Lubrication, Maintenance and Tribotechnology
LUBMAT 2023, Preston, UK

The Goldilocks Zone and the Coefficient of Friction of Threaded Fasteners

W. Eccles [1]

[1] Bolt Science Limited, Chorley, PR7 2PY, U.K.

Keywords: Threaded Fasteners, Machine Elements, Friction and Wear, Lubrication

ABSTRACT

Most engineering products use threaded fasteners with most of these fasteners being tightened. The purpose of tightening the fasteners is to produce a clamp force on the joint. It is this clamp force acting on the joint interface that is the critical parameter to ensure the joint's structural integrity. The clamp force resulting from the tightening depends on the magnitude of the tightening torque and the friction present under the nut face and in the threads. A great deal of emphasis is placed upon applying the correct torque value to a fastener. Occasionally, but to a much lesser extent in many instances, users may also ensure that the friction is within the expected limits.

To prevent the possibility of overtightening, the tightening torque is commonly determined using the minimum anticipated friction conditions. The fastener friction can be controlled, to some degree, using lubricants. Such a lower limit will indirectly define the maximum clamp force provided by the tightening process. An upper friction limit needs to be specified so that the minimum clamp force that would be achieved is controlled. For most joint designs, it is the minimum clamp force that determines whether the joint will be successful or not. Consequently, there are conflicting requirements between engineering a joint that can be efficiently tightened whilst reducing the risk from self-loosening. This paper discusses these conflicting requirements and how they create an optimum range for threaded fasteners, a 'Goldilocks Zone'. Such a zone is one in which the friction is neither too low nor too high. The paper also discusses a case study of an accident that occurred because of the fastener friction falling outside this zone.

1. INTRODUCTION

The correct tightening of threaded fasteners is critical for dynamically loaded joints. The tightening induces a tension, a preload, into the fastener that is reacted by a clamp force acting on the joint. The magnitude of the clamp force is the critical parameter in ensuring that the joint's structural integrity is safeguarded. Since the fastener preload force is difficult to measure on a production basis, controlling the torque applied to the fastener is the most common tightening approach. The torque being related to the fastener preload by the thread size and geometry and the magnitude of the friction present in the threads and under the nut face.

In volume production, a large amount of emphasis is given to ensuring that the torque being applied to the fastener is that which is defined in the design specification. To achieve a consistent preload, the fastener friction must be controlled as well as the magnitude of the torque being applied. Historically, a far greater emphasis has been on controlling the applied torque rather than controlling (or considering) the magnitude of the friction present.

Threaded fasteners that have been tightened are held in place by friction. If friction is below a certain threshold, once the socket on the tightening tool is removed following tightening, the fastener will immediately unwind. This will occur when the loosening torque present from the preload acting on the helix of the thread is greater than the frictional resistance torque. The value of friction needed to achieve this is very low, a coefficient of friction of the order of 0.02 to 0.03 is required, these values are somewhat below that currently achievable by the best of lubricants. There is a concern, however, that at low, but achievable friction levels, the fastener would unwind at the slightest micromovement of the joint. Accordingly, there is a practical requirement, based over concern about this spontaneous loosening, to place a lower limit on the fastener friction level. Most large OEM's place such a lower limit on fastener friction, defining the limit that they will accept in their purchase standards.

The specification for the tightening torque is determined using the minimum anticipated friction value to prevent the possibility of overtightening. Since such a lower limit will indirectly define the maximum preload, an upper friction limit needs to be specified so that the minimum preload that would be achieved is controlled. As the thread friction increases, so does the torsion sustained by the fastener and the torque needed to achieve a specific preload value. Since yielding of the fastener occurs as a combination of tensile and torsional shear stress, and the preload is dependent upon the tensile stress, as the thread friction increases, the maximum preload achievable decreases. Hence, from a preload and tightening torque perspective, it is desirable that the friction is kept as low as possible.

