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Prevailing torque and residual prevailing torque of Bolt-Nut connections having slight pitch difference

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ABSTRACT

In this study, by applying three-dimensional FEM, the nut screwing process is analyzed to obtain the prevailing torque T_p confirming anti-loosening. Since the nut unscrewing properties after the nut tightening is more important to prevent the nut self-loosening, the nut unscrewing is also analyzed to obtain the residual prevailing torque T_p^u newly defined in this process. Both results of T_p and T_p^u are in good agreement with the experimental results. With increasing the pitch difference, the residual prevailing torque T_p^u increases as well as the prevailing torque T_p.

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KEYWORDS

Bolt-Nut connection; pitch difference; anti-loosening performance; residual prevailing torque; three dimensional FEM

1. Introduction

The bolt-nut connection is an important fastening element used frequently with low cost. As an example, more than 3000 various bolts-nuts are used in a car. Those various bolts-nuts are standardized by ISO, JIS and DIN based upon the closely related metric system. Under dynamic loading, the bolt clamping force is often reduced to zero when the returning rotational force exceeds the frictional force. This is often happening because of the spiral shape of the bolt-nut thread. As a result, a lot of accidents caused by loosening are reported in vehicles, aircraft and the like. For that reason, good anti-loosening performance and high fatigue strength are always required for bolt-nut connections, and a lot of research was conducted to prevent loosening (Bhattacharya, Sen, and Das 2010; Izumi, Yokoyama, Iwasaki, et al. 2005; Izumi, Yokoyama, Teraoka 2005; Izumi et al. 2009; Chen, Shimizu, and Masuda 2012; Noda, Kuhara, et al. 2008; Noda, Xiao, et al. 2008; Noda et al. 2016; Ranjan, Vikranth, and Ashitava 2013; Liu, Gong, and Ding 2018) and improve fatigue strength (Pedersen and Pedersen 2008; Honarmandi, Zu, and Behdinan 2005; Slovinsky and Hutapea 2010; Sawa et al. 2014; Majzoobi, Farrahi, and Habibi 2005; Noda, Xiao, and Kuhara 2011; Noda et al. 2016; Hirai and Uno 2005; Zhou et al. 2015; Lee, Kim, and Han 2014; Chakherlou et al. 2013; Chen et al. 2015, 2016).

To improve fatigue strength and anti-loosening with low cost, slight pitch differences were studied between bolt-nut threads previously (Chen et al. 2015, 2016; Noda et al. 2016). The axi-symmetric FEM analysis clarified the stress reduction at the thread root to demonstrate the fatigue strength improvement (Chen et al. 2015, 2016). The experiment by varying the pitch difference revealed that the prevailing torque T_p is closely related to the anti-loosening (Noda et al. 2016). Since the pitch difference nut can be manufactured similarly to the normal nuts, the estimated cost can be only about 1.5 times of the normal nut cost.

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Since the axi-symmetric FEM cannot treat 3 D spiral thread, in this paper a three-dimensional EFM will be applied to screwing the nut to evaluate the prevailing torque T_p. Since the nut unscrewing properties after the nut tightening is closely related to preventing the self-loosening, the residual prevailing torque T_p^u will be also discussed during unscrewing the nut by comparing with experiments. Those obtained values of T_p and T_p^u are useful for discussing suitable pitch difference nuts to provide good anti-loosening performance.

2. Experimental method

2.1. Definition of screwing, tightening, untightening and unscrewing processes

As shown in Fig. 1, the screwing process and the tightening process should be distinguished before and after the nut touching the clamped body. Moreover, in Fig. 1, the untightening process and the unscrewing process are also illustrated. Figure 2 illustrates the relationship between torque and number of nut rotation for the pitch difference $\alpha = 0$ and $\alpha \neq 0$ during the screwing and tightening processes. Figure 3 illustrates the contact status of thread for the pitch difference nut during the screwing and the tightening process. Position A is where the nut starts to contact with bolt and Position B is where prevailing torque appears. Position C is where prevailing torque is increasing, Position D is where the whole nut thread is completely screwed onto the bolt, and Position E is where the nut starts to contact with the clamped body. Position F is where the clamping force is increasing. Position G is where the nut is tightened completely. Moreover, there are 4 processes in Fig. 3. The process from Position A to Position E is called screwing process, the process from Position E to Position G is called tightening process. After that, the nut is rotated in reverse direction, the process Position $\mathrm{G^u}$ to Position $\mathrm{E^u}$ is called untightening process, the process from Position $\mathrm{E^u}$ to Position $\mathrm{A^u}$ is called unscrewing process. In Fig. 2, when a normal nut is screwed onto bolt, the torque occurs from Position E and increases rapidly as shown by black line. When a pitch difference nut is screwed onto bolt, the torque occurs from Position B and increases until Position D then becomes constant. After nut contacts the clamped body, from Position E the torque increases rapidly as shown by blue line in Fig. 2.

