IOP Conference Series: Materials Science and Engineering

PAPER • **OPEN ACCESS**

Suitable pitch difference to realize anti-loosening performance for various bolts-nuts diameter

To cite this article: S Kubo *et al* 2018 *IOP Conf. Ser.: Mater. Sci. Eng.* **372** 012007

IOP Conf. Series: Materials Science and Engineering **372** (2018) 012007 doi:10.1088/1757-899X/372/1/012007

Suitable pitch difference to realize anti-loosening performance for various bolts-nuts diameter

S Kubo[1], K Tateishi[1], N-A Noda[1,2], Y Sano[1], Y Takase[1] and K Honda[1]

[1]Mechanical Engineering Department, Kyushu Institute of Technology, 1-1 Sensui-cho Tobata-ku, Kitakyushu-shi, 804-8550, Japan

E-mail: noda@mech.kyutech.ac.jp

Abstract. In bolt-nut connection, the anti-loosening performance and high fatigue strength are always required with low cost to ensure the connected structure's safety. In the previous study, a suitable pitch difference between the bolt-nut was obtained as $\alpha = 33\ \mu m$ for M16 JIS bolt-nut through loosening experiment and FEM simulation for tightening process. However, other bolt-nut diameters have not been considered yet. In this paper, therefore, suitable pitch difference is considered for various diameters to realize anti-loosening performance. Since bolt-nut thread geometries are different depending on the diameter, they are expressed as approximate formula. Then, loosening force and anti-loosening force are considered by varying the diameter. Finally, suitable pitch difference $\alpha_{min}^{suit} < \alpha < \alpha_{max}^{suit}$ was determined from mechanical condition.

1. Introduction

In a wide industrial field, bolt-nut joint is unitized as a machine element and is of great importance. However, the anti-loosening and anti-damage performances are always required. When vibration is added, the rotational loosening force on the spiral shape of the screw becomes larger than the friction between the screw surfaces, then the loosening occurs. Most previous studies have been mainly focusing on anti-loosening performance [1-4]. Only a few studies contribute toward improving fatigue strength [5-7]. To improve both the fatigue life and the performance of anti-loosening, a slight pitch difference between bolt and nut was previously proposed [8-12]. Then, the anti-loosening performance was confirmed for M16 bolt-nut connection by using FEM simulation as well as experiment [13-15]. However, other bolt-nut diameters have not been considered yet. Therefore, in this paper, the anti-loosening will be studied for different bolt-nut diameter to clarify the suitable pitch difference theoretically and experimentally. This special nut may improve the fatigue strength [13-15] without using additional components. Moreover, this special nut can be manufactured just by changing feed speed of a numerical control (NC) lathe.

2. Effect of the pitch difference on the rotation

2.1. Bolt-nut specimens

Japanese Industrial Standard (JIS) M16 bolt-nut connections with strength grade 8.8 are employed. The bolt material is chromium- molybdenum steel SCM435, and the nut material is medium carbon steel S45C quenched and tempered, whose properties are indicated in table 1, and whose stress–strain curves are shown in figure 1, respectively. The normal M16 bolt and nut have the same pitch dimension as 2000 μm. Herein, the nut pitch is assumed to be (2000+α) μm. Figure 2 shows a contact status between bolt and nut threads during the tightening process. Notation $\boldsymbol{C_x}$ is the horizontal clearances between bolt and nut.

Table 1. Material properties of bolt and nut.

	Young's modulus (GPa)	Poison's ratio	Yield strength (MPa)	Tensile strength (MPa)
SCM435 (Bolt)	206	0.3	800	1200
S45C (Nut)	206	0.3	530	980

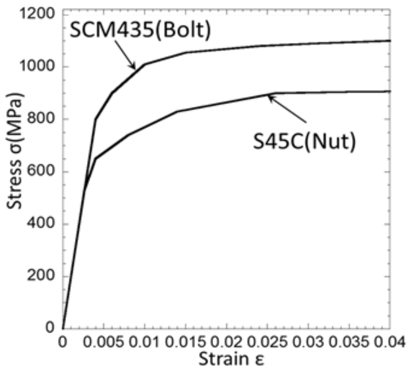

Figure 1. Stress strain relation for SCM435 and S45C.

