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Stress Reduction Effect and Anti-Loosening Performance of Outer Cap Nut by Finite Element Method*

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Abstract

Previously several kinds of anti-loosening bolts and nuts were invented. However, they usually need a certain amount of prevailing torque even before the nut touches a clamped member. A new outer cap nut named "Super loose proof (SPR)" has been developed to overcome such inconvenience. At first this outer cap nut can be rotated smoothly by hand until the nut touching the clamped member. After fastening the outer cap nut, anti-loosening performance can be realized by deforming the outer cap and producing thread contact force at the outer cap region. In this study, stress concentration and tightening-loosening behavior are analyzed by axi-symmetric and three-dimensional finite element methods. Under a certain bolt-axial force, the load distribution of the first thread decreases more than 12% with increasing initial clearance of outer cap nut. Stress concentration appearing at the first thread of the bolt is about 10% smaller than that of conventional nut, reflecting the increase of the thread contact force at the outer cap region. On the other hands, it is found that anti-loosening performance of SPR can be realized when the outer cap has high yield stress.

Key words: Finite Element Method, Stress Concentration, Contact Problem, Fixing Element, Machine Element, Bolted Joint, Screw Thread

1. Introduction

Threaded fasteners are widely used to assemble mechanical products and structure because of their convenience and low cost. However, self-loosening often occurs when the threaded fasteners are subjected to dynamic external loads, such as impacts and vibrations(1). In order to prevent such self-loosening, some anti-loosening bolts and nuts were developed(2-6). However, for example, double nuts need two tightening processes, and another example, Super Slit Nut needs a certain amount of torque by using wrenches even before the nut touches clamped member(6). At the top of high tower, therefore, it is

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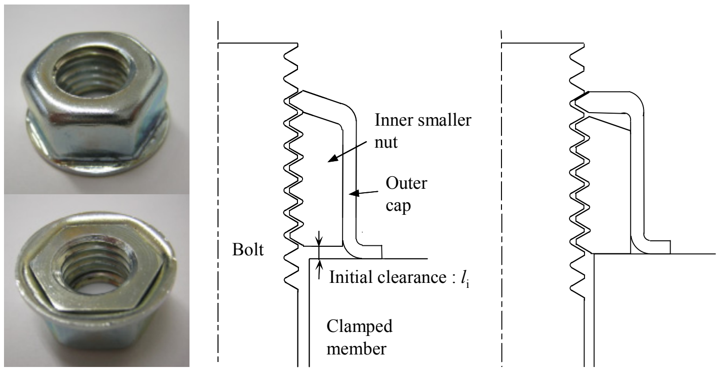

(a) photograph (b) before fastening (c) after fastening

Fig. 1 Super Loose Proof (Anti-loosening outer cap nut)

inconvenient to use such special threaded fasteners.

As a new generation of anti-loosening nut, a new Outer Cap Nut named Super Loose Proof, has been developed to overcome such inconvenience (see Fig. 1). The outer cap nut includes a smaller size nut, which is pressed into the outer cap, and there is an initial clearance at the bearing surface of the nut (see Fig. 1 (b)). The outer cap nut can be tightened easily by hands before contact to a clamped member. After the outer cap contacts with the clamped member, the bearing surface of the outer cap cannot move so that the deformation starts at the top of the outer cap. Through fastening process, the inner smaller size nut can move down because of the initial clearance, and accordingly, clearance comes out at the top of inner nut (see Fig. 1 (c)). This top clearance promotes the deformation of the outer cap, and therefore, the spring back of the outer cap causes large thread contact force between the nut and bolt. The thread contact force may prevent the loosening of the nut. In other words, there is no anti-loosening force before fastening so that the outer cap nut can be rotate easily by hand; however, the anti-loosening thread contact force can be realized after tightening in order to prevent loosening.

