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Three-dimensional finite element analysis for prevailing torque of bolt-nut connection having slight pitch difference

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Abstract In a vast industrial field, the bolt-nut connection is widely used and unitized as an important machine component. In order to improve safety and for cost reduction, cheap bolt-nut connections with excellent anti-loosening performance are always needed. In this paper, the slight pitch difference is considered between the nut and bolt threads to improve the anti-loosening by focusing on the prevailing torque T_p. In the first place, during the nut screwing process, the bolt axial force F_α caused by the pitch difference is examined because previously F_α was studied by the axisymmetric FEM simulation as a substitute of the prevailing torque T_p. Next, the prevailing torque T_p is directly analyzed by changing the pitch difference through the 3D FEM simulation. The obtained prevailing torque needed for the nut rotation is in consonance with the measured value experimentally. By considering manufacturing error sometimes included, finally, the real pitch difference α of the nut is identified from the number of nut thread in contact with the bolt thread, which is measured experimentally. Then, the correctness of this α is confirmed through the prevailing torque obtained by 3D FEM simulation.

1. Introduction

The bolt-nut connections are vital joining components vastly used to link mechanical components conveniently at low cost. The history of bolt-nut connection was fully reviewed in Ref. [1]. To ensure the safety of the connected structures, excellent anti-loosening performance and high fatigue strength are always needed. A lot of researches have been done mainly concentrating on anti-loosening performance [2-10], and several researches have contributed to the fatigue strength improvement [11-21]. Usually, anti-loosening performance in bolt-nut connections cannot be achieved without diminishing fatigue strength and without increasing the cost. Most of the special bolt-nut connections have either several additional components or complex geometry, resulting to a complicated manufacturing process and high market price, which is normally more than three times higher compared to the normal bolt-nut connection. This study therefore concentrates on the pitch difference between bolt and nut to improve both anti-loosening performance and fatigue strength. The proposed bolt-nut connection in this study can be produced in the same way as the normal bolt-nut connection. The cost is expected to be about 1.5 times of the cost of the normal bolt-nut connection by considering the adjustment of thread tap pitch and the inspection procedure on the pitch difference.

Our previous experimental studies have clarified that a suitable pitch difference may improve the fatigue life and also the anti-loosening performance [22-24]. Our previous analytical studies also have expressed the fatigue life improvement by applying the finite element method (FEM), but they cannot express the nut rotation because the axi-symmetric FEM modeling is applied without considering thread spirals [22-24]. Therefore, in this study, a three-dimensional finite element method (3D FEM) will be applied to the nut screwing process.

In the first place, the 3D FEM simulation will be compared with the axi-symmetric FEM simulation used previously in terms of the bolt axial force caused by the pitch difference. Then, the

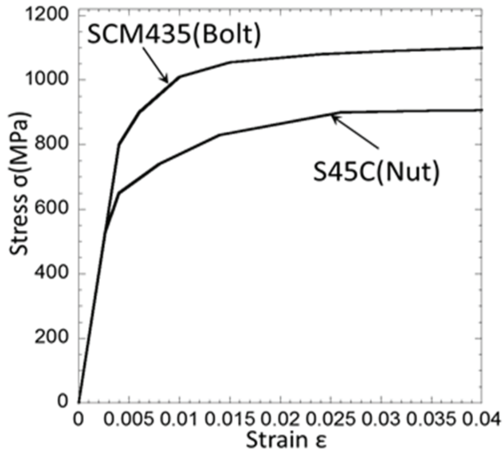


Fig. 1. Stress-strain relation for SCM435 and S45C.

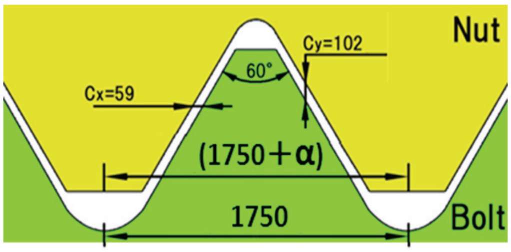


Fig. 2. Pitch difference and clearance between M12 bolt and nut threads (unit: μm).

validity of the analyses will be confirmed in the screwing process. Next, the 3D FEM simulation will be compared with the experimental study in terms of the prevailing torque characterizing the anti-loosening performance. Then, the usefulness of the slight pitch difference will be confirmed experimentally and analytically. Finally, how to correct the manufacturing error of pitch difference will be discussed with confirming the validity from the prevailing torque in the experiment and FEM simulation. It will be shown that 3D FEM simulation is always in good agreement with the experiment and a few exceptions may be corrected as shown in this last example.

