Lubrication of Fasteners with Different Geometries Using Minimal Treatments

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Controlling the applied torque during the tightening or driving of a fastener is an important assembly consideration. Modification of the coating lubricity has a great effect on torque requirements. For the applicato, applying a coating, traditional methods of controlling lubricity are complicated by the fact that fasteners with different geometries may exhibit different coefficients of friction, with identical coatings, under identical test conditions necessitating adjustment of the finishing steps based upon fastener geometry. In this paper a means to avoid this dilemma, based upon a coating system that adapts to different geometries during the tightening process is presented.

Introduction and background:

Fastener engineering is an important and dynamic endeavor. Innovative fasteners evolve very quickly in response to demands for automation, rapid assembly, weight reduction, appearance, and lower costs. Underlying all fastener-engineering requirements is the demand for safe and highly reliable six sigma compliant products.

The fastener assembly process generally relies upon the fastener obtaining a suitable clamping load in order to reliably hold a joint intact for the service life of the joint. It is difficult to directly measure the clamping load during factory assembly and alternative methods of assuring proper clamping load during assembly have been devised. Often these methods may rely upon measuring the force applied to the fastener during the assembly process. For rivets this might be the impact or broach load. For blind rivets or rivet 'nuts' measuring the differential pressure or pulling energy of a pneumatic hammer or impact tool might be employed.

With threaded fasteners measurement of the torque, or the shape of a torque curve vs. angle to sense yield, sometimes combined with a final turn of the fastener (torque plus angle) can be used to ensure proper clamp load, if the coefficient of friction (cof or μ) is known, or assumed to be invariant within a collection of similar fasteners, and, prior to assembly, off line, a highly accurate method to monitor assembly, such as using a load washer, was employed to correlate torque, torque plus angle, changes in torque curve, etc.[1].

In the past, assuring that fasteners have reproducible coefficients of friction has relied upon various methods, including application of a lubricant on the surface of the fastener. Zinc and zinc alloy plated fasteners with hexavalent chromium based chromates were often treated by spin drying and subsequent immersion into a solution of diluted lubricating material followed by another spin drying step, a treatment technology referred to as dry film lubrication (DFL) with post treatment steps of treat (chromate)-dry-treat (lubricate)-dry. The replacement of hexavalent chromium, principally by solutions of trivalent chromium passivation, has been accompanied by increased demands for corrosion performance. To accommodate corrosion performance requirements a post passivation immersion into a seal has become a common practice. It is possible to apply a dry film lubricant to a dry passivated zinc surface. Often this becomes a treat-dry-treat (seal)-dry-treat (lubricate)-dry process sequence. To shorten the process sequence

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[1] SAE short course #95030, Threaded Fasteners and the Bolted Joint, presented by Jess J. Conner.

lubricant can be added to the seal, this is often referred to as an integral lubricant (IL) seal. With the IL the process sequence simplified to treat (passivate)-dry-treat (IL seal)-dry, and with improvements has further simplified to treat (passivate)-treat (IL seal)-dry.
All of the above strategies allowed the amount of lubricant applied to a fastener to be varied. The integral lubricant seal was initially formulated so that different amounts of lubricant could be added to the seal to allow different cof to be achieved. Recent demands by automotive OEM's have required that the IL seal be premixed, that is, the amount of lubricant in the seal be constant. This demand was created by the need for consistency between applicators supplying the same fastener to an assembly line. The assembly line, using automated assembly methods, requires that batch-to-batch adjustment of fasteners will not be necessary.

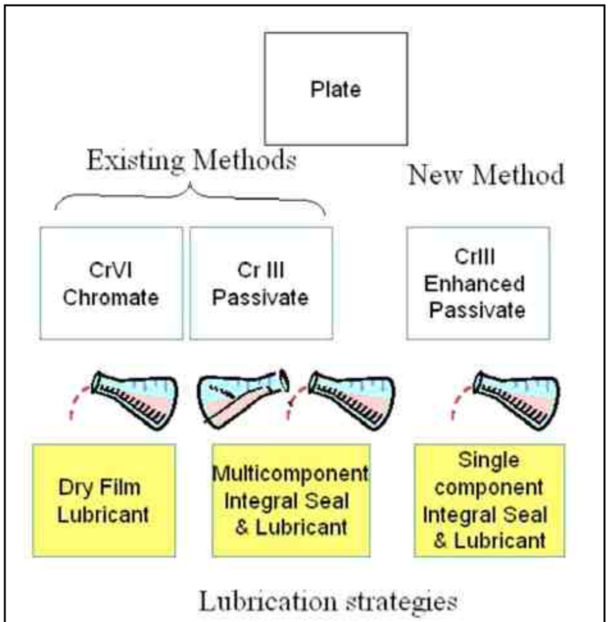

Figure 1 Schematic representation of existing and new fastener lubrication technologies.