Therefore the outer cap nut can be used as conveniently as standard nuts because of no prevailing torque necessary before fastening. Also it should be noted that this outer cap nut has another advantage. Usually, the first thread of the standard nut burdens 25%-35% of the total load. On the other hand, the outer cap nut may improve the load distribution because the outer cap bears the largest portion of the total load. In this study, tightening-loosening behavior of the outer cap nut will be analyzed by axi-symmetric and three-dimensional finite element methods. Also, the load distribution of each thread will be compared with the case of standard nut; then, the stress reduction effect of the outer cap will be investigated.

2. Analysis method

For standard nut and bolt, it is known the first thread from the bearing surface of the nut bears about 30% of the total loads. The load distributions of the second and third decrease gradually. In the previous studies, the axi-symmetric analysis model was studied by the FEM ignoring the lead angle of the thread[7-10]. Moreover, the spiral shape of the thread was taken into consideration by 3-D FEM recently[11]. In this paper, we will employ threaded fastener involving an M12-6H/6g (JIS) thread, with a cylindrical clamped member whose inner radius, outer radius, and thickness are 13, 50, 35mm, respectively. Then, the results of the standard nut and those of the outer cap nut are compared and discussed. In this study, two models are considered, the axi-symmetric model ignoring the lead angle 2.7°, and the 3-D model considering the lead angle. In both models, the Young's modulus and Poisson's ratio of all materials (bolt, nut, and clamped member) are 205GPa and 0.3 respectively. In this study, six types of the material of the outer cap are considered, (1) Elastic material, (2) SS400, (3) S45C, (4) SCM440, (5) SUP9, (6) SUP10M. The stress and strain relations are shown in Fig. 2. For the outer cap nut, the inner dimension of the outer cap is smaller than the outer dimension of the inner nut (small hexangular nut), and the inner nut is pressed into the outer cap.

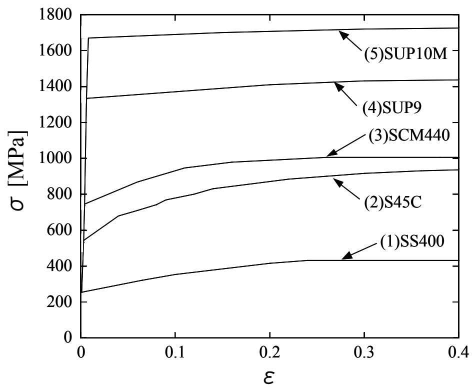

Fig. 2 Relation between stress and strain

2.1. Axi-symmetric analysis

In the axi-symmetric model 1(see Fig. 3), the thread shapes at both ends are complete, and the number of the threads is 6 with the nut thickness of 10.5mm. However, in the analysis of model 2 and model 3, the effect of the incomplete threads on the nut will be taken into consideration[10]. In model 2, the first thread is incomplete and only half of the thread remains, but the last one is complete, with the nut thickness of 9.625mm. In model 3, both of the ends are incomplete, and half of the first thread and three quarters of the last thread remain, with the nut thickness of 9.1875mm. Model 1, 2, 3 have different nut thicknesses in order to compare their load distributions at the six threads. An example of FEM mesh is shown in Fig. 4. The tolerance class is 6H/6g. The nut, bolt, and clamped member are combined as shown in Fig. 4 after mesh generated. The bolt and nut are divided into square elements automatically, and the parts of the thread root should be divided in detail. The clamped member is divided into the square elements and the contacting part is divided minutely. The total number of nodes and elements are 23,868 and 23,236. Both coefficients of friction between threads (μth) and between the bearing surfaces (μb) are 0.15. In order to simulate nut rotation process in the axi-symmetric model, the axial