2. Three-dimensional FEM simulation during the nut screwed onto the bolt

In this study, the Japanese Industrial Standard (JIS) M12 and M10 bolt-nut connections are used. The strength grade is 8.8. Table 1 shows material properties of bolt-nut materials JIS SCM435 and JIS S45C quenched and tempered. Fig. 1 shows the stress-strain curves of those materials. In Secs. 3 and 4, JIS M12 bolt-nut thread will be considered, which usually has the same pitch 1750 μm, but in this study, the nut pitch is presumed to be equal or slightly larger than the bolt pitch as shown in Fig. 2. Three types of pitch difference α = 30 μm, α = 40 μm and α = 50 μm will be discussed with the clearance in the x-direction C_x = 59 μm will be assumed.

As shown in Figs. 3 and 4, during the screwing process of

Table 1. Material properties of bolt and nut.

	Young's modulus E (GPa)	Poisson's ratio v	Yield stress σ_y (MPa)	Tensile strength σ_B (MPa)
SCM435 (bolt)	206	0.3	800	1200
S45C (nut)	206	0.3	530	980

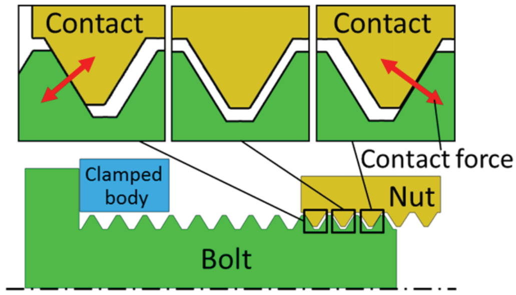


Fig. 3. Contact status when the prevailing torque appears between bolt and nut.

the nut, so called prevailing torque T_p occurs even before the nut contacts with the clamped body [3-5]. Note that tightening torque T is different from prevailing torque T_p as shown in Fig. 4. The tightening torque T is defined in the tightening process in Fig. 4(b) after the nut comes into contact with a clamped body and the prevailing torque T_p is defined in the screwing process in Fig. 4(a) before the nut comes into contact with the clamped body. In this study, the 3D FEM simulation for the prevailing torque is compared with the experiment. This screwing experiment is conducted under lubrication surface state of Molybdenum disulfide grease spray PRO (manufactured by Azette Co., Ltd.). In this study, the friction coefficient μ = 0.12 is used in the FEM simulation in accordance with the Ref. [25]. The validity of μ = 0.12 is also confirmed by the preliminary analysis for the screwing in Fig. 4(a).

Fig. 5 shows the three-dimensional mesh, and the FEM software ANSYS 16.2 is employed for analysis. To simplify the simulation, the hexagonal nut and bolt head are substituted by cylinders. The helical threads of the bolt and nut are subdivided into meticulous elements compared to the other parts [2]. Penalty contact solution method and material non-linearity are considered in this analysis. Boundary conditions are provided in the following way. The side surface of the bolt head is fixed as shown in Fig. 6. Then, a series of screwing angles θ are applied on the side surface of the nut. The start position of the FEM simulation is where the prevailing torque appears. Therefore, the start position varies relying on the pitch difference. For example, the start position of the α = 40 μm nut is 3.5 cycles of the nut rotation. And the end position of the simulation is where the nut is screwed into the bolt after 8 cycles of the nut rotation where the nut is supposed to touch the clamped body. The axisymmetric FEM analysis is also conducted in a similar way of the previous study [24].

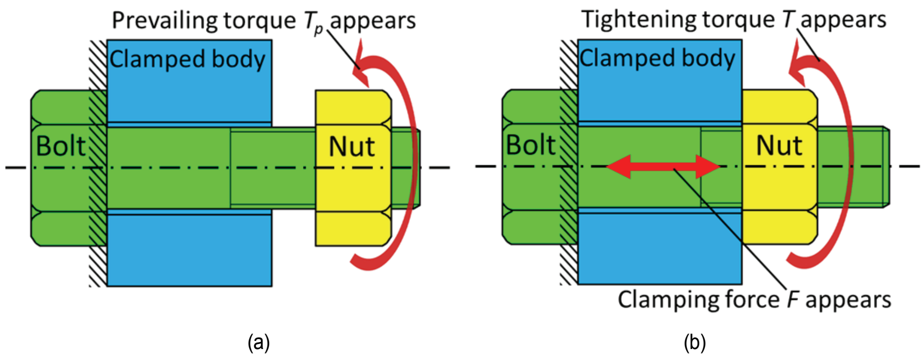


Fig. 4. Schematic illustration for (a) screwing process; (b) tightening process.