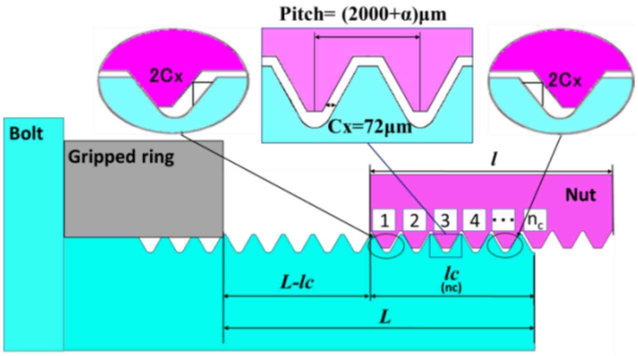

Figure 2. Schematic diagram of bolted joint.

2.2. Occurrence condition of Tp

Notation l_c is defined as the distance where the bolt-nut threads contact takes place, which can be obtained geometrically as shown in equation (1)

$$(n_c - 1)\alpha = 2C_x$$

$$C_x = \frac{C_y}{\tan\theta}, \qquad l_c = n_c p \tag{1}$$

In equation (1), p is the pitch of bolt (for example, p=2 mm for M16), α is the pitch difference, n_c is the number of nut thread in contact in figure 2 except for n=1, θ is the thread angle (=60°) and C_x and C_y are the horizontal and vertical clearances between bolt and nut threads. Note that equation (1) is valid when n_c < 8 which is the total nut threads number 8 and the nut length < 16 mm.

2.3. Pitch difference α vs prevailing torque

To obtain the relationship between torque and clamping force, the torque was controlled by using an electric torque wrench, and the clamping force was measured by using the strain gauge attached to the clamped body surface as shown in figure 3. The uniaxial strain gauge with a length of 2 mm KFG-2 (Kyowa Electronic Instruments Co., Ltd.) was used in this measurement. Before the experiments, calibration tests were performed by compressing the clamped body to obtain the relationship between the clamping force and surface strain as shown in figure 3. Similar tests were performed to calibrate the torque wrench as shown in figure 4.

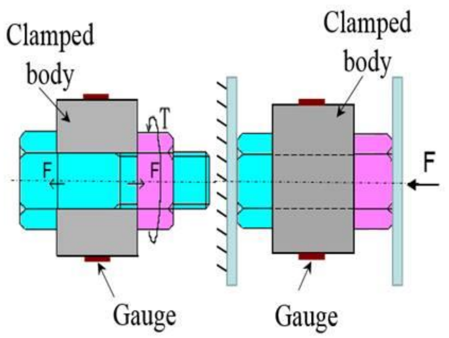

Figure 3. Calibration method for bolt axial force measurement.

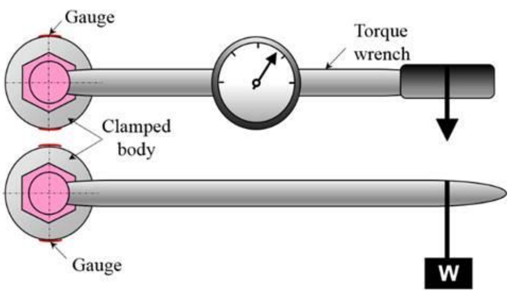

Figure 4. calibration method for torque wrench.

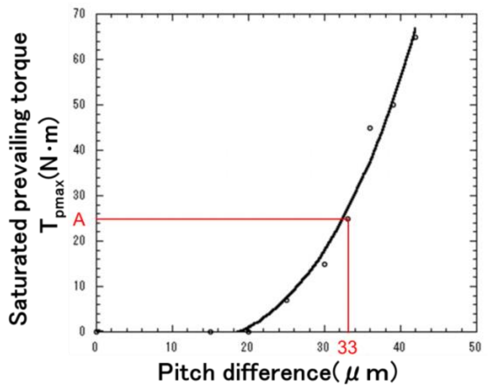

Figure 5. Relationship between torque and Pitch difference (M16).

