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FRICTION COEFFICIENT INFLUENCE ON THE FASTENING TORQUES OF A BOLTED JOINT

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Abstract.
One of the most common ways a component or a system of components fail is due to the wrong clamp force applied by a nominal torque that results on a bolted joint getting loose or breaking during operation. Still, one of the several methods available to join and connect parts within a component is by using threaded elements. The objective of a joining bolt is to join different surfaces and avoid their separation. While united, loads like torque and forces appear in the joined parts. The significant moments that are revealed during the joining process are named as pretension torque, grip torque and spin torque. The distribution and magnitude of these moments are affected by several factors. To understand this variance of clamp forces achieved by the same applied torque, mathematical models based on literature were applied first to understand and model the friction coefficient influence on common torques applied on threaded bolts. To verify this approach an experiment using a device capable of measuring the actual friction coefficient during the torque application was used to compare the results. During the machine cycle a motor was responsible for fastening the bolted joint while a load cell measured the clamp force until it reached the desired value and the real friction coefficient was one of the results available. The experiment was repeated several times to validate the method and the results gathered were within the expectation showing that the torque required to achieve the same clamp force can vary a lot due to the friction coefficient variance depending on material, surface finish, presence of dirt and debris in between the threads. Therefore it was verified that using only torque based on moment application for bolted joints can be inaccurate on achieving the desired clamp force and the recommended method to avoid this is by fastening with a torque angle.

Keywords: *fastening torques, friction coefficient, Variance*

1. INTRODUCTION

1.1 Establishing the subject

According to Juvinall (1983), a lay person may consider threaded bolts (bolts, nuts, etc.) as one of the simplest parts of the existing machine elements. Given the right observation, the engineer will verify that these components, that apparently are simple, have a vast variety. An example given by Juvinall (1983) in its book is that the fuselage of an aircraft can use up to 2.4 million fasteners that can cost around 750 thousand dollars.

In Collins (2006) work, the selection of the fastening type and method depends on various factors, including direction and size of the load, if the load is symmetric or eccentric, if the parts to be assembled have the same or different materials, dimensions, thickness, geometry and weight, the part's alignment, if the joint is to be permanent or not, and last but not least, the cost of the joint assembly.

Therefore bolted joints, including threated bolts, machine bolts and studs are the most selected and used type of union by engineers and machine designers.

1.2 Objective

The objective of this paper is to experiment and understand the influence of the friction coefficient on the fastening torques and moments of a bolted joint by varying the materials of the assembly. Using a device that is capable of measuring the real friction coefficient during the torque application on a threaded bolt, the experimental results will be compared to the mathematical models available in order to verify both approaches.

2. BOLTED JOINTS

2.1 Threaded fastening elements

The success or failure of a project can depend on the appropriate selection of the joining elements. Threaded bolts can be used on fixing parts as well as to move loads. (Norton, 2000)

Figure 1 shows the basic elements of a threaded joint. It consists on an external element, such as a threaded bolt, machine bolt, stud, assembled in an internal threaded element, such as a nut, or threaded holes. Of one the advantages of this type of union is that the element's material can be the same or different. (Collins, 2006).

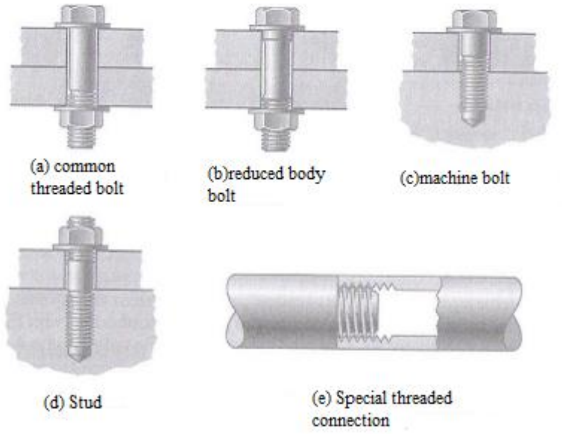

Figure 1. Different types of fastening elements
Source: Collins (2006)

