A KENÉS HATÁSA CSAVARKÖTÉS CIKLIKUS MEGHÚZÁSA ÉS LAZÍTÁSA ESETÉN

EFFECT OF LUBRICATION DURING CYCLIC TIGHTENING ON THE BOLT PRELOAD FORCE

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ABSTRACT This paper investigates how repeated tightening and loosening affects the generated preload in a bolted joint. Experiments were conducted on a zinc-coated bolt with three different diameter sizes. The torque tension relationship was compared experimentally of four lubrication types based on the torque coefficient factor.

1. INTRODUCTION

Threaded fasteners are widely used in many industrial applications due to their advantages. In a mechanical structure, the bolted joint is a typical fastener used to keep two or more mechanical elements together. This semi-permanent connection method is widely used, since it efficiently facilitates assembly/disassembly functionality at a low cost [1], [2]. When the bolt is tightened, the effect of the thread helical geometry generates tension in the bolt. This tension is called the preload force. The primary function of this tension is to prevent the separation of the joint components.

During the design stage, estimating a proper value of the preload force is a critical task to prevent joint failure. For a typical joint shown in Figure 1, Motosh [3] in his approach proposed that the input torque to the turning head has three components:

$$T_{input} = T_{Pitch} + T_{Underhead} + T_{Thread} \tag{1}$$

These components are equivalent to the following:

$$T_{Pitch} = F_{clamping}\left(\frac{P}{2\pi}\right) \tag{2}$$

$$T_{Underhead} = F_{clamping}\left(\frac{\mu D}{2}\right)_{underhead} \tag{3}$$

$$T_{Thread} = F_{clamping}\left(\mu_{Thread}\frac{D_{Thread}}{2\cos\beta}\right) \tag{4}$$

In this expression the losses from the input torque T_{input} are to overcome two frictional torque components: the torque under the joint turning head $T_{Underhead}$ and the torque at the threads level T_{Thread}. The remaining component T_{Pitch} is the torque responsible for generating the tension by stretching the bolt. The approximation of the three components are 50%, 40%, and 10%, respectively [2]

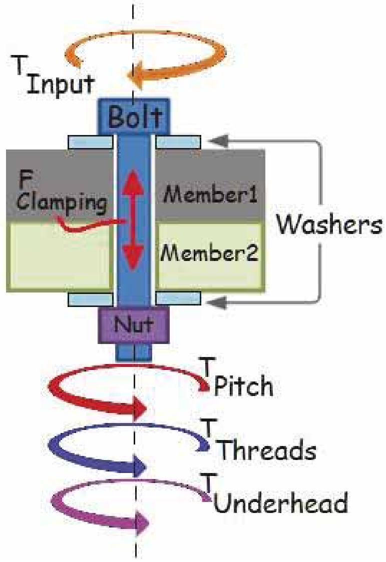

Figure 1. Typical bolted joint

Another representation for the torque-tension relationship, according to the DIN EN ISO 16047 [4] standard, can be given in equations (5) and (6).

$$T = F\left(\frac{1}{2}\cdot\frac{P+1{,}154\cdot\pi\cdot\mu_{th}\cdot d_2}{\pi-1{,}154\cdot\mu_{th}\cdot\frac{P}{d_2}} + \mu_b\cdot\frac{D_o+d_h}{4}\right) \tag{5}$$

$$T = K\cdot F\cdot D \tag{6}$$

Equations (7) and (5) are derived based on the bolted joint geometrical and frictional parameters. On the other hand, Equation (6) is a short-form of the torque-preload relationship. It is empirically derived based on the bolt nominal diameter *D* and the experimentally measured *torque coefficient K*, also called the "*nut factor*". Here, *K* is a dimensionless constant that combines the influence of all the variables that affect the preload (F), even those not defined or complicated to quantify.

The reasonable preload force range depends on selecting a proper input torque value based on the bolt yield strength. To increase the bolting performance, it is essential to avoid undertightening and overtightening that eventually lead to joint failure, e.g., self-loosening and fatigue. Moreover, for safety reasons, bolt manufacturers prescribe not to reuse the bolt/nut once they are dismantled

This paper investigates the generated preload's behavior in a bolted joint when the same bolt is repeatedly tightened, loosened, and retightened using the same input torque. The test was made under different lubrication conditions. The investigation is considered as a simulation for the case when the fasteners are reused in some industrial applications.

