Tribological Considerations of Threaded Fastener Friction and the Importance of Lubrication

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Abstract

The torque-tension relationship of threaded fasteners affects almost all engineering disciplines. Tribological processes at fastener interfaces manifest as the system's friction coefficient. Lubrication-related influences are usually described empirically using K or μ. The drive towards lightweight fastener materials in engineering systems and lubricants with reduced environmental impact is challenging existing knowledge and industrial practice in a range of applications, many safety critical. More comprehensive understanding is needed to achieve repeatable friction during assembly and re-assembly, resistance to loosening and fretting during operation, and effective anti-seize for disassembly with a growing range of materials and lubricants. The lubricants considered showed three predominant lubrication mechanisms: plastic deformation of metal powders; burnishing/alignment of molybdenum disulphide, MoS_2; and adhering/embedding of non-metal particles. Multivariate analysis identified key sensitivities for these mechanisms. Assembly generated changes at fastener surfaces and in the lubricating materials. Re-assembly exhibited significant reductions in friction.

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1. Introduction and Tribological Context

1.1. Threaded Fasteners and Torque Controlled Tightening

As most threaded fasteners are torque-controlled tensioned, interface frictional properties are crucial to accuracy and reliability, especially if fastener tension is predicted by the torque-tension relationship rather than directly measured [1]. Few systematic investigations of fundamental tribological mechanisms have been published, and most tribologically-focussed studies have not focussed on the lubricant.

The need to reduce environmental impact has driven changes in available/desirable lubricant ingredients. This dovetails with engineering trends towards lighter, higher strength materials for improved efficiency and performance. To adapt and optimise threaded fastener systems to the significant changes introduced by these developments (and others) requires understanding of the fundamental tribological mechanisms.

In torque-controlled tensioning, the tension-torsion relationship can be modelled using variants of the 'Long Form' equation [2-5]. In this study, the following form was used:

$$\boldsymbol{T} = \boldsymbol{F} \times \left[(\boldsymbol{P}/2\boldsymbol{\pi}) + (\boldsymbol{\mu} \times \mathbf{0.577} \times \boldsymbol{d_2}) + (\boldsymbol{\mu} \times \mathbf{0.5} \times \boldsymbol{d_f})\right] + \boldsymbol{T_P} \tag{1}$$

where P is the thread pitch, d_2 is the thread pitch diameter, d_f is the bearing surface friction diameter. This uses the 'total friction coefficient' (μ) approach in ISO 16047 [6] and DIN 65946 [3], assuming identical friction coefficients at the bolt head / nut face (μ_H) and thread (μ_T); the error from this assumption is reportedly 1-2% [3]. The DIN 65946 provision for prevailing torque fasteners adds a constant prevailing torque (T_P) [3]. Long form equations are more accurate than k factor equations [5,7].

Bickford estimated that 75 variables affect the tension-torsion relationship and 30-40 variables affect the friction coefficient [2]. Several have large effects on friction coefficients, especially fastener material [8,9], size [10], lubricant formulation [7,11], use of washers [12,13], and repeat assemblies [7,11,14,15].

Accurate and reliable torque-controlled tightening depends on the friction coefficient in two key aspects:

- the ***magnitude*** should be neither too large [2,16,17], nor too low [18]. Some OEM specifications give acceptable ranges of 0.12-0.18 and 0.09-0.14 [19], and 0.10-0.16 [12].

- ***variation*** in friction from fastener-to-fastener should be minimised to reduce tension variation [20,21].

Because ~90% of input torsional energy is dissipated as friction, friction variation has disproportionate influence on tension variation [2,22,23]. More accurate but more complex methods can reduce variance, e.g., yield tensioning [1,24], ultrasonic tension measurement [25–27] and loads cells, etc. [10,22,28,29].

1.2. Repeat Assembly

In the overwhelming majority of precision and critical systems, fasteners are used once. In some systems, fasteners are re-used, e.g. the A-A-59004A standard, where torque setting parameters are calculated using re-lubricated, re-assembled fasteners [30]. Whilst prohibited in some systems and not ideal practice, fastener re-use could save cost, time and waste and increase the sustainability of the system. Re-use is not recommended unless changes to the system arising from re-use can be confidently accounted for.

Eccles et al [11,19] showed that tension achieved in the same unlubricated BZP fastener reduced progressively for 4-5 repeat assemblies using the same assembly torque, after which an equilibrium was reached. Zinc coating removal occurred first, then abrasive wear progressed to adhesive wear. Others observed larger friction coefficient on second assemblies, typically with unlubricated fasteners [31–33] but not always [7,14]. Friction coefficient variation can also increase with repeat assemblies [34]. Nassar and Sun [15] showed that initial surface roughness affected the progression of friction. Others showed that friction coefficient can be significantly lower on second assemblies [7,35], suggesting 'running-in' of surface [7]. Typically, this was only observed for well-lubricated fasteners [33]. This reduced friction coefficient can cause overtightening failures if not compensated for [36]. Overall, well lubricated fasteners have lower friction coefficient on repeated assembly and poorly lubricated fasteners have progressively higher friction coefficient [2,21].

1.3. Anti-Seize Materials

A good lubricant should (i) generate accurate [20,34,37] and repeatable frictional properties [2,20,21] on assembly, (ii) resist loosening [18] and fretting wear during operation [38], and (iii) allow controlled and damage-free release on disassembly [27,34,39]. Other researchers have reviewed lubricant technologies:

- Eccles [11] compared oils, grease and MoS_2 coatings: Greases generated lower variation in repeat assemblies versus oil, but that copper-containing grease performed best,
- a variety of lubricating solids have been considered, e.g. copper [7,11,40], nickel [7,16,34], calcium fluoride [7], silver [24,35,42,43], zinc [16], graphite [43,44], MoS_2 [7,11,16,36] and PTFE [7,37], but often individually rather than comparatively, sometimes as a model compound [7,16] or as a commercial formulation [37,44,45],
- generic oils and greases were often used where lubrication was not a focus [20,31,35,40,41],
- resin-bonded and deposited coatings can be beneficial in controlling friction and resisting seizure/galling [34,43,46],
- threadlocker adhesives can provide lubrication but extreme care is required to control friction increase during curing [41,47];
- coated fastener components and inserts, e.g. silver [35,43,44], nickel [24], can provide galling resistance on assembly, with or without a lubricant.

In fretting, using similar lubricant formulations to anti-seize compounds, Waterhouse and Allery [48] showed that copper powders formed lubricating films by being plastically deformed into a thin layer at the interface; non-metallic powders, e.g., boron carbide, were embedded into surfaces, and increased adhesion between the surfaces and fatigue life. In rolling element bearings: Dwyer Joyce observed copper particles plastically deforming in a rolling elastohydrodynamic contact, whereas ceramic particles fractured in the inlet region before embedding in the surfaces [49]; Nikas described differences between these brittle (friable) and ductile behaviours [50], and that increasing the hardness of the particle caused deeper, shorter and steeper dents, though did not consider embedment [51]; Dwyer-Joyce described brittle particle breakup dependence on fracture toughness and particle size [52]; Hagan described a critical size below which brittle particles cannot be fragmented [53]; Axen, typical of studies into abrasive wear (rather than friction) observed increased wear rates above critical ratios of substrate and abrasive hardness [54]; Others reported increased wear rate when the hardness ratio is >0.8 [55].

Various established lubricant technologies exist. Differences in functional mechanism are sometimes identified, but few studies compare different lubricants and lubrication mechanisms.

[7,11,16,24,34]. There are no published studies that consider second-order effects that could provide explanation of the tribological differences observed.

1.4. Load Distribution

Fasteners are typically tensioned to a prescribed proportion of the minimum 0.2% proof stress in the bolt tension area [6,34,56,57]: The real yield point is higher for safety [58,59]. The target percentage of theoretical 0.2% proof stress depends on the system, duty cycle and industry [6,19,34,56,57,60-66]. In this study, 75% of the nominal 0.2% proof tension was used unless stated otherwise.

Stresses concentrate, particularly at thread roots [67,68], but also at the underhead face [69]. Several models describe the load supported by each thread pitch [11,63,70–76]: Kenny and Patterson reviewed earlier work [77]. Some are summarised in Table 1 with any relative or normalised values adjusted to percentages. In this study, Brutti's model with a washer [75] was used (Table 1): ISO geometry with a freely rotating washer best described the fastener sets. Washers can reduce the proportion of load carried by the first thread [75].

Table 1: Load Distributions on Threads under Axial Loading

% Axial Tension Carried by Each Thread										
Author		Fastenal [63]	Majzoobi et al[1] [72]	Seika [74]	Fukuoka et al [73]	Liu et al [71]	Dragoni[1] [70]	Brutti[1] (washer) [75]	Brutti[1] (no washer) [75]	Kenny and Patterson[2] [76]
Year		2005	2014	1974	1986	2017	1997	2017	2017	1985
Thread Number from Loaded Nut Face	1	35	35	18	29	29	31	33	35	40
	2	25	22	19	18	21	23	23	24	27
	3	18	21	17	15	17	18	16	16	16
	4	-	-	16	14	13	13	12	11	9
	5	-	-	15	12	11	9	9	8	5
	6	-	-	15	12	10	6	7	6	3

[1] ISO Metric Thread Form

[2] Sopwith's Model [78] applied to ISO Metric Thread Form

1.5. Contact Pressure and the Influence of Washers

Various authors have calculated contact pressure values >500MPa [19], up to 700MPa [79], and 1100MPa [80]. Tronci et al [35] estimated that many contact points were <150MPa but others were greater than the 750MPa needed to strip silver plating from the surface.

In this study, the approximate load bearing area of a thread pitch was calculated as the area of a frustrum cone. As described by Eccles [81], the effect of lead angle (4.2° on M12 fasteners) on area is negligible. For an idealised M12 fastener, A_{Thread} is approximately 46×10^{-6} m^2. Uniform radial contact pressure on the threads was assumed, which generates an insignificant 0.8% error, [8,82].

For an ideal fastener, contact pressure is significantly higher at the hole edge and decreases to the edge of the bearing area. Stephen [83] reported pressure ratios of between 1:3 to 1:5 from the outside to the inside of the bearing face. Nassar [8,84] calculated that an exponential decrease in contact pressure most closely reproduced experimentally derived values, attributing the peak pressure values to concentration at hole edges. Using the mean radius, as in many versions of the long form torsion-tension equation, would generate ~12% variance with the absolute experimental data in comparison to the more accurate exponential distribution [8]. In this study, the mean radius was used to derive friction coefficient values, but contact pressure calculations were made with reference to a 1:4 distribution from the outside to the inside radius of the bearing faces, i.e., mid-range in the work of Stephen [83].

Nominally flat washers tend to distribute the underhead and nut loads over a larger area at the component surface, reducing contact pressure [85–87]. The maximum contact stress at the joint interface is generally considerably lower than at the underhead and nut bearing faces [88,89,90].

The friction coefficient equations within ISO 16047 [6] and DIN 65946 [3] do not account for the influence of washers on the friction diameter D_f, (Equation 1). In this study, rotating washers were used for all test conditions, which is a deviation from ISO 16047 and DIN 65946: Rotating washers are common in a variety of industries but their influence is rarely reported in literature. The friction radius of the centre of the nut face was used, as in ISO 16047 [6] and DIN 65946 [3]. Calculations by the authors, not reported here, estimate that the variance caused by this assumption is ~2% for a 24mm diameter, freely rotating washer and, because it is applied throughout this study, does not affect the internal coherence of the data presented.

