Bolt Clamping Force versus Torque Relation (*F-T* Relation) during Tightening and Untightening the Nut Having Slight Pitch Difference

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Abstract

In this study, tightening and untightening experiments were conducted for pitch difference nuts towards realizing good anti-loosening performance. By applying the three-dimensional FEM, the relationship between the tightening force F and the tightening torque T was clarified by varying the pitch difference α during the whole nut tightening/untightening process. The well-known F-T formula available for tightening the normal nut (α=0) agrees with the FEM results and the F-T formula for untightening the normal nut (α=0) was newly proposed. Then, the loosening resistance ability of the pitch difference nut was clarified focusing on the untightening in comparison with the normal nut.

Keywords

Bolt-Nut connection; Pitch difference; Anti-loosening; Screwing; Tightening; Untightening, Unscrewing

1. Introduction

The bolt-nut connection is an essential machine element widely used and standardized. About 3000 bolts are used in a car with low cost. To fix the engine under high intensity of vibration, for example, good anti-loosening performance is requested. A lot of studies are available for improving anti-loosening[1-10)] as well as for reducing stress concentration[11-24)]. However, such special nuts usually require more components or special geometric shapes with complicated manufacturing processes and high cost.

The bolt-nut connection with pitch difference was proposed and studied mainly to improve the fatigue properties with low cost. In our recent studies, the anti-loosening performance and the fatigue strength improvement were verified under a certain range of pitch difference[22-24)]. Furthermore, in the preceding paper, the authors clarified that the prevailing torque is closely related to the anti-loosening performance experimentally and analytically [24,25)].

Prevailing torque is defined as the torque necessary for screwing a nut before clamping. Consequently, unlike a free spinning nut, a so-called prevailing torque nut has rotation resistance during screwing and unscrewing[25)]. It should be noted that the residual prevailing torque during unscrewing is more useful for anti-loosening compared to the prevailing torque during screwing[26)]. This is because the residual prevailing torque may express the remaining rotational resistance under no clamping force[26]. Recently the authors studied the screwing and unscrewing the pitch difference nut without considering the nut tightening although that may affect the anti-loosening.

In this paper, therefore, the clamping force F vs tightening torque T relation during the tightening/untightening process will be discussed. Since bolt- nut connections are used to clamp mechanical elements, the F-T relation is essential. It will be shown that when the nut with pitch difference is forcibly loosened, a large loosening resistance torque appears. Practically this loosening resistance torque may prevent the nut from self-loosening very well. By comparing the F-T relation between the pitch difference nut and normal nut, the superiority of loosening resistance ability will be clearly explained.

2. Experimental method and results for tightening and untightening

2.1 Screwing and tightening process

In this study, the clamping force and tightening torque relation will be discussed experimentally and analytically for normal nut and pitch difference nut. Fig.1 illustrates the screwing process and the tightening process, which should be distinguished from before and after the nut touching the clamped body. Moreover, in Fig.1, the untightening process and the unscrewing process are also illustrated. The tightening and untightening processes are specially conducted analytically and experimentally in this study. Fig.2 illustrates the contact status of thread for the pitch difference nut during screwing, tightening, untightening, unscrewing process. The nut position change $A \rightarrow B \rightarrow \ldots \rightarrow F \rightarrow G \rightarrow G^u \rightarrow F^u \rightarrow \ldots \rightarrow B^u \rightarrow A^u$ defined in the preceding paper[25] will be also used in this study.

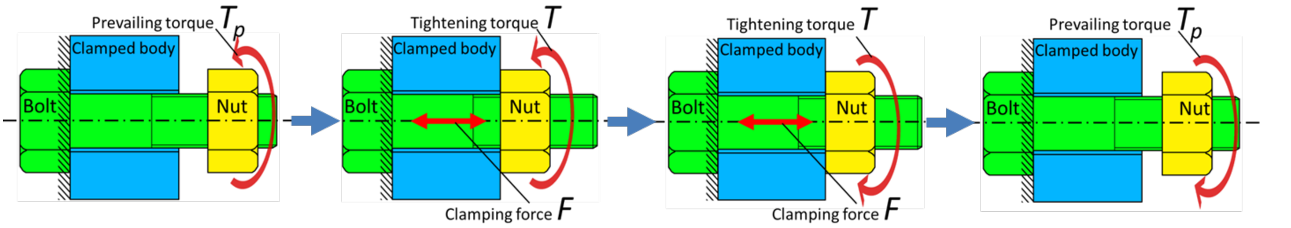
2.2 Specimen and experimental condition

Fig.3 shows JIS M12 bolt-nut considered in this study. Fig.4 illustrates the clearance and the pitch difference α between bolt and nut threads. The clearance in the axial direction C_x =59μm. The bolt pitch is the same of JIS M12 bolt pitch p=1750μm. The nut pitch (1750+α)μm is larger than the bolt pitch by αμm as shown in Fig.4. In this study α=28μm, α=40μm, α=45μm are considered. Fig.5 shows the thread contact status when the prevailing torque T_p appears. Fig.6 shows the stress strain relation for SCM435 and S45C used in the analysis. Table 1 shows the material properties of bolt and nut. The axisymmetric finite element analysis showed that the stress distribution of pitch difference nut $\alpha > 0$ is totally different from the one of normal nut $\alpha = 0$ [22-28]. For $\alpha = 0$, the stress at thread root gradually decreases from the bearing surface side to the other side[23,27]. In contrast, for the pitch difference bolts $\alpha > 0$, the stress at thread root gradually increases from the bearing surface side to the other side[23]. This is the reason why the fatigue life of bolt nut connections can be improved by introducing a suitable pitch difference[22,23,28].

Fig.7 illustrates the boundary conditions for experiment. The bolt head is fixed and the nut is screwed onto the bolt manually by torque wrench until the nut is in contact with the clamped body;

then, the nut is tightened through the device. Fig.8 illustrates the experimental device based on JIS B 1084 standard. During tightening, the tightening torque T, the clamping force F and the nut rotation angle θ are recorded. The molybdenum disulfide grease spray is used on the thread surface as lubricating oil; then, friction coefficient in thread surface μ_s and the friction coefficient in bearing surface μ_w are also measured by using the device during the nut tightening. During the nut untightening, the torque is measured manually by using a torque wrench and reading the clamping force recorded on the device at the same time.

To provide a suitable bolt clamping force is important in many engineering applications. It is recommended that the bolt nominal stress not be allowed to exceed 70% of the yield strength. When the bolt nominal stress reaches the yield stress σ_y= 800MPa, the bolt clamping force can be calculated as $F_{100\%}$= 68kN. The clamping forces $F_{25\%}$= 16.8kN can be calculated when the bolt stress is 25% (200MPa) of the yield stress. Also, the clamping forces $F_{50\%}$= 33.7kN can be calculated when the bolt stress is 50% (400MPa) of the yield stress. Fig.9 shows the experimentally obtained relationship between the clamping force F and the tightening torque T when α=0. As shown in Fig.9, to generate clamping force $F_{25\%}$, the tightening torque should be $T_{25\%}$=45Nm. To generate clamping force $F_{50\%}$, the tightening torque should be $T_{50\%}$=85Nm. Therefore, in this study, the pitch difference nuts are tightened under $T_{25\%}$=45Nm and $T_{50\%}$=85Nm.