The factors that influence these two conflicting requirements are discussed in this paper. In particular, the paper discusses how the acceptable range of values for the coefficients of friction, the 'Goldilocks Zone', have been established. The paper also covers how in practice friction is controlled and measured in threaded fasteners and gives a case study of an accident that occurred because of the fastener friction falling outside the desired zone.

2. THE TIGHTENING TORQUE - BOLT PRELOAD RELATIONSHIP

Since the fastener preload force is difficult to measure on a production basis, controlling the torque applied to the fastener is the most common tightening approach. Tightening a nut by applying torque induces torsion into the fastener as well as tension. The torsion is primarily due to thread friction. The higher the thread friction, the higher will be the torsional stress. Yielding in fasteners during the tightening process is due to the combined effects of both axial tensile stress and torsional stress. The higher the torsional stress, lower will be the tensile stress at yield and so consequently, lower will be the bolt preload.

One approach is to combine both the axial tensile and torsional shear stresses using the Von-Mises criteria so that a consistent percentage utilisation of the bolt's yield strength is achieved [1]. This allows the bolt's preload to be computed:

$$F = \frac{A_0.v.R_{p0.2\min}}{\sqrt{1+3\left[\frac{3}{2}.\frac{d_2}{d_0}\left(\frac{P}{\pi.d_2}+1.155\mu_t\right)\right]^2}}$$

Once the bolt preload F is known, the relationship between the bolt preload F and the tightening torque T can be determined. For thread angles of 60 degrees (which is applicable to metric and unified thread forms) T is given by:

$$T = F\left[\frac{P}{2\pi}+0.577\,d_2\,\mu_t+\mu_n\frac{D_e}{2}\right]$$

Where:

T	Total tightening torque.
F	Bolt preload.
v	Proportion of the yield strength to be utilised – normally taken as 0.9.
$R_{p0.2min}$	Minimum yield strength or the 0.2% proportional stress limit.
d_2	Basic pitch diameter of the thread.
P	Pitch of the thread.
d_0	Diameter of the relevant bolt section to be used in the calculations.
A_0	Area related to diameter d_0. Normally, As is used, the thread's stress area.
μ_t	Coefficient of friction for the threads.
μ_n	Coefficient of friction for the nut face or bolt head.
D_e	The effective bearing diameter of the nut or bolt head.

Due to issues in measuring the thread and head friction values on installed fasteners, the total coefficient of friction, μ_{total}, is often used in torque calculations. The total coefficient of friction is a weighted average of the nut face and thread friction values (defined in ISO 16047 [2]). In practice, the coefficient of friction falls within a range of values for a particular fastener size and coating condition rather than being a single specific value. As the friction value increases, so does the torque value needed to achieve the specified bolt preload. It is recommended that the minimum value in the range is used to determine the tightening torque. Such a minimum value will produce a lower bound torque value. Accordingly, when such a value is specified, a fastener will not be over tightened so that the stress will be greater than the criteria selected. As a result, the greater the range in the possible friction values, the greater will be the potential scatter in the bolt preload. This is one reason why some control is needed on the upper friction value in the range.

3. THE GOLDILOCKS ZONE FOR THE COEFFICIENT OF FRICTION

3.1. Coefficient of Friction too low

The value of the fastener coefficient of friction influences the ease and rate that self-loosening occurs. As the fastener friction decreases, the torque needed to overcome friction also decreases, but the torque needed to stretch the fastener remains constant. The stretch torque is a function of the thread pitch and the preload and does not include friction. Accordingly, as the friction decreases the proportion of the tightening torque required to stretch the bolt increases. If the decrease in friction is continued, a point is reached where the thread extension torque becomes equal to the friction torque. An example of this is shown in figure 1.

