (Koga and Isono, 1986). Koga's work was later challenged as being contradictory to other experimental results which showed that axial vibrations seldom caused loosening (Zadoks and Yu, 1993).

Kumehara et al. (1980) investigated the effect that axial ultrasonic vibration has on the self-loosening of threads. They concluded that self-loosening would occur when the vibration was sufficient to cause joint separation. They refer to the amplitude of the vibrational stress on the surface to achieve this condition as the "Critical Amplitude at the Separation between the Bearing Surfaces". Such conditions are rarely encountered in practice since bolts are typically tightened to relatively high stress values, usually between 50% and 80% of the yield strength of the bolt. Later, other researchers (Vinogradov and Huang, 1989) showed a loosening mechanism involving high frequency excitation causing microslip in the threads to occur. They did note, however, that the frequencies involved were much higher than the operating frequencies encountered in most industries. They also found that transverse vibration was more likely to cause loosening than axial vibration.

Fu discusses the loosening of threaded couplings used in percussive rock drilling (Fu, 1993). His explanation of the loosening process was based upon by Goodier and Sweeney (1945), suggesting that loosening is driven by fluctuation in the bolt tension. The coupling illustrated in the paper would result in minimal thread extension making it prone to loosening from embedding and similar non-rotational processes. This possibility is not discussed by Fu.

2.7 LOOSENING BY TRANSVERSE VIBRATION

2.7.1 The work of Junker

The most influential paper on the self-loosening of threaded fasteners to-date was by Gerhard H. Junker. Junker (1969) reports a theory developed to predict self-loosening under vibratory loading occurs. Junker found that transverse dynamic loads generate a far more severe condition for self-loosening than dynamic axial loads. The reason for this is that radial movement under axial loading is significantly smaller than that which is sustained under transverse loading.

Junker showed that preloaded fasteners self-loosen when relative movement occurs between the mating threads and the fastener bearing surface. Such relative movement will occur when the transverse force acting on the joint is larger than the frictional resisting force generated by the bolt's preload. For small transverse displacements, relative motion can occur between the thread flanks and bearing area contact surface. Once the thread clearances are overcome the bolt will be subject to bending forces, and if the transverse slippage continues slippage of the bolt head bearing surface will occur. According to Junker, once this is initiated the thread and the bolt head will be momentarily free from friction. The internal off torque, present as a result of the preload acting on the thread helix angle, generates a rotation between the nut and the bolt. Under repeated transverse movements this mechanism can completely loosen fasteners.

The paper describes a series of tests together with a test machine he developed (illustrated in figure 2.8) to investigate the effect of transverse movement on preloaded threaded fasteners. The test machine allows a cyclic transverse displacement to be imparted into a bolted joint. A load cell within the joint allows continuous monitoring of the bolt load as transverse motion is applied to the bolted joint. On the machine, a test specimen bolt passes through a bush that clamps the load cell to a fixed base plate. The nut is attached to the bolt and onto a moving base. The moving and fixed base plates are separated by needle roller bearings. The purpose of these bearings is to reduce any friction between the joint surfaces that would resist transverse movement to a minimum.

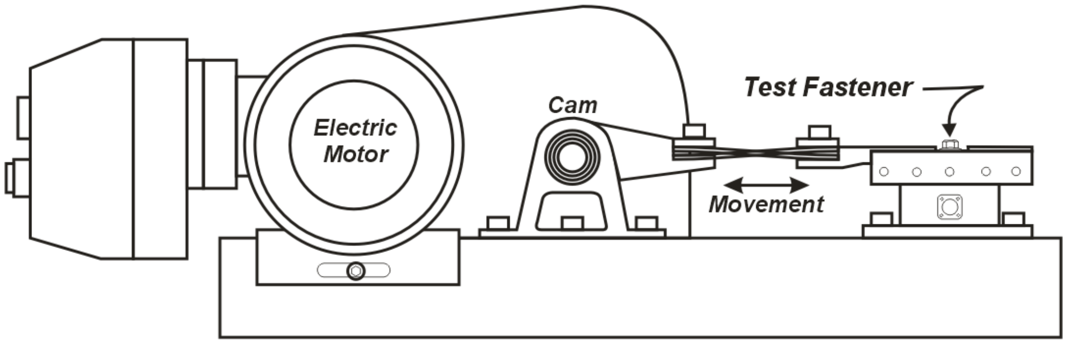

Figure 2.8 The Junker Test Machine

An eccentric cam connected to an electric motor generated the transverse movement. The motion was transmitted through flexure plates into the moving base of the joint. A load cell between the flexure plates and the joint was used to measure the magnitude of the transverse force being applied. Transverse movement was measured by a displacement transducer, whilst a potentiometer was used to simultaneously measure nut rotation.

A graph was produced from the test results in which bolt preload is plotted against the number of cycles of transverse displacement. Such graphs have subsequently been referred to as a preload decay curves. Junker found that the rate of loosening was dependent upon the magnitude of the amplitude of transverse displacement, but independent of the frequency of movement. The test machine allowed, for the first time, the performance of various fastener designs to be quantitatively compared in terms of their resistance to self-loosening.

Junker proposed that a criterion of the locking ability of a screw was the vibration product – this is the product of transverse force times joint displacement. The ability of the fastener to absorb a particular value of vibration product was, he found, an indicator of the fastener's locking ability. A particular fastener could either withstand a particular vibration product or would self-loosen in a relatively short period of time. He tested the same type of fastener to a number of different levels of the vibration product. Junker found that if the preload level was not going to decrease then the fastener had to continue to absorb this amount of energy. On fasteners that self loosen, initially it would absorb the set energy, but this would decrease as the test cycles progressed and the preload decreased.

Junker's influence on the subject of self-loosening has been substantial. The standard method to assess the locking ability of a fastener is by using a Junker or transverse displacement test machine. There are several designs of these machines in use, but all substantially based upon the principle of measuring bolt preload as the fastener is subjected to cycles of transverse displacement. His notion that the locking ability of a fastener should be based upon the vibration product has not generally been adopted, the preload decay curve becoming the de-facto assessment method. The DIN standard DIN 65151 (Deutsche Norm 1994) provides details of the Junker apparatus and the preload decay curve as the assessment method.

2.7.2 Research on transverse slip subsequent to Junker

Later researchers investigating loosening by transverse slip built upon Junker's theory. Finkelston (1972) reports on work completed at the Standard Pressed Steel Company. Tests were completed using a transverse test machine and a range of specific fastener characteristics. Results were presented in the form of preload decay curves and were based upon standard 3/8 inch fasteners. Increasing the preload in a bolt was found to increase the vibration loosening resistance. If the dynamic forces acting on the joint were not high, increased preload was sufficient to prevent loosening. The test involved measuring the loosening characteristic of bolts tightened to 6000 lbf relative to bolts tightened to 4000 lbf. The results under the severe vibration condition of the test are shown in the loosening curves of figure 2.9.

It is generally accepted (Junker, 1969) that the loosening torque acting on a fastener is directly proportional to the thread's helix angle, which in turn is dependent upon the thread pitch. The larger the thread pitch, as in the case of coarse threaded fasteners, the more quickly they would be anticipated to loosen. This is confirmed by Finkelston (1972) in a series of tests described comparing 3/8 inch UNF self-locking nuts to 3/8 inch UNC self locking nuts. As anticipated, the fine pitch nuts loosened at a slower rate than the coarse pitch nuts.

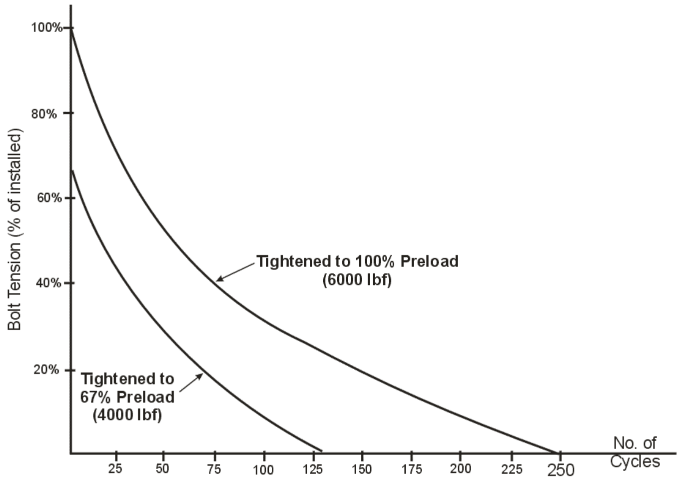

Figure 2.9 The Effect of Preload on the Self-loosening Characteristics of a Fastener (Finkelston (1972))

Many types of self-locking fasteners ("locknuts") use a prevailing torque feature to generate a frictional resistance to thread rotation. Finkelston also completed a series of tests to compare locknuts with a prevailing torque to free spinning nuts. He found that the effect of the prevailing torque was to reduce the rate of loosening and to prevent complete loss of preload when joints are subjected to solely transverse vibration.

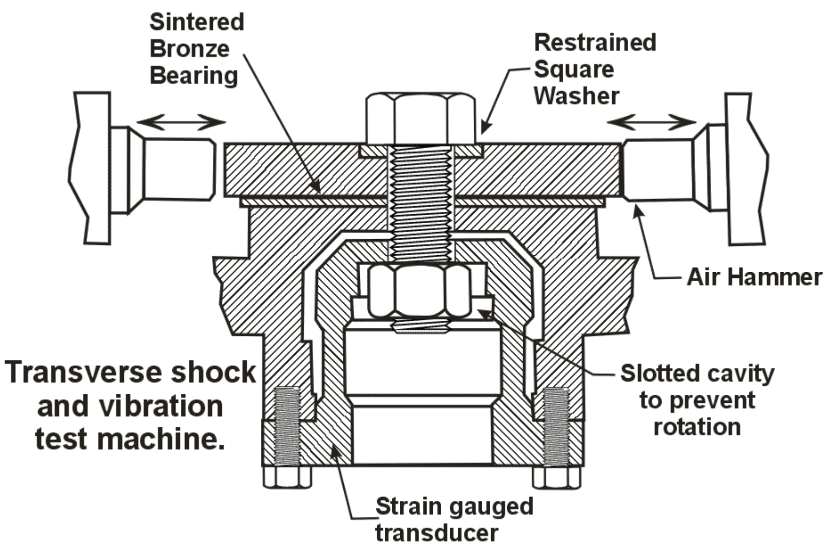

Figure 2.10 Test Apparatus used by Pearce (1973)

Pearce (1973) reports on tests conducted on a small range of fastener locking methods. The test machine is illustrated in figure 2.10 used induced repeated transverse shocks into the joint at a frequency of 16.7 Hz (1000 cycles per minute) with amplitude of ± 3.8 mm (0.15 inches). Fasteners to be tested were inserted through the top of the machine and were secured by a nut in slotted cavity to prevent rotation. A restrained sintered bronze bearing was used to minimise the frictional resistance to the transverse movement. The nut was seated on a strain-gauged transducer that measured the bolt preload. In tests completed by Junker and others, the transverse movement was induced by an eccentric driver arrangement, Pearce used a pair of air hammers to induce the transverse movement. This would induce shock as well as movement into the joint.

Pearce (1973) reported test results that appear to be comparable to those produced from a more conventional eccentrically driven transverse test machine. The tests were conducted on 3/8 - 16 SAE Grade 5 hexagon headed bolts with a phosphate and oil finish. The results of the tests are shown below in figure 2.11. As can be seen, the helical spring lock washer performed only slightly better than a plain bolt. A plastic inserted pellet produced better results, but was still far short of the best results produced by free spinning locking fasteners (that have a circumferential row of teeth under a flanged head) and chemical locking coatings. (Dri-Lok is a pre-applied thread-locking adhesive that is activated by assembly forces, releasing anaerobic adhesive which locks the mating threads. Loctite 242 is a liquid applied adhesive that cures when confined in the absence of air between the thread surfaces.)

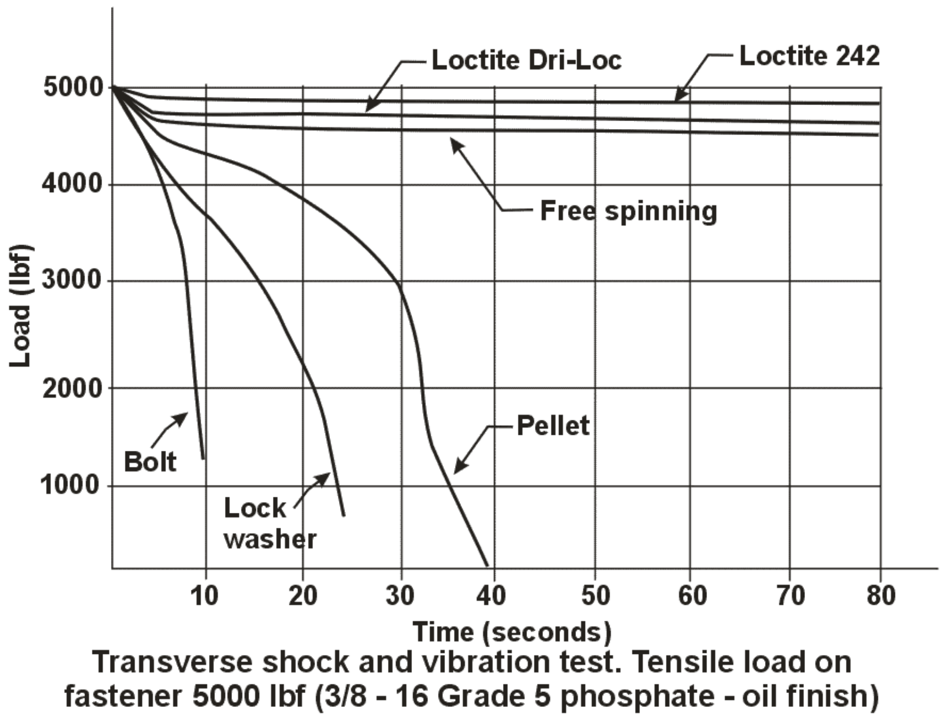

Figure 2.11 Test results from Pearce (1973)

During the 1970s and early 1980s automotive and aerospace companies completed extensive tests (Riches, 1971; Riches, 1975; Light, 1976; Dick, 1983; Light, 1983; Dick, 1984) used Junker machines to assess the loosening resistance of many of the fasteners then used. The effect of this work was to rationalise the type of fasteners used eliminating those from their group standards that were ineffective in resisting loosening.

Haviland (1981) showed that transverse joint movement and subsequent loosening can arise from other mechanisms besides direct shear loading. Differential thermal expansion due to temperature differences or dissimilar joint material can lead to joint slip. Bending of the joint can also lead to displacement and joint slip.

Sakai (1978) also made an important contribution in showing that there is an additional loosening torque, besides the torque resulting from the preload acting on the thread helix, due to the movement of the nut thread on the bolt thread. This is discussed in more detail in chapter 6.

Yamamoto and Kasei (1984) established an equation for determining the transverse force required to cause slip on the nut face allowing for bolt bending. This led to the concept of critical slip discussed in section 2.7.4. They promoted the idea that a cause of loosening is elastic torsion generated in the bolt shank during relative motion in the mating threads, similar in many respects to Sakai's work. Kasei et al. (1989) developed this work and showed that some loosening is possible without bearing surface slip occurring. Although rapid loosening does occur with transverse slip, their tests and experimental work focused on the possibility of it happening when the existence of microscopic sliding was absent or minute. They considered this stage to be the beginning of the loosening process and one that led to allowing transverse joint movement and significant fastener preload loss. To investigate the process they constructed a test apparatus that allowed the measurement of the torque in the fastener during testing. Their tests did indicate that this loosening process does occur, and is relatively small, but can be significant since any loosening will result in a preload reduction allowing further loosening to be made more readily. Kasei and Sawai (1986) also applied the staircase method, commonly used in fatigue testing, to the evaluation of the locking performance of threaded fasteners.

Kerley (1987) used a structured method, called retroduction, for the analysis and testing of the self-loosening of fasteners. His test method involved a cantilever beam consisting of two plates clamped together by a single bolt, the beam being excited via a shaker table with frequencies from 20 to 2000 Hz. His conclusions were that low frequency, not high frequency, caused self-loosening during sine and random vibration on a cantilever.

Zadoks and co-workers (Zadoks and Yu, 1993; Zadoks and Yu, 1997; Zadoks and Kokatam, 2001) reported on a series of theoretical studies into self-loosening of threaded fasteners completed over a several year period. Both classical and computer based techniques were used in their studies. Although a considerable amount of work was completed, their conclusions did not reveal any significant new findings.

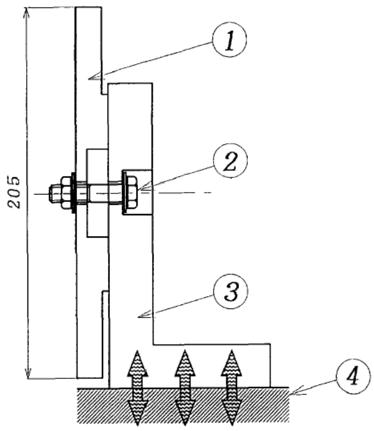

Figure 2.12 Acceleration apparatus built by Sase and co-workers (1996)

Sase et al. (1996) completed a study to evaluate the effectiveness of anti-loosening nuts. The apparatus they used for the investigation was an eccentric cam driven by a motor that forced oscillations to be applied to a beam, that in turn resulted in produced forced transverse movement of the bolt and the nut. The preload in the fastener was continuously measured during the test. They concluded that prevailing torque nuts consisting of a nylon ring and similar all metal nuts were ineffective in preventing loosening. Serrated flange nuts were shown to have the ability to suppress loosening to some extent. They found a double nut combination and a special coned/eccentric nut to be the best performing nuts. Sase et al. (1996) also produced another loosening device for their investigations which was based upon inducing transverse loading into a bolted assembly by accelerating the test apparatus. The equipment is shown in figure 2.12. A plate (1 in figure 2.12) of mass 5 kg was secured by a bolt and nut (2) onto a rigid angle plate (4) which was mounted on the base of an accelerator. The angle of turn of the nut during the test was recorded by video, the clamping force before and after the test was also recorded. The authors proposed this testing method for the objective evaluation of fasteners, quoting several advantages such as frequency and amplitude adjustments are easy and tests can be rapidly completed.

Satoh et al. (1997) completed some important work on the influence that paint can have on the self-loosening of fasteners. A large proportion of practical joints are separated by a paint film yet most of the research into the subject has been completed on machined and clean surfaces. They found that almost complete pretension loss could occur under relatively small dynamic transverse displacements when thick paint films are used. This was partly from creep/embedding effects and partly from self-loosening. A hydraulic pulse-testing machine illustrated in figure 2.13 was used to conduct the tests and set to produce amplitude of +/- 0.3 mm at a frequency of 10 Hz. They also used a lubricant under the bolt head (a 'torque conditioning paste') that they state had a significant influence on the self-loosening characteristics.

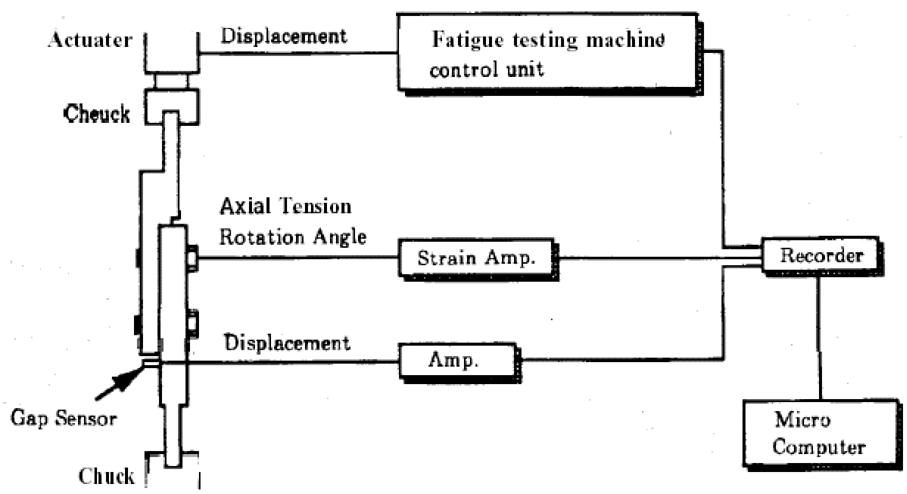

Figure 2.13 Test Apparatus (from Satoh et al. (1997))

In an experimental study, Dong and Hess (1999) used an inertial loaded compound cantilever beam apparatus, similar to that used earlier by Kerley (1987), to study the effect that thread dimensional conformance has on the vibrational loosening characteristics of fasteners. Their study showed that the resistance to vibration and shock-induced loosening improves with increased preload, finer thread pitch, increased head and thread friction, tighter tolerances, lower vibration level and higher vibration frequency. They used a compound beam and applied a 60 Hz sine wave at a level of 25g. A sudden drop in the response acceleration detected loosening of the test specimens. This study was criticised by Matievich (1999) for testing at an unrealistically low preload. Using a Junker machine Matievich concluded that there was no significant difference for the test fasteners to loosen as a function of pitch diameter size. In a later study (Dong and Hess, 2000), investigated the influence of thread non-conformity by using the NASM 1312-7 test method (National Aerospace Standard 1997). In this study they found that undersized bolt threads and oversized nut threads could have a significant effect on loosening. The loosening time for a thread within specification was as much as 97% greater than the expected value for time to loosen for non-conforming product. Hess's conclusion that thread fit has a significant effect on self-loosening was subsequently confirmed by Nassar and his co-workers (Nassar and Housari, 2006; Housari, Nassar et al. 2007).

Pai and Hess (2001) used a finite element model to study different loosening processes caused by localised or complete slip at the bolt head or in the threads. They concluded that their FEM model was capable of modelling the factors that influence slip and self-loosening. In a further paper Pai and Hess (2004) stated that a fastener can loosen at roughly half the shear load required for complete head slip. They state that this is critical for joint design, but current design methodology (VDI, 2003) is based upon sizing the fastener so that the friction grip provided by the preload prevents complete slip. A significant amount of experience in the application of this methodology indicates that fasteners do not come loose if designed on a friction grip basis.

Pai and Hess (2003) also investigated the ideal location for fasteners in a structure to avoid the tendency for self loosening. They modelled a compound beam that was secured by a single bolt using the FE method and experimentally assessed their model by testing. They suggest that fasteners should be placed at vibration anti-nodes in assemblies so that the local shear force acting is minimised. Sanclemente and Hess (2007) completed a parametric study investigating the resistance to loosening caused by some basic parameters such as preload, material elastic modulus, nominal diameter, thread pitch, hole fit and lubrication. Based upon a statistical analysis they concluded that the preload and fastener elasticity are the most influencing parameters.

In some novel work Antonios et al. (2006) investigated the ability of a washer made from shape memory alloy (SMA) to compensate for preload loss from loosening. After a certain amount of preload is lost a heater enveloping the washer is activated allowing an axial constrained recovery of the SMA and control of the bolt preload.

Nassar and his co-workers have completed several studies over a several year period (Nassar and Housari, 2005; Housari and Nassar, 2006; Nassar and Housari, 2006; Nassar and Housari, 2006; Nassar and Yang, 2009) on various aspects of the self-loosening of fasteners. The majority of these studies repeated work completed earlier elsewhere. In one study on the effect of adhesive coating on the performance of threaded fasteners (Ganeshmurthy, Housari et al. 2007) it was found that the locking adhesive used was successful in preventing the nut from becoming detached from the bolt, but a significant amount of preload was lost.

Hashimura (2007) investigated the transition between loosening and fatigue of threaded fasteners when subjected to transverse vibration. He found that if bolts had loosened within 10^3 ~10^4 vibration cycles, damage such as crack nucleation at the root of the thread was not observed. However, if loosening is not observed until approximately 10^5 ~10^6 cycles, a crack was observed at the root of all threads in all his experiments. His test apparatus used air vibrators to induce transverse movement into a plate clamped with a bolt whose preload was monitored by a load cell. He investigated the effect of material strength and tightening method on the loosening-fatigue characteristics of bolts transversely loaded. He found, like other researchers (Burguete, 1995; VDI, 2003), that bolt material strength does not influence the loosening-fatigue life. He also found that tightening the bolt into the plastic region also decreased the loosening-fatigue life. (The author of this thesis notes that in practice this may not be the case since, if additional preload is achieved by either increasing the bolt material strength or by tightening into the plastic region, then the resistance to joint slip will be increased and the loosening-fatigue performance improved.)

In a study on the locking performance of threaded inserts (Cheatham et al. 2008) it was found that inserts which had a prevailing torque feature exhibited loosening characteristics similar to prevailing torque nuts. That is, when tested on a Junker type machine a significant amount of preload loss occurred initially but a fraction of the residual preload is retained.

2.7.3 Design of fasteners to resist self-loosening

For over 150 years developments have been made in the design of fasteners to resist self-loosening. Since Junker's work (1969), study has been made into the locking performance of specific types of fasteners. In cases when it is not possible to prevent transverse joint movement by having a sufficient preload, then use of a fastener that resists loosening maybe a suitable approach.

The use of two nuts to prevent self-loosening is well known. For the method to work a thin nut must be tightened first to a preload typically in the order of 30% to 50% of the target preload. The main nut is then tightened on the top of the thin nut which is held with a spanner whilst the main nut is tightened to the full preload. This approach results in the pressure flanks of the main nut threads and the upper thread flank of the thin nut preventing transverse movement of the bolt thread. Rossides (1982) investigated the efficacy of the use of the double nut system of locking both via a theoretical study and experimentally (1982) by using a Junker machine. Sawa and Yasumasa (2005) investigated the double nut system using several finite element models while Izumi et al. (2008) also used FEM to study the double nut method. The findings of Rossides and Sawa are that the double nut system is highly effective in preventing loosening. Rossides did show in his theoretical work that to achieve an optimum result the preload of the thin nut had to be adjusted to suit the joint/bolt stiffness.

Sase et al. (1998) and Sase and Fujii (2001) completed a series of studies into a modified thread form they called a Step Lock Bolt. This thread design has steps on the thread to resist loosening (figure 2.14). The steps are placed on the external thread

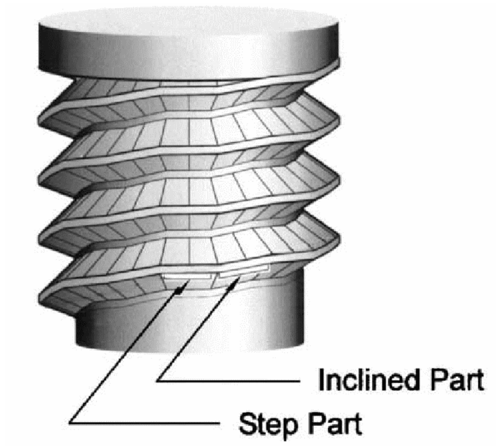

Figure 2.14 The Step Lock Bolt (Sase and Fujii (2001))

and the SLB is used with a conventional nut. The basis of the concept is that any torsion in the bolt, that induces self-loosening will be suppressed because the clamping force is supported on the step parts of the thread. Tests completed by Sase and his colleagues investigated the effect of nut/bolt hardness on anti-loosening performance. They also investigated the effect that the length and number of steps per pitch would have on loosening characteristics. The thread on which they based their investigations had a 8mm diameter and a 0.75 mm pitch. They found that a bolt of hardness Hv380 and a thread design having 10 steps per pitch with a 50% step-space ratio, coupled with a nut hardness of Hv260, prevented loosening over a range of initial clamping force from 4 to 16 kN. The tests were based upon using a transverse displacement apparatus. The difficulties in producing the thread rolling dies for this thread form may prevent its widespread use.

Shape memory alloys (SMA) are a relatively new type of metal with the potential to ‘remember’ a previous shape when activated by a temperature change. An SMA nut can be designed so that when it is assembled onto a bolt at a low temperature, when the joint returns to normal ambient temperature the nut will experience a phase change from a martensitic structure to austenitic which results in a size reduction. This reduction in size results in a clamp force being created on the bolt thread. Zhang et al. (2000) studied the effectiveness of using shape memory alloy to improve the loosening resistance of fasteners. The SMA they used was a titanium-nickel alloy. When tested using a NASM test (National Aerospace Standard, 1997), the SMA nut showed no indication of loosening. Subsequently, Shen et al. (2002) studied the loosening characteristics of nuts made from an iron based SMA. Although such materials do present an interesting phenomenon, it is uncertain whether the material offers sufficient strength to match that which is achieved by high tensile steel used in the higher grades of nuts. The researchers had used preloads that were approximately a third of that used in practice for the size of fastener tested.

Takemasu and Miyahara (2005) evaluated a unique product called a double thread bolt. This bolt has a left-hand thread rolled over a standard right hand thread. The pitch of the left-hand thread being half that of the right hand thread. First a standard nut is tightened onto the bolt with a left-hand nut being tightened on top of it subsequently. Takemasu and Miyahara used the NASM test (National Aerospace Standard, 1997) to evaluate the design and found that it did not loosen. The author of this thesis has also evaluated samples of this product and found that the fine threaded nut stripped when tested on a Junker machine. The force produced by the loosening torque was sufficient to strip the fine thread. Increasing the nut strength could rectify this problem.

Sawa et al. (2006) evaluated a number of fastener locking devices for effectiveness at different preload levels and found that many were only partially effective. Bhattacharya et al. (2008) completed tests on several different types of locking fasteners but the amplitude of the transverse displacement used was small (0.175 mm) for the size of fasteners tested (M16) resulting in a modest loosening tendency.

2.7.4 Critical slip

Under small transverse joint movements the fastener will initially bend rather than suffering slip of the bearing surface. As the movement increases, the bending resistance will increase until it is sufficient to overcome the friction grip of the fastener on the bearing surface. The amount of movement to cause bearing surface slip is referred to as the 'critical slip'. Bearing surface slip is critical for self-loosening to occur. Consequently, a fastener will not rotate loose until the critical value of slip is reached.

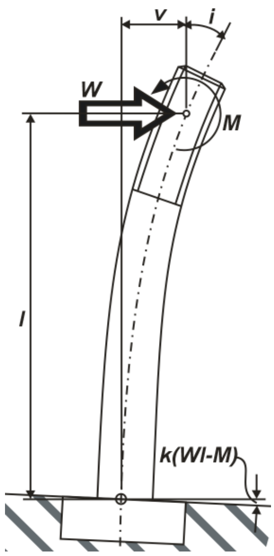

Figure 2.15 Bolt bending and rotation under transverse loading (from Yamamoto and Kasei (1984))

The value of the critical slip is dependent upon several factors; the key ones include: the preload, fastener diameter and the bearing face friction value. Repeated slip below this critical value will result in wear on the joint interfaces and some preload loss which will in turn lower the critical slip value. Under repeated slip conditions, the repetitive bending can also lead to fastener fatigue failure.

Yamamoto and Kasei (1984) presented a theoretical analysis to allow the critical slip to be determined. Experimental work was also completed to seek to verify their findings. Figure 2.15 is taken from their paper. It shows a bolt which is regarded as a beam loaded by a transverse force W, a reaction moment from the mating nut is located at distance 'l' from the bearing surface of the bolt head. Yamamoto and Kasei assumed that the bearing surface of the bolt head tilted against the surface of the joint, in proportion to the moment Wl-M with a coefficient of proportionality k. They developed the following equation for relating the transverse force W to fastener deflection v:

$$W = \frac{1}{l^2\left(\frac{l}{3EI}+k\right)}v + \frac{\frac{l}{2EI}+k}{l\left(\frac{l}{3EI}+k\right)}\frac{L}{4}\frac{\mu_w F_f}{\cos^2\alpha} \tag{2.1}$$

Where E is the Young's Modulus, I is the second moment of area of the fastener, α is half the flank angle (30 degrees for metric threads), L is the nut height, μ_w is the coefficient of friction between the bearing surface of the fastener and the moving joint surface and F_f is the preload. Slip will occur when the value of W becomes W_1 expressed as:

$$W_1 = \mu_w F_f \tag{2.2}$$

On the basis of the deflections that occur in the fastener threads, Yamamoto and Kasei showed, from theoretical considerations, that an elastic torsion would be developed in the fastener shank. This torsion, and a moment on the bearing surface, was a driver for the loosening mechanism under transverse vibration. Five years later Kasei et al. (1989) published further developments in their theory.

Kaminskaya and Lipov (1990) used Yamamoto and Kasei findings on work they completed on the loosening of bolts on a milling machine. They developed an equation to determine the preload needed to resist loosening in the general case of a three-dimension force and three moments.

In 2004, Izumi et al. (2004) presented a paper that analysed Yamamoto and Kasei's experimental findings using a three-dimensional finite element model. They found that the FEM results and to a lesser extent, theory (equation 2.3 below), resulted in a smaller value of the critical slip than what was measured experimentally. They thought that this might be due to the stiffness of the test machine and the roughness of the surface. They refined Yamamoto and Kasei's model to arrive at an equation for critical slip as:

$$S_{cr} = 2F_s\left\{\mu_w\left(\frac{l_g^3}{3EI_g}+\frac{l_p^3}{3EI_p}+\frac{l_g l_p l_n}{EI_g}+k_w l_n^2\right)-\frac{m}{4}\frac{\mu_s}{\cos^2\alpha}\left(\frac{l_g^2}{2EI_g}+\frac{l_p^2}{2EI_p}+\frac{l_g l_p}{EI_g}+k_w l_n\right)\right\} \tag{2.3}$$

The term m is the height of the nut. The equation incorporates the proposal of Nakamura et al. (2001) who performed a finite element analysis to obtain the inclination coefficient of the bolt head which they quote as:

$$k_w = 0.168\left(\frac{1}{d}\right)^3 \quad \left[1/\left(kN\ mm\right)\right] \tag{2.4}$$

In the analysis of Izumi et al, relative slip of the fastener bearing surface occurs when the shear loading F_r reaches the maximum static friction force $\mu_w F_s$ where μ_w is the friction coefficient of the bearing surface.

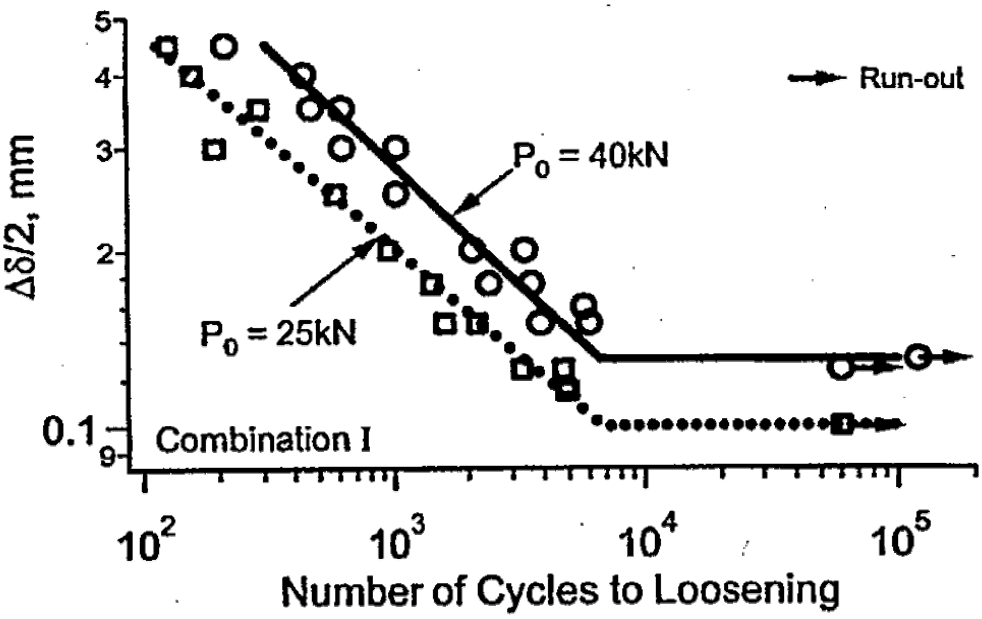

Figure 2.16 Loosening curves at two different preload levels from Jiang et al. (2004)

Jiang et al. (2004) conducted experimental studies in which they used the term endurance limit, rather than critical slip, to refer to the amplitude of vibration needed to achieve loosening for a given clamp length of a fastener. Other researchers subsequently developed their work (Zhang et al. 2006). They produced results showing that there was a critical amplitude of vibration needed, for a given fastener clamp length and preload, for self-loosening to occur. Figure 2.16 illustration is taken from Jiang et al. (2004) and shows that when vibration amplitude is plotted against the number of cycles to loosening, a curve similar to those produced for fatigue results. Hashimura (2007) continued with the approach of Jiang and his co-workers and studied the interaction between fastener loosening and fatigue strength under transverse vibration. Nishimura et al. (2007) developed Izumi's work by completing finite element and experimental studies. They provided a more detailed study of the inclination coefficient for the bolt head, which they found, was dependent upon the fastener preload.

Sakai (2008) simplified earlier work to some extent and showed that the clearance in the threads also influenced critical slip value. He showed that the critical slip was made up of two components. The first component is the tilting of the bolt in the internal thread due to the thread clearances. The second component is due to bending of the bolt as a result of the friction grip of the bolt head on the bearing surface. The amount of bolt bending being within the unconstrained and fully constrained head conditions.

2.8 THE FRICTION COEFFICIENT OF THREADED FASTENERS

Many equipment manufacturers specify that new fasteners should be used if the assembly is dismantled. However, because of a lack of availability of fasteners, or for economic reasons, fasteners are often re-used following the disassembly of a joint. A number of researchers have reported a change in the friction characteristics of fasteners on repeated tightening. Morgan and Henshall (1996) report that wheel studs can experience a significant (50%) reduction in their axial tension after several re-uses. However, they found that recovery to the original condition could be achieved by the use of oil as a lubricant. Jiang and Zhang and Park (2002) report that a doubling in thread friction can occur on distorted head type prevailing torque nuts.

Previous studies (Sakai, 1978; Jiang et al. 2002) have established that the friction coefficients are largely independent of the speed of tightening of the fastener and substantially independent of the preload. Jiang, Zhang and Park (2002) also concluded that the thread friction coefficient is substantially independent of the bolt's preload and that the bearing head friction coefficient decreased with increasing preload. They also noted that repeated tightening and loosening generally increases the friction present in bolted joints especially when the contact surfaces are coated. Their tests were conducted on flanged nuts and they speculated that this was attributable to a change in the pressure distribution over the contact area with increasing clamp force.

Despite these studies, there appears to have been little work conducted on the re-use of the commonly used electro-zinc plated nuts, bolts and washers. Jiang, Zhang and Park (2002) completed some work on some combinations of zinc plated washers, nuts and bolts, but their focus was on additional finish conditions typically used in the automotive industry. Electro-zinc plating (EZP) without additional surface treatments is so widely used on fasteners because it offers good corrosion protection at a low cost.

2.9 THE EFFECT OF TRANSVERSE VIBRATION ON THE FRICTION FORCES ACTING ON THREADED FASTENERS

In 1978 Sakai (1978) showed that the apparent friction coefficient of a bolt with the clamped joint in the sliding state was between 0.005 and 0.02. He stated that this was measured by determining the torque needed to tighten a bolt whilst the joint was being subjected to transverse slip. He also showed that, from theoretical considerations, there was an additional loosening torque component acting when the thread surfaces were sliding over each other. He stated that a necessary condition for the bolt to self-loosen with the clamped parts slipping was for the apparent friction coefficient to be smaller than 0.03. In 2001 Sakai (2001) built upon his previous work and investigated further the role that friction plays in the loosening process. He re-stated that the coefficient of friction in the direction of rotation around the bolt axis when the bolt head and thread surfaces are sliding should be extremely small, a range 0.00 ~ 0.02 is quoted. Sakai advocates tribological research to seek surface treatment conditions to maintain a high coefficient of friction under slip conditions.

In recent years several researchers have investigated the loosening process by using detailed three-dimensional finite element models. Such models have been developed by Zadocks and Yu (1993; 1997), Pai and Hess (Pai and Hess, 2002; Pai, 2004), Izumi et al (2005), Shoji and Sawa (2005), Fujioka and Sakai (2005), Wang and Jiang (2006), Nisimura et al (2007) and Izumi et al (2008). Due to the nature of the method, the friction coefficient is an input rather than an output from the models. Additionally these researchers had to make other assumptions regarding the frictional force in the circumferential direction when transverse slip occurred.

In 2002 Pai and Hess (2002) used a detailed three-dimensional model to investigate the loosening process. With their model they investigated the effect of three different levels of friction and the effect of localised slip on the loosening process. They developed a theory (Pal and Hess, 2004) in which loosening, caused by complete slip as indicated by Junker (1969), could also result from the accumulation of localised slip in the form of elastic deformation. Their test data indicated that a fastener could loosen at roughly half the shear load required for complete head slip. The model assumed that once the tangential friction forces are overcome, either from complete slip or an accumulation of localised slip, the fastener would be free to rotate in the circumferential direction.

In 2005, Izumi et al published an investigation into the loosening process by using a three-dimension finite element model in which they assumed the friction coefficient was a constant 0.15 (Izumi et al. 2005). One of their conclusions was that complete thread slip occurred prior to bolt head slip which was previously considered the initial point of loosening. In the same year Shoji and Sawa also used a three-dimensional finite element model to investigate loosening, this time based upon a constant friction value of 0.1 (Shoji and Sawa, 2005). They concluded that alternating shear stress on the contact surfaces of the threads induces the loosening process.

In 2005 Fujioka and Sakai (2005) showed that friction had a significant influence on the rotational loosening of fasteners. They investigated the theory that friction can influence loosening by using a fine mesh three-dimension finite element model. They found that it was not just the overall magnitude of the friction that was important, but the relative value of the thread friction compared to the head face friction.

Housari and Nassar (2006) developed a mathematical model to assess the effect that variations in the static friction coefficient has on the loosening rate. However, they did not discuss how the apparent friction in a circumferential direction, when transverse slip occurred, might be influenced by the static friction value. Wang and Jiang (2006) used FEM to study the role that a stick-slip mechanism plays in the self-loosening of threaded fasteners. Their study found that localised stick-slip action results in the gradual displacement between two contact surfaces, this contributes to the rotation of the nut. They assumed a friction coefficient of 0.09. Nisimura et al (2007) used a friction coefficient of 0.2, Izumi et al (2008) used 0.15 and appeared to ignore the reduction in the friction force in a circumferential direction when transverse slip of the contact surfaces occurs.

3. OVERVIEW OF THE EXPERIMENTAL PROGRAMME

3.1 INTRODUCTION

A bolted joint would not remain tight without the presence of friction. When a bolt is tightened the preload acting on the thread helix results in an off torque that, without friction, would cause the nut to spin off. By resisting movement, friction also plays a central role in determining the conversion of applied torque into bolt preload. Based upon the literature research completed in chapter 2, as well as the authors own experience, the role that friction plays in the loosening process has not been extensively studied. Friction is central to threaded fastener tightening and loosening processes and deserves further study. The studies in the Thesis can be considered in three groups. Namely:

1. The first phase of testing involved the measurement of friction exhibited when threaded fasteners are tightened, removed and re-tightened. The focus of the study was the effect on the friction coefficient of re-used electro-zinc plated fasteners. The reason why it was decided to investigate this particular type of finish is that it is one of the most popular in use today.

2. A large number of tests were completed using a standard fastener transverse vibration test machine commonly referred to as a 'Junker machine' which induces transverse vibration on a joint. These tests largely replicated tests that had previously been completed by other researchers. Focus was made on the loosening characteristics of prevailing torque nuts. The essential characteristic of such nuts is that a torque is needed to rotate the nut down the thread of an untightened bolt. This type of nut is probably the most common type of anti self-loosening fastener in present use.

A standard Junker machine was modified to allow axial loading to be imparted into the bolted joint whilst it was experiencing transverse vibration. This novel arrangement was used to conduct original studies into the complete self-loosening of prevailing torque nuts under circumstances when transverse vibration is accompanied by axial load.

3. In the final series of original tests, the self-loosening forces acting on a fastener being subjected to transverse vibration were investigated. A study was also made of the drop in the circumferential frictional forces when transverse joint movement was being experienced.

3.2 TESTS ON THE DETERMINATION OF THE FRICTION COEFFICIENT OF FASTENERS

Fasteners are often re-used, either for economic reasons or due to lack of immediate availability of a replacement. (Such as when replacing a punctured car tyre). Re-use of fasteners involves repeated sliding contact between surfaces. Such sliding can have a detrimental effect on the performance of the fastener.

The effect of such repeated sliding contact was investigated experimentally by the study of repeated use of electro-zinc plated (EZP) fasteners. Zinc plating is an extremely common coating used on fasteners. The tests involved measuring the coefficient of friction present between the threads and under the nut face. To do these tests, novel apparatus was designed and made.

In this series of tests the approach was to measure the applied torque, the thread reaction torque and the bolt preload simultaneously. From these measurements the thread friction and nut face friction torque was established allowing the coefficient of friction to be determined for the thread and nut face.

To investigate the effect of sliding distance on the nut face/washer interface, a test apparatus was used that allowed a nut to be rotated on a test surface indefinitely at a pre-determined load. (Normally the bolt load changes when the nut rotates which makes comparison with conventional friction tests difficult.) Based upon studies of existing methods of measuring thread and nut face friction, a design was developed that would economically allow the preload, the total applied torque and the thread torque component to be measured. The test apparatus consisted of a load cell and thrust bearing mounted in series. A bolt passed through the middle of the assembly. A series of disc springs was used in order to increase the resilience of the bolted assembly so that once the bolt had been tightened it would be insensitive to clamp force variation due to wear on the nut face.

3.3 LOOSENING OF THREADED FASTENERS BY TRANSVERSE VIBRATION

Junker and other researchers (Junker (1969), Finelston (1972), Haviland (1981)) have shown that transverse vibration could fully loosen non-locked threaded fasteners. A large number of tests were completed to investigate the self-loosening of fasteners under transverse vibration. The tests were completed using a standard transverse vibration test machine commonly referred to as a "Junker machine".

The focus of these tests was to study the performance of a particular type of lock nut referred to generically as a "prevailing torque" nut. This popular form of lock nut has a locking feature in which a small torque, the “prevailing torque”, is needed to rotate the nut down the thread of an untightened bolt. This type of nut is in extensive use across most industries and is probably the most common type of locking device for threaded fasteners.

There are many varieties of prevailing torque nuts, but in general they can all be classified into one of two categories, those with a non-metallic insert and those which are all-metal. Non-metallic insert nuts typically generate a prevailing torque by incorporating in the top of the nut a polymer insert that is deformed by the bolt thread. All-metal nuts generate prevailing torque by incorporating a means of deformation in their form. Tests were performed on both types of nut.

Further tests have been completed to establish the effect on the prevailing torque of re-use of the nut. The performance requirements for such nuts are specified in the ISO 2230 standard (British Standards, 2008). The standard stipulates that a minimum level of prevailing torque, for a given thread size, must be maintained for up to five re-uses. In many industries such nuts are re-used, and in some industries, such as the rail industry, re-tightened if found to be loose. As a performance requirement for re-use, the standard specifies that the nut is to be untightened and then re-tightened. The tests completed compared this standard procedure to a test involving the nuts being self-loosened due to transverse vibration. Self-loosening, which happens in practice, should be more severe in that many repetitions of relative thread movement will occur during the loosening process and these will lead to wear of the prevailing torque locking arrangement in the nut. This wear can be anticipated to lower the prevailing torque to a greater extent than that which occurs by manually untightening the nut.

3.4 COMPLETE SELF-LOOSENING OF PREVAILING TORQUE FASTENERS

Instances of the detachment of prevailing torque nuts from bolts have occurred with sometimes catastrophic consequences, but to-date, there has been a lack of understanding of the precise conditions that can lead to such separation. One reason for the lack of understanding is that such complete loosening has not been reproducible on a Junker machine.

The author's study of incidents in which detachments of prevailing torque nuts have occurred indicate that the joint has been be subjected to axial as well as transverse loading. Previous published research had indicated that axial loading alone acting on a joint does not result in any significant self-loosening (Goodier and Sweeney (1945), Sauer et al. (1950)). As part of the author's research programme the suspicion arose that axial loading in the presence of transverse joint movement would affect the loosening characteristics of prevailing torque nuts.

In order to investigate the causes of the detachment of prevailing torque nuts from bolts, the Junker machine used in previous experiments was modified to allow axial as well as transverse loading to be introduced into a joint. Miniature hydraulic jacks were used to allow axial loads to be imposed upon the joint whilst transverse movement was occurring. The arrangement allowed axial loading alone, transverse displacement alone or a combination of both to be imposed on the joint. The results revealed that by combining axial loading with transverse vibration complete loosening of prevailing torque nuts could occur. Further investigation revealed the precise conditions under which such detachment occurs.

3.5 THE EFFECT OF TRANSVERSE VIBRATION ON THE FRICTION FORCES ACTING ON THREADED FASTENERS

Central to understanding the root cause of the loosening is knowledge of the forces tending to loosen fasteners and why there is an apparent reduction in the friction coefficient under certain circumstances. A series of tests was completed measuring both the torque acting in the loosening direction and the apparent reduction in the friction coefficient of a threaded fastener when a joint is experiencing repeated transverse movement.