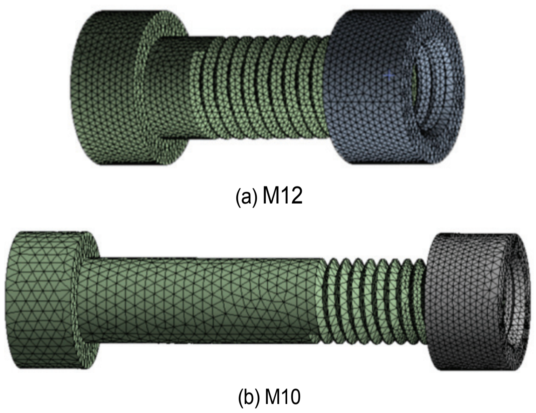


Fig. 5. FEM mesh model for M12 bolt-nut used in this study and M10 bolt-nut is discussed in appendix.

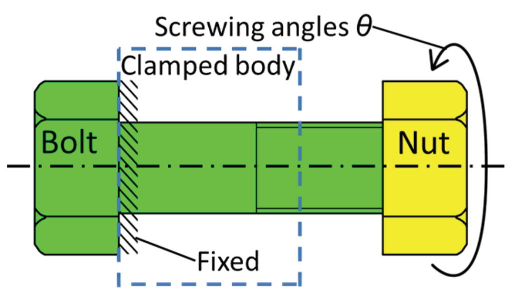


Fig. 6. Boundary conditions.

3. Three-dimensional FEM simulation to obtain bolt axial force between the threads

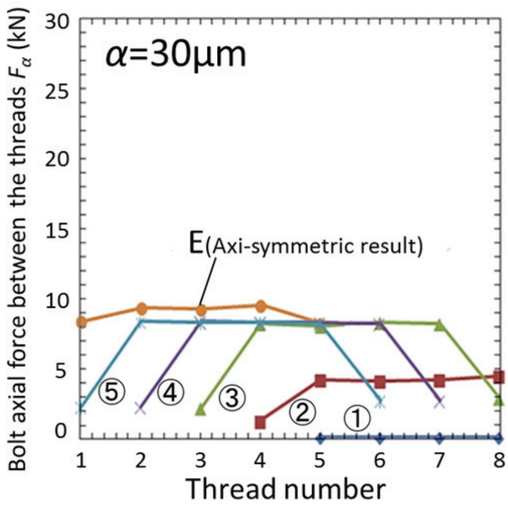
Since the pitch of nut is slightly larger than the pitch of bolt, a tensile force in the bolt axial direction F_α appears between the bolt threads as shown in Fig. 7(c). This bolt axial force F_α is closely related to prevailing torque T_p [24]. It should be noted that F_α is different from the clamping force F indicated in Fig. 4(b). The bolt axial force F_α between bolt threads occurs from the accumulation of pitch difference in the screwing process.

Fig. 7(a) shows F_α when α = 30 μm from position ① to position ⑤ whose positions are illustrated in Fig. 7(c). Position ① is where the nut is screwed onto bolt 4 cycles without appearing F_α and T_p. Position ② is where the nut is screwed 1 more cycle from position ①. When α = 30 μm, both F_α and T_p appear between positions ① and ②. Finally, position ⑤ is where the nut is screwed onto the bolt 8 cycles where the nut is supposed to touch the clamped body. From position ① to positions ②, ③, the whole nut is being screwed onto the bolt, and therefore the accumulated pitch difference affects the results. From position ③ to positions ④, ⑤, however, the pitch difference cannot be accumulated anymore since the whole nut thread is already screwed onto the bolt.

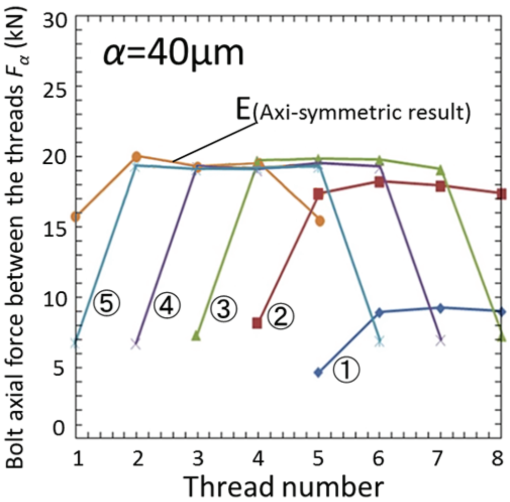
In a similar way, Fig. 7(b) shows F_α when α = 40 μm from position ① to position ⑤ illustrated in Fig. 7(c). When α = 40 μm in Fig. 7(b), even at position ①, the bolt axial force F_α appears since the nut threads have already contacted with the bolt threads different from when α = 30 μm. As shown in Figs. 7(a) and (b), the bolt axial force F_α of α = 40 μm is larger than F_α of α = 30 μm. If the bolt axial force F_α is larger, the prevailing torque T_p is also larger, which may contribute better anti-loosening performance. As shown in Figs. 7(a) and (b), during the nut being screwed onto the bolt, 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.