To account for torsional stresses generated overcoming friction, some studies and industry standards recommend a target *von Mises* stress rather than tensile stress [58,91-92], including the VDI 2230 method [1]. Eccles recommends 75% theoretical yield/proof tension if torsion cannot be compensated for, the approach taken in this study, 90% theoretical yield/proof tension if it can be [60].

1.6. Aims

This study contains detailed analysis of the varying tribological mechanisms observed between different lubricant technologies. Changes to fastener surfaces were observed, but also changes arising from transformation of the lubricant ingredients were considered. Multivariate statistical analysis was used to identify second-order tribological parameters that describe the variations in friction across different materials and lubrication mechanisms. These aspects of lubricated fastener friction have not previously been published.

More accurate torque-controlled tensioning can improve threaded fastener performance, increase joint reliability, and optimise the strength-to-weight ratio of assemblies.

2. Experimental Methods and Data Analyses

2.1. The Test Rig

A tension-torsion rig was used to determine fastener friction coefficients during assembly, shown schematically in Figure 1, where a hydraulic transducer measured axial tension and a torque transducer simultaneously measured total torque. Because total torque was measured, rather than separating bearing friction and thread friction torque, total friction coefficient was determined (Equation 1).

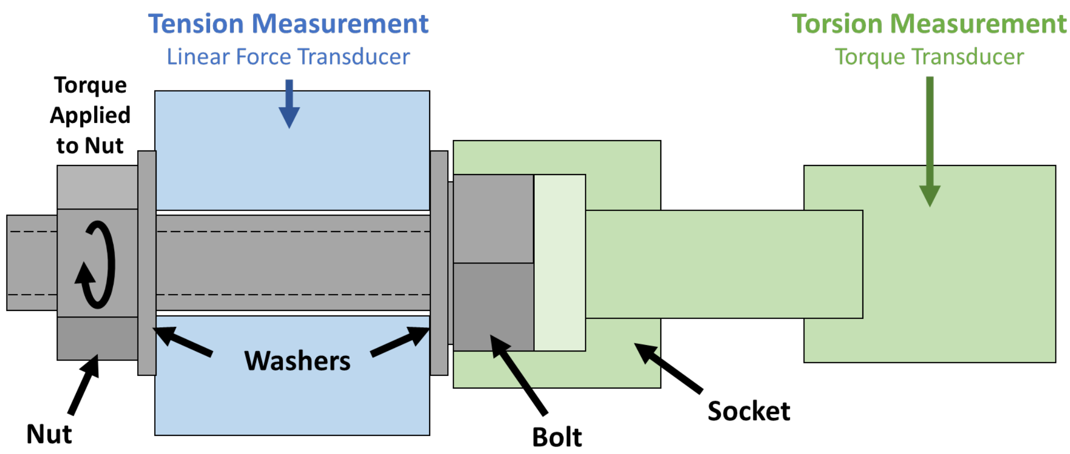

Figure 1: Schematic Diagram of Tension-Torsion Rig

2.1 The Assembly Process

Nuts were run down by hand until snug, apart from prevailing torque fasteners which were run down and snugged using a wrench. Fastener tension was achieved by applying a torque via a hand wrench. Unless otherwise stated, fasteners were tensioned to 75% of their theoretical 0.2% proof tension. When dissimilar materials were used (e.g. Ti6Al4V bolts with aluminium 7075 nuts/washers) the theoretical yield tension was 75% of the lower yield stress material.

Unless otherwise stated, 5 tests were performed for each test condition. Repeatability was calculated as the 95% confidence interval of these values. As discussed previously, repeatability in fastener assembly friction is not merely for statistical significance but is also a performance indicator, i.e. smaller 95% confidence intervals indicate a more repeatable assembly fastener-to-fastener.

2.2. Fasteners

The fasteners used in this study were ISO M12 coarse thread hexagon head set screws with full nuts **[93]** and Form A washers **[94]**. A range of typical materials was chosen, as described in Table 2. Some high temperature fasteners were used where additional features were present, e.g., threaded inserts with prevailing torque features, silver and/or nickel plated nuts/inserts, and crimped nuts to provide prevailing torque. Where present, prevailing torques were measured during run-down of the nut and compensated using the DIN 65946 method when calculating the friction coefficient **[3]**. Tronci *et.al.,* **[35]**, using a low viscosity turbine oil as a lubricant showed that silver coatings can wear through or delaminate during assembly, particularly on prevailing torque features where sliding distances under increased contact pressure are higher. However, in this study, where silver or nickel coatings were used, coating integrity was assumed when calculating interface properties. Fasteners were degreased with a residue-free hydrocarbon solvent before use.

Table 2: Fastener Material Combinations Tested

Fastener Material Combinations		
Bolt	**Nut (if different to bolt)**	**Washer (if different to bolt)**
8.8 carbon steel plain finish		
8.8 carbon steel, hot dip galvanised		
8.8 carbon steel bright zinc plated (BZP)		
10.9 carbon steel plain finish		
12.9 carbon steel plain finish		
A2-70 stainless steel		
A4-70 stainless steel		
A2-70 stainless steel	A2-70 stainless steel. Nickel plated nut, Silver plated prevailing torque insert	A2-70 stainless steel
A2-70 stainless steel	A2-70 stainless steel. Nickel plated nut, Uncoated prevailing torque insert	A2-70 stainless steel
A2-70 stainless steel	8.8 carbon steel plain finish	8.8 carbon steel plain finish
8.8 carbon steel plain finish	A2-70 stainless steel	A2-70 stainless steel
8.8 carbon steel plain finish	8.8 carbon steel hot dip galvanised	8.8 carbon steel hot dip galvanised
8.8 carbon steel plain finish	8.8 carbon steel BZP	8.8 carbon steel BZP
6061 aluminium		
7075 aluminium (clear anodised)		
Titanium Grade 5 (Ti6Al4V)	7075 aluminium (clear anodised)	7075 aluminium (clear anodised)
Inconel 718 plain finish	Inconel 718 plain finish	8.8 carbon steel plain finish
Inconel 718 plain finish	Inconel 718 silver coated	8.8 carbon steel plain finish
Inconel 718 plain finish	Inconel 718 silver coated crimped nut	8.8 carbon steel plain finish

2.3. Lubricants

A variety of lubricants were used: some specific anti-seize and assembly lubricants together with others not specifically designed for threaded fasteners which were used for comparison, Table 3. Because these were commercial formulations, limited composition information is given. Broadly, with reference to other work, copper would be expected to behave as a 'ductile' solid whilst non-metal solid lubricants would be expected to be 'brittle' and 'friable' [49,52]. The friction mechanisms of MoS_2 are quite different, [95–97].

Table 3: Lubricants Tested

Technology	Ref	Type
Metal Powder	1	Copper-containing grease with graphite and MoS_2
	2	Copper-containing grease with aluminium, graphite and MoS_2
	3	Copper-containing grease: Low formulation
	4	Nickel-containing paste
Non-Metal Paste	5	Calcium fluoride and mica paste
	6	Proprietary Formulation 1
	7	Proprietary Formulation 2
MoS_2 Lubricants	8	MoS_2 in calcium-thickened grease
	9	MoS_2 in clay-thickened grease
	10	Dry Film Coating 1: MoS_2 & aldehyde resin - Aerosolised
	11	Dry Film Coating 2: MoS_2 & phenolic resin

	12	Dry Film Coating 3: MoS_2 & inorganic resin - Aerosolised

2.4. Application of Lubricants, Greases and Pastes

Lubricants were applied using best practice for their respective types. Thin layers of pastes and greases were applied using a brush [34] to the threads, nut face, bolt underhead and both surfaces of the washers [34,43,92]. Dry film aerosol coatings were applied using light passes across the surface from around 30cm using a thoroughly-agitated aerosol and allowed to fully cure before assembly.

2.5. Example Results Indicating Second-Order Influences

Two examples are shown below that illustrate that, whilst there are obvious changes in frictional properties between different materials, surfaces and lubrication, there also appear to be second-order effects that affect the frictional properties.

Surface Finish: When unlubricated, 8.8 carbon steel fasteners with a plain finish gave relatively low friction coefficients with relatively low variance, Figure 2Figure 2**Error! Reference source not found.**. Bright Zinc Plated (BZP) finishes had higher friction coefficients and larger friction variance due to their tendency for surface galling and coating delamination. Hot Dip Galvanised fasteners had almost impractically high friction coefficients and variance due to the high roughness and coating thickness of the galvanised coating - nuts in these fastener sets were tapped with a larger root diameter to accommodate the coating (see, e.g., SAE J1648 [98]).

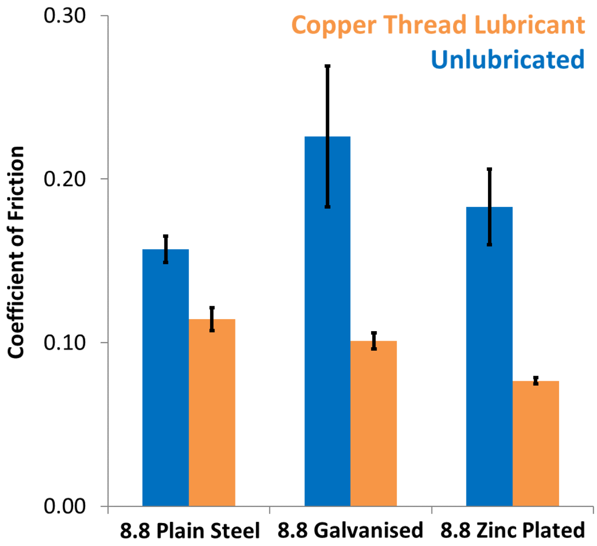

Figure 2: Variation in Friction Coefficient between Lubricated (Orange) and Unlubricated (Blue) 8.8 Carbon Steel Fasteners with Different Surface Finishes

Lubricating each fastener type with a general-purpose, copper-containing, anti-seize lubricant, reduced their friction coefficients, and variance was similarly reduced, Figure 2. As expected, the reduction effect was largest for the 'hot dip' galvanised fasteners. The low friction coefficient for BZP fasteners was probably due to the low initial roughness of the BZP coating, being 0.33µm R_a on the nut face versus approx. 0.79µm R_a for the plain finish. It is proposed that a more complete lubricant film may have formed for this combination of low surface roughness and anti-seize lubricant, which reduced direct contact between the fastener surfaces and generated lower friction.

The 'friction coefficient' is a system response, not a lubricant property. As would be expected for boundary lubrication, interactions between the lubricant and the contacting surfaces have a substantial influence on friction. A 'friction coefficient value for a lubricant' should not be assumed to apply for all fastener materials that it is applied to [16].

Fastener Material: Figure 3**Error! Reference source not found.** shows tension-torsion traces for different grades of plain finish carbon steel fasteners, 8.8, 10.9 and 12.9, [58], lubricated with copper-containing anti-seize lubricant. The tension-torsion traces for 10.9 and 12.9 fasteners were not significantly different but the friction coefficient for 8.8 fasteners was significantly higher over the same range of tension. The composition of these steels are relatively similar, ruling out a chemical effect [58]. The higher friction for the 8.8 fasteners was not due to yield because the differences are manifest at low tensions, well below the yield tension value even when accounting for stress concentration [67,69]. Therefore, it was hypothesised that differences in other material properties and interactions with the lubricant accounted for these differences in friction coefficient.