The Junker machine used in the previous tests was modified to allow torque measurements to be completed on threaded fasteners whilst they were subjected to transverse vibration. The machine was used to perform three types of tests:

1. Measurement of the torque tending to loosen fasteners whilst they were subjected to transverse vibration.
2. Tightening of fasteners whilst they were subjected to transverse vibration.
3. Measurement of the torque needed to rotate nuts in the tightening direction whilst they were subjected to transverse vibration.

The first series of tests involved measurement of the loosening torque being exerted on M8 electro-zinc plated bolts and plain non-locking nuts whilst being subjected to transverse vibration. A torque transducer was mounted on the moving top plate of the machine so that it would react the loosening torque being generated by the fastener undergoing transverse slip.

The second series of tests reproduced the work of another researcher (Sakai, 1978) and allowed the overall friction coefficient of the fastener to be determined following the DIN 946 method (Deutsche Norm, 1991). The torque needed to tighten an M8 plain non-locking electro-zinc plated nuts onto bolts under the conditions of transverse slip was measured.

In the third series of tests an electro-zinc plated flanged nut (with a Spiralock thread form) was locked onto an M6 threaded rod using a high strength chemical threadlocker so that no self-loosening would occur. The torque needed to rotate the fastener under transverse vibration was measured. From the torque measurements obtained it was possible to determine the circumferential coefficient of friction of the nut face under conditions of transverse slip.

4. DETERMINATION OF THE FRICTION COEFFICIENT OF FASTENERS

4.1 INTRODUCTION

Without friction, threaded fasteners could not retain their tension without the addition of a locking mechanism and would subsequently be highly restricted in their application. When a nut is tightened onto a bolt, it is friction that holds it in place preventing it from spinning off. Friction is essential, but too high a friction value can also be a problem to the functioning of a threaded fastener. In the majority of applications, it is the preload that is the primary design requirement. The preload results from the axial stress generated when the bolt is tightened. High thread friction results in a high torsional stress in the threaded section that limits the axial stress available and hence the preload achievable from the tightening process. Preload is difficult and expensive to measure and so usually its control is indirectly achieved by specifying a tightening torque. Friction plays a large part in determining the effectiveness of the conversion of applied torque to fastener preload. To control preload by torque control it is essential to have some limits placed on the range of the coefficient of friction.

When the application sustains purely static loading, it is desirable to keep the friction as low as possible so that the torsional stress in the bolt is minimised. In dynamic applications, a concern is that low friction will aid self-loosening. Most of the major automotive companies specify the acceptable friction range on fasteners they use so as to provide some control over the torque-tension relationship. For example based upon the author's investigations and discussions, Ford, Renault and Peugeot-Citroen specify an acceptable friction range of 0.12 to 0.18. German manufacturers typically specify a lower minimum value, for example Porsche and Volkswagen specify 0.09 – 0.14. Company standards are used (such as the Ford standard WZ 101 (Ford, 2002)) for checking the compliance of the fasteners, such standards being usually based upon relevant national standards (such as DIN 946 (Deutsche Norm, 1991)) adjusted to suit the particular requirements of the company.

In the automotive industry, zinc flake and other coatings that incorporate PTFE or other dry lubricants are typically used to control the friction characteristics. In other industries, electro zinc plated (EZP) coatings are used which typically do not have a coating to modify the friction characteristics. As part of this research project, the friction characteristics of fasteners coated with electro-zinc plating were investigated.

There is a relationship between the torque applied and the preload generated (Deutsche Norm, 1991). This torque-tension relationship involves the thread and nut dimensions and the coefficient of friction in the threads and under the nut face. This relationship is presented in equation 4.1.

The tightening torque is usually determined by using an expression that assumes a fixed (coulomb) value for the coefficient of friction, so that a particular pre-stress is applied to the bolt when it is tightened. The target pre-stress applied to the bolt varies between industries. In the petro-chemical industry, typically a target pre-stress of 50% (ASME, 2010) of the bolt material's minimum specified yield strength is used. In automotive engineering applications, this pre-stress is typically around 75% of the minimum yield strength value of the bolt material (Eccles, 1993). If the coefficient of friction increases, then the pre-stress and subsequently the clamp force, will decrease for a given torque value.

4.2 TEST METHODS

In this study, two methods were used to investigate the friction coefficient changes in the contact interfaces of bolts, nuts and the clamped surface material, specifically:

1. Tests on fasteners during repeated tightening, to establish how the friction coefficient, and the clamp force generated by a constant tightening torque changed as a result of the repeated tightening.
2. Measurements of the nut face friction under conditions of constant pressure, to investigate how the friction coefficient changes with the angular rotation.

The dimensions of the fasteners used in this series of tests are given in table 4.1.

Bolt Property Class (ISO 898)	8.8
Thread Diameter D	12 mm
Thread Pitch P	1.75 mm
Basic Pitch Diameter of Thread d_2	10.863 mm
Included Thread Flank Angle	60 deg.
Bearing Diameter of Nut d_o	18.5 mm
Hole Diameter of Washer d_i	13.0 mm
Effective Bearing Diameter of Nut D_e	15.75 mm
Thickness of electro-zinc plating	0.005 mm – 0.009 mm

Table 4.1 – Fastener Dimensional Details

4.3 DETAILS OF THE TEST APPARATUS

4.3.1 Measurement of the Nut Face and Thread Friction Coefficients

To determine the friction coefficient between the materials at the nut face and those in the threads, a test apparatus was designed and built. The test apparatus included a specially designed test frame with torque, angle and clamp force measuring instruments connected to a computer via an analogue to digital converter to allow data to be collected during the experiments. The tightening torque was applied by an electronic torque wrench that could simultaneously measure the angle of rotation applied to the nut or bolt head as well as the applied torque. The test apparatus comprised a load cell to measure the bolt's clamp force and a torque transducer to measure the thread extension and friction torque. The thread friction and extension torque are isolated by the use of a thrust bearing that allows the reaction to be made by a torque transducer as shown in figure 4.1.

Cabling connects the electronic hand wrench to the A/D converter allowing sampling of the torque-angle measurements. The sampling rate is variable, but an acquisition speed of 10 samples per second was found to produce consistent results. Software was written to read, process and draw graphs of the results.

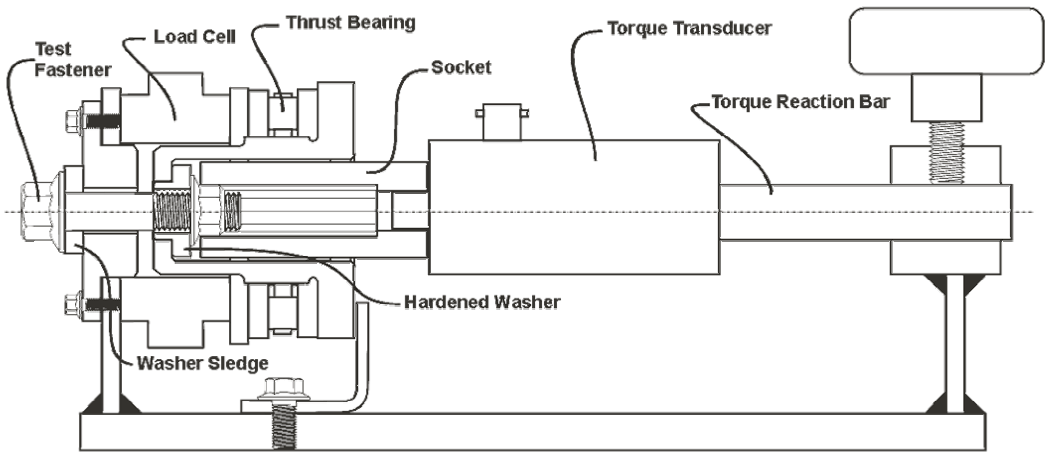

Figure 4.1 - Section through the Test Rig

Upon completion of the test, the test data consisting of applied torque, thread torque and bolt preload was processed allowing the head and thread friction coefficients to be computed for each measured sample interval.

A constant tightening torque of 80 Nm was used in the tests described here. The 80 Nm torque was selected so that the induced stress would not exceed the bolt's yield strength. If the friction coefficient increases on re-tightening, an increase in the tightening torque is necessary in order that a similar clamp force is maintained. This was not done in these tests since the majority of engineering specifications detail a constant torque value. The purpose of this study was to investigate the consequences of re-tightening; any re-tightening would usually be completed by tightening up to the original first tightening torque value.

In this series of tests a minimum of 10 samples were tested. Each sample was tightened 10 times to assess the effect which re-tightening would have on friction.

4.3.2 Measurement of the Nut Face Friction Coefficients under a Constant Load

When a nut was tightened in the experiment the pressure on the nut face changed gradually as the tightening progressed. The tightening was normally stopped when a specified torque was reached making it difficult to establish the effect that the sliding distance of the two surfaces had on the friction coefficient. To investigate the effect that sliding distance had on the nut face/washer interface, a test apparatus was used that allowed a nut to be rotated on a test surface indefinitely at a pre-determined load. The test arrangement is shown in figure 4.2.

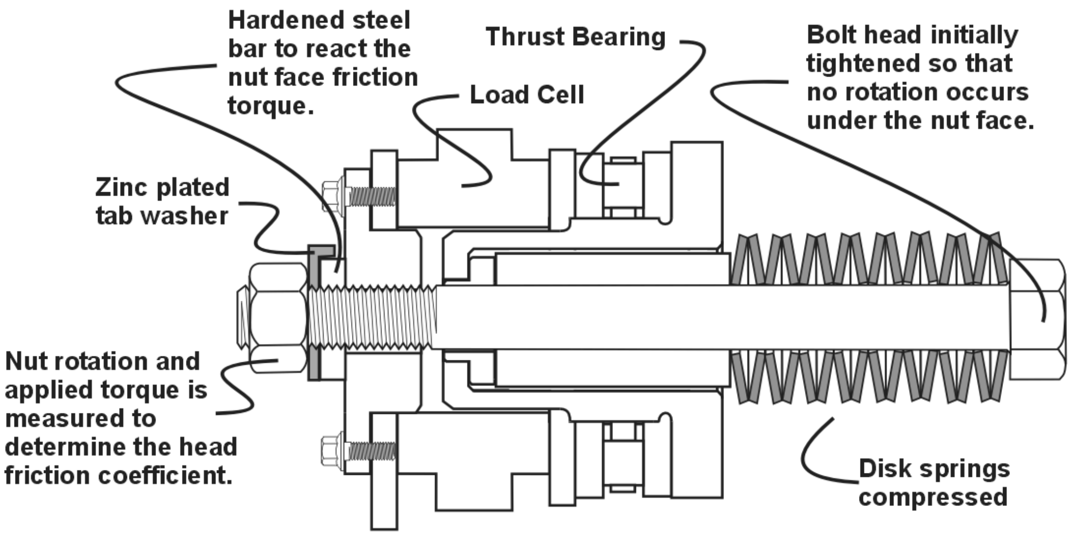

Figure 4.2 Test apparatus for determining the nut face friction coefficient under a constant load

The test apparatus consisted of a load cell and thrust bearing mounted in series. A bolt passed through the middle of the assembly. A series of disc springs was used in order to increase the resilience of the bolted assembly so that once the bolt had been tightened it would be insensitive to clamp force variation due to wear on the nut face. An EZP coated tab washer was used to ensure that the rotating surface was the nut onto the washer rather than the washer onto the support. The bolt head was tightened so that no relative rotation occurred between the nut and the washer. Tests were completed at three values of bolt load, 10 kN, 20 kN and 30 kN.

The test consisted of applying torque to the nut and letting the bolt rotate. Measuring the torque required to rotate the nut against the tab washer allowed the nut face friction coefficient to be determined.

The nut rotation angle was measured simultaneously with the clamp force. It was found that there was a limit to the number of revolutions the nut could be rotated. The applied torque gradually increased resulting in a point being reached when the tab on the washer failed, resulting in the rotating interface being the washer against the hardened steel bar. The test was terminated when this occurred. In these tests a minimum of 10 samples were tested.

4.4 COMPUTATION OF THE FRICTION COEFFICIENTS

The motion of a nut along a thread may be considered to be that of a body moving up an inclined plane under the action of a horizontal force applied at the mean radius of the thread. The development of a screw thread is an inclined plane whose height is equal to the pitch p and the base equal to the mean circumference, $\pi \cdot d_2$, where d_2 is the thread basic pitch diameter. The force F is the force in the fastener.

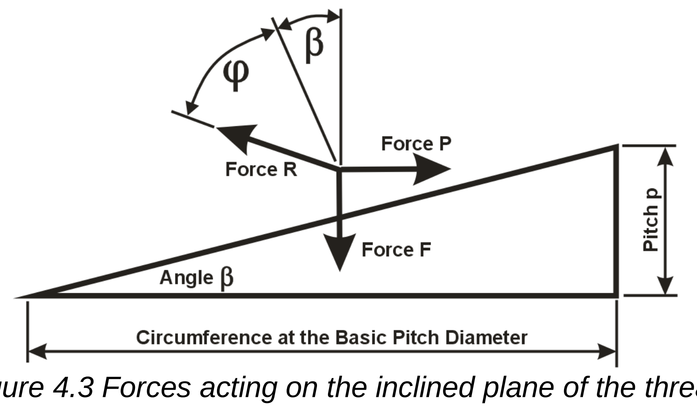

Figure 4.3 Forces acting on the inclined plane of the thread

From figure 4.3, it should be noted that when the force *P* is horizontal, the reaction force *R* would be at an angle φ to the vertical, hence:

$$Tan\,\varphi = \frac{Force\,P}{Force\,F} = \mu_t \qquad (4.1)$$

Where μ_t = the coefficient of friction for the threads.

The force diagram for the inclined plane allows a relationship between forces *P* and *F* to be stated:

$$P = F\,Tan(\beta + \varphi)$$

Since the torque acting on the threads is $T_{tm} = P\,d_2 / 2$ substituting the value of *P* i.e.

$$P = \frac{2T_{tm}}{d_2} = F\,Tan(\beta + \varphi)$$

hence

$$T_{tm} = (F\,d_2 / 2)Tan(\beta + \varphi) \qquad (4.2)$$

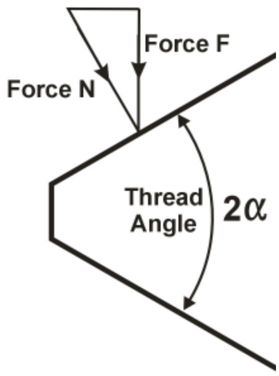

Figure 4.4 Forces acting on the thread flank

Figure 4.4 shows a section through a thread with the forces acting upon it, the thread flank angle is denoted as 2α. In order to account for the inclination of the normal force in this plane, from the diagram:

$$N\cos(\alpha) = F \;\; or \;\; N = \frac{F}{\cos(\alpha)}$$

The normal frictional resistance is:

$$\mu_t N = \frac{\mu_t F}{\cos(\alpha)} = \mu' F$$

where $$\mu' = \frac{\mu_t}{\cos(\alpha)} \tag{4.3}$$

From Equation 4.2, expanding the term in the brackets results in:

$$T_{tm} = F\frac{d_2}{2}\left[\frac{Tan\,\beta + Tan\,\varphi}{1 - Tan\,\beta\, Tan\,\varphi}\right]$$

From figure 4.3 $$Tan\,\beta = \frac{p}{\pi\, d_2}$$

And from equations 4.1 and 4.3: $$Tan\,\varphi = \mu' = \frac{\mu_t}{\cos(\alpha/2)}$$

Substituting this value in the equation for T_{tm}

$$T_{tm} = F\frac{d_2}{2}\left[\frac{\dfrac{p}{\pi d_2} + \dfrac{\mu_t}{\cos(\alpha)}}{1 - \dfrac{p}{\pi d_2}\dfrac{\mu_t}{\cos(\alpha)}}\right]$$

Since the product of the lower line is very small, a close approximation is:

$$T_{tm} = F\frac{d_2}{2}\left[\frac{p}{\pi d_2} + \frac{\mu_t}{\cos(\alpha)}\right] \tag{4.4}$$

The frictional torque developed by the nut or bolt head (depending upon which is being rotated) during tightening is:

$$T_{nut} = F\,\mu_n \frac{D_e}{2} \tag{4.5}$$

where:

μ_n =Frictional coefficient under the nut

D_e =Diameter of the circle where the friction can be considered to act.

Combining the thread and nut face torque gives:

$$T = F\left[\frac{p}{2\pi} + \frac{d_2}{2}\frac{\mu_t}{\cos(\alpha)} + \mu_n \frac{D_e}{2}\right] \quad (4.6)$$

The thread flank angle on metric and Unified thread forms is 60 degrees, hence for metric threads, equation 4.6 simplifies to:

$$T = F\left[0.159p + 0.578\mu_t d_2 + \frac{D_e}{2}\mu_n\right] \quad (4.7)$$

The value of D_e can be taken to be:

$$D_e = \frac{d_0 + d_i}{2} \quad (4.8)$$

The tests conducted consisted of tightening the nuts in the test apparatus whilst measuring the applied torque T, thread reaction torque T_{tm} and the clamp force F generated by the tightening process. Equation 4.7 can be re-written as:

$T = F\left[0.159p + 0.578\mu_t d_2\right] + F\left[\frac{D_e}{2}\mu_n\right]$ i.e. $T = T_{tm} + (T - T_{tm})$

From this, it is a simple matter to show that the friction coefficients for the thread μ_t, and the nut face μ_n are:

$$\mu_t = \frac{\frac{T_{tm}}{F} - 0.159p}{0.578d_2} \quad (4.9) \qquad\qquad \mu_n = \frac{2(T - T_{tm})}{D_e F} \quad (4.10)$$

where

T	Total tightening torque
F	Bolt preload
d_2	The basic pitch diameter of the thread
p	Pitch of the thread
D_e	The effective bearing diameter of the nut
d_o	The outer bearing diameter of the nut
d_i	The inner bearing diameter of the nut face

These are the formulas presented in the DIN 946 (Deutsche Norm, 1991) standard. (Note: During the course of this investigation the ISO 16047 (British Standards, 2005) standard was introduced that is substantially the same as the DIN 946 standard. This had no implications on the work described here. A free body diagram for a nut being tightened onto a bolt is shown in figure 1.2 which may be of assistance in understanding the distribution of forces within the joint.

To ensure that the nut surface rotated against the washer surface rather than the washer rotating against the support, a “tabbed washer” was used to prevent it rotating. It is worth noting that because of the very high values of friction after several tightenings, the tab on the washer failed (sheared) in two instances resulting in rotation occurring on the washer to support face rather than the nut to washer face. Data from the tests when such a failure occurred were ignored.

4.5 RESULTS OF TESTS ON ELECTRO-ZINC PLATED FASTENERS

4.5.1 Nut Face and Thread Friction Coefficients

A summary of the results is shown as a boxplot in figure 4.5. The centre lines in the boxes represent the sample medians. The ends of the boxes represent the upper and lower quartile of each sample and the whiskers indicate the extents.

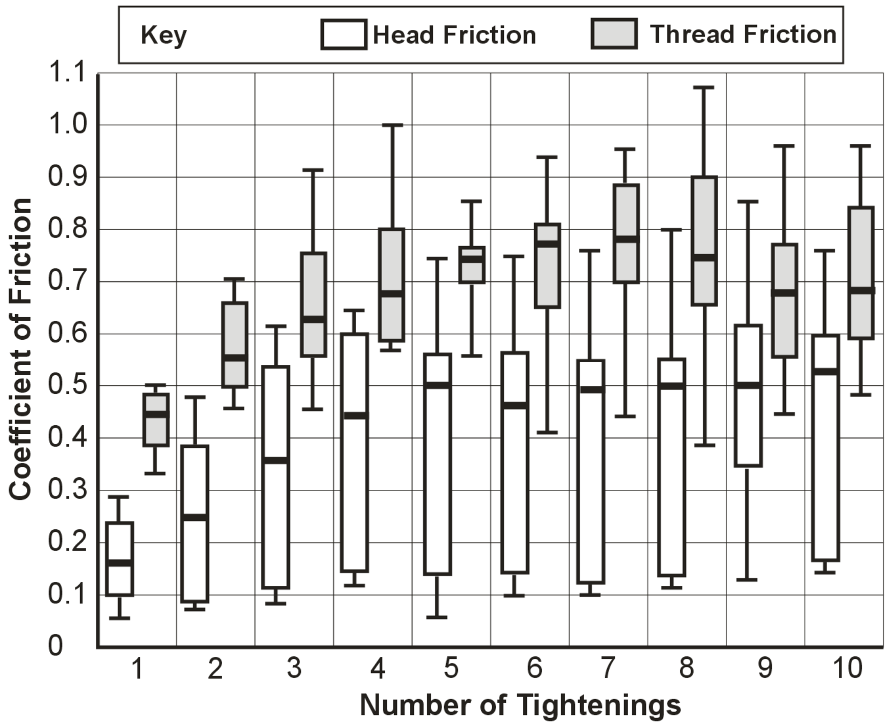

Figure 4.5 – Effect of re-tightening the nut on the friction coefficient. Ten tests were completed for each tightening.

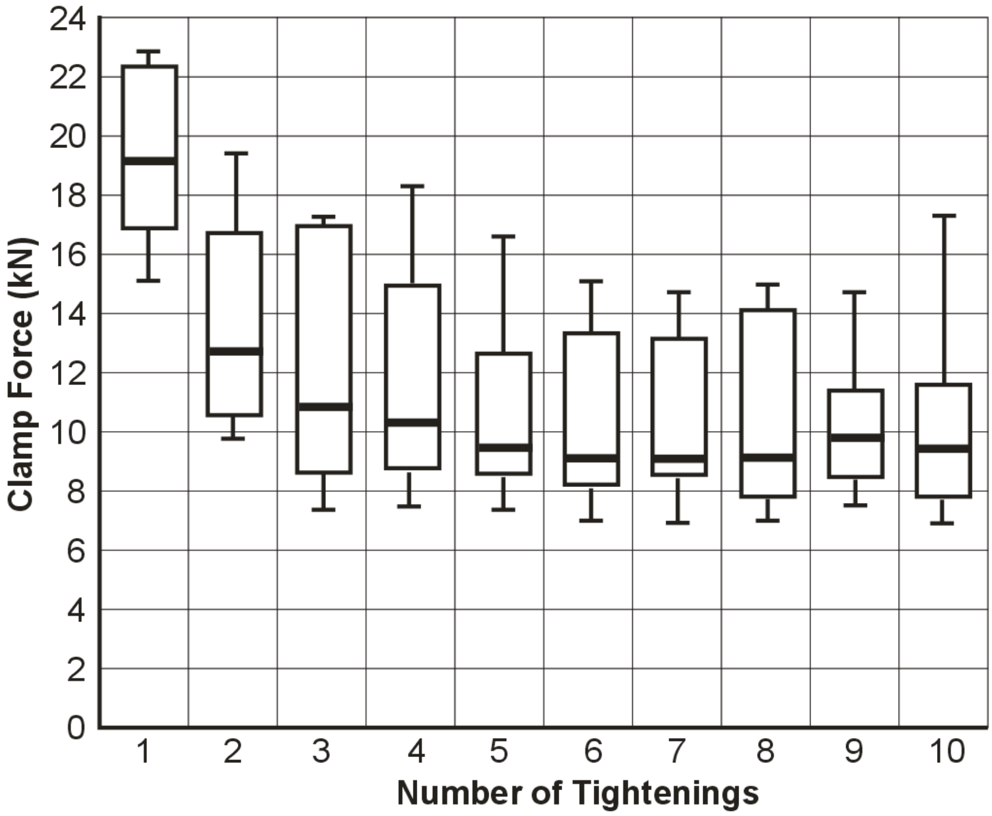

Figure 4.6 – Effect of re-tightening the nut on the clamp force generated. Ten tests were completed for each tightening.

As can be seen from these results:

- The median friction coefficient increases steadily during the first four tightenings and then stabilises at a value approximately twice that observed during the first tightening.
- The trend was for the clamp force produced by the bolt to decrease as the number of tightenings increased and the friction increased. Again, the clamp force stabilised after four tightenings. This is illustrated in figure 4.6. A constant tightening torque of 80 Nm was used for all the tightenings.
- Significant scatter was observed in the results, the scatter in the head friction being particularly pronounced. The scatter observed in the friction from EZP fasteners is one reason why high volume car manufacturers have tended to move away from this type of finish to a zinc flake finish. Such a finish usually incorporates a top coat of PTFE or similar solid lubricant to minimise frictional scatter. In practice, a large scatter in friction can result in an inefficient joint design. The range in the friction coefficient has a direct influence on the fastener strength/size that is determined for a loaded joint. For a given size/strength of fastener, the upper limit of the friction determines the minimum clamp force that will be achieved for a given tightening

torque value. Such a clamp force has to be sufficient to resist joint separation from direct forces and transverse movement from shear forces. The lower limit of friction will result in the maximum clamp force, for the specified tightening torque, that the fastener must sustain without failing. If the friction scatter is increased the net effect is that the optimum bolt size is also increased. If close control of the clamp force is required, control of the scatter in the friction is also needed.

The distribution of load in screw threads has been studied by a number of researchers (Yakushev, 1964). An authoritative paper by Sopwith (1948) suggested that the nearer the thread surface was to the joint face, the higher was the proportion of the overall load it sustained. The load distribution in the threads is dependent upon several variables, one of which is friction present in the threads. The higher the friction, the greater is the proportion of the loading that acts on the threads nearest the joint face.

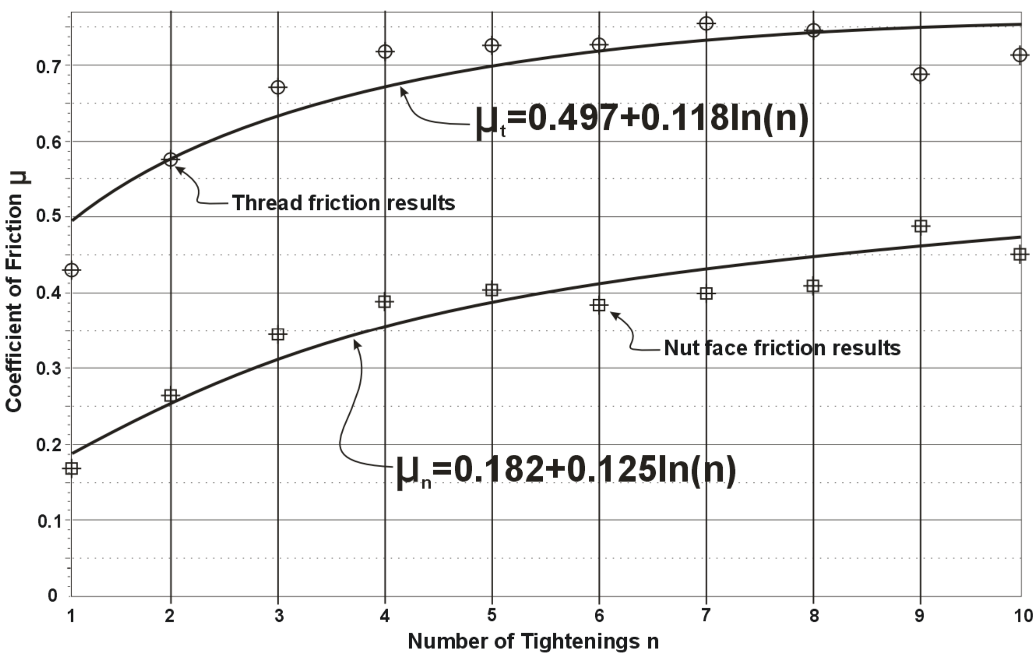

Figure 4.7 The Variation in the Friction Coefficient with the Number of Tightenings. (Mean values plotted.)

As the friction increases in the thread, the proportion of the loading sustained by the first thread increases. The maximum pressure on the threads (derived using Sopwith's equations (1948)) for the first tightening is typically 500 MPa (bolt loading 24 kN) compared to 175 MPa for the nut face pressure. This higher surface pressure explains, to some extent, the reason why the friction coefficient is typically higher in the threads than it is on the nut face for EZP coated fasteners.

Figure 4.7 illustrates an empirical fit was made to establish the relationship between the friction coefficient and the number of tightenings. It was found that an asymptotic relationship fitted, with reasonable accuracy to the variation in nut face and thread friction coefficients as the number of tightenings progressed. For the nut face friction coefficient, the following equation produced a good fit:

$$\mu_n = 0.182 + 0.125\ln(n) \qquad (4.11)$$

where n is the number of tightenings sustained by the fastener.

For the thread friction coefficient, the following equation produced a good fit with the test results:

$$\mu_t = 0.497 + 0.118\ln(n) \qquad (4.12)$$

The values using these equations are compared with the experimental results in figure 4.7. These equations can be used to develop the torque-tension relationship to allow the number of tightenings sustained by the fastener to be considered. Based upon the work completed, the torque-tension relationship (equation 4.1) can be adjusted so that the bolt preload is defined in terms of the applied torque and the friction characteristics. For EZP fasteners, the relationship becomes:

$$F = \frac{T}{\left[0.578\mu_t d_2 + 0.159p + \dfrac{D_e \mu_n}{2}\right]} \qquad (4.13)$$

in this case: $\mu_t = 0.497 + 0.118\ln(n)$ *and* $\mu_n = 0.182 + 0.125\ln(n)$

A comparison between the experimental results and the results is shown in table 4.2.

Table 4.2 presents a summary of the results and a comparison between measured and predicted bolt preloads. Average values are quoted.

Tightening Number	Mean Experimental Results		Values based upon mathematical model		Actual Mean Bolt Preload	Predicted Bolt Preload	Percentage Error of Predicted versus Actual Preload
	Nut Face Friction μ_n	Thread Friction μ_t	Nut Face Friction μ_n	Thread Friction μ_t	newtons	newtons	
1	0.164	0.430	0.182	0.497	18722	16556	11.5%
2	0.263	0.579	0.269	0.579	13363	13262	0.8%
3	0.341	0.678	0.319	0.627	11080	11892	7.3%
4	0.383	0.717	0.355	0.661	10261	11074	7.9%
5	0.402	0.729	0.383	0.687	9972	10515	5.4%
6	0.388	0.729	0.406	0.708	10116	10100	0.2%
7	0.400	0.752	0.425	0.727	9814	9768	0.5%
8	0.418	0.742	0.441	0.742	9722	9512	2.2%
9	0.489	0.685	0.456	0.756	9492	9285	2.2%
10	0.449	0.713	0.470	0.769	9655	9083	5.9%

Table 4.2 – Comparison of Actual versus Predicted Friction Coefficient and Bolt Preload Values

4.5.2 SEM Investigation

A series of inspections was conducted using a scanning electron microscope[1] to observe the surfaces in the contact between re-tightening operations. In this series of tests the nut was tightened against a hardened steel washer without EZP. Scanning electron microscopy of bolts was conducted to develop some understanding of the effects leading to changes in the friction coefficients. Interest was centred on the threaded region of the bolt since previous tests had indicated that the largest changes in the friction coefficient were occurring in the threads. After the bolt had been tightened once and then released, surface damage could be observed on the pressure flanks of the thread. The two photographs shown in figures 4.8 and 4.9 show the condition of the thread flanks after one tightening. As can be seen in the higher magnification photo, region (a) is where zinc has been transferred from the nut thread onto the bolt thread. Region (b) shows the coating fractured with partial removal of some of the zinc. Region (c) shows evidence of abrasive wear, possibly due to a wear debris particle and region (d) is a wear particle left on the surface of the thread.

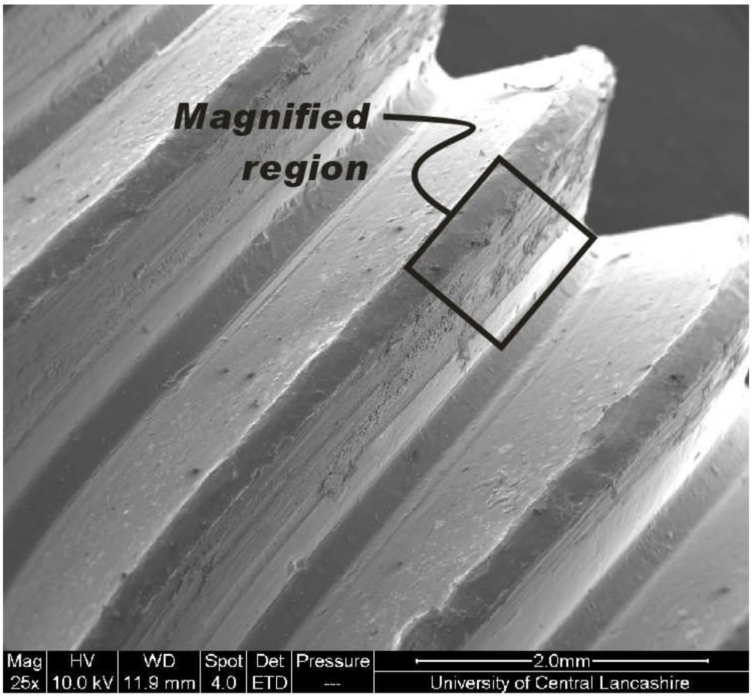

Figure 4.8 Thread Tightened Once - 25x Magnification – Rectangular region magnifed in figure 4.9

[1] A FEI Quanta 200 Scanning Electron Microscope was used in these tests.

The thread coefficient of friction was considerably higher than the nut face friction and showed a decreasing trend as the preload increased.

After five tightening and untightenings, significant wear and surface damage had occurred on the pressure flanks of the threads. Figure 4.10 shows the surface of the pressure flank surface of the thread after 5 tightenings. Figure 4.11 shows a magnified view of the surface exhibiting the zinc becoming detached from the steel substrate that may have occurred as a result of adhesion effects. The nut face surface after the first tightening was relatively smooth, by the fifth tightening, debris and wear particles can be observed on the surface. However, the surface was in a better condition than that of the thread flanks.

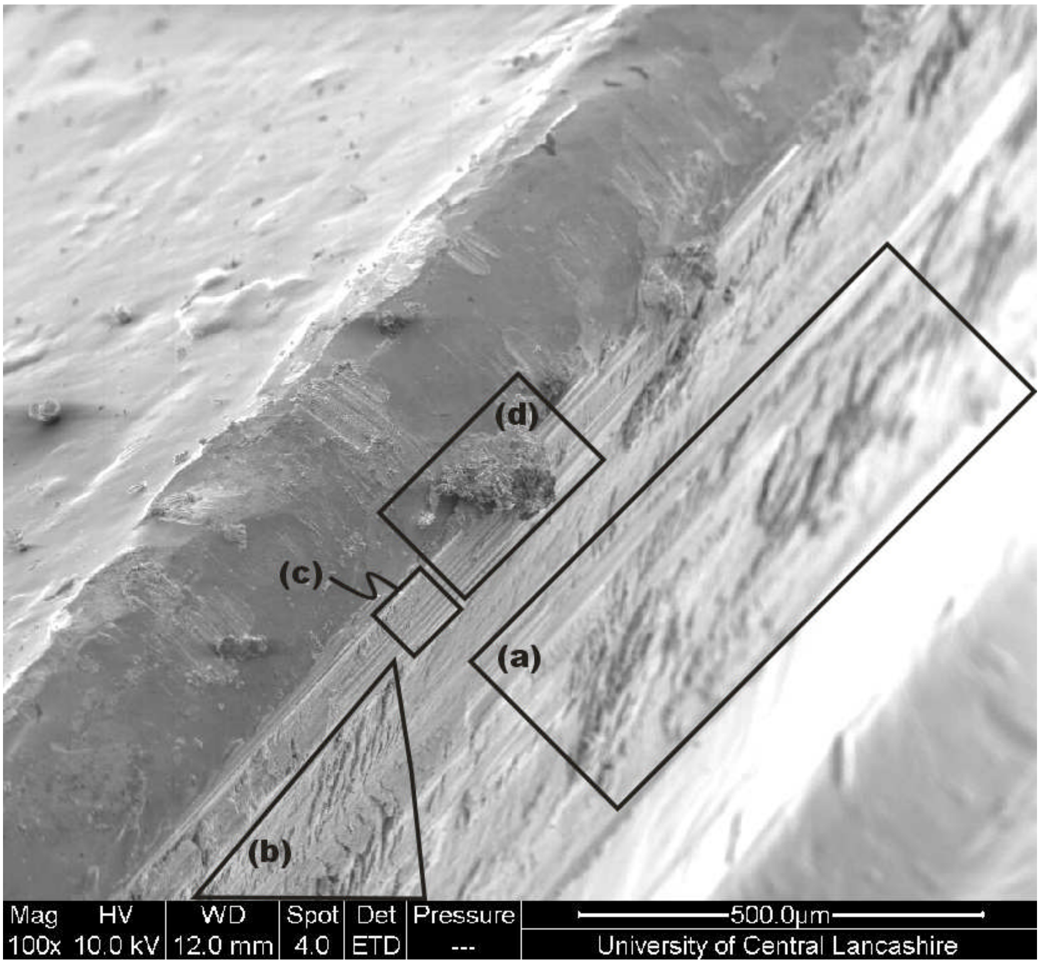

Figure 4.9 Thread Tightened Once - 100x Magnification

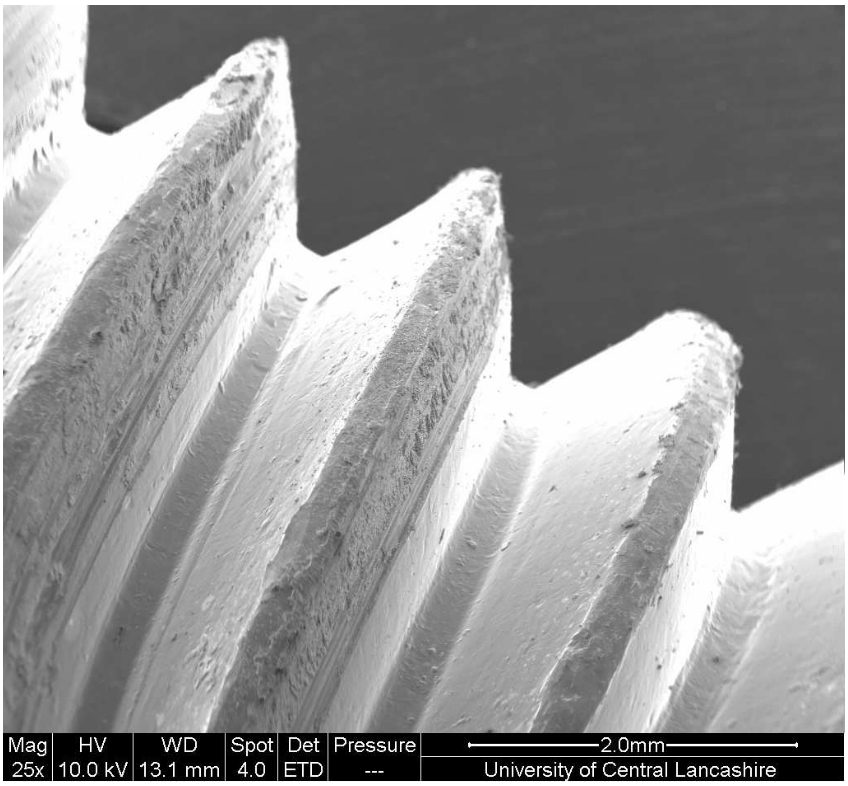

Figure 4.10 Thread Tightened Five Times - 25x Magnification

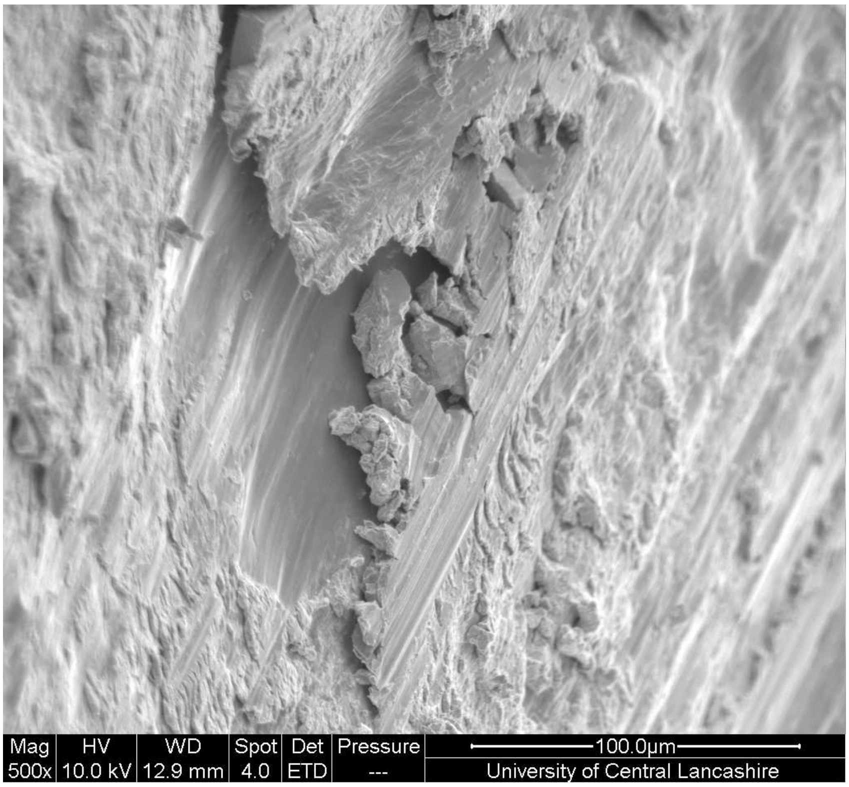

Figure 4.11 Thread Tightened Five Times - 500x Magnification

After ten tightenings and untightenings, severe wear had occurred on the threads; this is shown in figure 4.12. The flank of the thread closest to the joint (region A of the thread in figure 4.12) is the highest loaded in the assembled joint and shows severe scoring, threads further into the nut show less severe wear patterns (B), but evidence can be seen of wear particles being broken away from the surface (C). The thread closest to the joint surface is also distorted.

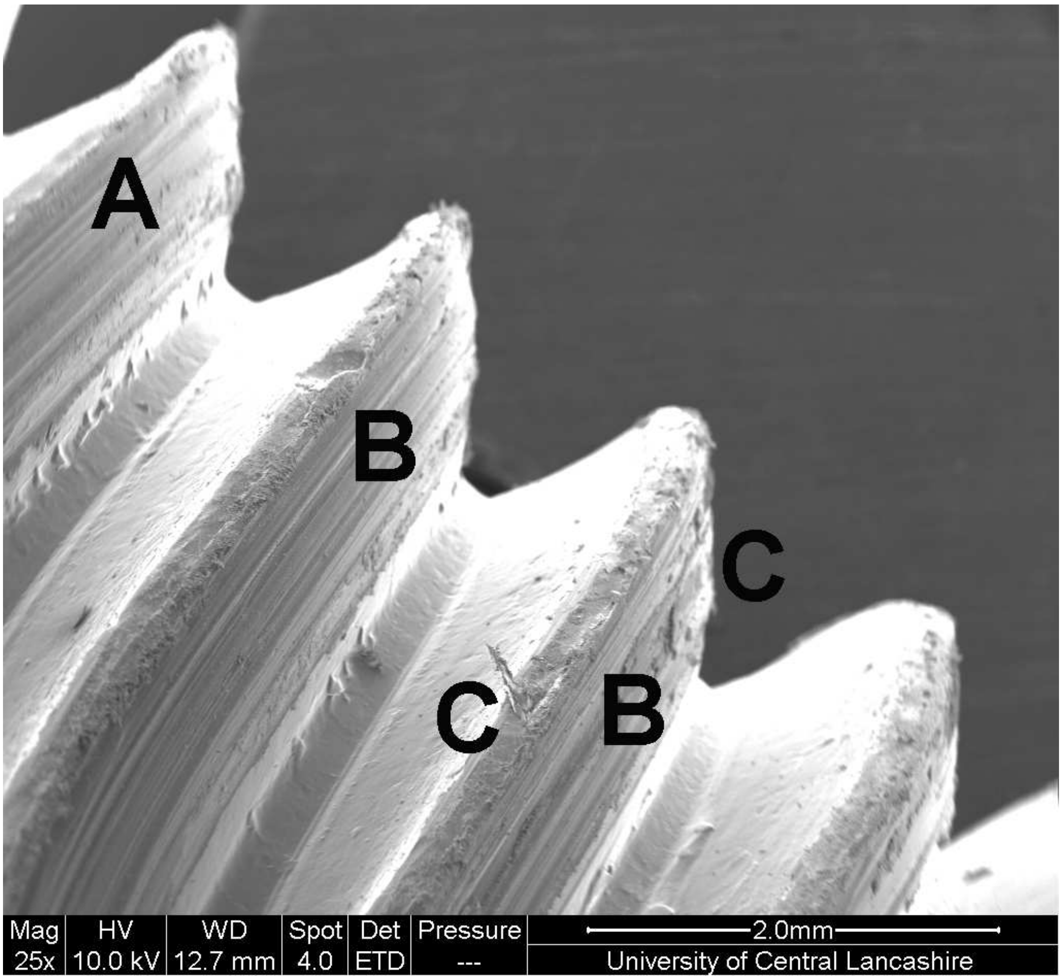

Figure 4.12 Thread Tightened 10 Times – 25x Magnification

4.6 NUT FACE FRICTION COEFFICIENT TESTS UNDER CONSTANT LOAD

Table 4.3 shows the results of the nut face friction coefficient tests. In order to quantify the sliding distance before a significant change in the friction coefficient occurred, the angle at which the friction coefficient became greater than 1.5 times its initial value was measured.

Applied Load (kN)	Initial Friction Coefficient	Angle at which Friction = 1.5 x Initial Value	Sliding Distance based upon mean friction radius (mm)
10	0.13	2655	365
10	0.11	3005	413
10	0.16	595	82
10	0.16	3013	414
10	0.13	1010	139
20	0.105	732	100
20	0.12	990	136
20	0.14	879	121
20	0.11	827	114
20	0.12	1064	146
20	0.18	2250	309
30	0.12	1355	186
30	0.13	1216	167
30	0.12	838	115
30	0.11	1070	147
30	0.13	1450	199
30	0.10	1244	171

Table 4.3 – Nut Face Friction Coefficient Results

The 1.5 factor was chosen based upon study of the angle – friction coefficient graphs so that breakdown in the condition of the friction surfaces could be distinguished from the normal variability of the friction coefficient. This angle indicating surface breakdown is documented in the table together with the sliding distance based upon the mean radius of the bearing surface of the nut. The nominal bearing pressures under the face of the nut were 73.5 MPa, 147 MPa and 220.5 MPa for the 10 kN, 20 kN and 30 kN loads respectively. The average sliding distance prior to a significant change occurring in the friction characteristics was 195 mm with a standard deviation of 105 mm.

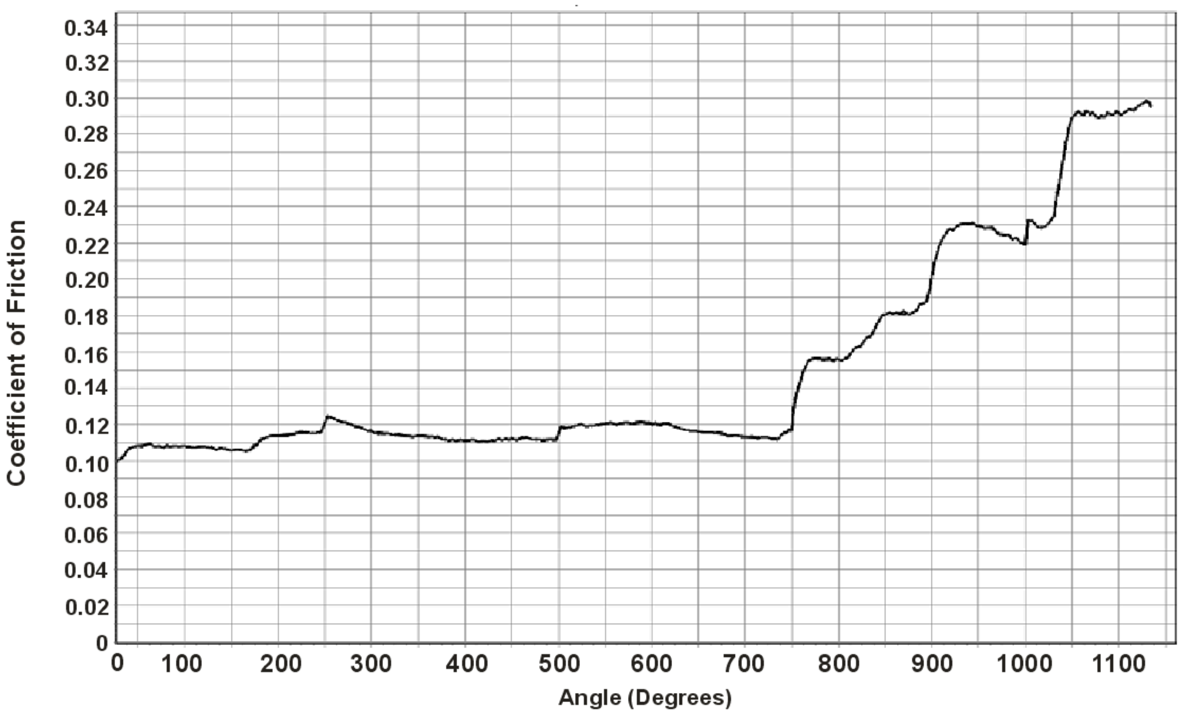

Figure 4.13 – Graph Showing the Onset of Surface Failure – 20 kN Load

Referring to table 4.3, under this test it was found that initial friction coefficients were reasonably stable i.e. the friction did not change significantly as the nut rotated. However after a critical angle of rotation, a transition point was reached when the friction coefficient started to increase, leading typically to a tripling in its value. A graph illustrating how the friction varied with the angle of nut rotation is shown in figure 4.13. The mean initial value of the friction coefficient was 0.13 with a standard deviation of 0.02.