While it is desirable to have a process that uses only a single lubricant containing solution it has proven to be very difficult to use such a process if a wide variety of threaded fasteners, possessing differing geometries, are processed through such systems.
Even if the property classes of the fasteners as well as bolt diameter and thread pitch are constant, we find that small variations in head geometry, such as concave head contact[2] will produce variations in torque, that exceed specified limits, if identical clamp loads are expected. For example, property class 8.8[*], M8 by 1.25 bolts with a thick flange head, thin flange head, and a hexagonal head have variations in torque at identical clamp loads, or pick points.
Careful examination of the bearing surfaces that the different fasteners contacted reveals that there are wear tracks of different dimension in all three cases. Furthermore, the speeds at which fasteners are turned or driven is usually quite slow, less than a meter per second, which indicates that fasteners are generally in a lubrication regime referred to as boundary layer lubrication. Consequently, we deduce that the challenge of creating a process with consistent chemical composition relies upon chemistry that can adapt to variations in wear under boundary layer conditions.
Wear volume is given as:

[2] ISO 15071:1999(E), from International Organization for Standardization
[*] Property class of metric bolts describes the ultimate tensile strength, UTS, and the yield by a number embossed on the head of the bolt. For example PC 8.8 has UTS of 800 MPa and yield of 0.8x800Mpa, or 640MPa. This, in turn, can be roughly related to hardness e.g., PC 10.x, 1000 MPa UTS, would have an H_k of around 325, roughly corresponding to Rockwell C of 33.

$$V=kLS/H$$

Where V is the wear volume, L in the normal load (Newtons), S is the sliding distance (meters), H is hardness (Pascals), and k is the wear coefficient (dimensionless).
Tribologists, studying wear and coefficient of friction (cof or μ), have observed that the wear coefficient is proportional to the coefficient of friction raised to a power between two and four.
The coefficient of friction is defined as:

$$\mu=F/L$$

Where F is the friction force, and L is the normal force. This is in nearly all cases a fractional value, and with fasteners typically between 0.1 and 0.2, so that a small variation in μ can produce a large variation in wear coefficient.
The preceding equation describes coefficient of friction for a simple sliding system. For a fastener that is being tightened by turning, the equation must be modified, and this is typically done as follows:

$$T=F\mu$$

Where T is the torque force (Nm).
This is then expanded, based upon a model for a threaded joint, which has three main components to the torque force, thread friction, under head friction, and bolt stretch, as follows:

$$T=F(P/2\pi + \mu_t r_t/\cos\beta + \mu_h r_h)$$

Where P is the thread pitch, μ_t the three friction coefficient, r_t the radius of the thread ($d_t/2$), β the half angle of the threads, μ_h the under head friction coefficient, and r_h the effect contact radius of the joint, usually given by:

$$r_h=0.5(d_o+d_h)/2=0.5D_b$$

Where d_0 is the diameter of the outer bearing surface and dh is the diameter of the hole in the bearing surface. This is an important assumption, and there are issues, for example, if a under head of a bolt is canted, so that a region near the outer diameter of the under head is the principle region which makes contact during tightening, isn't the 'average' contacting area, different than the preceding equation? There have been efforts to attempt to modify the torque equation to take into account such variations[3] but, for the present, the preceding equations are those used to derive most standards.
With these equations, a further simplification is made. Assume:

$$\mu_t=\mu_h=\mu_{total}$$

Then, with β given by 30^o (an ISO standard), the torque equation may be rearranged to:

$$\mu_{tot}=((T/F)-(P/2\pi))/(0.578d_t+0.5D_b)$$

Then solved for T:

[3] Said Nassar, *Bearing Friction Torque in Bolted Joints,* DAAE07-03C-L110, 12 Feb. 2004,

$$T=F(P/2\pi+\mu_{tot}(0.578d_t+0.5D_b)$$

Which becomes the working equation for torque tension measurements where F, in kN, the clamp load, is measured using a washer equipped with a load cell, and T, in Nm, is measured using a torque transducer, as a function of pitch, thread diameter and contact area, all in mm.

Using these equations we may observe how measurement would vary if a surface has a constant coefficient of friction but the contact area varies. We can also observe how torque values can vary if the coefficient of friction varies. In figures two and three graphs of such variations, for a typical PC 8.8 M8x1.25 fastener are presented.

From these data we conclude that an effective coating must allow for variation in cof as wear proceeds, that the variation in cof need to be relatively small, and that as wear proceeds, cof should decrease. If the lubricant does not change then the concentration of the lubricant must change, and it should be less concentrated near the surface, than at the interface with the next layer. We now refer to such coatings as ‘inverted’ lubrication.