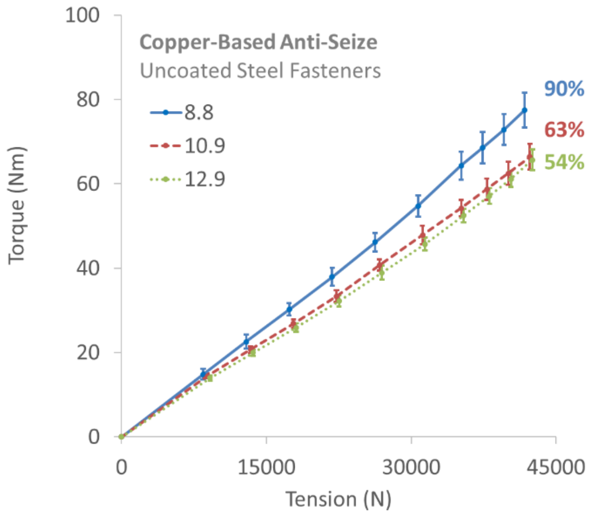

Figure 3: Tension-Torsion Relationships for Different Grades of Plain Finish Carbon Steel Fasteners Lubricated with a Copper-Containing Grease (The Percentage Values Indicate the Highest Proportion of Yield Tension Achieved).

3. Functional Surface Changes

Different anti-seize lubricant technologies have different friction mechanisms. To highlight these different responses, a series of fasteners were assembled to 75% 0.2% proof tension, dismantled, cleaned, degreased and then visually analysed using a confocal microscope, comparing to a fresh fastener surface. To compare the effect of surface treatment, 8.8 plain finish, 8.8 BZP and A2-70 fasteners were used. Four different lubricants were used, covering a range of lubrication mechanisms:

- a copper-containing grease, Reference 1 in Table 3,
- two metal-free pastes, References 5 and 6 in Table 3,
- a dry film, resin-bonded MoS_2 coating, Reference 10 in Table 3,

Assembly was conducted on the tension-torsion rig to confirm that the coefficient of friction was representative of typical conditions, which was the case for all tests. Images were taken of two key interfaces:

- the loaded nut face at 75% proof stress: $P_{average}$ ~ 350MPa, P_{max} ~ 710MPa for 8.8 steel, and $P_{average}$ ~ 241MPa, P_{max} ~ 485MPa for A2-70 stainless steel;
- the load-bearing surface of the first engaged thread, i.e., the most highly loaded surface at 75% proof stress: P_{Thread} ~ 255MPa for 8.8 steel, and P_{Thread} ~ 170MPa for A2-70 stainless steel.

Average Roughness (R_a), Skewness (R_{sk}) and Maximum Peak (R_p) were measured at a variety of points across the surfaces. Error bars describe 95% confidence intervals for these measurements. The frictional mechanisms and surface parameters were comparable for both surfaces. To avoid duplication, images and surface profilometry of one surface from each interface are shown and reported.

3.1. Nut Face Images for Plain Finish 8.8 Steel Fasteners:

The nut face images and corresponding R_a, R_{sk}, R_p and friction coefficient values for plain finish 8.8 steel fasteners are shown in Figures 4a-h. Figure 4d shows the 'fresh' nut face with concentric grooves from the facing operation. Non-metal pastes generated the highest coefficients of friction, Figure 4a, the higher of which shows a layer of particles across the interface and that the top surfaces of groove features have been worn down, Figure 4e. The particle layer survived the cleaning process, suggesting that particles were embedded or very well adhered. R_a was significantly lower, Figure 4a, but R_{sk} did not significantly change, Figure 4b. R_p was significantly increased compared to the fresh surface, Figure 4c, suggesting that abrasion grooves and embedded particles have increased the height of some parts of the surface locally. It appeared that the machining grooves were in the order of waviness, so the changes in the general profile of these was not reflected in e.g. R_p.

The second non-metal paste showed a similar interface layer but this was removed from some of the grooves, Figure 4f, suggesting either poorer adhesion or that particles were not as well embedded in the surface. R_a, R_{sk} and R_p varied widely, Figures 4a-c, which reflected that the interface layer was not uniform across the surface, though a complete interface layer was probably present during assembly. These differences in embedment correlate well with the surface morphologies observed by Axen et al [54].

The MoS_2 coating generated a significantly lower friction coefficient; some particle/debris occurred at the surface but groove peaks were not as extensively removed,

Figure 4g and R_a and R_{sk} did not vary significantly from that of the original surface. R_p increased significantly.

Use of a copper-containing grease generated a low friction coefficient with very little evidence of a debris layer on the surface, Figure 4h, though the tops of the grooves were significantly flattened. As such, R_a and R_{sk} were slightly reduced, reflecting a more conformal surface. R_p was slightly increased.

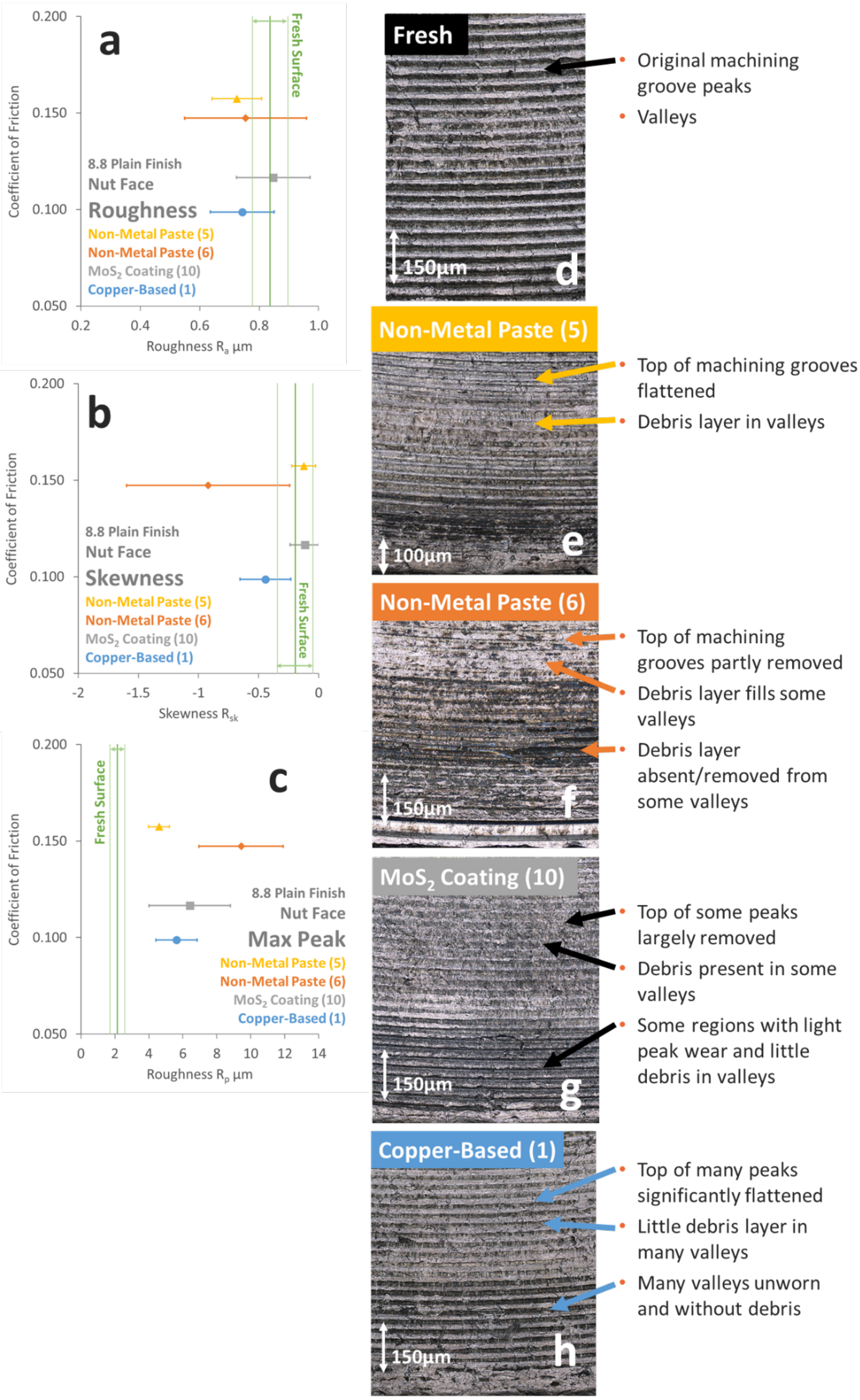

Figure 4: Surface Changes for 8.8 Plain Finish Nut Faces Following Assembly to 75% Froof Stress Using Metallic (Copper and MoS_2) and Non-Metallic Pastes, with a Fresh Nut Face for Comparison

3.2. Thread Phase Images for Plain Surface 8.8 Fasteners:

Thread surfaces for a plain finish 8.8 were initially black oxide (Fe_3O_4), Figure 5d, with wide ranges of R_p and $R_{sk,}$ Figure 5b-c. Following assembly with the higher friction non-metal paste, the oxide layer was completely removed with fine grooves and embedded particles present across the surface, Figure 5e, these features increased R_a from the fresh surface (Figure 5a). R_{sk} was slightly positive, indicating that the formation of fine grooves generated new peaks as well, to R_p was lower than for the fresh surface.

The lower friction non-metal paste generated a fuller debris layer in the worn parts of the thread, Figure 5f, however, some oxide layer remained near the thread tip and root. Though surface features were significantly changed, R_a was comparable to the original surface (Figure 5a): R_{sk} was comparable to the value when the other non-metal paste was used, Figure 5b.

The MoS_2 coating generated a significantly higher R_a, with a greater variation than for the original surface, Figure 5a. Deeper scores were present, perhaps suggesting some adhesive wear mechanisms, Figure 5g. These scores seem to generate new peaks, so R_p is very high, Figure 5c. Areas between the scored areas were mildly abraded and local variations in the wear mechanism caused significant local variation in R_{sk}, Figure 5b.

When copper-containing grease was used, the surface changes were less extensive than when other lubricants were used, Figure 5h whilst some small grooves and mild abrasion were present. A significant quantity of oxide layer remained with some copper particles remaining adhered to the surface. R_a and R_p were significantly lower than the original surface, Figure 5a,c, and R_{sk} was lower than when other lubricants were used, Figure 5b, indicating a more conformal surface than the original.