4.7 DISCUSSION AND CONCLUSIONS

Based upon the experiments completed and the measurements made, a change in the friction between the contact surfaces of the bolt/nut thread and nut face occurs upon repeated tightening. This breakdown of these surfaces resulting from fracture and partial removal of the coating results in an increase in the friction coefficient present which causes a reduction in the clamp force provided for an assembly when tightened to a specified torque value. This reduction in the clamp force is significant, the clamp force on the sixth re-tightening typically being half that obtained on the first tightening which could have an adverse effect on the structural integrity of the assembly.

The process of the friction increasing to a peak value prior to falling to a stable value is achieved is described by Suh and Sin (1981). The process is dependent upon the distance slid by the two surfaces and is equated, in the re-use of fasteners, with the number of times the fastener has been tightened. This change in the friction conditions is considered to be due to damage to the contact surfaces resulting from sliding and the high pressures involved. The primary processes for causing the friction variation are the ploughing effect of trapped wear particles and adhesion. Wear particles trapped between the surfaces results in ploughing by the entrapped particles further increasing friction. Levelling off of the frictional resistance occurs as the number of particles entrapped between the surfaces becomes constant.

Higher levels of friction were noted for the threads than were present under the nut face. This is thought to be due to a number of factors. Thread friction is somewhat complicated by the non-linear pressure distribution on the thread surfaces and that this non-linearity is itself partially a function of the friction value (Sopwith, 1948). The higher pressures being present in the thread that leads to increased wear particle generation and ploughing. The enclosed nature of the mating threads also results in wear particles created at the surface closest to the joint face, which is under the highest pressure, being moved further up the thread as the nut rotates. Work hardening is occurring due to thread deformation (that can be seen in figure 4.11). Under such circumstances, besides surface friction there will also be a component of the overall friction force due to work deforming the thread (Bellemare et al. 2008). The thread contact geometry and the elastic-plastic properties of the steel will influence the magnitude of this effect. A third factor that will influence the friction force measured is due to distortion of the thread flanks as the tightening process progresses. An implicit assumption in the determination of the friction coefficient is that the flank angle is a constant 60 degrees.

The distorted pressure flank that can be seen in the SEM photographs will increase the frictional force even if the friction coefficient remained constant.

Although the magnitude of the thread friction coefficient is higher than that present under the nut face, the friction becoming approximately constant after five or six tightenings is similar to the effect observed for the nut face friction.

It was also noted that the magnitude of the thread friction coefficient became dependent upon the pressure (clamp force) as the number of tightenings increased. The mechanism for this effect is not completely understood, but is thought to be due to the increase in pressure causing a change from wear particle ploughing to adhesive friction.

Tests completed to determine the nut face friction coefficient under a constant loading clearly indicated that a change occurs in the friction characteristics after a certain amount of angular movement. By using the mean nut radius, this angular movement was converted to a linear movement. The average distance prior to a significant change in friction occurring was 195 mm. The initial average friction coefficient of 0.13 is less than the mean value noted in the tightening tests (mean value on first tightening of 0.19). However the scatter in the results on the tightening tests observed on the nut face tests under constant loading was less than that observed in the full tightening tests.

Based upon these results, the reuse of unlubricated EZP threaded fasteners cannot be recommended. The increase in friction resulting from reuse could have an adverse effect on a joint’s structural integrity due to the corresponding decrease in the generated clamp force that occurs.

5. LOOSENING OF FASTENERS BY TRANSVERSE VIBRATION

5.1 INTRODUCTION

One reason why bolted joints suffer integrity problems is due to self-loosening of the fasteners. Junker (1969) and other researchers have shown that transverse vibration could fully loosen non-locked threaded fasteners. Frequently the Engineering Designer specifies a lock nut to attempt to prevent such loosening. The term 'lock nut' is generic and is usually used to describe any nut that incorporates some feature to prevent self-loosening. The vast majority of lock nuts do not fully lock the nut to the bolt and will sustain some degree of self-loosening under transverse vibration. In this study the loosening characteristics of a popular form of lock nut are investigated. Prevailing torque nuts have a locking feature in which a small torque (the "prevailing torque") is needed to rotate the nut down the thread of an untightened bolt. A large number of transverse vibration tests have been performed on various types of prevailing torque nuts to investigate their loosening characteristics. Further tests have been completed to establish the effect on the prevailing torque of re-use of the nut.

5.2 DETAILS OF THE TEST MACHINE

The machine that was used in the experiments conducted by the author is usually described as a Junker machine, after its inventor Gerhard Junker and is described in his major paper on fastener loosening (Junker, 1969). The machine allows tests to be completed to the DIN 65151 standard (Deutsche Norm, 1994). An outline of the machine is shown in figure 5.1. The test machine was obtained from SPS Technologies. The machine required some renovation to make it operative. This included the provision of new instrumentation, hardened bearing plates and adapters. The instrumentation was connected to an analogue to digital converter so that the results could be recorded directly onto a computer. Software was written (in the Delphi language by the author) to allow the results to be displayed graphically.

Figure 5.2 shows the test arrangement when the loosening characteristics of a nut is to be tested. The machine has a fixed frequency of transverse motion of 12.5 Hz and provides a displacement amplitude of +/- 0.65 mm. By the use of a suitable adapter (1), the machine allows screws, nuts or lock washers to be tested. An electric motor drives a cam that induces transverse reciprocating movement into a top plate (2), the top plate rests upon needle roller bearings (3) that rests on a fixed support (4) so that friction is minimised. The test fastener passes from an adapter (1) within the machine and through a fastener bearing plate (5). The adapter (1) clamps a load cell (6) to allow the fastener clamp force to be measured during tightening and testing. When a screw is being tested an adapter with a tapped hole is used. When nuts are being tested the bolt (7) passes through the adapter so that the nut (8) is on top of the machine. If desired the bolt can be prevented from rotating by placing a socket through the adapter and onto the bolt head.

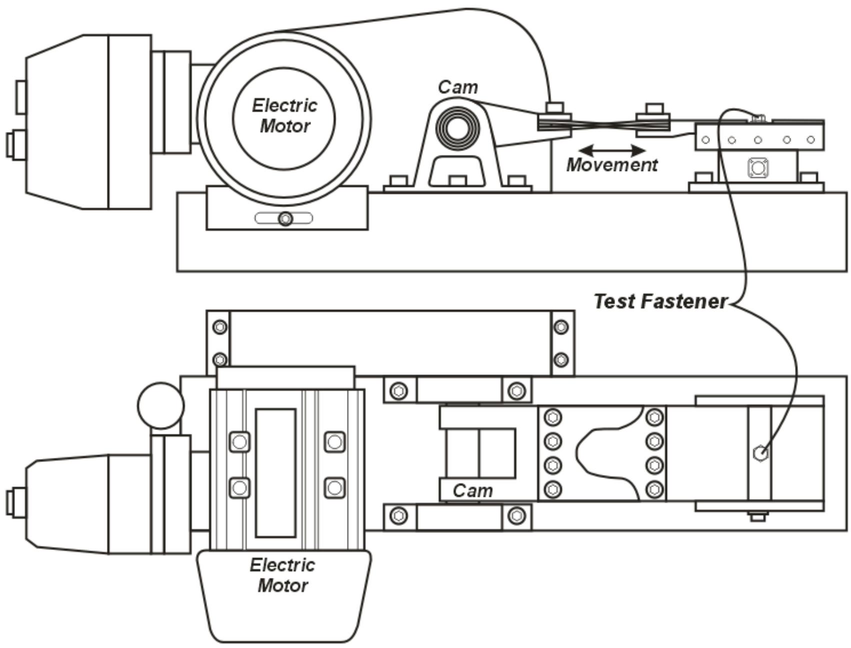

Figure 5.1 Diagram of the Junker Machine used in the Experiments

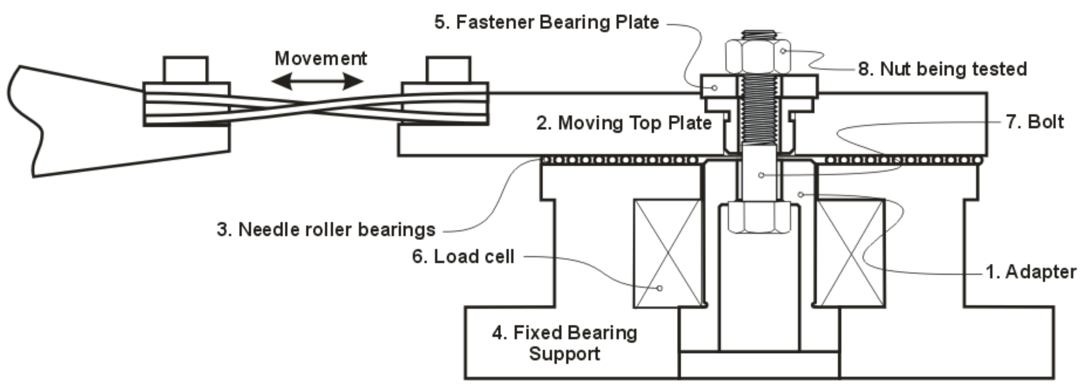

Figure 5.2 Section through the test machine showing the arrangement for testing a nut.

A typical test would involve tightening the nut so that there was a specific preload in the bolt. A specific preload rather than a tightening torque was used to allow the same starting point to be used to assist in comparing different types of fasteners.

5.3 LOOSENING CHARACTERISTICS OF VARIOUS TYPES OF FASTENERS

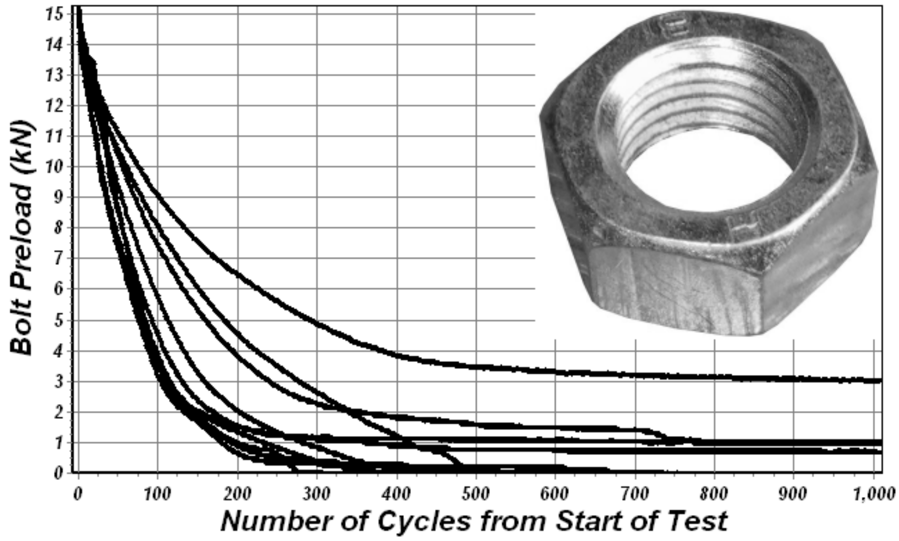

Figure 5.3a Plain nut

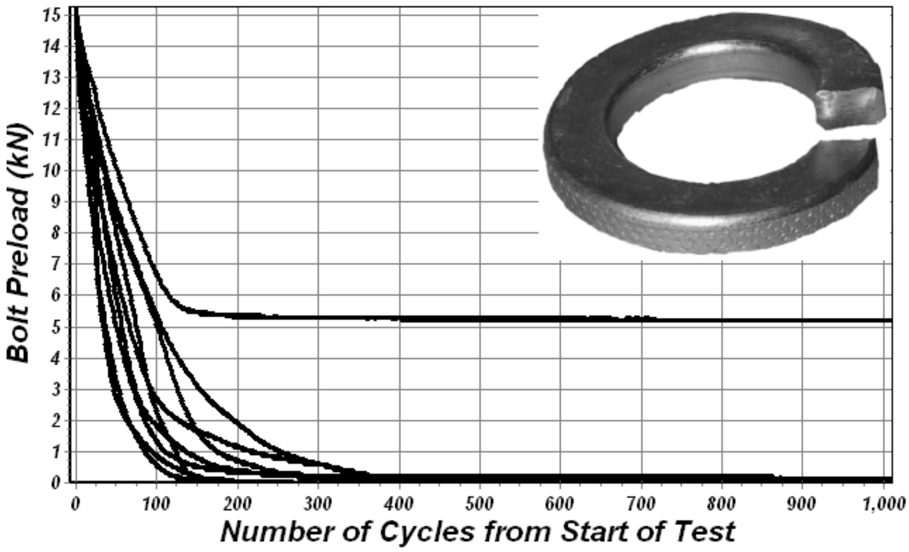

Figure 5.3b Plain nut with a helical spring washer

Figure 5.3 Preload decay curves for a plain nut and plain nut with a helical spring washer

5.3.1 Introduction

Transverse vibration tests were completed on a number of different types of locking devices in order to assess their effectiveness and to develop an understanding of the loosening process. A preload of 15 kN was used since this value is typical of what would be used in practice for the M8 fasteners used in the tests. The test duration of 1000 test cycles was selected since based upon experience the preload had usually stabilised by this number of cycles. That is, usually the nut had come completely loose or the preload stabilised to a specific value. The DIN 65151 standard (Deutsche Norm, 1994) states that the aim of the test is to allow a comparative evaluation of various locking elements under defined test conditions, the duration can be greater or less than 1000 cycles so long as the conditions are the same for each fastener. By measuring the bolt preload during the test a graph of preload versus time or preload versus number of cycles can be constructed that is referred to as a preload decay curve. For the ideal fastener, the preload decay curve would be a horizontal line indicating that no preload would be lost during transverse slip of the joint. However this is generally not the case even with fasteners that do not self-loosen during the test, some preload loss occurs due to embedding.

5.3.2 Plain Nuts with and without helical spring washers

A series of tests were completed on plain nuts having an electro-zinc plating finish to allow a basis to judge the effectiveness of various types of locking devices. Tests were also completed on plain nuts with helical spring washers. The preload decay curves are shown in figures 5.3a and 5.3b. New fasteners were used in each test, a total of 10 tests were performed for each graph.

It should be noted that on tests with and without the washer fitted, the nut usually comes completely loose but there are occasions when a residual preload can be retained. A measure of the rate of loosening can be established by determining the slope of the straight-line portion of the loosening curve. A boxplot of the results is shown in figure 5.4. These results confirm the visual interpretation from the preload decay curves, that nuts with helical spring washers self-loosen faster than plain nuts by themselves.

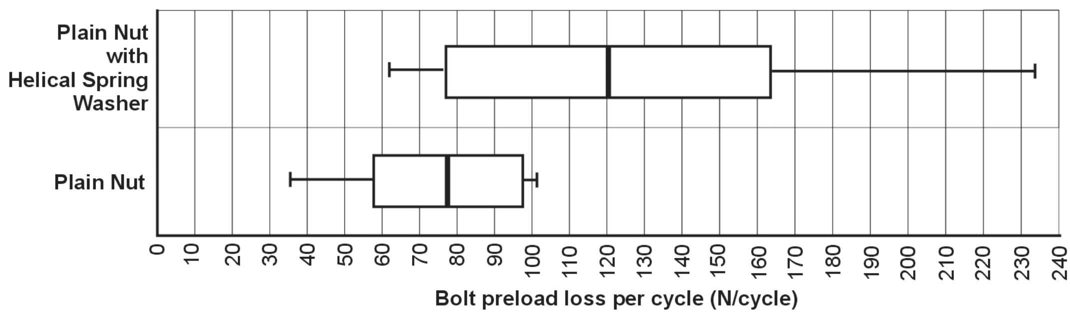

Figure 5.4 Boxplot comparing the slope of the preload decay curves

In the case of the plain nut, it is friction alone that prevents the nut from loosening. When the nut is self-loosening, patches of higher friction on the thread or nut face interface can arrest the loosening process. One reason for the poor performance of the washers is that the tang of the washers failed to bite into the nut surface. It was first reported by Junker that helical spring washers are ineffective in preventing loosening (Junker, 1969). However they are still used today in many applications. Efforts have been made in recent years to prevent their use by removal of the standards that define their dimensions (Esser and Hellwig, 2004).

5.3.3 Tests on prevailing torque nuts

Prevailing torque nuts are commonly used to try to prevent self-loosening and/or detachment of threaded fasteners. This type of nut was developed over one hundred years ago (Johnson, 1877) and have the essential characteristic that a torque is needed to rotate the nut down the thread of an untightened bolt. One advantage of this type of nut is that the locking feature can be verified at the time of assembly by measuring the prevailing torque. This type of nut is in extensive use across most industries and is probably the most common type of locking device for threaded fasteners.

There are many varieties of such nuts, but in general they can all be classified into one of two categories, those with a non-metallic insert or those which are all-metal. Non-metallic insert nuts typically generate a prevailing torque by incorporating a polymer insert in the top of the nut that is deformed by the bolt thread. The all-metal variety achieves the prevailing torque by one of the following methods:

a) By distorting the threads at the top of the nut.
b) By introducing slots and then deforming them
c) By making the top threads elliptical shaped.
d) By introducing spring steel inserts.

Tests were completed on a range of prevailing torque nuts commonly available in the U.K. The preload decay curves are presented in figure 5.5, typically five or more tests on each type of fastener were performed. This type of nut does not prevent self-loosening. However under purely transverse vibration, this form of nut retains a small amount of residual preload, in general. In 2004, DIN published a document that categorised prevailing torque nuts as loss prevention devices i.e. the nut may come loose but it would be retained on the bolt (Esser and Hellwig, 2004).

In general, the results between the various types of prevailing torque nuts are similar. The nut initially rotates when subjected to the transverse vibration with a significant amount of preload being lost prior to an asymptotic preload value being reached. The prices of the nuts differ substantially, a non-metallic insert nut being approximately 1.5 times the cost of a plain nut. The price of all-metal prevailing torque nuts varies between 20 to 55 times the cost of a plain nut depending upon the make.

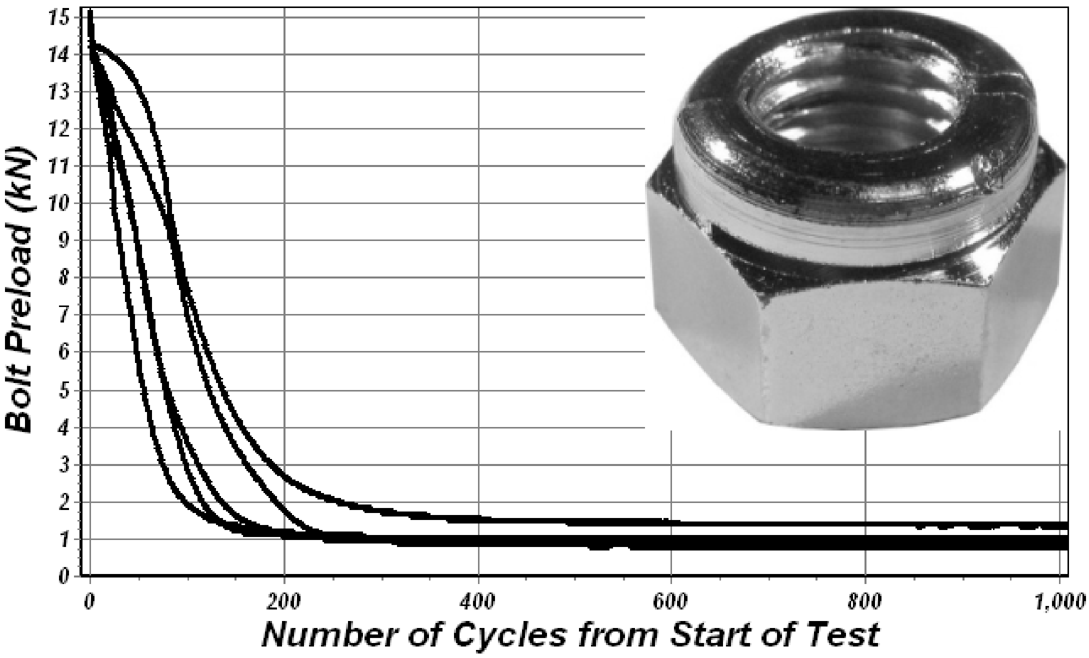

Figure 5.5a Preload decay curves for the Aerotight nut

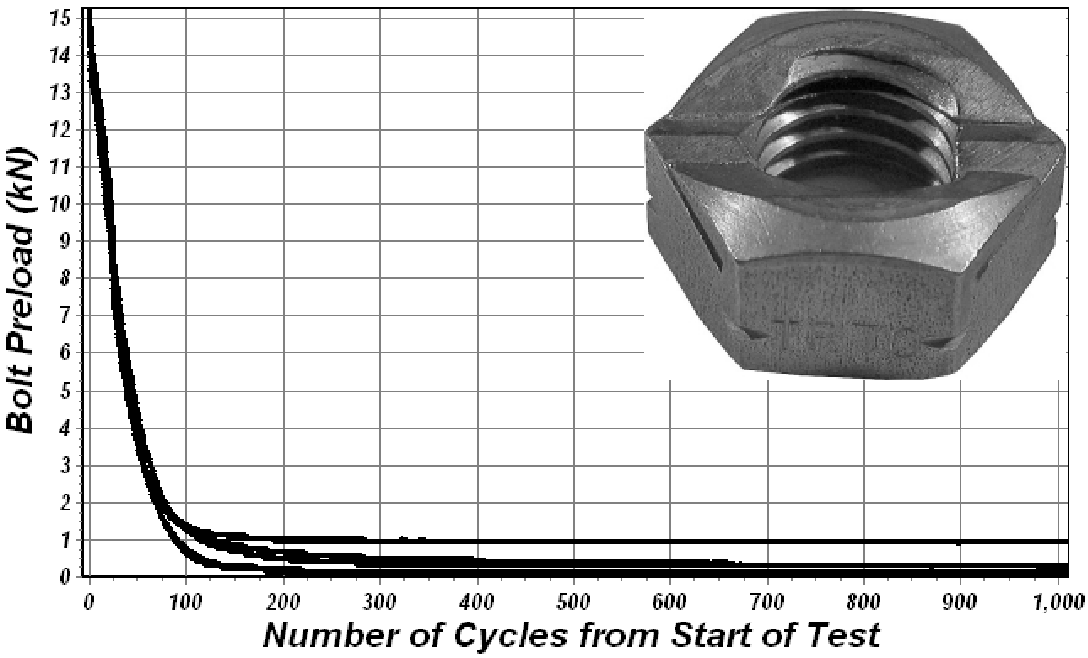

Figure 5.5b Preload decay curves for the Binx nut

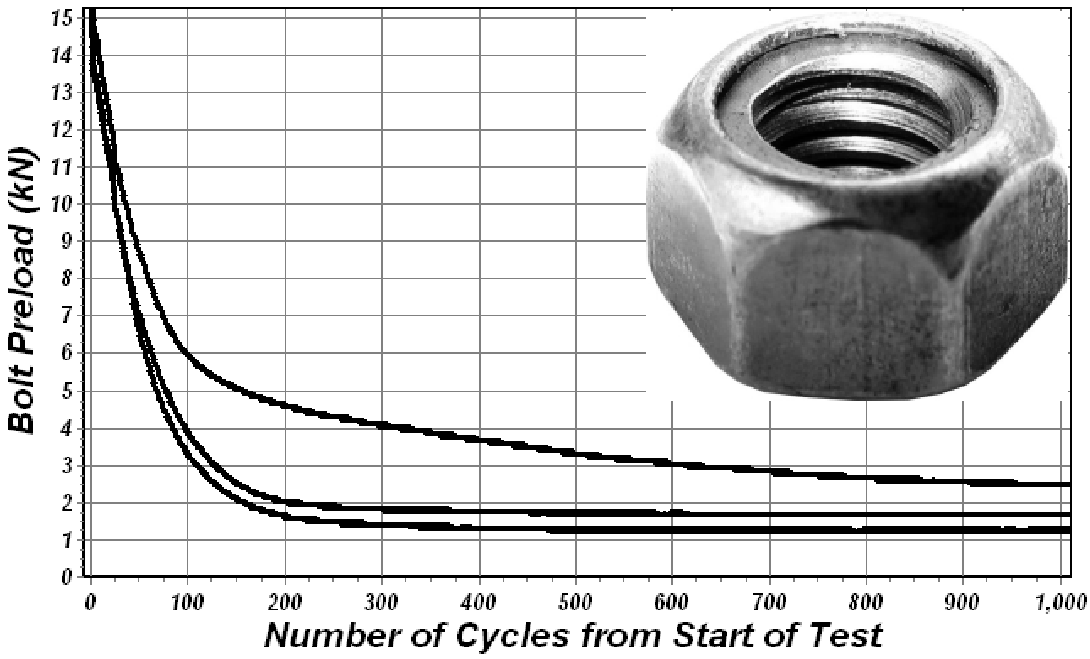

Figure 5.5c Preload decay curves for the Flaig and Hommel nut

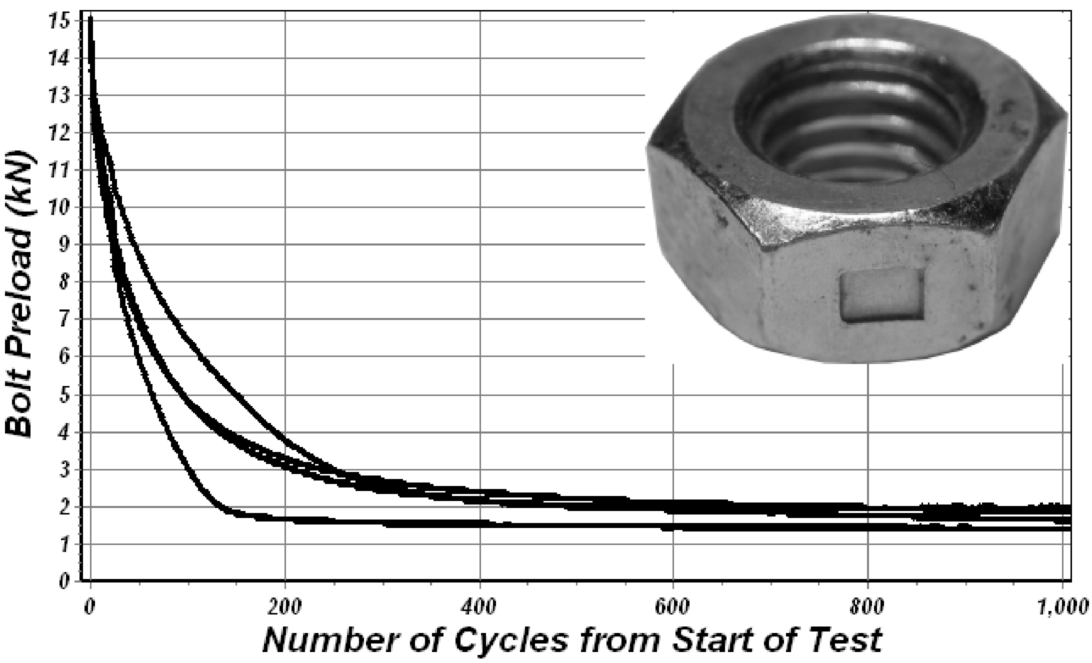

Figure 5.5d Preload decay curves for the Biway nut

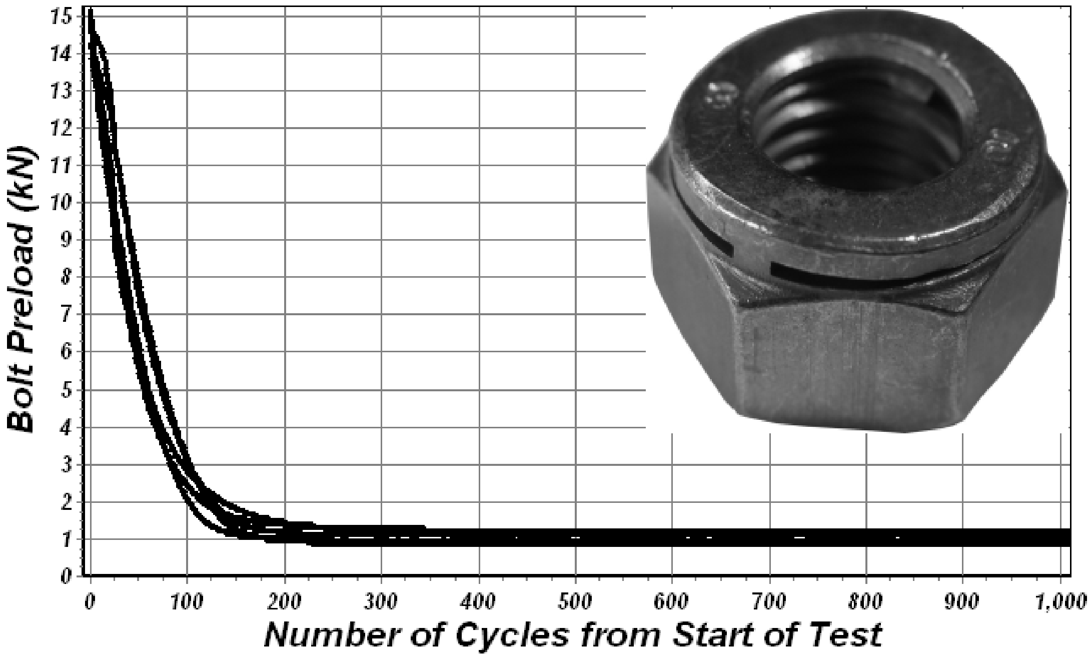

Figure 5.5e Preload decay curves for the Philidas Industrial nut

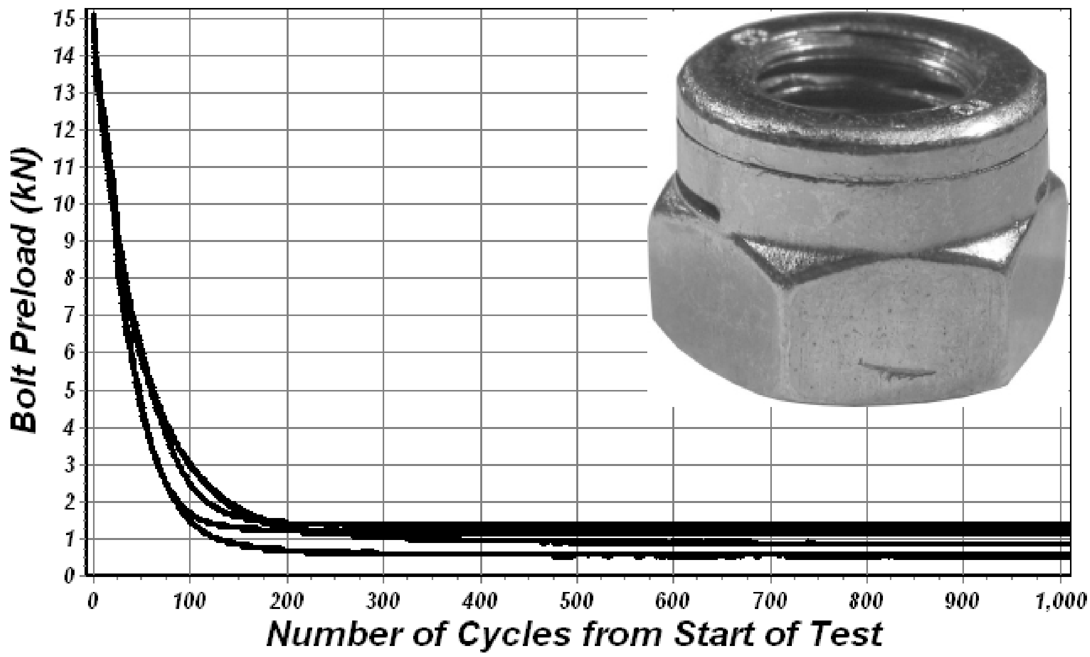

Figure 5.5f Preload decay curves for the Philidas Turret nut

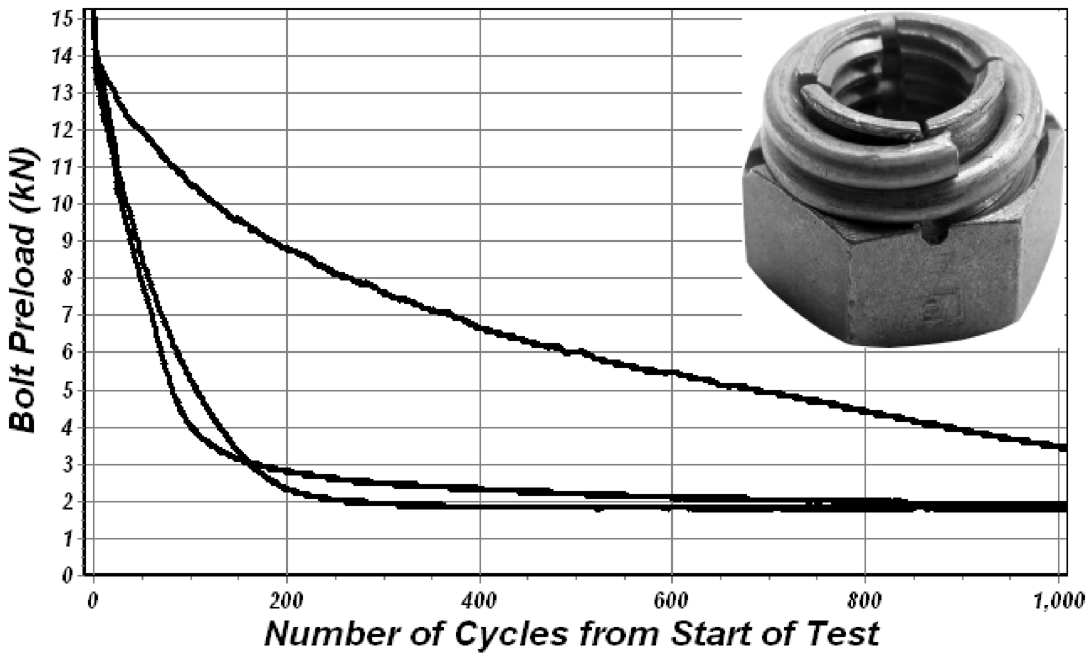

Figure 5.5g Preload decay curves for the Vargal nut

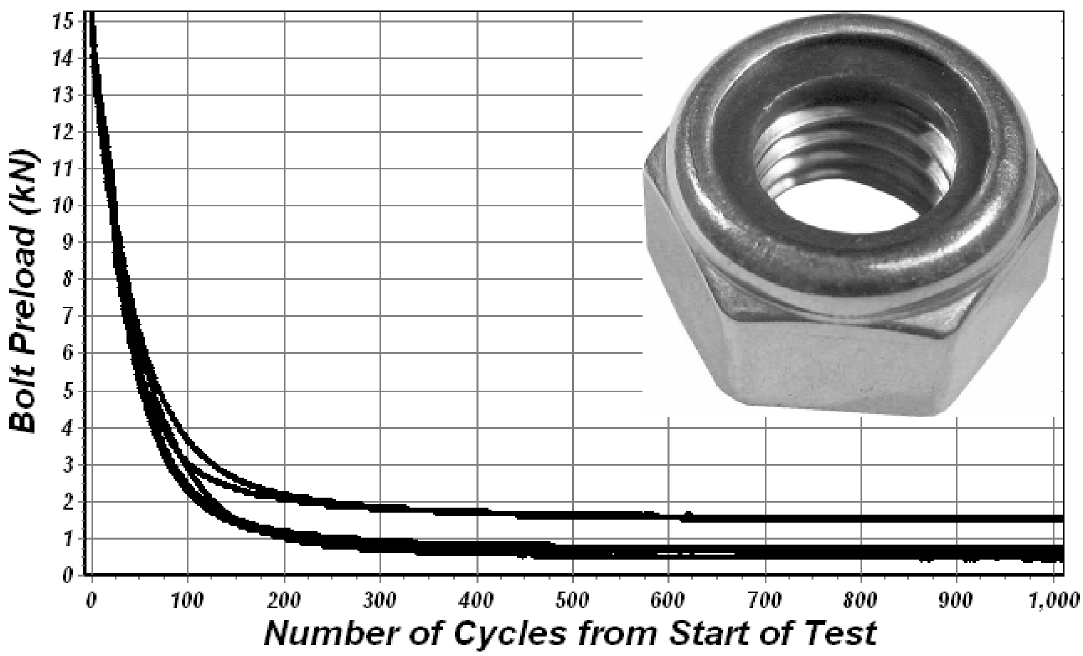

Figure 5.5h Preload decay curves for the Nylon insert nut

5.4 THE EFFECT OF LOOSENING ON THE PREVAILING TORQUE

5.4.1 Introduction

It has been noted in literature that the prevailing torque can decrease with re-use (Sase et al. 1996) with non-metallic insert nuts. The performance requirements for such nuts are specified in the ISO 2230 standard (British Standards, 2008). The standard stipulates that a minimum level of prevailing torque, for a given thread size, must be maintained for up to five re-uses. A number of manufacturers of such nuts claim that the nuts are re-useable, with some industries, such as on railway track, routinely re-tightening them if/when they loosen (RAIB, 2008).

The standard gives a detailed test procedure that should be followed to assess the retention of the prevailing torque when re-used. The test involves tightening the nut on a bolt and measuring the prevailing-on torque and subsequently tightening to a specific preload before releasing the load and measuring the prevailing-off torque. The nut is then rotated down the bolt a further four times with the fifth removal prevailing-off torque being measured. The standard imposes a maximum value for the first installation prevailing-on torque and a minimum value for the first and fifth removal prevailing-off torque. For the M8 nuts tested in this series of tests, the maximum value for the first installation prevailing-on torque is 6 Nm, the minimum value for the first removal prevailing-off torque is 0.85 Nm and 0.6 Nm for the fifth. (The prevailing-on torque is the prevailing torque in the tightening direction and the prevailing-off torque is the prevailing torque in the loosening direction.)

The author speculated that there would be a difference between the performance of nuts when tested to the ISO procedure compared to nuts that self-loosened as a result of transverse vibration. This matters since the procedure is used as a measure of the resistance to self-loosening of the fastener and if this assumption is flawed then joint integrity may be an issue in some applications. Self-loosening, which happens in practice, should be more severe in that many repetitions of relative thread movement will occur during the loosening process that lead to wear of the prevailing torque locking arrangement in the nut. To assess if this was the case, a series of tests was completed on the Binx all-metal prevailing torque nut and the non-metallic insert prevailing torque nuts. Tests were completed both to the standard procedure and to a test involving the nuts being self-loosened due to transverse vibration. A light grease was used (Lubriseal) for all the tests so that the thread and nut face friction would not change significantly as a result of re-tightening. This was done so that greater consistency could be achieved with the vibration tests; the test procedure does permit the use of a lubricant. A Norbar 0 – 10 Nm electronic torque transducer was used to complete the torque measurements.

With the self-loosening tests, the nuts were tightened to the preload required by the standard (15.9 kN) and allowed to self-loosen. Any residual preload after 1000 cycles was removed and the prevailing-off torque measured as per the standard. Repeating the loosening test on the same bolt was found to be a problem in some cases as fatigue failure occurred.

5.4.2 Tests on the Binx Nut

The Binx nut is one of the most popular all-metal prevailing torque nuts used in the U.K. For the tests completed to the ISO 2320 procedure 10 nuts were used and 15 nuts were used in the transverse vibration tests. Five bolts failed by fatigue during the transverse vibration tests. The bolt was subjected to substantial bending stresses during the test, cumulative fatigue damage arises by the repetitions involved in this series of tests. The results of the tests are presented in the boxplot in figure 5.6.

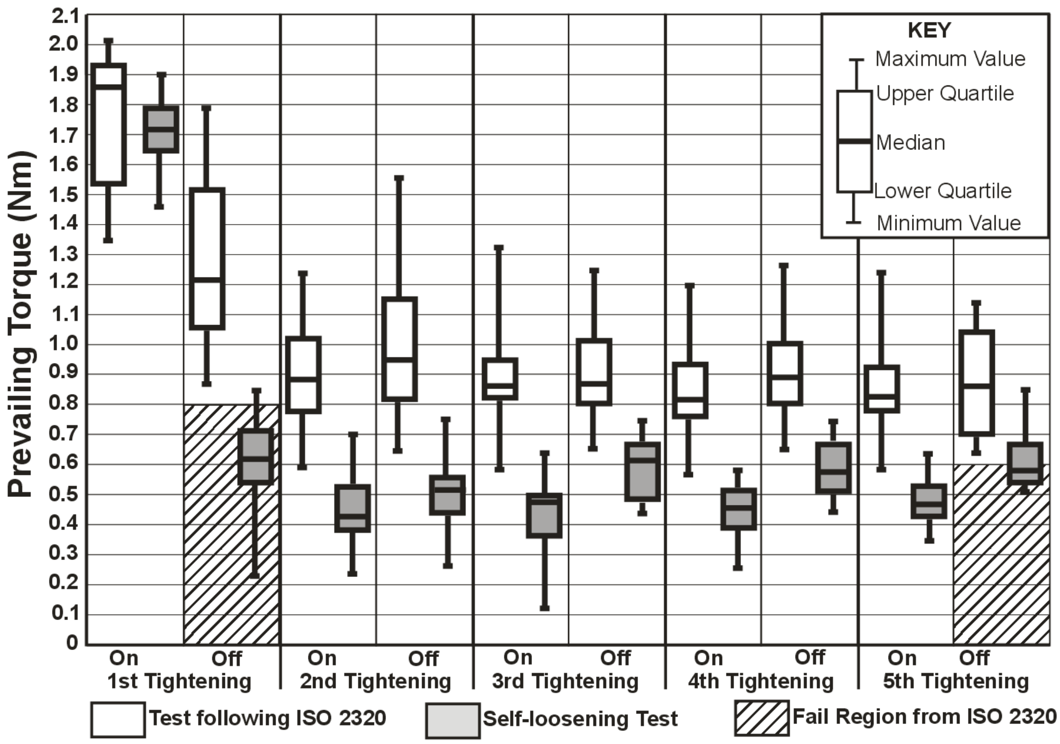

Figure 5.6 Prevailing torque tests on M8 Binx nuts. (Columns marked 'On' refer to the prevailing on torque and columns marked 'Off' to the prevailing off torque.)

As can be seen from figure 5.6, there was a significant drop-off in the prevailing torque after the first tightening, the drop-off being significantly larger for the transverse vibration tests. All the nuts passed the minimum first prevailing-off torque requirement of 0.8 Nm, whereas, with the self-loosening test, all but one failed. Following the first tightening, subsequent tightening and loosening of the nuts produced only a minor change in the prevailing torque.

5.4.3 Tests on non-metallic insert nuts

Non-metallic insert nuts have a plain polymer ring at the top of the nut that generates a prevailing torque when threaded onto a bolt. A series of tests was completed to measure the prevailing torque in the same manner as the all-metal nuts. The results are shown in figure 5.7. In these tests all the nuts passed the minimum first prevailing-off torque requirement of 0.8 Nm. With the self-loosening test, all but one passed.

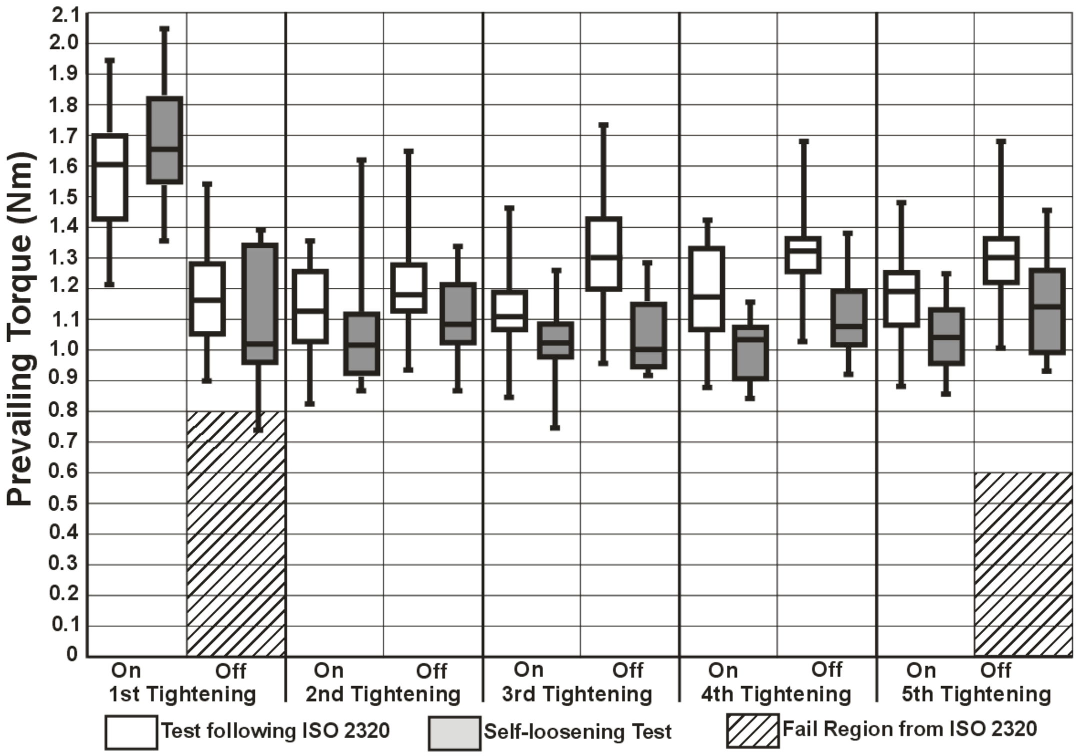

Figure 5.7 Prevailing torque tests on non-metallic insert nuts

5.4.4 Comparison of Test Results from Metallic and Polymer Insert Prevailing Torque Nuts

A key issue in preventing structural integrity problems relates to how well the fastener retains preload when subjected to transverse vibration. The ISO 2230 standard (British Standards, 2008) uses prevailing torque as a locking criteria since it is easy to measure, the retained preload under a transverse vibration test is more closely related to what is required in practice. Figure 5.8 presents a comparison of the residual preload after 1000 transverse vibration cycles between the two types of nuts

Comparing the boxplots of the all-metal prevailing torque nut results (figure 5.6) with that of the non-metallic insert prevailing torque nuts (figure 5.7) it can be seen that:

1. The self-loosening test reduced the prevailing torque to a larger extent than the standard test. This was particularly significant for the all-metal type of prevailing torque nut.
2. The drop-off in the prevailing torque was less with the non-metallic insert prevailing torque nuts than with the all-metal type for both the standard and the self-loosening tests. The reason for this is uncertain but it may be related to how the prevailing torque of the nylon insert being less affected by vibration than the all-metal type of nut.
3. The reduction in the prevailing torque was less significant than had been reported in a previously published study (Sase et al. 1996) when both types of nut were subjected to repeated loosening tests. This could be due to differences in the detail design of the nuts.