In Figs. 7(a) and (b), F_α distribution is symmetric except for ① and ②. This is because 3D FEM nut model considers the chamfer at both ends of the nut as shown in Fig. 7(c). Since there is chamfer at the left end of nut, at positions ① and ② F_α distribution is not symmetric at the both ends.

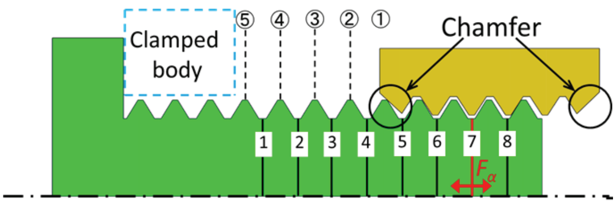
In Figs. 7(a) and (b), the orange line shows the axisymmetric FEM results when the nut at position E in Fig. 7(d). From the comparison between the axisymmetric and 3D results, it is shown that the maximum values coincide with each other within 10 % difference. In the previous study, because of the axi-symmetric FEM simulation without considering 3D spiral and nut rotation, the anti-loosening performance was discussed in terms of the bolt axial force F_α indirectly. In this study, therefore, 3D EFM analysis will be applied directly to calculating the prevailing torque T_p during the nut being screwed onto the bolt as well as the bolt axial force F_α.



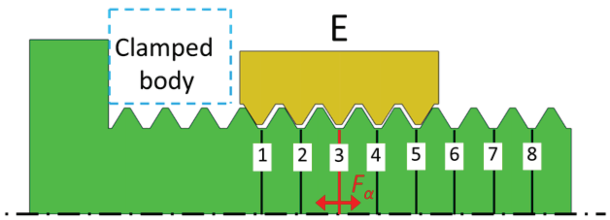
(a) α = 30 μm



(b) α = 40 μm



(c) Nut position ①, ②, ③, ④, ⑤ in Figs. 7(a) and (b) for 3D model having chamfer



(d) Nut position E in Figs. 7(a) and (b) for axi-symetric model without considering chamfer

Fig. 7. Bolt axial force appearing between the threads F_α in the screwing process for M12.

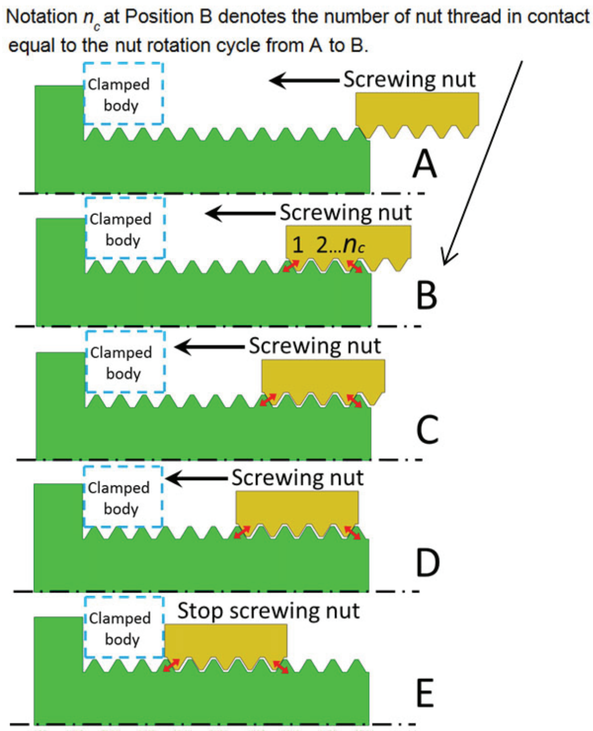


Fig. 8. Screwing process of nut (position A: Strat of screwing, position B: T_p just appears, position C: T_p is increasing, position D: T_p is saturated).

4. Three-dimensional FEM simulation to obtain prevailing torque

The author's previous experiment showed that under suitable pitch difference between bolt and nut, the anti-loosening performance can be improved significantly compared to the normal nut [24]. To evaluating anti-loosening performance, the prevailing torque can be used conveniently for special bolt-nut connections.