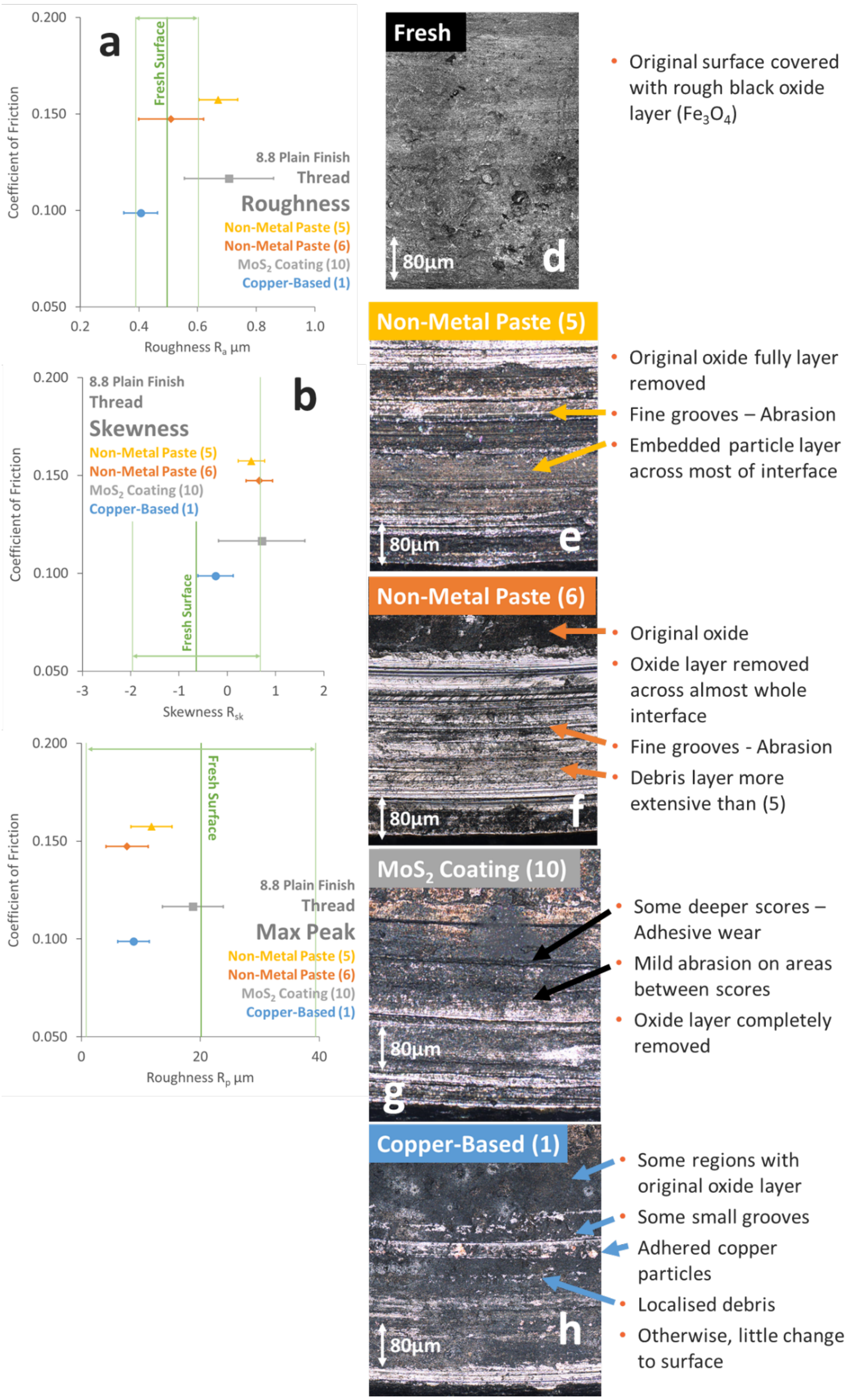

Figure 5: Surface Changes For 8.8 Plain Finish 'First Engaged Thread' Following Assembly to 75% Proof Stress

3.3. BZP 8.8 Nut Face and Thread Surface Images:

The 8.8 BZP surface coatings were much more durable than black oxide with varying degrees of the coating removed. When lubricated with non-metal pastes, large regions of the BZP coating were removed from the nut faces, Figures 6e and f, exposing substrate steel. On these exposed areas, one non-metal paste, (5), formed an adhered/embedded particulate layer (6e): The other non-metal paste, (6), fine grooves on the substrate were visible through the particulate layer, suggesting a less well adhered adhered interface layer, (Figure 6f). For threads lubricated with non-metal pastes, the BZP coating was completely removed and fine grooves were present, Figure 7e and f.

When lubricated with an MoS_2 coating, a significant proportion of BZP coating remained on the nut face, Figure 6e, but was completely removed from the thread surface, Figure 7e, where deeper grooves suggested some adhesive wear, increasing R_a, Figure 7a, this being the probable root cause for the high friction coefficient.

When lubricated with a copper-containing grease, very little BZP coating was removed from either surface, Figure 6h and 7h, except for some areas where delamination had initiated on the thread. Unlike the result for plain finish 8.8 carbon steel fasteners, no copper remained adhered to the surface. It appears that plastic deformation of copper between the fastener surfaces largely separated them and provided effective protection from boundary friction and wear. As a result, the friction coefficient was exceptionally low, Figures 6a and 7a.

As indicated in Figures 6d and 7d, fresh BZP coating surfaces were fairly irregular and some small zinc beads were present (note particularly the high and variable skewness of thread surfaces in Figures 7a and b, and the high initial values of R_p in Figures 6c and 7c) and with features defined by the coating and not the underlying substrate surface. Therefore, as lubrication was influenced by coating removal, little can be reasonably interpreted from changes in roughness and skewness from the original surface.

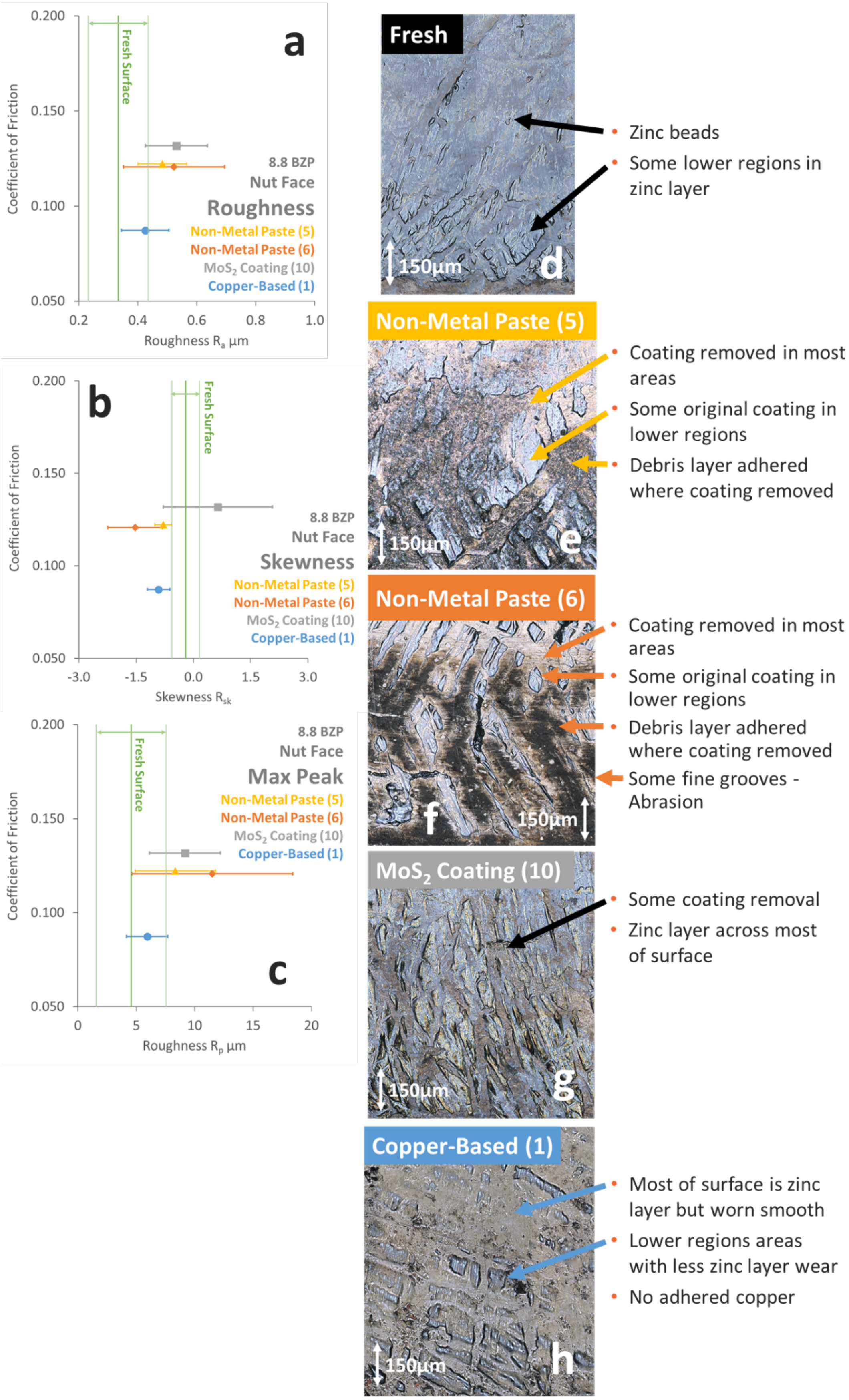

Figure 6: Surface Changes For 8.8 BZP Nut Face Following Assembly to 75% Proof Stress

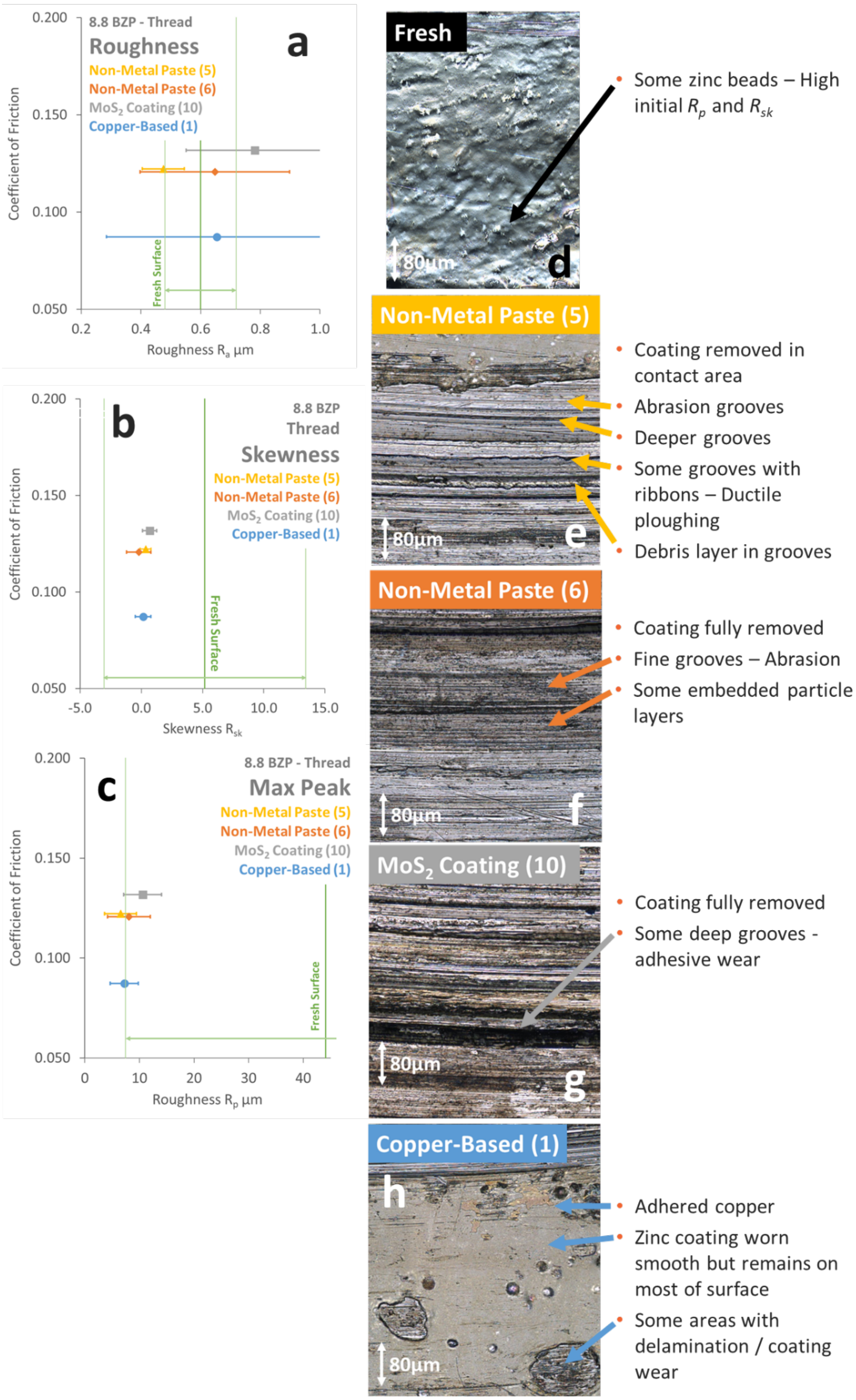

Figure 7: Surface Changes For 8.8 BZP 'First Engaged Thread' Following Assembly to 75% Proof Stress