2.2. Specimens

Figure 4 shows the dimensions of JIS M12 bolt-nut used in this study. Figure 5 indicates the clearance and pitch difference between the bolt and nut. The clearance in the axial direction is $C_x = 59\,\mu\mathrm{m}$. The pitch of nut is larger than the pitch of bolt by α μm. The pitch of M12 bolt is $p = 1750\,\mu\mathrm{m}$, and the pitch of nut is $1750+\alpha$ μm. Our previous experimental study proved that the anti-loosening performance is weaker when the pitch difference is smaller (Noda et al. 2016). And larger pitch difference nuts sometimes cannot be screwed onto the bolt. Therefore, the suitable range of pitch difference can be in the range $30\,\mu\mathrm{m} \leq \alpha \leq 50\,\mu\mathrm{m}$. Then, the pitch differences α can be chosen as $\alpha = 35\,\mu\mathrm{m}$, $\alpha = 40\,\mu\mathrm{m}$ and $\alpha = 50\,\mu\mathrm{m}$. Figure 6 shows the contact status when the prevailing torque appears between bolt and nut. Table 1 shows the material properties of bolt and nut. Figure 7 shows the stress-strain relation for SCM435 and S45C.

2.3. Experimental condition

Figure 8 shows the experimental condition. The clamped body and bolt head are fixed. The experimental conditions of the four processes are as follows.

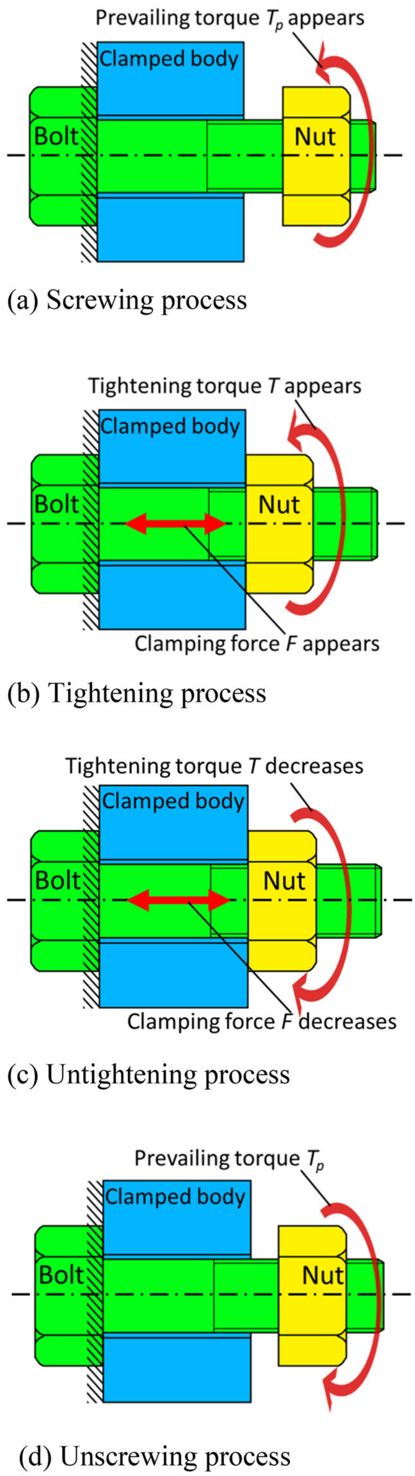
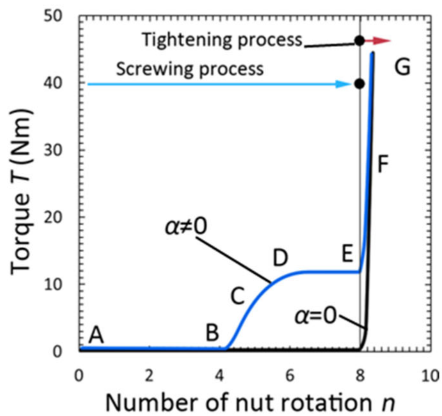
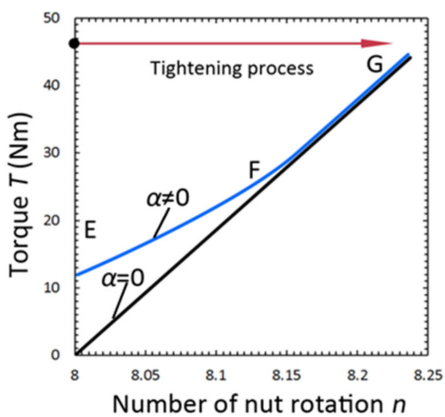