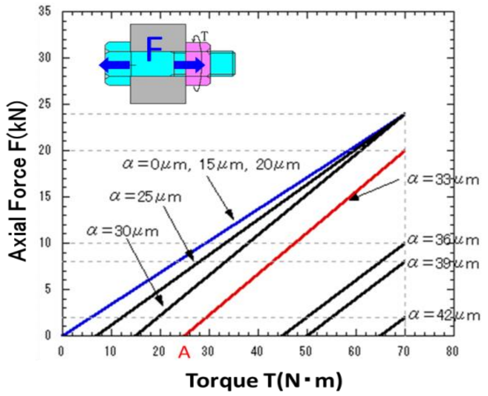

Figure 6. Relationship between torque and axial force (M16).

Figure 5 shows the prevailing torque T_p when the nut rotation is large enough by varying the pitch difference. From figure 5, it is seen that saturated prevailing torque T_p is nearly proportional to the pitch difference α in the range of $15\ \mu\text{m} < \alpha < 42\ \mu\text{m}$. Figure 6 shows the tightening torque vs clamping force experimentally obtained. When $\alpha = 0{\sim}30\ \mu\text{m}$, the torque–clamping force relationship is the same as the one of α = 0 μm. When $\alpha = 33\ \mu\text{m}$, the prevailing torque of 25 N·m is required before the nut touches the clamped plate. Under the same tightening torque of 70 N·m, the clamping force was reduced to 20 kN. When α = 39 μm, under a torque of 70 N·m the axial force decreased significantly to 8 kN, which was only 1/3 of the axial force of α=0 μm.

3. Loosening experiment

3.1. Device and conditions

The bolt-axial force 24 kN was considered for the standard bolt–nut, and the corresponding tightening torque was 70 N·m. It is tighten ring that insert to bolt at 70 N·m for nut. And experiment machine is built up. Also an extra length is 10mm. As shown in figure 7, the experimental device was an impact-vibration testing machine based on NAS3350 (National Aerospace Standard), whose vibration frequency was about 30 Hz, and vibration acceleration is 20 g. The maximum vibration cycle of NAS3350 is 30 000, therefore, if the number of vibration cycles was over 30 000, the anti-loosening performance may be considered to be good enough.

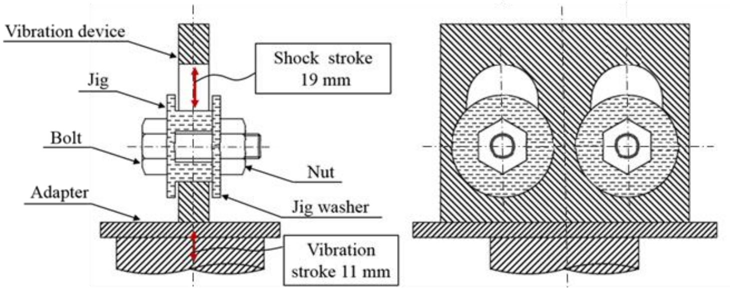

Figure 7. Loosening experimental device based on NAS3350.

3.2. Results

Table 2 lists the number of cycles for the start loosening and the nut dropping. Table 2 also lists the prevailing torque measured in the loosening experiments and the bolt axial forces estimated from figure 6. For each pitch difference α, two specimens were tested. For α=0 μm and $\alpha = 15$ μm, the nuts dropped at about 1000 cycles. For $\alpha = 33$ μm, no loosening was observed until 30 000 cycles and prevailing torque was 25 N·m. For $\alpha = 42$ μm, no loosening was observed until 30 000 cycles although the axial force was estimated to be only1~4 kN. It maybe concluded that if α is too small, the anti-loosening cannot be expected and if α is too large, the clamping ability is not good enough. By considering both the anti-loosening and clamping abilities, α=33~42 μm can be selected as the most suitable pitch difference.

Table 2. Anti-loosening Performance (M16).