3.4. A2-70 Nut Face and Thread Surface Images:

Some subtly different behaviour was observed for A2-70 fasteners on both thread and nut face surfaces, Figures 8 and 9. One non-metal paste (6) formed an adhered/embedded interface layer which generated a high friction coefficient, Figure 8f. The other non-metal paste (5) generated fine grooves on both nut face and thread, with an adhered/embedded layer of particles and residue on only part of the thread surface, implying a less well adhered interface layer, Figures 8e and 9e, which generated a significantly lower friction coefficient, Figure 8a.

When lubricated with an MoS_2 coating, A2-70 surface roughness significantly increased and the skewness significantly decreased, Figures 8a and b, Figure 9a and b, though little visible change was apparent, Figures 8g and 9g. The friction coefficient was relatively low, Figure 8a, indicating that the lubricating layer was effective at reducing wear and adhesion of the surfaces. The high R_p on the thread suggests that some galling has occurred, causing ploughing of the surfaces, Figure 9c.

When lubricated with a copper-containing grease, the thread surface appeared to be relatively unchanged, Figure 9h, though the reduced roughness and skewness suggested that some of the raised features from the manufacturing process were worn down and conformity of the surfaces had increased, Figure 9a and c. However, abrasion and some deeper galling/scuffing features were observed on the nut face, Figure 8h, which were the probable root cause of the higher friction coefficient, Figure 8a, and the higher R_p, Figure 8c.

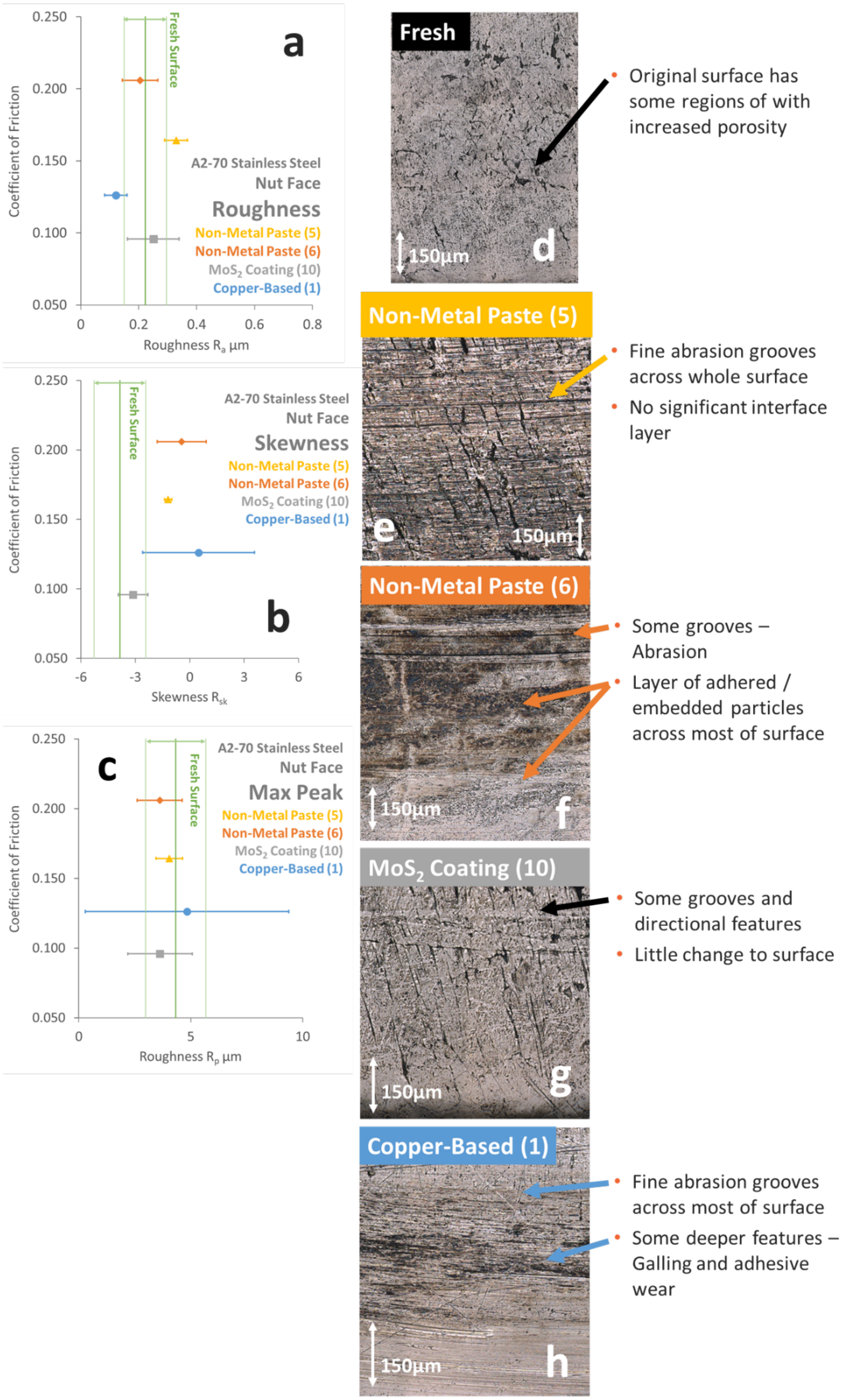

Figure 8: Surface Changes For A2-70 Stainless Steel Nut Face Following Assembly to 75% Proof Stress

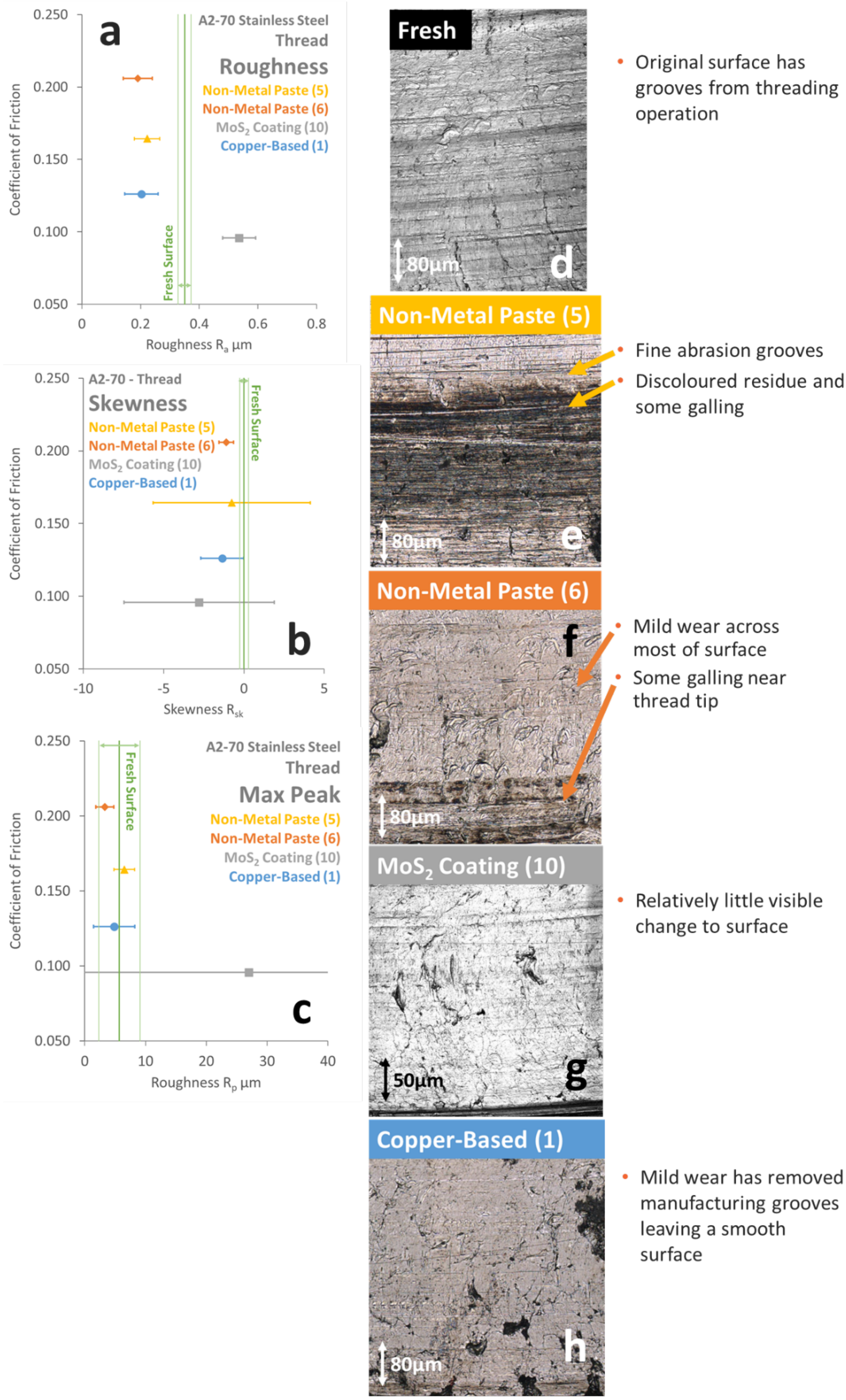

Figure 9: Surface Changes for A2-70 Stainless Steel First Engaged Thread Following Assembly to 75% Proof Stress

3.5. **Lubricant Technologies:**

Across the different fastener materials, different lubricant technologies exhibited different characteristic mechanisms:

- **Non-metal Pastes:** An interface layer of particles formed when particles became embedded into, or adhered to surfaces, which resisted adhesive wear and galling. Friction was controlled by shear of this layer and/or the interface of the particles and surface. Differences in the relative adhesion/embedding of the particles in this interface layer appeared to influence the global friction coefficient.
- **MoS_2 Coatings:** MoS_2 appeared to generate a thin lubricating layer across the surface which appeared to prevent (or resist the propagation of) adhesive wear of the fastener surfaces. Burnishing of MoS_2 onto the surface under shear could be the formation mechanism for this layer. It is hypothesised that the relatively low shear strength of MoS_2 [95,99], in comparison to the higher energy dissipation in the embedment/adhesion of solid particles, was the reason for the lower friction coefficient of the MoS_2 coating than non-metal pastes on the same fastener material.
- **Copper-Containing Grease:** Copper particles appeared to be plastically deformed at the interface rather than adhering/embedding. Most copper particles were removed in the cleaning process, indicating weak adherence. It is hypothesised that copper particles acted to separate the fastener surfaces and prevent their adhesion. The low shear strength of copper relative to the fastener materials and the interface layer of copper that reduced contact between the surfaces appeared to be the cause of the lower friction coefficients.