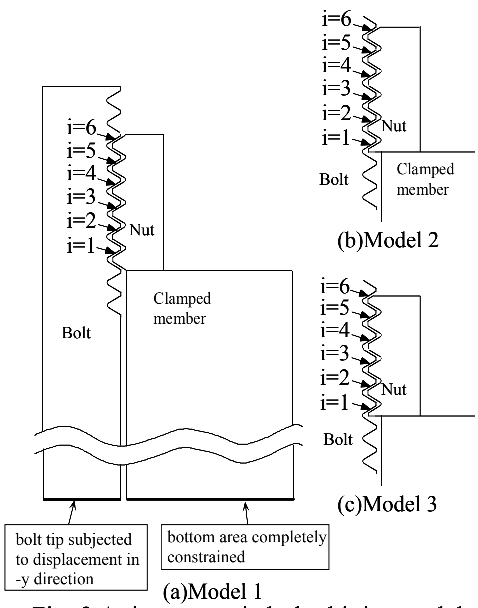

Fig. 3 Axi-symmetric bolted joint model

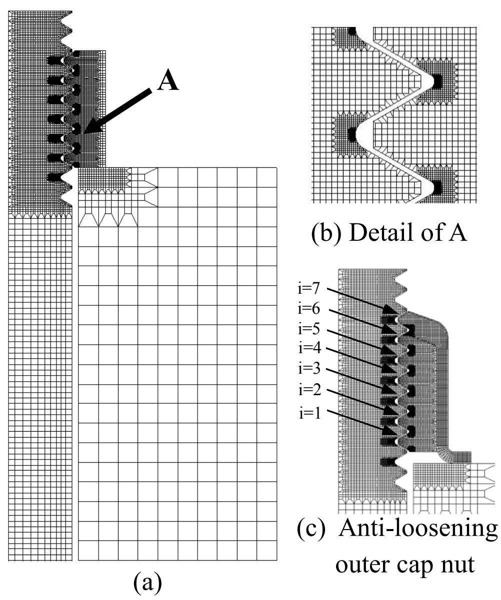

Fig. 4 Axi-symmetric finite element model for bolted joint

displacement is forced on the head of bolt, and the side surface of the clamped member is fixed in order to produce the bolt-axial force. The boundary conditions are shown as Fig. 3.

In the analysis of the outer cap nut, modeling of the bolt and clamped member is similar to that of the standard nut, and the inner nut and the outer cap nut are combined as shown in Fig. 4 (c) after mesh generated. The total number of the elements and the nodes are 25,453 and 26,276, respectively. First, the hexangular inner nut is pressed into the outer cap. The dimension of the outer cap is smaller by 0.05mm than the outer dimension of the inner nut. The constrained condition is the same as the standard nut. Moreover, both coefficients of fiction between the outer cap and the inner nut and between bearing surfaces (μb) are 0.15. In this model, No.6 and No.7 thread are the threads of the outer cap. No.7 thread is complete and No.6 one is incomplete.

2.2. Three-dimensional analysis

In the 3D model, the effect of lead angle can be considered. Assuming M12, the inner radius, the outer radius, and the thickness of the cylindrical clamped member are 14mm, 50mm, and 35mm, respectively. The tolerance class is 6H/6g. The nut, bolt and clamped member in Fig. 5 (b), (c) are combined together after mesh generated as shown in Fig 5 (a).

Figure 5 (a) shows the model of standard nut and bolt where the number of the elements is 20,884 and the number of the nodes is 6,762. In 3D models the detail shape of the curvature of the bottom of thread is not considered. Both models (b), (d) include the incomplete threads, considering the real threads. To investigate the effect of the incomplete thread, another nut similar to model (b) is also considered where there is no incomplete part. After the nut contacts the clamped member, the nut will rotate 90°. Both coefficients of friction between threads (μth) and between the bearing surfaces (μb) are 0.15. To prevent self-loosening, the bolt head is fixed and no rotation is assumed between the bolt head and bottom surface of the clamped member.

Figure 5 (d) shows the outer cap nut combined together with inner nut. The dimensions are indicated in Fig. 5 (e). In this analysis, the total numbers of the elements and nodes are 37,184 and 10,992, respectively. The inner diameter of outer cap is smaller by 0.05mm than the outer diameter of the inner nut before it is pressed into the inside. When the inner nut is pressed into the outer cap, the initial stress is produced at that time. Then, the bearing surface of the cap rotates 105° in the tightening orientation. Afterwards the nut rotates -45° until the outer cap separates from the clamped member. The contact condition and the boundary condition are the same as those of the standard nut. Both coefficients of fiction