2. EXPERIMENTAL SET-UP METHOD AND PRELOAD MEASUREMENT

The tests were conducted on bolt-nut pairs with a zinc-coated surface finish. Three distinct sizes: M6, M8, and M10, were used with a mating nut. Four lubrication cases were considered to simulate the impact of the joint contact surface quality. The preparation of the cases is as follows. The first one is the as-is, representing the out-of-the-box state, with no operation on the surface. For the remaining three cases, the bolts and nuts are cleaned using Loctite SF 7061 to have a surface free of contamination. The second, dry case is the cleaned and not lubricated one. Two types of lubricants were added to the cleaned surfaces, representing the third and fourth cases as follows: solid molybdenum disulphide powder (MoS_2) and engine motor oil (15W-40). Twenty new bolts/nuts are assigned for each lubrication case for each bolt size.

The lubrication was added to the joint contact surfaces: at the threads and under the turning head (nut) only before the first tightening. A thin layer of the solid MoS_2 powder was applied for the third case, and a few drops of 15W-40 oil were applied for the last case. The measurements and the data collection were realized similarly to that described in a previous paper [5].

Two torque control methods were used. A torque wrench was used for tightening the nut to a specific torque based on the bolt size. After the tightening was complete, a strain gauge with a data acquisition system was used to measure and record the peak force generated in the bolt; then, the nut was released. This process forms one cycle, and this cycle is repeated ten times for each bolt under the same torque value. A total of 240 new bolts/nuts were used in the experiments. Table 1 summarizes the geometrical and technical information and the calculated parameters for the tested bolts.

Table 1. Tested bolt specifications

Size		M6	M8	M10
Torque (N.m)		10	20	40
d_1(mm)		5.188	7.188	9.188
d_2(mm)		9.75	10.75	11.75
Metric thread profile angle, β (°)		60	60	60
computed friction angle ρ' (°)		6.587		
Thread lead angle α (°)		4.386	3.168	3.168
Thread pitch (mm)		1.25	1.25	1.25
grade	Bolt	8.8		
	Nut	8		

3. RESULTS

3.1. Generated preload

The lubrication condition influences the initially generated preload force, as shown in Figure 2. Each bar in the diagram is labeled with the preload mean of twenty measurements. For the three different bolt sizes, the torque conversion to preload is enhanced when lubrication is added compared to as-is and dry cases. The initial preload is the lowest for the dry case. This means when the contact surface is free of any kind of lubrication, more torque is consumed to overcome friction. The highest preload mean based on the bolt size was for the following lubrication: oiled and MoS_2 are almost the same for the M6, MoS_2 for the M8, and oiled for the M10.

Figure 3 shows how the generated preload behaves during the tightening/unscrewing cycles. The following can be concluded:

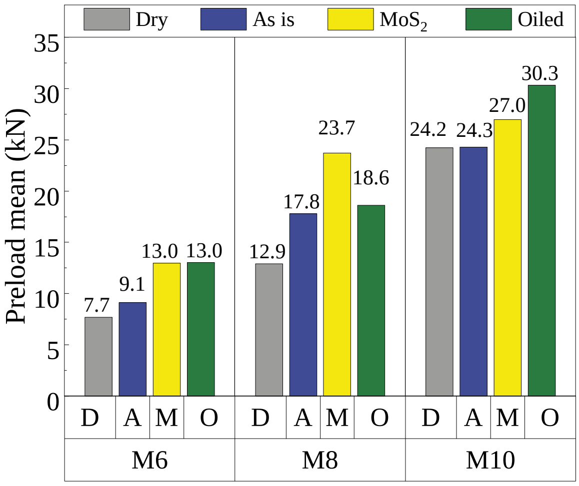

Figure 2. Initially generated preload force for the three diameters under different lubrication cases

1. Among different lubrications for each bolt size, the lowest bolting performance was for the dry case.
2. The generated preload for the as-is case is slightly better than the dry one. In contrast to the M10 size, they are almost identical, which can be related to the surface state of the out-of-the-box was clean surface (no contamination).
3. For the MoS_2 lubrication, the preload is enhanced. However, it keeps decreasing as a function of tightening numbers, in a similar trend to the as-is and dry lubrication.
4. Applying oil film gives the best bolting performance; the generated preload during the repetition for each size is almost stabilized around the first preload value.
5. For all diameters, there was a kind of surface wear-in period in the first 5-6 tightenings, where the successive preload forces decreased quickly. Then the preload force oscillated with a small amplitude around a constant value.