Fig. 8 schematically illustrates the nut screwing process from position A to position E in a different way from Fig. 7(c). Here, position A is defined as where the screwing starts with threads contact without appearing the prevailing torque T_p. Position B is defined as where the prevailing torque T_p just appears. At position B, notation n_c is defined as the number of nut threads in contact except n_c = 1 when T_p just appears as shown in Fig. 8 [24]. Position C is a sample position where the prevailing torque T_p is increasing. Position D is defined as where the nut is completely screwed onto the bolt and therefore the prevailing torque T_p becomes saturated. Position E is defined as where the screwing ends since the nut is touching the clamped body. Note that the geometrical locations of positions A, B and D vary depending on the pitch difference α.

Fig. 9 illustrates the relation between the prevailing torque T_p and the number of nut rotation n for α = 40 μm. Here, the red line shows the prevailing torque T_p^{FEM} obtained by 3D simulation.

In comparison, the black line shows the experimentally obtained prevailing torque T_p^{Exp} measured at the interval of 45

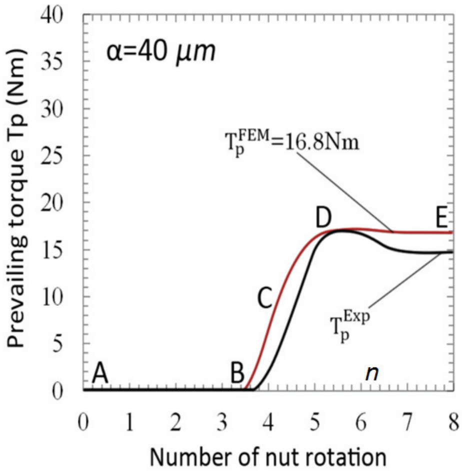


Fig. 9. Prevailing torque T_p in the screwing process.

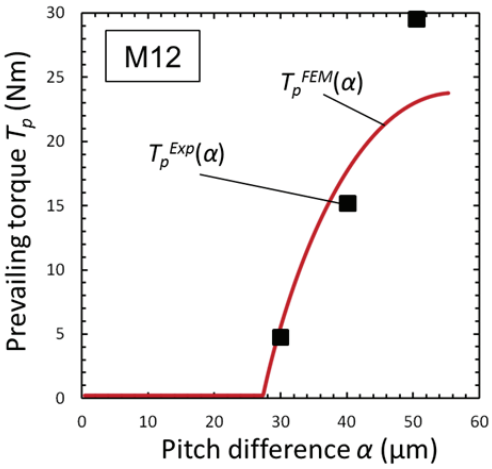


Fig. 10. Prevailing torque vs. pitch difference of M12.

degree of the nut rotation angle by using torque wrench under the fixed bolt head. As shown in Fig. 9, the prevailing torque appears at n = 3.5 times of the nut rotation when the nut is screwed onto the bolt. Then, the prevailing torque initially increases with increasing the nut rotation and becomes saturated during positions D and E as T_p^{FEM} = 16.8 Nm. The prevailing torque may be predicted by 3D FEM simulation by varying the pitch difference α without doing experiment. The prevailing torque T_p^{FEM} obtained by FEM is in good agreement with the experimental result.

The small difference between T_p^{FEM} and T_p^{Exp} can be explained in the following way. In the experiment, the bolt nut contact threads are worn down during the screwing process; and therefore, the frictional condition such as the frictional coefficient is changing through generating and dropping out of the wear debris. However, the friction coefficient μ = 0.12 used in FEM has to be fixed in the screwing process due to the restriction of the software. This is the reason why the FEM prevailing torque is almost constant between positions D to E.

Fig. 10 shows the relationship between the prevailing torque and pitch difference α of M12 bolt-nut connections. The red line shows 3D FEM results T_p^{FEM} (α) and solid square (■) shows experimental results. The results of FEM coincide with the experimental results well except for α = 50 μm. The difference of α = 50 μm can be explained from frictional condition severity due to large α.

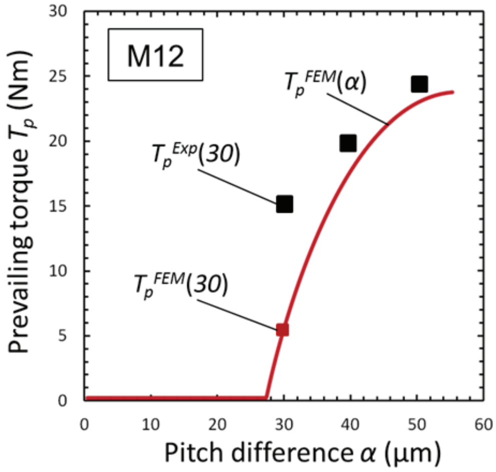


Fig. 11. Prevailing torque vs. pitch difference of M12 manufactured in a different way of Fig. 10.