(a) Screwing process (b) Tightening process (c) Untightening process (d) Unscrewing process

Fig.1 Schematic illustration for (a) screwing process (b) tightening process (c) untightening process and (d) unscrewing process.

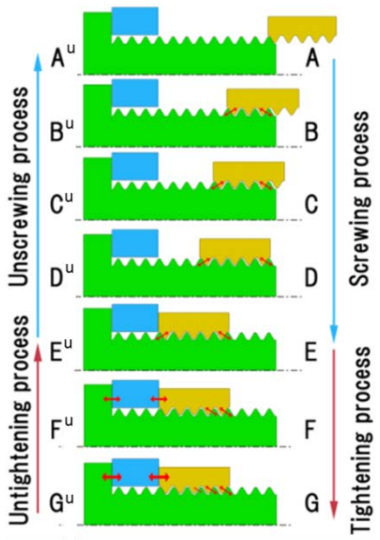


Fig.2 Screwing and tightening process of nut.

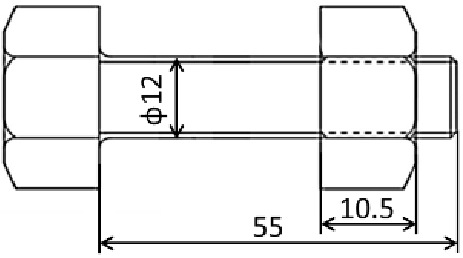


Fig.3 M12 Bolt-nut specimen (mm).

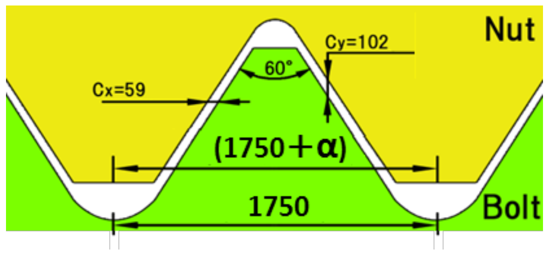


Fig.4 Pitch difference α and clearance between bolt and nut threads (μm).

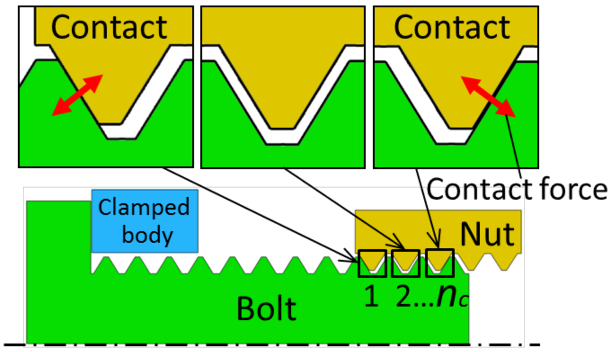


Fig.5 Contact status when the prevailing torque appears between bolt and nut.

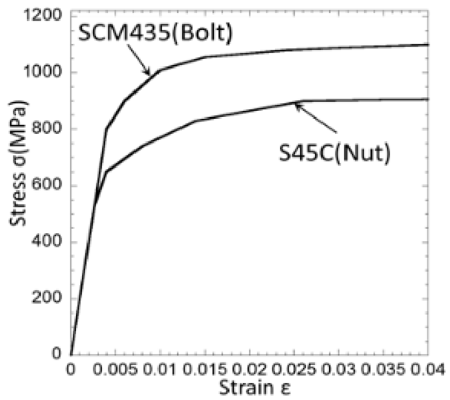


Fig.6 Stress strain relation for SCM435 and S45C.

Table 1 Material properties of bolt and nut.

	Young's modulus E (GPa)	Poisson's ratio ν	Yield strength σ_y (MPa)	Tensile strength σ_B (MPa)
SCM435 (Bolt)	206	0.3	800	1200
S45C (Nut)	206	0.3	530	980

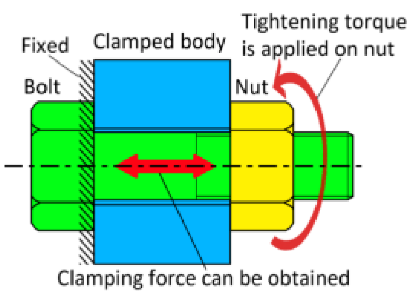
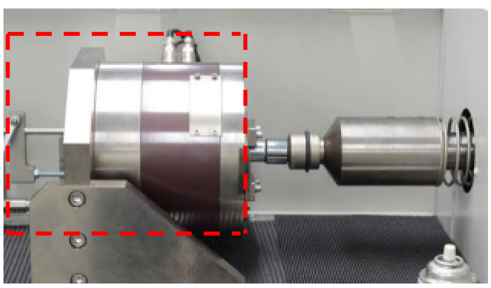
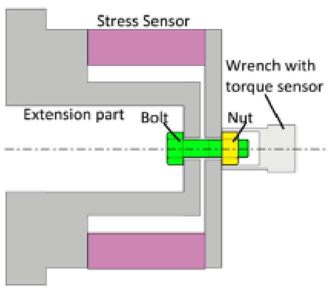


Fig.7 Boundary conditions of nut tightening experiment.



(a) Photo of tightening experiment device.



(b) Schematic illustration of tightening experiment device.

Fig.8 Nut tightening experiment device based on JIS B 1084

2.3 Experimentally obtained *F*- *T* relation during tightening and untightening

Fig.10 shows the experimentally obtained clamping force F vs tightening torque T relation when the pitch difference α=28μm, α=40μm, α=45μm. The points in Fig. 10 denote the experimental results, and the smooth lines denote F-T relation obtained by applying the least squares method. It is seen that the pitch difference α=28~45μm is suitable for tightening and untightening. For example, the previous studies showed a larger pitch difference $\alpha \geq 60$μm is not suitable for JIS M16 bolt-nut connections because the nut could not be screwed onto the bolt due to self-locking [24,25]. In Fig.10 (a), the nuts are tightened in the range $T \leq T_{25\%}$ =45Nm. In Fig.10 (b), the nuts are tightened in the range $T \leq T_{50\%}$=85Nm. In those figures, the clamping force F appears when the tightening torque larger than the prevailing torque as $T>T_p$. The clamping force F increases up to the maximum value $F_{25\%}$=16.8kN and $F_{50\%}$=33.5kN; then, the tightening torque reaches $T=T_{25\%}$ and $T_{50\%}$. The untightening process starts after reaching the torque $T=T_{25\%}$ or $T_{50\%}$. For a certain period, the nut rotates together with the bolt without changing the contact status, and during this period the tightening force F is almost unchanged. After the magnitude of reversing torque $|T|$ reaches a threshold value $|T| = T_{slip}$, the magnitude $|T|$ starts decreasing. This is because after $|T| = T_{slip}$ the nut starts slipping and rotating relatively to the bolt. Here, T_{slip} can be defined in Eq. (1) as shown in Fig.10 and Fig.12.