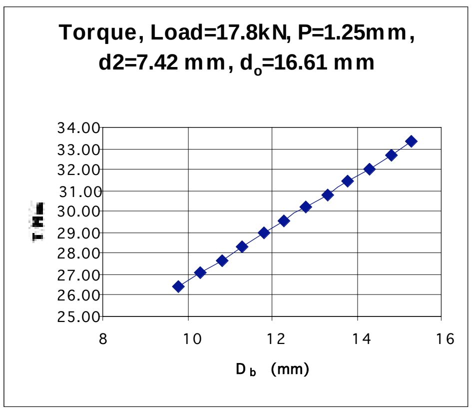

Figure 2 Variation of effective contact area effects observed torque.

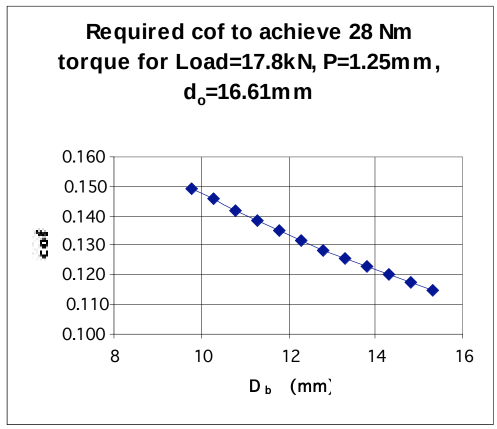

Figure 3 Small variations in cof can produce constant torque values as effective contact area changes.

We have created inverted lubrication coatings by several means, including:

1. Using a series of dips, each one having less lubricant or a lubricant with higher cof. We've tested this system using a series of seals, partially cured between coatings then fully cured after all coatings are applied. And, we have tested used this strategy with electroless nickel coatings, dip spin and electrophoretic paints, and with electrodeposits using included lubricant. We have done studies with zinc alloys, putting the most slippery electrodeposits at the bottom of a stack, and then applying subsequent alloy layers, each one less slippery. In all cases, wear rate is improved based upon reciprocating pin, Taber or Falex testing.
2. Another system we’ve tested 'bleeds' in lubricant during the coating phase, e.g. during spray painting, flame spray or vacuum coating,
3. And, another system that is now commercial, employs only a Cr^3 passivate and a seal. The passivate is what we call an 'enhanced passivate'. That is a passivate that has lubricant and is formulated so that the passivate layer is not

harmed by the lubricant, and a thin film of lubricant is present on the intact surface of the passivate when parts are withdrawn from solution. A second dip is made into a seal with lubricant. During the drying step, diffusion takes place and an inverted gradient is created in the seal with more lubricant adjacent the passivate than at the surface. When a fastener lacking asperities rubs the surface, the cof is adequate. But, when a fastener having significant asperities, or a non uniform contact area is used, the coating wears to localized regions of lower cof, so that overall, the tightening is unaffected. We test these types of coating using torque tension, broach load, and impact hammer test instruments.

This paper will describe how 'enhanced passivation', combined with a single additive integral seal, can achieve 'inverted lubrication', and produce fasteners which meet torque requirements even when fastener geometries vary.

Experimental:

The fasteners used for these torque tension experiments were:

1. M10x1.50x45 10.9 Property Class with Large Flange Head
2. M8x1.25x40 9.8 Property Class with Flange Head
3. M10x1.50x60 9.8 Property Class with Hex Head
4. M8x1.25x30 8.8 Property Class with Integral Washer

The bolts each have a different head geometry and different torque performance. The torque tension test method used in these experiments was Ford WZ 101, which requires plain steel nut and washer with specified roughness for the washer. Coefficient of friction was calculated from the formula presented in the introduction. Measurements were made using RST Torque Tension and TesT Torque Tension machines. The solutions used in the testing are commercial products[4]. Corrosion testing was done per Ford APGE at National Exposure Laboratories, Toledo, OH. Morphology and composition of the passivate and seal were examined using a Hitachi field emission SEM. Samples were fractured then sputter coated with ~50Å of gold prior to examination in the SEM.

Results:

Experiment 1: Effect of Bolt Geometry on Coating Wear Through

In this experiment the same bolt was tightened ten times with a fresh nut and bearing surface each time. The fasteners were zinc plated, followed by passivation in a trivalent passivate, then application of a topcoat with integral lubricant. In the first two tests a conventional passivate was used, in the third an enhanced passivate was used.

The first chart shows data from the integral washer bolt with no enhancer. As can be seen, the torque increased somewhat but the coating seems to be stable, due to the friction surfaces being the lubricated integral washer and underhead of the bolt.

[4] Corrolux® 530L process from Atotech, USA.