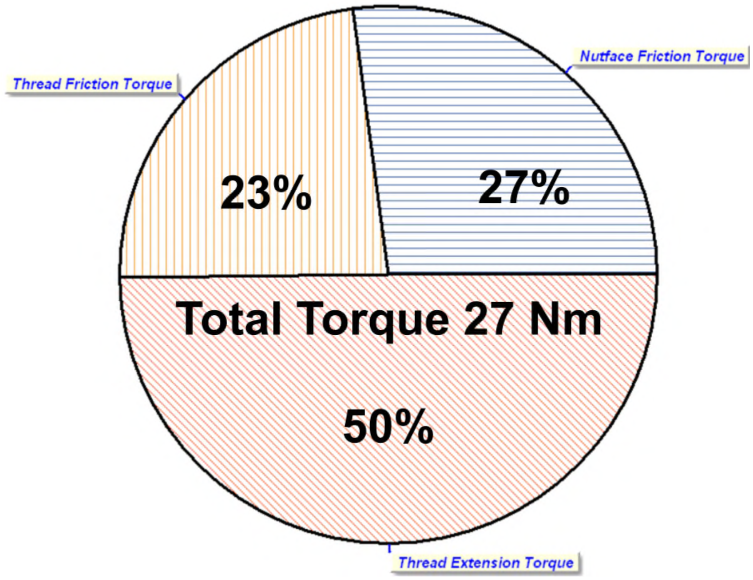

Figure 1: Example Torque Distribution Chart for an M12 8.8 Bolt Fully Tightened with μ=0.02

The chart in figure 1 shows the torque distribution for an M12 property class 8.8 bolt at a friction level of 0.02. These values were computed using the torque equation presented earlier in this paper. If friction reached such a low level, that is, at a value of 0.02 or below, the bolt or nut, would unwind as soon as the socket on the tightening tool was removed from the nut. It would not maintain the preload. It is the stretch torque component that drives the loosening process. There is a concern that at low levels of friction, but above the 0.02 value, would allow the bolt to rotate too readily. High performance greases on some smooth surfaces allow a fastener friction coefficient of the order of 0.04 to 0.05 to be achieved.

There is a concern that very low levels of friction would too readily allow self-loosening to occur when joints are dynamically loaded. Accordingly, industry bodies such as VDI (Verein Deutscher Ingenieure) advise, and many manufacturers specify in their internal standards, that a friction range is required in their purchased fasteners. VDI 2230 [1] advises that the fastener friction be of friction coefficient class B. This allows the range in the coefficient of friction to be within 0.08 to 0.16. As can be seen in figure 2, volume manufacturers specify their own ranges. The target mean value being usually between 0.11 to 0.15, dependent upon the manufacturer.

For structural threaded fasteners, EN 14399-3 [3], defines a K factor (alternatively referred to as a “nut factor” or the “torque coefficient”) to express desired friction levels. Figure 2 shows some of K from EN 14899-3 and some industrial standards converted to their approximate coefficient of friction values. As can be seen, the range specified approximates to VDI friction coefficient class A. Lubrication would be required to reach these friction levels. The EN standard specifies tightening to a high percentage of the tensile strength of the bolt and are not intended to be re-used. In general terms, the variation in the applied forces acting on structural joints, the vibration levels sustained, and the possibility of load reversals, are not as great as those that can be experienced in mechanical engineering.