Pitch difference α(μm)	Cycles for dropping	Prevailing torque (N·m)	Axial force(kN)
0	751	0	24
	876		
15	813	0	24
	1528		
33	30000	25	14~24
	30000		
42	30000	67	1~4
	30000	57	

4. Suitable pitch difference for various bolt-nut diameter

4.1. How to estimate suitable pitch difference

In this section, suitable pitch difference will be considered for various thread nominal diameters. First bolt-nut geometries are different depending on the diameter. Figure 8 shows JIS clearance C_x between the bolt and nut threads by varying nominal thread diameter d. It is seem that the clearance C_x can be expressed as equation (2)

$$C_x = 1.4882d + 41.239 \quad (2)$$

Similarly, figure 9 shows JIS nut thickness h by varying nominal thread diameter d. In this study, we have to estimate loosening force F_l which is closely related to the volume of the nut. Then, h is an important dimension. It is seen that nut thickness can be expressed as equation (3).

$$h = 0.3099d + 0.4253 \quad (3)$$

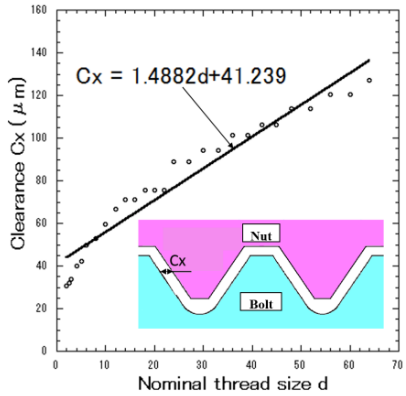

Figure 8. Relationship between Nominal thread size d and Clearance C_x.

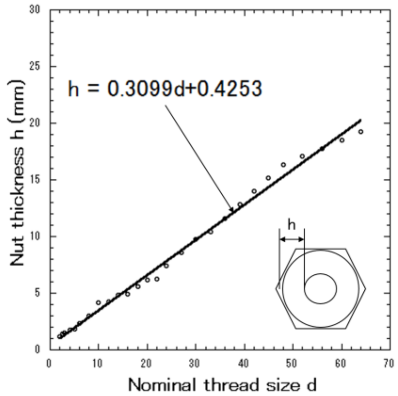

Figure 9. Relationship between Nut size d and Nut thickness h.

In the following discussion, loosening force F_l and anti-loosening force F_a are assumed. Figure 10 shows the relationship between F_l and F_a. On the one hand, loosening force F_l can be proportional as equation (4) since F_l should be proportional to the nut weight.

$$F_l \propto \gamma V \cong \gamma(\pi d)hl \tag{4}$$

Here, γ= specific gravity, V= the volume of nut, l= the length of nut.

On the other hand, anti-loosening force F_a should be proportional to be contact surface area between the nut and bolt threads.

$$F_a \propto \tau_s(\pi d)l_r \tag{5}$$

In equation (6) τ_s = average shear stress between the nut and bolt threads, and l_r is the length of the nut threads in contact in figure 10. Therefore, loosening condition can be expressed as equation (6) assuming loosening and anti-loosening forces should be balanced.

$$F_l = KF_a \tag{6}$$

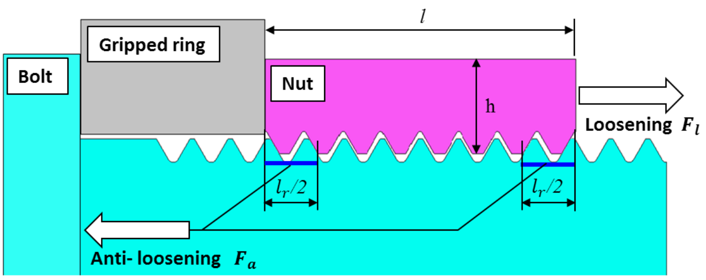

Figure 10. Relationship between loosening force F_l and anti-loosening force F_a.

