ELŐFESZÍTŐ ERŐ VÁLTOZÁSA CSAVARKÖTÉSBEN

BOLT PRELOAD VARIATION IN CYCLIC TIGHTENING

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ABSTRACT. This paper investigates the preload of a bolted link under repeated tightening/loosening cycles. The theoretical and the empirical torque-tension relationship is shortly presented. Experimental data of zinc-coated bolts are presented for two lubrication cases.

1. INTRODUCTION

The tightening process generates preload force in the bolted link. This force is essential to transmit power and to prevent the separation of the clamped components. The estimation of the preload is usually made in function of the bolt material's yield strength, which varies depending on manufacturers [1]. Too low or too high preload can lead to the failure of the joint, therefore, several methods have been introduced to control the bolt clamping force, for example, torque or angle control, bolt elongation control, and torquing control [2]. During tightening, approximately 10%-20% of the applied torque generates a tension in the bolt to hold clamped parts together, while the remaining part is lost to overcome friction on the thread and under the bolt head [3], [4]. Figure 1 illustrates how the torque is distributed.

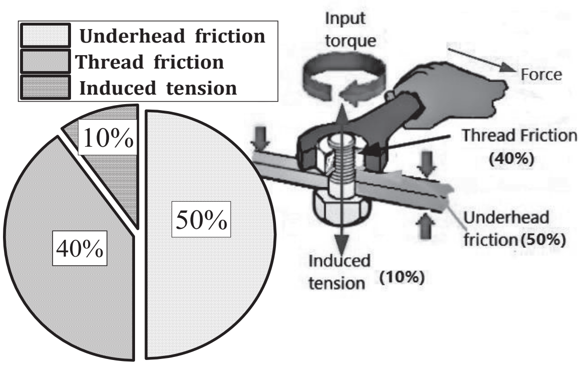

Figure 1. Tightening torque composition [4]

In the machine element theory, usual equations follow this triple division of the tightening torque. This idea was presented by Motosh in 1976 [5]:

$$T_{in} = F_P\left(\frac{P}{2\pi} + \frac{\mu_t r_t}{cos\beta} + \mu_n r_n\right) \tag{1}$$

as well as in the standard DIN 946/VDI 2230 [6]:

$$M_A = F_V\left(0.159P + 0.578d_2 \cdot \mu_G + \frac{D_{Km}}{2} \cdot \mu_K\right) \tag{2}$$

and in the standard ISO 16047 [7]:

$$T = F\left(\frac{1}{2} \cdot \frac{P + 1{,}154 \cdot \pi \cdot \mu_{th} \cdot d_2}{\pi - 1{,}154 \cdot \mu_{th} \cdot \frac{P}{d_2}} + \mu_b \cdot \frac{D_o + d_h}{4}\right) \tag{3}$$

These equations can be generalized as $T=F\cdot X$, where the constant X reflects the geometrical and frictional parameters of the joint. The problem with this structure is that it is very difficult to measure friction under the bolt head and on the thread. Moreover, the friction coefficients change during each tightening, and it is difficult to have a constant guess value.

However, it exists another kind of expression that relates the tightening torque T to the generated bolt clamping force F, in function of the bolt diameter D, and the factor K named "nut factor" (Torque coefficient ISO 16047 [7]) as follows :

$$T = K \cdot F \cdot D \tag{4}$$

Here, the nut factor K contains all thread geometry as well as friction-related constants that are either unknown or difficult to measure. ASME Standard PCC-1 [8] states that "K is an experimentally determined, dimensionless constant related to the coefficient of friction". Equation (4) has a simple form and is easy to use, as it contains standard measurable data.

In industrial applications, manufacturers often prescribe to use a new bolt and washer after disassembly for safety reasons. In other applications, dismantled fasteners are reused due to economical reasons, poor maintenance, or no fastener availability.

In this paper, we focus on the variation of the preload force in bolts used more than one time. We do not observe a variation of the friction at given places, but try to have an overall view of the phenomenon. For this reason, we realized experiments that simulate typical tightening-releasing processing on the same bolt. A typical example of this situation is the case of vehicle wheel bolts. They are regularly released then tightened, for example in case of winter-summer tire changes or brake pad repairs. If only winter-summer tire changes are considered, it takes ten tightening cycles over five years of service.