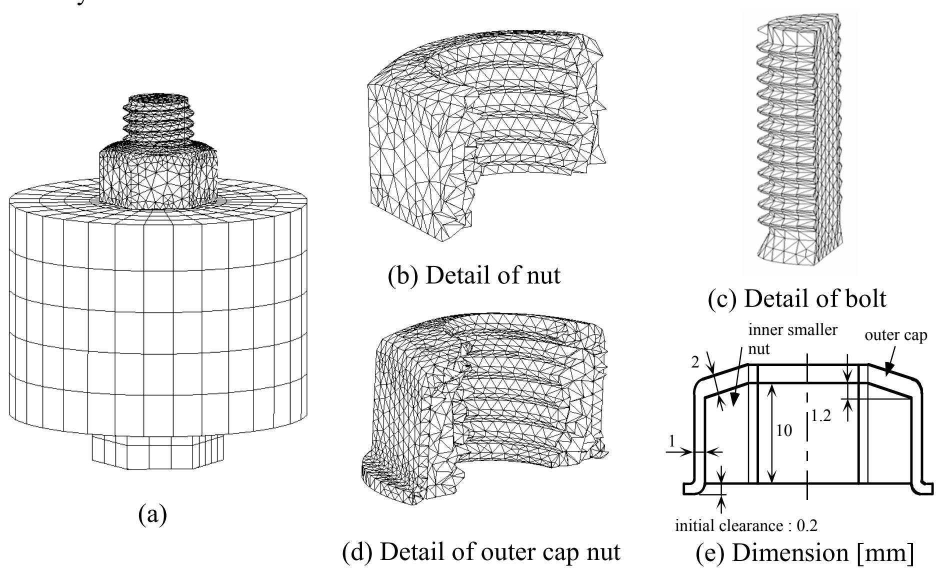

Fig. 5 Three-dimensional finite element model for bolted joint (a) (b) (c) and outer cap nut (d) (e)

between the outer cap and the inner nut and between bearing surfaces (μb) are 0.15.

For both axi-symmetric and 3D analyses, contact elements with Coulomb friction are applied to the nut, bolt, and clamped member. Then, we adopt the direct constraints method for the solution of contact problems. No special interference elements are required in this method and complex changing contact conditions can be simulated since no prior knowledge of where contact occurs is necessary. Therefore, when contact occurs, the contact between node and surface is constrained to normal direction and contact elements slip to tangential direction of the surface. The bilinear model is also adopted as the friction model. Relation between the sliding displacement and the friction force is linear in the small area of relative sliding displacement, and the friction force becomes constant beyond the range in this model.

3. Stress reduction effect of outer cap nut

3.1. The load distribution of the normal thread

First of all, the standard nut is analyzed through axi-symmetric and the 3D models in comparison with the outer cap nut. During tightening, the bolt-axial force will be burdened at each thread. Then, it is known that the first thread from the bearing surface of the nut bears about 30% of the total load. For the axi-symmetric models 1-3, the load distributions of each thread are shown in Fig. 6 as dashed lines when the bolt-axial force F_b is 45kN. The result of the 3D model is also indicated in Fig. 6 when the rotate angle is 90° as a solid line. Here, the solid line in Fig. 6 (a) shows the results of the incomplete threads in Fig. 5 (b). On the other hand, the solid line Fig. 6(b) shows the results of complete threads by deleting the incomplete threads in Fig. 5 (b). The models in Fig. 6 (b) have five threads with the smaller nut thickness. In Fig. 6, the y-axis is the load distribution, and x-axis is the number of each

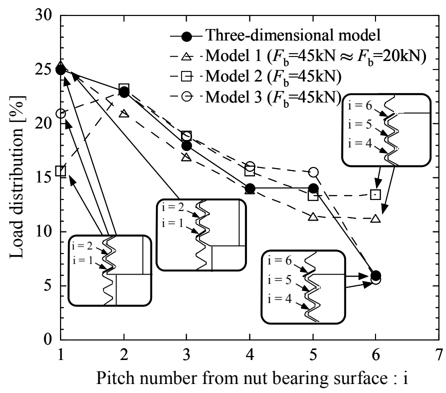

Fig.6(a) Load distribution of bolt-axial force of standard nut with incomplete threads at both ends