There are several types of fastening elements available in the market. For each project or application a given element with a specific thread or body may be used according to a standard. A vast variety of materials are also available. Several nut types are also available. (Faires, 1976)

2.2 Thread Classification and Geometry

For Shigley (2006), screw threads follow a basic terminology. According to Fig. 2, the pitch is the distance between adjacent thread forms measured parallel to the thread axis. The pitch in U.S. units is the reciprocal of the number of thread forms per inch N, the major diameter (d) is the largest diameter of a screw thread, the minor (or root) diameter (dr) is the smallest diameter of a screw thread, the pitch diameter (dp) is a theoretical diameter between the major and minor diameters, the lead l, not shown on Fig. 2, is the distance the nut moves parallel to the screw axis when the nut is given one turn, for a single thread the lead is the same as the pitch.

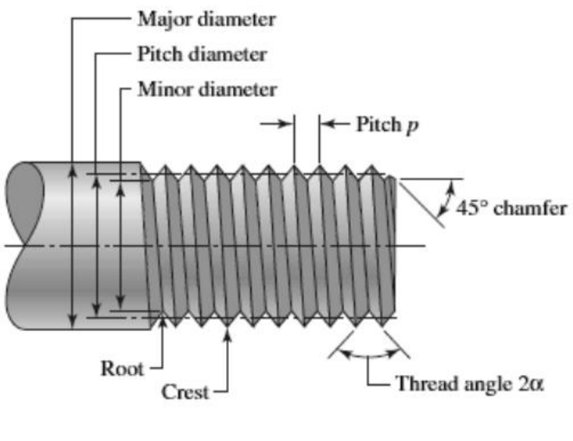

Figure 2. Terminology of screw thread
Source: Shigley (2006)

For the metric profile the thread's angle is 60 degrees. (Shigley, 2006 and Juvinall, 1983).

2.3 Bolt Torque related to Bolt Tension

One of the main desired behaviors of a bolted joint is to prevent separation between the contact surfaces of the parts of the joint. According to Shigley (2006), high preloads are desirable in important bolted connections, there must be means of ensuring that the preload is actually developed when the parts are assembled. Figure 3 shows a bolted connection with the loads represented by the letter P.

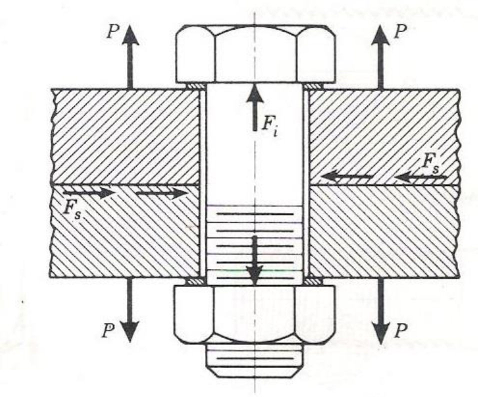

Figure 3. Bolted connection with loads during work cycle
Source: Shigley (2006)

In the work by Bickford (1995) the overall length of the bolt can be measured using a micrometer once it is assembled, the bolt elongation due to the preload F_i can be computed using the Eq. 1.

$$\delta = \frac{F_i l}{(AE)} \tag{1}$$

Where δ is the elongation, F_i is the force being applied on the bolt l is the bolt's length, A the cylinder area and E is the Young's Modulus.

However it is usually impractical to measure the bolt's elongation during its assembly, either the threaded end is often in a blind hole or in an assembly line it would take too much time to measure. In such cases torque wrenching, pneumatic impact wrenching or the turn of the nut method are commonly used. (Shigley, 2006).

According to Shigley (2006), torque wrenches have a built in dial that indicates the proper torque.

