PCB PIEZOTRONICS

FUNDAMENTALS OF TORQUE-TENSION AND COEFFICIENT OF FRICTION TESTING

Fundamentals of Torque-Tension
and Coefficient of Friction Testing
Understanding the Basics of Tightening Threaded Fasteners

More and more fastener coatings are being required to not only comply with corrosion specifications, but to also fulfill specified tightening characteristic requirements. These requirements include at least a torque-tension test and at times a coefficient of friction test. The torque-tension test usually consists of an acceptance input torque window at a specified tension or clamp load value, while the coefficient of friction test requires an acceptance coefficient of friction window (typically the reference coefficient of friction μ_{ges}) at a specified tension or clamp load value.

Prior to understanding the "whys" of this testing it is necessary to know some of the "hows" and "whats" of basic tightening of threaded fasteners. The following sections will attempt to answer all of these questions.

Fastener Loads During Tightening

When a bolt is tightened, by applying a torque to the bolt head or nut, the bolt gets stretched causing the clamping plates to be compressed together.

The nomenclature used to describe the loads remains true to its German roots from the VDI and DIN specifications where:

F_M: bolt tension or preload prior to any joint settling.

F_K: clamp load in the clamped pieces

M_A: applied torque or input torque

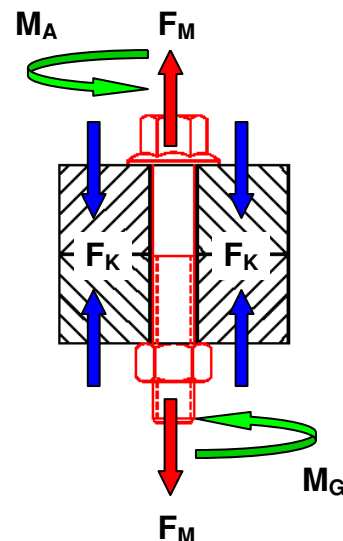M_G: thread torque or the reaction torque to the input torque

Figure 1: *Fastener Loads during Tightening*

The thread torque can be described as the torque it takes to keep the nut from turning when an input torque is applied to the bolt head.

Bolt Tension and Clamp Load

As the bolt is being tightened, the bolt is actually being stretched causing tension while the clamped pieces are being compressed causing clamp load. This phenomenon can be shown in the traditional joint diagram in Figure 2.
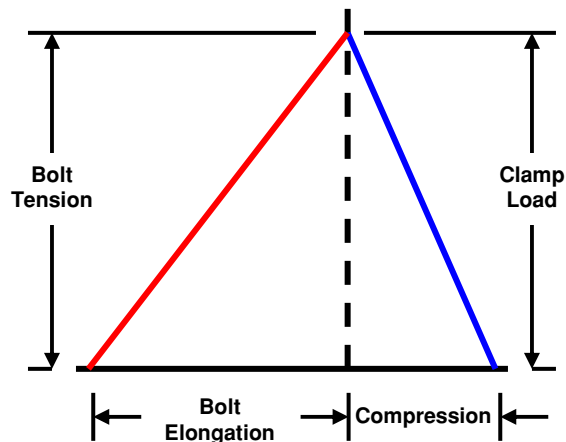

Figure 2: *Joint Diagram*

The slope of the curve represents the resilience (elasticity) of the material being stretched or compressed.

Note: The resilience is the inverse of the spring rate given in units of length/force (i.e. in/lb, mm/N).

The slope is a function of the material properties of the part being stretched or compressed along with the length of the part being stretched or compressed and remains a constant until the material experiences yield or permanent deformation. This being the case, it can be shown that to obtain a certain bolt stretch, for a given material and geometry, it will always require the same amount of bolt tension. The following equation is the basic equation for the elastic resilience

$$\delta = \frac{l}{E * A}$$

δ: Elastic Resilience
l: Length
E: Young's Modulus
A: Cross-Sectional Area

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The length and cross-sectional area are strictly taken from the geometry of the bolt or clamped parts and the Young's Modulus is a material property describing elasticity. Based on this the slope of the joint diagram will change when permanent deformation sets in due to possible changes in both length and area.