5. Three-dimensional FEM simulation to detect and correct manufacturing process error in the pitch difference

5.1 Manufacturing process error in the pitch difference

The error of the pitch difference is depending on the manufacturing method. Fig. 11 shows another relationship between the prevailing torque T_p and pitch difference α of M12 bolt-nut connections manufactured in a different way of Fig. 10. The red line shows the prevailing torque T_p^{FEM} (α) obtained by 3D FEM simulation in comparison with the experimental results indicated by solid square (■). It is seen that most of the experimental results are in good agreement with T_p^{FEM} (α) although T_p^{Exp} (30) is different from T_p^{FEM} (30). This is due to a manufacturing process error of the pitch difference of the nut. To confirm and correct this, the following experiment is conducted.

5.2 How to detect and correct manufacturing process error from the number of nut thread in contact

In Fig. 8 in Sec. 4, notation n_c is defined as the number of nut thread in contact except n_c = 1. In Ref. [24], number n_c is discussed by using the axi-symmetric modelling. In this study, number n_c is obtained by using 3D CAD and denoted by n_c^{Real} (α) by calculating the nut rotation from positions A to B in Fig. 8. By varying the pitch difference α, Fig. 12 shows the number of nut thread in contact $n_c^{Real}(\alpha)$ obtained by 3D CAD as the red line.

Table 2. Correction of pitch difference α of M12.

Target α	Real α'	$n_c^{Exp}(\alpha) = n_c^{Real}(\alpha')$
30	34	3.5
40	39	3.1
50	50	2.4

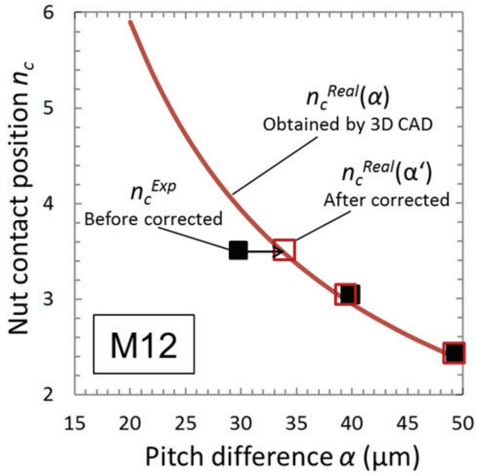


Fig. 12. Number of nut thread in contact n_c of M12.

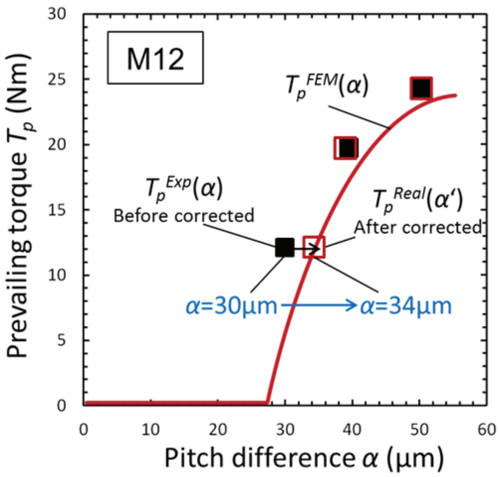


Fig. 13. Comparison of prevailing torque $T_p^{Exp}(\alpha)$, $T_p^{Real}(\alpha)$ and $T_p^{FEM}(\alpha)$.

To compare with $n_c^{Real}(\alpha)$, experimentally obtained $n_c^{Exp}(\alpha)$ is also plotted as the solid square (■) in Fig. 12. To obtain $n_c^{Exp}(\alpha)$ in Fig. 12, a dial type torque wrench whose product name TOHNICHI DB 50N is used during the nut being screwed onto the bolt. Then, $n_c^{Exp}(\alpha)$ is defined as the nut rotation number when the prevailing torque T_p^{Exp} = 0.1 Nm appears. In Fig. 12, open red square (□) denotes the corrected result $n_c^{Real}(\alpha')$ by using 3D CAD.