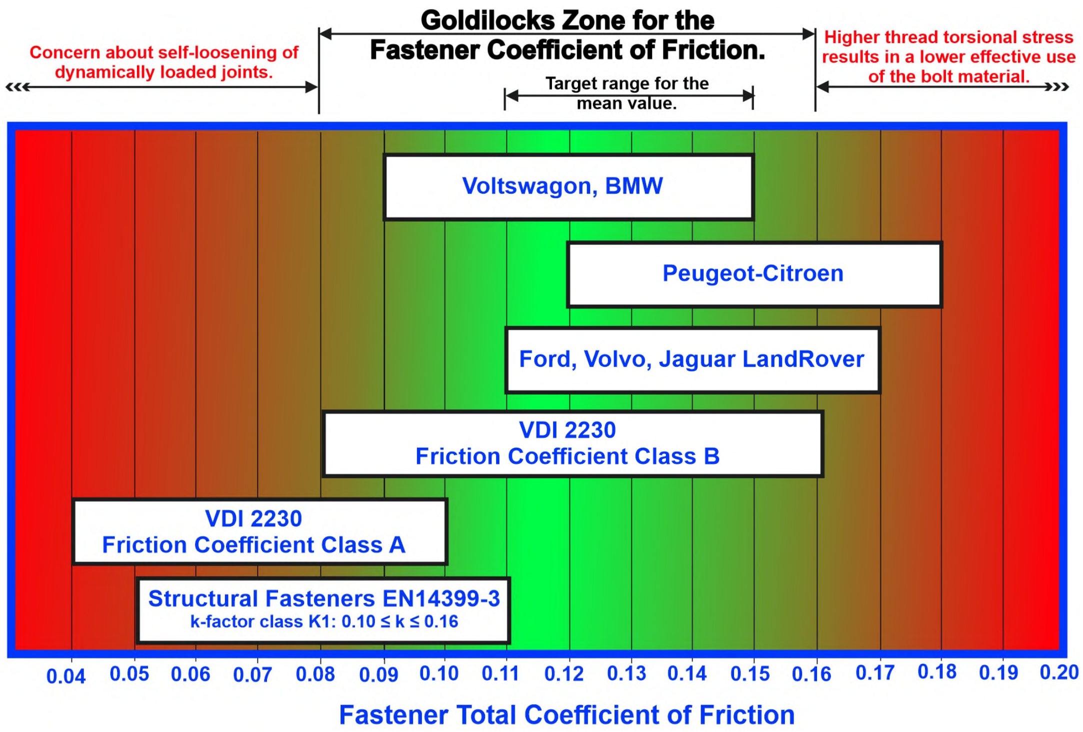

Figure 2 Specified Friction Ranges for Threaded Fasteners

3.2. Coefficient of Friction too high

The upper limit of the range of acceptable values is required so that the scatter in the preload is not excessive. Some coatings and materials have a very high coefficient of friction in the unlubricated condition. For example, friction coefficients can exceed 0.3 for galvanized coatings and stainless steel.

Tightening a bolt using torque so that a preload is produced is not the same as stretching it in a tensile test machine. A tensile test produces only axial stress in the thread. Whereas, if the bolt is torque tightened, a torsional stress is induced as well as an axial stress. The torsional stress is the result of friction in the threads as well as the torque needed to stretch the bolt. Accordingly, since the bolt yields as a result of the combined stress, a lower axial stress, and hence preload, will be present when the bolt yields when torque tightened than is the case if the bolt was tested in a tensile testing machine.

Increasing the thread friction will increase the torsional stress and decrease the axial stress available. Subsequently, lower will be the preload that can be achieved before yielding occurs. Most bolts are not tightened to yield but this effect is still present. At high levels of friction, the preload will be less than what is achievable at more moderate levels.