$$T_{slip} \equiv Max|T| \ \ when \ \ F = F_{25\%} \ or F_{50\%} \ \ and \ T < 0 \qquad (1)$$

With decreasing $|T|$, the clamping force F decreases. Finally, the nut is apart from the clamped body as F=0 although the reverse torque still remains as $T_P^u \equiv |T|$ as shown in Fig.10 for α=28μm, α=40μm, α=45μm. The torque defined in Eq. (2) was named the residual prevailing torque T_P^u [26].

$$T_P^u \equiv |T|_{\alpha>0} \ \ when \ \ F = 0 \ and \ T < 0 \qquad (2)$$

The preceding paper discussed the residual prevailing torque T_P^u in comparison with the prevailing torque T_p commonly used [25) 26)]. Instead of Eq. (2), the prevailing torque T_p can be defined in Eq. (3).

$$T_P \equiv |T|_{\alpha>0} \ \ when \ \ F = 0 \ and \ T > 0 \qquad (3)$$

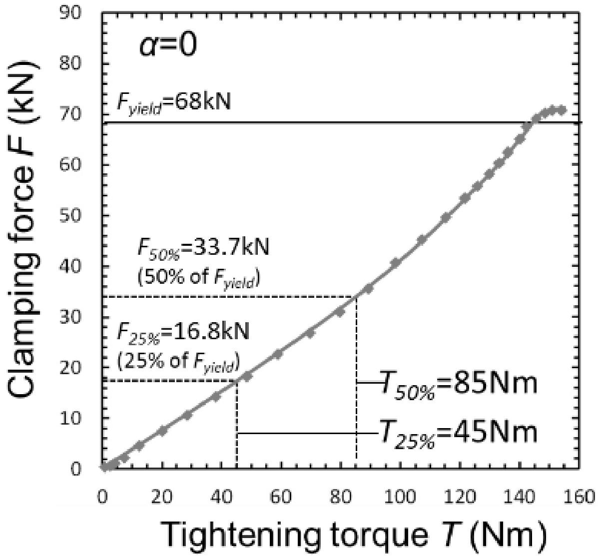
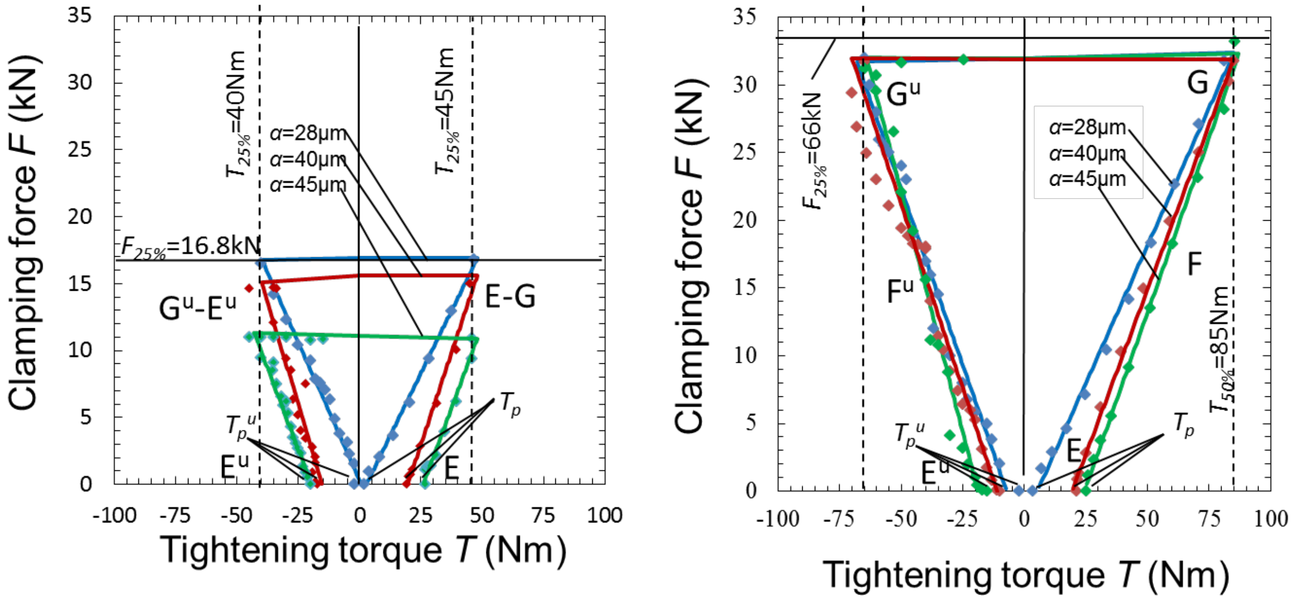


Fig.9 Experimental results for lamping force F vs tightening torque T for α=0.



(a) F-T relation under $T \leq T_{25\%}$ =45Nm.

(b) F-T relation under $T \leq T_{50\%}$ =85Nm.

Fig.10 Experimental results for clamping force F vs tightening torque T

(Points denote the experimental results and the smooth lines denote F-T relation obtained by applying the least squares method.)

3. Analytical method and results for tightening and untightening

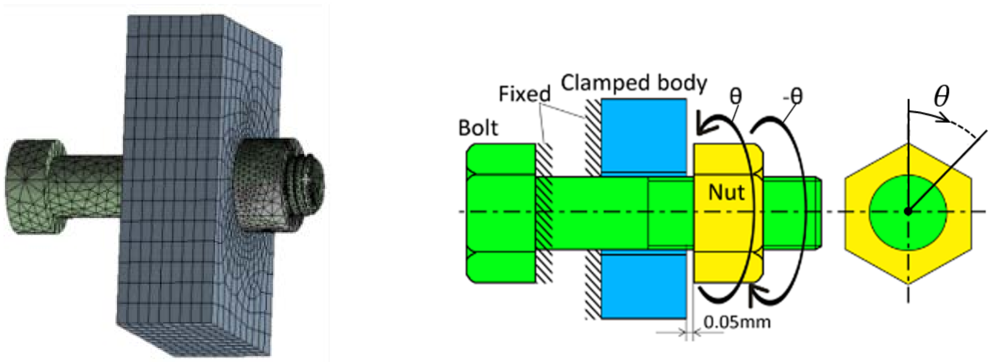
3.1 Analysis method

In this study, the clamping force vs tightening torque relation (F- T relation) will be discussed analytically as well as experimentally. In this paper, the analytical results represent the numerical simulation results obtained by using finite element method. Figure11 shows (a) the 3D FEM mesh composed by tetrahedron solid elements and (b) analytical boundary conditions for normal nut and pitch difference nut. To simplify the model, the hexagonal bolt head and nut are replaced by the cylinder as shown in Fig.11 (a). To express the spiral shapes of the bolt and nut thread, the minimum element dimension in the spiral part of the thread is 0.048 mm. The number of elements is 9.3×10^4, and the number of nodes is 15.1×10^4. Analysis software ANSYS 16.2 is used to conduct non-linear, quasi-static, elastoplastic and contact analysis. Fig.11 (b) shows the analytical boundary conditions where the bolt head is fixed and nut rotation angles $\pm\theta$ are applied on the nut. In order to save the calculation time, the analysis is performed from the Position E in Fig.2. The friction coefficients measured in the experiment are in the range μ_s=0.11-0.15 at the thread surface and in the range μ_w=0.16-0.18 at the bearing surface. The ranges of those friction coefficients coincide with the values obtained by Udagawa[30]. To fit the simulation results with the experimental results, the friction coefficient at the thread surface are chosen as μ_s=0.14 and the friction coefficient at the bearing surface are chosen as μ_w=0.17. The simulation of screwing process can be described in the following way.