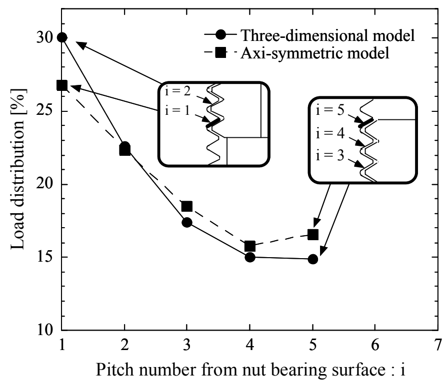

Fig. 6(b) Load distribution of bolt-axial force of standard nut (Comparison between axi-symmetric and 3D models) thread from the bearing surface of the nut. From the bearing surface of the nut, the first thread is called No.1 thread, and the second is No.2 thread, and so on. As shown in model 1 in Fig. 6 (a), the difference between F_b=45kN and F_b=20kN is less than 0.1%.

In the model 1, No.1 thread bears the largest loads, and No.2 and No.3 threads bear smaller loads gradually. The load distribution of No.1 thread in model 1 is larger than that in the model 2. However, in model 2, the load distributions from No.2 threads to No.6 threads all increase. In model 2, No.2 bears the largest loads, because No.1 thread is incomplete although No.1 thread bears the largest load in general. Similar results were reported by Fukuoka(10).

Also for the 3D model, the results of the incomplete thread in Fig. 6 (a) can be compared with the results of complete threads in Fig. 6 (b). In Fig. 6 (a), the rigidity of No.1 and last threads decreases because they are incomplete. From both results of the axi-symmetric and 3D models, it may be concluded that the incomplete thread usually decreases the load distribution of the thread compared with the one of complete thread.

3.2. The load distribution and stress concentration of the outer cap nut

When the nut rotates 105° in the 3D model, the load distribution of each thread is compared with the axis-symmetric model in Fig. 7. Figure 7 (a) shows the results when the outer cup is elastic, and Fig. 7 (b) shows the results for SS400. Here, the bolt-axial forces in these models are almost the same. In the 3D models in Fig. 7, No.7 thread is incomplete, and No.6 and No.7 threads are on the outer cap. On the other hand, in the axi-symmetric models in Fig. 7, No.1 and No.7 threads of the cap are complete, and No.6 thread is incomplete. For both models, outer cap carries the largest load. However, because of the difference, in the 3D model No.6 thread on the outer cap nut bears the largest load, and in the axis-symmetric model No.7 thread bears the largest load.

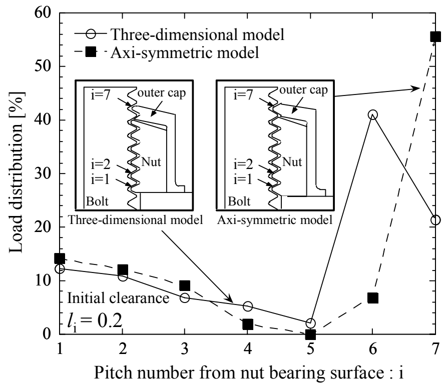

Fig. 7(a) Comparison between 3D and axi-symmetric models for load distribution of outer cap nut (Elastic outer cap, l_i=0.2mm, θ=105˚, F_b=52.9kN)

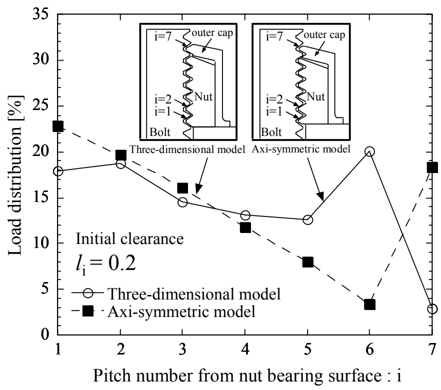

Fig. 7(b) Comparison between 3D and axi-symmetric models for load distribution of outer cap nut (SS400 outer cap, l_i=0.2mm, θ=105˚, F_b=44.6kN)