Following the work of several authors, Shigley (2006), Collins (2006), Bickford (1995), Norton (2000), Juvinall (1983), VDI2230 Standard (2001) and many others, although the friction coefficient varies a lot, a good estimate of the torque required to produce a given preload can be computed by Eq.2.

$$T = \frac{F_i d_m}{2}\left(\frac{1+\pi\mu d_m sec\alpha}{\pi d_m - \mu l sec\alpha}\right) + \frac{F_i \mu_c d_c}{2} \tag{2}$$

Where d_m is the average of the major and minor diameters, μ is the friction coefficient, α is the thread angle, μ_c the collar's friction coefficient and d_c the collar's mean diameter.

Since tan λ = *l/πdm*, and the mean collar diameter d_c = d + (1,5d/2), where d is the pitch diameter (Shilgey, 2006) by dividing the numerator and denominator of the first term by *πdm* it gets:

$$T = \left[\left(\frac{d_m}{2d}\right)\left(\frac{tg\,\lambda + \mu sec\alpha}{1-\mu tg\,\lambda\, sec\alpha}\right) + 0{,}625\mu_c\right] F_i d \tag{3}$$

Rearranging the terms so the force is the function of the applied torque on Eq. 3, it gets:

$$F_i = \frac{T}{d\left[\left(\frac{d_m}{2d}\right)\left(\frac{tg\,\beta+\mu sec\alpha}{1-\mu tg\beta sec\alpha}\right)+0{,}625\mu_c\right]} \tag{4}$$

The friction coefficient on threads and collars are within a range between 0.08 and 0.20, depending, overall, on the surface finish, thread precision and lubricants. (Junivall, 1983).

According to the works of Shigley (2006) and Faires (1976), the friction coefficient value, is on average 0.15 for both μ_c and μ.

An important note to be pointed out that will determine the experiment of the research is the selection of a single bolt torque to compare the experimental and analytic results. A method that is commonly used and present in most of the references is the use of ISO Standards with standards torque values for a given screw diameter, as seen on tab. 1

Table 1. Torque values for metric threads class 10.9 per ISO898 -1
Source: ISO 898-1 (2009)

Diameter	Pitch	Area	Proof Load	Torque k=0,18
mm	mm	mm²	N	Nm
5	0,8	14,2	11800	7,9
6	1	20,2	16700	13,5
8	1,25	36,6	30400	33
10	1,5	58	48100	65
12	1,75	84,3	70000	113
14	2	115	95500	180
16	2	157	130000	281
20	2,5	245	203000	547

3. METHODS AND RESULTS

3.1 Experiment description

The experience was done on a device capable of measuring the acting friction coefficient during the torque application on a selected bolt. Figure 4 represents part of the used device. The bolts used on this experiment was a class 10.9 metric M12 x 1.75 mm pitch.