The bolt elongation can be directly correlated to the angle of turn in a bolt when a torque is applied. The elongation of the bolt is measured in degrees of rotation rather than units of length. Much like the joint diagram the slope of the clamp load-angle curve is only a function of the resilience of the material. The difference being that the slope is a combination of resilience between both the bolt and clamped parts. A permanent deformation of either the bolt or the clamped parts will cause a change in slope in the clamp load-angle curve. The following figure shows a clamp load-angle curve tightened to joint failure.
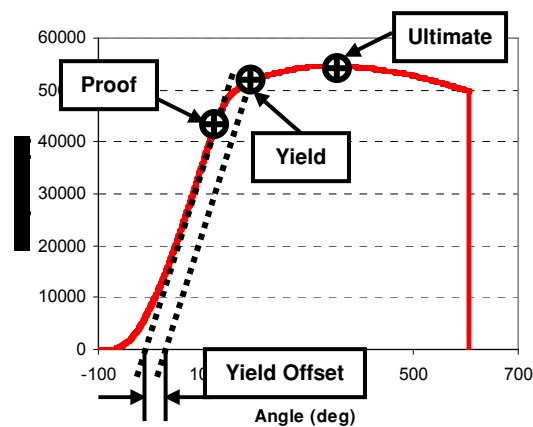

Figure 3: *Clamp Load-Angle Failure Curve*

Figure 3 describes the key points on the clamp load-angle curve; the methods to measure the points are as follows:

Proof Load: A projection is made tangent to the linear portion of the clamp load-angle curve. The point at which the slope of the curve deviates from the projection is known as the Proof Load. It is in this linear range that the entire lot of equations regarding fastener tightening and friction are valid. Staying within the elastic range also provides the most consistent torque-tension and angle relationship between common fasteners.

Yield Load: The projection made to determine the proof load is offset a given number of degrees (based on the resilience of the joint). The point at which the offset projection intersects the clamp load-angle curve is known as the Yield Load.

Ultimate Load: The Ultimate Load is measured as the peak clamp load value.

Input Torque, Thread Torque, and Underhead Torque

Input torque can be thought of as the amount of work (or energy) applied to a threaded fastener causing the bolt to turn an "x" amount of degrees and in turn stretching the bolt producing a clamp load. The input torque is divided into two factors: thread torque and underhead torque.

Thread Torque: The reaction torque to the input torque, or the amount of the torque it takes to keep the nut from turning when an input torque is applied to the bolt head.

Underhead Torque: The amount of the input torque it takes to overcome friction in the bearing surface of the part of the fastener being turned.

By definition the Input Torque is described as:

$M_A = M_G + M_K$

M_A: Input Torque
M_G: Thread Torque
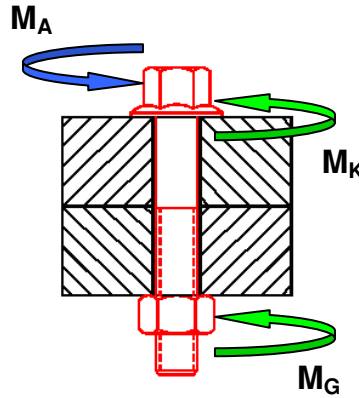M_K: Underhead Torque

Figure 4: *Torques*

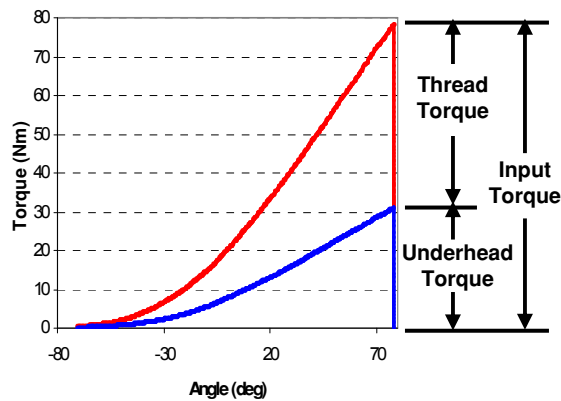Figure 5 graphically describes the input torque definition.