Figure 1. Schematic illustration for (a) screwing process (b) tightening process (c) untightening process and (d) unscrewing process.



(a) Variation of torque T in screwing and tightening process.



(b) Variation of torque T in tightening process.

Figure 2. Variation of torque due to the nut rotation in screwing and tightening process.

i. The screwing process in Fig. 1(a) whose experiment is based on JIS B 1056. Screwing the nut onto the bolt by torque wrench in Fig. 8 from starting position where the rotation cycle is 0 as shown in Fig. 3 Position A until the nut connects with clamped body as shown in Fig. 3 Position E, and measure the relationship between the prevailing torque T_p and the rotation cycles of nut n.

ii. The tightening process in Fig. 1(b) whose experiment is based on JIS B 1084. After screwing, the nut will be tightened by experimental device as shown in Fig. 9. The clamping force F in tightening process can be measured by the sensor in experimental device. In addition, the friction coefficient in thread surface μ_s, and bearing surface μ_w also can be measured. When the tensile stress generated in the bolt reaches the yield stress $\sigma_y = 800\,\text{MPa}$, the clamping force in bolt is $F_{100\%} = 68\,\text{kN}$. And the clamping force $F_{25\%} = 16.8\,\text{kN}$ of normal nut can be found from experimental result when the tensile stress in the bolt is 25% (200 MPa) of the yield stress. Here, the reason why the tightening force of 25% of the yield stress is selected is that the torque T changes at Position F (see Fig. 2b). From the experimental

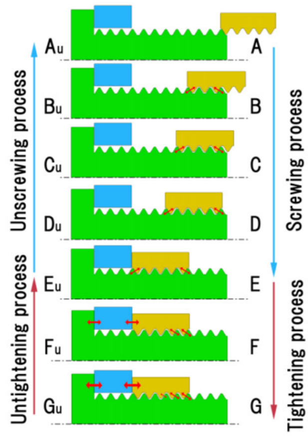


Figure 3. Screwing and tightening process of nut.

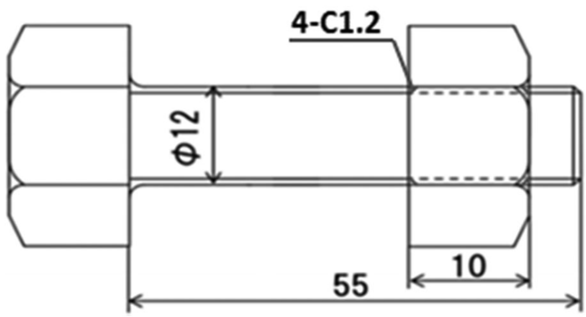


Figure 4. M12 Bolt-nut specimen (mm).

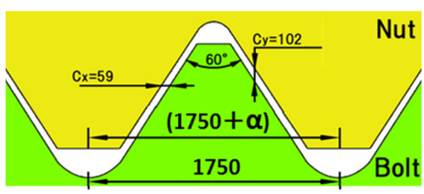


Figure 5. Pitch difference and clearance between bolt and nut threads (μm).

result of tightening of normal nut, it can be seen that in order to generate clamping force $F_{25\%}$, the needed tightening torque $T_{25\%} = 45\,\text{Nm}$. The $T_{25\%} = 45\,\text{Nm}$ is selected as maximum tightening torque for $\alpha = 35\,\mu\text{m}$, $40\,\mu\text{m}$ and $50\,\mu\text{m}$.

iii. The untightening process as shown in Fig. 1(c). The untightening torque in untightening process cannot be measured by the experimental device, so it will be measured by torque wrench. Whenever the torque reduces by 10 Nm, the clamping force F should be counted once on the experimental device until the nut separates with the clamped body.

iv. The unscrewing process as shown in Fig. 1(d). The measurement method of unscrewing process is same with screwing process.