4. Effect of Multiple Assemblies on Fastener Friction

To further explore the characteristic friction mechanisms of different lubricant technologies, a series of experiments was conducted to consider the evolution of these mechanisms and system responses over multiple assemblies. A first batch of five fastener sets were assembled in accordance with standard practice; then in accordance with typical reuse practice [30], were dis-assembled, the individual components cleaned and degreased before being relubricated and re-assembled. Five assembly/disassembly procedures were sequentially conducted on each fastener set. A second batch of five fastener sets were assembled and disassembled five times each. However, for this batch, the fasteners were lubricated during the first assembly only. Thus, the original lubricant film was used throughout the testing of this batch.

4.1. Multiple Assemblies of Plain Finish 8.8 Carbon Steel Fasteners and a Copper-Containing Grease:

The effect of multiple assemblies on fastener friction for a copper-containing grease on plain finish 8.8 carbon steel fasteners is shown in Figure 10. For both sets of fasteners, the friction coefficient was ~10% lower on second assembly relative to the first, then decreasing further through to the fifth assembly. The reduction in friction coefficient was slightly greater when the lubricant was only applied once but with greater variance. As shown in Figures 4 and 5, a copper-containing grease caused surfaces to decrease in roughness, increase in conformity and 'run-in'. Presumably, for second and subsequent assemblies, there was more separation of the fastener surfaces. As shown in Figures 4 and 5 previously, for the same assemblies, copper particles appeared to plastically deform under shear and load at the interface. As this consumes energy when the lubricant was removed and reapplied, the process was repeated with fresh copper particles. Therefore, when the lubricant film was applied once and was present for all 5 assemblies, the friction coefficient was lower, i.e. the interface layer was still sufficient to control friction and less energy was consumed deforming the copper particles.

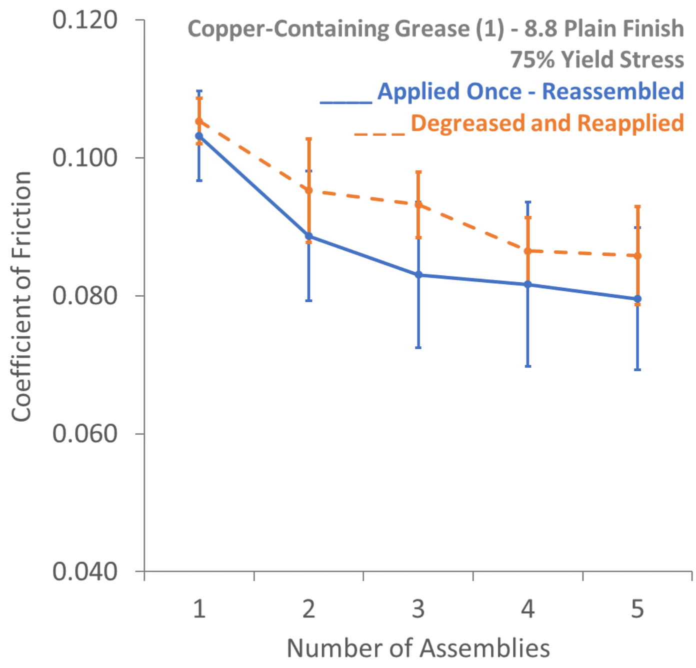

Figure 10: Variation of Friction Coefficients for Repeat Assemblies of Plain Finish 8.8 Carbon Steel Fasteners Lubricated with a Copper-Containing Grease

4.2. Multiple Assemblies of A2-70 Stainless Fasteners Using a Non-Metal Paste:

The effects of multiple assemblies on A2-70 stainless steel fastener friction using a non-metal paste is shown in Figure 11. There was lower reduction in the coefficient of friction from first to second assemblies using the non-metal paste, ~7%, compared to the copper-containing grease. It is interesting that there was a negligible difference in friction coefficient on second assembly between the once-applied and re-applied lubrication conditions. A slight reduction in coefficient of friction was seen for the once-applied lubricant for subsequent, third, etc., assemblies onwards. Figures 8 and 9 show that the non-metal paste formed an interface layer of embedded and loosely compacted particles. The small differences in friction coefficient show that formation and function of the interface layer consumed a similar amount of energy whether the lubricant was fresh or used, particularly when compared to the copper-containing grease.

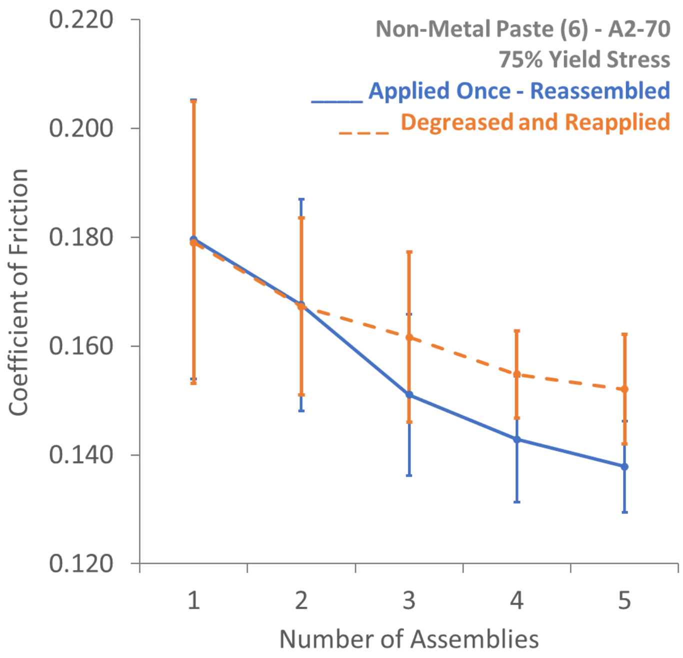

Figure 11: Friction Coefficient Variation for Repeat Assemblies of A2-70 Stainless Steel Fasteners Lubricated with a Non-Metal Paste

4.3. Multiple Assemblies of Plain Finish 8.8 Carbon Steel Fasteners Using an MoS_2 Paste:

The effect of multiple assemblies on fastener friction coefficients for an MoS_2 paste on plain finish 8.8 carbon steel fasteners is shown Figure 12. When the lubricant was reapplied, the friction coefficient was ~9% lower on second assembly but ~20% lower when the lubricant was reused. As indicated in Figures 4 and 5 previously and as described in previous studies [95,97,99–101], MoS_2 films are formed on a surface with deformed lamellae aligned with the axis of shear, usually in the early stages. The aligned MoS_2 film produced lower friction under comparable conditions [97,99,102] and could allow surfaces to be run-in [95,103], Figures 4 and 5.

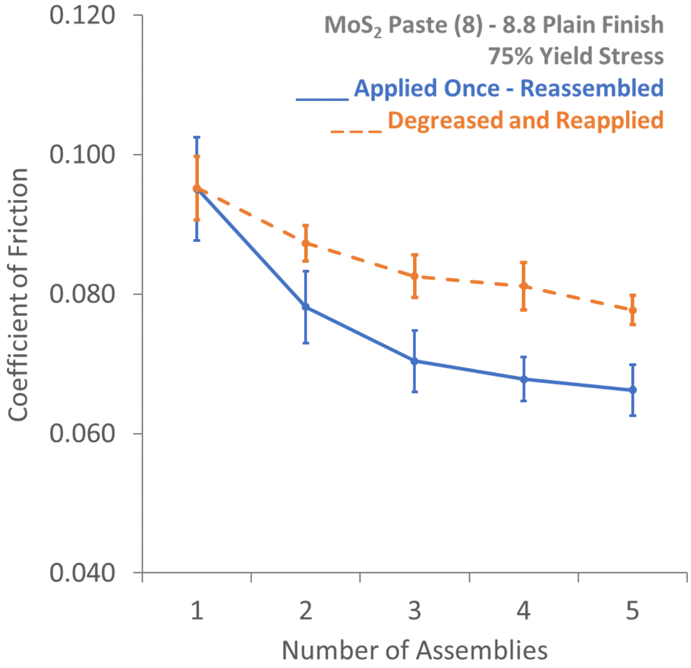

Figure 12: Variation of Friction Coefficient for Repeat Assemblies of Plain Finish 8.8 Carbon Steel Fasteners Lubricated with an MoS_2 Paste

4.4. Comparison Between Lubricants:

When a lubricated fastener is assembled, some energy is used to deform/wear the fastener surface and further energy was used to deform/transform the lubricant. As described by Farr [104] for MoS_2 – '*it is unrealistic to distinguish between the running in of the lubricant film from that of the lubricated surfaces in an arbitrary way, for the two processes occur simultaneously'*. As such, if fasteners are reused, the original torque settings may no longer be valid. Recalculation or validation is needed to prevent potential overtightening if the friction coefficient is lower,

The different lubrication mechanisms for copper-containing greases, non-metal pastes and MoS_2 greases consumed different quantities of energy during the formation and function of interface layers. These different lubrication mechanisms generated different responses on second and subsequent assemblies: copper particles plastically deformed and MoS_2 formed a basally-aligned film, both of which required less energy to shear after formation. Conversely, non-metal paste films were formed by embedding and shear of an interface layer of particles, which required comparable energy to continue shearing as to form initially, Thus, if fasteners are to be reused, cleaning and reapplying the lubricant is recommended, particularly for lubricants that are more sensitive to re-application, i.e., copper and MoS_2. Where provision for re-use exists within fastening systems and standards, the variation in friction must be accounted for.

5. Multivariate Analysis of Results

Having identified and confirmed the different characteristic friction mechanisms for different lubricant technologies, the different system parameters and responses that affect each of these were investigated using multivariate statistical analysis. A linear regression analysis was performed on data from a range of fastener material, lubricant and surface finishes combinations, as in Table 2, and a range of lubricants, as in Table 3. Not all possible lubricant/fastener combinations were tested, but there were sufficient for statistically meaningful results. Where data points were from fasteners with prevailing torque features, the global friction coefficient was separated from the prevailing torque using the DIN 65946 method [3].

5.1. Initial Data Screening Steps:

Separate regressions were performed on data from metal-containing lubricants, MoS_2-based lubricants and non-metal pastes.

An initial screening step removed any variables with very low influence (Pearson Correlation Coefficient <0.35) and, to improve regression accuracy, groups of similar variables (e.g., surface energy parameters) were reduced to the variable with the highest Pearson Correlation Coefficient for the friction parameter under consideration. Linear regression was performed on the remaining variables, eliminating the least influential variables until the analysis of variance (ANOVA) p-value significance was ≤ 0.05 and t-values for each coefficient were >2. The key variables in the regression analysis are shown in Table 4 and, where required, these are defined below. For reasons of commercial sensitivity, the ranges of these parameters are reported rather than specific values. The sources of this data are also identified in Table 4 where possible.