Figure 5: *Input Torque Definition*

The thread torque is a combination of two factors: thread friction torque, pitch torque.

Thread Friction Torque: The frictional portion of the thread torque describing the amount of torque it takes to overcome the friction in the engaged threads.

Pitch Torque: The geometrical factor of the thread torque describing the amount of torque that is used to stretch the bolt.

The pitch torque is calculated by the following equation:

$$\text{Pitch Torque} = \text{Thread Pitch} * \frac{\text{Clamp Load}}{2\pi}$$

The thread friction torque is then calculated by subtracting the pitch torque from the thread torque. Graphically, this makes the input torque definition look like Figure 6.
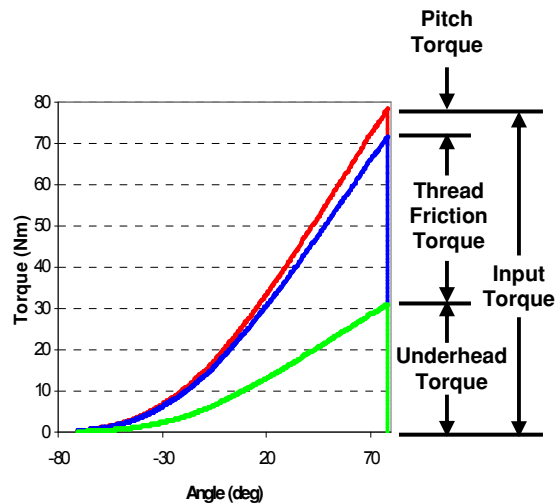

Figure 6: *Expanded Input Torque Definition*

Thread,Underhead,andReferenceFrictionCoefficients

The amount of thread torque and underhead torque present in a given tightening can be described using friction coefficients. Friction coefficients are always described in terms of two materials, platings, coatings, etc., and are defined as follows:

Thread Friction Coefficient: The value describing the friction between mating threads.

Underhead Friction Coefficient: The value describing the friction between the bearing materials of the bolt and first clamping plate or the nut and first clamping plate (depending on which end of the fastener is being turned).

Reference Friction Coefficient: The value describing the overall friction in the joint combining aspects from both the threaded and bearing surfaces.

The friction coefficients are calculated values using the measured input torque, thread torque, and clamp load along with some geometric characteristics. The equations are as follows (from the DIN 946 specification):

Thread Friction Coefficient (μ_G)

$$\mu_G = \frac{\left[\dfrac{M_G}{F_M}\right] - 0.159 * P}{0.578 * d_2}$$

M_G: Thread Torque
F_M: Assembly Preload (Clamp Load)
P: Thread Pitch
d_2:   Thread Pitch Diameter

Underhead Friction Coefficient (μ_K)

$$\mu_K = \frac{2 * (M_A - M_G)}{D_{KM} * F_M}$$

$$D_{KM} = \frac{d_w + d_h}{2}$$

M_A: Input Torque
D_{KM}:   Mean Bearing Diameter
d_w: Bearing Surface Diameter
d_h: Through Hole Diameter

Reference Friction Coefficient (μ_{ges})

$$\mu_{ges} = \frac{\left[\dfrac{M_A}{F_M}\right] - 0.159 * P}{0.578 * d_2 + \left[\dfrac{D_{KM}}{2}\right]}$$