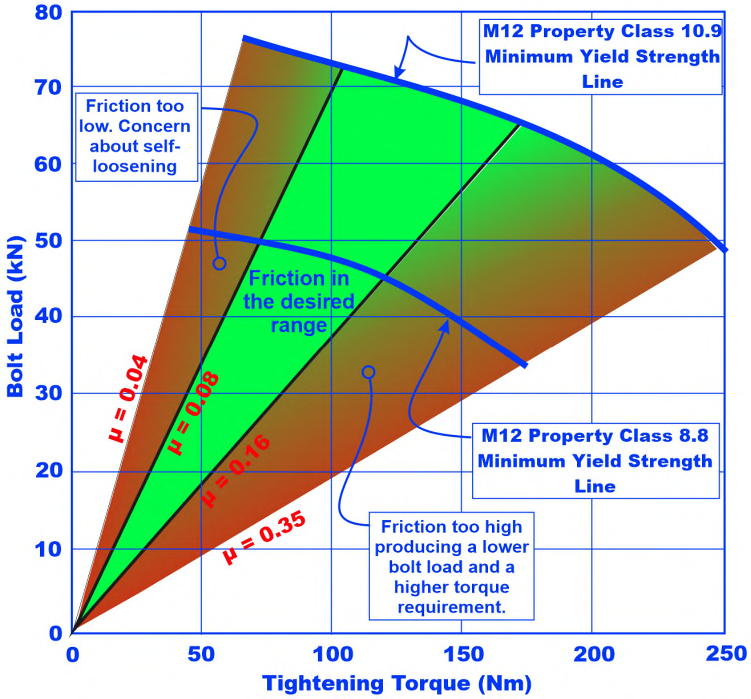

Figure 3 An example of how the coefficient of friction influences the tightening torque and achievable bolt preload.

Using the equations presented earlier in this paper, the effect of friction on the preload and the torque value can be established for any size of fastener. Figure 3 shows the behavior of M12 bolts of property classes 8.8 and 10.9 tightened to their minimum yield strength. For a property class 8.8 bolt, if the coefficient of friction was 0.12, it would require a torque of 92 Nm and would provide 47 kN of preload. Whereas the same bolt with a coefficient of friction of 0.35 would require a torque of 171 Nm and the preload would be 34 kN. It can be appreciated from figure 3 that it is desirable to keep the friction reasonably low. Also, it can be seen from the chart the sensitivity of the tightening torque to the friction level present.

3.3. The Goldilocks Zone

From a purely preload and bolt stress standpoint it is best that the fastener friction is minimised. Too low a friction value raises the concern that at low levels it would too readily allow self-loosening on dynamically loaded joints. Although research is sparse about the issue, the 0.02 to 0.08 coefficient of friction range is a “no-go” zone for fasteners for many manufacturers. Setting the lower limit too high results in inefficient tightening, a higher torque is required and the maximum preload achievable is lowered. Once a lower limit is defined, an upper limit is required to limit the scatter in the bolt preload that will be potentially experienced. As shown in figure 2, different manufacturers have different lower friction limits. To allow for the inherent variability associated with friction, a 0.06 difference is usually specified between the minimum and maximum friction values.

4. CONTROLLING THE COEFFICIENT OF FRICTION

Traditionally, a lubricant such as a grease, is applied to both the nut face and the thread to control the coefficient of friction of the fastener prior to its assembly. The disadvantage of this in volume assembly is that the lubricant could be missed off and secondly, it may inadvertently contaminate other surfaces. Most dynamically loaded joints rely upon friction grip between the inner joint surfaces to transmit any shear loading. For such joints, the highest possible friction is desirable that would be compromised if such surfaces were inadvertently contaminated with grease. Accordingly, for volume production, a dry lubricant is usually used on fasteners to control friction for fasteners used in volume production.

A section through a modern electroplated coating is shown in figure 4. The main functions of a fastener coating are to prevent corrosion and to control the fastener's friction properties. The thickness of coating that can be applied is limited since the threads must be mated together, this would be compromised by too thick a coating. A conversion coating, such as a chromate, chemically combines with the metal layer to enhance corrosion resistance. This can be further enhanced using a sealant that acts as a barrier to the metal layer.