The molybdenum disulfide grease spray PRO (manufactured by Azette Co., Ltd.) is used on thread surface only as the lubricating oil.

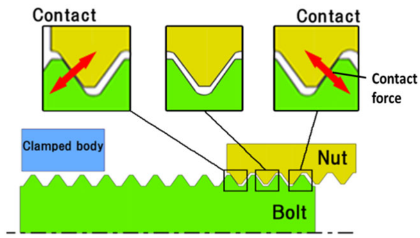


Figure 6. Contact status when the prevailing torque appears between bolt and nut.

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

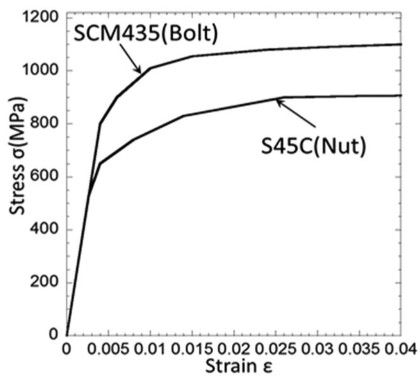


Figure 7. Stress strain relation for SCM435 and S45C.

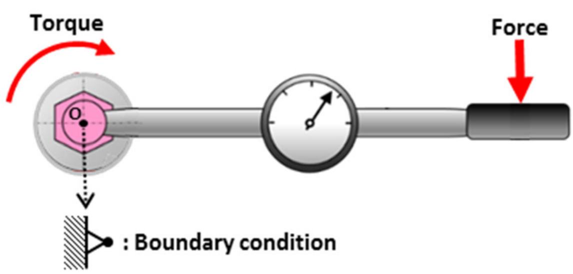
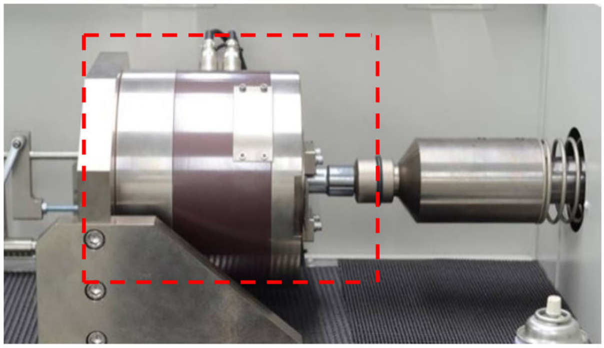


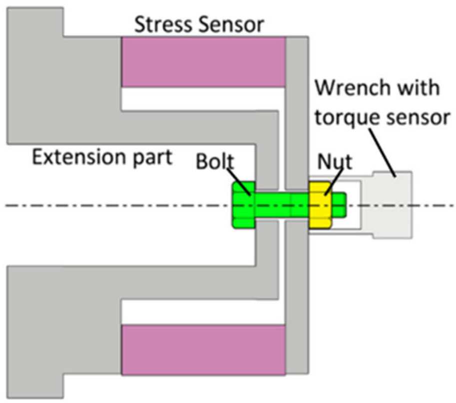
Figure 8. Torque measuring method and device (Torque wrench).

3. Analytical method and results in comparison with the experiment

Figure 10(a) shows the FEM mesh of the bolt-nut connection constructed for FEM software ANSYS 16.2. The elements are 3 D tetrahedron solid element. Because to express the spiral shapes of the bolt and nut thread is important, the minimum element in the spiral part of the thread is 0.048 mm. As shown in the appendix, the bolt axial force obtained by the 3D model in Fig. 10 is in good agreement with the axi-symmetric results. It is confirmed that prevailing torque T_p does not change when the element size smaller than 0.048 mm. The number of elements is about 80000, and the number of nodes is about 151000. To simplify the model, the hexagonal part of



(a) Photo of tightening experiment device.



(b) Schematic illustration of tightening experiment device.

Figure 9. Nut tightening experiment device based on JIS B 1084.

bolt head and nut are replaced by the cylindrical shape, and half of the clamped body is omitted. Penalty contact solution method and material non-linearity are considered in this analysis. Figure 10(b) shows the boundary conditions; the bolt head and the left surface of clamped body are fixed and a rotation angle $\pm\theta$ is applied to the nut. The friction coefficient can be measured by the tightening experiment device shown in Fig. 9. The experimentally obtained friction coefficient is in the range $\mu_s = 0.11–0.14$ at the thread surface and in the range $\mu_w = 0.16–0.18$ at the bearing surface. From the reference (The Japan Research Institute for Screw Threads and Fasteners 1982; Sakai, 1978), when the molybdenum disulfide grease spray is used on thread surface only, the friction coefficient is $\mu_s = 0.12$ at thread surface, and the friction coefficient is $\mu_w = 0.17$ at bearing surface. Those two friction coefficients are used in the following analyses.