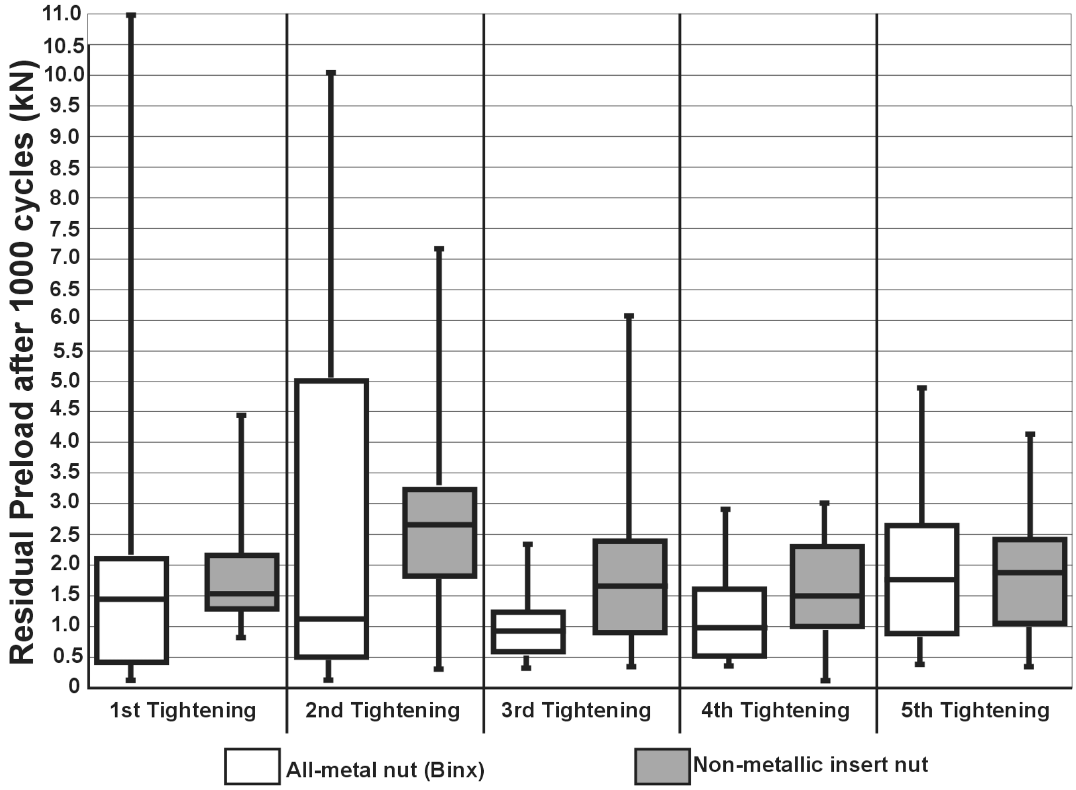

Figure 5.8 Residual bolt preload after 1000 transverse vibration cycles for two types of prevailing torque nuts

Figure 5.8 shows the scatter that was encountered in the residual preload after 1000 cycles is larger for both types of nuts. As can be seen from the diagram, the scatter was significant for both types of nuts. The scatter is thought to be due to frictional variation and thread defects being encountered as the nut rotated on the bolt thread. The author believes small thread nicks can result in the rotation being halted.

Based upon these tests on two types of prevailing torque nuts, no evidence was found that the prevailing torque reduced to zero as a result of re-tightening. Self-loosening was found to be a more severe case than the ISO 2230 test in terms of the reduction in the prevailing torque that occurs with re-use. This was the case for both types of nuts but in particular for the all-metal type of nut.

6. COMPLETE SELF-LOOSENING OF PREVAILING TORQUE FASTENERS

6.1 INTRODUCTION

As discussed in chapter 5, prevailing torque type nuts are commonly used to try to prevent self-loosening and/or detachment of threaded fasteners. There have been instances when new prevailing torque nuts that still retain a prevailing torque have become detached from assemblies. One example relates to the use of a prevailing torque nut fitted to a control rod on the elevator control system on a Tupolev 154M operated by China Southwest Airlines[2]. Over time the nut self-loosened and became detached from the bolt. As a direct result of this loss, on the 24th February 1999 whilst approaching the coastal city of Wenzhou, the aircraft suddenly pitched down and crashed resulting in the death of all the 61 people on board. The causes of such failures have not been understood since the standard method of test, the Junker DIN 65151 (Deutsche Norm, 1994) test, has been unable to reproduce such complete loosening.

Goodier and Sweeney (1945) followed by Sauer et al. (1950) showed that dynamic axial loading would only loosen nuts by at most a few degrees of rotation. Junkers' work (1969) showed that transverse vibration could completely loosen fasteners. Subsequent researchers have built upon this early work but to date have not demonstrated the precise conditions that can lead to the detachment of prevailing torque nuts.

Based upon the study of problematic joints in service, the author speculated that axial loading combined with transverse movement might induce complete self-loosening and subsequent detachment of prevailing torque nuts. The present standard test method (DIN 65151) based upon purely transverse vibration is not able to replicate some

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[2] See http://aviation-safety.net/database/record.php?id=19990224-0

service experience for prevailing torque nuts. Combined axial and shear loads on joints are common in many applications such as for example the wheel to hub joints on a road vehicle (shear loads can be imposed onto the joint by braking and axial loads from cornering).

6.2 MODIFICATIONS MADE TO THE TEST MACHINE

The Junker machine shown in figures 5.1 and 5.2 was modified to allow an axial load to be imparted into a bolted joint whilst simultaneously applying cyclic transverse displacement. This was achieved by using miniature hydraulic cylinders to impart the axial load into the joint with the oil pressure being used to regulate the magnitude of the loading. The arrangement is illustrated in figures 6.1 and 6.2.

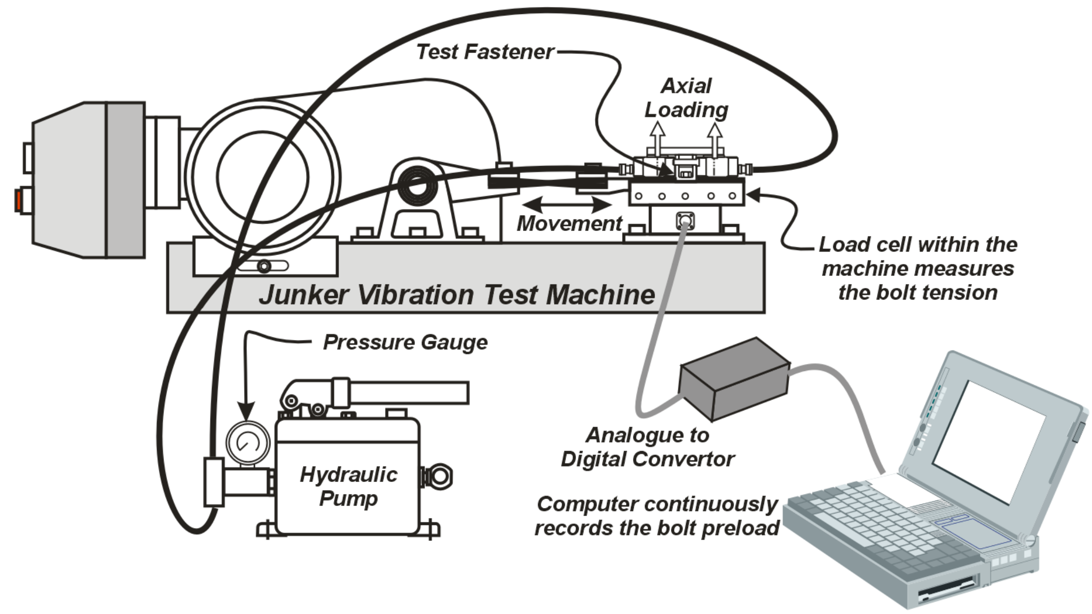

Figure 6.1 – Overall view of the test machine

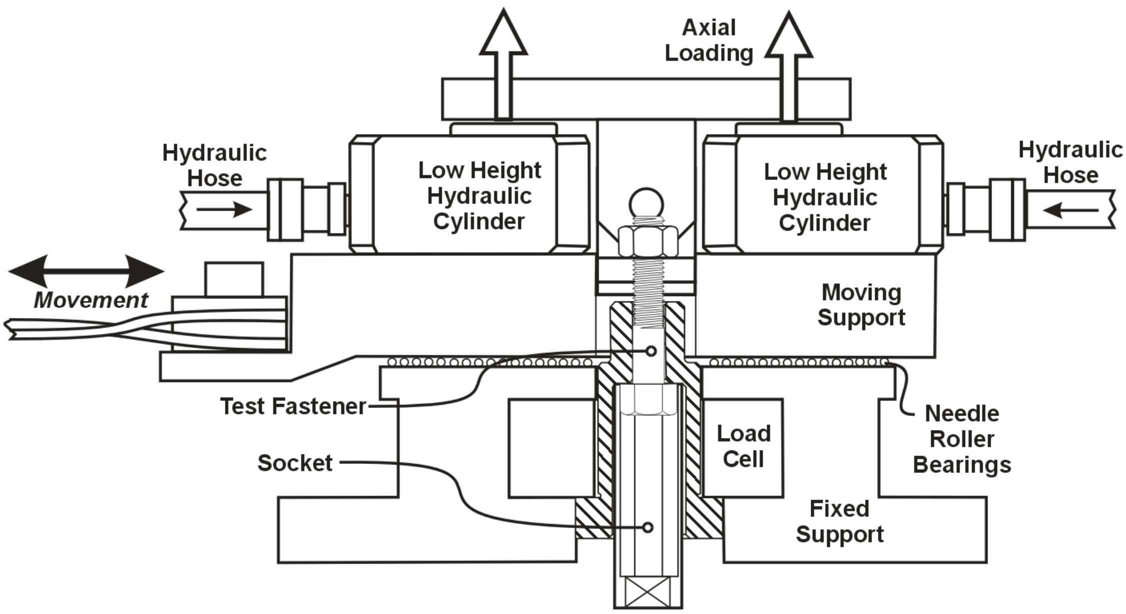

Figure 6.2 Section through the test machine

A new moving support was made which had a deep recess where the bolt was located. This allowed an assembly to be fitted that transferred the load imparted from the hydraulic cylinders into the bolt being tested. The hydraulic cylinders were secured to the moving support so that the whole load assembly moved back and forth during the test.

The tests involved tightening M8 electro-zinc plated bolts to a nominal preload of 15 kN prior to the start of transverse motion. An axial load was then applied to the joint, the axial load being smaller in magnitude than the preload. Due to the mechanics of the way the applied axial load is sustained by the joint, this caused the bolt load to be increased only marginally. This is due to the joint and bolt forming a system of balanced springs. The bolt acts as a tension spring and the joint, a compression spring, the tensile and compressive loads balancing each other. The bolt is stretched by only a small additional amount, and hence only sustains a small proportion of the axial load. The majority of the axial load reduces the compression sustained by the joint.

It was found that when transverse movement commenced, appropriate loading conditions led to prevailing torque nuts initially suffering a rapid loss of preload until the load in the bolt approached the axial load being applied by the hydraulic system. Nut rotation continued under the axial loading from the jacks. At the completion of the test, the hydraulic pressure was released reducing the load in the bolt to zero. This process is illustrated in figure 6.3.

Tests that involved an intermittent axial loading were completed by repeatedly increasing and then releasing the oil pressure being applied to the hydraulic cylinders.

6.3 RESULTS

Several types of all-metal prevailing torque nuts have been tested and have been reported in Chapter 5. As can be seen in figure 5.5 they were found to have similar loosening characteristics under a standard DIN 65151 test. All-metal prevailing torque nuts typically initially self-loosen, but the loosening then stops so that a residual preload is retained. A major factor in determining the value of the retained residual preload is the level of the prevailing torque of the nut. In general, the higher the prevailing torque, the higher will be the retained residual preload. (Torsional stress induced into the fastener for a given level of preload will also be higher.) In total over 50 nuts were tested, and, in general, the prevailing torque of the better performing nuts remained reasonably constant when they were re-used up to five times. The tests were conducted to establish the value of axial loading which would result in continued self

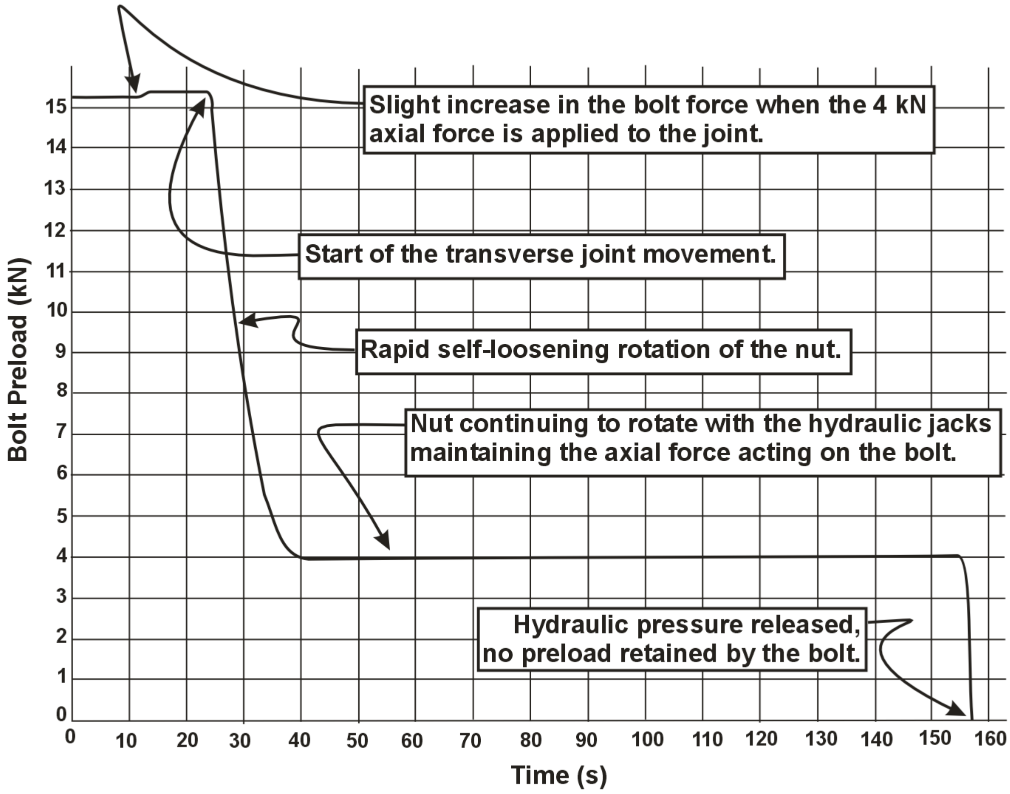

Figure 6.3 - Typical preload decay graph with transverse joint displacement and axial loading applied to the joint.

loosening of M8 prevailing torque nuts. The conditions for these tests are presented in table 6.1 along with a summary of some of the results.

The procedure for testing involved first determining the residual preload retained by the prevailing torque nut with zero axial loading applied to the joint using a conventional Junker test. The result from one such test are shown in the graph in figure 6.4a. The residual preload retained by the bolt in this case is 3.1 kN. Figures 6.4b and 6.4c show decay curves when axial loading of 1.1 kN and 2.7 kN was applied. In both situations the bolt preload was retained at approximately the same level as if no axial loading has been applied. However figure 6.4d shows the decay curve when an axial load of 3.1 kN is applied. In this case the nut continued to rotate, the axial load was maintained and compensated for the load loss that occurred from nut rotation. At the end of the test when the axial load was removed by releasing the pressure from the hydraulic cylinders, the residual preload was observed to be zero.

Test	Axial Loading	Retained bolt preload at the end of the test	Details
1	None	0.8 kN	
2	5 kN Intermittent	0 kN	Same nut as test 1
3	None	3.2 kN	
4	1.1 kN Constant	3.4 kN	Same nut as test 3
5	2.7 kN Constant	2.7 kN	Same nut as test 3
6	3.1 kN Constant	0 kN	Same nut as test 3
7	4.1 kN Constant	0 kN	Same nut as test 3
8	None	0.8 kN	
9	1.1 kN Constant	0 kN	Same nut as test 8
10	5.3 kN Constant	0 kN	Same nut as test 8
11	5 kN Intermittent	0 kN	Same nut as test 8
12	None	0.4 kN	
13	0.4 kN Constant	0.4 kN	Same nut as test 12
14	1 kN Constant	0 kN	Same nut as test 12
15	0.6 kN Constant	0 kN	Same nut as test 12
16	None	0.7 kN	
17	0.8 kN Constant	0 kN	Same nut as test 16
18	5 kN Intermittent	0 kN	Same nut as test 16
19	5 kN Intermittent	0 kN	Same nut as test 16
20	5 kN Constant	0 kN	Same nut as test 16
21	5 kN Intermittent	0 kN	Same nut as test 16
22	None	0.3 kN	
23	0.7 kN Constant	0 kN	Same nut as test 22
24	0.1 Constant	0.1 kN	Same nut as test 22
25	0.7 kN Intermittent	0 kN	Same nut as test 22
26	None	0.7 kN	
27	0.5 kN Constant	2.4 kN	Same nut as test 26
28	2.2 kN Constant	0 kN	Same nut as test 26
29	3.5 kN Intermittent	0 kN	Same nut as test 26

Table 6.1 Results from Junker tests with and without axial loading being present. The initial preload in all the tests was 15 kN. The test duration was typically 2 minutes (1500 transverse movement cycles).

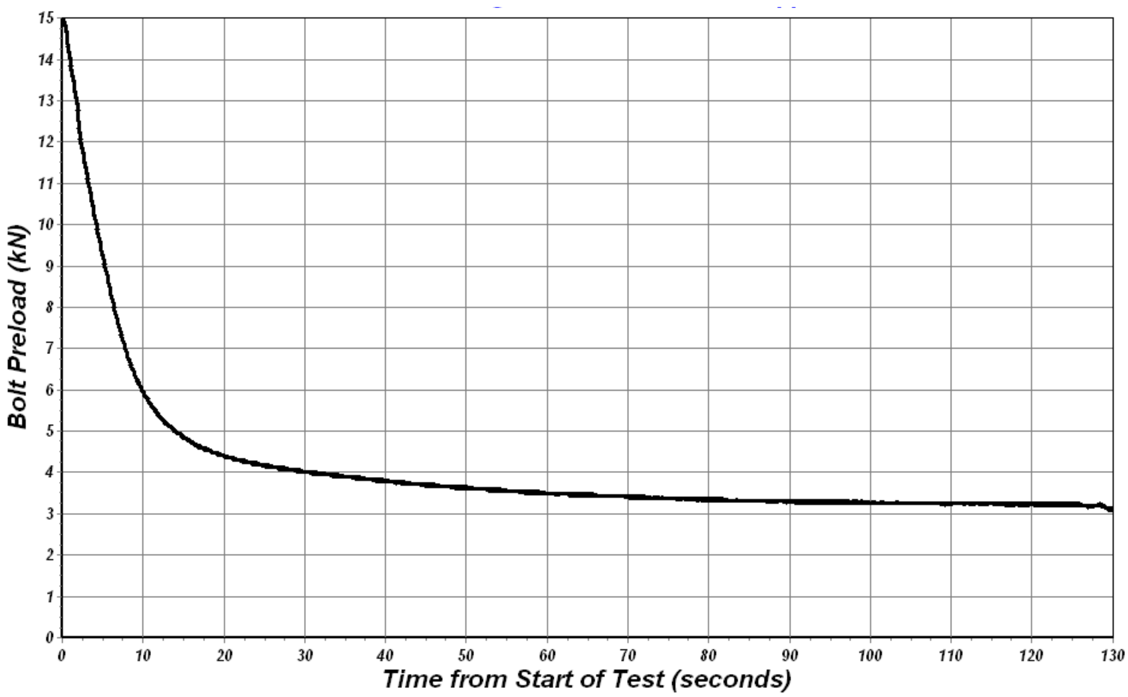

Figure 6.4a Effect of transverse joint displacement and axial loading on the loosening of a M8 all-metal prevailing torque nut - No Axial Loading Applied

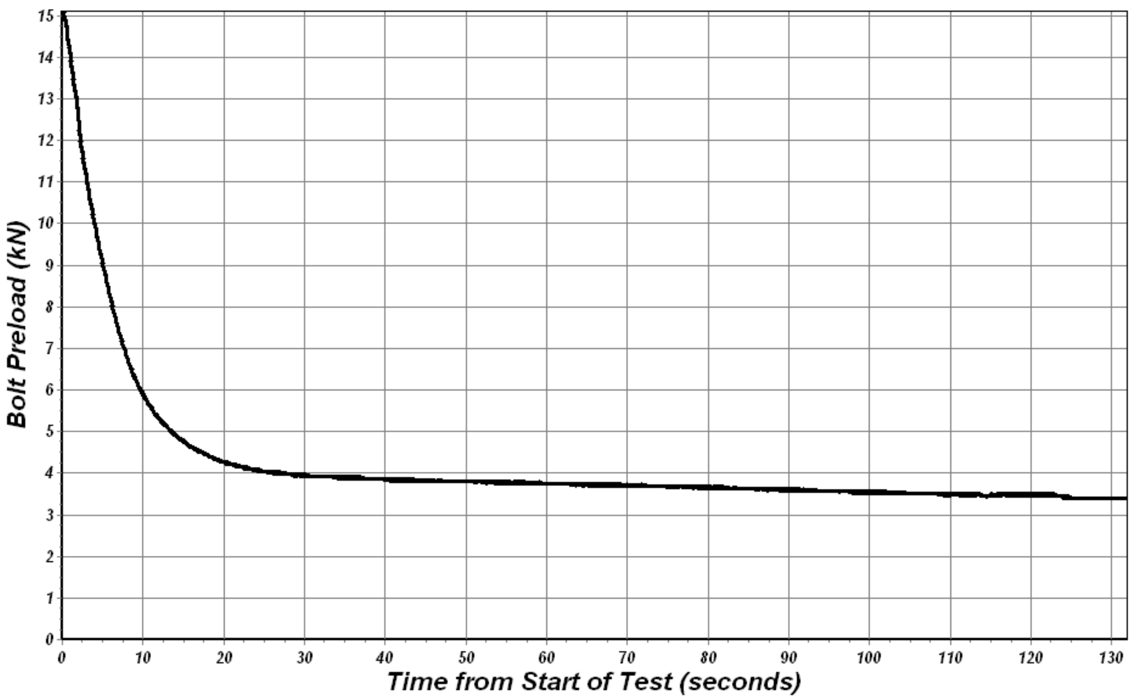

Figure 6.4b Effect of transverse joint displacement and axial loading on the loosening of a M8 all-metal prevailing torque nut - 1.1 kN Axial Loading Applied

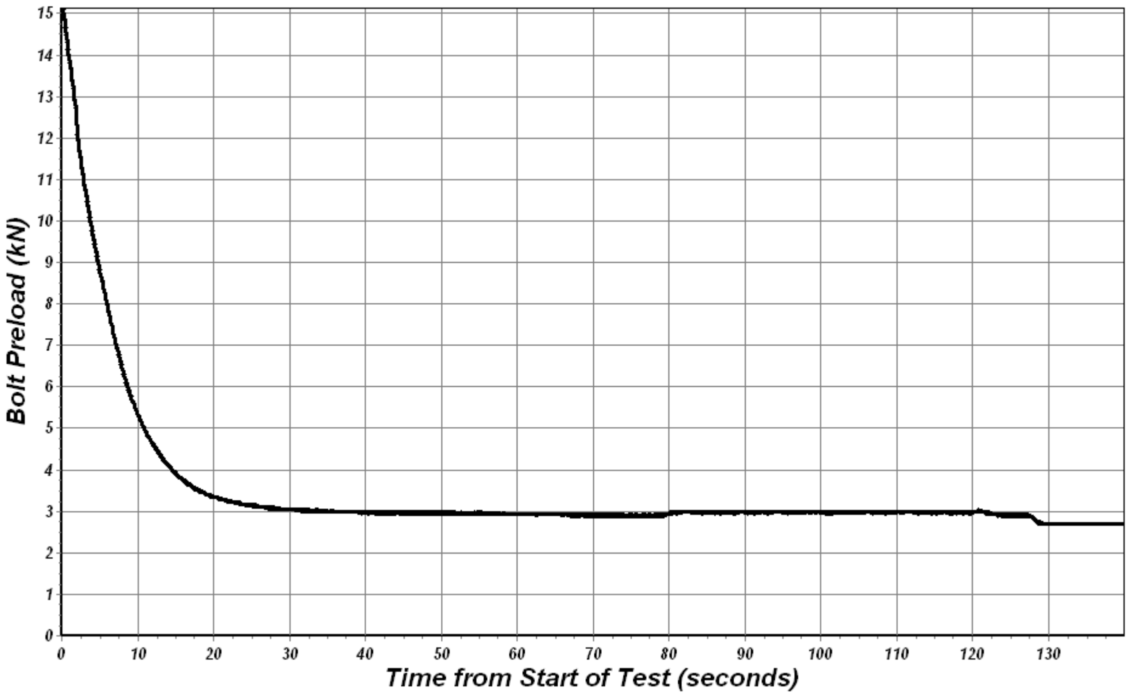

Figure 6.4c Effect of transverse joint displacement and axial loading on the loosening of a M8 all-metal prevailing torque nut - 2.7 kN Axial Loading Applied

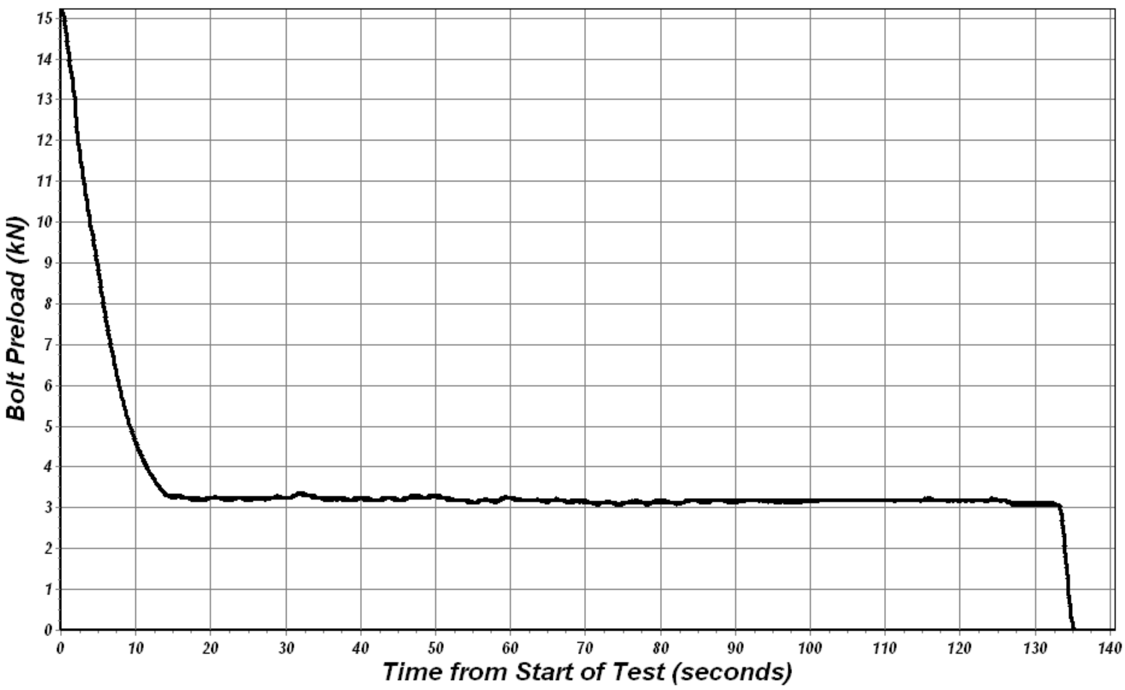

Figure 6.4d Effect of transverse joint displacement and axial loading on the loosening of a M8 all-metal prevailing torque nut - 3.1 kN Axial Loading Applied

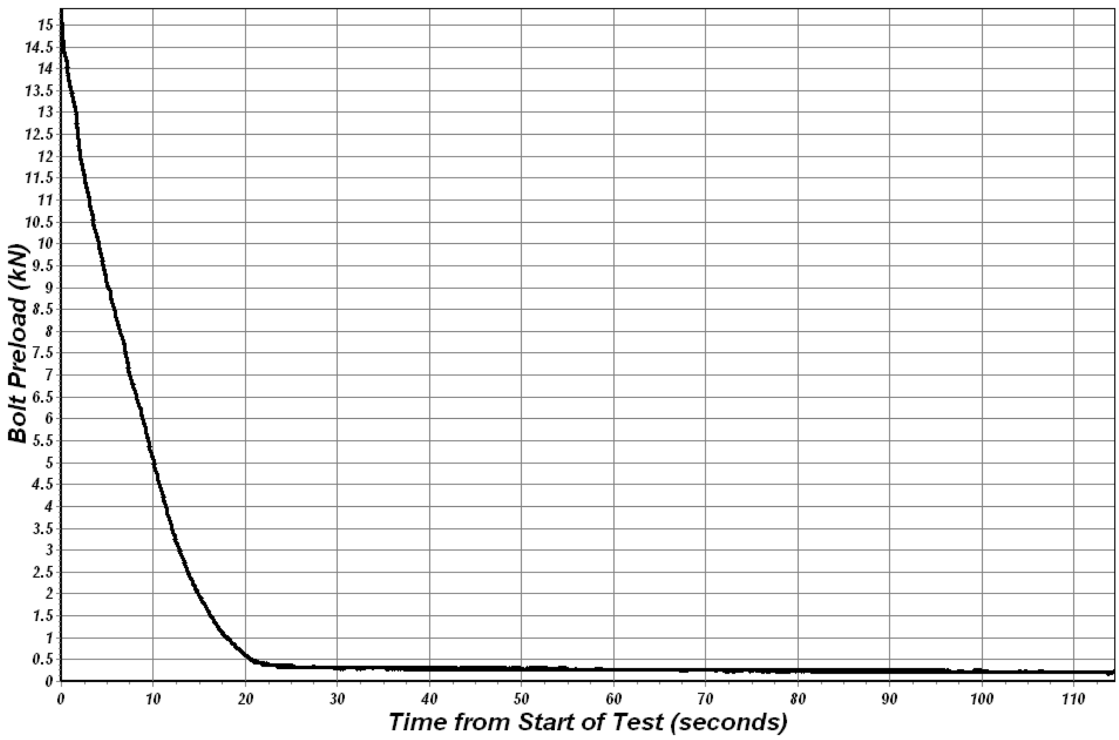

Figure 6.5a – Effect of transverse joint displacement and an intermittent axial loading on the loosening characteristics of M8 all metal prevailing torque nuts - No axial loading – 0.3 kN bolt preload retained

In many real life applications, the axial loading is intermittent rather than having a constant magnitude. For example loading on wind turbine securing bolts varies depending upon the wind speed. The loading on many joints in a car structure depends upon the terrain and the driving speed. In mechanical engineering, constant loading is the exception rather than the rule. Figure 6.5 illustrates results from a number of tests examining the effect that an intermittent axial loading had on the self loosening of M8 prevailing torque nuts. Figure 6.5a shows the results of a test on a M8 all-metal prevailing torque nut with zero axial loading. The retained preload for this particular nut was approximately 0.3 kN without axial loading. Figure 6.5b illustrates the effect of an intermittent axial load of 0.7 kN being applied to the same nut. At each instance of loading, further rotation of the nut occurred which incrementally reduced the preload. Figure 6.5c shows the results of another M8 all-metal prevailing torque nut with zero axial load. For this nut, the retained preload was approximately 2.4 kN. Figure 6.5d illustrates the effect of an intermittent axial load of 3.5 kN. Again, further rotation of the nut occurred that reduced the bolt preload to zero. Even after the preload had been reduced to zero, nut rotation could be observed when the axial load was applied whilst transverse joint movement was occurring.

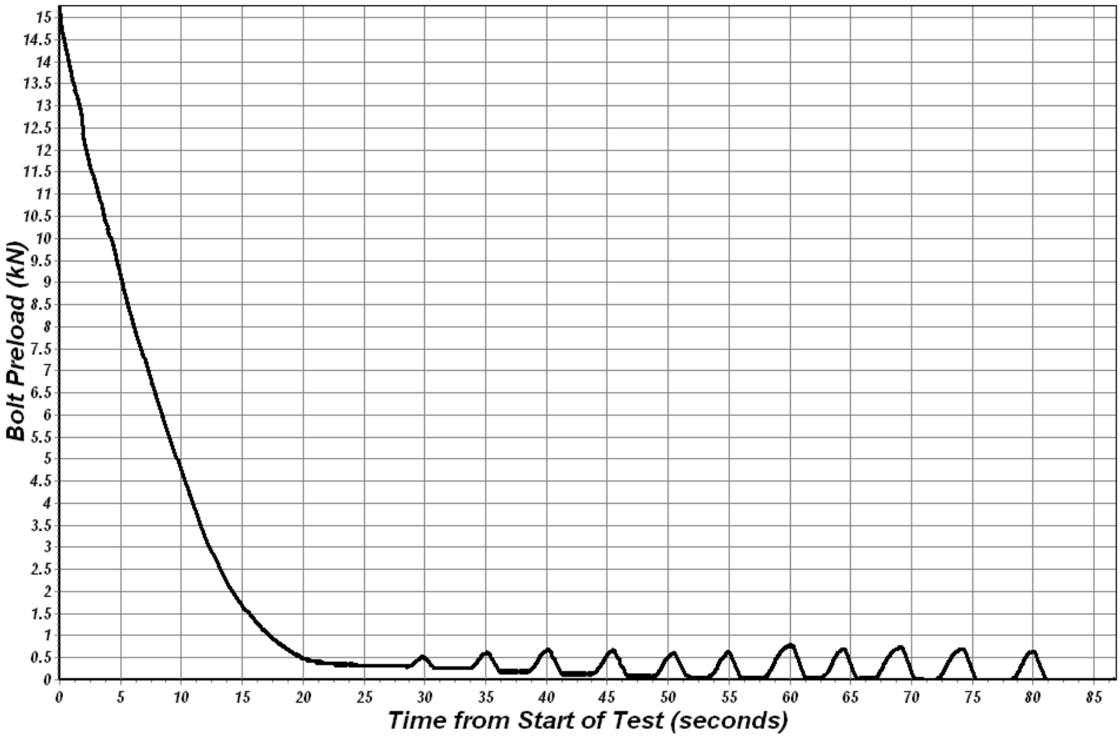

Figure 6.5b – Effect of transverse joint displacement and an intermittent axial loading. Nut as in test a) with 0.7 kN intermittent axial loading

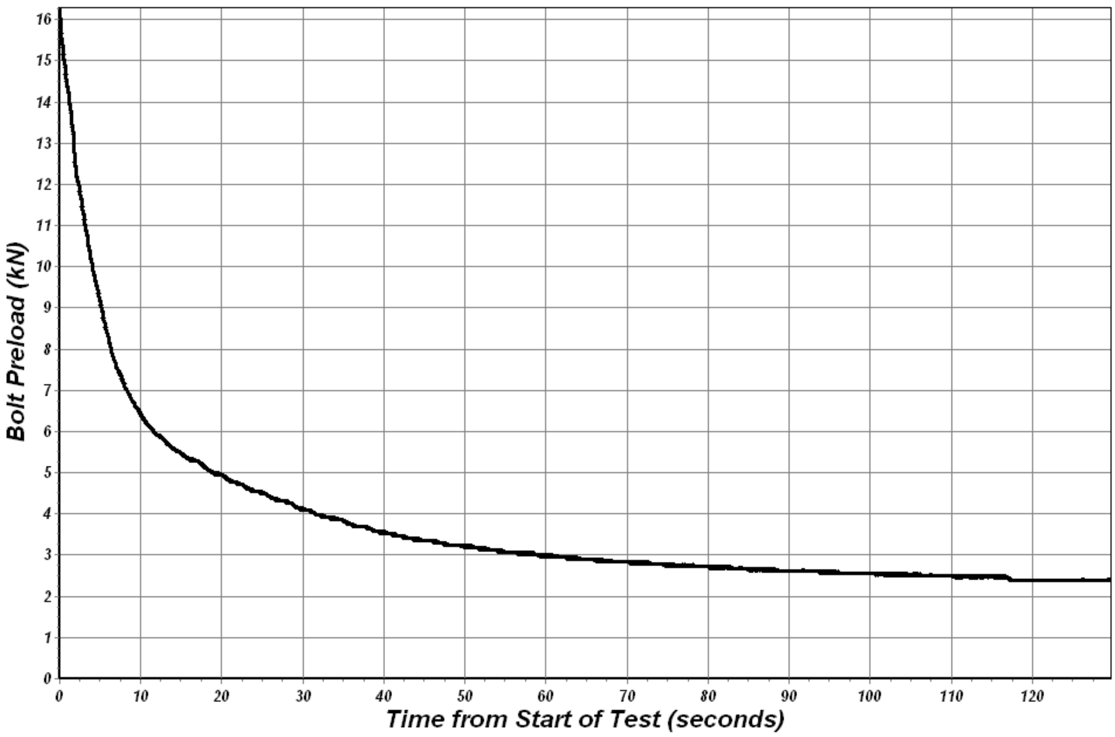

Figure 6.5c – Effect of transverse joint displacement and an intermittent axial loading No axial loading 2.4 kN bolt preload retained.

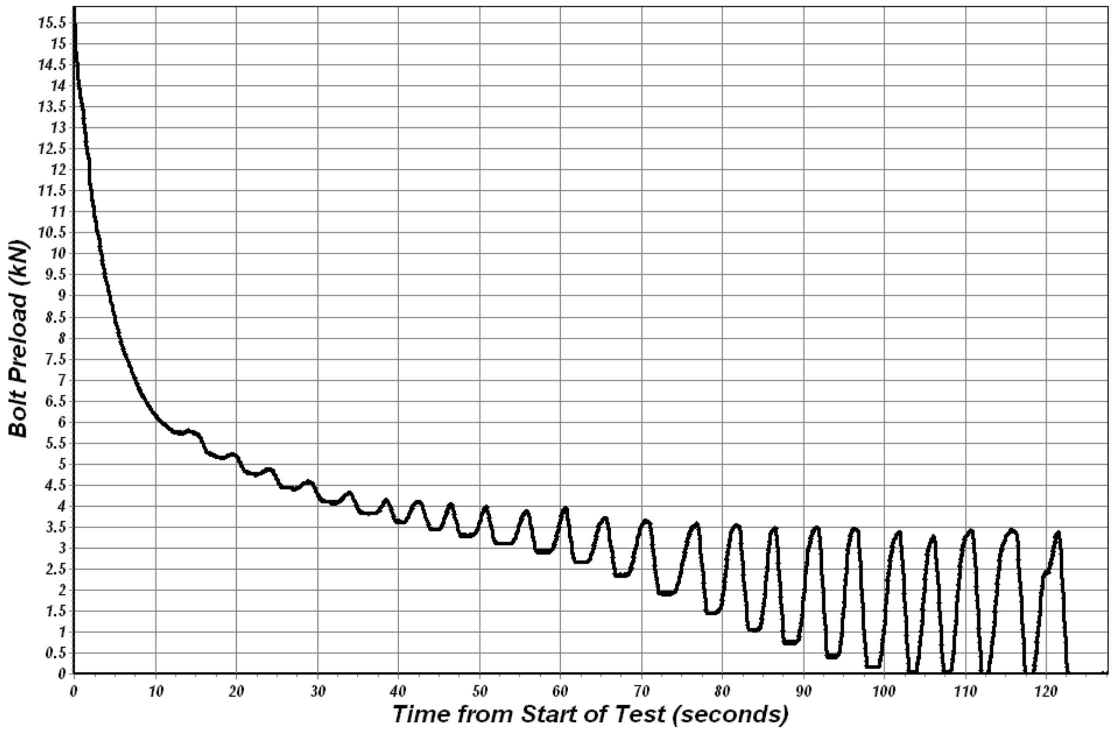

Figure 6.5d – Effect of transverse joint displacement and an intermittent axial loading. Nut as in test c) with 3.5 kN intermittent axial loading

6.4 ANALYTICAL MODEL

In order to be able to build an analytical model of the loosening process, forces resisting loosening and those promoting loosening need to be established. The relationship between the torque applied to a nut and the preload generated by the bolt involves the thread and nut dimensions and the coefficient of friction in the threads and under the nut face. There are various forms of the torque-tension equation for threaded fasteners. One form presented here is commonly used for free spinning (plain) nuts, the tightening torque T being given by (Kobayashi and Hongo, 1998):

$$T = \frac{F}{2}\left[\frac{p}{\pi} + \frac{\mu_t d_2}{\cos\alpha} + D_e \mu_n\right] \qquad (6.1)$$

where

D_e	The effective bearing diameter of the nut
F	Bolt preload
T	Tightening torque applied to the fastener
d_2	The basic pitch diameter of the thread
d_i	The inner bearing diameter of the nut face
d_o	The outer bearing diameter of the nut
p	Pitch of the thread
α	The half included flank angle for the threads
μ_n	Coefficient of friction for the nut face or bolt head (whichever is rotated during tightening)
μ_t	Coefficient of friction for the threads

The value of D_e is taken as the mean of the outer and inner bearing diameter of the nut :

$$D_e = \frac{d_o + d_i}{2} \qquad (6.2)$$

The first term in the brackets in equation 6.1 is the torque to stretch the bolt, this torque always acts in the loosening direction and its magnitude is independent of friction. The second and third terms in the brackets of equation 6.1 represent the torque needed to overcome thread friction and the torque needed to overcome nut face friction. When transverse slip of the joint occurs, the nut slides over the joint and simultaneous slip occurs between the nut and the bolt threads. Under such conditions the resistance to rotation is significantly reduced. The prevailing torque present from the nut resists loosening rotation, but in addition to the bolt stretch torque acting in the loosening direction, a further torque acts when slippage occurs on the threads. As shown by Sakai (2001) this torque is due to differences in the resistance offered between the ascending and descending sides of the nut thread whilst it is sliding on the bolt thread.

This is illustrated in figure 6.6. The thread surface is helical and is inclined relative to the slip direction by the lead angle β . The inclined surfaces, when a translational movement occurs, have an ascending side and a descending side which leads to a difference in the forces between the two sides which in turn generates a loosening torque. From Sakai's work (Sakai 2001), figure 6.7 shows linear slip occurring on a square thread. Friction acts on the area dA_s of the thread surface and the difference in forces resulting when slip occurs results in a loosening torque T_{ss} that (from (Sakai 2001)) can be expressed as:

$$
\begin{aligned}
T_{ss} = &\int_{r=r_1}^{r=r_2}\int_{\theta=0}^{\theta=\pi}\frac{F}{A_{slip}}dA_{slip}\left(\mu_t\cos\beta+\sin\beta'\right)\cos\beta' r\sin\theta \\
&-\int_{r=r_1}^{r=r_2}\int_{\theta=0}^{\theta=\pi}\frac{F}{A_{slip}}dA_{slip}\left(\mu_t\cos\beta-\sin\beta'\right)\cos\beta' r\sin\theta
\end{aligned}
\tag{6.3}
$$

where r_1 and r_2 are the inside and outside diameters respectively of the screw contact surface. The first term represents the ascent torque and the second term the descent torque acting on the bolt.

The area of the screw contact surface A_{slip} is given by:

$$
A_{slip} = \frac{\pi\left(r_2^2 - r_1^2\right)}{\cos\beta} \tag{6.4}
$$

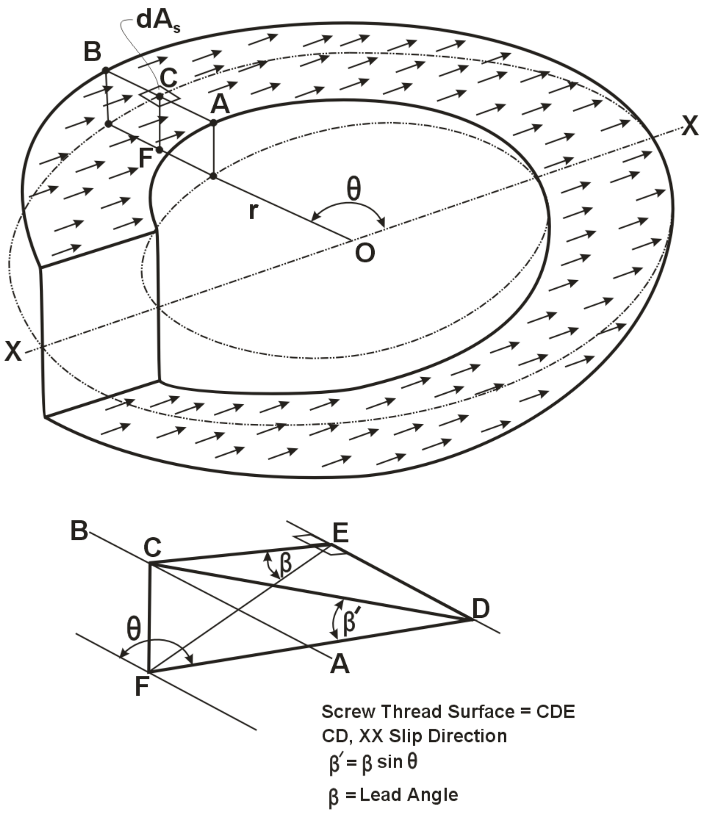

Figure 6.7 Linear slip occurring on a square thread

From figure 6.7, $\tan\beta^{'} = \tan\beta\sin\theta$ and since the lead angle β is small (3 degrees for the M8 threads used in this study) we have $\beta^{'} = \beta\sin\theta$ and $\cos\beta^{'} = 1$ In this case, T_{SS} will assume the following approximate equation:

$$T_{ss} = \frac{2F\beta}{A_{slip}} \int_{r=r_1}^{r=r_2} \int_{\theta=0}^{\theta=\pi} r^2 \sin\theta \, d\theta dr \qquad (6.5)$$

Integration of equation 6.5 leads to:

$$T_{ss} = \frac{\pi F\beta}{3A_{slip}}\left(r_2^3 - r_1^3\right) = \frac{F\beta\left(r_2^2 + r_2 r_1 + r_1^2\right)}{3\left(r_2 + r_1\right)} \qquad (6.6)$$

The author's extension to Sakai's work (2001) involves using the basic pitch diameter of the thread, denoted by d_2., and taking this as being equal to r_1+r_2, Also from this r_1= d_2 - r_2

Substituting these values into equation 6.6 gives: $T_{ss} \cong \frac{1}{4} F d_2 \beta$ (6.7)

If p is the pitch of the thread and d_2 the basic pitch diameter, the lead angle is the transverse movement of the screw completed in one revolution divided by the circumference around the thread at the pitch diameter:

$$\beta = Tan^{-1}\left(\frac{p}{\pi d_2}\right) \qquad (6.8)$$

Since the angle is small, equation 6.8 can be approximated as:

$$\beta = \frac{p}{\pi d_2} \qquad (6.9)$$

Substituting this value into equation 6.7 gives: $T_{ss} = \frac{Fp}{4\pi}$ (6.10)

From equation 6.1, the first term in the equation represents the torque needed to stretch the fastener which always acts in the loosening direction. The second term represents the torque needed to overcome friction in the threads and the third term is the torque needed to overcome friction under the nut face. Friction resists nut rotation in both the loosening and tightening directions. Considering the loosening direction and adding the prevailing torque which also resists nut movement in any rotation direction gives the torque T_L needed to loosen a nut as:

$$T_L = F\left[-\frac{p}{2\pi} + \frac{\mu_t d_2}{2\cos\alpha} + \mu_n \frac{D_e}{2}\right] + T_P \qquad (6.11)$$

The above equation applies to static situations. Due to the negative term, the loosening torque is typically 15% to 20% lower than the tightening torque which can be readily practically demonstrated. (This applies when the nut is untightened immediately after being tightened. If the joint is left for a period of time friction changes can occur that can affect the situation such that the loosening torque can be either smaller or greater than the initial tightening torque.) When transverse slip is occurring to the joint, the torque acting in the loosening direction will be the torque given by equation 6.10 and the first term in equation 6.11. Combining these terms to establish the total torque acting in the loosening direction T_{loosen} under transverse slip conditions gives:

$$T_{loosen} = \frac{Fp}{4\pi} + \frac{Fp}{2\pi} = \frac{3Fp}{4\pi} \qquad (6.12)$$

Under the conditions of slip under the nut face and in the threads, friction has been overcome in the transverse direction by external forces to the joint. Under such conditions Junker (1969) considered that a non-locking nut would be free from friction in the circumferential direction. Sakai (2001) has concluded that under slip conditions, the resistance of the nut to rotation is extremely small, he quotes friction coefficients of between 0.00 and 0.02. Strictly what are quoted are not true friction coefficients since friction has already been overcome by the external force causing transverse joint slip. The friction values he quotes are based upon the additional force/torque to rotate the nut under slip conditions. Applications which include this effect include floor polishing machines, the machine being easier to move when the polishing disk is rotating. Also, a cork is easier to remove from a bottle if the cork is first rotating before being pulled. Overcome friction in one direction and the resistance to movement in another direction reduces dramatically. To differentiate between thread and head friction values under static conditions and values under the conditions of transverse slip, the terms μ_{ns} and μ_{ts} will be used for the nut face and thread friction coefficients in the rotational direction when transverse slip is occurring. Also the term T_{Ps} will be used to denote the prevailing torque under transverse slip. The torque T_R that resists loosening under the conditions of transverse slip is then given by:

$$T_R = F\left[\frac{\mu_{ts} d_2}{2\cos\alpha} + \mu_{ns}\frac{D_e}{2}\right] + T_{Ps} \qquad (6.13)$$

For rotation of the nut in the loosening direction to occur, the torque acting in the loosening direction T_{loosen} must be greater than the torque T_R resisting loosening, hence for loosening to occur:

$$\frac{3FP}{4\pi} > F\left[\frac{\mu_{ts} d_2}{2\cos\alpha} + \mu_{ns}\frac{D_e}{2}\right] + T_{Ps} \qquad (6.14)$$

Since the thread and head friction coefficients in the rotational direction when transverse slip is occurring (μ_{ts} and μ_{ns}) are close to zero (Sakai, 2001), equations 6.14 simplifies to give the condition for self loosening to occur with prevailing torque nuts:

$$\frac{3Fp}{4\pi} > T_{ps} \qquad (6.15)$$

For the M8 nuts used in this test series, the retained preload is typically in the range of 1 to 3 kN. This implies that T_{ps} is in the range of 0.3 Nm to 0.9 Nm. The relevant standard quotes a maximum first assembly prevailing torque of 6 Nm for this size and grade of nut and a minimum fifth removal prevailing torque of 0.6 Nm. The prevailing torque measured on the nuts used in these tests was between 1.5 Nm and 2.3 Nm. The frictional drag generating the prevailing torque is likely to be reduced under transverse slip conditions which is one explanation for the difference. The prevailing torque is generated by pressure applied by the nut to the bolt thread. Just as transverse joint movement reduces the rotational resistance of a plain nut, a prevailing torque will probably also be reduced under such conditions when transverse slip of the threads is occurring.