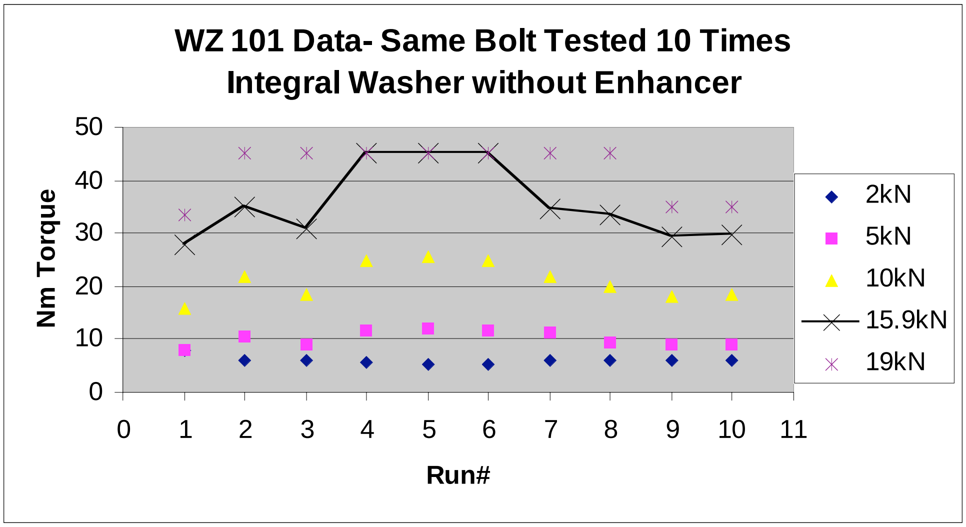

Figure 5: Black line- pick point tension (45 Nm= machine shut off point)

The next chart shows data from the same procedure using an M8 9.8 PC flange head bolt. The effect of the coating breakdown caused by friction between the bolt and steel washer is clearly seen, with the accompanying increases in torque required to reach the "pick point" tension (note that the highest torque value measured in this test was 45 Nm; the actual torque was higher):

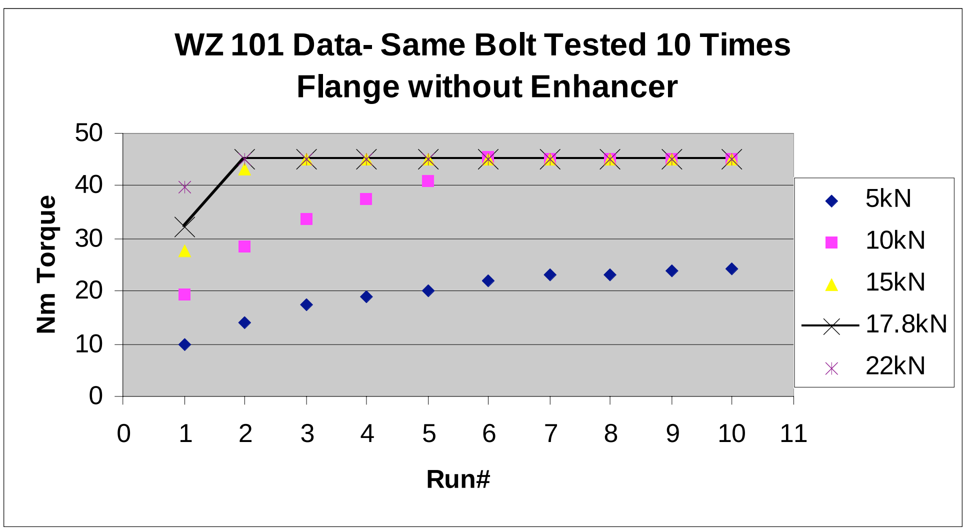

Figure 6: Black line- pick point tension (45 Nm= machine shut off point)

In the third test the same flange head bolt was used except with enhanced passivate in place of conventional passivate. The inverted lubricant mitigates the effect of coating wear through and reduces the increase in friction over ten tightenings:

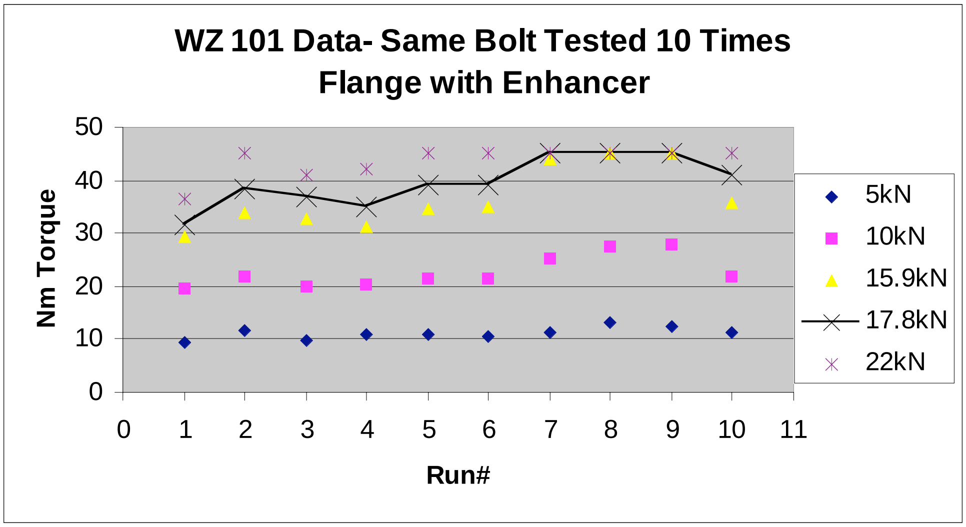

Figure 7- (45 Nm= machine shut off point)

The data from these tests show that the use of inverted lubrication can reduce the torque increases caused by wearing of the coating on flange head threaded fasteners.

Experiment 2: Mitigation of Torque Differences Caused by Bolt Geometry
In this experiment mixed batches of the four bolt types were coated together in a sequence of zinc plate, enhanced passivate, and seal. The amount of enhancer was varied from 0 to 10 milliliters per liter. Torque tension testing was performed according to Ford WZ 101. The experiment was repeated four times with different seal formulations containing increasing amounts of lubricant. The results are summarized in the following charts:

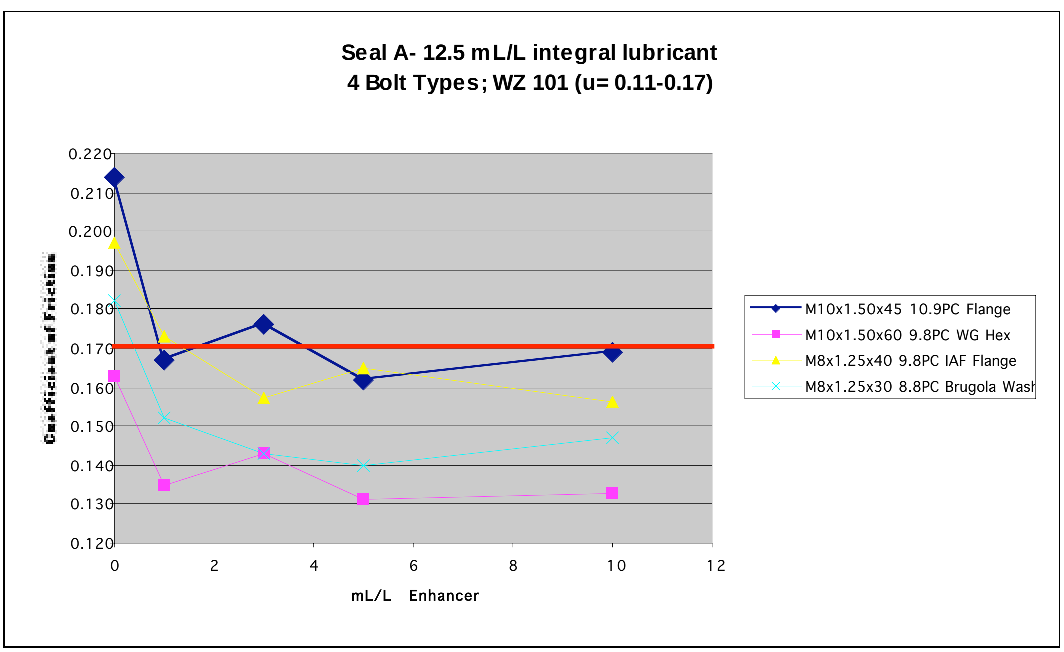

Figure 8- Spec requirement= total coefficient of friction of 0.11-0.17 with 6 sigma accuracy