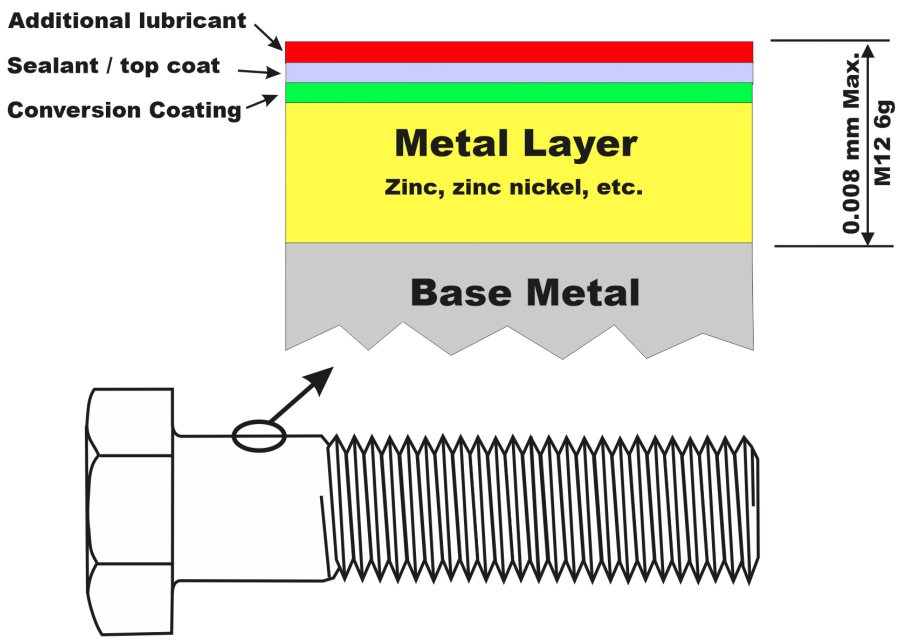

Figure 4 Section through an electroplated coating on a fastener

Lubricant topcoats are often based upon PTFE or proprietary compounds such as Torque 'N' Tension friction control fluid (from MacDermid Enthone). Different formulations allow the friction value to be adjusted to meet the requirements of the friction specifications of major fastener users such as car companies. Lubricant topcoats are also used on non-electroplated coatings such as zinc flake.

Section A.2.1 of ISO 4042 [4] states: "*For fasteners with ISO metric thread, at least one of the mating threaded fasteners should be lubricated for a consistent torque/clamp force relationship and to achieve a specific clamp force.*" Frequently the end-user is unaware of such a requirement and unless specified, no additional lubricant on the coating will be applied to the fasteners. This will be to the detriment of the fastener performance and the joint integrity.

5. CASE STUDY.

On 26 August 2020, a freight train travelling from Milford Haven in Wales derailed near Llangennech in Carmarthenshire. The freight train comprised 25 tank wagons, each with 75 tonnes of diesel and fuel oil. Ten of the wagons derailed spilling 446000 litres of fuel, three of the tanks caught fire consuming about 116000 litres. The remaining fuel spilled into an area of special scientific interest and a special area of conservation. Residents were evacuated late at night by emergency services, and it took 33 hours for the fire and rescue service to extinguish the fire.

The UK Rail Accident Investigation Branch (RAIB) investigation concluded in their report [5] that the derailment was likely to be due to the brakes being applied on one set of wheels during the journey. The evidence indicates that the brakes had been applied due to a fault on a relay valve joint in which two M10 nuts had become loose. Due to subsequent movement of an 'O-ring' that was present within the joint, air was allowed to pass and activate the brake actuators. The wagon with the brakes applied derailed when passing a set of points, due to a flat being worn on the wheels, decoupling itself from the train and derailing several of the wagons behind it.