The FEM analysis is performed from the Position B in Fig. 3 where the prevailing torque starts to occur. In the analysis the nut is rotated through Position B→C→D→E→F→G, and then the nut is rotated to the reverse direction through Position G^u→F^u→E^u→D^u→C^u→B^u. The whole analysis process is carried out continuously.

Figure 11 shows the torque variation in the screwing process, tightening process, untightening process and unscrewing process when the pitch difference when $\alpha = 35\,\mu m$, $\alpha = 40\,\mu m$, $\alpha = 50\,\mu m$. In Fig. 11, the analytical results are in good agreement with the experimental results in the screwing process. In the unscrewing process, the analytical results are larger than the experimental result especially when $\alpha = 50\,\mu m$. This is because when $\alpha = 50\,\mu m$ the larger contact stress causes

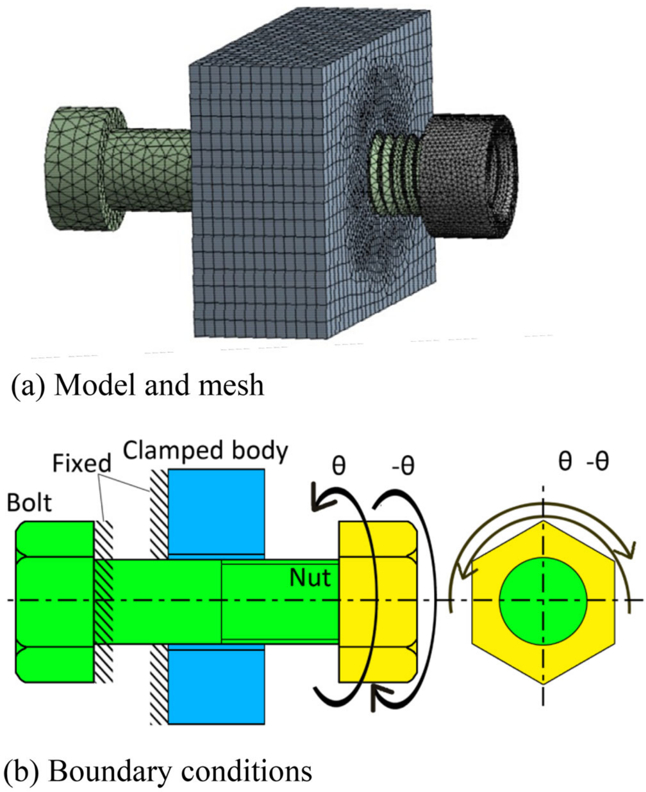


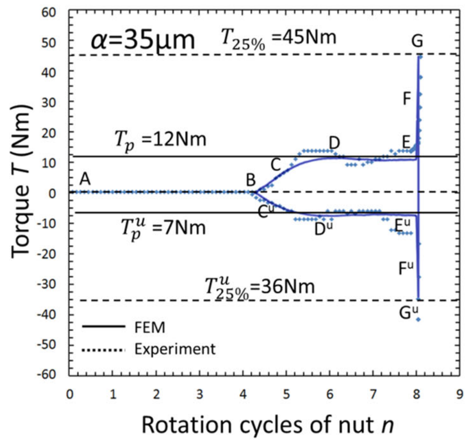
Figure 10. FEM model and boundary conditions for tightening and untightening process.

wear on the thread surface but the wear cannot be considered in the analysis. Figure 12 shows the bolt thread surface observation for the nut position D in Fig. 11(d). As shown in Fig. 12(a), when $\alpha = 0\,\mu$m, no scratch can be seen on the thread surface. In Figs. 12(b)–12(d), scratched wear can be seen. When $\alpha = 50\,\mu$m, wear debris can be seen and such large amount of wear causes torque reduction. In Fig. 2, the blue line shows the analytical result of tightening torque T and the nut rotation n relation for pitch difference $\alpha = 35\,\mu$m. The result of $\alpha = 35\,\mu$m coincides with the result of normal nut $\alpha = 0$ at Position F→G. The analysis verifies that at Position F the nut threads contact status becomes completely the same as the contact status of the normal nut (see Fig. 3 Position F→G)