As shown in Fig. 12, a difference can be seen between $n_c^{Exp}(30)$ = 3.5 and $n_c^{Real}(30)$ = 4.0. This is because the bolt-nut connection experimentally used as α = 30 μm has some manufacturing error regarding the pitch difference α. Therefore, $n_c^{Exp}(30)$ = 3.5 should be regarded as $n_c^{Real}(34)$ = 3.5. This is because they should have the same number of nut thread in contact. In other words, the target α = 30 μm should be regarded as α' = 34 μm in reality.

5.3 Confirmation of real pitch difference from T_p^{FEM}

Fig. 13 shows the relationship between the prevailing torque $T_p(\alpha)$ and pitch difference α. In Fig. 13, the solid black square (■) denotes experimentally obtained prevailing torque $T_p^{Exp}(\alpha)$ during position D-E in Fig. 8. The red line shows the corresponding saturated prevailing torque $T_p^{FEM}(\alpha)$ analytically obtained. The open red square (□) denotes real prevailing torque $T_p^{Real}(\alpha')$. As shown in Table 2, the real pitch difference α' can be obtained from $n_c^{Real}(\alpha')$, and real prevailing torque $T_p^{Real}(\alpha')$ has the same value with $T_p^{FEM}(\alpha)$ before corrected. The validity of α' can be confirmed from the coincidence of $T_p^{Real}(\alpha')$ and $T_p^{FEM}(\alpha)$.

As shown in Fig. 13(a), for M10 a difference can be seen between $T_p^{FEM}(30)$ is different from $T_p^{Exp}(30)$. However, after $T_p^{Exp}(30)$ is corrected to $T_p^{Real}(34)$ by using $n_c^{Real}(34)$, they are in good agreement. In a similar way, all values of $T_p^{Exp}(\alpha)$ denoted by the solid black square (■) can be changed to $T_p^{Real}(\alpha)$ denoted by open red square (□) as shown in Fig. 13(a). It is confirmed that real α can be obtained in terms of the prevailing torque. A large manufacturing error of the pitch difference can be found by confirming the prevailing torque as well as by confirming the number of nut thread in contact n_c.

6. Conclusions

In this paper, to improve the anti-loosening performance with low cost, a slight pitch difference was considered between bolt and nut threads by focusing on the prevailing torque T_p. A three-dimensional FEM simulation was applied to the nut being screwed onto the bolt by changing the pitch difference. The conclusions can be summarized in the following way.

(1) During the nut being screwed onto the nut, the bolt axial force F_α was obtained through the axisymmetric FEM and 3D FEM simulations. The results are in good agreement within 10 % difference. Since the bolt axial force F_α is closely related to the prevailing torque T_p, simple axisymmetric simulations can be used conveniently in some practical situations.

(2) During the screwing process of the nut, the prevailing torque T_p was directly analyzed by the 3D FEM simulation. The FEM results are in good agreement with the experimental results in most cases. In this simulation, the prevailing torque T_p appears after the nut thread is in contact with the bolt thread. Then, T_p increases with nut rotation, and finally reaches a certain saturated value when all the nut threads are screwed onto the bolt. Larger pitch difference produces larger T_p, which may contribute anti-loosening performance.

(3) Some manufacturing process errors in the pitch difference may be detected and corrected by measuring the prevailing torque T_p as well as the number of nut thread in contact n_c.

In this study, the real pitch difference α was obtained from the number of nut thread in contact n_c by using 3D CAD and the correctness of this α was confirmed from the prevailing torque by using 3D FEM.

Nomenclature

α : Pitch difference
C_x : Clearance in the x-direction
μ : Friction coefficient
T_p : Prevailing torque
F_α : Bolt axial force between the threads
n_c : The number of nut threads in contact
T_p^{FEM} : Prevailing torque obtained by 3D FEM
T_p^{Exp} : Prevailing torque obtained by experiment
T_p^{Real} : The real prevailing torque
n_c^{Exp} : The number of nut threads in contact obtained by experiment.
n_c^{Real} : The number of nut threads in contact obtained by CAD

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Table A.1. Correction of pitch difference α of M10.

Target α	Real α'	$n_c^{Exp}(\alpha) = n_c^{Real}(\alpha')$
20	26	4.25
30	27	4.00
35	33	3.15
40	36	2.90
45	43	2.40

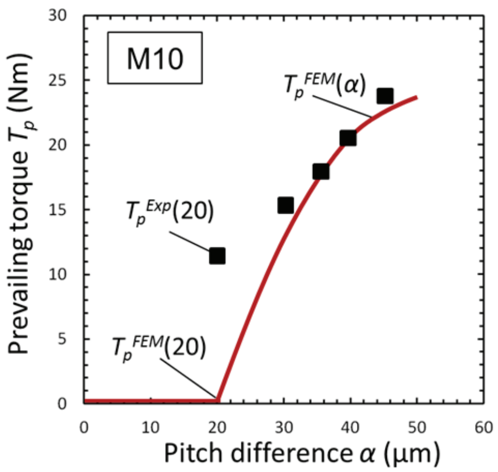

Fig. A.1. Prevailing torque vs. pitch difference of M10.