In this analysis, the nut rotation angle $\theta=\theta_{25\%}$ and $\theta=\theta_{50\%}$ should be provided as the boundary condition although they are unknown. Here, the nut rotation angle $\theta_{25\%}$ generates the clamping force $F_{25\%}$ and the nut rotation angle $\theta_{50\%}$ generates the clamping force $F_{50\%}$. Therefore, the following two steps (i) (ii) are applied to the analysis for α=28, α=40, α=45μm.

(i) Apply an enough rotation angle θ_{large} =200～300° to the nut. Since the rotation angle θ_{large} is large enough, the mean tensile stress in the bolt exceeds the yield stress. The rotation angles $\theta_{25\%}$ and $\theta_{50\%}$ corresponding to the tightening forces $F_{25\%}$ and $F_{50\%}$ can be found in this step.

(ii) Apply the obtained rotation angles $\theta_{25\%}$, $\theta_{50\%}$ to the nut. Then, the nut is tightened so that the clamping force reaches $F_{25\%}$ and $F_{50\%}$. After that, apply the reverse the nut rotation $-\theta_{25\%}$ and $-\theta_{50\%}$ for the untightened analysis.



(a) FEM mesh for M12 bolt nut.

(b) Boundary conditions.

Fig.11 FEM mesh and boundary conditions for tightening and untightening process.

3.2 Analytically obtained *F*- *T* relation during tightening and untightening processes

Fig.12 shows the analytically obtained F- T relation for pitch difference α=40μm. In Fig.12, the points denote the experimental results, and the lines denote FEA results. In Fig.12 (a), the nut is tightened in the range $T \leq T_{25\%}$ =45Nm with $F_{25\%}$=16.8kN. In Fig.12 (b), the nut is tightened in the range $T \leq T_{50\%}$=85Nm with $F_{50\%}$=33.5kN. In those figures, the clamping force F appears when the tightening torque larger than the prevailing torque as $T>T_p$. In Fig.12, the analytical result is denoted by the solid line, the experimental result is denoted by the line with the plots and the result of α=0 is denoted by the light gray line. As shown in Fig.12, overall the analytical results are in good agreement with the experimental results.

Table 2 and Table 3 compare the analytical and experimental results for $T \leq T_{25\%}$ =45Nm and $T \leq T_{50\%}$=85Nm. Here, ① prevailing torque T_p, ② clamping force $F_{25\%}$, $F_{50\%}$ ③ maximum reversing torque T_{slip} defined in Eq. (1), ④residual prevailing torque T_P^u defined in Eq. (2) are focused as indicated in Fig. 12. As shown in those Tables, most of the results coincide with each other within 10% difference. Under a certain clamping force F, the experimental torque is slightly different and often larger than the analytical torque presumably because of the following reasons. First, during the screwing/tightening for relatively large α, the contact threads may wear down causing wear debris, which cannot be considered in the analysis. Second, by using the experiment device in Fig.8 the tightening torque can be measured automatically, but the untightening torque had to be measured manually including relatively larger error. Third, in this experiment, since the tightening torque was applied relatively faster, the lubricating oil film becomes thinner causing larger friction.

Since the analytical and experimental results coincide with each other, the F-T relations can be discussed through the FEM simulation. Fig.13 shows the analytically obtained F-T relations for α=28, 40, 45μm. As shown in Fig.13 (a) when $T \leq T_{25\%}$ =45Nm, with increasing the pitch difference α, the magnitude of reversing torque $|T|$ slightly increases. In Fig.13 (b) when $T \leq T_{50\%}$=85Nm, with increasing the pitch difference α, the reversing torque $T < 0$ changes smaller. As shown in Fig.13 (a), when the tightening torque T is relatively smaller, the clamping force F increases relatively larger with increasing α. With increasing α, the residual prevailing torque T_P^u increases independent of the tightening torque as well as prevailing torque T_p.

To clarify the plastic deformation effect, Fig.13 (b) shows the F-T relation obtained by the elastic analysis compared to the elastoplastic analyses based on Fig.6. The elastic results show that the residual prevailing torque T_P^u is almost the same as prevailing torque T_p. Instead, the elastoplastic results show that the residual prevailing torque T_P^u is smaller than the prevailing torque T_p. As shown in Table 3, the residual prevailing torque T_P^u obtained by elastoplastic FEA is in good agreement with the experimental results. It may be concluded that elastic tightening the nut is suitable for preventing loosening. In Fig.13(c), F-T relations obtained by the formula for normal nut α=0 discussed in the next section are also indicated to understand the tightening and untightening difference.