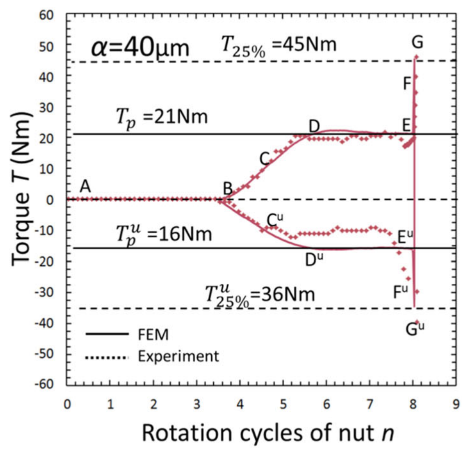
4. Prevailing torque T_p and residual prevailing torque T_p^u

Figure 13 shows the equivalent stress σ_{eq} for $\alpha = 35\,\mu$m. Figure 13(a) shows σ_{eq} when the tightening process starts. In Figure 13(a), since σ_{eq} is smaller than nut yield stress $\sigma_y = 530\,\text{MPa}$ for most region, plastic region is very limited as shown in the red color region. Figure 13(b) shows σ_{eq} when the nut is tightened under tightening torque $T_{25\%} = 45\,\text{Nm}$ with clamping force $F_{25\%} = 16.8\,\text{kN}$, which is corresponding to 25% of the bolt yield stress $\sigma_y = 800\,\text{MPa}$. In Figure 13(b), since σ_{eq} is larger than nut yield stress $\sigma_y = 530\,\text{MPa}$ as shown in the red color region, the plastic region becomes larger. The residual prevailing torque smaller than prevailing torque can be explained from the plastic deformation during the tightening process.

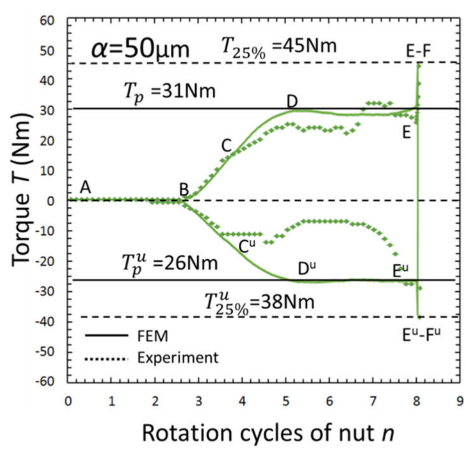
Figure 14 illustrates the prevailing torque T_p -α relation and the residual prevailing torque T_p^u-α relation obtained by applying FEM when the pitch difference $\alpha = 25$, 30, 35, 40, 45, 50 μm.



(a) α=35 μm

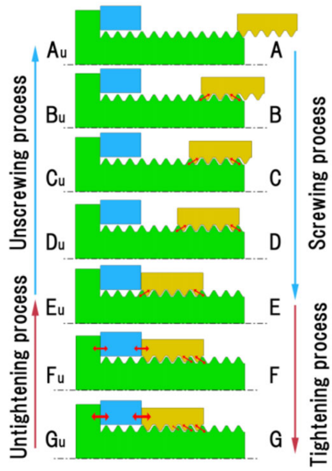


(b) α=40 μm



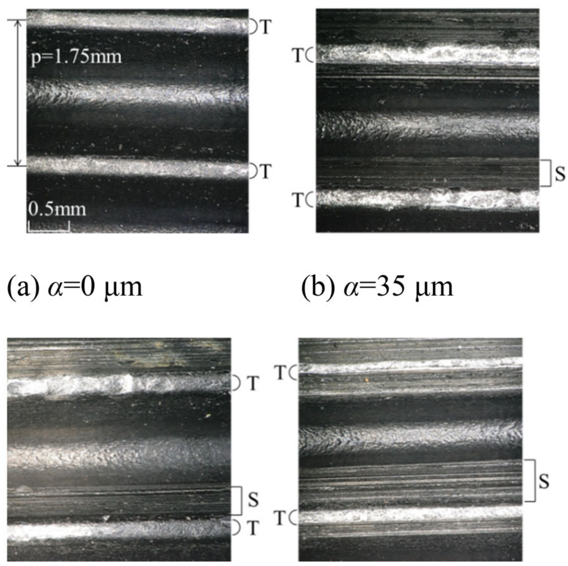
(c) α=50 μm

Figure 11. Variation of torque in screwing, tightening, untightening and unscrewing process of nut.