Therefore, we have

$$\gamma(\pi d)hl = K\tau_s(\pi d)l_r, = \frac{\gamma hl}{\tau_s l_r} = \frac{\gamma}{\tau_s}K' \text{ , } K' = \frac{lh}{l_r} \tag{7}$$

Here, K and $K'$are proportional constants. From figure 2 and equation (1) by using equation (7), the number of nut thread in contact in figure 2 will be expressed since n_c is important for anti-loosening.

$$l_r = \frac{lh}{K'}$$

From figure 2 with equation (1)

$$l_r = l - l_c = l - n_c p \tag{8}$$

$$n_c = n\left(1 - \frac{h}{K'}\right) \tag{9}$$

From table 2, suitable prevailing torque range to prevent loosening can be expressly as 30<T_p<60 for M16. Similarly, suitable pitch difference can be expressed as $\alpha_{min}^{suit} = 34\mu m < \alpha^{suit} < \alpha_{max}^{suit} = 41\ \mu m$. From equation (1), we have $\alpha = 2C_x/(n_c - 1)$. Assume $n_{c\,max}$ is the number of nut threads for α_{min}^{sut} and $n_{c\,min}$ is the number of nut threads for α_{max}^{sut}. Then,

$$n_{c\,max} = n\left(1 - \frac{h}{K'_{max}}\right) \tag{10}$$

$$n_{c\,min} = n\left(1 - \frac{h}{K'_{min}}\right) \tag{11}$$

Here, K'_{max} is the proportional constant for α_{min}^{sut} and K'_{min} is the proportional constant for α_{max}^{sut}. Therefore, generally, the suitable α can be expressed as

$$\frac{2C_x}{n_{c\max} - 1} < \alpha < \frac{2C_x}{n_{c\min} - 1} \tag{12}$$

By substituting equation (8) into (9),

$$\frac{2Cx}{n\left(1 - \frac{h}{K'_{max}}\right) - 1} < \alpha < \frac{2Cx}{n\left(1 - \frac{h}{K'_{min}}\right) - 1} \tag{13}$$

Since the values of the proportional constants K and K' are unknown, in this study, the loosening experimental results for M16 in table 2 is used. From $\alpha_{min}^{suit} = 34\ \mu m < \alpha^{suit} < \alpha_{max}^{suit} = 41\ \mu m$ for M16 with$C_x = 72\ \mu\ m$, equation (14) is obtained by using equation (1).

$$4.5 \leq n_c \leq 5.2 \tag{14}$$

By substituting $n_{c\,max}$ and $n_{c\,min}$ into (8), we have

$$5.6 \leq l_r \leq 7 \tag{15}$$

Here, For M16 bolt–nut connection, substituting by $l = 16\ mm, h = 5\ mm$(M16), equations (2) and (3).

$$K'_{min} = 11.4 \leq K' \leq K'_{max} = 14.3 \tag{16}$$

Assumed this K'_{max} and K'_{min} can be for various nominal thread diameter. By using equation (2) and equation (3) into (13)

$$\frac{2.9764d + 82.478}{\mathrm{n}\left(1 - \frac{0.3099\mathrm{d} + 0.4253}{14.3}\right) - 1} < \alpha < \frac{2.9764d + 82.478}{\mathrm{n}\left(1 - \frac{0.3099\mathrm{d} + 0.4253}{11.4}\right) - 1} \tag{17}$$

Table 3. Suitable pitch difference estimated for M12~M16.

Diameter	Pitch p(mm)	Nut length l(mm)	Number of threads n	Nut thickness h(mm)	Clearance C_x(μm)	Suitable Pitch difference range α(μm)
M12	1.75	10	5.7	4.1	59.1	38.6<α<44.8
M14	2	11	5.5	4.8	62.1	46.5<α<56.4
M16	2	13	6.5	5.4	65.1	42.6<α<53.5
	2	16	8	5.4	65.1	32.6<α<40.4

4.2. Validity of equation (17)

Suitable pitch difference can be estimated by using equation (17). Table 3 shows suitable pitch difference obtained from equation (17) for M12~M16 bolt-nut connection. As an example, for M12 the suitable α can be expressed as $38.6\ \mu m < \alpha < 44.8\ \mu m$. By using Junker vibration test, we have confirmed that $\alpha = 40\ \mu m$ have no loosening equation (17) is valid. Therefore, the validity of equation (17) is confirmed for M12.