3.2. Nut factor K

To examine the relationship between the input torque and generated preload, the short-form equation (6) was used. The nut factor was calculated using the rearranged equation (7). The nut factor K was calculated for every tightening measurement. After that, the nut factor mean was computed for each bolt size under the four lubrication cases for each replication using equation (8).

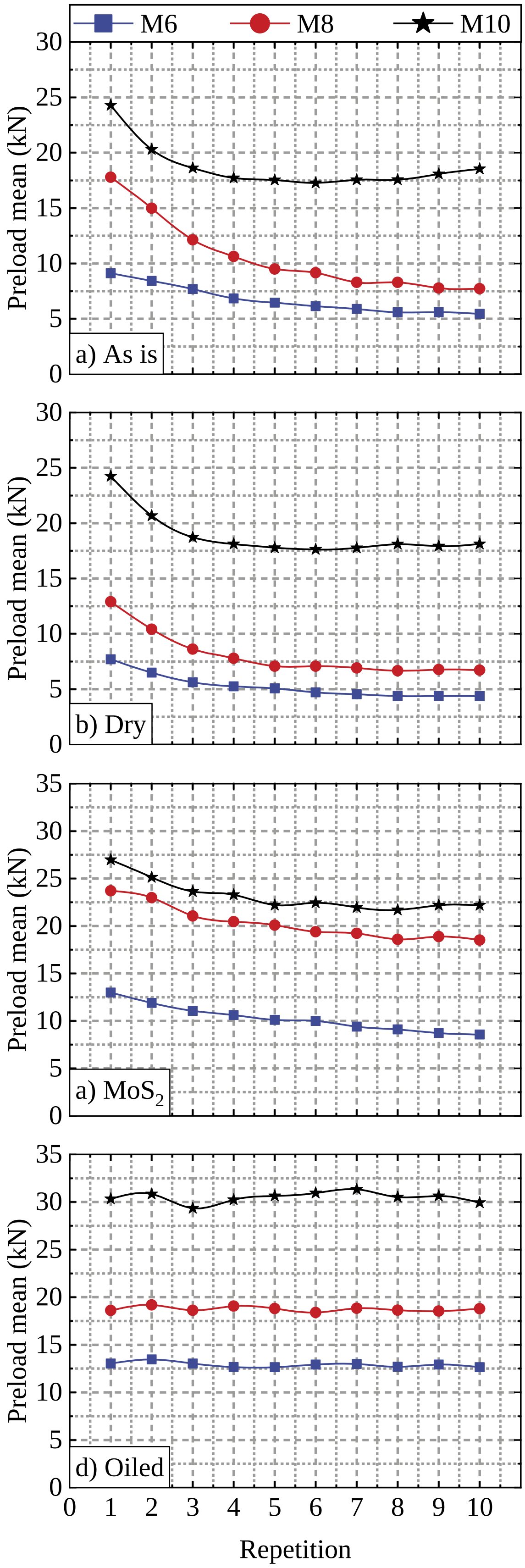

Figure 3. Preload variation during repeated tightening under different lubrication conditions

$$K = \frac{T_{input_D}}{D * F_{Measured}} \quad (7)$$

$$K_{Mean_R} = \frac{\sum_{B=1}^{N} \frac{T_{input_D}}{D * F_{Measured_B}}}{N} \quad (8)$$

In the equations, K is the nut factor. $K_{Mean\ R}$ represents the nut factor mean indexed by repetition *R=1,2, 3 ... 10. B* is the bolt number, *N=20* is the total number of tested bolts for each case, and D is the bolt nominal diameter. $T_{input,\ D}$ indicates the input tightening torque for the bolt nominal diameter, and $F_{Measured\ B}$ is the experimentally measured preload. In this investigation, the input torque is constant for each bolt size. It is not changed following the lubrication condition. Here, the measured preload and the calculated nut factor are inversely related; a higher nut factor indicates poor bolting performance and vice versa.