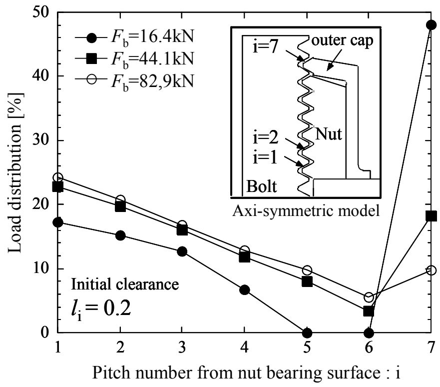

Fig. 8 Effect of bolt-axial force F_b by axi-symmetric model
(SS400 outer cap, l_i=0.2mm)

In the axi-symmetric model, the load distribution of each thread under the different bolt-axial force is shown in Fig. 8 when the outer cap is SS400. The load distribution of the each thread varies depending on the bolt-axial force because of the difference of the plastic strain. With increasing the bolt-axial force, the load distribution of outer cap decreases.

In Fig. 9, the load distributions of each thread are shown for various materials when the rotation angle is 105°. At this time, the bolt-axial forces of the bolt are 44.6kN - 52.9kN. From Fig. 9, the outer cap bears the largest load, and the peak value increases with increasing yield stress.

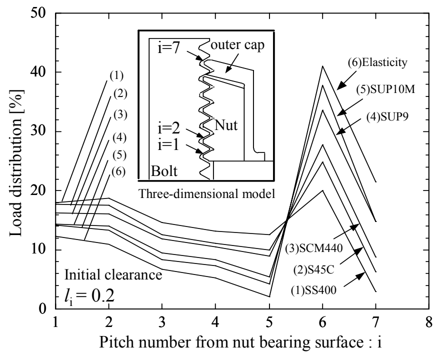

Fig. 9 Effect of material difference of outer cap nut by 3D model
(l_i=0.2mm, θ=105°, F_b=44.6kN-52.9kN)

In Fig. 10, effect of initial clearance is shown when the material of the outer cap is SS400. The initial clearance l_i is changed as 0mm, 0.01mm, 0.2mm, 0.5mm, 1.0mm. Comparing the standard nut (l_i=0), the load distribution of No.7 thread becomes larger and that of No.1 becomes smaller, when the initial clearance varies form 0 to 1.0mm. From Fig. 9, it is found that the load distribution of No.1 decreases to 22% when l_i=0.2mm.

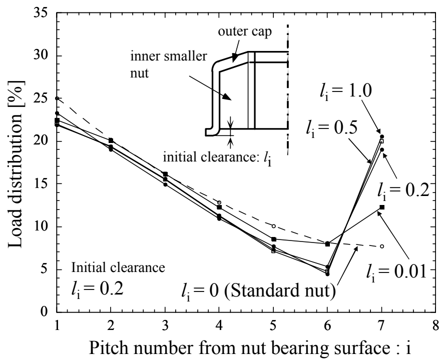

Fig. 10 Effect of initial clearance of outer cap nut by axi-symmetric model
(SS400 outer cap, l_i =0.2mm, F_b≈45kN)

The stress concentration factor K_t is shown in Fig. 11 where K_t is defined as $K_t = \sigma_i / \sigma_n$, σ_i is the equivalent stress at the thread root of the bolt, and σ_n is the normal stress at the minimum section. In general, the maximum stress appears at the first thread root from the bearing surface of the nut[6-9]. However, as shown in Fig. 11, the larger stress appears at No.7 for outer cap nut. If the material of the outer cap has high yield stress (see SUP9, SUP10M), the stress concentration decreases by 15% - 18% than that of the standard nut although higher stress appears near the outer cap nut. The stress of No.1 thread of the bolt decreases by 11% and the stress of No.7 thread is also lower than that of No.1 thread the standard nut, if the material of the cap is S45C or SCM440. These materials are most desirable for stress reduction.