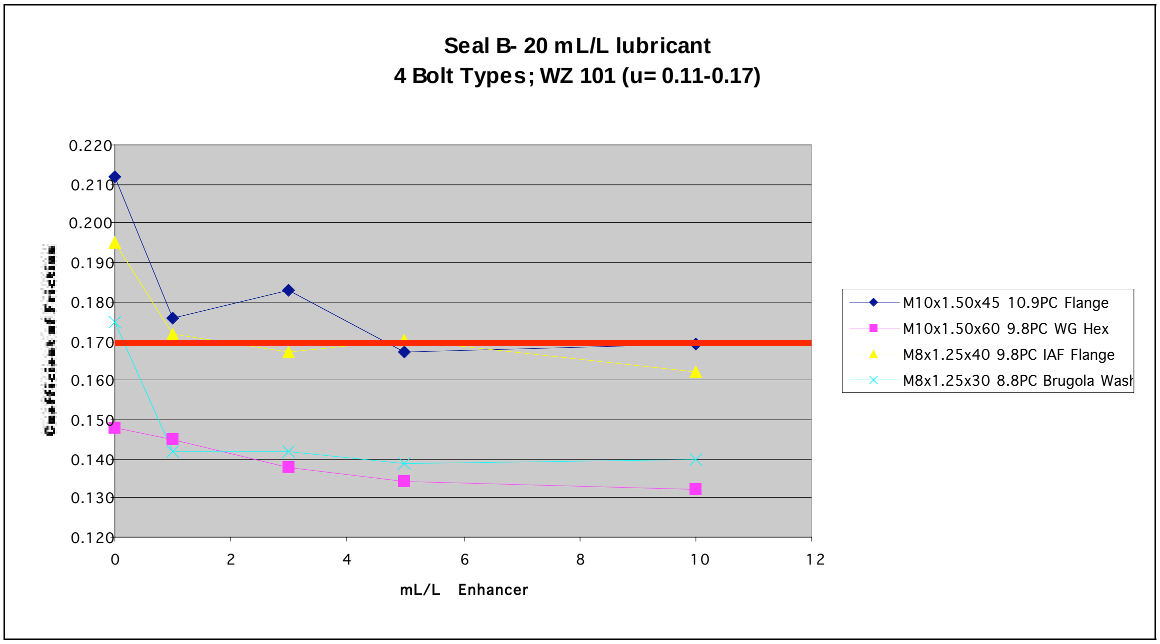

Figure 9- Spec requirement= total coefficient of friction of 0.11-0.17 with 6 sigma accuracy

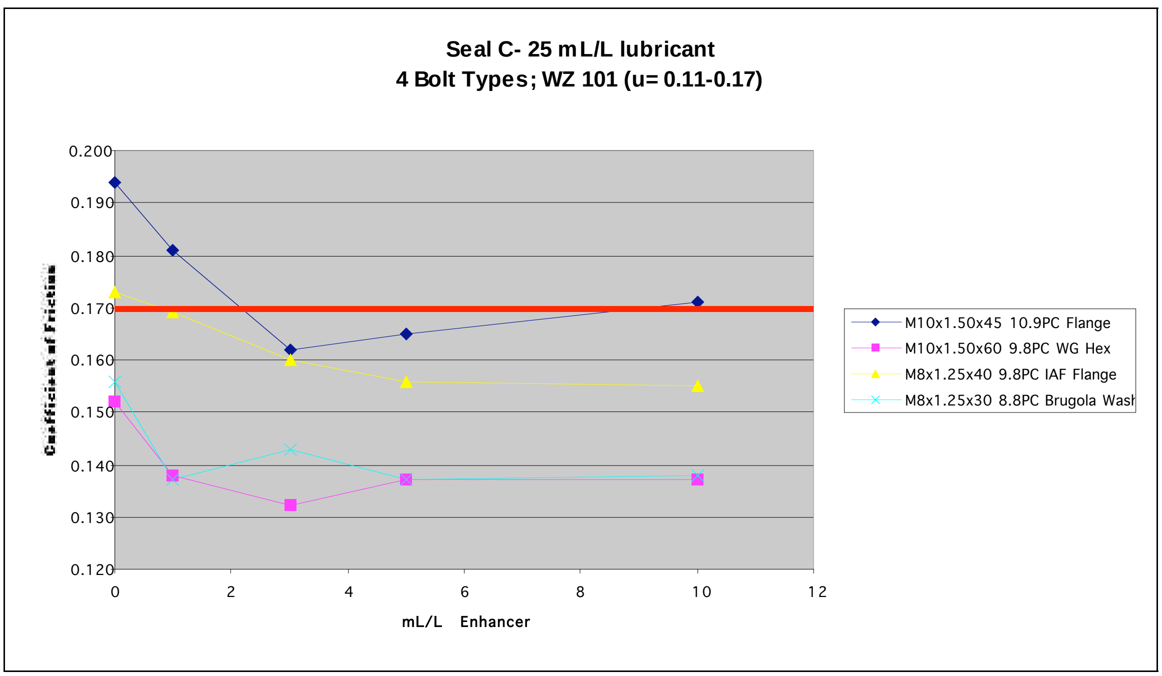

Figure 10- Spec requirement= total coefficient of friction of 0.11-0.17 with 6 sigma accuracy