Typically, 80% – 90% of the input torque is used to overcome friction in the threaded and underhead regions leaving only 10% – 20% of the input torque left to stretch the bolt producing clamp load. The distribution of which area (thread or underhead) that will take more torque to overcome friction depends on the bearing surface area along with the friction coefficient between the mating surfaces.
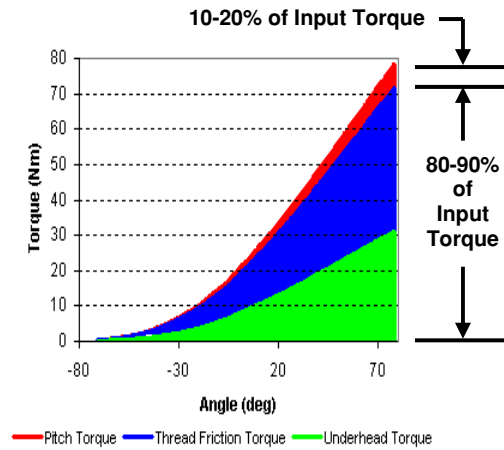

Figure 7: *Input Torque Usage*

Knowing that pitch torque is directly related to clamp load (for a given bolted joint), if the pitch torque is increased (by lowering friction in the thread or underhead areas) more clamp load will be attained at a given torque value.

Controlling the friction coefficients, in both the thread and underhead regions, will result in more consistent and predictable clamp load values.

Effects of Varying Friction Coefficients

Consider the example of tightening a certain fastener to 50 Nm and view the effects of changing the friction coefficients have on the attained clamp load. Four test samples of the same lot were tested under different lubrication conditions: no lubrication, lubricated underhead, lubricated threads, and lubricated underhead and threads
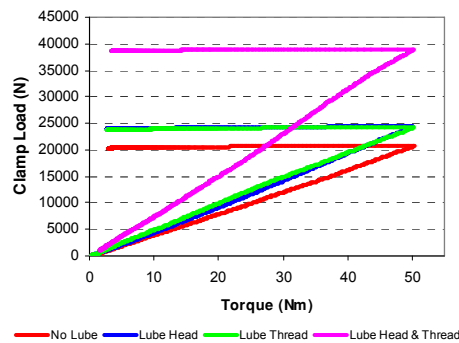

Figure 8: *4 Samples Tightened to 50 Nm*

Each sample had the torque separated into its three parts giving the following:
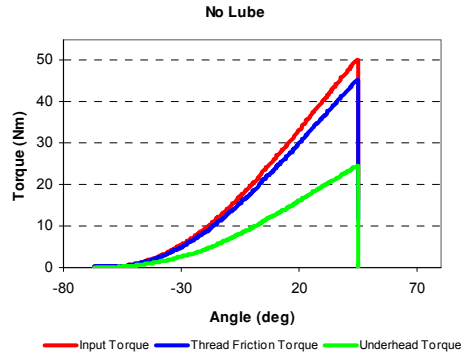
No Lube

Figure 9: *No Lubrication*

Figure 9 shows the torque breakdown of the sample with no lubrication. This will be considered that baseline sample. Figures 10 and 11 give the torque breakdown for samples with lubricated underhead and threads respectively.
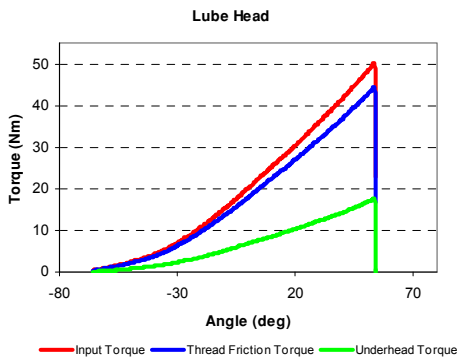
Lube Head