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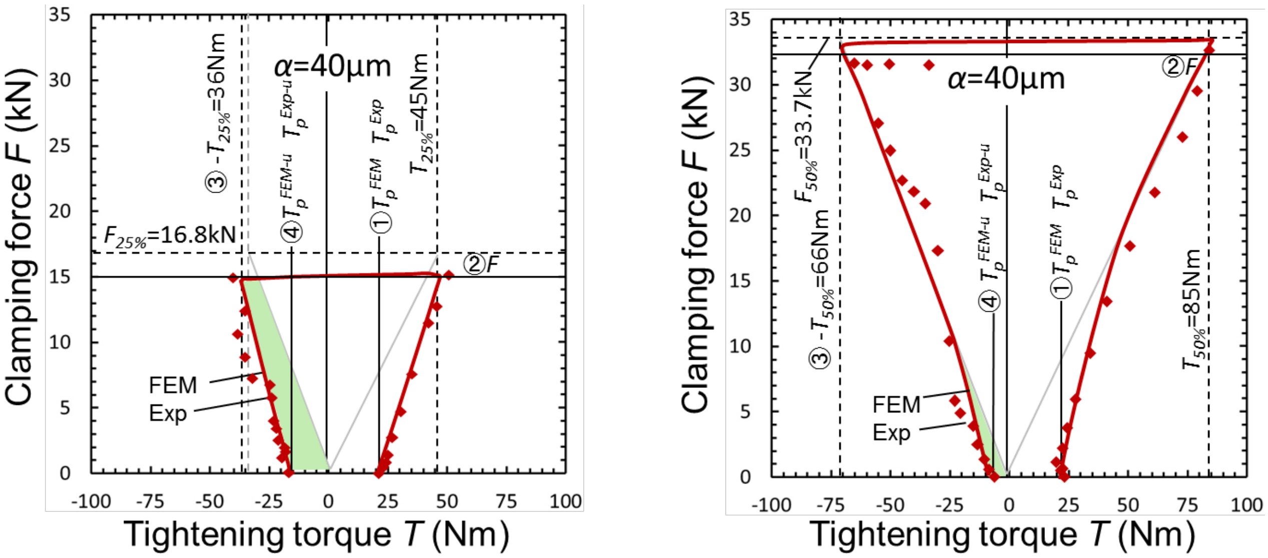
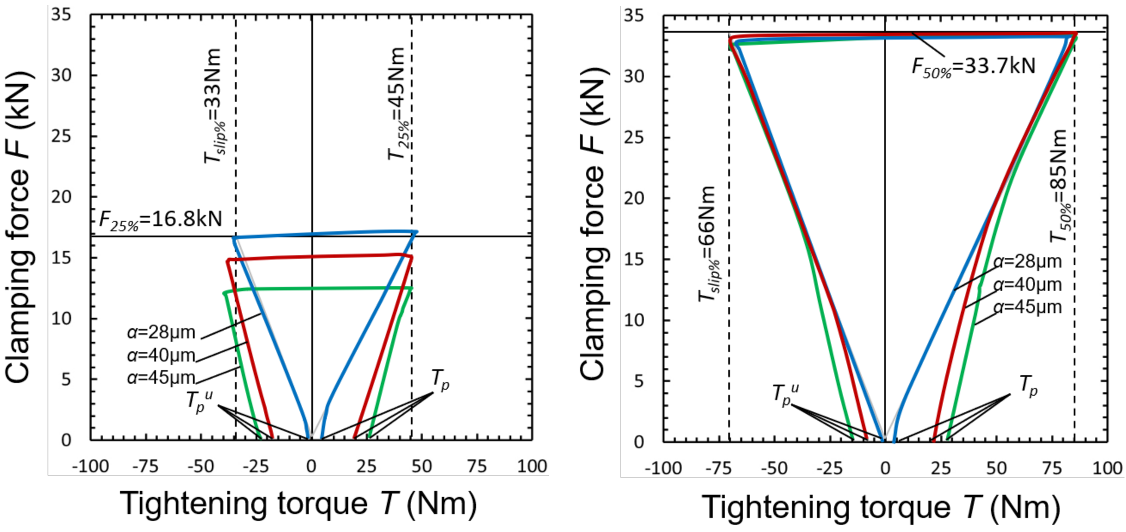


(a) F-T relation under $T \leq T_{25\%}$ =45Nm.

(b) F-T relation under $T \leq T_{50\%}$ =85Nm.

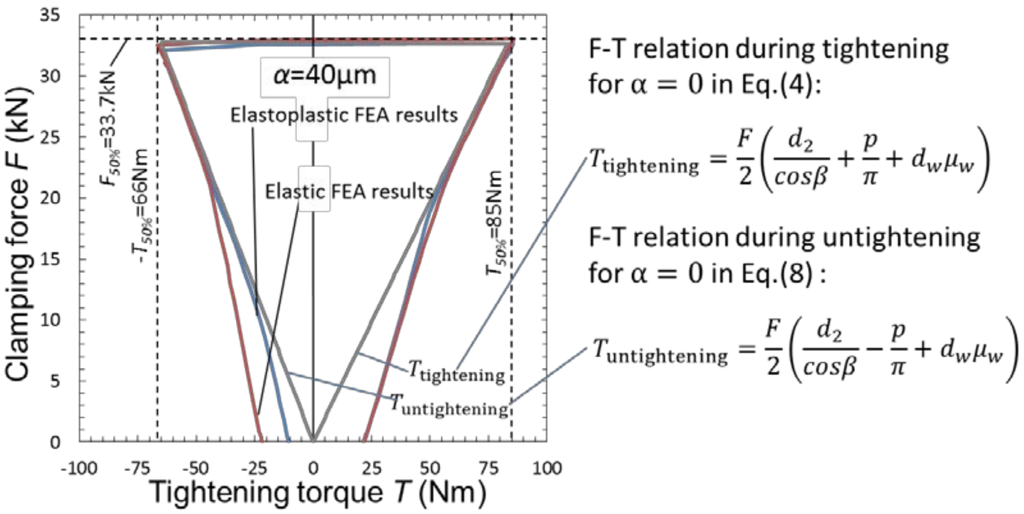
Fig.12 FEM and experimental results for clamping force F vs tightening torque T for α=40μm.

(Points in Fig.12 denote the experimental results and lines denote FEA results)



(a) F-T relation under $T \leq T_{25\%}$ =45Nm.

(b) F-T relation under $T \leq T_{50\%}$ =85Nm.



(c) *F*-*T* relation for α=40μm under $T \leq T_{50\%}$ =85Nm obtained by elastic analysis in comparison with the elastoplastic analysis

Fig.13 FEM results of clamping force *F* vs tightening torque *T*.

Table 2 Comparison of T_p, $F_{25\%}$, T_{slip}, T_p^u when $T \leq T_{25\%}$=45Nm.

	① T_p^{FEM} (Nm)	(T_p^{EXP})	② $F_{25\%}^{FEM}$ (kN)	($F_{25\%}^{EXP}$)	③ T_{slip}^{FEM} (Nm)	(T_{slip}^{EXP})	④ T_p^{u-FEM} (Nm)	(T_p^{u-EXP})
α=28μm	4.7	(4.3)	17.0	(16.8)	33.5	(35.0)	2.1	(2.0)
α=40μm	21.5	(19.5)	15.2	(14.9)	36.0	(40.0)	16.5	(17.0)
α=45μm	27.0	(26.9)	12.5	(10.9)	39.2	(45.0)	23.5	(26.0)

Table 3 Comparison of T_p, $F_{50\%}$, T_{slip}, T_p^u when $T \leq T_{50\%}$=85Nm..

	① T_p^{FEM} (Nm)	(T_p^{EXP})	② $F_{50\%}^{FEM}$ (kN)	($F_{50\%}^{EXP}$)	③ T_{slip}^{FEM} (Nm)	(T_{slip}^{EXP})	④ T_p^{u-FEM} (Nm)	(T_p^{u-EXP})
α=28μm	4.7	(4.2)	33.0	(31.8)	66.1	(65.0)	0	(2.0)
α=40μm	21.5	(21.3)	33.5	(31.8)	66.8	(70.0)	10.8	(10.0)
α=45μm	27.0	(24.9)	32.9	(33.2)	66.5	(65.0)	16.4	(15.0)

4 *F*-*T* formula for normal nut

The *F*-*T* relation was discussed experimentally and analytically in the above sections. For normal nut, Equations (4) - (7) are commonly used [29)] to connect components properly by providing suitable *T* and *F*. Fig.14(a) illustrates that the torque *T* is composed of three different types of torques as shown in Equation (4), that is, $T=T_{Thread}+T_{Axial}+T_{Bearing}$ during the tightening process. Here, the notation T_{Thread} denotes the friction torque at thread surface, T_{Axial} denotes the torque due to bolt elongation, and $T_{Bearing}$ denotes the friction torque at bearing surface. The FEM analysis results can be compared with those theoretical formulas.

$$T = \frac{F}{2}\left(\frac{d_2}{\cos\beta}\mu_s + \frac{p}{\pi} + d_w\mu_w\right)$$

$$= T_{Thread} + T_{Axial} + T_{Bearing} \tag{4}$$

$$T_{Thread} = \frac{F}{2}\cdot\frac{d_2}{\cos\beta}\mu_s \tag{5}$$

$$T_{Axial} = \frac{F}{2}\cdot\frac{p}{\pi} \tag{6}$$

$$T_{Bearing} = \frac{F}{2}d_w\mu_w \tag{7}$$

In Equations (4) - (7), d denotes the nominal diameter of the screw, d_2 denotes the effective diameter of the screw, p denotes the pitch, d_w denotes the equivalent diameter of the friction on the bearing surface, β denotes the half angle of the screw thread, d_0 denotes the outer diameter of the nut bearing surface, and d_h denotes the inner diameter of the nut bearing surface. Table 4 shows the dimensions and friction coefficients for M12 bolt and nut necessary for Equations (4) - (7).