(d) Screwing and tightening process of nut.

Figure 11. Continued.



(a) α=0 μm (b) α=35 μm

(c) α=40 μm (d) α=50 μm

Figure 12. Bolt thread surface after screwing experiment (S: Scratch wear. T: Top of thread.).

Those curves in Fig. 14 are affected by the plastic deformations at bolt threads due to the tightening torque and the pitch difference α especially when α is large. It can be seen that the residual prevailing torque T_p^u increases as well as the prevailing torque T_p and with the increasing of pitch difference α, and for each pitch difference α the prevailing torque T_p is always larger than its residual prevailing torque T_p^u.

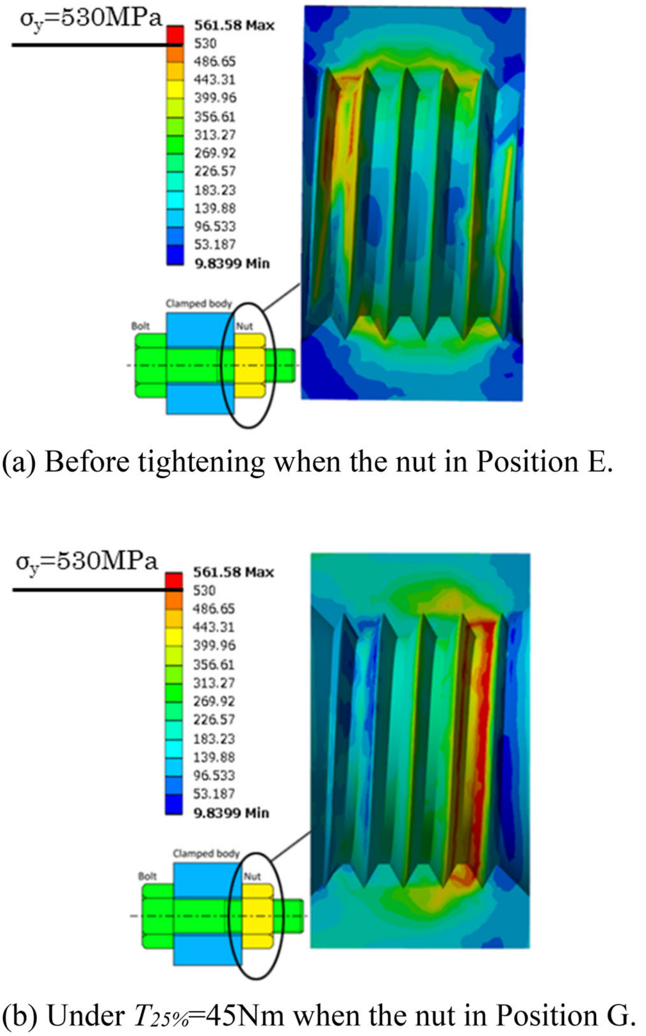


Figure 13. Equivalent stress σ_{eq} before tighten and when $\alpha = 35\mu$m nut is tightened under tightening torque $T = T_{25\%}$.

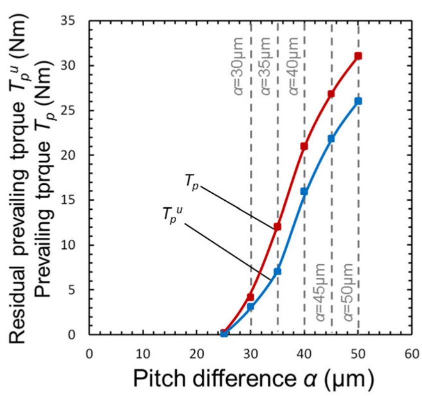


Figure 14. Prevailing torque T_p and residual prevailing torque $T_p{}^u$ vs pitch difference α.

5. Conclusions

It is well known that the prevailing torque for special bolt-nut connections is useful for evaluating anti-loosening performance. In this study, several slight pitch differences were introduced between the bolt-nut connections toward realizing anti-loosening performance. By applying the three-dimensional FEM, the nut screwing process was analyzed to obtain the prevailing torque T_p. Since the nut unscrewing properties after the nut tightening is more important to prevent the nut self-loosening, the nut unscrewing is also analyzed to obtain the residual prevailing torque T_p^u. The conclusions can be summarized in the following way.