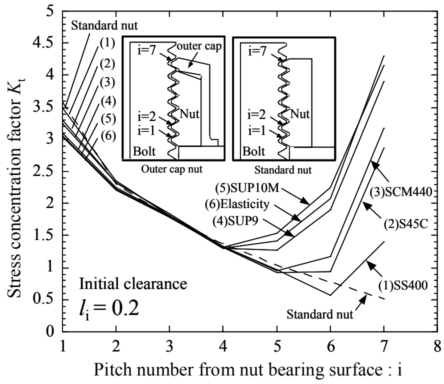

Fig. 11 Stress concentration factor of outer cap nut by axi-symmetric model (l_i =0.2mm, F_b =44.6kN-52.9kN)

4. The anti-loosening performance of the outer cap nut

When the material of the outer cap is elastic, the relation between clamping force and rotation angle is shown in Fig. 12. Also, the relation between tightening torque and rotation angle is shown in Fig. 13 (a), and the detail of A is as shown in Fig. 13 (b). Shown as Fig. 13, when the outer cap nut rotates 18°(see ①), the thread of the outer cap begins contacting the thread of the bolt, and the inner nut starts moving down to the clamped member; consequently the tightening toque increases. When the rotation angle is 80° (see ②), the inner nut begins contacting the clamped member, and therefore the tightening toque increases more. After the rotation angle reaches 105°, the rotation direction is reversed, and the tightening torque decreases gradually. At the rotation angle of 20° (see ③), the outer cap nut separates from the clamped member. However, 8–10Nm torque is still necessary to rotate the outer cap nut. The previous experimental results[12] show that this magnitude of the prevailing torque is large enough to pass the self-loosening test with extremely severe vibration and impact as specified in NAS3354.

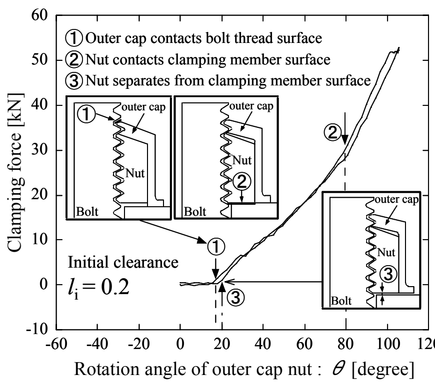

Fig. 12 Relation between clamping force and rotation angle of outer cap nut by three-dimensional model (Elastic outer cap, l_i =0.2mm)

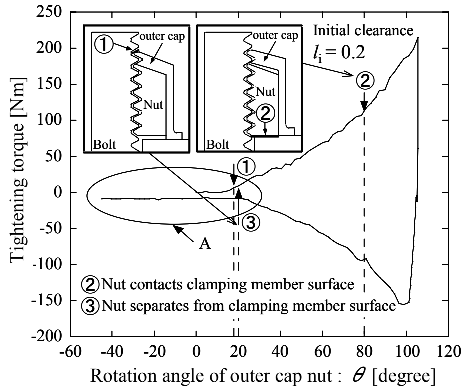

Fig. 13(a) Relation between tightening torque and rotation angle of outer cap nut by three-dimensional model (Elastic outer cap, l_i =0.2mm)

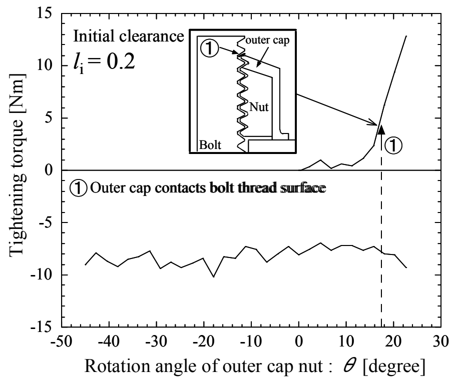

Fig. 13(b) Detail of A

In Fig. 14, the load of each thread of the outer cap nut is shown at the rotation angle is -45 degree. From this figure, it is seen that the prevailing torque is generated due to spring back effect after the deformation of outer cap. The No.6 thread consists of the portions of the inner nut and outer cap. The directions of the contact forces are indicated as the arrows in Fig. 14. The directions of the forces in No.6 thread are in the different directions for the portions of the inner nut and outer cap.