2. METHODOLOGY AND EXPERIMENTAL PROCEDURE

The investigation is made on zinc-coated bolts of size M6 and M8, with a mating nut. Figure 2 illustrates the experimental procedure. For each size, forty brand new bolts and nuts are used. For setting the friction parameters, two cases were considered: the as-is case and the oiled case. For the oiled case, the parts were cleaned using Loctite SF 7061. Then standard 15W-40 motor oil was applied at the threads and contact surfaces under the bolt heads, only before the first tightening, for twenty bolts in a given size. The remaining twenty bolts represented the as-received, out-of-the-box friction condition.

The measurement was realized as follows. The bolt head was fixed in a wise. A specific force measuring sensor was placed between two specific washers on the bolt, then a nut was tightened on the top (Figure 3). The tightening torque was applied with a torque wrench. An identical torque setting was used for all the bolts of identical diameter size.

The data acquisition system consisted of an HBM KMR+ bolt force sensor, an HBM Quantum X data collector device, and a computer. For each tightening, the peak bolt force was recorded, then the bolt was loosened. Tightening and loosening were forming one cycle, and this has been repeated 20 times for each bolt.

The technical information, the calculated parameter, and the assumed value of the coefficients of friction [1] for the tested bolt are summarized in Table 1. A total of 40 bolts were used for the experiment. Figure 4 presents the oiled samples of the two sizes, while Figure 5 illustrates the experimental setup.

Table 1. Bolt geometry specifications

Size		M6	M8
Grade	Bolt	8.8	8.8
	Nut	8	8
Thread pitch (mm)		1	1.25
Metric thread profile angle, β (°)		60	60
Tightening torque (N.m)		10	20
d_1 *(mm)*		5.35	7.188
d_2 *(mm)*		8.5	10.75
Assumed μ_T *and* μ_H		0.1	0.1
Computed angle ρ' (°)		6.587	6.587
Thread lead angle α *(°)*		3.037	2.847