Fig.14(b) illustrates that the reversing torque $T < 0$ is also composed of three different torques as $T = T_{Thread} - T_{Axial} + T_{Bearing}$. During the untightening, T_{Thread} and $T_{Bearing}$ are still acting as resistance torques, but T_{Axial} is acting in the same direction of untightening as shown in Equation (8).

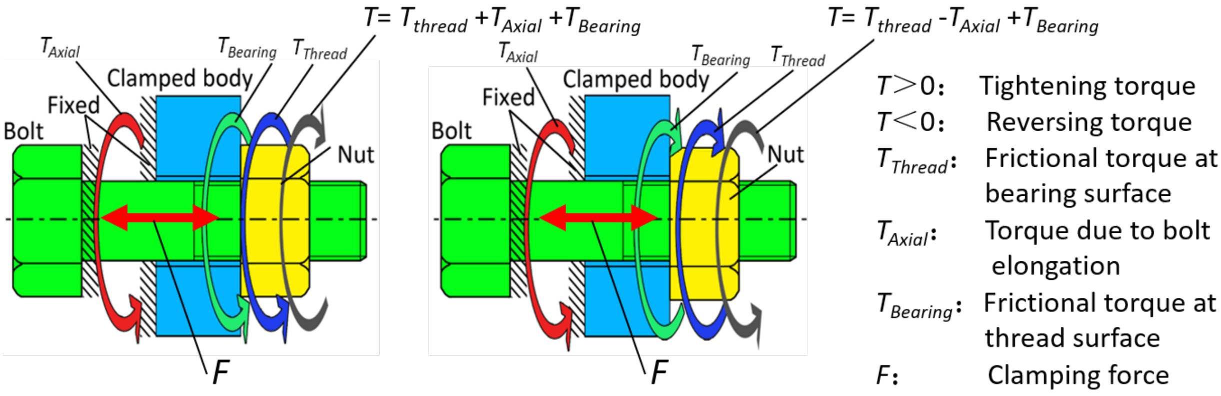
$$T = \frac{F}{2}\left(\frac{d_2}{\cos\beta}\mu_s - \frac{p}{\pi} + d_w\mu_w\right)$$

$$= T_{Thread} - T_{Axial} + T_{Bearing} \tag{8}$$

Fig.15(a) shows the relationship between clamping force F and tightening torque T during the tightening/untightening process of a normal nut when $T \leq T_{50\%}$. The black line shows the tightening torque T, the green shows $T_{Bearing}$ and the red shows $T_{Thread} \pm T_{Axial}$. Besides, the solid line shows the FEM analysis results, and the broken line shows values of Equations (4) - (8). It is seen that the FEM analysis results and Equations (4) - (8) are in good agreement. It is therefore confirmed that the reversing torque $T < 0$ can be expressed as shown in Equation (8).

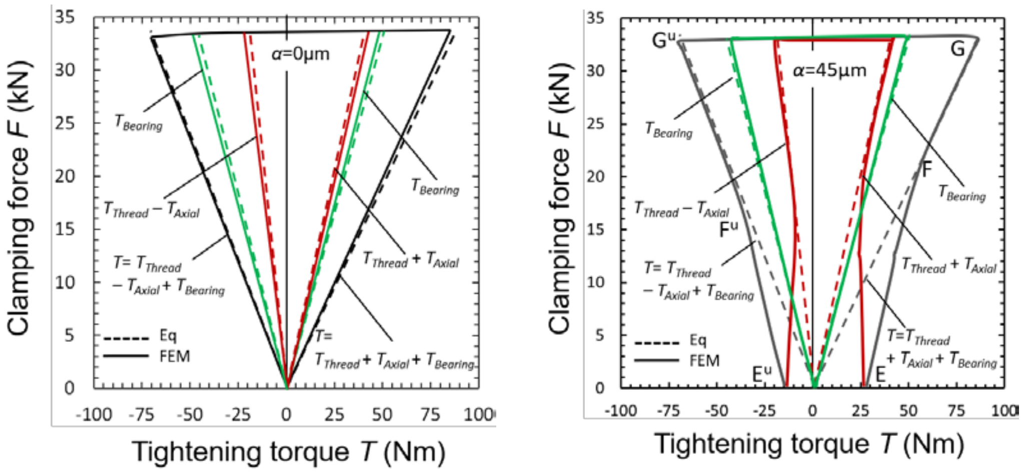
Table 4 Dimensions and friction coefficients for M12 bolt and nut in Equation (4) - (8).

Pitch diameter d_2 (mm)	Half angle of thread β (°)	Friction coefficient		Pitch p (mm)	Bolt bearing surface outer diameter d_0 (mm)	Bolt hole diameter d_h (mm)
		Thread surface μ_s	Bearing surface μ_w			
10.863	30	0.14	0.17	1.75	18	13.2



(a) Tightening (b) Untightening

Fig.14 Torques in tightening/ untightening process.



(a) α=0 (b) α=45μm

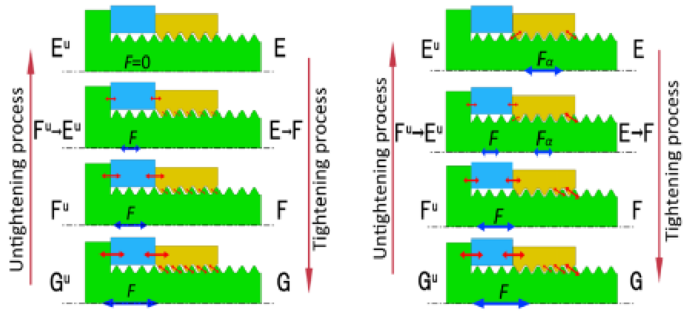
Fig.15 FEM results for clamping force F vs tightening torque T when $T \leq T_{50\%}$

Fig. 15 (b) shows the F-T relation obtained by FEM for the pitch difference nut α=45μm. It is seen that the analytical results coincide with Equation (4) from the nut Position F → G. In other words, The F - T relation for the pitch difference nut can be predicted by the formulas (3)-(7) after Position F. Regarding the tightening in Fig.15 (a), the F - T relation for the normal nut is linear. However, as shown in Fig. 15(b), the F - T relation is non-linear initially but becomes linear and coincides with the F - T relation of the normal nut from Position F to G. The reason can be explained as follow.