When a tensile axial load F_A is applied to a joint held together by a tightened bolt, the bolt does not sustain the full effect of the load, but usually only a small part of it. The majority of the applied load, typically 90% for structural joints, reduces the clamp force on the joint provided by the bolt. The remaining 10% of the load will increase the force in the bolt. This applies until joint separation occurs, that is, the applied force exceeds the clamp force on the joint and a gap occurs in the joint. Under conditions of transverse joint slip, once self-loosening reduces the bolt preload such that joint separation will occur, it is the tensile axial load F_A that will be the cause of further nut rotation. The bolt would essentially act as if it had not been tightened, all the axial load being sustained as a tensile load in the bolt. Under the such conditions, F_A can replace F in equation 6.13.which can be re-arranged to give the following condition for F_A in order to ensure that continued rotation of the nut would not occur:

$$\frac{4\pi T_{ps}}{3p} > F_A \qquad (6.16)$$

If the condition given by equation 6.16 was not met, continued rotation would occur until the nut detached from the bolt. For free spinning nuts under transverse slip conditions, T_{ps} is zero or close to zero and subsequently detachment of the nut from the bolt can occur with even very small axial loads applied to the joint.

6.5 DISCUSSION

Based upon the experiments completed and the measurements made, the effect of applying an axial load when transverse joint movement is occurring is to aid the self loosening tendency of prevailing torque nuts. Whether or not complete loosening will occur depends upon the magnitude of the applied axial load.

Tests have also shown that when an intermittent axial load occurs when a joint experiences transverse slip this can also result in the complete self-loosening of prevailing torque nuts and their possible detachment from bolts. In mechanical engineering applications, dynamic conditions often result in any axial loading being applied to a joint being variable in magnitude. The results of these tests point towards an occasional peak in axial load being especially detrimental to the security of prevailing torque nuts if the loading is assumed to be purely shear in nature.

The feature that increases friction that results in the prevailing torque on the majority of this type of nut is located generally on the top of the nut. Junker (1969) speculated that some degree of resistance to loosening will be provided by such features restricting the thread movement of the nut relative to the bolt. However, tests have indicated that this effect can be routinely overcome under the right conditions.

6.6 CONCLUSIONS

It has been shown that under certain conditions loosening and the complete detachment of prevailing torque nuts can occur. Such a loosening process involves the application of an axial load when joint slip is occurring. The tests described here indicate that if the magnitude of the axial loading exceeds the residual preload in the bolt retained from sustaining transverse movement alone, all-metal prevailing torque nuts can completely detach from bolts.

A key finding of this study is that prevailing torque nuts cannot be defined as loss prevention devices when axial loading as well as transverse movement is applied to a joint. This implies that the present standard test method (DIN 65151) should be revised to allow a true assessment of the locking ability of certain categories of fasteners.

Axial loading, applied whilst transverse joint slip is occurring, also affects the loosening characteristics of standard plain nuts. On a standard Junker test, plain nuts generally stop rotating when the preload reaches zero, that is, the nut is retained on the bolt. In the presence of an axial load, such rotation does not stop and will continue until the nut detaches from the bolt. There are a number of examples of accidents resulting from nuts becoming completely detached from bolts. One such accident occurred on Friday 23 February 2007 when a train derailed on points at Lambrigg, near Grayrigg in Cumbria (RAIB, 2008). A significant factor in the occurrence of this accident was nuts becoming detached from the bolts allowing the switch rail to be struck by the inner faces of passing train wheels. This resulted in the subsequent failures of other parts of the switch structure and ultimately the derailment of the train. Previous attempts to explain why such nuts become detached have relied upon an appropriate axial vibration frequency being applied (Hess et al. 1996; Hess and Davis 1996; Hess and Davis, 1996; Hess, 1997; Basava and Hess, 1998) to the assembly so that the nut rotates in the off direction.

Applications that involve shear and axial loading being simultaneously applied to a joint are numerous in engineering. The reason why prevailing torque nuts have historically been frequently specified is the belief that if they came loose, they would not become detached from the bolt. This work has shown that the engineer may need to be more circumspect in this regard and give some study as to the magnitude of any axial load that may be acting on the joint. Knowledge of the circumstances under which a prevailing torque nut can become completely detached from the bolt will enable more reliable and safer designs to be engineered to prevent failures and accidents.

Based upon the work of Junkers and other researchers, it is known that self loosening of threaded fasteners can be prevented if sufficient preload is generated so that friction grip between the joint plates prevents the occurrence of transverse movement. In applications in which overload conditions can occasionally cause transverse joint movement, prevailing torque fasteners are frequently used in the belief that although partial loosening may occur, the nut will not become detached from the bolt. In the light of the research presented here, this criterion requires to be updated to include the simultaneous effect that axial loading can have on the loosening process. Additionally, repeated re-use/tightening of prevailing torque nuts can reduce the prevailing torque due to thread wear and hence their effectiveness in resisting loosening. This work has concluded that the maximum simultaneous axial load that can be sustained when transverse joint movement occurs is equivalent to the preload retained by the prevailing torque nut on a normal Junkers test.

7. THE EFFECT OF TRANSVERSE VIBRATION ON THE FRICTION FORCES ACTING ON THREADED FASTENERS

7.1 INTRODUCTION

The vast majority of threaded fasteners retain their preload because of friction present on the thread and head contact surfaces that resists self-loosening. In 1969, Junker in his groundbreaking paper (1969) showed that transverse vibration could completely loosen plain non-locking threaded fasteners. He stated that once relative motion occurs between the threaded surfaces and other contact surfaces of the clamped parts because of a tangential external force, the bolted connection would become free of friction in a circumferential direction. In the paper he reports on some tests that he completed to measure the torque needed to loosen a threaded fastener subjected to transverse vibration. He found that when the induced transverse force exceeded a certain level, negative torque values resulted indicating that the fastener would self-loosen. The bolt preload acting on the thread, which is sloped, creates a torque in the circumferential direction that results in self-loosening of the fastener.

Based upon the literature search there seems to be an absence of studies investigating how the friction force reduces in the circumferential direction when transverse slip is occurring. This chapter presents the results of a series of experiments measuring the friction forces involved under such slip conditions.

7.2 DETAILS OF THE TESTS COMPLETED

The Junker machine that was used in previous experiments was developed to allow torque measurements to be completed on threaded fasteners whilst they were subjected to transverse vibration. Wear was found to be an issue in these tests and was reduced by the use of a thin coating of grease on the threads and nut bearing surfaces.

This machine was used to perform three series of tests:

1. Measurement of the torque tending to loosen fasteners whilst they were subjected to transverse vibration.
2. Tightening of fasteners whilst they were subjected to transverse vibration.
3. Measurement of the torque needed to rotate nuts in the tightening direction whilst they were subjected to transverse vibration.

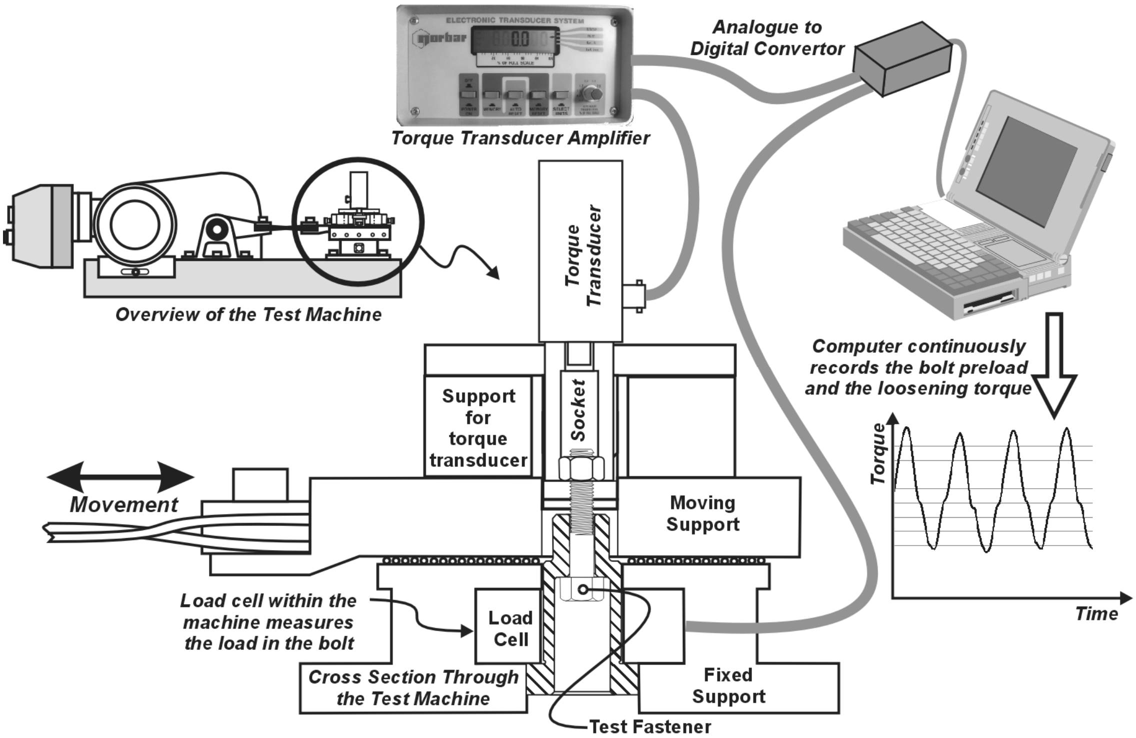

Figure 7.1 Test Arrangement to determine the loosening torque under transverse vibration

The first series of tests involved measurement of the loosening torque being exerted on M8 electro-zinc plated bolts and plain non-locking nuts whilst being subjected to transverse vibration. A torque transducer (a Norbar 0 to 10 Nm transducer) was mounted on the moving top plate of the machine so that it would react the loosening torque being generated by the fastener undergoing transverse slip. A socket connected the nut to the torque transducer. Figure 7.1 illustrates the arrangement used.

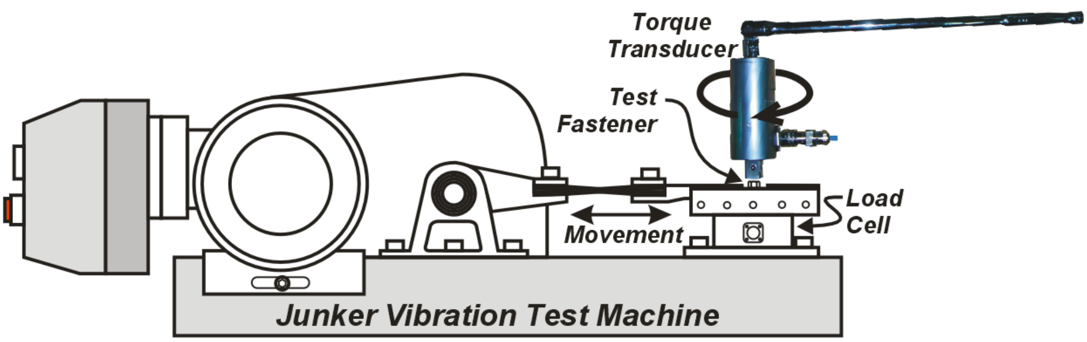

Figure 7.2 Test Arrangement to determine the tightening torque under transverse vibration

The second series of tests reproduced the work of Sakai (1978) and allowed the overall friction coefficient of the fastener to be determined following the DIN 946 method

(Deutsche Norm, 1991). The torque needed to tighten a M8 plain non-locking electro-zinc plated nuts onto bolts under the conditions of transverse slip was measured. Figure 7.2 shows the test arrangement used. The torque was applied to the nut in a slow and deliberate manner by hand.

In the third series of tests an electro-zinc plated flanged nut (with a Spiralock thread form) was locked onto a M6 threaded rod using a high strength chemical threadlocker (Loctite 272) so that no self-loosening would occur. The torque needed to rotate the fastener under transverse vibration was measured. From the torque measurements obtained it was possible to determine the circumferential coefficient of friction of the nut face under conditions of transverse slip. Figure 7.3 illustrates the test arrangement used. In order to reduce friction on the bottom face, the nut clamped a thrust bearing. In order to allow the drag torque of the bearing to be determined, tests were completed with an identical thrust bearing mounted on the moving surface, and the torque needed to rotate the assembly measured. It was assumed that one bearing would have half the frictional drag torque of two. The torque absorbed by the rotation of the nut could be established by subtracting the bearing torque from the overall applied torque.

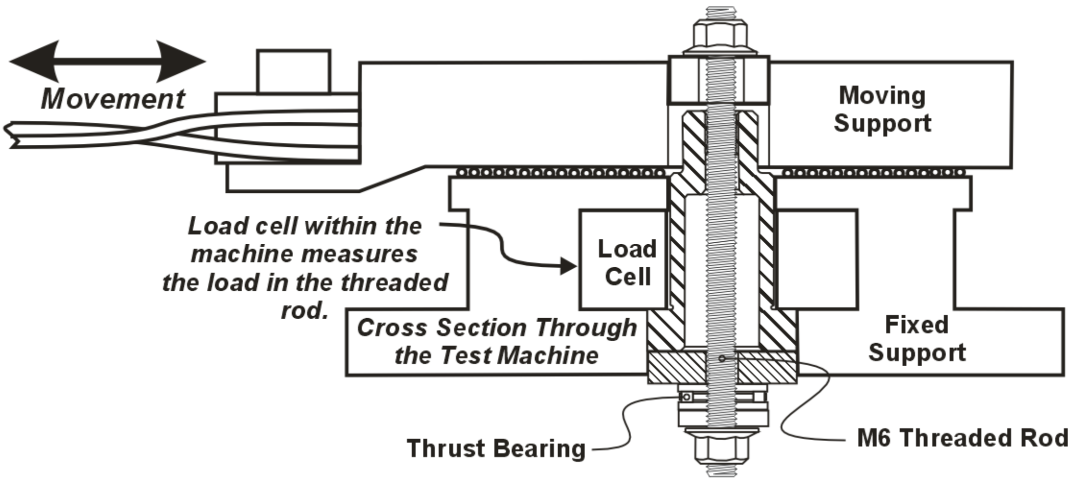

Figure 7.3 Test Arrangement to determine the nut bearing torque under transverse vibration

7.3 EXPERIMENTAL RESULTS

7.3.1 Loosening Torque Measurements

Measurements were made of the loosening torque with preloads ranging from 1 to 15 kN in 1 kN increments using the arrangement shown in figure 7.1. A typical result from the loosening tests is shown in figure 7.4. The loosening torque varies between a lower and upper limit in a roughly sinusoidal form corresponding to the frequency of the transverse vibration i.e. 12.5 Hz. Figure 7.5 shows the upper and lower torque limits measured over a one second period for bolt preload values between 1 and 15 kN.

Figure 7.6 plots the average values for the upper and lower loosening torque values predicted by equation 6.12. The lower loosening torque value, for a particular preload is constant and is the torque due to the preload acting on the inclined helix of the thread. The upper loosening torque value corresponds to the lower torque value plus the torque due to the movement of bolt thread passing over the nut thread when transverse joint movement is occurring.

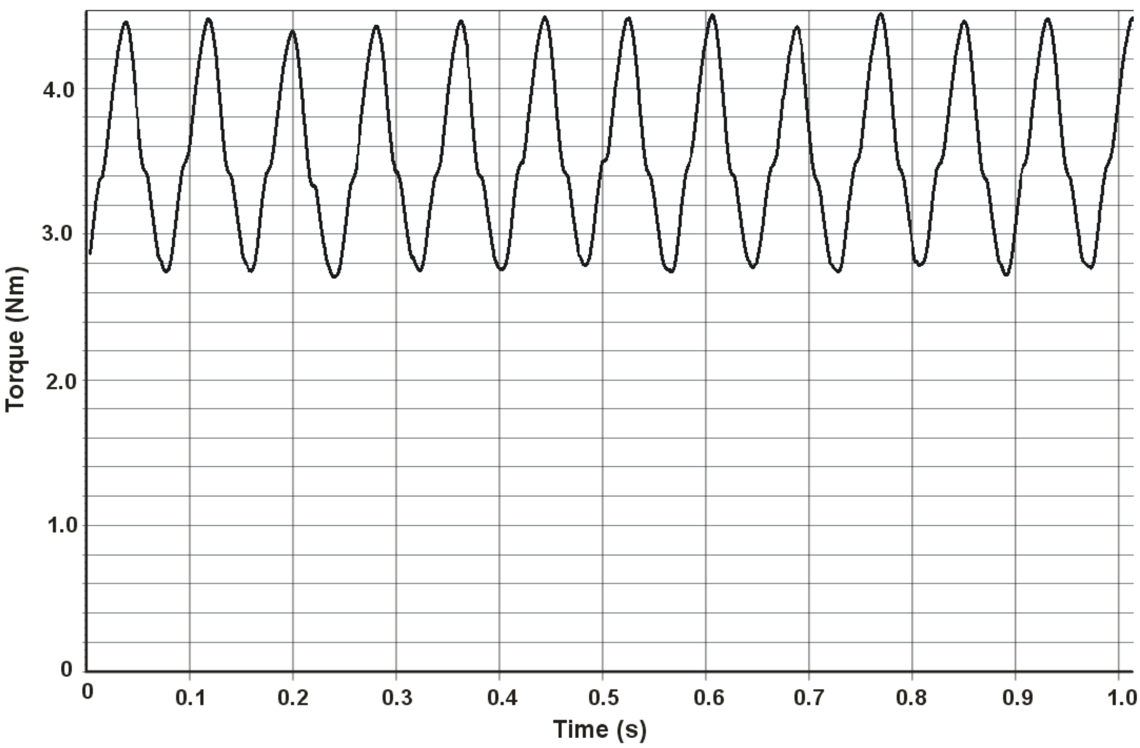

Figure 7.4 Loosening torque measured for a M8 Fastener over a one second period

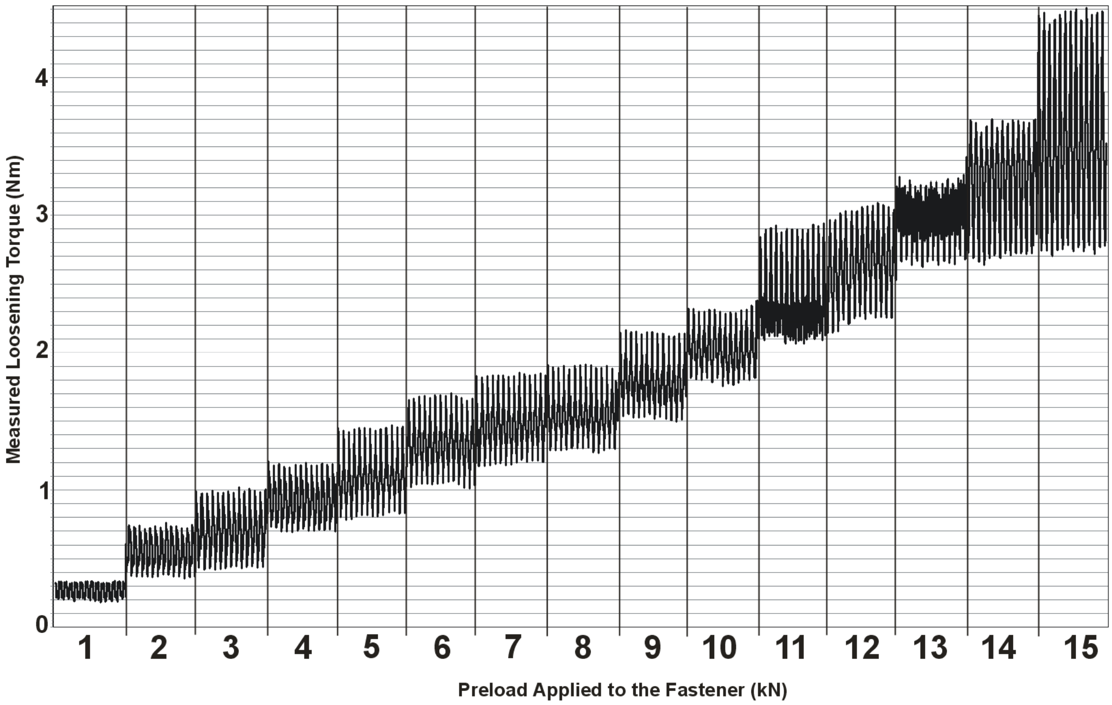

Figure 7.5 Measure of the loosening torque over one second periods for preload values from 1 to 15 kN

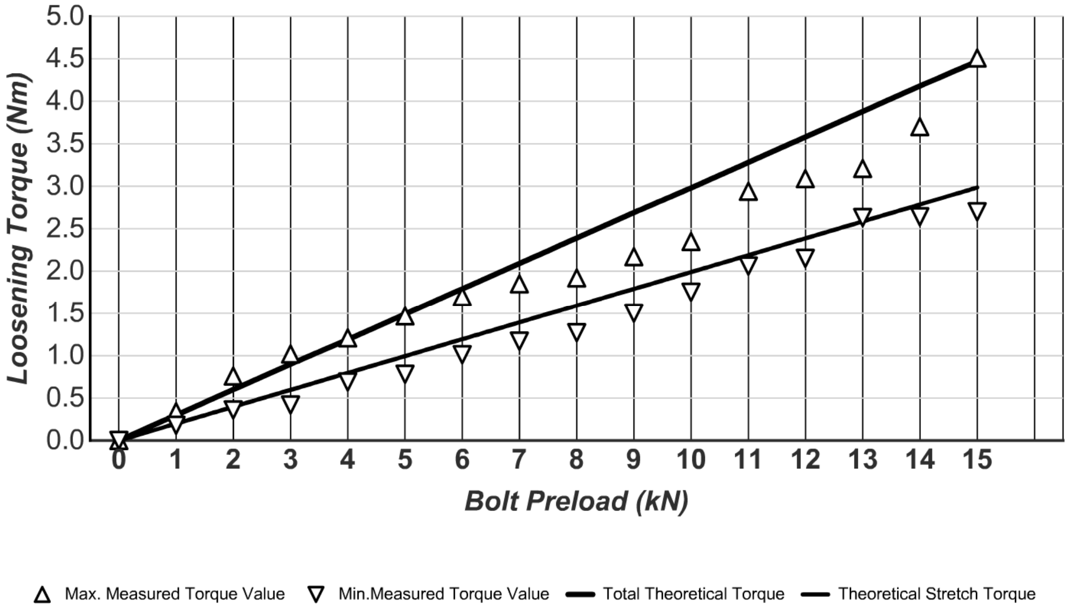

Figure 7.6 Comparison of the upper and lower limits of the loosening torque for bolt preload values from 1 to 15 kN

7.3.2 Tightening Torque Measurements

Tightening torque measurements were made up to a nominal preload of 10 kN since it was found that at very rapid wear was occurring on the threads at higher preloads. The graph in figure 7.7 shows experimental measurements of the difference in the torque-tension relationship between a bolt tightened with and without transverse vibration acting.

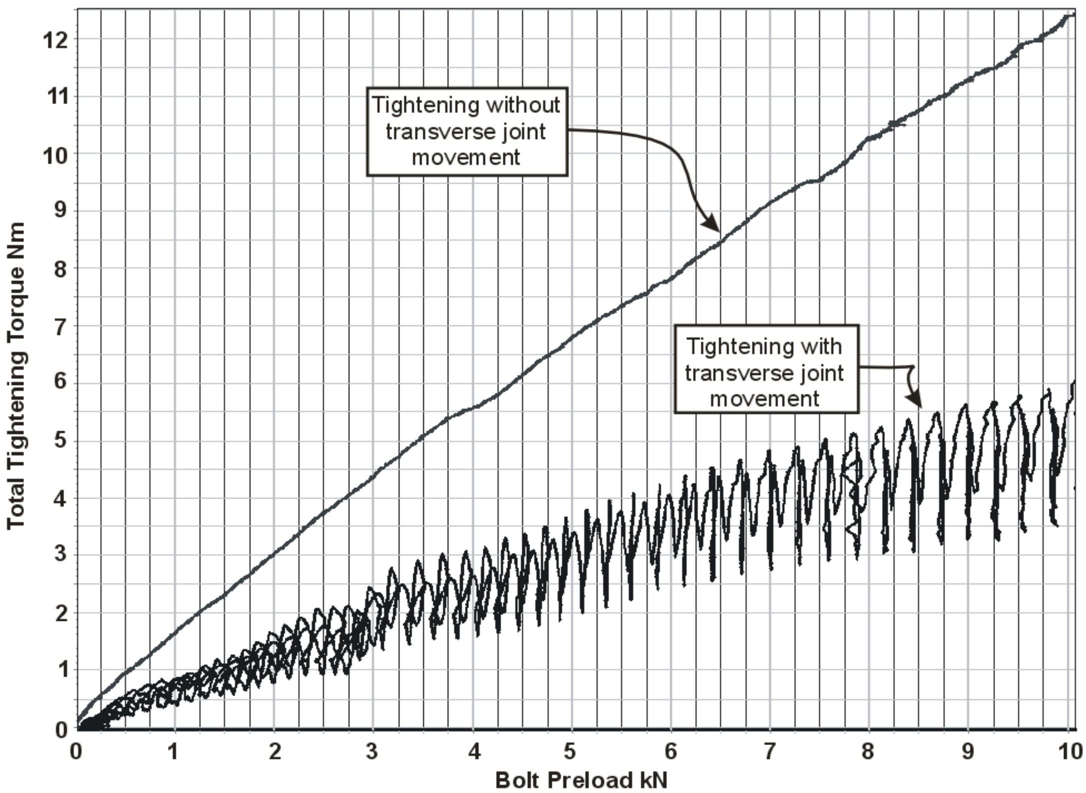

Figure 7.7 Torque-bolt preload characteristics when tightening a bolted joint with and without transverse movement

Without vibration, the torque-tension relationship is linear. With vibration, the magnitude of the torque fluctuated corresponding to the load applied by the transverse vibration. The torque required to tighten the nut was significantly lower when vibration was present. Based upon the torque and preload measurements made and the dimensions of the bolt, the coefficient of friction measured in a rotational sense between the nut face and threads can be determined. Figure 7.8 shows this coefficient of friction varies with preload during tightening whilst the joint is under transverse vibration. This graph was produced by measuring the applied torque and preload whilst the nut was being tightening and computing the coefficient of friction from these values. The cyclic variation in the friction coefficient is due to the transverse motion.

From figure 7.8 the upper limit of friction can be seen to decrease as the preload increases. To obtain some indication of the scatter in the results, the upper and lower limits of the torque were determined for bolt preload values between 1 and 10 kN for 15 fasteners. A boxplot of the results is presented in figure 7.9.

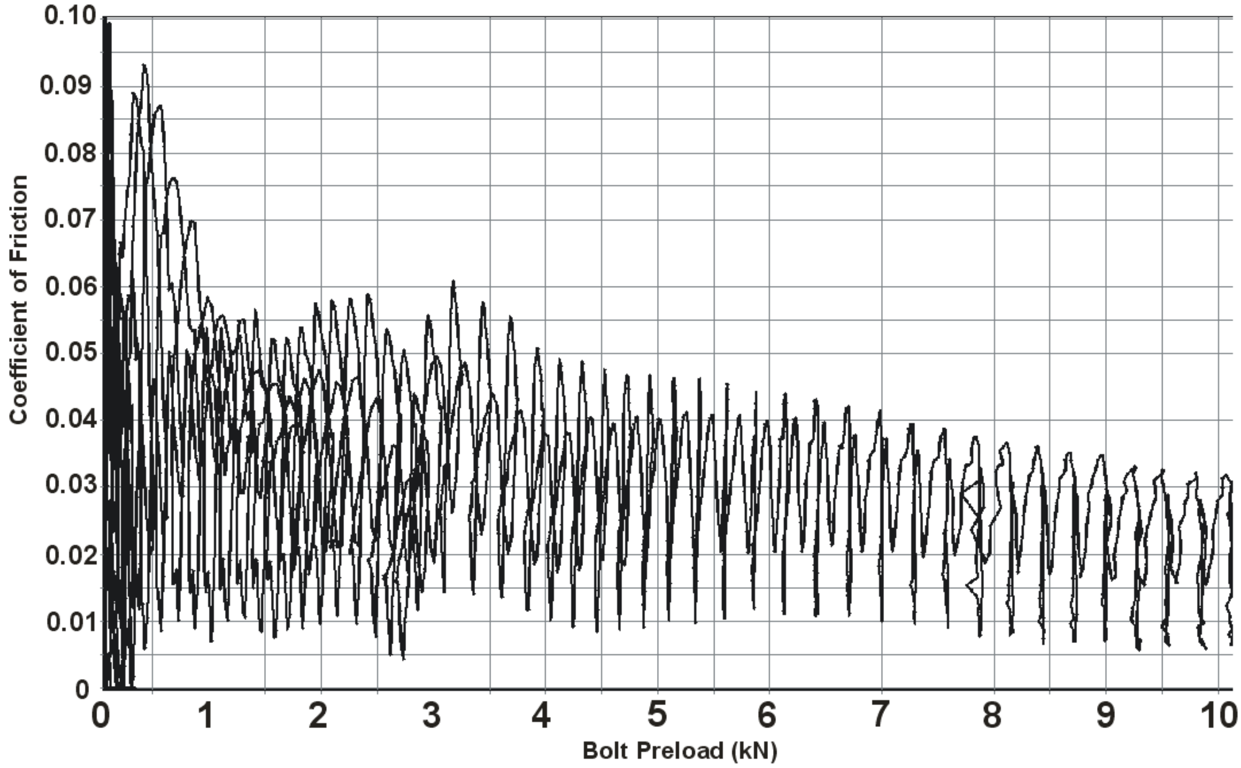

Figure 7.8 Measurement of the coefficient of friction for a fastener with a joint undergoing

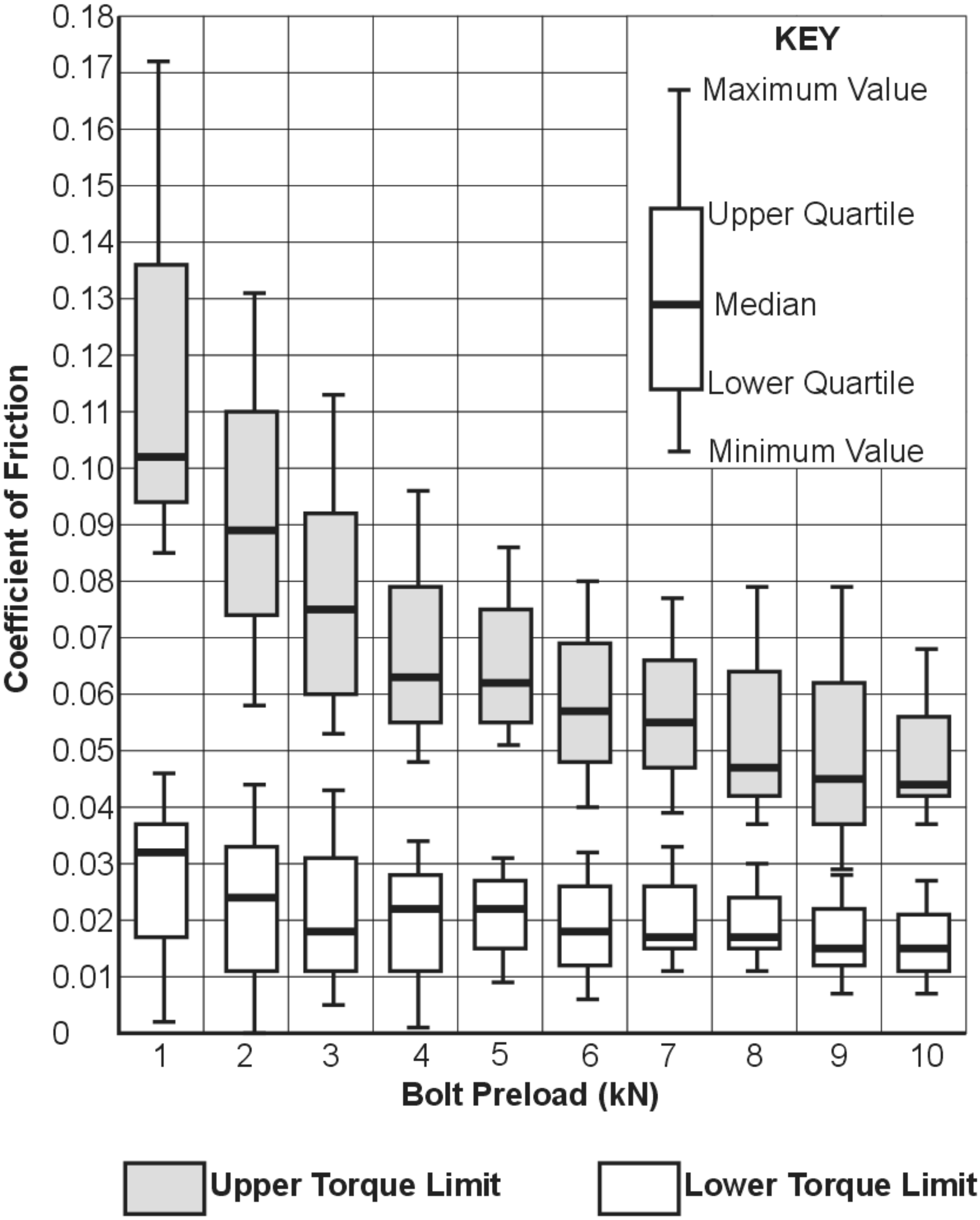

Figure 7.9 Boxplot of upper and lower friction values during transverse joint movement

7.3.3 Nut Rotation Torque Measurements

To investigate the torque needed to rotate a nut under a fixed load whilst undergoing transverse slip, the test arrangement shown in figure 7.3 was used. The nuts were locked onto the threaded rod at a particular preload. A thrust bearing was used on the lower nut to reduce the rotation torque needed. To establish the frictional drag torque of the bearing a number of calibration experiments were completed. This was done by placing an identical bearing under the top nut and measuring the torque to rotate the assembly whilst it was being subjected to vibration. For the calibration tests the clearance between the threaded rod and the side of the adapter it passes through was increased to ensure that bending would occur rather than the bolt head slipping. Table 7.1 presents a summary of the results for five tests at a 4 kN loading.

	Mean torque (Nm)	Torque standard deviation (Nm)	Minimum torque (Nm)	Maximum torque (Nm)	Coefficient of friction range (for each bearing)
1	0.274	0.020	0.192	0.333	0.0017 – 0.0030
2	0.343	0.053	0.211	0.470	0.0019 – 0.0042
3	0.256	0.030	0.143	0.320	0.0013 – 0.0029
4	0.294	0.047	0.172	0.396	0.0015 – 0.0035
5	0.336	0.028	0.263	0.417	0.0023 – 0.0037
Average/ Max/Min	0.301		0.192	0.470	0.0017 – 0.0042

Table 7.1 Results of the calibration tests

Tests on assemblies where the top nut was subjected to transverse slip were completed at preload values between 1 and 4 kN. At the lower preload values greater variability was observed. Table 7.2 presents the results for tests at a preload value of 4 kN. A graph illustrating this set of results is shown in figure 7.10 A total of nine tests were completed at a preload of 4 kN with the nut sliding under transverse vibration. The results of the tests are shown in the table 7.2. The mean torque was determined by averaging all the torque values measured in the sample interval.

	Mean torque (Nm)	Torque standard deviation (Nm)	Minimum torque (Nm)	Maximum torque (Nm)
1	0.314	0.033	0.211	0.405
2	0.182	0.024	0.122	0.266
3	0.242	0.031	0.123	0.309
4	0.252	0.036	0.175	0.360
5	0.292	0.047	0.160	0.409
6	0.262	0.048	0.163	0.385
7	0.237	0.020	0.160	0.292
8	0.320	0.039	0.222	0.426
9	0.277	0.033	0.191	0.386
Average/ Max/Min	0.264		0.122	0.426

Table 7.2 Measurements of the torque required to rotate a M6 flanged nut with transverse vibration at a 4 kN preload.

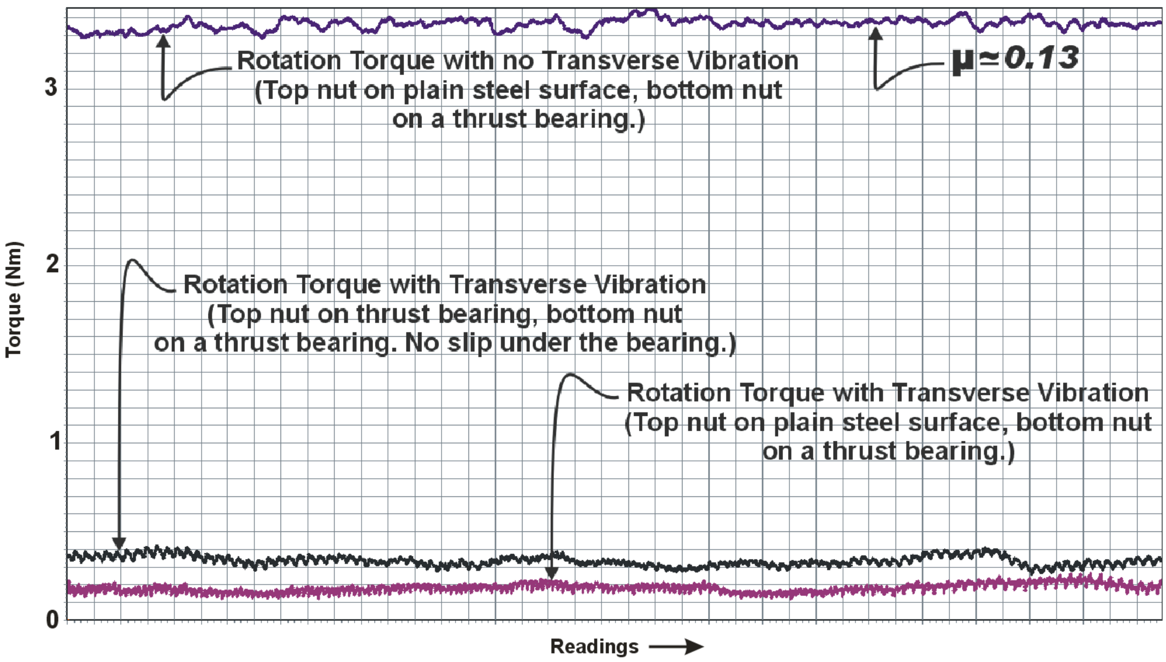

Figure 7.10 Torque required to rotate a M6 flanged nut with and without transverse vibration at a 4 kN preload (friction values are the range of measured values).

As can be seen from figure 7.10, it was found that the torque needed to rotate the nut under conditions of transverse slip with a thrust bearing on the lower nut was lower than that needed to rotate the assembly with a thrust bearing top and bottom

From equation 4.2 the value of the effective bearing diameter of the nut D_e can be determined for the size of nut used in these tests d_i = 8.8 mm and d_o = 13 mm, giving:

$$D_e = \frac{d_o + d_i}{2} = \frac{13 + 8.8}{2} = 10.9\ mm$$

The torque needed to rotate the bearing at the bottom of the threaded rod will be assumed to be half the value used to rotate the threaded rod assembly with bearings top and bottom. Hence based upon the average value from table 7.1:

Bearing drag torque = 0.301 /2 = 0.150 Nm

Using the average torque value from table 7.2, the torque needed to rotate the nut = 0.264 – 0.150 = 0.114 Nm

The apparent bearing head friction in the circumferential direction when the nut is experiencing transverse slip can be determined using:

$$\mu = \frac{2T}{FD_e} = \frac{2x0.114}{4x10.9} = 0.0052$$

Based upon mean torque values from each test shown in table 7.2, the minimum friction value is 0.0015 and the maximum 0.0078.

7.4. DEVELOPMENT OF THE ANALYTICAL MODEL

In the previous chapter a model of the loosening process was developed showing the magnitude of the loosening torque acting on a threaded fastener when the joint was subjected to transverse slip. This chapter will consider the magnitude of the friction in the circumferential direction whilst under such conditions. This model is based upon the one presented in chapter 6, the objective being to measure the coefficient of friction in the circumferential direction whilst the fastener sustains transverse slip.

If $\mu_{totalslip}$ is taken as a reference coefficient for characterising the overall friction behaviour of a bolt/nut assembly under transverse slip conditions (the coefficient of friction for the nut face is taken to be the same as the coefficient of friction of the threads). Substituting the value of $\mu_{totalslip} = \mu_{ts} = \mu_{ns}$ into equation 6.14 gives:

$$\frac{3FP}{4\pi} > F\left[\mu_{totalslip}\left(\frac{d_2}{\cos\alpha} + \frac{D_e}{2}\right)\right] + T_{Ps} \qquad (7.1)$$

The value of $\mu_{totalslip}$ that would lead to self-loosening can be determined for a particular fastener size by considering when the two terms above are equal. Re-arranging equation 7.2 to give the value of μ_{loosen}, the maximum friction value that would lead to self-loosening when the joint is being subjected to transverse slip leads to:

$$\mu_{loosen} = \frac{\dfrac{3P}{4\pi} - T_{Ps}}{\dfrac{d_2}{\cos\alpha} + \dfrac{D_e}{2}} \qquad (7.2)$$

Plain nuts were used in this series of tests with T_{Ps} equal to zero. For the nut and bolt dimensions used in these tests, the value of μ_{loosen} is 0.032 (i.e. a value of friction equal to or less than this value will lead to self-loosening).

If a nut is tightened on a joint being subjected to transverse slip, for the nut to rotate the torque will have to overcome the frictional forces (given by equation 6.13) and the torque acting in the loosening direction (given by equation 6.12). Combining these two equations and using $\mu_{totalslip}$ to establish the torque needed to tighten a nut under transverse slip conditions T_{slip} gives:

$$T_{slip} = F\left[\frac{3P}{4\pi} + \mu_{totalslip}\left(\frac{d_2}{\cos\alpha} + \frac{D_e}{2}\right)\right] + T_{Ps} \qquad (7.3)$$

Re-arranging equation 7.2 to give the value of $\mu_{totalslip}$ in terms of the other values:

$$\mu_{totalslip} = \frac{\dfrac{\left(T_{slip} - T_{Ps}\right)}{F} - \dfrac{3p}{4\pi}}{\dfrac{d_2}{2\cos\alpha} + \dfrac{D_e}{2}} \qquad (7.4)$$

7.5 DISCUSSION

What is initially surprising from the results presented in figures 7.5 and 7.6 is that the torque range is not between zero and an upper limit but rather varies between two positive limits. That is, there is a loosening torque that persists to act on the fastener that is approximately proportional to the preload throughout the transverse movement cycle. Figure 7.6 plots the average values for the upper and lower loosening torque relative to that predicted by equation 6.12. The measured lower torque values are close to that required to stretch the bolt (which always acts in the loosening direction). Although the results are not fully conclusive, the upper torque value is close to that predicted when the bolt stretch torque is added to the thread sliding torque predicted by Sakai (2001). This implies that the apparent friction between the nut and the joint surface must be close to zero throughout the transverse slip cycle. If this were not the case, the loosening torque would be likely to decrease to a low value. The tests completed to measure the torque needed to rotate a nut, without tightening it, under conditions of transverse slip indicates that the friction in the circumferential direction is exceptionally small. This is one explanation as to why the loosening torque does not cycle between zero and a maximum value. The frequency of the variation in the torque value is the same as that of the transverse joint movement.

The results shown in figure 7.8 indicate that exceptionally low friction in the circumferential direction is present between the fastener and the joint when transverse joint movement is occurring. Figure 7.9 shows a boxplot summarising tests on 15 fasteners. From these results the following observations are made:

1. The lower friction value based upon lower torque limit measured is broadly constant and is typically below the friction value needed to cause self-loosening to occur (0.032 for this size and geometry of fastener).

2. The upper friction value is typically greater than that needed to cause self-loosening and is larger at a low bolt preload than it is at higher preload values.

3. At preload values of 2 kN and above, the upper friction values measured are substantially lower than those observed when no transverse movement is occurring.

4. During part of the transverse slip cycle, the friction value is greater than that which would cause self-loosening. This would imply that nut rotation only occurs during part of the transverse slip cycle.

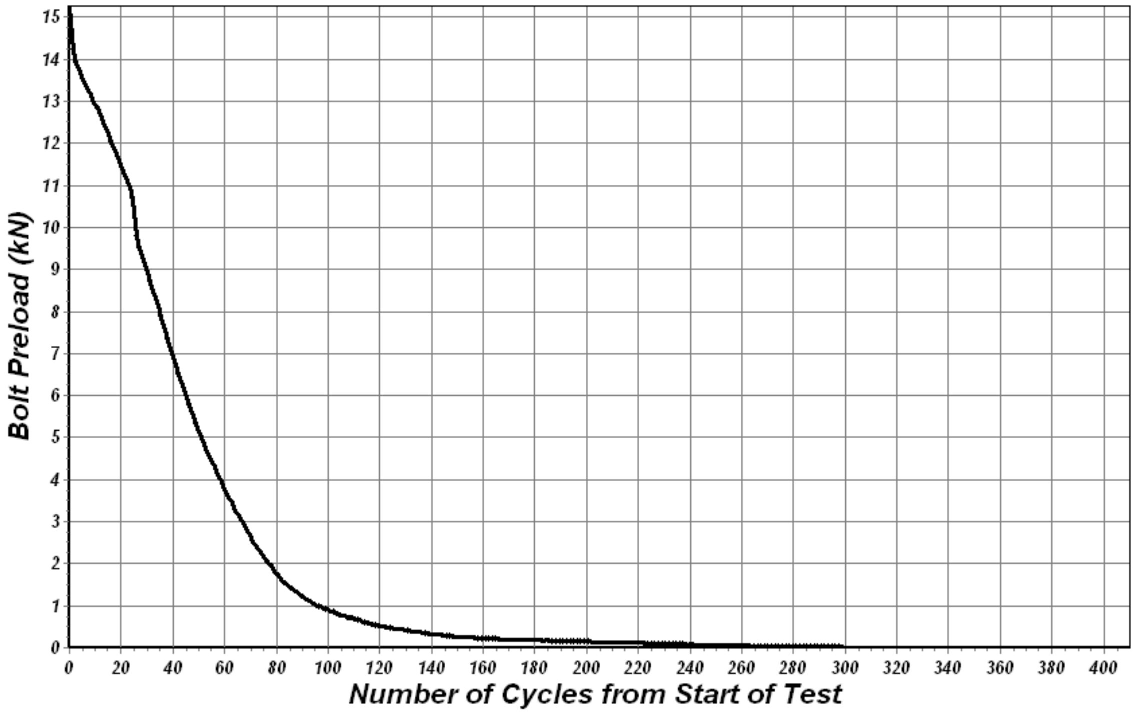

Figure 7.11 Preload decay curve for a M8 plain nut

The torque fluctuation measured in the loosening and tightening experiments under transverse vibration did not occur in the nut bearing face torque tests. This observation points towards the conclusion that the cause of the torque fluctuation is movement in the threads. The apparent friction values in these tests of 0.0015 to 0.0078 were typically significantly lower than those observed in the tightening tests. The friction computed in the tightening tests was an average value of the thread and nut bearing face friction values. This points towards the circumferential friction coefficient in the threads being greater than that on the bearing surface during conditions of transverse slip.

A typical result from a Junker test performed on an M8 plain nut is shown in figure 7.11. Initially there is a very rapid drop off in preload that occurs as a result of embedding loss (which is plastic deformation of the surface irregularities of the thread and nut face contact areas). Following this, the rate at which preload is lost per cycle is approximately constant. At around 2 kN, the rate of preload loss begins to decrease such that it takes twice the number of cycles to drop the remaining 2 kN as it did from 15 kN to 2 kN. This can be related to the results shown in figure 7.9. As the preload decreases below the 2 kN mark, the proportion of the transverse cycle that the friction value is below the critical self loosening value decreases. This would directly influence the self-loosening rotation rate and hence the rate at which the bolt preload decreases.