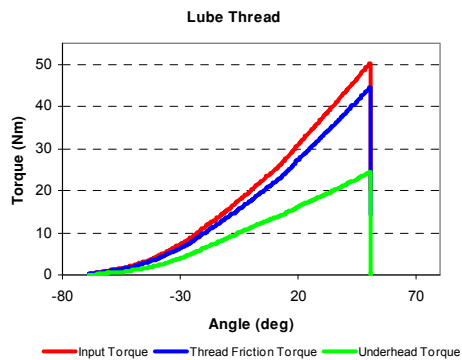**Figure 10**: *Lubricated Underhead*

Lube Thread

Figure 11: *Lubricated Threads*

Figure 12 shows the sample with both the underhead and thread regions lubricated.
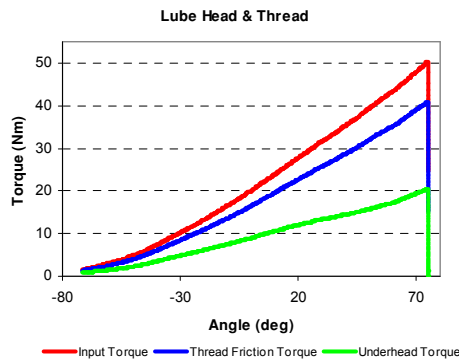
Lube Head & Thread

Figure 12: *Lubricated Underhead and Threads*

Reviewing Figures 8-12 it can be seen that increasing the pitch torque by lowering the friction coefficients will cause an increase in clamp load.

Table 1: *Example Data (Samples Tightened to 50 Nm)*

Sample	Clamp Load	Pitch Torque
No Lubrication	20752 N	5.0 Nm
Lubricated Underhead	24425 N	5.8 Nm
Lubricated Threads	24235 N	5.7 Nm
Lubricated Underhead & Threads	38953 N	9.2 Nm

Friction Coefficient Ranges

It is important to note that for each coating/coating, material/material combination there is a range of values that define the frictional characteristics. These ranges are determined by calculating the ±3-sigma friction coefficient values from a particular lot of samples tested. Using the calculated standard deviation (*n*-1 formula) of the lot tested and assuming a normal distribution of results the ±3-sigma friction values can be calculated using the following equations:

Standard Deviation (σ)

$$\sigma = \sqrt{\frac{\sum(x - \overline{x})^2}{n - 1}}$$

x: Data Value
$\overline{x}$:   Mean of Data Values (average)
n: Number of Samples

Mean ($\overline{x}$)

$$\overline{x} = \frac{\sum x}{n}$$

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±3 Sigma Values (±3σ)

$$-3\sigma = \bar{x} - 3 * \sigma$$

$$+3\sigma = \bar{x} + 3 * \sigma$$

Typically a test lot with 10-30 samples will be tested to determine the friction range. The ±3-sigma values estimate the range limits that the entire population will fall into based on the small number of samples tested. The model will allow for 99.73% of the population to fall within the limits. The following figure graphically describes a normal distribution with the calculated values.
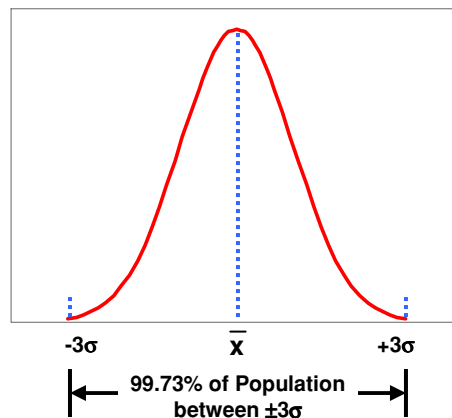

Figure 13: *Normal Distribution*

Note: The larger the sample size the better the distribution model will be. The increase in the confidence of the model becomes negligible after 30 samples.

With geometric parameters assumed constant and using a highly capable tightening tool, the ±3 sigma clamp load range at a given torque value will be directly proportional to the ±3 sigma range in the friction coefficients (from the variances in the three torque factors).