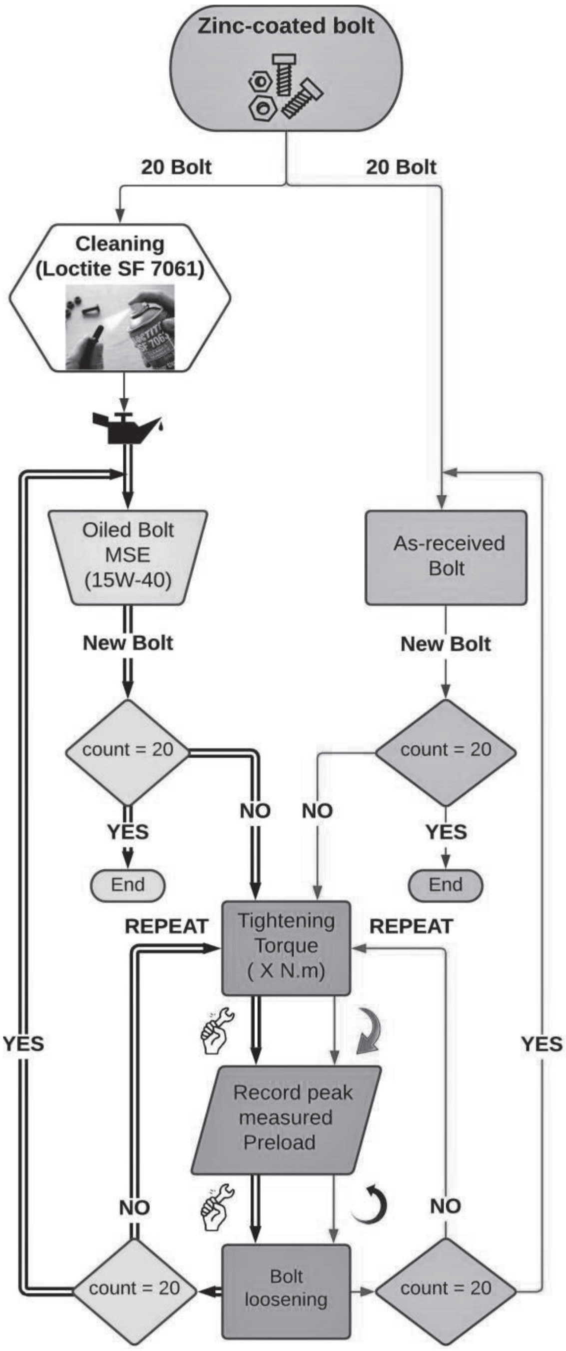

Figure 2. Experiment process flowchart

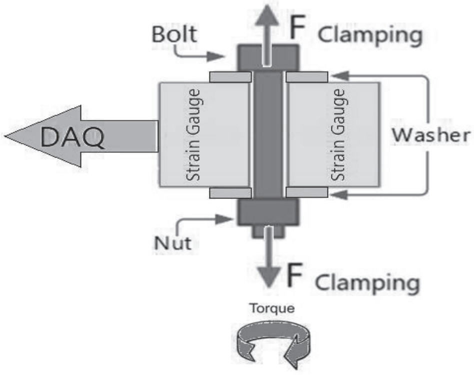

Figure 3. Configuration of the tested bolted joint

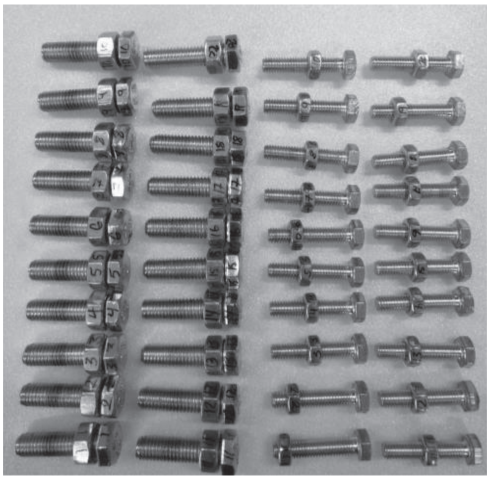

Figure 4. Sample of the tested bolts

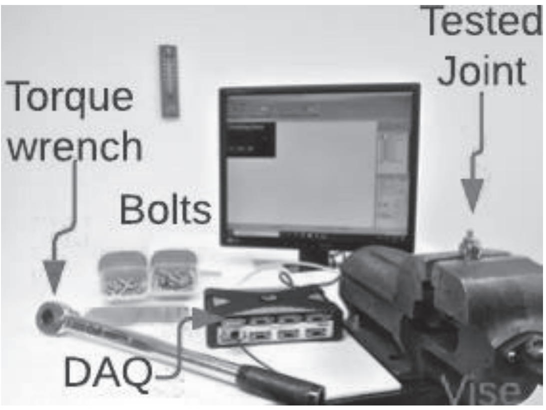

Figure 5. Experimental setup

3. RESULTS

3.1. Preload force

Table 2 shows the mean of the measured preload value for the first tightening compared to the theoretically calculated one, using the selected coefficient of friction for both threads and the under the nut contact surface.

In the case of the as-is M6 bolt, the measured bolt force is only half of that computed. For the oiled case, measured and computed values show a good correlation. There was no such problem in the case of M8 bolts. Generally, the measured force values were higher than the theoretical value.

Table 2. Calculated preload and mean of the measured preload for the first cycle.

	Theoretical preload (kN)			Measured preload First tightening (kN)	
Size	eq(1)	eq(2)	eq(3)	Received	Oiled
M6	11.19	11.19	11.15	6.45	11.67
M8	17.36	17.36	17.31	17.78	18.61