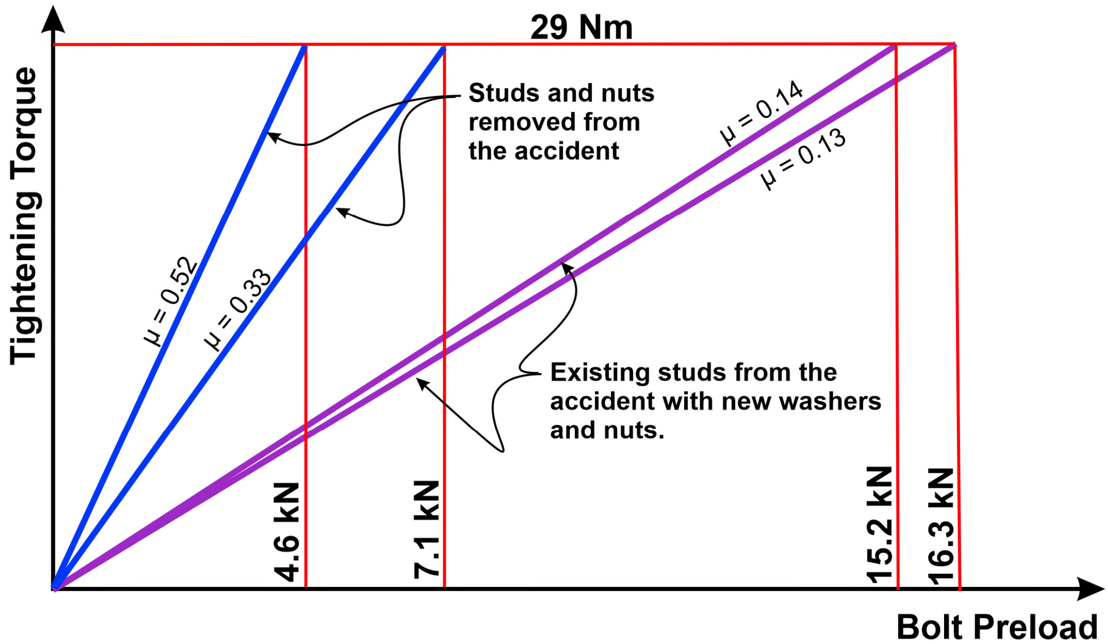

Figure 5 Re-used nuts compared with new nuts; all the tests being conducted on existing studs.

When a previous service was completed on the relay valve, nuts had been re-used and washers missed off the assembly. Figure 5 shows the results of tests conducted to establish the torque-tension relationship for used and new fasteners. Tightening the two nuts from the accident assembly to 29 Nm tightening torque resulted in a preload of 4.6 kN and 7.1 kN. When new nuts and washers were used on the existing studs, the results were 15.2 kN and 16.3 kN. As can be seen, the coefficient of total friction for new nuts was in the 0.13 to 0.14 range, within the Goldilocks zone. The re-used nuts displayed a coefficient of total friction of between 0.33 to 0.55, well outside this desirable zone.

A Preload Requirement Chart provides a graphical representation of the forces acting on the joint interface. Such a chart for the joint with re-used nuts is shown in figure 6. The forces acting on the joint are due to inertia and can be determined from the mass of the assembly and the acceleration levels that the rail vehicle is subjected. The key requirement from a bolted joint is that there must always be a residual clamp force acting on the joint once any relaxation losses, such as embedding, and the applied forces are considered. Accordingly, on the chart, there should be a gap between the minimum preload and the total preload requirement. As can be seen from the chart, this is not the case for the joint when previously used nuts are used without the washers being present.