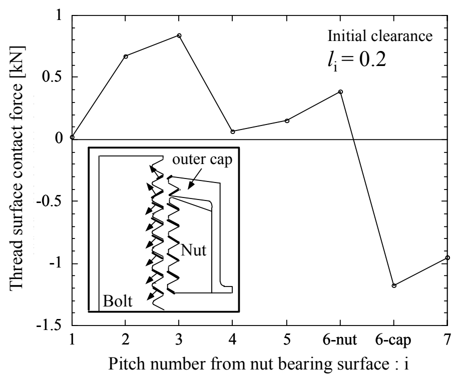

Fig. 14 Thread surface contact force by three-dimensional model (Elastic outer cap, l_i =0.2mm, θ= -45°)

Effect of material difference is shown in Fig. 15. As shown in Fig. 15, when the material of the cap is SS400, S45C and SCM440, the anti-loosening performance is not enough. This is because these materials have large plastic strain in the outer cap when the inner nut is pressed into the outer cap. Therefore the spring effect of the outer cap is too small for those materials. The elastic behavior of the outer cap is very important to realize the anti-loosening performance.

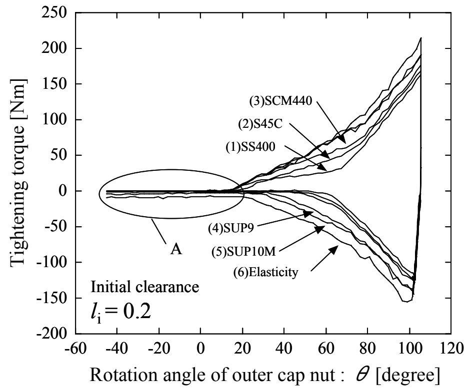

Fig. 15(a) Effect of material difference for tightening torque by three-dimensional model (l_i =0.2mm, θ= -45˚)

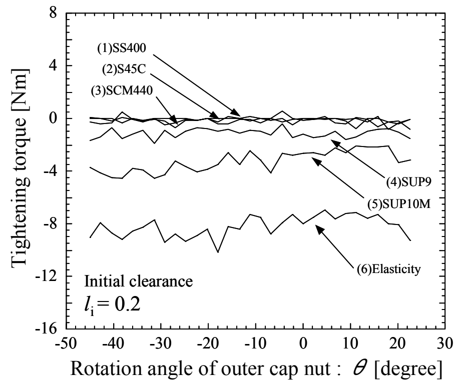

Fig. 15(b) Detail of A

5. Conclusions

In the previous studies, several anti-loosening bolts and nuts have been developed. However, for example, double nuts need two tightening processes, and another example, Super Slit Nut cannot be used without wrench. In order to overcome such inconvenience, the outer cap nut is developed as a new generation of anti-loosening nut. In this study, the stress reduction effect and anti-loosening performance of the outer cap nut are discussed by the application of axi-symmetric and 3D FEM modeling. Then, the following conclusions can be made:

(1) Since real bolts and nuts always have incomplete thread, the effect on the load distribution was investigated for the standard nut. It is found that the incomplete thread decreases the largest load distribution of complete thread because the stiffness of incomplete thread is smaller. The load distribution of No.1 thread is lower by 30%-25% than the No.1 complete thread model. The results are shown in Fig. 6. (a), (b).

(2) Usually, the first thread of the standard nut burdens 25%-35% of the total bolt-axial force. Since the outer cap bears the largest load, the outer cap nut can improve the load distribution under proper bolt-axial force (F_b=44.6kN-52.9kN) compared with the case of standard nut.

(3) The effect of material difference was investigated for the stress reduction and anti-loosening performance of outer cap nut. It is found that the maximum equivalent stress in the thread root of the bolt decreases by 11% and the anti-loosening performance can be realized with increasing the yield stress of outer cap.

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