Fig.16 illustrates the contact force as the red arrow during the tightening/untightening process. For the normal nut α=0 as shown in Fig.16 (a), the direction of the contact force at each thread remains unchanged during the whole tightening/untightening process. On the other hand, when α=45μm in Fig.16 (b), at the beginning from Position E, the contact force appears at both nut ends with the different left side surface at No.1 thread. With increasing the clamping force F during the tightening process from Position E, the contact force at left side decreases and the contact force at right side increases. At this time, the axial force between the threads F_α due to the pitch difference accumulated during the screwing process decreases gradually but the tightening force F increases at the same time. In other words, the tightening force F of the nut with pitch difference increases more rapidly compared to the normal nut with increasing T.

As shown in Fig.16 (b), from Position F, the contact force at left side of nut disappears completely, at this time the axial force between thread F_α disappears and released to the clamping force F completely. Since the contact status becomes the same during Position F→G as shown in Fig.16 (b), the slope of the F - T relations becomes the same for α=0 and α=45μm. Similarly, during the untightening the nut α=45μm, the contact status becomes different at Position F^u as shown in Fig.16 (b), the slope of the F - T relations becomes different during Position F^u→E^u.

Fig.17 shows the 3D contact status of nut thread during tightening/untightening process when pitch difference α=45μm obtained by analysis. Here the yellow zone shows near contact where the nut thread is near the bolt thread but no contact. The orange zone shows contact state where the threads are sliding with contact each other. The red zone shows adhesion state where the threads contact without sliding. When the nut rotation is reversed during Position G→G^u, the bolt and the nut rotate together without sliding.



(a) α=0 (b) α=45μm

Fig.16 Contact status in tightening/untightening process of nut.

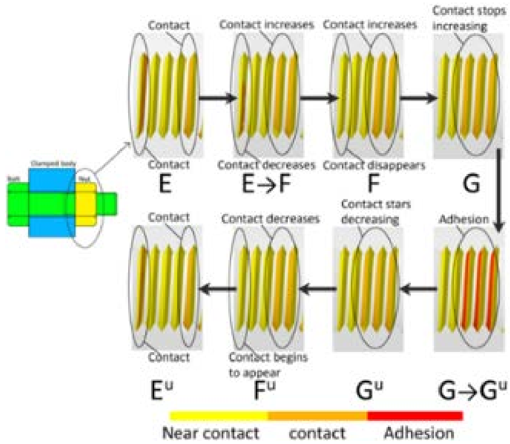


Fig.17 Contact status of bolt nut threads when α=45μm.

5. Difference of *F*-*T* relation between the pitch difference nut and the normal nut

Fig.18 illustrates the F-T relation of the pitch difference nut. The green line denoting α=45μm is compared with the gray line denoting α=0. The nut position change E→F→G in Fig.18 (b) is illustrated in Fig.16. The clamping force F of α=0 appears by applying T >0. Instead, the clamping force F of α =45μm appears by applying $T \geq T_p$ when the torque exceeds the prevailing torque.

After the clamping force reaches $F=F_{25\%}$ or $F=F_{50\%}$ defined in Fig.9, untightening torque is applied as $T < 0$. Then, clamping force F decreases with decreasing $|T|$. As shown in Fig.18, under the same tightening force F, the magnitude T of α=45μm is equal to or larger than the magnitude T of α = 0 as $|T|_{\alpha=45} \geq |T|_{\alpha=0}$. In Fig.18, the light green zone illustrates the difference between α=0 and α=45μm defined in Eq. (9).

$$T_R^u \equiv |T|_{\alpha>0} - |T|_{\alpha=0} \;\; when \;\; T < 0 \qquad (9)$$

Such torque difference in Eq.(9) can be regarded as the loosening resistance torque T_R^u contributing to anti-loosening. Even when F=0 the reverse torque is not zero as $|T|>0$, which was named residual prevailing torque in the previous paper[25]. The residual prevailing torque may represent the anti-loosening performance [5,6,24,25,26] although it is defined only when F=0. In this paper, therefore, more general concept, "loosening resistance torque" is newly introduced as shown in Eq. (10). Equation (10) represents loosening resistance when the clamped force is still enough $F\geq 0$ during untightening.

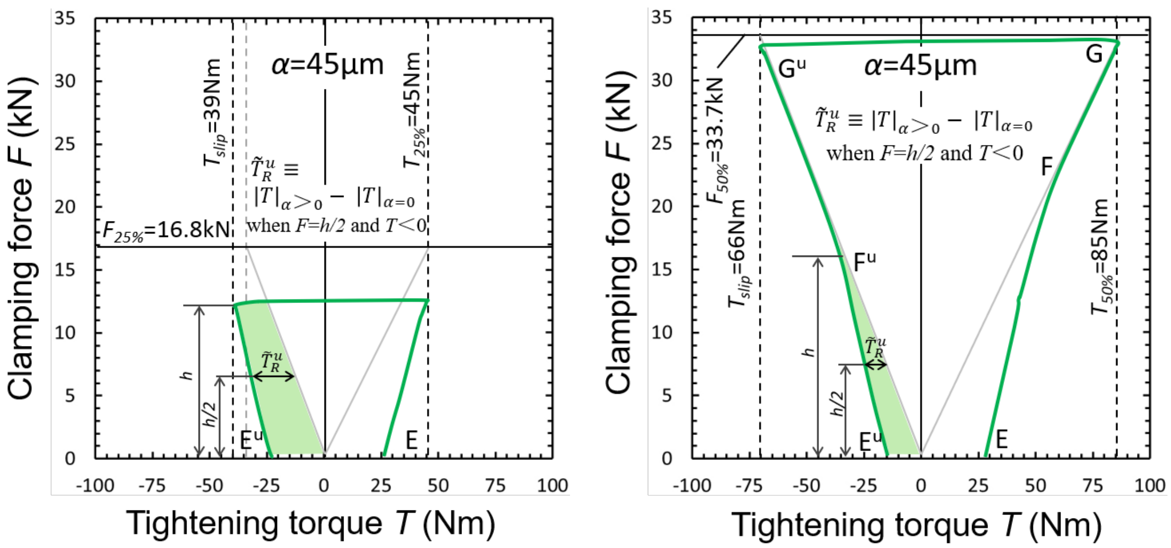
For normal nut $\alpha=0$, Fig.18 shows no loosening resistance $|T|=0$ at F=0 although under sufficient F>0 the loosening torque $|T|>0$. Instead, for α=45μm, the following loosening resistance $T_R^u>0$ can be expected compared to $\alpha=0$ even when F=0.

$$T_R^u > 0 \;\; when \; 0 \leq F < h \tag{10}$$

Here, $F=h$ is the upper limit providing $T_R^u > 0$. In other words, in Fig.18 (a), $F=h$ is the maximum clamping force; but in Fig.18 (b), h is the clamping force at Position F^u in Fig.16 where nut threads at both ends starts contacting to the bolt threads. Since $T_R^u>0$ when $0\leq F<h$, loosening resistance torque T_R^u can be represented by median value $\tilde{T}_R^u$ at $F=h/2$ as shown in Eq. (11).

$$\tilde{T}_R^u \equiv |T|_{\alpha>0} - |T|_{\alpha=0} \;\; when \;\; F=h/2 \;\; and \; T<0 \tag{11}$$

Fig.19 illustrates median loosening resistance torque $\tilde{T}_R^u$ vs. pitch difference α. It can be seen that $\tilde{T}_R^u$ increases with the increasing pitch difference α. As shown in Fig.19, the loosening resistance torque $\tilde{T}_R^u$ can be larger when the tightening force is not very large.