The tightening tests were completed to investigate the magnitude of the circumferential friction of the fastener when the joint is undergoing transverse movement. As can be seen from figure 7.7, the torque needed to tighten a fastener to a preload value is substantially smaller than is the case when no transverse movement is present. A cyclic torque variation is displayed during the tightening due, it is thought, to the transverse movement causing the thread sliding torque to vary. The torque variation is greater than that anticipated from the theory presented. For example at a 10 kN preload the average difference between the peak torque and lower torque measured was 3 Nm, whereas the torque variation anticipated from theory would be in the order of 1 Nm. One explanation of this is the inadequacy of the present theory. The assumption made in section 6.4 in the derivation of the loosening torque T_{ss} (arising from differences in the resistance offered between the ascending and descending sides of the nut thread) that the nut and bolt threads were perpendicular to each other. The assumption is illustrated in figure 7.12. In practice, when a joint is subjected to transverse movement, some degree of bolt bending occurs so that the nut and bolt threads do not remain perpendicular. The bending of the bolt will also result in the surface pressure increasing on one side of the nut surface. This is illustrated in figure 7.13. The bending of the bolt will result in an angular difference between the bolt and nut threads and subsequently a change in the load distribution and the interaction between the threads under slip conditions. This will in turn affect the loosening torque not explained in the equations presented in section 6.4.

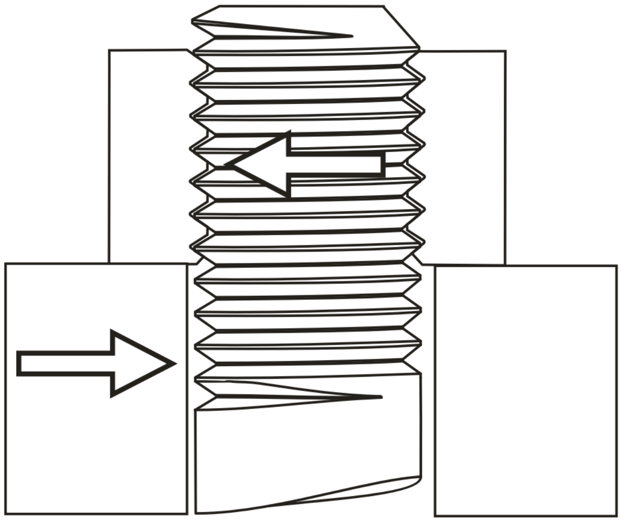

Figure 7.12 Bolt and nut threads remaining perpendicular during transverse slip

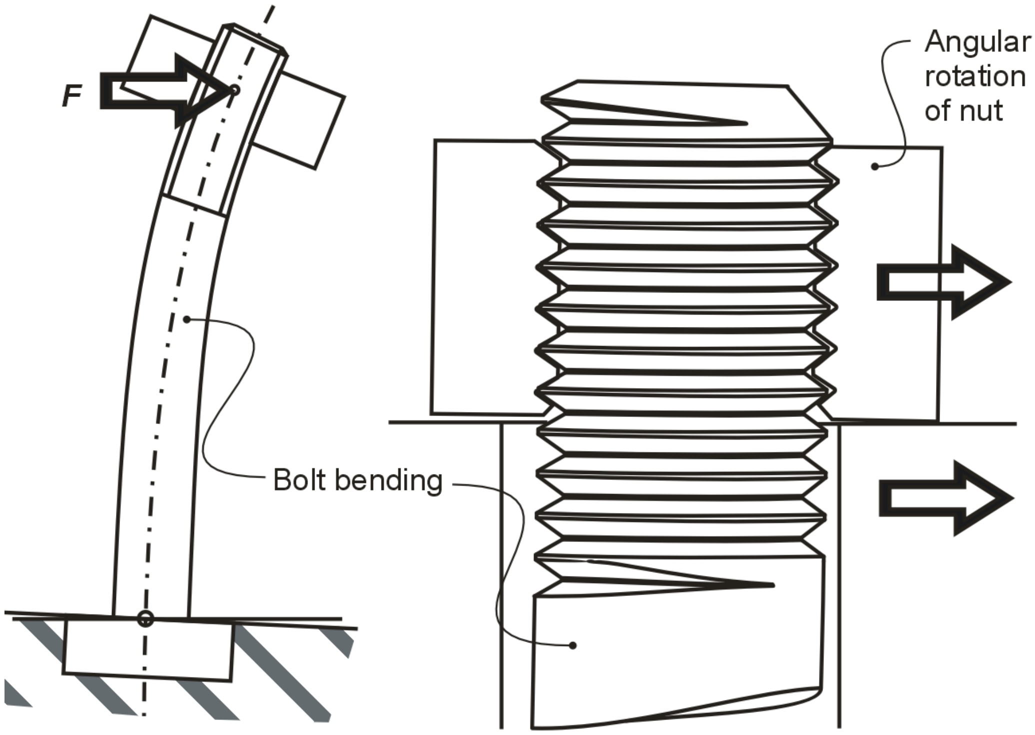

Figure 7.13 Bolt bending causing angular rotation of the nut

7.6 CONCLUSIONS

The main conclusions obtained in this study can be summarised as follows:

1. The loosening torque measured under transverse joint movement is close to that theoretically predicted. The lower limit of the torque measured at each cycle is close to the bolt stretch torque. This torque is preload dependent and is normally resisted by friction in the threads and on the nut face. The peak torque is the addition to the stretch torque of the torque generated as a result of thread movement.

2. The findings of these tests are in broad agreement with the earlier work of Sakai (1978; 2001). Under conditions of transverse slip very low values of friction in the circumferential direction are measured. However what was found in this study was that the friction value varied within the vibration cycle. At lower levels of preload the proportion of the cycle in which the friction was lower than the critical level decreased. Such a decrease would slow the rate of loosening that is typically observed in a Junker vibration test.

3. It can be inferred from the results that the nut is not in continuous rotation during the loosening process. Typically it is only during part of the transverse movement cycle that friction is at a low enough level for self-loosening to occur. It can also be inferred that as the bolt preload decreases there is a smaller proportion of the cycle when friction is below the self-loosening critical level. For these tests, at low preload levels (<= 1 kN), the upper friction value approximately corresponds to that present for the static case. (When there is no transverse joint movement).

The results indicate that head friction in the rotation direction is lower than that of the thread under transverse slip conditions. The results in this study also indicate that the head friction was relatively constant whereas the thread friction varied within the vibration cycle. Izumi et al (2005) suggested, based upon FE modelling, that complete thread slip is achieved prior to bolt head slip and torsion is induced into the bolt shank. However, if as this present study indicates, the friction is less under the head, then this is unlikely to occur.

8. CONCLUSIONS

8.1 INTRODUCTION

Threaded fasteners are used in almost every engineering product, but their complexity in the detail of their operation is often under-estimated. This complexity is due in part to the multifaceted nature of friction. The preload being imparted into a joint depends upon two thread surfaces being rotated on each other under load and the frictional resistance sustained. The vast majority of joints rely upon this preload for their structural integrity.

Friction plays a central part in the use of threaded fasteners. Without friction, threaded fasteners would not retain the preload; the nut would spin off after tightening when the socket was removed. The conversion of the torque applied to the nut into preload is largely controlled by friction.

Two of the significant issues involved in the use of threaded fasteners have been investigated in this study. The first part of the study investigated the relationship between torque and preload. In particular, a study was made of the changes in the frictional forces that occur with the re-use of electro-zinc plated fasteners. The second and major part of the study, considered the self-loosening of threaded fasteners.

8.2 DETERMINATION OF THE FRICTION COEFFICIENT OF FASTENERS

Experiments to determine the friction coefficient of fasteners indicated that a change in the friction between the contact surfaces of the bolt/nut thread and nut face occurred upon repeated tightening of fasteners with a zinc electro plated finish. This caused a reduction in the preload when the fastener was tightened to a specified torque value. The reduction in the preload was found to be significant, the preload on the sixth re-tightening typically being half that obtained on the first tightening. Such a reduction would have an adverse effect on the structural integrity of the assembly.

Higher levels of friction were noted for the threads than for under the nut face. This is thought to be due to several factors. Thread friction is somewhat complicated by the non-linear pressure distribution on the thread surfaces and this non-linearity is itself partially a function of the friction value. The higher pressures that are present in the thread lead to increased wear particle generation and ploughing. The enclosed nature of the mating threads also results in wear particles, created at the surface closest to the joint face, which is under the highest pressure, being moved further up the thread as the nut rotates. Energy dissipation also arises during the tightening process due to thread deformation. In such circumstances, besides surface friction there will be a component of the overall friction force due to work deforming the thread. The elastic-plastic properties of the steel will influence the magnitude of this effect and hence is likely to be higher on lower strength fasteners than those of higher strength. The strength of the nut relative to that of the bolt will also influence this effect. It is known that thread deformation is less when a high strength nut is used (Alexander, 1977). The higher strength nut assists in resisting thread bending, which in turn improves the resistance to thread stripping. The relative rigidity of the nut and bolt threads also has a further influence. An implicit assumption in the determination of the friction coefficient is that the flank angle is a constant 60 degrees. The distorted pressure flank of the threads nearest the joint surface that was observed in this study will increase the frictional force even if the friction coefficient remained constant because an increase in the flank angle will have a greater the wedging effect. Effects such as those just described are not adequately covered in present approaches to the computation of the torque-tension relationship of fasteners.

Although the magnitude of the thread friction coefficient is higher than that which is under the nut face, the increase in the nut face friction follows a similar increasing trend on re-tightening becoming approximately constant after five or six tightenings. It was also noted that the magnitude of the thread friction coefficient became dependent upon the pressure (clamp force) as the number of tightenings increased. The mechanism for this effect is not completely understood, but is thought to be due to the increase in pressure causing a change from wear particle ploughing to adhesive friction.

Tests completed to determine the nut face friction coefficient under a constant loading clearly indicated that a change occurs in the friction characteristics after a certain amount of angular movement. By using the mean nut radius, this angular movement was converted to a linear movement. The average distance prior to a significant change in friction occurring was 195 mm. The initial average friction coefficient of 0.13 is less than the mean value noted in the tightening tests (mean value on first tightening of 0.19). However the scatter in the results on the tightening tests observed on the nut face tests under constant loading was less than that observed in the full tightening tests.

Based upon these results, the reuse of unlubricated EZP threaded fasteners cannot be recommended. The increase in friction resulting from reuse could have an adverse effect on a joint's structural integrity due to the corresponding decrease in the generated clamp force that occurs. EZP is a common type of finish used on fasteners.

Many prevailing torque nuts for example use such a finish. The present tests on re-use of such nuts, such as ISO 2320 (British Standards, 2008), assess the prevailing torque characteristics with re-use but not the effect that such re-use will have on friction and subsequently the bolt preload.

8.3 LOOSENING OF FASTENERS BY TRANSVERSE VIBRATION

The loosening of threaded fasteners is a complex phenomenon. Some degree of preload reduction usually occurs when a joint is placed under an external load as a result of non-rotational loosening involving localised collapse and plasticity of the contact surfaces. For a fastener to come completely loose however, usually involves some degree of self-loosening i.e. fastener rotation. Previous research indicated that transverse vibration is the principal cause of self-loosening. In this study a large number of tests have been performed to study the loosening characteristics of various types of fasteners. The focus was on prevailing torque nuts, a common type of lock nut. The tests confirmed the findings of previous research in that under transverse vibration this type of nut initially rotates until a relatively small asymptotic preload value is reached. It was also found that all the prevailing torque nuts tested had similar loosening characteristics.

Novel investigations were conducted into the loss of prevailing torque when a fastener was reused. The present standard test code defined in the ISO 2230 standard (British Standards, 2008) is applicable for a joint being deliberately disassembled for maintenance or other purposes. In some industries, for example the rail industry, the practice is to retighten nuts that are found loose. Self-loosening involves relative movement of the threads. This movement could be anticipated to have a greater effect on the efficacy of the prevailing torque feature than normal untightening. This was indeed found to be the case. Based upon these findings, it would be wise to discard any nut found to have self-loosened and replace with new following an investigation into the cause of self-loosening. The finding that nylon insert nuts performed better than the all-metal type of nut is of economic interest. A non-metallic insert nut costs typically between 10 to 40 times less than the all-metal type of nut. The reason for the better performance may be that the thread movement caused elastic movement of the locking element rather than wear and plastic deformation that would be likely to occur with the all-metal type of nut.

8.4 COMPLETE SELF-LOOSENING OF PREVAILING TORQUE FASTENERS

The structural integrity of the majority of bolted joints is dependent upon achieving and maintaining sufficient preload. If the preload is insufficient to prevent joint slip from applied shear loads, self-loosening of the fastener becomes likely. Prevailing torque nuts are used to limit the loosening and ensure that the nut does not become detached from the bolt. What this study has revealed is that if an axial load is also acting on a joint which is experiencing transverse slip, prevailing torque nuts can continue to self-loosen leading to their possible detachment from bolts.

It is the tensile load in the bolt that generates the loosening torque under transverse slip conditions. The tensile load can be from the initial preload or an axial loading. The other effect of the transverse slip is to apparently reduce the frictional resistance in the circumferential direction to a very low value. These two effects are responsible for self-loosening.

On the majority of assemblies, fasteners are tightened prior to the application of an external load. The bolt acts as a tension spring and the joint a compression spring, the tensile and compressive loads in the spring system balancing each other. When the joint sustains an external load, the bolt is stretched by only a small additional amount since the joint stiffness is usually far greater than the bolt stiffness. Subsequently, the bolt usually only sustains a small proportion of the external axial load. This assumes that the external load does not completely relieve the compression on the joint from the preload. If the compression is completely relieved, joint separation occurs and the bolt sustains the full magnitude of the external load. The mechanism by which prevailing torque nuts can become detached from bolts can be explained as follows. Under the appropriate conditions of transverse slip, self-loosening will occur reducing the preload to a fraction of its original value. If an external axial load is also acting once the preload reduces to a value so that joint separation occurs, it is axial load that will drive the loosening process. As long as the axial load is maintained and transverse slip is occurring, the nut will continue to rotate until it becomes detached.

The study reported in this thesis also sheds some light on why detachment of the nut from the bolt can readily occur. Frequently on a standard Junker test, once a plain nut has come loose it stops rotating as the absence of the preload eliminates the torque driving the loosening process. In practice the presence of corrosion would sometimes inhibit rotation once the preload is eliminated. However, the detachment of plain non-locking nuts is commonplace. Often such detachments have been put down to 'vibration' and some work has been completed on this aspect (Hess and

Sudhirkashyap, 1997). This thesis, however, proposes another explanation, plain nuts in the presence of transverse vibration and axial loading will readily become detached. The presence of even a small constant or intermittent axial load acting on the joint in the presence of transverse slip will lead to the continued rotation of plain nuts until they detach.

8.5 THE EFFECT OF TRANSVERSE VIBRATION ON THE FRICTION FORCES ACTING ON THREADED FASTENERS

Considering that friction effects are central to the self-loosening of threaded fasteners, it is surprising that other researchers have not studied this aspect in more detail. The apparent reduction in frictional resistance in the circumferential when transverse slip is experienced and the torque in the untightening direction exerted on the nut under such conditions, are the key to the understanding the self-loosening phenomenon.

One surprising aspect of the tests was that the loosening torque range under transverse slip conditions was not between zero and an upper limit, but varied between two positive limits. That is, there is a loosening torque that persists to act on the fastener that is approximately proportional to the preload throughout the transverse movement cycle.

The results also indicated that exceptionally low apparent friction in the circumferential direction is present between the fastener and the joint when transverse joint movement occurs. It was found that the frictional resistance varied during the transverse slip cycle. The lower friction value based upon the lower torque limit measured was broadly constant and was typically below the friction value needed to cause self-loosening to occur. The upper friction value was typically greater than that needed to cause self-loosening and was larger at a low bolt preload than at higher preload values.

At preload values of 2 kN and above, the upper friction values measured were substantially lower than those observed when no transverse movement occurred. During part of the transverse slip cycle, the friction value was greater than that which would cause self-loosening. This would imply that nut rotation only arises during part of the transverse slip cycle.

The torque fluctuation measured in the loosening and tightening experiments under transverse vibration did not occur in the nut bearing face torque tests. This observation points towards the conclusion that the cause of the torque fluctuation is movement in the threads. The apparent friction values in these tests of 0.0015 to 0.0078 were typically, significantly lower than those observed in the tightening tests. The friction computed in the tightening tests was an average value of the thread and nut bearing face friction values. This points towards the apparent friction in the circumferential direction in the threads being greater than that on the bearing surface during conditions of transverse slip.

8.6 APPLICATION OF THE RESULTS OF THIS RESEARCH

Key points relating to the application of this research include:

- The re-use of unlubricated fasteners with an EZP coating is not advisable for critical applications. Since it is normal practice for the fasteners to be tightened to a specific torque value, the preload achieved will reduce if the fasteners are re-used. The addition of a suitable lubricant, such as molybdenum disulfide, will stabilise the friction coefficient so that a similar preload would be achieved when the fasteners are re-tightened.
- A key finding of this research is how prevailing torque nuts can become detached from bolts. They can no longer be defined as loss prevention devices because their locking performance depends upon the loading conditions to which they are subjected. A modification of the DIN 65151 test is needed so that the effect of axial loading can be included. Design Engineers should also consider the loading conditions that joints will be subjected to prior to specifying torque prevailing nuts.
- The investigations completed on the loosening torque being applied to fasteners and the friction forces present in the circumferential direction whilst they are subjected to transverse vibration will be useful in developing understanding as to why fasteners self-loosen.

9. FURTHER WORK

9.1 INTRODUCTION

Although threaded fasteners can be considered a mature technology there are significant problems with their reliable use. This in part arises from the inability to inexpensively measure the fastener preload in an assembled joint. The structural integrity of the majority of bolted joints depends upon the preload generated by the tightening process achieving a certain minimum value. As a consequence of the difficulties in measuring the preload, other approaches are introduced such as controlling the torque applied to tighten the fastener. As discussed previously, the applied torque only controls the fastener preload indirectly. To achieve some degree of control over the preload using the applied torque, the friction conditions need to be known. The first part of this study investigated friction related to tightening of fasteners. Later the friction forces acting on a fastener experiencing transverse slip were investigated.

9.2 EFFECT OF RETIGHTENING FASTENERS ON THE FRICTION COEFFICIENT

In a study completed on retightening wheel fasteners Morgan and Henshall (1996) demonstrated a substantial decrease of up to 60-70 percent in the preload for degreased fasteners in self-finish after 5 re-tightenings compared to the as-received performance. The study showed that full recovery of the preload on subsequent tightenings could be achieved by applying engine oil. Wheel fasteners typically have a self-finish i.e. have no coating. An investigation into whether the application of lubrication would recover the first time tightening characteristics of coated fasteners would assist in developing guidelines on reuse.

During this study a number of fasteners were tested having a zinc flake coating (Geomet). It was observed that a change in the frictional characteristics occurred with fasteners having this type of finish. These fastenings are commonly used in automotive applications and further work on other fastener finishes would be of interest.

As part of this study, a series of tests were completed by the author on the effect of lubricants in the repeated use of EZP threaded fasteners with the findings being presented at LUBMAT 2006 (Eccles, 2006). The results indicated that the use of certain lubricants, such as molybdenum disulphide paste, lithium grease and Copaslip, could prevent friction increase on re-tightening. The use of oil did not prevent an increase in friction, the friction coefficient approximately doubling after the tenth cycle. Re-use of stainless steel bolts indicated that a factor of three increase occurs. Even the use of molybdenum disulphide paste only moderated the friction increase, but did not prevent it. Whilst many equipment manufacturers advocate that new fasteners must be used if their product is disassembled, fasteners are widely re-used, especially if made in stainless steel. Investigation into appropriate dry lubricant coatings that will allow fastener reuse without a decrease in properties would be economically and environmentally beneficial.

9.3 LOOSENING OF PREVAILING TORQUE NUTS

Prevailing torque nuts have made an important contribution to fastener security since they eliminate the possibility of the locking element being omitted as can be the case with lock washers. However this study has shown that they are not effective under certain conditions.

Tests have been conducted on an all-metal variety of prevailing torque nut. Beside the non-metallic insert nut, many other types of locking devices have a prevailing torque feature. Adhesive locking involves the application of a liquid to the threads that cures to form a void filling solid that subsequently restricts relative movement. Adhesive having too high a bond strength has the consequence that disassembly is hindered with the nuts often having to be cut off. Also high bond strength adhesives tend to exhibit high friction values. Such high strength adhesives, called "studlocker" adhesives, are often used to lock studs in place. However, when they are used on bolt threads the preload is restricted, relative to what could be achieved, due to the high torsional stresses induced due to the higher friction. Owing to these factors, medium strength adhesives are frequently used. Under a conventional Junkers test, such medium strength threadlockers exhibit self loosening characteristics similar to prevailing torque nuts. Investigating the effect of axial loading and transverse vibration on adhesive locking and other types of prevailing torque fasteners could extend this study. Prevailing torque features are also applied to bolts, such as a fused polymer patch applied to the thread, such fasteners behave in a similar manner to prevailing torque nuts on a conventional Junker test.

Prevailing torque nuts retain less preload on a transverse vibration test than would be anticipated. The expectation is that self-loosening would stop when the loosening torque was equal to the prevailing torque. Typically this is not the case, a lower preload being retained than would be expected by the statically measured prevailing torque. The static measurement of the prevailing torque (without transverse vibration) may be the cause of the anomaly. The magnitude of the prevailing torque may reduce under transverse vibration conditions. The relative movement of the threads may influence the pressure being applied by the nut to the bolt thread by the prevailing torque feature and hence the torque being generated. To assess if this hypothesis is true, a comparison is needed between the prevailing torque measured statically with that under transverse vibration conditions. This will not necessarily be easy to complete since a test method would need to be developed that could measure the prevailing torque under such conditions.

The current study has revealed a need for the development of a modified standard to assess the loosening characteristics of threaded fasteners. The present Junker test as defined in the DIN 65151 standard (Deutsche Norm, 1994) has been found to be inadequate to fully loosen prevailing torque nuts. The development of an updated standard is needed that could evaluate the effect of axial loading in addition to transverse slip on the resistance to loosening of fasteners. Presently fastener companies promote the results of a Junker's test as proof that a fastener will retain preload under vibration. This study has revealed that this is not always the case. To develop an improved standard, further work is needed to investigate the level of axial loading appropriate for a given size of fastener.

9.4 FRICTION FORCES ACTING UNDER TRANSVERSE SLIP

The torque fluctuation that arises when tightening the fastener, when transverse movement is occurring, is higher than that which current theory predicts. One possible explanation for this is that there is a direct shear force acting on the fastener when the torque is applied via a wrench. This shear force could influence the results. The tests should be repeated with a pure torque being applied to assess if the torque fluctuation reduces.

The results from the current study point towards the nut rotation being discontinuous during the transverse movement cycle as the friction value fluctuates above and below the critical self-loosening value. Future work should include the measurement of this rotation during the transverse movement cycle and determination of how the nut rotation varies with each stage of the vibration cycle.

As the static friction coefficient increases, so does the resistance of a fastener to self-loosening. Further work should be done to measure the apparent friction coefficient under transverse slip for different fastener finish (friction) conditions. It is possible that as the static friction coefficient increases, the period of the transverse movement cycle when the apparent friction dips below the critical level decreases.

A test should be devised to allow the determination of both the nut face and thread friction coefficients under transverse slip conditions. It is known that the pressure distribution under the nut face is non-linear (Cullimore and Eckhart, 1972). The highest pressure is next to the hole and decreases towards the outer face of the nut. Further understanding of the loosening process would be gleaned by measuring the pressure distribution under the nut during transverse slip conditions. Due to the side loading it is anticipated that the pressure distribution would be asymmetric under such conditions.

9.5 SELF-LOOSENING AND FATIGUE FAILURE

The two common failure modes of threaded fasteners under transverse vibration are self-loosening and fatigue. In general, if the fastener does not self-loosen when subjected to transverse vibration and the bending stress in the bolt is sufficiently high then fatigue will occur. The ‘critical slip’ concept which applies to non-locking fasteners was discussed in section 2.7.4. On fasteners having some locking facility, slip which would self-loosen non-locking fasteners can be sustained. The bending stress in the bolt will be increased when larger joint slip is encountered. The machine used by the author had a transverse amplitude of +/- 0.65 mm, M8 bolts would usually sustain a fatigue failure between 1000 and 3000 cycles. A graph illustrating fatigue failure is shown in figure 9.1.

In the test there was no self-loosening of the nut, preload loss was as a result of non-rotational loosening. Initially there was a rapid reduction in the preload. This was probably due to a combination of embedding loss as a consequence of flattening of the surface roughness and localised plastic deformation in the threads. The gradual loss of preload that followed the initial rapid drop was probably as a result of continued plastic deformation in the bolt thread. The rapid drop off in the preload that occurs at approximately 1600 cycles is as a result of the fatigue crack propagating through the bolt. Complete fracture occurs at 1700 cycles.

The thread in the nut used had an asymmetric thread form that enabled the bolt thread crest to contact the nut thread on a 30 degree ramp rather than the conventional 60 degrees. As the bolt was tightened the bolt thread was pulled onto this ramp preventing relative movement under vibration. On tests performed on this type of thread and another similar thread form (the ‘Bump thread’), prevention of thread movement would appear to lead to a reduction in the fatigue life. This may be as a result of the locking method inducing higher bending stresses into the bolt. If thread movement under transverse movement can occur the bending stresses would decrease since the bending moment would be reduced due to the end restraint being less.

The common failure modes for bolts in joints undergoing transverse slip are self-loosening and fatigue failure. Investigation into how the thread profile could be optimised to provide self-loosening resistance whilst maintaining a high resistance to fatigue would be a useful.

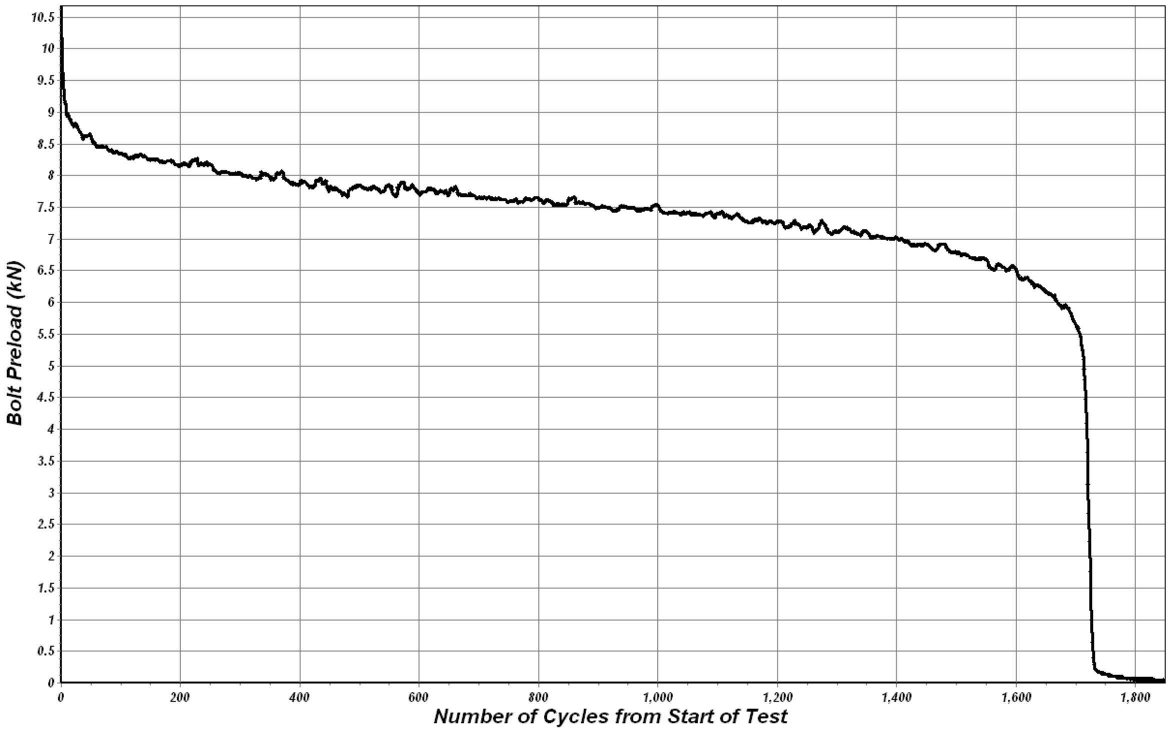

Figure 9.1 Fatigue failure of a M8 bolt with a nut having an asymmetric thread form

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The Effect of Lubricants on the Repeated Use of Threaded Fasteners

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ABSTRACT

Threaded fasteners are used widely in critical applications in engineering plant and equipment world-wide. They are usually supplied dry, with a surface finish applied for corrosion protection purposes but without any lubricant being present. Practice varies between industries and companies on the use of lubricants with fasteners. Due to a lack of availability, or for economic reasons, fasteners are often re-used following disassembly of a joint. To understand the impact of differing practice, this study reports on the effect that a range of lubricants (including oil, lithium grease and molybdenum disulphide) can have on the repeated use of fasteners compared to fasteners without a lubricant applied. The study focuses on fasteners having an electro-zinc plated (EZP) finish that are commonly used across a wide range of industries. Results of torque-tension tests, used to determine friction coefficients during tightening, are reported together with the effect that repeated use, with and without, lubricants has on the friction coefficient.

A significant reduction (54%) in the bolt preload occurs between the first and tenth tightening of an EZP bolt tested in the "as-received" condition with no lubricant applied; with a 51% reduction occurring between the first and fourth tightenings. This contrasts with the cases where lubricants were used where there was no significant reduction in the preload values between the first and tenth tightening. These results indicate that the use of an appropriate lubricant can ensure that a consistent preload is achieved from bolts re-tightened to a constant torque value. The effect of using a dry lubricant on stainless steel also resulted in a consistently higher preload when the fastener was reused compared to the unlubricated condition.

This study shows that there exists a potential for large economic savings if lubricants are employed on re-used fasteners. Presently when parts are removed from assemblies for repair or servicing, OEM's typically recommend using new fasteners. By ensuring that a consistent preload is achieved when the fasteners are re-tightened, the use of lubricants could allow fasteners to be reused in many instances.

Keywords: Fastener, Lubricant Testing, Bolt, Nut, Preload, Coefficient of Friction

1. INTRODUCTION

Replacing parts on most items of mechanical equipment requires the removal and subsequent re-tightening of threaded fasteners. It is the fastener's preload (the clamp force that is generated when the fastener is tightened) that ensures the structural integrity of the joint. The preload clamps the joint together and prevents relative movement between its constituent parts. Insufficient preload generally results in the joint moving and the fasteners failing by fatigue or self loosening and subsequently being dislodged leading to significant problems [1-4].

In the majority of applications, the preload provided by the fastener is indirectly controlled by specifying a tightening torque. The relationship between the applied torque and preload during the tightening process is linear and so increasing the torque value leads to a proportional increase in the preload. To provide guidance and a measure of quality control, many manufacturers provide the appropriate tightening torque to which specific fasteners should be tightened on assembly drawings and in their maintenance handbooks, . Such torque specifications are usually based upon values quoted in a table that was compiled on the basis of knowing the friction conditions present in the fastener derived from tests.

It is not widely appreciated that re-tightening a threaded fastener can significantly affect its performance. Damage to the fastener's surface, incurred during the tightening process, affects the friction characteristics that prevail under the nut face and the bolt head. If the friction increases as a result of surface damage sustained during previous tightenings then the fastener's preload will decrease for a given tightening torque. It is usual to specify a single tightening torque in a maintenance handbook, independent of the number of times the fastener has been tightened. If the performance of the product was proven based upon the preload achieved on the fastener's first tightening then there is the risk of structural integrity problems if a decrease in preload occurs as a result of an increase in friction due to the fastener being re-tightened.

Many OEM's specify that if a part is removed, the fasteners should be replaced. However in many applications this is not feasible for economic or practical reasons. The market for threaded fasteners is considerable. According to the Freedonia Group [5], the world-wide demand for fasteners in 2006 is expected to be US$46.3 billion. The policy of disposing of fasteners that are capable of providing a satisfactory function can be regarded as wasteful in both in economic and environmental terms.

There has been little published work on the influence that re-tightening has on the performance of threaded fasteners. One paper by Morgan and Henshall [6] reports on the effect of re-tightening M22 x 1.5 nuts on commercial vehicle wheels with and without a lubricant (engine oil). Their report indicated that degreasing the nuts resulted in a 70% reduction in the preload after three tightenings.

In this present study, the effect of re-tightening M12 x 1.75 electro-zinc plated (EZP) bolts and nuts on their torque-preload characteristics is investigated. Fasteners are commonly zinc plated to prevent corrosion of the underlying steel - the zinc providing both a barrier and sacrificial protection. The fasteners were not degreased in the tests but used in their 'as received' condition. The majority of the tests were performed on property class 8.8 bolts [7] however a number of tests were also completed on stainless steel bolts and nuts of property class A4-70 [8]. Stainless steel fasteners are used across a range of industries for their corrosion resistance properties and their retention of fracture toughness at low temperatures. A common issue with the assembly of such fasteners is galling in the threads that can result in problematic tightening.

2. EQUIPMENT

To allow the bolt preload to be measured, a test apparatus was designed and built. This device included a specially designed test frame, with torque, angle and clamp-force measuring instruments. These were connected to a computer via an analogue to digital converter to allow data to be collected throughout an experiment.

Nuts were "hand tightened" using an electronic torque wrench which was used for tightening the nuts had cabling that connected to the A/D converter allowing sampling of the torque-angle measurements to be made. A 12 bit analogue to digital (A/D) converter was used to sample the analogue signals from the load cell and the electronic torque wrench. The sampling rate is variable, but an acquisition speed of 10 samples per second was found to produce consistent results. The test apparatus allowed either the bolt head to be rotated with the nut held stationary or the nut to be rotated with the bolt head stationary. In the tests reported here, the nut was rotated with the bolt head held stationary. The equipment is illustrated in figure 1.

Figure 2 illustrates the torque/preload measuring device. It consists of a load cell to measure the preload and a torque transducer to measure the thread torque. A roller thrust bearing allows the thread torque to be reacted by the torque transducer. With knowledge of the overall applied torque, the thread torque and the preload, the device allows the thread and under head friction coefficients to be determined.

For one set of tests, standard property class 8.8 bolts were used with property class 8 nuts and plain washers were used under the nut face. Bolts, nuts and washers had an EZP finish, the coating thickness typically varied between 5 μm to 9 μm. A constant tightening torque of 50 Nm was used in the tests. It was found that if a higher torque was used, the bolt yield point would be reached when a high

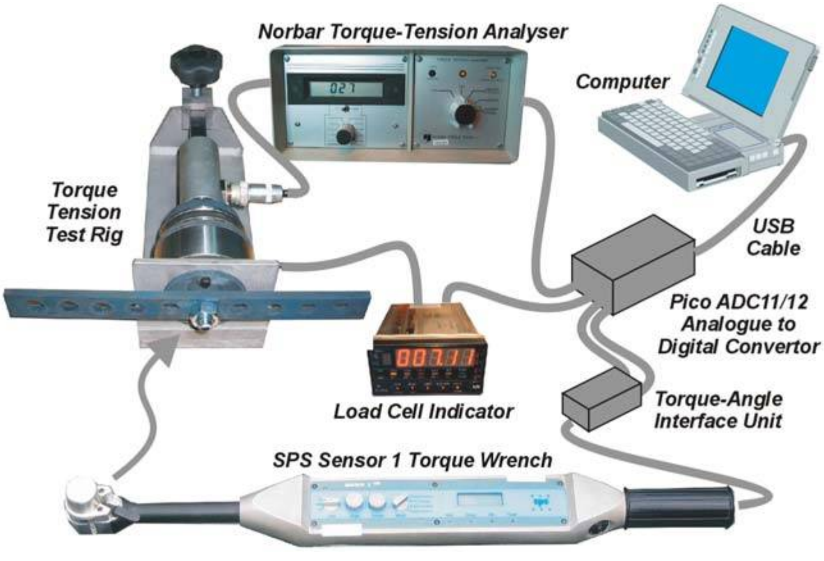

Figure 1 - Overview of the Test Apparatus

performance lubricant was used. To ensure that the nut surface rotated against the washer surface rather than the washer rotating against the support, a tabbed washer was used to prevent the washer rotating.

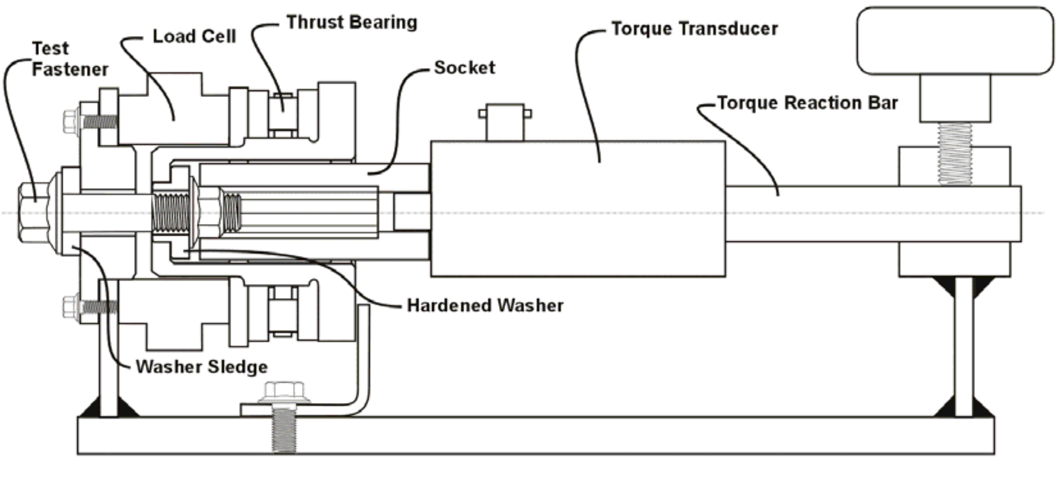

Figure 2 - Section through the fastener test apparatus

3. TEST CONDITIONS / SPECIMENS

Two series of tests were performed:

1. Tests on as received property class 8.8 EZP bolts and nuts in the following conditions:
    - As received with no lubricant applied.
    - With a molybdenum disulphide paste applied.
    - With Copaslip[1] applied.
    - With litium grease applied (Duckhams LB10 lithium grease)
    - With a light oil applied (Castrol Everyman Oil)

In each case, the lubricant was applied prior the the first tightening to the threads and the nut bearing face but was not re-applied between subsequent re-tightenings.

2. Tests on as received stainless steel property class A4-70 bolts and nuts in the following conditions:
    - As received with no lubricant applied.
    - With a dry coating of molybdenum disulfide applied via a spray prior to the first tightening.

4. RESULTS

Bolt preloads arising during multiple tightening on the EZP bolts and nuts to a torque of 50 Nm are summarized in figure 3.

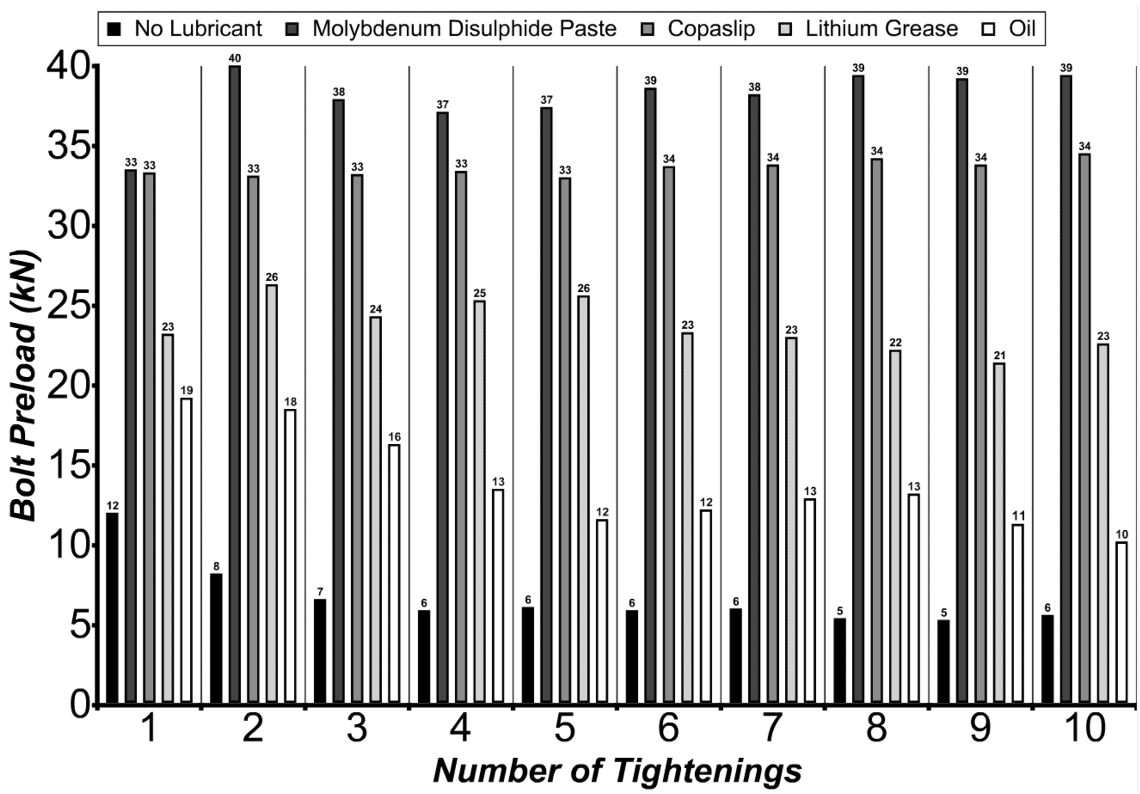

Figure 3 - The Effect on the Bolt Preload of Repeated Tightening of EZP Bolts

[1] Copaslip is a high temperature anti-seize compound that incorporates a non-melt grease with copper. It is a registered trademark.

Bolt preloads arising during multiple tightening on the stainless steel bolts and nuts to a torque of 50 Nm are summarized in figure 4.

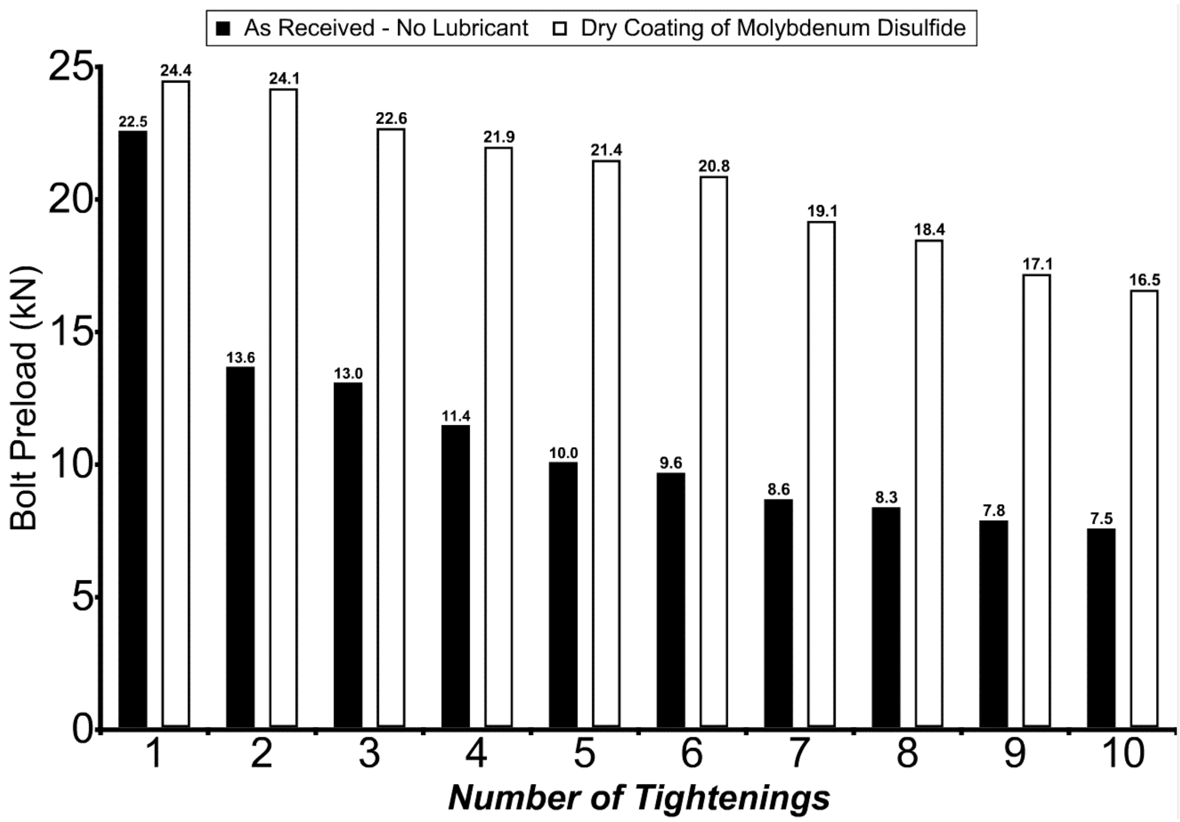

Figure 4 - The Effect on Bolt Preload of Repeated Tightening of Stainless Steel Bolts

5. MATHEMATICAL MODEL

A mathematical model relating the applied torque to the tension induced into the bolt is described by [9].

$$T = \frac{F}{2}\left[\frac{p}{\pi} + \frac{\mu_t d_2}{\cos\beta} + D_e \mu_n\right] \qquad \text{Equation 1}$$

$$\text{with } D_e = \frac{d_o + d_i}{2} \qquad \text{Equation 2}$$

where

T	Total tightening torque
F	Bolt preload
μ_t	Coefficient of friction for the threads
d_2	The basic pitch diameter of the thread
β	The half included angle for the threads
p	Pitch of the thread
μ_n	Coefficient of friction for the nut face or bolt head
D_e	The effective bearing diameter of the nut

d_o	The outer bearing diameter of the nut
d_i	The inner bearing diameter of the nut face

The three terms inside the bracket in equation 1 represent the torque resulting from the circumferential component of the normal reaction between the nut and bolt threads due to the thread's helix angle. The second term is from torque to overcome the friction between the bolt and nut threads and the final term the torque needed to overcome friction under the nut face.

As can be seen from inspection of the equation 1, if the terms in the bracket are constant during the tightening process a linear relationship exists between the applied torque and the preload. A typical graph of change in preload as torque increases from the authors' experimental data is shown in figure 3. A roughly linear relationship between torque and the bolt preload is evident.

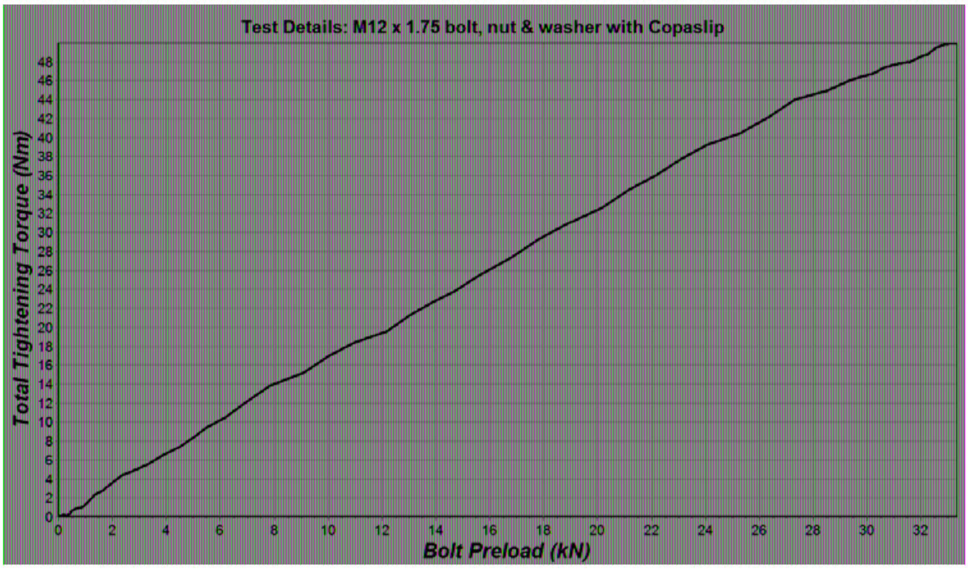

Figure 5 - Torque-Force Graph

Based upon this experimental confirmation, it is safe to determine the coefficient of total friction μ_{tot} by the formula (from [10])

$$\mu_{tot} = \frac{\frac{T}{F} - 0.15915p}{0.57735d_2 + \frac{D_e}{2}} \qquad \text{Equation 3}$$

6. DISCUSSION

From inspection of figure 3 it is evident:

- A significant reduction (54%) in the bolt preload occurs between the first and tenth tightening of an EZP bolt, a 51% reduction occurs between the first and fourth tightenings.
- Use of a specialist high pressure lubricant (molybdenum disulfide or Copaslip) applied under the nut face and in the threads results in no significant change in the bolt preload between the first and tenth tightenings. Use of a lithium grease also provided a consistent bolt preload between the first and tenth tightenings.

- The use of a light machine oil resulted in a significant decrease (47%) in the bolt preload value. This was probably a result of the oil being displaced from the moving contact surfaces as a result of the high contact pressures involved.