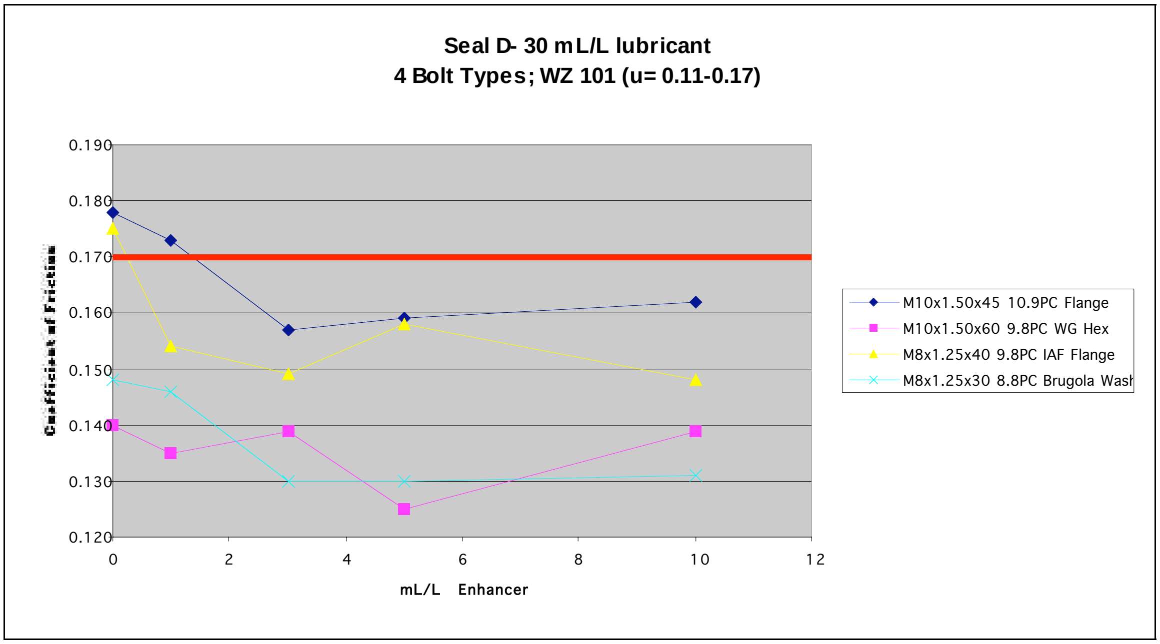

Figure 11- Spec requirement= total coefficient of friction of 0.11-0.17 with 6 sigma accuracy

Coefficient of friction decreased as expected with increasing lubricant amounts in the seal. The average high to low range of the coefficient of friction tended to decrease as well with increasing enhancer concentration in the passivate.

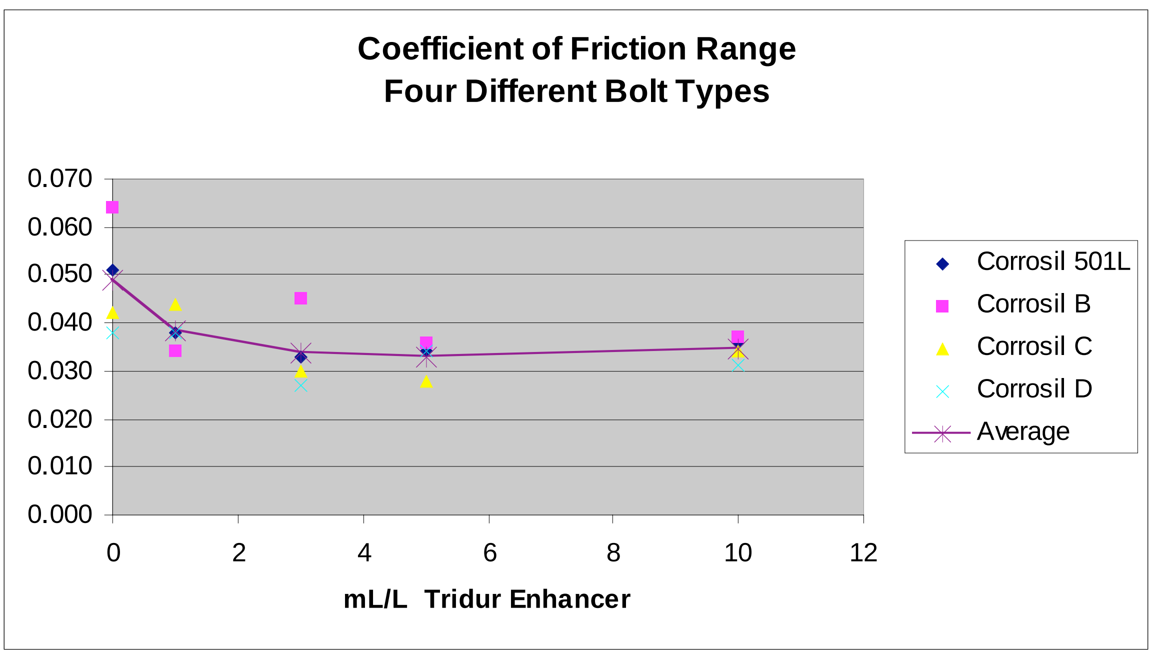

Figure 12

The data from Experiment 2 indicates that torque differences between bolts with differing geometries can be mitigated to simplify meeting the torque requirement.