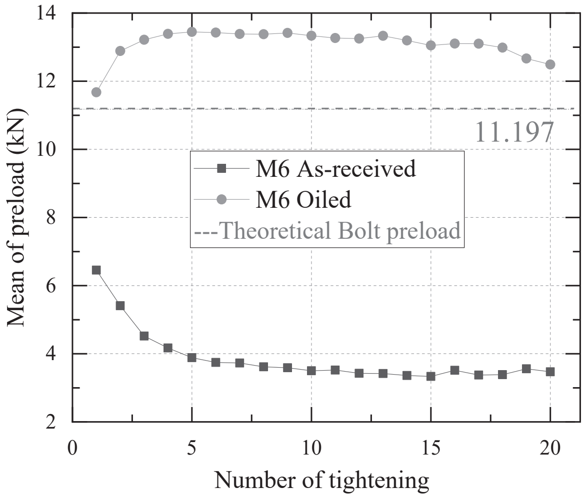

Figure 6. Effect of repeated tightening on the M6 zinc-coated bolt

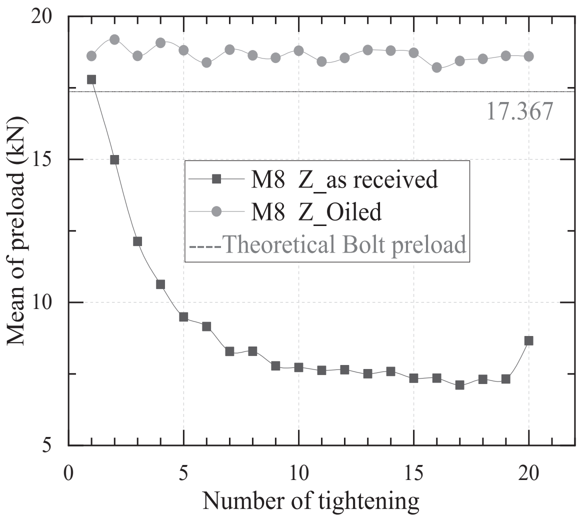

Figure 7. Effect of repeated tightening of the M8 zinc-coated bolt

Figure 6 and Figure 7 show the mean bolt preload force value from twenty tightenings. Blue lines show the as-received surface condition, while grey lines show the oiled surface condition. For the as-received case, we can see that there is a kind of wear-in period in the first 5-6 tightenings, where the successive preload forces decrease quickly. The magnitude decrease is 39,81% for the M6 size and 46,64% for the M8 size. During the next tightenings, the preload force decrease continues, but at a much smaller slope.

In the lubricated case, we can still see a wear-in for the M6 case. Here, the preload force increases slightly during the first 5 tightenings. The magnitude increase is 15,16%. In the case of the M8 size, the preload force oscillates with a small amplitude around a constant value.

3.2. Nut factor

From the measured data, a nut factor *K* has been computed both for M6 and M8 sizes, upon the rearranged equation (4):

$$K = \frac{T_{tightening}}{F_{measured} \cdot D_{nominal}} \quad (5)$$

Here the tightening torque was constant for each bolt size. Figure 8 shows the computed values of the nut factor. We can observe that in the lubricated case, the nut factor is almost identical and constant, with a mean value of 0,15, regardless of the bolt size and the tightening torque.

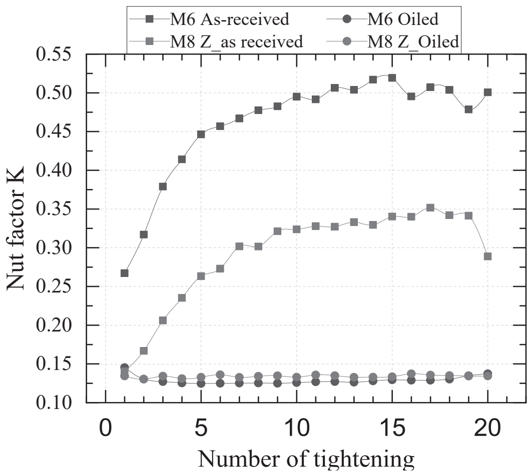

Figure 8. Calculated nut factor during repeated tightenings for M6 and M8 bolts

We can also observe, that in the as-received case, though the curve shapes are similar, the numerical values are different. Three zones can be recognized on the curves. During the first five tightenings, the friction, thus the nut factor increases quickly. From five to fifteen tightening, there is still an increase, but it is much smaller. Finally, after fifteen tightenings, the nut factor values seem to stabilize, but the trend is not clear. The maximum value is 0,519 for the M6 size and 0,352 for the M8 size.

Note that in the as-received case, the formation of a small amount of coating metal powder has been observed on the mating surfaces during the successive tightenings. This powder was not removed between the successive tightenings.

4. CONCLUSION

Measurements have been made to define how the preload force changes in a bolted link during successive tightenings and releases. It has been presented that the measured values differ from those given by the theory in the case of M6 bolts. Due to uncertainties in the definition of friction, the nut factor has been computed. It has been shown that oil lubrication helps to keep the preload force at a constant value both for M6 and M8 size. It has also been shown that in the as-received case the preload force decreases strongly. This decrease must be considered in industrial applications.

5. REFERENCES

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