(a) F-T relation when α=45μm under $T\leq T_{25\%}$=45Nm. (b) F-T relation when α=45μm under $T\leq T_{50\%}$=80Nm.

Fig.18 FEM results for clamping force F vs tightening torque T.

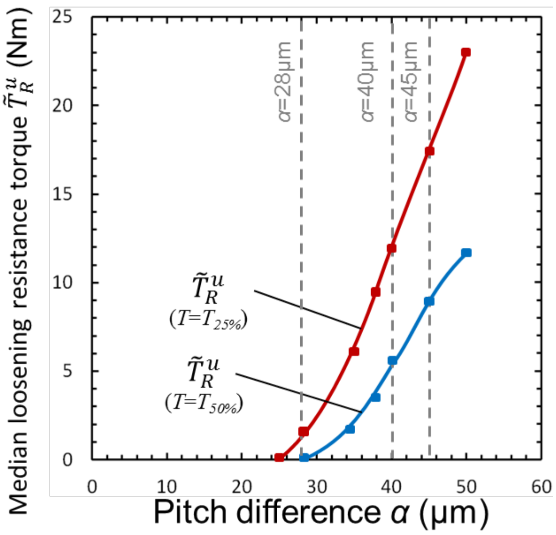


Fig.19 Median loosening resistance torque $\tilde{T}_R^u$ vs pitch difference α.

As shown in Fig.19, during untightening, the loosening resistance torque $\sim\tilde{T}_R^u$ appears. Such loosening resistance is caused by the remaining pitch difference after screwing, tightening and untightening. Fig.20 illustrates the thread deformation when α=45μm and $T\leq T_{25\%}$. In Fig.20, the nut is at Position B^u after the nut past through all screwing, tightening, untightening, unscrewing processes indicated in Fig.2 as A→B→…→F→G→G^u→F^u→…→B^u. At Position B^u, there is no thread contact anymore. In Fig.20, the thread deformation is enlarged by 10 times. Due to the contact about upper 80% portion of the thread, the upper 30% of the bolt thread is deformed with the largest displacement +74μm at the top of No.8 thread. However, the lower 70% thread portion is not deformed very much. It may be concluded that the pitch difference does not change due to contact except upper thread portion. This is the reason why the loosening resistance torque $\sim\tilde{T}_R^u$ can be expected as shown in Fig.19. The loosening resistance torque can be larger when the tightening force is not very large.

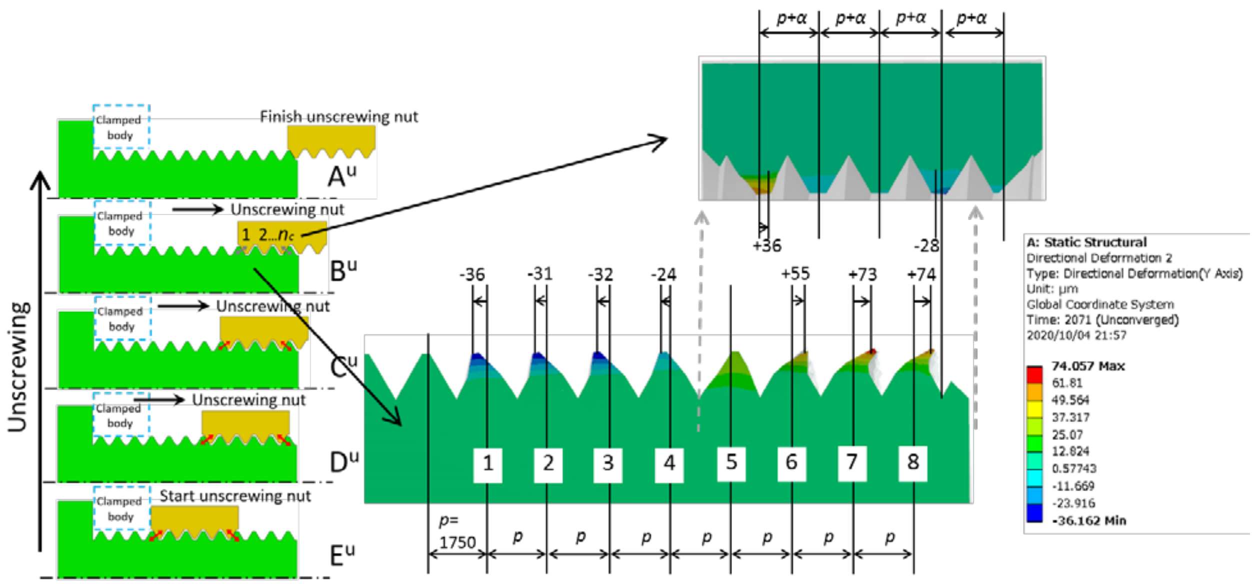


Fig.20 Thread section enlarged by 10 times when α=45μm at Position B^u (Unit μm)

6. Conclusions

In this study, tightening and untightening experiments were conducted for pitch difference nuts towards realizing good anti-loosening performance. By applying the three-dimensional FEM, the clamping force F and tightening torque T relation was discussed during the whole nut tightening/untightening process. The conclusions can be summarized in the following way.

(1) The relationship between the tightening force F and the tightening torque T was clarified by varying the pitch difference nut experimentally. The F-T relation can be predicted by the application of 3D FEM since the results are in good agreement during both tightening and untightening processes.

(2) The well-known F-T formula available for tightening the normal nut α=0 agrees with the FEM analysis results within 5%. The F-T formula for untightening the normal nut α=0 was newly proposed. The F-T relation of the pitch difference nut coincides with the F-T relation of the normal nut after Position F in Fig. 16 where the nut contact status becomes the same as the nut contact status of the normal nut.

(3) The thread deformation analysis showed that the pitch difference does not change due to the tightening and untightening except the upper thread portions.

(4) The difference of the F-T relation was clarified between the pitch difference nut and the normal nut. The loosening resistance torque T_R^u can be defined as the difference during the untightening process. With increasing the pitch difference α, median loosening resistance torque $\tilde{T}_R^u$ increases suggesting that the pitch difference nut has superior anti-loosening performance. The loosening resistance torque T_R^u can be larger when the tightening force is not very large.

In this study, FEM results were compared with the experimental results drawing the conclusions summarized above. To quantify the pitch difference α effect in the F-T behavior, however, a 3D FE simulation is needed. In contrast, the normal nut α=0, simple classical formulas (4)- (8) provide very accurate F-T results without costly FE models. In this sense, it is useful to generalize equations (4) - (8) to include an additional term that quantifies the prevailing torque controlled by α. This generalized F-T formula would be really useful, avoiding complex 3D simulations. In this sense, something similar should be done in the further study to predict self-loosening with equation (8) in the reference[31)].

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