5. Conclusion

In the previous study, a suitable pitch difference between the bolt-nut was obtained as $\alpha = 33\ \mu m$ for M16 JIS bolt-nut through loosening experiment and FEM simulation for tightening process. In this paper, suitable pitch difference was considered for other nominal thread diameters to realize anti-loosening performance. Suitable pitch difference was determined by assuming that loosening force should be balanced to the anti-loosening force. Then, the suitable pitch difference was expressed as approximate formula. By using Junker vibration test, we have confirmed that the proposed formula is valid for M12 since $\alpha = 40\ \mu m$ have no loosening equation (17) is valid.

References

[1] Hard Lock Kogyo KK 2002 Hard lock nut *Japan Patent* 2002-195236
[2] Izumi S, Yokoyama T, Teraoka T, Iwasaki A, Sakai S, Saito K, Nagawa M and Noda H 2005 Verification of anti-loosening performance of super slit nut by finite element method *Trans Jpn Soc Mech Eng* **71** 380-6
[3] Chen D H, Shimizu E and Masuda K 2012 Relation between thread deformation and anti-loosening effect for nut with circumference slits *Trans Jpn Soc Mech Eng* **78** 390-402
[4] Goodier J N and Sweeney R J 1945 Loosening by vibration of threaded fastenings *Mechanical Engineering* **67** 798-802
[5] Nishida S I 1944 Failure Analysis in Engineering Applications, Butterworth Heinemann, Oxford
[6] Maruyama K 1975 Stress analysis of a bolt-nut Joint by the finite element method and the copper-electroplating method (3nd Report) *Trans Jpn Soc Mech Eng* 2292-302
[7] Noda N A, Chen X and Kuhara M 2008 The reduction of stress connection by tapering threads *Trans Jpn Soc Mech Eng* **74** 1305-13
[8] Xiao Y, Wang Q, Noda N A, Akaishi Y, Takase Y and Nishida S 2011 Stress reduction effect of tapering thread bolts and nuts which have slightly different pitches *Trans Soc Automotive Engineers Jpn* **42** 927-33
[9] Akaishi Y, Chen X, Yu Y, Tamasaki H, Noda N A, Sano Y and Takase Y 2013 Fatigue strength analysis for bolts and nuts which have slightly different pitches considering clearance *Trans Soc Automotive Engineers Jpn* **44** 1111-7
[10] Chen X, Noda N A, Maruyama H, Wang H, Sano Y and Takase Y 2014 Experimental investigation on the suitable pitch difference improving the fatigue strength significantly for high strength bolts and nuts *Proceedings of the 3rd International Conference on Fracture Fatigue and Wear* **2** 195-200
[11] Chen X, Noda N A, Wahab M A, Akaishi Y I, Sano Y, Takase Y and Fekete G 2015 Fatigue failure analysis in bolt-nut connection having slight pitch difference using experiments and finite element method *Acta Polytechnica Hungarica* **12** 61-79
[12] Takase Y, Noda N A, Chen X, Sano Y, Fujisawa R, Maruyama H and Wahab M A 2015 Fatigue strength evaluation for bolt-nut connections having slight pitch difference considering incomplete threads of nut *Proceedings of 4th International Conference on Fracture Fatigue and Wear* **3** 43-9
[13] Noda N A, Sano Y, Takase Y, Chen X, Maruyama H, Wang H and Fujisawa R 2015 Anti-loosening performance of special bolts and nuts having enhanced fatigue life by introducing pitch difference *Transactions of Society of Automotive Engineers of Japan* **46** 121-6

[14] Chen X, Noda N A, Wahab M A, Sano Y, Maruyama H, Wang H, Fujisawa R and Takase Y 2015 Fatigue life improvement by slight pitch difference in bolt-nut connections *Journal of the Chinese Society of Mechanical Engineers* **37** 11-9

[15] Noda N A, Chen X, Sano Y, Wahab M A, Maruyama H, Fujisawa R and Takase Y 2015 Effect of pitch difference between the bolt-nut connections upon the anti-loosening performance and fatigue life *The Journal of Strain Analysis for Engineering Design* **96** 476-89