From inspection of figure 4 it can be seen that:

- The preload change on the tenth tightening on unlubricated stainless steel was 33% of that achieved on the first tightening (from 22.5 kN to 7.5 kN). The use of a dry coating of molybdenum disulfide on the stainless steel resulted in the preload on the tenth tightening being 68% of the first tightening (24.4 kN to 16.5 kN).

Table 1 - Friction Values

	μ_{tot} First Tightening	μ_{tot} Fifth Tightening	μ_{tot} Tenth Tightening
EZP bolt as received	0.276	0.566	0.620
EZP bolt with molybdenum disulfide paste	0.086	0.075	0.070
EZP bolt with Copaslip	0.086	0.087	0.083
EZP bolt with lithium grease	0.133	0.118	0.137
EZP bolt with machine oil	0.165	0.286	0.329
Stainless steel bolt – as received	0.137	0.332	0.449
Stainless steel bolt with a dry molybdenum disulfide coating	0.125	0.145	0.193

Table 1 presents coefficient of friction values calculated using equation 3. It can be seen that a lubricant can play a vital role in ensuring a relatively constant friction value between tightenings. VDI 2230 [9] indicates that a reliable design must be based upon the minimum bolt preload value. Having a large scatter in the preload results in bolts larger than would otherwise be needed having to be specified. The bolt diameter has to be sized based upon the maximum preload in order that the minimum preload still meets the functional requirement.

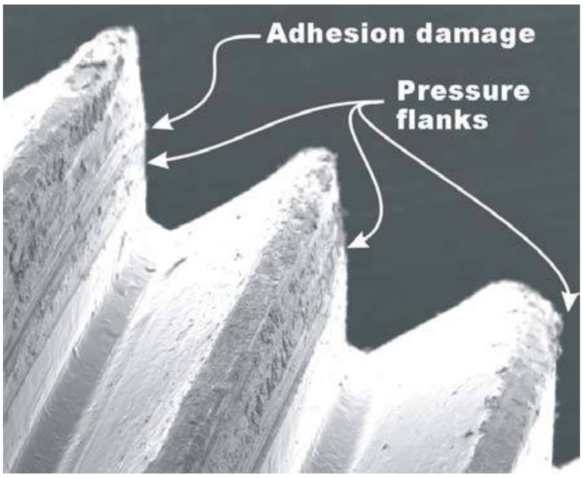

Figure 6 - The pressure flank of an unlubricated EZP bolt thread after five tightenings

As can be seen from the results, there is a significant increase in friction occurs with unlubricated EZP bolts as the number of tightenings is increased. Previous work [11] using a scanning electron microscope indicated that, in an unlubricated condition, a significant amount of surface damage can be observed on the pressure flanks of the threads. This is shown in figure 6 which illustrates damage on a thread pressure flank i.e. the side of the thread that sustains the loading, after five tightenings.

7. CONCLUSIONS

This study has shown that the use of high pressure lubricants, such as molybdenum disulfide, on threaded fasteners offers considerable potential in ensuring that a high and repeatable preload is achieved when the fasteners are re-tightened after parts are removed from an assembly. By reducing the thread friction and hence the torsional stress incurred during tightening, lubricants also provide the opportunity for a greater utilization of the bolt strength. That is, a greater preload is achievable using a lubricated fastener since the yield point is reached as a combination of tensile and torsional stresses. Decreasing the torsional stress due to lower friction allows a higher tensile stress to be present when the yield point is reached and hence a higher preload.

Considering that many companies specify that the fasteners should be replaced if parts are removed, this study shows that with the use of lubricants, fasteners tightened using the torque control method can be re-used so that an repeatable preload is achieved. Detailed advice on how the fasteners could be reused, by using a lubricant, could be provided in maintenance handbooks and similar publications. The potential economic savings that could be made if this was adopted would be considerable. This approach is also in line with the increasing importance of re-cycling and sustainability.

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Frictional changes during repeated tightening of zinc plated threaded fasteners

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ABSTRACT

Tightening a bolt to a torque value is the most popular means of controlling the preload provided by a threaded fastener. However, repeated tightening and loosening of a threaded fastener, can have a dramatic effect on the friction coefficient. These changes will, in turn, will affect the preload and can, therefore, have an adverse effect on the structural integrity of an assembly. This study investigates the effect of repeated tightening on electro-zinc plated (EZP) nuts, bolts and washers. It is found that significant wear of the contact surfaces of the bolt/nut thread and nut face occurs upon repeated tightening.

1. Introduction

Without friction, the use of threaded fasteners would not be possible without a locking mechanism and would be highly restricted. When a nut is tightened onto a bolt, it is friction that holds it in place preventing it from spinning off. Friction is essential, but too high a friction value can also be a problem to the functioning of a threaded fastener. In the majority of applications, it is the preload that is the primary design requirement. The preload results from the axial stress generated when the bolt is tightened. High thread friction results in a high torsional stress in the threaded section that limits the axial stress available and hence the preload achievable from the tightening process. Preload is difficult and expensive to measure and so usually its control is indirectly achieved by specifying a tightening torque. Friction plays a large part in determining the effectiveness of the conversion of applied torque to fastener preload. To control preload by torque control it is essential to have some limits placed on the range of the coefficient of friction.

When the application sustains purely static loading, it is desirable to keep the friction as low as possible so that the torsional stress in the bolt is minimised. In dynamic applications, a concern is that low friction will aid self-loosening. Most of the major automotive companies specify the acceptable friction range on fasteners they use so as to provide some control over the torque–tension relationship. For example, Ford, Renault and Peugeot-Citroen specify an acceptable friction range of 0.12–0.18. German manufacturers typically specify a lower minimum value, for example Porsche and Volkswagen specify 0.09–0.14. Company standards are used (such as the Ford standard WZ 101 [1]) for checking the compliance of the fasteners, such standards being usually based upon relevant national standards (such as DIN 946 [2]) adjusted to suit the particular requirements of the company.

In the automotive industry, zinc flake and other coatings that incorporate PTFE or other dry lubricants are typically used to control the friction characteristics. In other industries, electro zinc plated (EZP) coatings are used which typically do not have a coating to modify the friction characteristics.

There is a relationship between the torque applied and the preload generated [2]. This torque–tension relationship involves the thread and nut dimensions and the coefficient of friction in the threads and under the nut face. This relationship is presented in Eq. (1).

The tightening torque is usually determined by using an expression that assumes a fixed value for the coefficient of friction, so that a particular pre-stress is applied to the bolt when it is tightened. The target pre-stress applied to the bolt varies between industries. In the petro-chemical industry, typically a target pre-stress of 50% of the bolt material's minimum specified yield strength is used [3]. In automotive engineering applications, this pre-stress is typically around 75% of the minimum yield strength value of the bolt material [4]. If the coefficient of friction increases, then the pre-stress and subsequently the clamp force, will decrease for a given torque value.

Many equipment manufacturers specify that new fasteners should be used if the assembly is dismantled. However, because of a lack of availability of fasteners, or for economic reasons, fasteners are often re-used following the disassembly of a joint. A number of researchers have reported a change in the friction

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characteristics of fasteners on repeated tightening. Morgan and Henshall [5] report that wheel studs can experience a significant (50%) reduction in their axial tension after several re-uses. However, they found that recovery to the original condition could be achieved by the use of oil as a lubricant. Jiang, Zhang and Park [6] report that a doubling in thread friction can occur on distorted head type prevailing torque nuts.

Previous studies [6,7] have established that the friction coefficients are largely independent of the speed of tightening of the fastener and substantially independent of the preload. Jiang, Zhang and Park [6] also concluded that the thread friction coefficient is substantially independent of the bolt's preload and that the bearing head friction coefficient decreased with increasing preload. They also noted that repeated tightening and loosening generally increases the friction present in bolted joints especially when the contact surfaces are coated. Their tests were conducted on flanged nuts and they speculated that this was attributable to a change in the pressure distribution over the contact area with increasing clamp force.

Despite these studies, there appears to have been little work conducted on the re-use of the commonly used electro-zinc plated nuts, bolts and washers. Jiang, Zhang and Park [6] completed some work on some combinations of zinc plated washers, nuts and bolts, but their focus was on additional finish conditions typically used in the automotive industry. Electro-zinc plating (EZP) without additional surface treatments is so widely used on fasteners because it offers good corrosion protection at a low cost.

2. Test methods

In this study, two methods were used to investigate the friction coefficient changes in the contact interfaces of bolts, nuts and the clamped surface material, specifically:

1. tests on fasteners during repeated tightening, to establish how the friction coefficient, and the clamp force generated by a constant tightening torque changed as a result of the repeated tightening;
2. measurements of the nut face friction under conditions of constant pressure, to investigate how the friction coefficient changes with the angular rotation.

The dimensions of the fasteners used in this series of tests are given in Table 1.

3. Details of the test apparatus

3.1. Measurement of the nut face and thread friction coefficients

To determine the friction coefficient between the materials at the nut face and those in the threads, a test apparatus was designed and built. The test apparatus included a specially designed test frame with torque, angle and clamp force measuring instruments connected to a computer via an analogue to digital converter to allow data to be collected during the experiments. The tightening torque was applied by an electronic torque wrench (SPS Sensor 1 wrench) that could simultaneously measure the angle of rotation applied to the nut or bolt head as well as the applied torque. The test apparatus comprised a load cell to measure the bolt's clamp force and a torque transducer to measure the thread extension and friction torque. The thread friction and extension torque are isolated by the use of a thrust bearing that allows the reaction to be made by a torque transducer as shown in Fig. 1.

Cabling connects the electronic hand wrench to the A/D converter allowing sampling of the torque–angle measurements. The sampling rate is variable, but an acquisition speed of 10 samples per second was found to produce consistent results. Software was written to read, process and draw graphs of the results.

Upon completion of the test, the test data consisting of applied torque, thread torque and bolt preload was processed allowing the head and thread friction coefficients to be computed for each measured sample interval.

A constant tightening torque of 80 Nm was used in the tests described here. The 80 Nm torque was selected so that the induced stress would not exceed the bolt's yield strength. If the friction coefficient increases on re-tightening, an increase in the tightening torque is necessary in order that a similar clamp force is maintained. This was not done in these tests since the majority of engineering specifications detail a constant torque value. The purpose of this study was to investigate the consequences of re-tightening; any re-tightening would usually be completed by tightening up to the original first tightening torque value.

3.2. Measurement of the nut face friction coefficients under a constant load

When a nut was tightened in the experiment the pressure on the nut face changed gradually as the tightening progressed. The tightening was normally stopped when a specified torque was reached making it difficult to establish the effect that the sliding distance of the two surfaces had on the friction coefficient. To investigate the effect that sliding distance had on the nut face/washer interface, a test apparatus was used that allowed a nut to be rotated on a test surface indefinitely at a pre-determined load. The test arrangement is shown in Fig. 2.

The test apparatus consisted of a load cell and thrust bearing mounted in series. A bolt passed through the middle of the assembly. A series of disc springs was used in order to increase the resilience of the bolted assembly so that once the bolt had been tightened it would be insensitive to clamp force variation due to wear on the nut face. An EZP coated tab washer was used to ensure that the rotating surface was the nut onto the washer rather than the washer onto the support. The bolt head was tightened so that no relative rotation occurred between the nut and the washer. Tests were completed at three values of bolt load, 10, 20 and 30 kN.

The test consisted of applying torque to the nut and letting the bolt rotate. Measuring the torque required to rotate the nut against the tab washer allowed the nut face friction coefficient to be determined.

The nut rotation angle was measured simultaneously with the clamp force. It was found that there was a limit to the number of revolutions the nut could be rotated. The applied torque gradually increased resulting in a point being reached when the tab on the

Table 1
Fastener dimensional details.

Bolt property class (ISO 898)	8.8
Thread diameter D	12 mm
Thread pitch P	1.75 mm
Basic pitch diameter of thread d_2	10.863 mm
Included thread flank angle	60°
Bearing diameter of nut d_o	18.5 mm
Hole diameter of washer d_i	13.0 mm
Effective bearing diameter of nut D_e	15.75 mm
Thickness of electro-zinc plating	0.005–0.009 mm

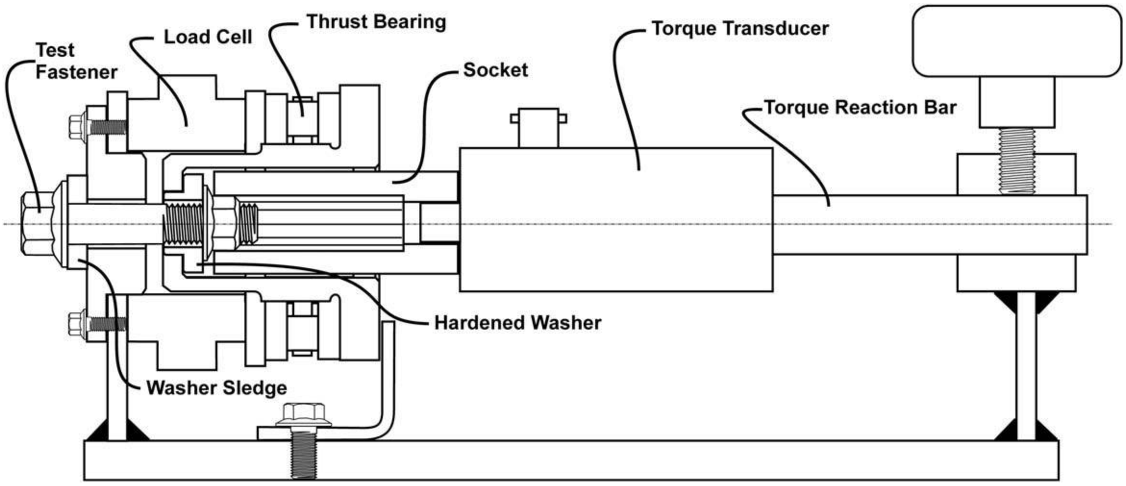

Fig. 1. Section through the Test Rig.

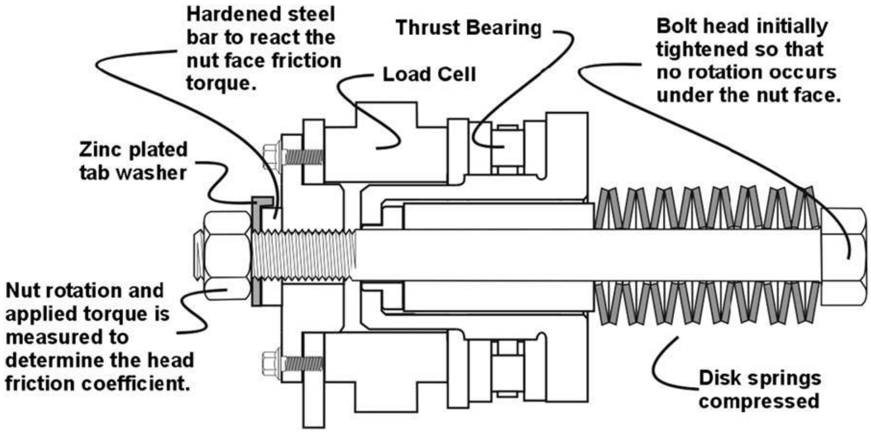

Fig. 2. Test apparatus for determining the nut face friction coefficient under a constant load.

washer failed, resulting in the rotating interface being the washer against the hardened steel bar. The test was terminated when this occurred.

4. Computation of the friction coefficients

The tests conducted consisted of tightening the nuts in the test apparatus whilst measuring the thread reaction torque and the clamp force generated by the tightening process. Thread and nut face friction coefficients were evaluated using DIN 946 [2] which gives:

$$T = F\left[0.159p + 0.578\mu_t d_2 + \frac{D_e}{2}\mu_n\right] \quad (1)$$

The value of D_e is taken to be:

$$D_e = \frac{d_o + d_i}{2} \quad (2)$$

where T is the total tightening torque, F the bolt preload, d_2 the basic pitch diameter of the thread, p the pitch of the thread, D_e the effective bearing diameter of the nut, d_o the outer bearing diameter of the nut, d_i the inner bearing diameter of the nut face.

(Note: During the course of this investigation the ISO 16047 [8] standard was introduced that is substantially the same as the DIN 946 [2] standard. This had no implications on the work described here.) Consequently if the applied torque T and the thread torque T_{tm} are measured together with the bolt preload F, the torque tension equation can be used to determine the instantaneous thread friction μ_t and nut face friction μ_n coefficients. The equations used in the analysis of the test results were thus:

$$\mu_t = \frac{\frac{T_{tm}}{F} - 0.159p}{0.577d_2} \quad (3)$$

$$\mu_n = \frac{2(T - T_{tm})}{D_e F} \quad (4)$$

where T_{tm} is the thread torque measured in the tests.

To ensure that the nut surface rotated against the washer surface rather than the washer rotating against the support, a “tabbed washer” was used to prevent it rotating. It is worth noting that because of the very high values of friction after several tightenings, the tab on the washer failed (sheared) in two instances resulting in rotation occurring on the washer to support face rather than the nut to washer face. Data from the tests when such a failure occurred were ignored.

5. Results of tests on electro-zinc plated fasteners

5.1. Nut face and thread friction coefficients

A summary of the results is shown as a boxplot in Fig. 3. The centre lines in the boxes represent the sample medians. The ends

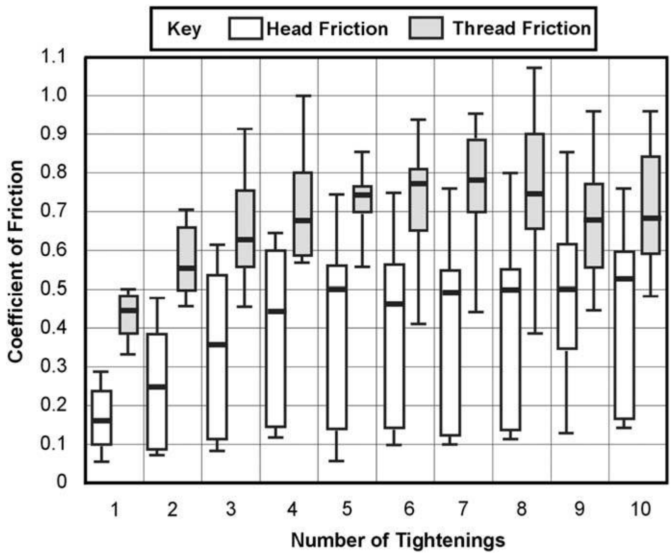

Fig. 3. Effect of re-tightening the nut on the friction coefficient.

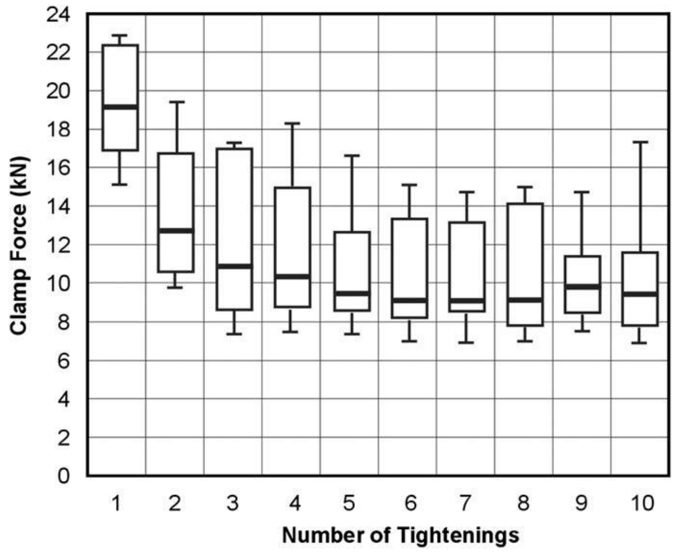

Fig. 4. Effect of re-tightening the nut on the clamp force generated.

of the boxes represent the upper and lower quartile of each sample and the whiskers indicate the extents.

As can be seen from these results:

- The median friction coefficient increases steadily during the first four tightenings and then stabilises at a value approximately twice that observed during the first tightening.
- The trend was for the clamp force produced by the bolt to decrease as the number of tightenings increased and the friction increased. Again, the clamp force stabilised after four tightenings. This is illustrated in Fig. 4. There was also a trend for the friction coefficient to decrease as the clamp force increased; this effect is illustrated in Fig. 5. A constant tightening torque of 80 Nm was used for all the tightenings.
- Significant scatter was observed in the results, the scatter in the head friction being particularly pronounced. The scatter observed in the friction from EZP fasteners is one reason why high volume car manufacturers have tended to move away from this type of finish to a zinc flake finish. Such a finish usually incorporates a top coat of PTFE or similar solid lubricant to minimise frictional scatter. In practice, a large scatter in friction can result in an inefficient joint design. The range in the friction coefficient has a direct influence on the fastener strength/size that is determined for a loaded joint. For a given size/strength of fastener, the upper limit of the friction determines the minimum clamp force that will be achieved for a given tightening torque value. Such a clamp force has to be sufficient to resist joint separation from direct forces and transverse movement from shear forces. The lower limit of friction will result in the maximum clamp force, for the specified tightening torque, that the fastener must sustain without failing. If the friction scatter is increased the net effect is that the optimum bolt size is also increased. If close control of the clamp force is required, control of the scatter in the friction is also needed.

The distribution of load in screw threads has been studied by a number of researchers [9]. An authoritative paper by Sopwith [10], indicated that the nearer the thread surface was to the joint face, the higher was the proportion of the overall load it sustained. The load distribution in the threads is dependent upon several variables, one of which is friction present in the threads. The higher the friction, the greater is the proportion of the loading that acts on the threads nearest the joint face.

As the friction increases in the thread, the proportion of the loading sustained by the first thread increases. The maximum pressure on the threads for the first tightening is typically 500 MPa (bolt loading 24 kN) compared to 175 MPa for the nut face pressure. This higher surface pressure explains, to some extent, the reason why the friction coefficient is typically higher in the threads than it is on the nut face for EZP coated fasteners.

Fig. 6 illustrates an empirical fit was made to establish the relationship between the friction coefficient and the number of tightenings. It was found that an asymptotic relationship fitted, with reasonable accuracy to the variation in nut face and thread friction coefficients as the number of tightenings progressed. For the nut face friction coefficient, the following equation produced a good fit:

$$\mu_n = 0.182 + 0.125\ln(n) \tag{5}$$

where n is the number of tightenings sustained by the fastener.

For the thread friction coefficient, the following equation produced a good fit with the test results:

$$\mu_t = 0.497 + 0.118\ln(n) \tag{6}$$

The predicted values using these equations are compared with the experimental results in Fig. 6. These equations can be used to develop the torque–tension relationship to allow the number of tightenings sustained by the fastener to be considered. Based upon the work completed, the torque–tension relationship (Eq. (1)) can be adjusted so that the bolt preload is defined in terms of the applied torque and the friction characteristics. For EZP fasteners, the relationship becomes:

$$F = \frac{T}{\left[0.578\mu_t d_2 + 0.159p + \dfrac{D_e\mu_n}{2}\right]} \tag{7}$$

In this case: μ_t=0.497+0.118 ln(n) and μ_n=0.182+0.125 ln(n)

A comparison between the experimental results and the results is shown in Table 3.

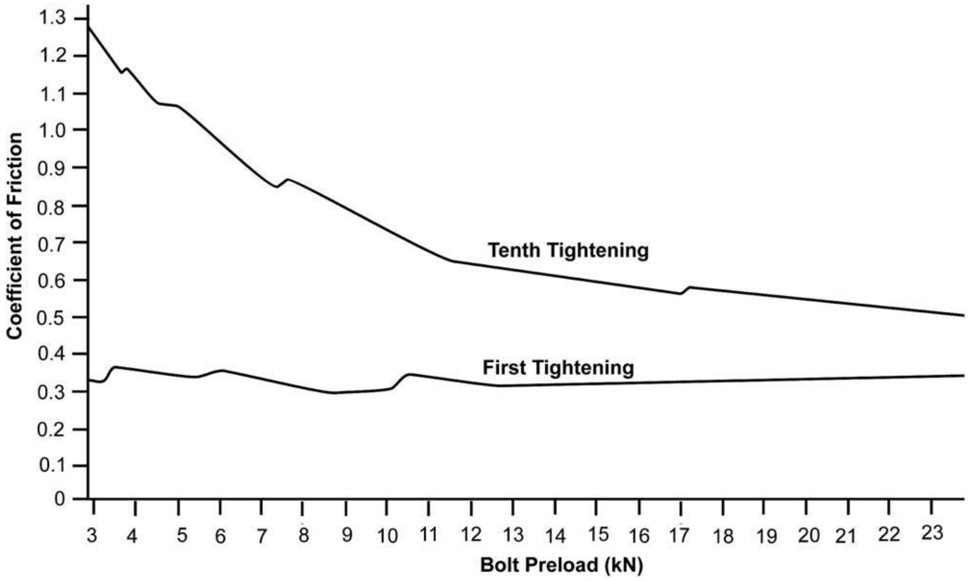

Fig. 5. Graph showing the effect of re-tightening on the variation of the thread friction coefficient.

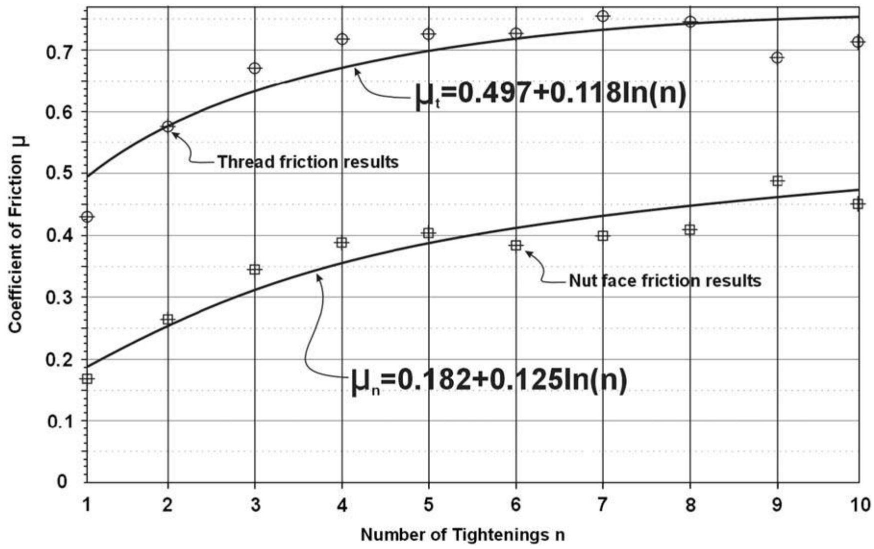

Fig. 6. The variation in the friction coefficient with the number of tightenings.

5.2. SEM Investigation

A series of inspections was conducted using a scanning electron microscope[1] to observe the surfaces in the contact between re-tightening operations. In this series of tests the nut was tightened against a hardened steel washer without EZP. Scanning electron microscopy of bolts was conducted to develop some understanding of the effects leading to changes in the friction coefficients. Interest was centred on the threaded region of the bolt since previous tests had indicated that the largest changes in the friction coefficient were occurring in the threads. After the bolt had been tightened once and then released, surface damage could be observed on the pressure flanks of the thread. The two photographs shown in Figs. 7 and 8 show the condition of the thread flanks after one tightening. As can be seen in the higher magnification photo, region (a) is where zinc has been transferred from the nut thread onto the bolt thread. Region (b) shows the coating fractured with partial removal of some of the zinc. Region (c) shows evidence of abrasive wear, possibly due to a wear debris particle and region (d) is a wear particle left on the surface of the thread.

The thread coefficient of friction was considerably higher than the nut face friction and showed a decreasing trend as the preload increased.

After 5 tightening and untightenings, significant wear and surface damage had occurred on the pressure flanks of the threads. Fig. 9 shows the roughness of the pressure flank surface of the thread after 5 tightenings. Fig. 10 shows a magnified view of the surface exhibiting the zinc becoming detached from the steel substrate that may have occurred as a result of adhesion effects. The nut face surface after the first tightening was relatively smooth, but by the fifth tightening, debris and wear particles can be observed on the surface. However, the surface was in a better condition than that of the thread flanks.

After 10 tightenings and untightenings, severe wear had occurred on the threads; this is shown in Fig. 11. The flank of

[1] A FEI Quanta 200 Scanning Electron Microscope was used in these tests.

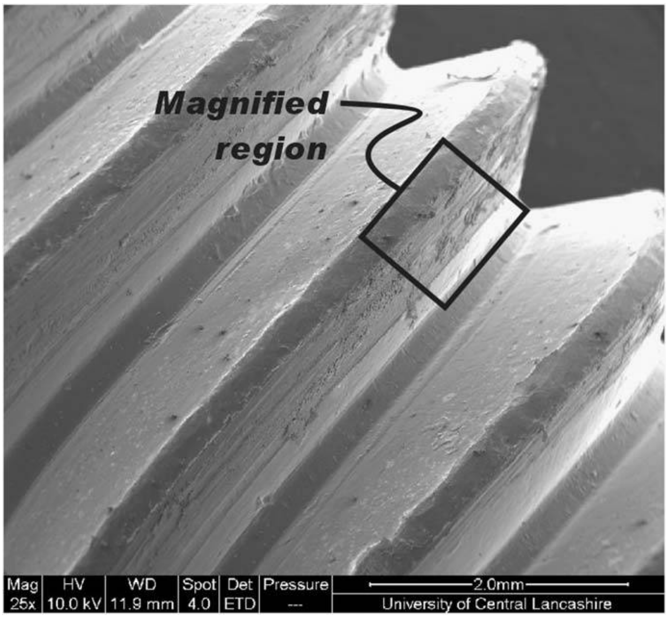

Fig. 7. Thread tightened once—25 × magnification—rectangular region magnified in figure 4.8.

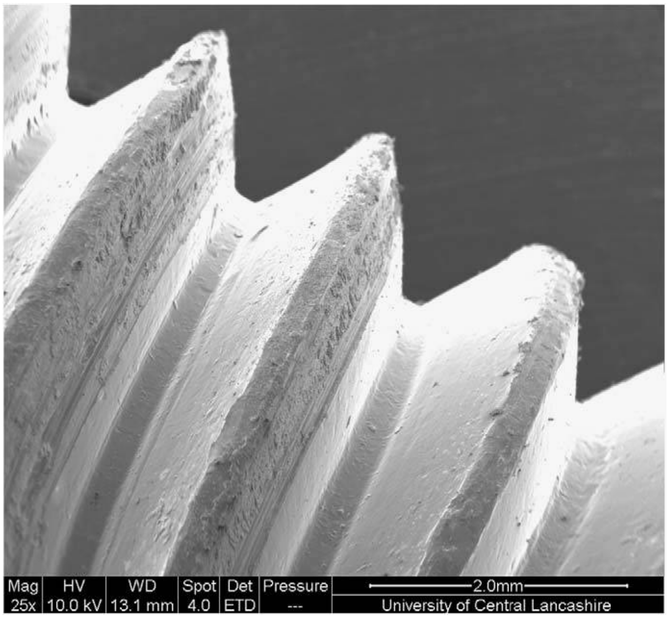

Fig. 9. Thread tightened five times—25 × magnification.

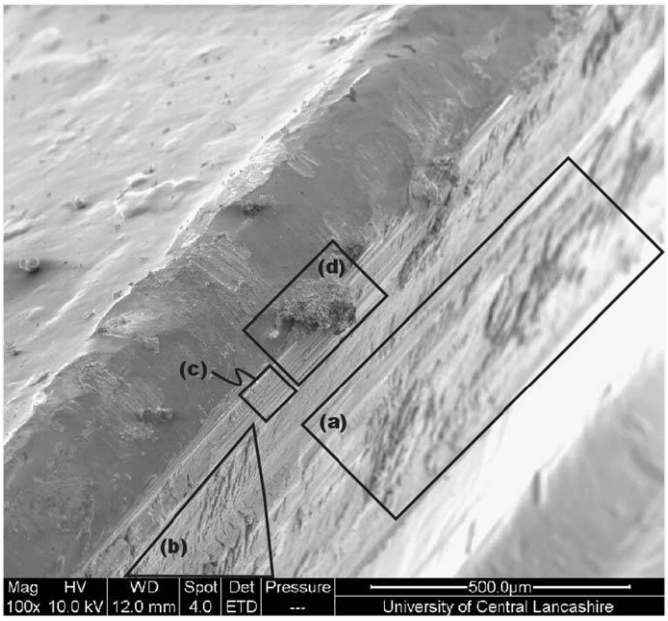

Fig. 8. Thread tightened once—100 × magnification.

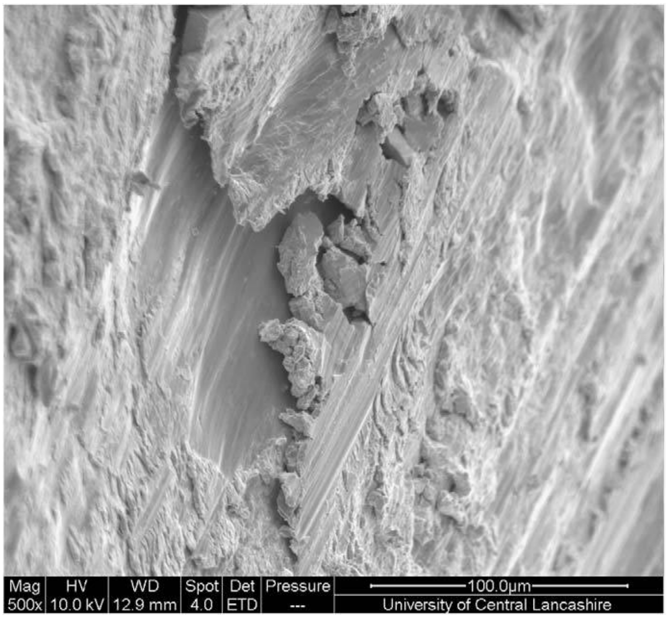

Fig. 10. Thread tightened five times—500 × magnification.

the thread closest to the joint (region A of the thread in Fig. 11) is the highest loaded in the assembled joint and shows severe scoring, threads further into the nut show less severe wear patterns (B), but evidence can be seen of wear particles being broken away from the surface (C). The thread closest to the joint surface is also distorted.

6. Nut face friction coefficient tests under constant load

Referring to Table 2, under this test it was found that initial friction coefficients were reasonably stable i.e. the friction did not change significantly as the nut rotated. However after a critical angle of rotation, a transition point was reached when the friction coefficient started to increase, leading typically to a tripling in its value. A graph illustrating how the friction varied with the angle of nut rotation is shown in Fig. 12. The mean initial value of the friction coefficient was 0.13 with a standard deviation of 0.02.

Table 2 shows the results of the nut face friction coefficient tests. In order to quantify the sliding distance before a significant change in the friction coefficient occurred, the angle at which the friction coefficient became greater than 1.5 times its initial value was measured. The 1.5 factor was chosen based upon study of the angle–friction coefficient graphs so that breakdown in the condition of the friction surfaces could be distinguished from the normal variability of the friction coefficient. For fasteners, the maximum friction value divided

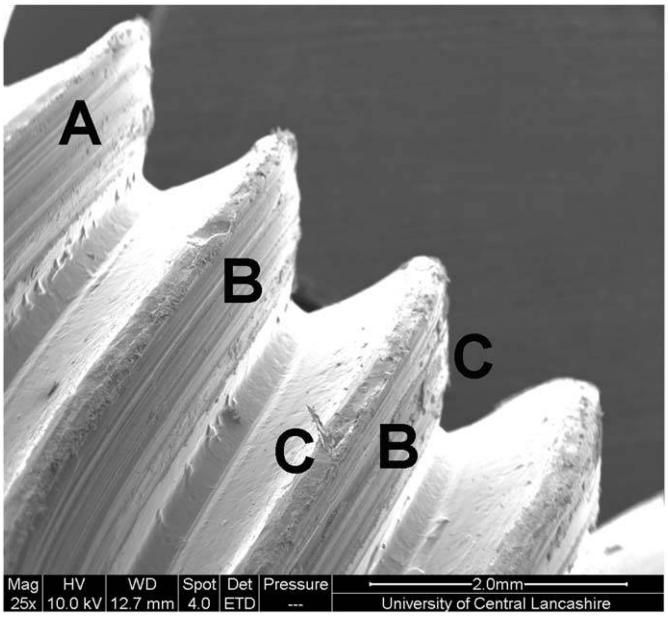

Fig. 11. Thread tightened 10 times—25 × magnification.

Table 2
Nut face friction coefficient results.

Applied load (kN)	Initial friction coefficient	Angle at which friction = 1.5 × initial value (deg)	Sliding distance based upon mean friction radius (mm)
10	0.13	2655	365
10	0.11	3005	413
10	0.16	595	82
10	0.16	3013	414
10	0.13	1010	139
20	0.105	732	100
20	0.12	990	136
20	0.14	879	121
20	0.11	827	114
20	0.12	1064	146
20	0.18	2250	309
30	0.12	1355	186
30	0.13	1216	167
30	0.12	838	115
30	0.11	1070	147
30	0.13	1450	199
30	0.10	1244	171

by the minimum value is typically of the order of 1.5. This angle indicating surface breakdown is documented in the table together with the sliding distance based upon the mean radius of the bearing surface of the nut. The nominal bearing pressures under the face of the nut were 73.5, 147 and 220.5 MPa for the 10, 20 and 30 kN loads, respectively. The average sliding distance prior to a significant change occurring in the friction characteristics was 195 mm with a standard deviation of 105 mm.

Table 3 presents a summary of the results and a comparison between measured and predicted bolt preloads.

7. Discussion and conclusion

Based upon the experiments completed and the measurements made, a change in the friction between the contact surfaces of the bolt/nut thread and nut face occurs upon repeated tightening. This breakdown of these surfaces results in an increase in the friction coefficient present which causes a reduction in the clamp force provided for an assembly when tightened to a specified torque value. This reduction in the clamp force is significant, the clamp force on the sixth re-tightening typically being half that obtained on the first tightening which could have an adverse effect on the structural integrity of the assembly. The use of appropriate lubricants can reduce the frictional scatter and the increase in friction as a result of fastener reuse [11].

The process of the friction increasing to a peak value and then decreasing before falling to a stable value is described by Suh and Sin [12]. The process is dependent upon the distance slid by the two surfaces and is equated, in the re-use of fasteners, with the number of times the fastener has been tightened. This change in the friction conditions is considered to be due to damage to the contact surfaces resulting from sliding and the high pressures involved. The primary processes for causing the friction variation are the ploughing effect of trapped wear particles and adhesion. Wear particles trapped between the surfaces results in ploughing by the entrapped particles further increasing friction. Levelling off of the frictional resistance occurs as the number of particles entrapped between the surfaces becomes constant.

Higher levels of friction were noted for the threads than were present under the nut face. This is thought to be due to a number of factors. Thread friction is somewhat complicated by the non-linear pressure distribution on the thread surfaces and that this non-linearity is itself partially a function of the friction value [10]. The higher pressures being present in the thread results in increased wear particle generation and ploughing. The enclosed nature of the mating threads also results in wear particles created at the surface closest to the joint face, which is under the highest pressure, being moved further up the thread as the nut rotates. Additionally, work hardening is probably occurring along with to thread deformation (as can be seen in Fig. 11). Under such circumstances, besides surface friction there will also be a component of the overall friction force due to the work required to deform the thread [13]. The thread contact geometry and the elastic–plastic properties of the steel will influence the magnitude of this effect. A third factor that will influence the friction force measured is due to distortion of the thread flanks as the tightening process progresses. An implicit assumption in the determination of the friction coefficient is that the flank angle is a constant 60°. The distorted pressure flank that can be seen in the SEM photographs will increase the frictional force even if the friction coefficient remained constant.

Although the magnitude of the thread friction coefficient is higher than that present under the nut face, the friction becoming approximately constant after five or six tightenings is similar to the effect observed for the nut face friction.

It was also noted that the magnitude of the thread friction coefficient became dependent upon the pressure (clamp force) as the number of tightenings increased. The mechanism for this effect is not completely understood, but is thought to be due to the increase in pressure causing a change from wear particle ploughing to adhesive friction.

Tests completed to determine the nut face friction coefficient under a constant loading clearly indicated that a change occurs in the friction characteristics after a certain amount of angular movement. By using the mean nut radius, this angular movement was converted to a linear movement. The average distance prior to a significant change in friction occurring was 195 mm. The initial average friction coefficient of 0.13 is less than the mean value noted in the tightening tests (mean value on first tightening

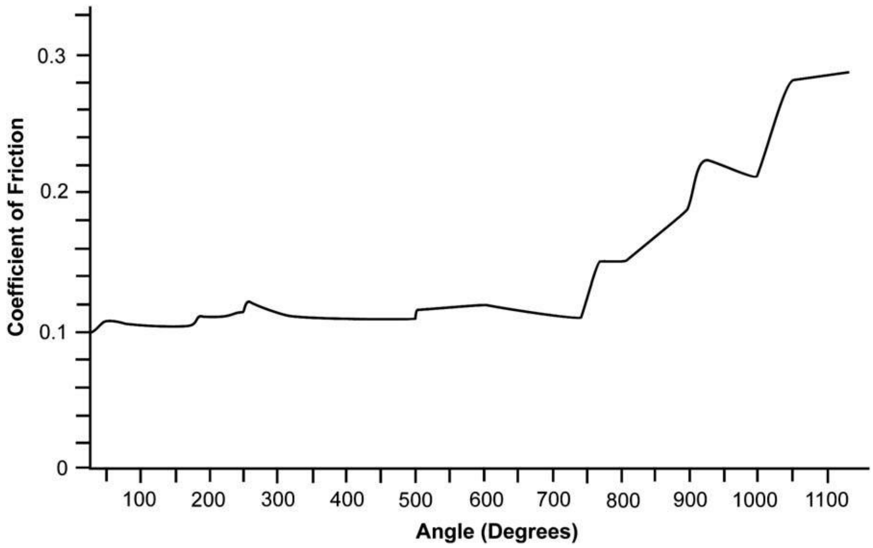

Fig. 12. Graph showing the onset of surface failure—20 kN load.

Table 3
Comparison of actual versus predicated friction coefficient and bolt preload values.

Tightening number	Mean experimental results		Predicted value based upon mathematical model		Actual mean bolt preload	Predicted bolt preload	Percentage error of predicted versus actual preload (%)
	Nut face friction (μ_n)	Thread friction (μ_t)	Nut face friction (μ_n)	Thread friction (μ_t)	Newtons	Newtons	
1	0.164	0.430	0.182	0.497	18722	16556	11.5
2	0.263	0.579	0.269	0.579	13363	13262	0.8
3	0.341	0.678	0.319	0.627	11080	11892	7.3
4	0.383	0.717	0.355	0.661	10261	11074	7.9
5	0.402	0.729	0.383	0.687	9972	10515	5.4
6	0.388	0.729	0.406	0.708	10116	10100	0.2
7	0.400	0.752	0.425	0.727	9814	9768	0.5
8	0.418	0.742	0.441	0.742	9722	9512	2.2
9	0.489	0.685	0.456	0.756	9492	9285	2.2
10	0.449	0.713	0.470	0.769	9655	9083	5.9

of 0.19). However the scatter in the results on the tightening tests observed on the nut face tests under constant loading was less than that observed in the full tightening tests.

Based upon these results, the re-use of unlubricated EZP threaded fasteners cannot be recommended. The increase in friction resulting from re-use could have an adverse effect on a joint's structural integrity due to the corresponding decrease in the generated clamp force that occurs.

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Towards an understanding of the loosening characteristics of prevailing torque nuts

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Abstract: Prevailing torque nuts are an extremely popular method of providing resistance to vibration-induced self-loosening of fasteners. Such nuts have a self-contained prevailing torque feature that provides a degree of resistance to rotation. Although such nuts are frequently used, it is not widely realized that they can occasionally come completely detached from bolts. The mechanism by which this can occur has hitherto been unidentified since it has not been possible to replicate detachment under laboratory testing. This article identifies a general condition that can result in the complete loosening and detachment of prevailing torque type nuts. This mechanism involves the application of an axial load when transverse joint slip is occurring. This article describes a modified Junker test machine that allows the application of axial loading to a joint while experiencing transverse displacement. Tests have been completed using an intermittent as well as a constant axial load. Loading in both modes has been demonstrated to result in the complete detachment of this nut type. Based on this investigation, if the magnitude of the axial loading exceeds the residual preload in the bolt retained from sustaining transverse movement alone, the all-metal type of prevailing torque nut can completely detach. Applications that involve shear and axial loading being simultaneously applied to a joint are numerous in engineering.

Keywords: prevailing torque type nut, self-loosening, Junker test, vibration, fastener, nut detachment

1 INTRODUCTION

Prevailing torque type nuts are commonly used to try to prevent self-loosening and/or detachment of threaded fasteners. Prevailing torque nuts were developed over 100 years ago [**1**] and have the essential characteristic that a torque is needed to rotate the nut down the thread of an untightened bolt. One advantage of this type of nut is that the locking feature can be verified at the time of assembly by measuring the prevailing torque. A problem with re-use of nylon insert nuts is that the prevailing torque can decrease as a result of wear [**2**]. Modern standards for all-metal prevailing torque nuts, however, stipulate that a minimum level of prevailing torque, for a given thread size, must be maintained for up to five re-uses [**3**]. This type of nut is in extensive use across most industries and is probably the most common type of locking device for threaded fasteners.

There are many varieties of such nuts, but in general they can all be classified into one of two categories, those with a non-metallic insert or those which are all-metal. Non-metallic insert nuts typically generate a prevailing torque by incorporating a polymer insert in the top of the nut that is deformed by the bolt thread. (A nut of this type is illustrated in Fig. 1(a)). The all-metal variety achieves the prevailing torque by either distorting the top nut threads by introducing slots (Fig. 1(b)), making the top threads elliptical shaped (Fig. 1(c)), or introducing spring steel inserts (Fig. 1(d)).

There have been instances when new prevailing torque nuts that still retain a prevailing torque have become detached from assemblies. The causes of such failures have not been understood since the standard method of test, the Junker (Deutsches Institut für Normung (DIN) 65151) test, has been unable to reproduce such loosening. One such failure is shown in Fig. 2;

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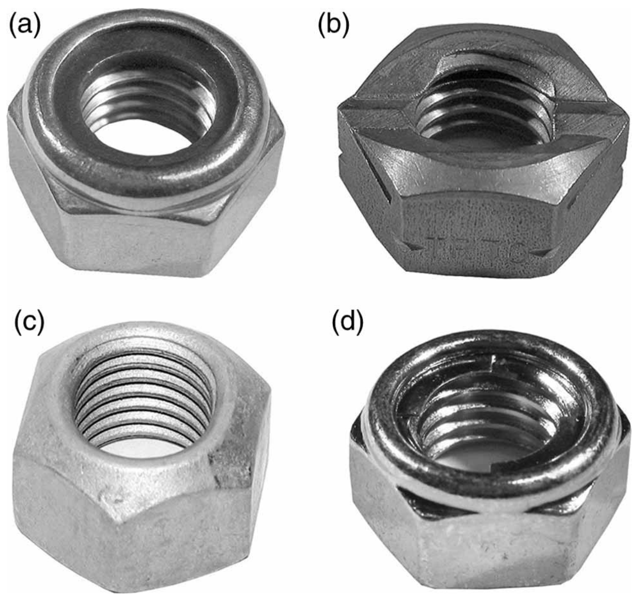

Fig. 1 The principal types of prevailing torque nuts: (a) non-metallic insert prevailing torque nut, (b) all-metal prevailing torque nut with slots used to distort the thread, (c) all-metal prevailing torque nut with the top of the thread elliptical shaped, and (d) all-metal prevailing torque nut with spring steel inserts

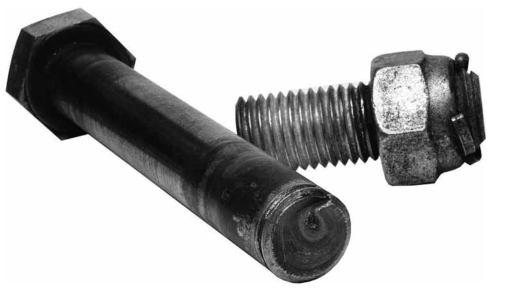

Fig. 2 Failed bolt illustrating the loosening of the non-metallic insert nut

it is an M24 bolt from a bus engine mounting. A non-metallic insert nut was used to resist any self-loosening tendency. However, it was found that this came loose and in at least one instance completely detached from the bolt. A split pin was introduced in an attempt to prevent this, but this was unsuccessful. Figure 2 shows the pin being sheared, the bolt failing by fatigue before this was achieved. The reason why the non-metallic insert nut was ineffective in preventing complete loosening and detachment was never established at the time of investigation into this failure.