Table 4: Variables Used in Multivariate Analysis, with reference to [105–111]

Variable	Definition	Variations			Data Source
Fastener Material					
-	Hardness (kg/mm²)	H_{Bolt} Bolt 38 - 355	H_{Nut} Nut 38 - 348	H_{Washer} Washer 38 - 230	Manufacturer Data / Specification
-	Surface Energy (mJ/m²)	γ_{Bolt} Bolt 39 - 1,700	γ_{Nut} Nut 39 - 1,700	γ_{Washer} Washer 39 - 1,700	[103–111]
Lubricant					
$\%_{Fluid}$ 44 - 66	% Fluid	-	-	-	Proprietary
$\%_{Solids}$ 35 - 100	% Lubricating Solids(1)	-	-	-	Proprietary
-	Hardness (kg/mm²)	H_{High} Highest in Formulation 25 - 550	H_C Composite Average(2) 25 - 288	H_{Low} Lowest in Formulation 3 - 80	Proprietary
-	Cube-Weighted Composite Hardness(3) (kg/mm²)	-	H_{C^3} Composite Average 25 -275	-	Proprietary
$\gamma_{Lubricant}$ 35 - 535	Surface Energy of Total Formulation(4) (mJ/m²)	-	-	-	Proprietary
-	Surface Energy of Lubricating Solids (mJ/m²)	$\gamma_{S\text{-}High}$ Highest in Formulation 24 - 1,770	γ_{Solids} Composite Average(5) 23 - 516	$\gamma_{S\text{-}Low}$ Lowest in Formulation 24 - 1,250	Proprietary
$\gamma_{Solids\%}$ 47 - 1,473	Specific Surface Energy of Lubricating Solids(6) (mJ/m²)	-	-	-	Proprietary
Tribological					
-	Rabinowicz Static Friction Parameter(7)	$(\gamma_{Bolt}+\gamma_{Solids})/H_{Bolt}$ vs Bolt Material 0.51 – 69.42	$(\gamma_{Bolt}+\gamma_{Solids})/H_{Nut}$ vs Nut Material 0.51 – 40.62	$(\gamma_{Bolt}+\gamma_{Solids})/H_{Washer}$ vs Washer Material 0.51 – 40.62	Calculated from Above
-	Log Rabinowicz Static Friction Parameter(8)	$\log(\gamma_{Bolt}+\gamma_{Solids})/H_{Bolt}$ vs Bolt Material -0.29 – 1.84	$\log(\gamma_{Bolt}+\gamma_{Solids})/H_{Nut}$ vs Nut Material -0.29 – 1.61	$\log(\gamma_{Bolt}+\gamma_{Solids})/H_{Washer}$ vs Washer Material -0.29 – 1.61	Calculated from Above
-	Contact Pressure (MPa)	P_{Head} Nut Face / Bolt Head 125 - 465	P_{Washer} Washer Average 40 - 147	P_{Thread} 1st Engaged Thread 95 - 352	Calculated in Section 2.2
Interactions					
-	Surface of Energy of Lubricating Solids vs Fastener	$\gamma_{S\text{-}High}/\gamma_{Bolt}$ Highest in Formulation 0.014 – 45.38	$\gamma_{Solids}/\gamma_{Bolt}$ Composite Average 0.027 – 37.77	$\gamma_{S\text{-}Low}/\gamma_{Bolt}$ Lowest in Formulation 0.014 – 32.05	Calculated from Above
-	Specific Surface of Energy of Lubricating Solids vs Fastener	$\gamma_{Solids\%}/\gamma_{Bolt}$ vs Bolt Material 0.014 – 13.22	$\gamma_{Solids\%}/\gamma_{Nut}$ vs Nut Material 0.014 – 13.22	$\gamma_{Solids\%}/\gamma_{Washer}$ vs Washer Material 0.014 – 13.22	Calculated from Above
-	Hardness of Lubricating Solids vs Fastener	$H_{Solids\%}/H_{Bolt}$ vs Bolt Material 0.077 – 7.57	$H_{Solids\%}/H_{Nut}$ vs Nut Material 0.077 – 7.57	$H_{Solids\%}/H_{Washer}$ vs Washer Material 0.13 – 7.57	Calculated from Above
-	Composite Hardness of Lubricating Solids vs Contact Pressure(9)	$H_{Solids\%}/P_{Head}$ Nut Face / Bolt Head 0.65 – 22.56	$H_{Solids\%}/P_{Washer}$ Washer Average 2.08 – 70.51	$H_{Solids\%}/P_{Thread}$ 1st Engaged Thread 0.86 – 29.69	Calculated from Above
-	Cube-Weighted Composite Hardness of Lubricating Solids vs Contact Pressure(9)	H_{C^3}/P_{Head} Nut Face / Bolt Head 0.65 – 34.26	H_{C^3}/P_{Washer} Washer Average 2.08 – 107.06	H_{C^3}/P_{Thread} 1st Engaged Thread 0.86 – 45.08	Calculated from Above

(1) Ignoring non-functional solids e.g. thickeners; (2) See Equation 2; (3) See Equation 3; (4) See Equation 4; (5) See Equation 5; (6) See Equation 6; (7) Described by Rabinowicz [105] to correlate with static friction of solids. Note that Rabinowicz used the hardness of the softest material. Here, the hardness of the fastener material was used;(8) Rabinowicz [105] used a logarithmic axis for the static friction parameter to obtain a linear fit of data vs static friction coefficient. Therefore, the logarithm was taken in case this improved correlation in linear regression. (9) Note that for these parameters, the indentation hardness was converted to MPa from the standard units of kg/mm² so that the ratio would be dimensionless. P_{Head} and P_{Washer} were average values.

5.2. Results from Multivariate Analyses

Because boundary friction of the interface lubricant films controlled overall friction behaviour of the fasteners, a number of parameters were defined to describe this. Considering that friction was likely to be influenced by the lubricating solids but that these were typically blends of multiple materials, a mass-weighted average was used, termed the composite hardness (H_C):

$$\boldsymbol{H_C = \sum m_1 H_1 + m_2 H_2 \dots m_i H_i} \qquad \textbf{(2)}$$

where m is the mass concentration of a lubricating solid and H_i is the indentation hardness of the material in kg/mm^2.

Considering the possibility that harder lubricating solids could have a disproportionate effect on friction, a cube-weighted composite hardness H_{C^3} was described to exaggerate the effect of harder materials:

$$\boldsymbol{H_{C^3} = \sqrt[3]{\sum m_1 {H_1}^3 + m_2 {H_2}^3 \dots m_i {H_i}^3}} \qquad \textbf{(3)}$$

This parameter does not really have quantitative meaning as a material property, but was acceptable as a semi-quantitative comparison. Similar to the composite hardness, a mass-weighted composite surface energy ($\gamma_{Lubricant}$) for the lubricant formulations was calculated as from the surface energy of each component (γ_i):

$$\boldsymbol{\gamma_{Lubricant} = \sum m_1 \gamma_1 + m_2 \gamma_2 \dots m_i \gamma_i} \qquad \textbf{(4)}$$

However, assuming that in slow speed boundary lubrication, the lubricating solids could have a greater influence on friction than a liquid phase, a composite surface energy (γ_{Solids}) was calculated using only the lubricating solids in the formulations:

$$\boldsymbol{\gamma_{Solids} = \sum m_{s1} \gamma_1 + m_{s2} \gamma_2 \dots m_{si} \gamma_i} \qquad \textbf{(5)}$$

where m_{si} is the mass-weighted proportion of the lubricating solids in the formulation only.

Considering the possibility that the coverage of the interface might have been a function of the proportion of lubricating solids in the formulation, a further composite surface energy ($\gamma_{Solids\%}$) of the lubricating solids was calculated but as a mass weighted proportion of the total formulation, i.e., assuming that the surface energy of the liquid phase was zero but still occupied some of the interface:

$$\boldsymbol{\gamma_{Solids\%} = \sum m_1 \gamma_1 + m_2 \gamma_2 \dots m_i \gamma_i} \qquad \textbf{(6)}$$

For unlubricated fasteners, no statistically significant models could be described by linear regression. It was not possible to meet the criteria for ANOVA significance nor coefficient *t*-values.

For metal-containing greases (copper, nickel etc.), the following regression models were described:

$$\boldsymbol{\mu_{metal} = 7.4 - 0.00023\,\gamma_{Solids} + 0.0011\,\gamma_{S-Low}/\gamma_{Bolt}} \tag{7}$$

$R^2 = 0.34$

$$\boldsymbol{95\%\ CI\ \mu_{metal} = 0.015 - 0.000023\,\gamma_{Solids} + 0.00033\,\gamma_{S-Low}/\gamma_{Bolt}} \tag{8}$$

$R^2 = 0.47$

Although friction coefficients and their variation were modelled by the same parameters, low R^2 values indicated that the variation in the data was poorly described by the variables considered here. Perhaps other, related but different, parameters to those described here were more influential for metal-containing lubricants. Further work is needed to explain lubrication mechanisms for these materials.

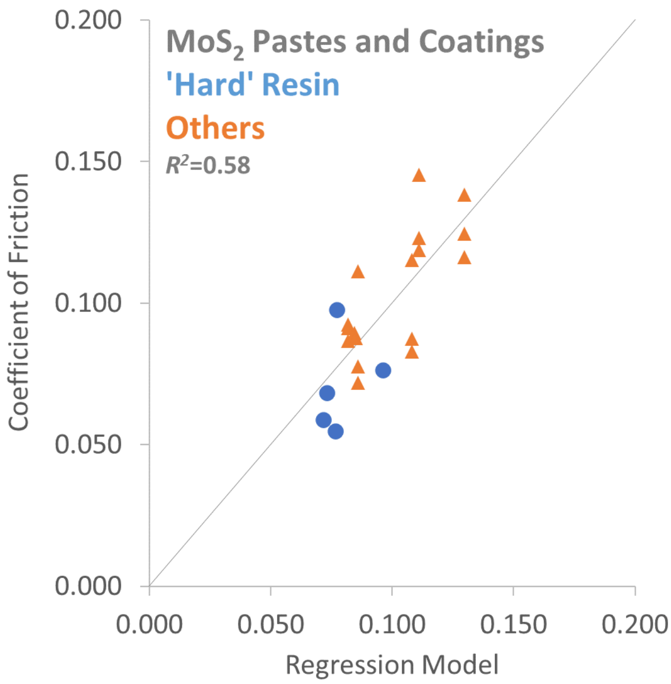

Figure 13: Comparison of Regression Model and Experimental Values for Friction Coefficients of MoS_2 Lubricants

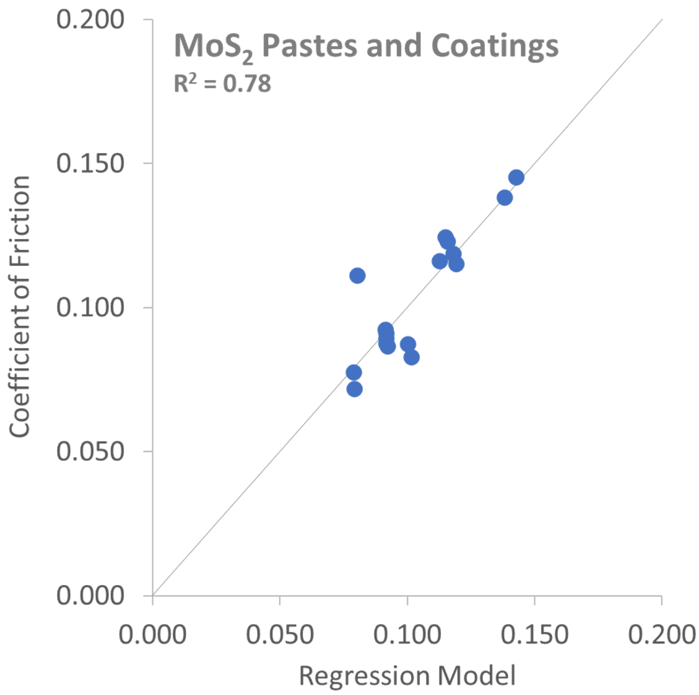

Figure 14: Comparison of Regression Model and Experimental Values for Friction Coefficients of 'Soft' MoS_2 Lubricants