The range of the calculated nut factor during the ten repetitions is plotted as a box plot in Figure 4. The nut factor mean is grouped by different lubrication conditions for comparison purposes. The following remark can be made:

1. In the case of as-is lubrication, approximately 50% of the nut factor value is within (0.21-.31) for the size M6 and M8, while for the M10, it is (0.21-0.23).
2. After cleaning joint contact surfaces (dry case), the data spread is similar to the as-is for the M10, with a higher nut factor value such that 75% lies within (0.3-0.4) for M6 and M8
3. When the MoS_2 was used, the nut factor range is reduced (0.16- 0.20) for the M8 and M10, while the M6 was between (0.12-0.135).
4. For the oiled case, the stability of the tightening process during repetition can be seen for the three different bolt sizes; the nut factor data spread is minimal among other lubrications cases, K almost 0.1 for M6, and 0.135 for M8 and M10,

4. CONCLUSION

The nut factor allows us to experimentally compare the outcomes of the tightening process without increasing the experiment's complexity. This method was used in this work to examine the effect of different joint contact surface states on the bolt preload under cyclic tightening. The type and presence of lubrication film at the contact surfaces matter for enhancing the surface contact quality. This stabilizes the preload during the repeated tightening /release cycles. Before reusing a bolt/nut in the assembly, it is important to apply lubrication to minimize input torque losses.

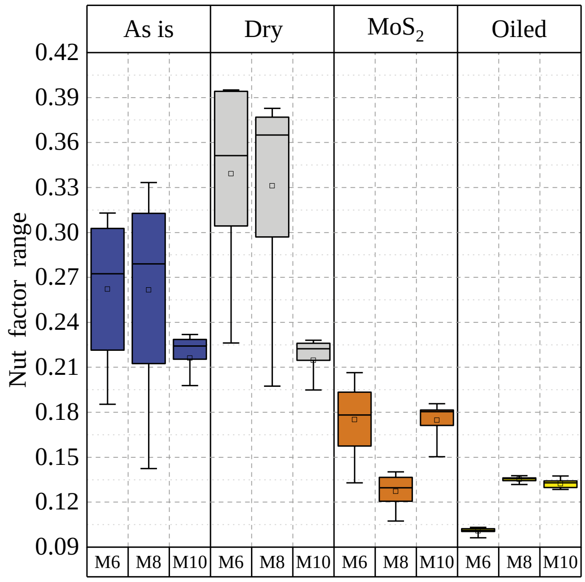

Figure 4 Nut factor range through the ten tightening repetitions

5. REFERENCES

[1] H. GONG, J. LIU, and H. FENG, “Review on anti-loosening methods for threaded fasteners,” *Chinese J. Aeronaut.*, vol. 35, no. 2, pp. 47–61, 2022, DOI: 10.1016/J.CJA.2020.12.038.

[2] J. H. Bickford, *An Introduction to the Design and Behaviour of Bolted Joints: Non-Gasketed Joints*, Fourth Edi. New York, NY, USA: CRC Press, Taylor & Francis Group, 2007.

[3] N. Motosh, “Development of design charts for bolts preloaded up to the plastic range,” *J. Manuf. Sci. Eng. Trans. ASME*, vol. 98, no. 3, pp. 849–851, 1976, DOI: 10.1115/1.3439041.

[4] ISO 16047, “Verbindungselemente – Drehmoment/Vorspannkraft-Versuch (ISO 16047:2005); Deutsche Fassung EN ISO 16047:2005.,” *Beuth Verlag Berlin, Ger.*, 2005.

[5] T. Alsardia and L. Lovas, “BOLT PRELOAD VARIATION IN CYCLIC TIGHTENING,” *GÉP*, vol. LXXII, no. 3–4, pp. 9–12, 2021.