Goodier and Sweeney [**4**] in 1945 tested axially loaded bolted joints dynamically and failed to completely loosen them. They also offered details of a possible mechanism for partial self-loosening. Sauer *et al.* [**5**] built upon their work by applying the axial loading rapidly by using a fatigue testing machine, but still failed to achieve complete loosening. Junker [**6**] reported on work he completed in the 1960s in which he found that transverse vibration could completely loosen fasteners. The test machine originally developed by Junker had the capability to allow a combination of cyclic axial and transverse vibration to be applied to a joint. He reported only on the effect of transverse vibration since he found that this mode of loading had a greater effect on self-loosening than other loading directions. Koga [**7**] reported on the effects of axial impacts on the loosening of fasteners. Subsequent papers in the literature largely focused on the effect of pure transverse vibration on the locking characteristics of fasteners.

Riches [**8**] was the first to investigate the loosening characteristics of commercially available prevailing torque nuts under transverse vibration. His work was part of a government contract and was not in the public domain at the time. His work showed that prevailing torque nuts do not fully loosen under transverse vibration. He also found that there was some correlation between the magnitude of the prevailing torque and the locking efficiency. Essentially, the higher the prevailing torque, the higher was the retained preload.

Finkelston [**9**] was the first to publish, in the public domain, the role that a prevailing torque plays in maintaining preload when a joint is subjected to transverse vibration. He found that a prevailing torque reduces the rate of preload loss and will also stop the loosening process when the prevailing torque counteracts the loosening torque. For the prevailing torque nuts, he tested (3/8″ – 16 tpi having a prevailing torque of 60 in-lb, which is 6.8 Nm), the loosening stopped with approximately 10 per cent of the initial preload remaining. In the 1970s and early 1980s large scale studies (reported in references [**8**], [**10**], and [**11**]) were completed by commercial organizations to assess the loosening resistance of a large number of proprietary locking devices. These tests included a large range of prevailing torque nuts. All these tests were completed under purely transverse vibration and, in general, it was found that prevailing torque fasteners retained a steady-state preload, but at levels well below the initial preload.

In 1994 the German standards authority DIN published the DIN 65151 standard [**12**] that assessed the loosening characteristics of fasteners based purely on transverse vibration. This standard became widely accepted by industry as the basis of assessing a fastener's locking/loosening characteristics.

In 1996, Sase *et al.* [**2**] completed tests on a range of M8 locking nuts including nylon insert and all-metal prevailing torque nuts. They evaluated a range of test approaches to loosen fasteners including axial vibration, transverse vibration, and impact loadings. They concluded that transverse vibration results in the greatest loosening tendency. They developed two loosening devices, one which applied low-frequency vibration and another that used a shaker table to impart high-frequency transverse vibration into a joint. The nylon insert nuts they tested displayed some resistance to loosening after their preload had reduced to about 25 per cent of the starting value. On the nuts they tested, the loosening resistance was lost when the nut was used repeatedly. On the third use, the nylon insert nuts had lost their efficiency and acted as plain nuts. (It should be noted that the relevant performance standard [**3**] requires that such nuts retain a minimum specified level of prevailing torque for five re-uses.) On the high-frequency test they completed, they found that nylon and metal-insert nuts did not significantly perform any better than conventional nuts.

In 2004, DIN published [**13**] a document that categorized prevailing torque nuts as loss prevention devices. The basis of the DIN document was again that the joints would be tested under purely transverse vibration. They stated that the jamming effect on the threads generated by the prevailing torque feature of the nuts cannot prevent the initial loosening of a bolted connection. However, such loosening would be halted as soon as the loosening torque was equal to the prevailing torque. Thus a certain amount of preload should be retained to prevent the bolted connection from falling apart.

In 2006, Sawa *et al.* [**14**], and later in 2008, Bhattacharya *et al.* [**15**] reported on transverse vibration tests completed on a range of locking mechanisms which included nylon insert nuts. In agreement with previous researchers, they found that with this type of nut a steady-state preload was achieved after $\sim$1000 test cycles. The transverse displacement that was used in the Bhattacharya *et al.* tests was relatively small (0.175 mm) for the sizes of fasteners that they tested (M10 and M16). Cheatham *et al.* [**16**] in 2008 reported on the loosening of threaded inserts under transverse vibration. They found that locking Heli-Coil inserts (which have a prevailing torque) also retained a steady-state preload, but at levels well below the initial preload.

Junker [**6**] showed that the rate of loosening was independent of the frequency of the transverse vibration. Other workers, such as Finkelston [**9**], showed that increasing the amplitude of vibration increases the loosening rate (the decrease in the preload per vibration cycle), but the characteristic shape of the loosening curves remains the same. Additionally, increasing the frequency or amplitude of transverse vibration does not lead to the detachment of prevailing torque nuts. Tests in which bolted joints have been excited by purely axial harmonic vibration have been conducted by Hess and Davis [**17–21**]. They found that plain non-locking fasteners could be moved in the tightening or loosening direction dependent on the frequency of the exciting vibration. The preload range that they investigated was, however, extremely low (from 0 to 100 N) and well below the range normally used in practice. It can be readily shown that transverse vibration will lead to loosening of prevailing torque nuts but a residual preload will be retained in the bolt and detachment will not occur. What has not been demonstrated to date are the precise conditions that can lead to the detachment of prevailing torque nuts when they have been tightened to preloads commonly used in practice.

Based on the study of problematic joints in service, the authors have speculated that axial loading combined with transverse movement may induce complete self-loosening and subsequent detachment of prevailing torque nuts. The present standard test method (DIN 65151 [**12**]) based on purely transverse vibration is not able to replicate some service experience for prevailing torque nuts. Combined axial and shear loads on joints are common in many applications. This includes the engine mounting joint bolt shown in Fig. 2 and wheel to hub joints on a car. (Shear loads can be imposed onto the joint by braking and axial loads from cornering.)

This article presents results from an experimental investigation of loosening of prevailing torque nuts. All-metal prevailing torque nuts have been tested on a modified Junker-type test machine that allows axial as well as dynamic transverse loads to be imposed on the fasteners. The effect on the loosening characteristics was investigated for both static and dynamic axial loading. The objective was to establish under what conditions complete loosening of the fasteners would occur. The experiments described in this article have demonstrated that the combination of axial and transverse load has a profound effect on the loosening of prevailing torque nuts and conditions that lead to nut removal can be unambiguously defined. This is novel work and explains why instances of complete detachment of prevailing torque nuts can occur in practice with sometimes-catastrophic consequences.

2 LOOSENING TESTS USING A MODIFIED JUNKER'S MACHINE

Junker showed that transverse joint movement can cause complete self-loosening of plain nuts. Based on this, he developed a test that is the present-day 'standard method' of assessing the locking performance of fasteners. This test was formalized as the DIN 65151 test [**12**]. The test involves inducing transverse movement into a joint while simultaneously measuring the fastener preload. A typical preload decay graph from such a test for a prevailing torque nut is shown in Fig. 3. After an initial stage of self-loosening, nut rotation stops leaving a residual preload in the fastener. The magnitude of the residual preload retained by a prevailing torque nut depends on the value of the prevailing torque, the friction conditions, and the amplitude of the transverse movement. The residual preload can

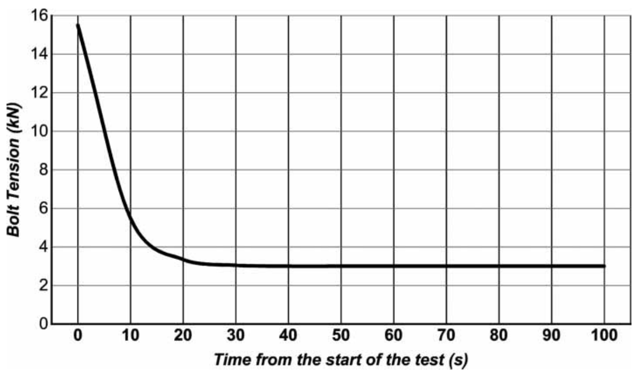

Fig. 3 Preload decay curve for a typical M8 all-metal prevailing torque nut in a Junker's test

also change with re-use since the prevailing torque can decrease when the nut is re-used.

For the work described in this article a Junker machine was modified to allow an axial load to be imparted into a bolted joint while simultaneously applying cyclic transverse displacement. The arrangement is illustrated in Figs 4 and 5. A load cell within the joint allows continuous monitoring of the bolt tension as transverse motion is applied to the bolted joint. On the test machine used in this series of tests, the transverse displacement was ±0.65 mm at a frequency of 12.5 Hz. The test bolt passes through a bush that clamps the load cell to a fixed base plate. The nut is attached to the bolt through a plate that is subjected to transverse movement. The moving and fixed base plates are separated by needle roller bearings. The purpose of these bearings is to minimize any friction between the joint surfaces that would resist transverse movement. An eccentric cam connected to an electric motor generates the transverse movement. Unlike a standard Junker test machine, the machine had a modification made to allow axial loading to be imparted into the fastener independent of the transverse movement. This was achieved by using miniature hydraulic jacks to impart the axial load into the joint with the oil pressure being used to regulate the magnitude of the loading. The bolt tension (preload) can be recorded continuously with the aid of a data-collection equipment.

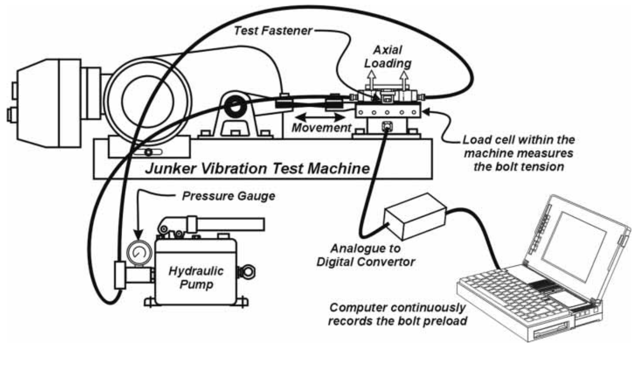

Fig. 4 Overall view of the test machine

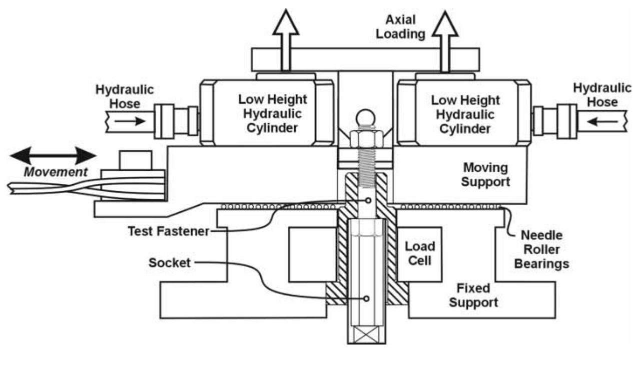

Fig. 5 Section through the test machine

3 EXPERIMENTAL RESULTS

The tests involved tightening M8 electro-zinc-plated bolts to a nominal preload of 15 kN prior to the start of transverse motion. An axial load was then applied to the joint, the axial load being smaller in magnitude than the preload. Because of the mechanics of the way the applied axial load is sustained by the joint, this caused the bolt preload to be only marginally increased. It was found that when transverse movement commenced, appropriate loading conditions led to prevailing torque nuts initially suffering a rapid loss of preload until the load in the bolt approached the axial load being applied by the hydraulic system. Nut rotation continues under the axial loading from the jacks. At the completion of the test, the hydraulic pressure is released reducing the load in the bolt to zero. This process is illustrated in Fig. 6.

Several types of all-metal prevailing torque nuts have been tested (but not reported here) and have been found to have similar loosening characteristics

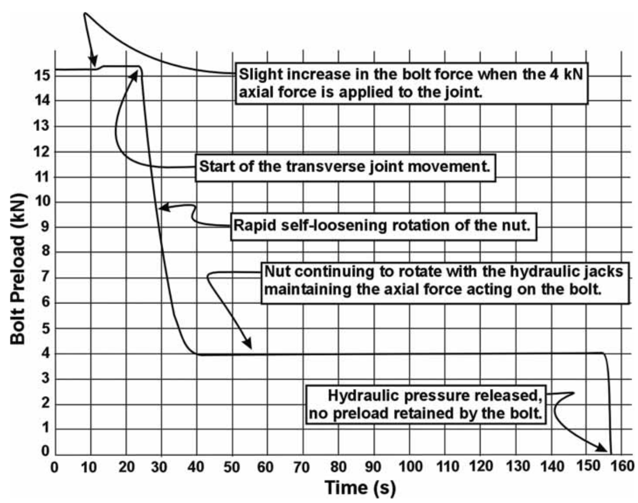

Fig. 6 Preload decay graph with transverse joint displacement and axial loading applied to the joint

under a standard DIN 65151 test. Figure 3 illustrates a typical preload decay curve for this type of nut in that they all initially self-loosen but retain a residual preload. A major factor in determining the value of the retained residual preload is the level of the prevailing torque of the nut. In general, the higher the prevailing torque, the higher will be the retained residual preload. (Higher also will be the torsional stress induced into the fastener for a given level of preload.) In total over 50 nuts were tested, and, in general, the prevailing torque of better quality nuts remained reasonably constant when they were re-used up to five times. In the series of tests reported here, tests were conducted to establish the value of axial loading, which would result in continued self-loosening of M8 prevailing torque nuts. The conditions for these tests are presented in Table 1 along with a summary of some of the results.

The procedure for testing first involved determining the residual preload retained by the prevailing torque nut with zero axial loading applied to the joint using a conventional Junker test. The results from one such test are shown in the graphs in Fig. 7(a). The residual preload retained by the bolt in this case is 3.1 kN. Figures 7(b) and (c) show decay curves when axial loading of 1.1 and 2.7 kN was applied. In both situations the bolt preload was retained at approximately the same level as if no axial loading has been applied. However, Fig. 7(d) shows the decay curve when an axial load of 3.1 kN is applied. In this case the nut continued to rotate, the axial load was maintained and compensated for the load loss that occurred from nut rotation. At the end of the test when the axial load was removed by releasing the pressure from the hydraulic cylinders, the residual preload was observed to be zero.

In many applications, the axial loading is intermittent rather than having a constant magnitude. Figure 8 illustrates results from a number of tests examining the effect that an intermittent axial loading had on the self-loosening of M8 prevailing torque nuts. Figure 8(a) shows the results of a test on a M8 all-metal prevailing torque nut with zero axial loading. The retained preload for this particular nut was ~0.3 kN without axial loading. Figure 8(b) illustrates the effect of an intermittent axial load of 0.7 kN being applied to the same nut. At each instance of loading, further rotation of the nut occurred, which incrementally reduced the preload. Figure 8(c) shows the results of another M8 all-metal prevailing torque nut with zero axial load. For this nut, the retained preload was ~2.4 kN. Figure 8(d) illustrates the effect of an intermittent axial load of 3.5 kN. Again, further rotation of the nut occurred that reduced the bolt preload to zero. Even after the preload had been reduced to zero, nut rotation could be observed when the axial load was applied while transverse joint movement was occurring.

Table 1 Results from Junker tests with and without axial loading being present

Test	Axial loading	Retained bolt preload at the end of the test (kN)	Details
1	None	0.8	
2	5 kN Intermittent	0	Same nut as test 1
3	None	3.2	
4	1.1 kN Constant	3.4	Same nut as test 3
5	2.7 kN Constant	2.7	Same nut as test 3
6	3.1 kN Constant	0	Same nut as test 3
7	4.1 kN Constant	0	Same nut as test 3
8	None	0.8	
9	1.1 kN Constant	0	Same nut as test 8
10	5.3 kN Constant	0	Same nut as test 8
11	5 kN Intermittent	0	Same nut as test 8
12	None	0.4	
13	0.4 kN Constant	0.4	Same nut as test 12
14	1 kN Constant	0	Same nut as test 12
15	0.6 kN Constant	0	Same nut as test 12
16	None	0.7	
17	0.8 kN Constant	0	Same nut as test 16
18	5 kN Intermittent	0	Same nut as test 16
19	5 kN Intermittent	0	Same nut as test 16
20	5 kN Constant	0	Same nut as test 16
21	5 kN Intermittent	0	Same nut as test 16
22	None	0.3	
23	0.7 kN Constant	0	Same nut as test 22
24	0.1 Constant	0.1	Same nut as test 22
25	0.7 kN Intermittent	0	Same nut as test 22
26	None	0.7	
27	0.5 kN Constant	2.4	Same nut as test 26
28	2.2 kN Constant	0	Same nut as test 26
29	3.5 kN Intermittent	0	Same nut as test 26

The initial preload in all the tests was 15 kN. The test duration was typically 2 min (1500 transverse movement cycles).

4 ANALYTICAL MODEL

In order to be able to build an analytical model of the loosening process, forces resisting loosening and those promoting loosening need to be established. The relationship between the torque applied to a nut and the preload generated by the bolt involves the thread and nut dimensions and the coefficient of friction in the threads and under the nut face. There are various forms of the torque-tension equation for threaded fasteners. One form presented here is commonly used for free spinning (plain) nuts, the tightening torque T being given [**22**]

$$T = \frac{F}{2}\left(\frac{p}{\pi} + \frac{\mu_t d_2}{\cos\alpha} + D_e \mu_n\right) \tag{1}$$

where D_e is the effective bearing diameter of the nut, F is the bolt preload, T is the tightening torque applied to the fastener, d_2 is the basic pitch diameter of the thread, d_i is the inner bearing diameter of the nut face, d_o is the outer bearing diameter of the nut, p is the pitch of the thread, α is the half included flank angle for the threads, μ_n is the coefficient of friction for the nut face or bolt head (whichever is rotated during tightening), and μ_t is the coefficient of friction for the threads.

The value of D_e is taken as the mean of the outer and inner bearing diameter of the nut [23]

$$D_e = \frac{d_o + d_i}{2} \tag{2}$$

The first term in the brackets in equation (1) is the torque to stretch the bolt, this torque always acts in the loosening direction and its magnitude is independent of friction. The second and third terms in the brackets of equation (1) represent the torque needed to overcome thread friction and the torque needed to overcome nut face friction. When transverse slip of the joint occurs, the nut slides over the joint and simultaneous slip occurs between the nut and the bolt threads. Under such conditions the resistance to rotation is significantly reduced. The prevailing torque present from the nut resists loosening rotation but in addition to the bolt stretch torque acting in the loosening direction, a further torque acts when slippage occurs on the threads. As shown by Sakai [24] this torque is due to differences in the resistance offered between the ascending and descending sides of the nut thread while it is sliding on the bolt thread. This is illustrated in Fig. 9. The thread surface is helical and is inclined relative to the slip direction by the lead angle β. The inclined surfaces, when a translational movement occurs, have an ascending side and a descending side which leads to a difference in the forces between the two sides which in turn generates a loosening torque. From Sakai's work [24], Fig. 10 shows linear slip occurring on a square thread. Friction acts

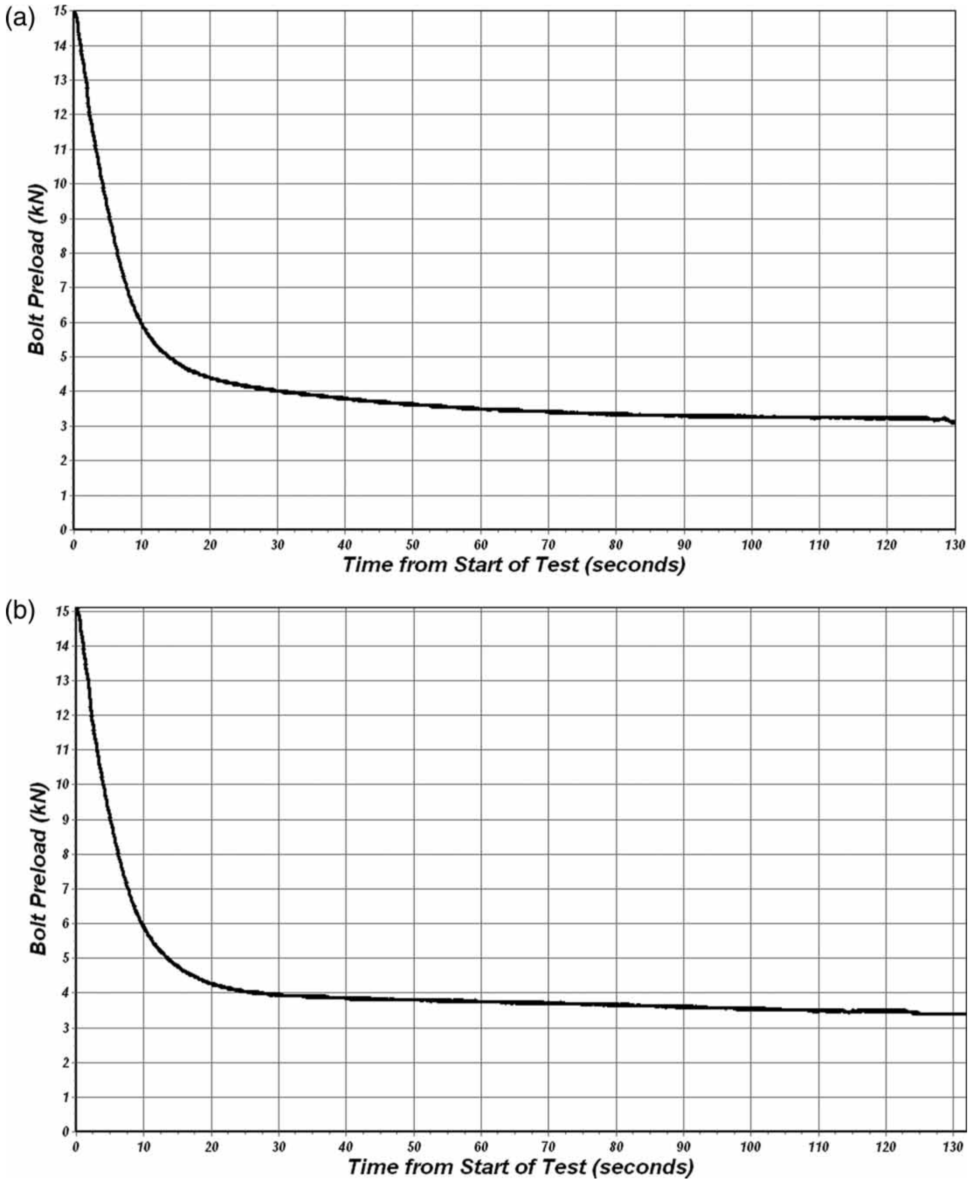

Fig. 7 Preload decay graph with transverse joint displacement and axial loading applied to the joint: (a) no axial loading applied, (b) 1.1 kN axial loading applied, (c) 2.7 kN axial loading applied, and (d) 3.1 kN axial loading applied

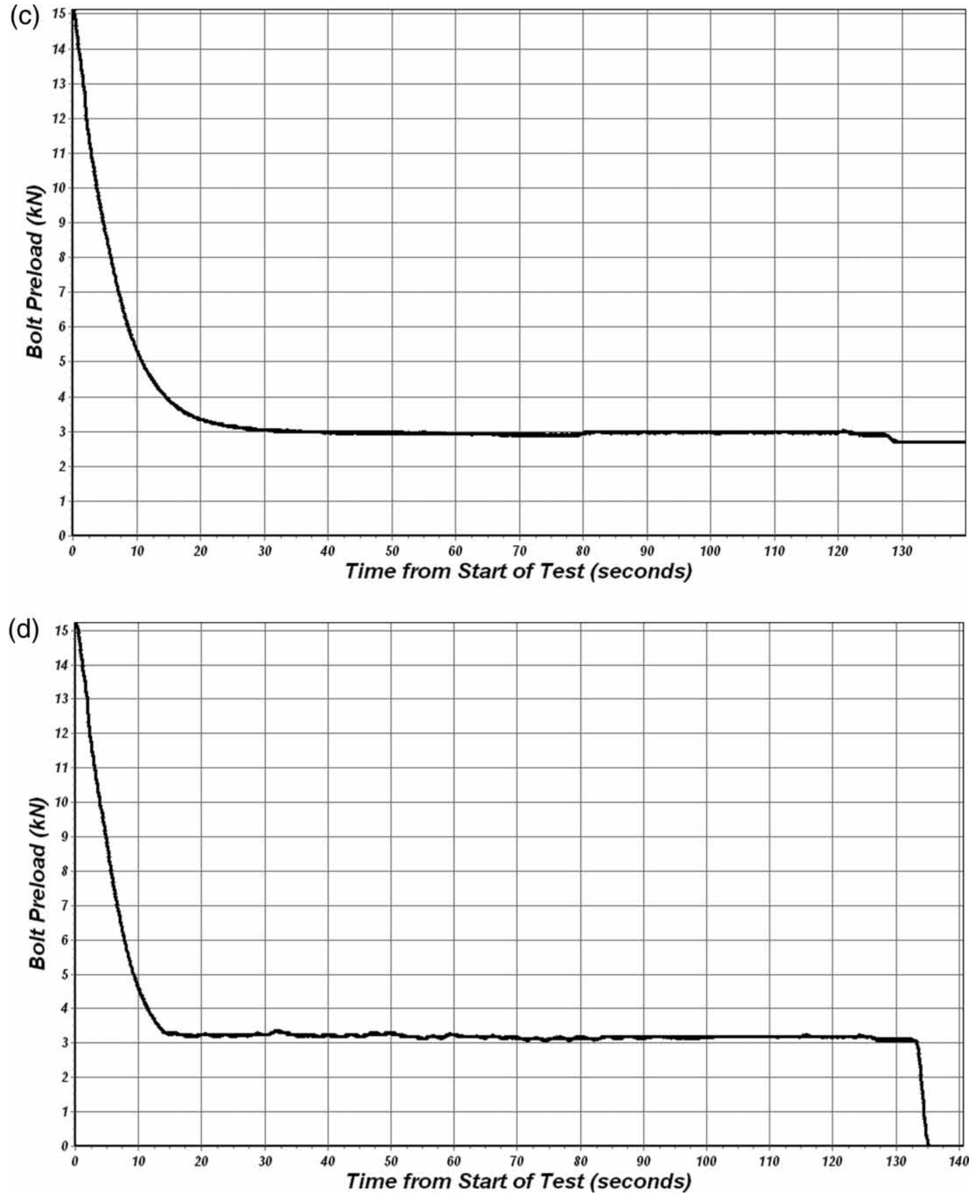

Fig. 7 (continued)

on the area dAs of the thread surface and the difference in forces resulting when slip occurs results in a loosening torque T_{ss} that (from reference [**24**]) can be expressed as

$$T_{ss} = \int_{r=r_1}^{r=r_2}\int_{\theta=0}^{\theta=\pi} \frac{F}{A_{slip}}\,dA_{slip}(\mu_t\cos\beta + \sin\beta') \times \cos\beta' r\sin\theta - \int_{r=r_1}^{r=r_2}\int_{\theta=0}^{\theta=\pi} \frac{F}{A_{slip}}\,dA_{slip}(\mu_t\cos\beta - \sin\beta') \times \cos\beta' r\sin\theta \quad (3)$$

where r_1 and r_2 are the inside and outside diameters, respectively, of the screw contact surface. The first term represents the ascent torque and the second term the descent torque acting on the bolt. The area of the screw contact surface A_{slip} is given by

$$A_{slip} = \frac{\pi(r_2^2 - r_1^2)}{\cos\beta} \quad (4)$$

From Fig. 10, $\tan\beta' = \tan\beta\sin\theta$ and since the lead angle β is small (3° for the M8 threads used in this study) we have $\beta' = \beta\sin\theta$ and $\cos\beta' = 1$ In this case, T_{ss} will assume the following approximate equation

$$T_{ss} = \frac{2F\beta}{A_{slip}}\int_{r=r_1}^{r=r_2}\int_{\theta=0}^{\theta=\pi} r^2\sin\theta\,d\theta\,dr \quad (5)$$

Integration of equation (5) leads to

$$T_{ss} = \frac{\pi F\beta}{3A_{slip}}(r_2^3 - r_1^3) = \frac{F\beta(r_2^2 + r_2r_1 + r_1^2)}{3(r_2 + r_1)} \quad (6)$$

The basic pitch diameter of the thread is denoted by d_2. In this case, it can be taken as $r_1 + r_2$, also from this $r_1 = d_2 - r_2$.

Substituting these values into equation (6) gives

$$T_{ss} \cong \frac{1}{4} F d_2 \beta \tag{7}$$

If p is the pitch of the thread and d_2 is the basic pitch diameter, the lead angle is the transverse movement of the screw completed in one revolution divided by the circumference around the thread at the pitch diameter

$$\beta = \tan^{-1}\left(\frac{p}{\pi d_2}\right) \tag{8}$$

Since the angle is small, equation (8) can be approximated as

$$\beta = \frac{P}{\pi d_2} \tag{9}$$

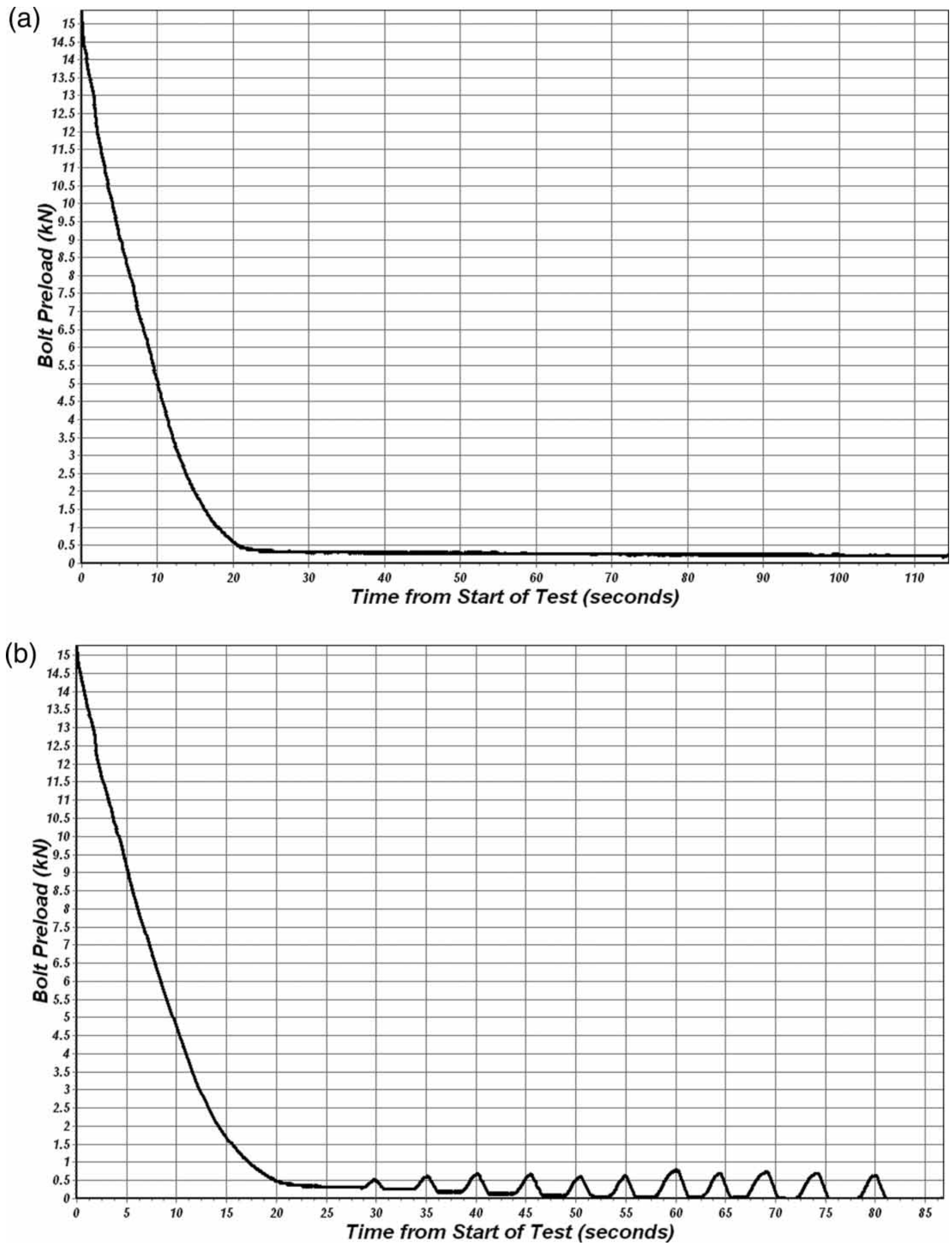

Fig. 8 Effect of transverse joint displacement and an intermittent axial loading on the loosening characteristics of M8 all metal prevailing torque nuts: (a) no axial loading −0.3 kN bolt preload retained, (b) nut as in test (a) with 0.7 kN intermittent axial loading, (c) no axial loading −2.4 kN bolt preload retained, and (d) nut as in test (c) with 3.5 kN intermittent axial loading

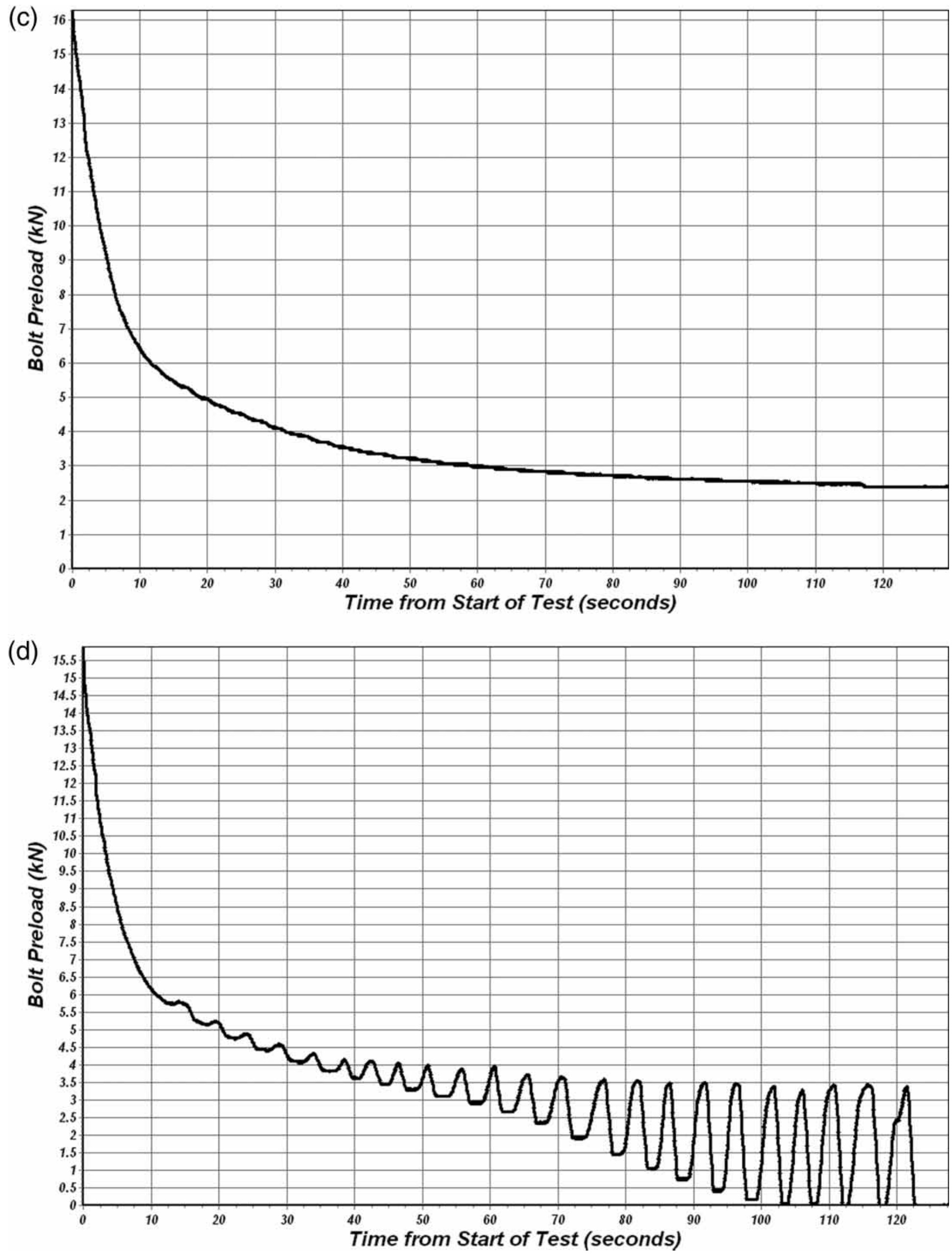

Fig. 8 (continued)

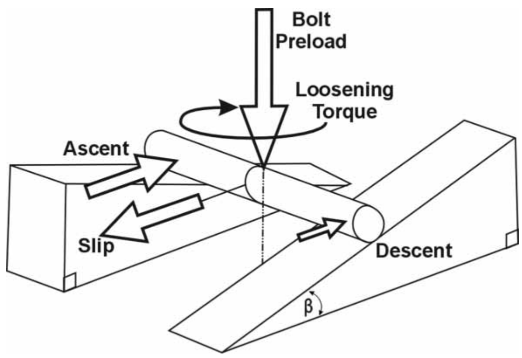

Fig. 9 Loosening torque generated by movement of the nut thread on the bolt thread

Substituting this value into equation (7) gives

$$T_{ss} = \frac{FP}{4\pi} \tag{10}$$

From equation (1), the first term in the equation represents the torque needed to stretch the fastener that always acts in the loosening direction. The second term represents the torque needed to overcome friction in the threads and the third term is the torque needed to overcome friction under the nut face. Friction resists nut rotation in both the loosening and tightening directions. Considering the loosening direction and adding the prevailing torque which also resists nut movement in any rotation direction gives

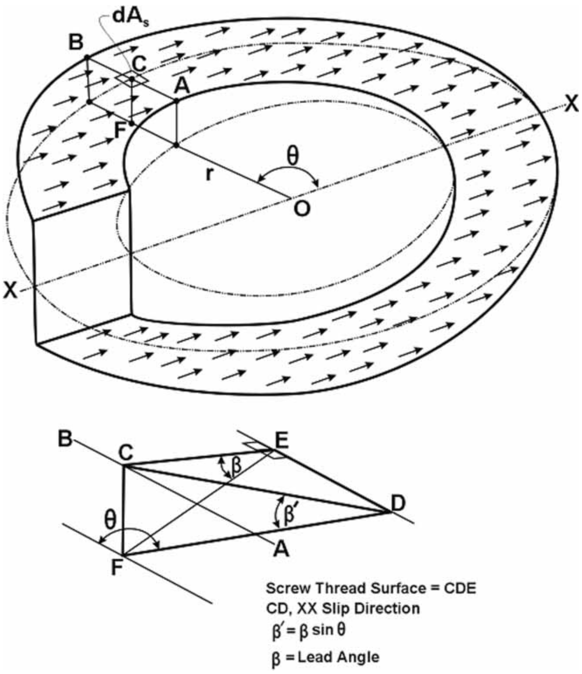

Fig. 10 Linear slip occurring on a square thread

the torque T_L needed to loosen a nut as

$$T_L = F\left(-\frac{P}{2\pi} + \frac{\mu_t d_2}{\cos\alpha} + \mu_n \frac{D_e}{2}\right) + T_P \tag{11}$$

The above equation applies to static situations. Because of the negative term, the loosening torque is typically 15–20 per cent lower than the tightening torque which can be readily practically demonstrated. (This applies when the nut is untightened immediately after being tightened. If the joint is left for a period of time friction changes can occur that can affect the situation such that the loosening torque can be either smaller or greater than the initial tightening torque.) When transverse slip is occurring to the joint, the torque acting in the loosening direction will be the torque given by equation (10) and the first term in equation (11). Combining these terms to establish the total torque acting in the loosening direction T_{loosen} under transverse slip conditions gives

$$T_{loosen} = \frac{FP}{4\pi} + \frac{FP}{2\pi} = \frac{3FP}{4\pi} \tag{12}$$

Under the conditions of slip under the nut face and in the threads, friction has been overcome in the transverse direction by external forces to the joint. Under such conditions Junker [**6**] considered that a non-locking nut would be free from friction in the circumferential direction. Sakai has concluded [**24**] that under slip conditions, the resistance of the nut to rotation is extremely small, he quotes friction coefficients of between 0.00 and 0.02. Strictly what are quoted are not true friction coefficients since friction has already been overcome by the external force causing transverse joint slip. The friction values he quotes are based on the additional force/torque to rotate the nut under slip conditions. Applications that include this effect include floor polishing machines, the machine being easier to move when the polishing disc is rotating. Also, a cork is easier to remove from a bottle if the cork is first rotating before being pulled. Overcome friction in one direction and the resistance to movement in another direction reduces dramatically. To differentiate between thread and head friction values under static conditions and values under the conditions of transverse slip, the terms μ_{ns} and μ_{ts} will be used for the nut face and thread friction coefficients in the rotational direction when transverse slip is occurring. Also the term T_{Ps} will be used to denote the prevailing torque under transverse slip. The torque T_R that resists loosening under the conditions of transverse slip is then given by

$$T_R = F\left(\frac{\mu_{ts} d_2}{\cos\alpha} + \mu_{ns} \frac{D_e}{2}\right) + T_{Ps} \tag{13}$$

For rotation of the nut in the loosening direction to occur, the torque acting in the loosening direction T_{loosen} must be greater than the torque T_R resisting loosening, hence for loosening to occur

$$\frac{3FP}{4\pi} > F\left(\frac{\mu_{ts} d_2}{\cos\alpha} + \mu_{ns} \frac{D_e}{2}\right) + T_{Ps} \tag{14}$$

Since the thread and head friction coefficients in the rotational direction when transverse slip is occurring (μ_{ts} and μ_{ns}) are close to zero [**24**], equation (14) simplifies to give the condition for self-loosening to occur with prevailing torque nuts

$$\frac{3Fp}{4\pi} > T_{ps} \tag{15}$$

For the M8 nuts used in this test series, the retained preload is typically in the range of 1–3 kN. This implies that T_{ps} is in the range of 0.3–0.9 Nm. The relevant standard [**3**] quotes a maximum first assembly prevailing torque of 6 Nm for this size and grade of nut and a minimum fifth removal prevailing torque of 0.6 Nm. The prevailing torque measured on the nuts used in these tests was between 1.5 and 2.3 Nm. The frictional drag generating the prevailing torque is likely to be reduced under transverse slip conditions, which is one explanation for the difference. The prevailing torque is generated by pressure applied by the nut to the bolt thread. Just as transverse joint movement reduces the rotational resistance of a plain nut, a prevailing torque will probably also be reduced under such conditions when transverse slip of the threads is occurring.

When a tensile axial load F_A is applied to a joint held together by a tightened bolt, the bolt does not sustain the full effect of the load, but usually only a small part of it. The majority of the applied load, typically 90 per cent for structural joints, reduces the clamp force on the joint provided by the bolt. The remaining 10 per cent of the load will increase the force in the bolt. This applies until joint separation occurs, i.e. the applied force exceeds the clamp force on the joint and a gap occurs in the joint. Under conditions of transverse joint slip, once self-loosening reduces the bolt preload such that joint separation will occur, it is the tensile axial load F_A that will be the cause of further nut rotation. The bolt would essentially act as if it had not been tightened, all the axial load being sustained as a tensile load in the bolt. Under such conditions, F_A can replace F in equation (15), which can be rearranged to give the following condition for F_A in order to ensure that continued rotation of the nut would not occur

$$\frac{4\pi T_{ps}}{3p} > F_A \tag{16}$$

If the conditions given by equation (16) were not met, continued rotation would occur until the nut detaches from the bolt. For free spinning nuts under transverse slip conditions, T_{ps} is zero or close to zero and subsequently detachment of the nut from the bolt can occur with small axial loads applied to the joint.

5 DISCUSSION

Based on the experiments completed and the measurements made, the effect of applying an axial load when transverse joint movement is occurring is to aid the self-loosening tendency of prevailing torque nuts. Whether or not complete loosening will occur depends on the magnitude of the applied axial load.

Tests have also shown that an intermittent axial load occurring when the joint is experiencing transverse slip can also result in the complete self-loosening of prevailing torque nuts and their possible detachment from bolts. In mechanical engineering applications, dynamic conditions often result in any axial loading being applied to a joint being variable in magnitude. The results of these tests point towards an occasional peak in axial load being especially detrimental to the security of prevailing torque nuts if the loading is assumed to be purely shear in nature.

The friction increasing feature that results in the prevailing torque on the majority of this type of nut is located generally on the top of the nut. Junker [**6**] speculated that some degree of resistance to loosening will be provided by such features restricting the thread movement of the nut relative to the bolt. However, tests by these authors have indicated that this effect can be routinely overcome under the right conditions.

6 CONCLUSIONS

It has been shown that under certain conditions loosening and the complete detachment of prevailing torque nuts can occur. Such a loosening process involves the application of an axial load when joint slip is occurring. The tests described in this article indicate that if the magnitude of the axial loading exceeds the residual preload in the bolt retained from sustaining transverse movement alone, all-metal prevailing torque nuts can completely detach from bolts.

A key finding of this study is that prevailing torque nuts cannot be defined as loss prevention devices when axial loading as well as transverse movement is applied to a joint. This implies are that the present standard test method (DIN 65151) should be revised to allow a true assessment of the locking ability of certain categories of fasteners.

Axial loading applied while transverse joint slip is occurring also affects the loosening characteristics of standard plain nuts. On a standard Junker test, plain nuts generally stop rotating when the preload reaches zero, i.e. the nut is retained on the bolt. In the presence of an axial load, such rotation does not stop and will continue until the nut detaches from the bolt. There are a number of examples of accidents resulting from nuts becoming completely detached from bolts. One such accident occurred on Friday 23 February 2007 when a train derailed on points at Lambrigg, near Grayrigg in Cumbria [**25**]. A significant factor in the occurrence of this accident was nuts becoming detached from the bolts, allowing the switch rail to be struck by the inner faces of passing train wheels. This resulted in the subsequent failures of other parts of the switch structure and ultimately the derailment of the train.

Applications that involve shear and axial loading being simultaneously applied to a joint are numerous in engineering. The reason why prevailing torque nuts have historically been frequently specified is the belief that if they came loose, they will not become detached from the bolt. This work has shown that the engineer may need to be more circumspect in this regard and give some study as to the magnitude of any axial load that may be acting on the joint. Knowledge of the circumstances under which a prevailing torque nut can become completely detached from the bolt will enable more reliable and safer designs to be engineered to prevent failures and accidents.

Based on the work of Junkers and other researchers, it is known that self-loosening of threaded fasteners can be prevented if sufficient preload is generated so that friction grip between the joint plates prevents the occurrence of transverse movement. In applications in which overload conditions can occasionally cause transverse joint movement, prevailing torque fasteners are frequently used in the belief that although partial loosening may occur, the nut will not become detached from the bolt. In the light of the research published here, this criterion requires to be updated to include the simultaneous effect that axial loading can have on the loosening process. Additionally, repeated re-use/tightening of prevailing torque nuts can reduce the prevailing torque due to thread wear and hence their effectiveness in resisting loosening. This work has concluded that the maximum simultaneous axial load that can be sustained when transverse joint movement occurs is equivalent to the preload retained by the prevailing torque nut on a normal Junkers test.

7 FURTHER WORK

These tests have been conducted on an all-metal variety of prevailing torque nut. Beside the non-metallic insert nut, many other types of locking devices have a prevailing torque feature. Adhesive applied to the threads is a commonly used locking method. Using too high a bond strength results in the nuts having to be cut off, so medium-strength adhesives are used so that removal is possible. Under a conventional Junkers test, such medium-strength threadlockers experience a degree of self-loosening similar to prevailing torque nuts. From a limited amount of work conducted to date, it appears that the presence of axial loading while transverse slip occurs also influences the loosening characteristics of other types of 'locking' devices. Further work in this area is needed.

In general, the prevailing torque is generated by pressure applied by the nut to the bolt thread; this appears to change if transverse movement of the threads relative to each other arises. The indications are that the value of prevailing torque under conditions of transverse slip is smaller than that which is measured under static conditions. The author intends to measure the prevailing torque under such conditions to facilitate the determination of the magnitude of the axial loading that the joint can sustain before complete self-loosening and subsequent possible detachment of the nut from the bolt.

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APPENDIX

Notation

A_{slip}	area of the screw contact surface
d_i	the inner bearing diameter of the nut face
d_o	the outer bearing diameter of the nut
d_2	the basic pitch diameter of the thread
D_e	the effective bearing diameter of the nut
F	bolt preload
F_A	the axial force applied to the joint
p	pitch of the thread
r_1	inside diameter of the screw contact surface
r_2	outside diameter of the screw contact surface
T	tightening torque applied to the fastener
T_{loosen}	total torque acting in the loosening direction
T_L	the torque required to loosen a bolt
T_{ps}	the prevailing torque from the nut while transverse slip is occurring
T_R	the torque resisting loosening
T_{ss}	loosening torque is due to differences in the resistance offered between the ascending and descending sides of the nut thread while it is sliding on the bolt thread
α	the half included angle for the threads
β	lead angle of the thread
μ_n	coefficient of friction for the nut face or bolt head (whichever is rotated during tightening)
μ_{ns}	nut face friction coefficient in the rotational direction when transverse slip is occurring
μ_t	coefficient of friction for the threads
μ_{ts}	thread friction coefficient in the rotational direction when transverse slip is occurring