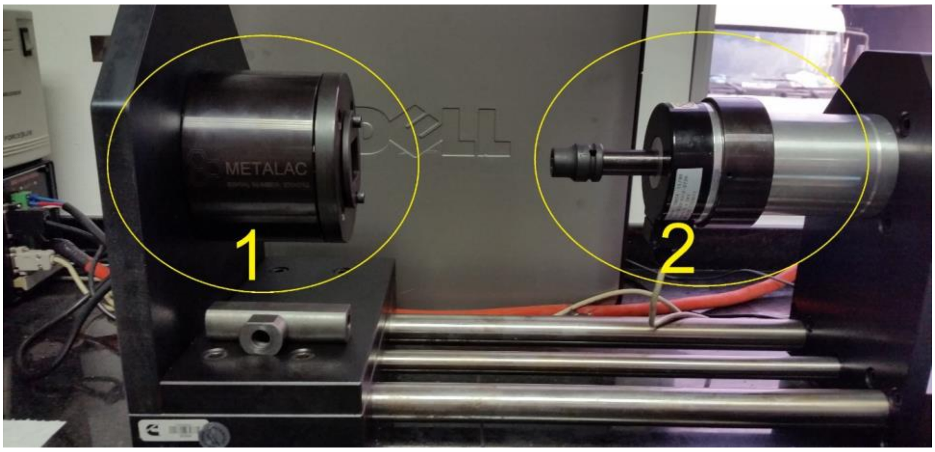

Figure 4. Device for friction coefficient measurement

The device on Fig. 4 is made by a Load Cell (Number 1 on Fig. 6) and a Stepper Motor (Number 2 on Fig. 6). Inside the load cell is assembled a standard thread where the chosen bolt will be tighten by the stepper motor.

Figure 5 shows the stepper motor tightening the bolt while strain gages measures the stresses on the screw's body, a computer connected to the device calculates the friction coefficients.

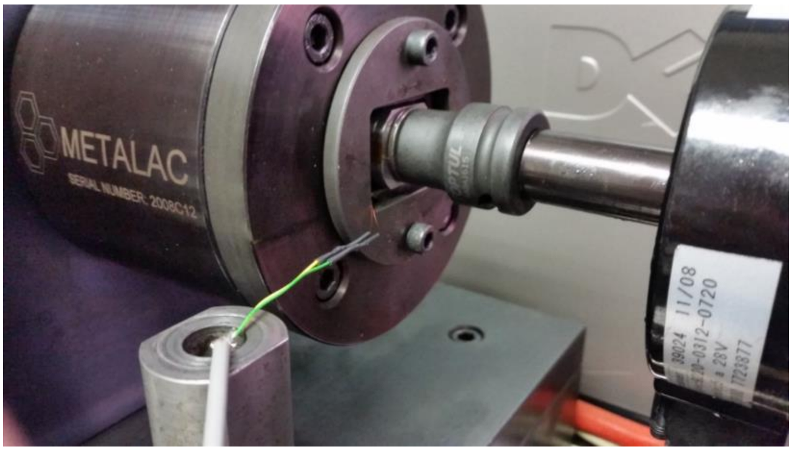

Figure 5. Device applying the torque to the bolt

3.2 Determing the maximum bolt's acting force

Prior to the experiment on the various bodies of proof a first test to determine and select the maximum force that the bolt can undergo while within its elastic limits needed to be made. This would help understanding the bolt's behavior and reactions to the torque being applied. Figure 6 shows a diagram of how the material behave until its failure.

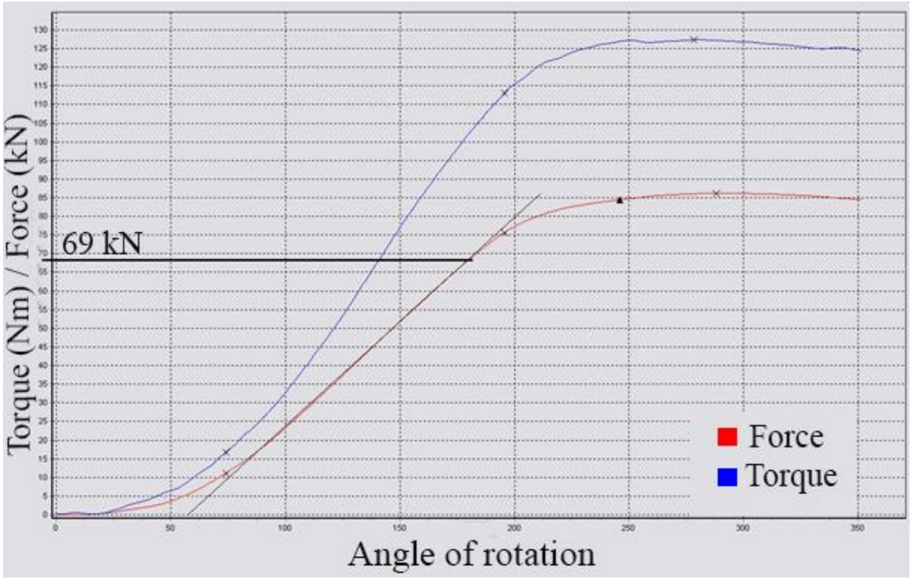

Figure 6. Diagram of the first test to determine the cap force for the experiment

The selected force that has been highlighted on Fig. 6 is of 69 kN, this is the force right before the plastic deformation of the bolt. If checked on tab. 1, for the 12mm diameter this is almost the same as the proof load per ISO 898-1.