For the MoS_2 lubricants, both resin-bonded dry film coatings and greases, the initial models generated R^2 values of 0.58 (μ) and 0.35 (*95% CI* μ), i.e., they did not describe variance in the dataset sufficiently. Plotting regression models and experimental data, e.g., Figure 13, showed that data from one of the resin-bonded dry film coatings had an outlying influence on the whole dataset. Therefore, this data was removed and regressions recalculated. As the removed data was from a coating with a harder resin, this was an interesting observation in itself. The remaining data, i.e., MoS_2 greases and resin-bonded coatings with 'softer' resins, could be loosely considered 'soft' MoS_2 lubricants as described by, e.g., Pilotti *et.al.* [112]. This data was described by the following regression models (see also Figure 14):

$$\boldsymbol{\mu_{MoS_2} = 0.060 + 0.000018\,\gamma_{Nut} + 0.0012\,\gamma_{Solids\%}/\gamma_{Nut}} \quad \textbf{(9)}$$

$R^2 = 0.78$

$$\boldsymbol{95\%\ CI\ \mu_{MoS_2} = 0.011 + 0.000018\,H_{Nut}} \quad \textbf{(10)}$$

$R^2 = 0.27$

The friction coefficient was most greatly influenced by the surface energy of the fastener (particularly the nut) and also the ratio of the surface energy of MoS_2 to the surface energy of fastener. The regression model in Equation 9 could be rewritten in the form:

$$\boldsymbol{\mu_{MoS_2} = 0.060 + \frac{0.000018\,\gamma_{Nut}^{\ 2} + 0.0012\gamma_{Solids\%}}{\gamma_{Nut}}} \qquad \textbf{(11)}$$

This form shows that the greater the surface energy of the fastener, the greater the friction coefficient, indicating that the most influential friction mechanism was adhesion. The lubrication mechanism of MoS_2 was influenced by adhesion of the fastener materials to each other and to the MoS_2. Presumably, the alignment under shear of MoS_2 particles and the formation of a lubrication film at the interface was dependent on adhesion of MoS_2 particles to the surfaces. Further, adhesion of the fastener surfaces to each other would increase the friction coefficient and would indicate lower effectiveness of the MoS_2 lubrication.

The low R^2 value of Equation 10, describing variation in coefficient of friction (*95% CI* μ), indicated that the regression model did not adequately describe variance in the dataset using the parameters considered.

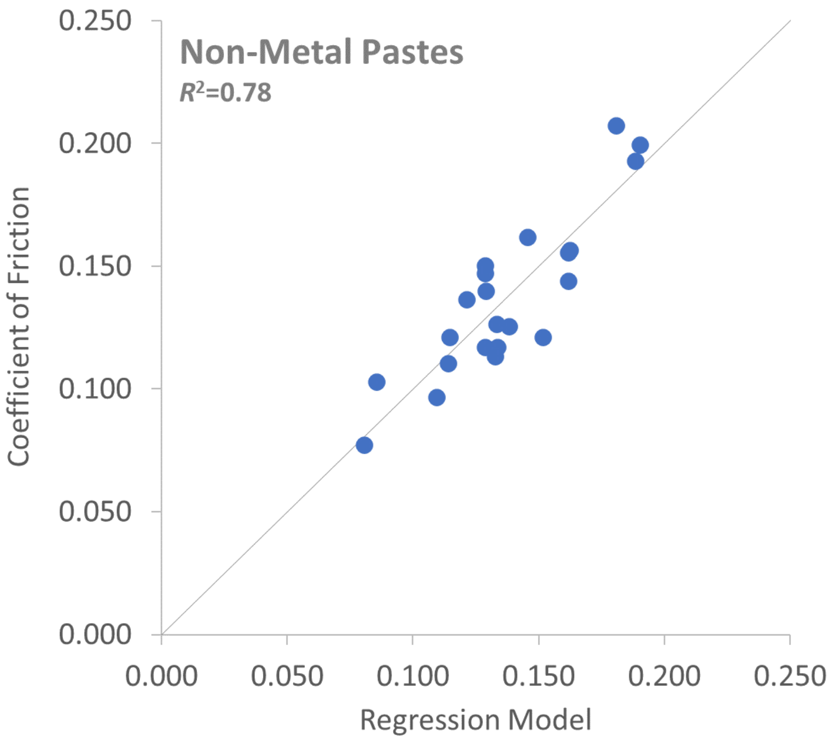

Figure 15: Comparison of Regression Model and Experimental Values for Friction Coefficients of Non-Metal Pastes

For non-metal pastes, the regression models are shown below and in Figure 15:

$$\boldsymbol{\mu_{NM} = -0.34 + 0.00012\ H_{Nut} + 0.0041 H_C - 0.040 \log(\frac{\gamma_{Bolt}+\gamma_{Solids}}{H_{Bolt}})} \quad \textbf{(12)}$$

$R^2 = 0.78$

$$\boldsymbol{95\%\ CI\ \mu_{NM} = 0.0029 + 0.000023\ H_{Nut} + 0.00035(H_{C^3}/P_{Head})} \quad \textbf{(13)}$$

$R^2 = 0.36$

The friction coefficient was influenced by the hardness of the fastener material and the composite hardness of the lubricating solids. The higher the hardness of the fastener material, the higher friction coefficient. The third term was a variant of the Rabinowicz static friction parameter. The greater the predicted static friction between the lubricating solids and the fastener surface, the lower the coefficient of friction. These observations appear to describe the parameters that were most influential in the formation of an interface layer of adhered/embedded particles, namely that the friction coefficient was lower when lubricating solids adhered to fastener surfaces and were insufficiently hard to cause abrasive friction/wear.

The low R^2 value of Equation 13, describing the variation in friction coefficient (*95% CI* μ), indicated that the regression model did not adequately describe the variance in the dataset using the parameters considered.

These analyses imply that changes in the surface energy of fastener and joint materials via, e.g., oxidation, surface treatment or contamination could significantly affect the friction of the system. MoS_2-lubricated systems would be most sensitive to changes in surface energy and that non-metal lubricated systems would be most sensitive to changes in hardness.

It is noted that these relationships are derived from finite datasets and should not be considered universal or predictive.

6. Implications for Industry

The following observations are especially relevant to industrial practice:

- There is no single friction coefficient value for a fastener lubricant; it will vary with fastener material. Sometimes there may be similar coefficients of friction for similar materials, Figure 3, but there are often significant differences, Figure 2. Friction coefficient must be regarded as a system response, not a material property.
- There are different lubrication mechanisms that affect lubricated friction in fasteners. In this study, plastic deformation of metals, burnishing and alignment of MoS_2, and adhering/embedding of non-metal particles were identified. Because these vary significantly with certain system parameters, care should be taken to ensure that torque settings are still valid when differences in these parameters are introduced, in particular:
    - for MoS_2 lubricants, changes in the surface energy significantly affect friction,
    - the carrier matrix also has a significant impact, e.g., a resin in a dry coating. Therefore, MoS_2 lubricants/coatings cannot be assumed to be defined by MoS_2 properties alone,
    - for non-metal pastes, friction can be sensitive to changes in fastener hardness.
- Because friction coefficients can vary greatly between different lubricant and material combinations, it confirms established best practice that torque settings should be calculated from data generated from the same lubricant-material combination as the final system.

  The assembly process transforms both fastener surfaces and the lubricant materials. Therefore, re-using an existing lubricant film should be avoided where possible. Friction coefficients can be greatly reduced on repeat assemblies, increasing the risk of over-tightening. All lubricants tested here showed this effect and MoS_2 lubricants were found to be especially sensitive. 'Best practice' is for fasteners to be cleaned and a fresh lubricant film reapplied between assemblies.
- Because a well-lubricated fastener surface 'runs in' on assembly, there is an increased risk of overtightening if the fastener is reused, even if fresh lubricant is applied. Therefore, where intentional re-use of fasteners occurs, changes in friction coefficient should be anticipated and accounted for.
- For pragmatism and simplicity, some industrial fastener standards permit the generation of torque charts, i.e. assume that friction coefficient will not vary significantly for a

given set of parameters, albeit providing cautions on their limitations e.g. SAE J2270 8.3.1.1 "…Torque tables identify torque with respect to fastener size, thread pitch, and material… Fastener torque tables do not always consider lubrication, self-locking feature, and other factors. The installer should be aware of the limitations of the tables" [43]. Engineers are encouraged to question assumptions about friction when calculating torque settings and to account for variation in friction as far as necessary, primarily by considering the friction coefficient as a system response.

- Trends toward sustainability in engineering include lightweighting projects, often substituting higher strength-to-weight materials like aluminium and titanium alloys. Changes in fastener and joint materials will probably generate differences in fastener friction. These effects must be allowed for.
- Even for the same materials, changes in surface treatments such as hardness and surface energy can affect fastener friction.

7. Conclusions

- Different lubrication mechanisms have been observed for different lubricant types, such as plastic deformation of metals, burnishing and alignment of MoS_2, and adhering/embedding of non-metal particles.
- Multivariate analysis has identified how these different mechanisms are sensitive to system parameters. Typically, interactions between the properties of the fastener materials and the lubricant materials:
    - the friction coefficients of MoS_2 lubricants were sensitive to the surface energies of the fastener material and MoS_2, i.e., the relative adhesions in the system.
    - the friction coefficients of non-metal pastes were most sensitive to the hardness of the fastener material and lubricant materials, and the Rabinowicz term for static friction, i.e., the ability of the lubricating solids to adhere, embed and form a lubricating interface layer.
- The assembly process changes the fastener surfaces (e.g. greater conformity) and also the lubricating materials (e.g. burnishing and alignment of MoS_2). These led to significant

reductions in friction coefficients in repeat assemblies. This should be allowed for, and adjusted to, when fasteners are re-used.

- The re-use of lubricant films has been shown to reduce the coefficients of friction even further with repeat assemblies, i.e., increasing the risk of over-tightening.
- Changes in fastener friction arising from changes in fastener materials, surface treatments, number of assemblies, etc. should be accounted for as standard practice. Friction coefficients should not be assumed to be constant.
- Fastener coefficients of friction and resulting torque settings are neither material properties nor lubricant properties, but must be regarded as system responses and should be calculated for each system.

8. Acknowledgements

Great appreciation is given to ROCOL. for allowing the publication of this work and to the other project students at the University of Bradford who laid the groundwork for the surface analysis techniques – Kevin Nicholas, Alexis Masclef and their supervisor, Dr David Bryant.

Dhari Aljeran contributed to this work as part of his studies at the University of Bradford. He is now at Higher Institute of Energy (HIE), Public Authority of Applied Education and Training (PAAET), Kuwait.

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