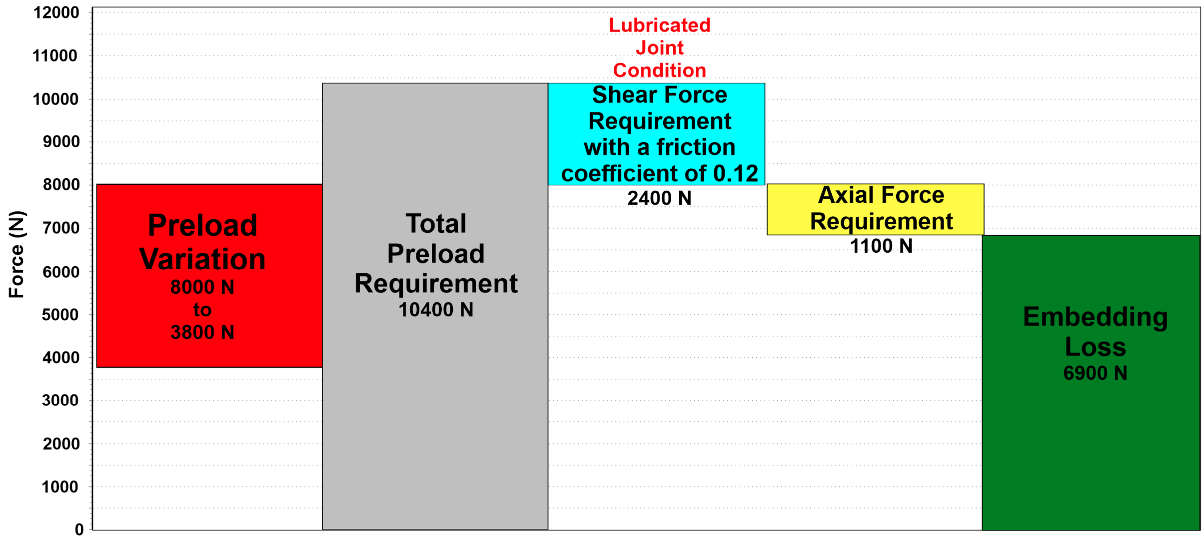

Figure 6 Preload requirement chart for the joint when re-used nuts are used.

In the analysis, consideration is given to the joint being both in the dry condition and lubricated, that is, there was some grease or oil present on the interface. Presence of grease would require a higher clamp force to prevent joint slip from occurring. If grease was present, it would also facilitate movement of the 'O ring'. The reason why the preload variation is slightly larger than that measured at 29 Nm is due to there being a tolerance on the torque value.

As can be seen from the chart, the embedding loss is likely to cause the nuts to loosen over time. That is, total preload loss is likely to occur if previously used nuts are used without the washers being present. This is likely to occur irrespective of whether there is grease present on the joint interface. Grease is likely to have been present since it was normal practice to apply grease to keep the ‘O rings’ from falling out of their respective recesses during assembly. Without specific work instructions to the contrary, grease deposits could contaminate the joint surface and reduce the friction present making the joint more likely to slip. The nuts coming loose allowed a gap to open in the joint. This provided opportunity for one of the ‘O rings’ to migrate within the joint allowing air to pass into the brake actuator.

The line was placed back into service in March 2021, over six months after the accident after a total of 37,500 hours of repair and environmental protection work. Some 30,000 tonnes of soil had been removed from the accident site. Mr Simon French, the Chief Inspector of Rail Accidents, said that there was *"a failure to appreciate*" the importance of the correct fastenings in the braking system. The case study focuses on the fastener related causes of the accident and what lessons can be learned and applied to both rail and other industries.

6. CONCLUSIONS

Historically a great deal of emphasis has been placed on controlling the torque applied to secure a threaded fastener. It is a common belief in industry that if the correct torque value is achieved, the structural integrity of the joint will be ensured. This is only the case if the fastener friction is also controlled. Many problems and accidents have resulted from the failure to control the fastener friction. Sometimes it is not even realized by assembly staff that it is even a factor in the integrity of a tightened fastener.

Large manufacturers tend to control the acceptable friction range for fasteners by their purchase specifications. This acceptable friction range, the Goldilocks range, is usually in the 0.08 to 0.16 range. Higher levels of friction require a higher tightening torque whilst producing a lower preload based upon a specified utilisation of the strength of the fastener. If the friction value is below 0.08, the concern is that the fastener would too readily allow self-loosening to occur when joints are dynamically loaded. This concern is currently not quantified, and more work is needed to assess the actual risk in using fasteners displaying a low level of friction in dynamic applications. There are advantages of low friction coefficients, such as a lower torque value being needed and a higher potential preload, in using fasteners with low friction levels.

REFERENCES

[1] VDI 2230 Part 1: Systematic calculation of highly stressed bolted joints. Joints with one cylindrical bolt.
[2] ISO 16047 Fasteners - torque clamp force testing.
[3] BS EN 14399-3 High-strength structural bolting for preloading -Part 3: System HR Hexagon bolt and nut assemblies
[4] ISO 4042: Fasteners - Electroplated coating systems.
[5] U.K. Rail Accident Investigation Branch Report 01/2022