Experiments 3-5 – Affect of enhanced passivation on uniformity of the passivate film and integral lubricant seal and the effect of 'inverted' lubrication on corrosion.

In the first of these experiments the amount of lubricant was varied in the passivate. Coupons were treated in lubricant 'enhanced' passivates and then fractured and examined by SEM (figure 13). The passivate layer is undisturbed by the lubricant and a thin lubricant film, with thickness proportional to concentration, is formed on top of the passivate film.

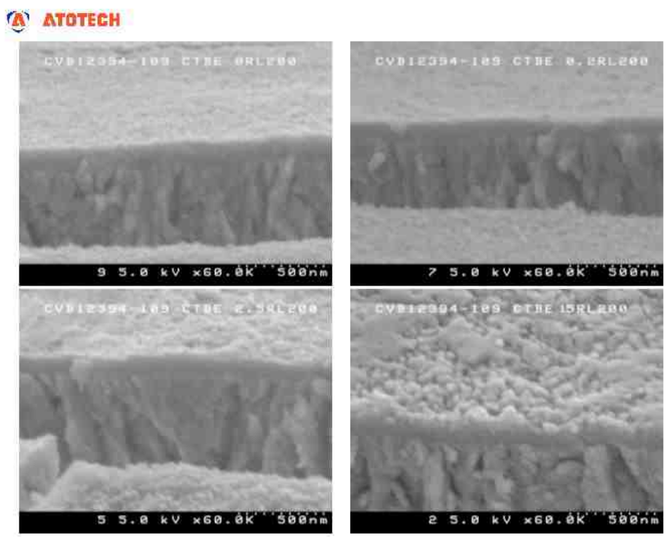

Figure 13. Addition of varying increasing amounts of lubricant (upper left clockwise) to a trivalent passivate creates two distinct layers and does not harm the passivate film.

Additionally, a coupon was processed through enhanced passivate and a series of three integral lubricant seals, each with less lubricant than the preceding treatment. After curing, the coupon was fractured and examined by SEM. In figure 14 the gradient of lubricant lubricant particles is obvious.

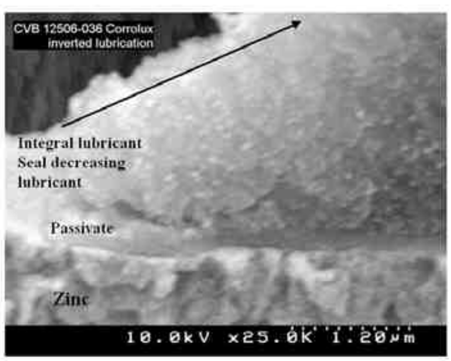

Figure 14. Enhanced passivation followed by dipping into integral lubricants with successively less lubricant then cured. The high contrast lubricant particles have

Numerous corrosion studies, ASTM B117 neutral salt spray, GM9540P, Ford APGE, etc. have been performed using the system of an enhanced passivate followed by one or more integral lubricant seals with controls of non enhanced passivate. In no experiments has the presence of lubricant enhanced passivate harmed the corrosion results. In fact, the corrosion performance appears to depend upon the chrome concentration of the passivate that is enhanced, the number of integral lubricant dips following enhanced passivation, and the heat treatment prior to corrosion testing. Routine ASTM B117 corrosion performance exceeding 216 hrs no white corrosion, and exceeding 360 hrs no red rust, with or without 4 hour 120oC bake has been observed for integral washer and flange head bolts with 8-10 μm of zinc, followed by enhanced passivation, and a single integral lubricant seal. Similar bolts subjected to APGE cyclic testing exceeds 12 cycles, no red rust, for unbaked bolts, and 10 cycles, no red rust, for bolts that were baked. These bolts also passed Ford WZ100 torque tension requirements with mean torque values +/-3 sigma, within the allowed limits of the specification.

Conclusions:

It has been shown from laboratory torque tension data that apparent effects caused by differences in fastener head geometry and increases in torque due to coating wear through, measured using conventional torque tension equipment and equations, can be mitigated by using a layered or gradient “inverted lubricant” approach. The advantages of this approach include fewer rejects and more ease of use in production due to fewer tanks and less bath adjustment or solution ‘change out’. The addition of an optional “dry film lubricant” after the seal step allows even greater flexibility in coating very high torque fasteners or fasteners requiring maximum lubricity such as rivets, ‘rivnuts’, or bolts with high surface hardness.