3.3 Determing the experimental friction coefficients

Once the maximum force has been selected the device is programmed to apply a torque on the bolt until its force, read by the load cell, reaches 69kN, at this point the friction coefficients were saved into a worksheet to be verified after the end of experiment. This was repeated 12 times. Table 2 shows the average friction coefficient both for the collar and threads.

Table 2. Experimental friction coefficients

Force (kN)	Torque (Nm)	µ	µc
69,1	94,5	0,075	0,065

3.4 Verifying the analytical approach

With the force, torque and friction coefficient values determined by the experiment, it is possible to verify how close the analytical approach is to reality.

Table 3 shows the thread geometry characteristics.

Table 3. Geometry for metric bolt M12 x 1.75
Source: Shigley (2006)

d	dm	Pitch	Area	α	tg β	sec α
[mm]	[mm]	[mm]	[mm²]	[degrees]	-	-
12	10,8625	1,75	92,6722	30	0,05128	1,1547

By using the thread geometry found on Tab. 3 (Shigley, 2006) and the friction coefficient values obtained on Tab. 2 in Eq. 4 and comparing the computed force to the measured force, the result is shown on Tab. 4.

Table 4. Variance of experimental and analytical results

M12x1,75 Class 10.9	Fi [kN] (Experimental)	Torque [Nm]	Fi[kN] (Analytical)	Variance [%]
	69	94,5	76,2	**9,3**

The variance is close but below 10%, which proves that the equations used by the analytical approach are a good approximation to the reality. If used the average friction coefficient value given by the literature, the variance would go up to 43%. Therefore the importance of using the right friction coefficient is very high, with a 43% variance the parts could separate or fail during its duty cycle.

To show even more the influence of the friction coefficient, Tab. 5 shows the resulted comparison on the desired force and the computed force by varying the friction coefficient values by 0.01 starting by the numbers obtained during the experiment. This emulates the changes in materials, lubrication, and presence of dust or metallic particles in between the threads.

Table 5. Variance of experimental and analytical results when using different friction coefficient

Fi [kN] (experimental) = 69		T_{orque} [Nm] = 94,5		Variance
M12x1,75 Class 10.9	µ	µc	Fi [kN] (Analytical)	%
	0,085	0,075	68,1	1,3
	0,095	0,085	61,9	11,5
	0,105	0,095	56,7	21,7

4. CONCLUSION

The research was able to verify the theory by comparing its approach with the results obtained during the experiment.

It has been shown that the variance of the friction coefficient values has a huge impact on the force the bolt will be on, this impacts directly on the assembled parts performance, life, and risk of failure. The variance of almost 50% on the force computed using the average coefficient for bolt friction shows that this approach is not recommended to be used in any bolted joint. One of the most common failure modes found in the industry is by the simplification of the bolted joint development, so for a simple and first approximation this simplification could be used, but the designer must be aware that by using it the actual force may not be sufficient to hold the parts together.

Since it is impractical to measure the bolts elongation on an assembly line, for instance, the solution to ensure that the right force is being applied is by using torque-angle tightening. This approach measures the angle of deformation on the bolts body and its proportional to its elongation and the friction coefficient does not influence this.

Overall the objective of verifying the influence of the friction coefficient on the fastening torques and moments of the bolted joint by varying the materials to be joined has been achieved.

5. AKNOWLEDGEMENTS

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7. RESPONSIBILITY NOTICE