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Appendix

In this paper JIS M12 bolt-nut manufacturing error was discussed but other bolt-nut dimensions can be treated similarly. This appendix deals with JIS M10 bolt-nut usually having the pitch 1500 μm. Here, five types of pitch difference α = 20 μm, α = 30 μm, α = 35 μm, α = 40 μm and α = 45 μm are discussed with the clearance in the x-direction C_x = 60 μm.

Fig. A.1 shows the relationship between the prevailing torque T_p and pitch difference α of M10 bolt-nut connections. It is seen that most of the experimental results are in good agreement with $T_p^{FEM}(\alpha)$ although $T_p^{Exp}(20)$ is different from $T_p^{FEM}(20)$. Fig. A.2 shows the number of nut thread in contact $n_c^{Real}(\alpha)$ obtained by 3D CAD as the red line of M10. By using the same method in Fig. A.2, M10 nut is also corrected. Table A.1 shows the corrected pitch difference α of M10 with $n_c^{Exp}(\alpha) = n_c^{Real}(\alpha')$.

Fig. A.3 shows the relationship between the prevailing torque $T_p(\alpha)$ and pitch difference. The corrected pitch difference of M10 is also confirmed by using same method in Fig. 13. For M10, although small differences can be seen, the results $T_p^{Real}(\alpha')$ and $T_p^{FEM}(\alpha)$ are in good agreement.

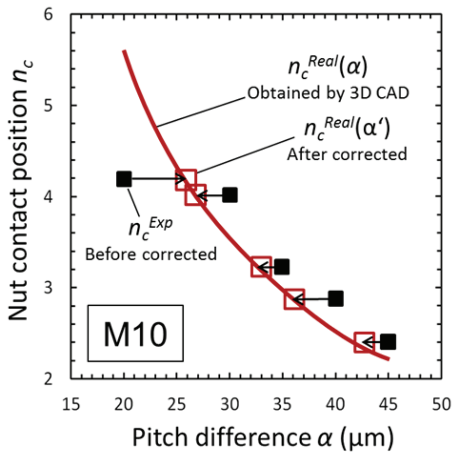

Fig. A.2. Number of nut thread in contact n_c of M10.

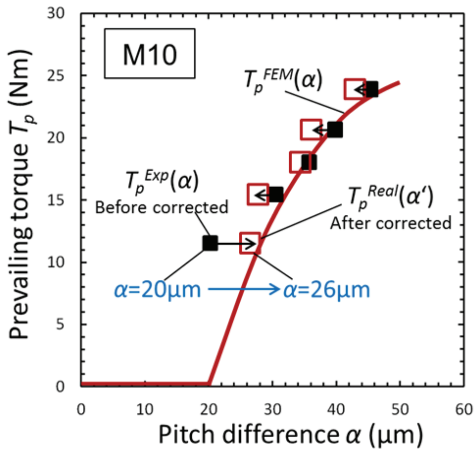

Fig. A.3. Comparison of prevailing torque $T_p^{Exp}(\alpha)$, $T_p^{Real}(\alpha)$ and $T_p^{FEM}(\alpha)$.

Nao-Aki Noda received his Ph.D. degree in Mechanical Engineering from Kyushu University, Japan in 1984. He has been doing research and teaching at Kyushu Inst. Tech., Kitakyushu, Japan, 1984-present. He is an author of Theory of Elasticity useful for engineers and a co-author of Safety Engineering for Workers in Industry and other several books. He is a co-editor of Stress Intensity Factors Handbook, Vol. 4 & 5, Advances in Finite Element Analysis for Computational Mechanics. He is a recipient of Outstanding Paper Medal of Japan Soc. Tech. Plasticity, Sokeizai Industry Technology award from the Materials Process Tech. Ctr., a fellow of JSME (Japan Soc. Mech. Engrs.) and a fellow of JSAE (Soc. Automotive Engrs. Japan). He also received JSMS Academic Contribution Award from Japan Soc. Material Science and JSME Materials and Mechanics Division Award from Japan Soc. of Mechanical Engineers. His achievements include researches in stress analysis for notched material testing specimens, and development for large ceramics structures used for steel manufacturing machinery.