1. The tightening torque T and the nut rotation n relation was investigated. Although the T-n relation varies depending on α, the T-n relation of $\alpha \neq 0$ coincides with the T-n relation of normal nut $\alpha = 0$ when the nut is tightened to Position F as shown in Fig. 2(b). This is because at Position F the nut threads contact status becomes completely the same as the contact status of the normal nut (see Fig. 3).
2. The prevailing torque T_p obtained by the FEM analysis is in good agreement with the experimental result (see Fig. 11). Also, the residual prevailing torque T_p^u obtained by FEM is in good agreement with the experimental result after the nut is untightened. The residual prevailing torque T_p^u is always smaller than the prevailing torque T_p because of the plastic deformation on nut thread.
3. With increasing the pitch difference, the residual prevailing torque T_p^u increases as well as the prevailing torque T_p (see Fig. 13). The prevailing torque T_p is in the range $T_p = 12$–31 Nm and the residual prevailing torque T_p^u is in the range $T_p^u = 7$–26 Nm depending on the pitch difference α. Those results suggest that suitable pitch difference may provide good anti-loosening performance.

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Appendix: Bolt axal force obtained by axi-symmetric analysis and three-dimensional analysis

In this study, prevailing torque which cannot be treated by the axi-symmetric analysis (Chen et al. 2015, 2016; Noda et al. 2016) was discussed by the three-dimensional analysis. To compare both results, a bolt axial force can be focused in this appendix. Since the nut pitch is slightly larger than the bolt pitch, the bolt axial force F_α appears in the axial direction as shown in Fig. A1 between the bolt threads. The previous study showed that this F_α is closely related to prevailing torque T_p (Noda et al. 2016). It should be noted that F_α in the screwing is different

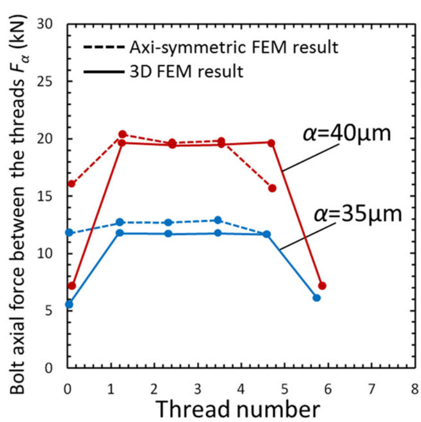
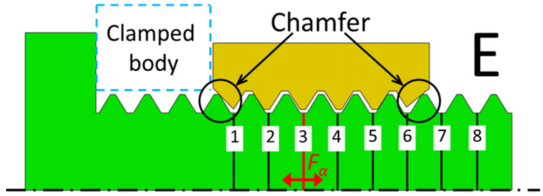
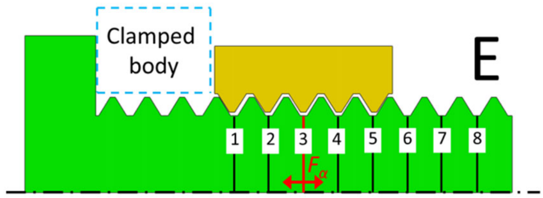


Figure A1. Bolt axial force appearing between the threads F_α in the screwing process.



(a) 3D model having chamfer.



(b) Axi-symmetric model without considering chamfer.

Figure A2. Nut position E and thread number in Fig. A1.

from the clamping force F in the tightening indicated in Fig. 1(b). The bolt axial force F_α between the bolt threads occurs from the accumulation of pitch difference in the screwing process. Figure A1 shows the axial force between bolt threads F_α for $\alpha = 35$, $40\,\mu$m, and whose position is illustrated in Fig. A2. Position E is where the nut is screwed onto the bolt 8 cycles where the nut is supposed to touch the clamped body.

If the bolt axial force F_α is larger, the prevailing torque T_p is larger, which may contribute better anti-loosening performance. As shown in Fig. A1, the bolt axial forces corresponding to the most outer nut threads become smaller than that in the middle part. This is due to the secondary outer nut threads being also in contact as well as the most outer nut threads. From the comparison between the axi-symmetric and the 3 D results, it is found that the maximum values coincide with each other within 10% difference. Figure A1 illustrates that the present prevailing torque is reliable and useful for discussing the anti-loosening performance.