Some friction ranges will be wider than others resulting in wider clamp load ranges at a given torque. The following two figures describe both a wide range and thin range of friction coefficients.
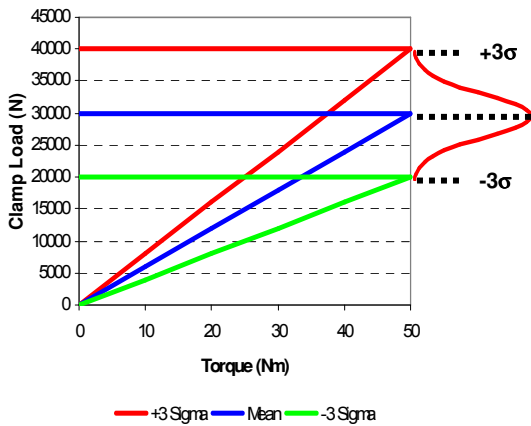
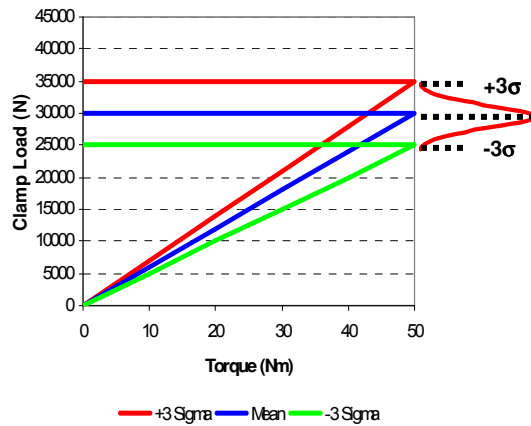**Figure 14**: *Wide Friction Range at Given Torque* **Figure 15**: *Thin Friction Range at Given Torque*

The wider the friction coefficient range the wider the clamp load range will be at a given torque value. It is important to control the friction coefficient range in order to obtain a more consistent bolted joint.

Torque-Tension Testing

Most torque-tension specifications will require tightening a specified bolt/nut/washer combination to a given clamp load value and returning a torque value. The nut and washer will typically have finish/plating, material, and dimensional requirements, while the bolt will have requirements for the material and dimensions. This reduces all of the variances in the joint to the bolt plating or coating.
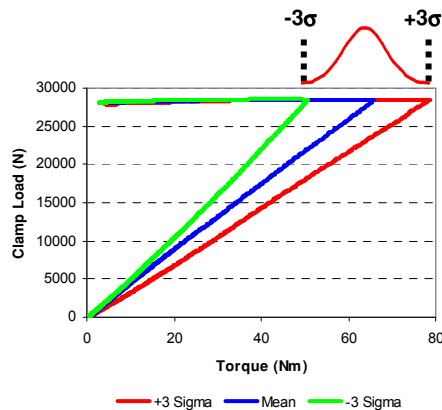

Figure 16*: Torque-Tension Test*

Figure 16 shows the results of a sample lot being tested and returning the ±3 sigma torque values at a specified tension value. The distribution will be compared to the specified acceptance distribution to determine torque-tension acceptance.

The coefficient of friction values can be calculated using the previously described equations along with the torque-tension data. The calculated data will be compared to the specified acceptance distribution data to determine coefficient of friction acceptance.

Conclusion

Torque values are strictly based on the friction present in the bolted joint assembly. Variances in the friction will lead to variances in the amount of clamp load/tension attained at a specified torque value.

Friction variances that can affect the tightening of a bolted joint occur in the underhead/bearing area and in the threads. The friction variances are always a function of two materials or platings of the mating parts. Coefficient of frictions can never be given in terms of one material or plating.

Testing of platings/coatings is used to limit the amount of variance allowed in the friction coefficients against a specified material. This allows for the end user to have confidence that the plating/coating being used will have a consistent relationship between torque and tension.

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