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Novel anti-loosening nut designed to have large and stable loosening resistance torque

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Abstract This paper presents a novel anti-loosening nut design, called “SL nut”. It features an anti-loosening ring with an interference fit with the bolt thread, thereby squeezing the bolt in the radial direction. Using the finite element analysis, the prevailing and loosening resistance torques of the SL nut were investigated during tightening and untightening processes. The nut demonstrated the largest prevailing and loosening resistance torque among the nuts with prevailing torque. Importantly, both torques remained steadily large during tightening and untightening process. The magnitude of the torques can be controlled by adjusting the overlap length of anti-loosening ring. This new type could become a cost-effective solution for a variety of industrial applications. Additionally, the approach we employed could spur further development of anti-loosening nuts.

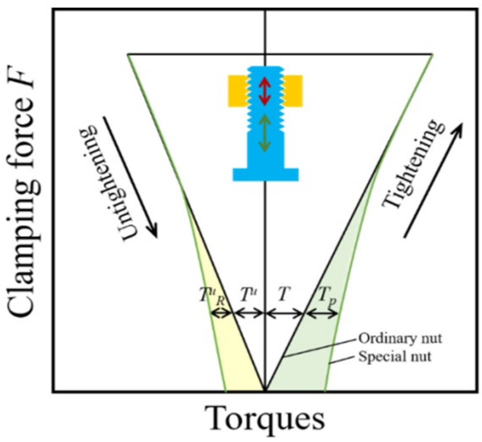
1. Introduction

Bolt nut connections are widely used in industry as they are low-cost and easy to maintain. For example, about 3000 nuts are used in one car, including the engine tightening part. Unfortunately, nuts can self-loose under an external force and cause accidents. Many studies have focused on loosening mechanism and improving the nut anti-loosening performance [1-26]. Many of them are designed to prevent loosening by changing the contact state. For example, the super lock nut changes the pitch between the second and third threads [5]. The U-nut is equipped with a friction ring on the top of the nut [6]. The outer cap nut is added a unique cap to prevent loosening [7]. The super slit nut has a reduced the pitch between the first and second threads and increased the anti-loosening part elasticity [9]. The pitch difference nut has slightly longer pitch than bolt [10-13].

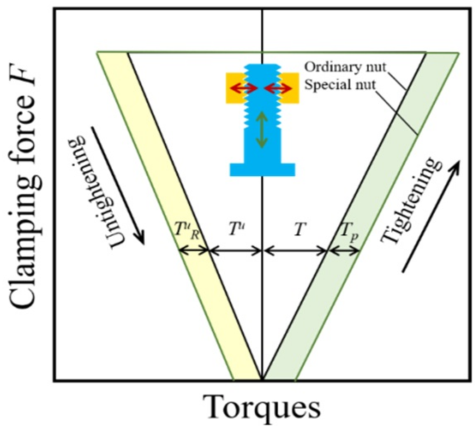
For tightening the above-mentioned anti-loosening nuts into the bolt, a larger torque should be applied compared with the ordinary nut. For example, Fig. 1(a) shows the *F-T* relation of ordinary nut and pitch difference nut. When the ordinary nut is tightened, only tightening torque T needed; but when the pitch difference nut is tightened, not only the tightening torque T but also an additional torque needed. This additional torque generally starts appearing at screwing process (before the nut touch with the clamped body), and it is named prevailing torque T_p [12, 13]. In the untightening process, for the pitch difference nut, also an additional torque named loosening resistance torque T^U_R needed. This is because the thread of pitch difference nut is specially designed; the nut needs to overcome greater friction when it is tightened or untightened. Many studies have shown that an increase in the nut's T_p can improve its anti-loosening performance [6-12]. Thus, T_p has been the major parameter for anti-loosening nuts evaluation. However, it not always provides a reliable information on the loosening process. To fill this gap, the loosening resistance torque (T^U_R) was proposed as a more suitable criterion for evaluating the anti-loosening performance [10]. While T_p resists the nut from being screwed into the bolt, T^U_R resists the nut from loosening. Without a significant thread deformation, their numerical values are similar. However, the threads wear and plastic deformation can cause a discrepancy between T_p and $T^U{}_R$ [13]. Therefore, it is more appropriate to use $T^U{}_R$ instead of T_p in evaluating the anti-loosening performance. This novel criterion has been employed for the pitch difference nut assessment [10-13]. Under a small clamping force this nut demonstrated a much larger T_p than the other special nuts [24]. Nevertheless, increasing the clamping force (F) comes with gradual decrease of T_p and $T^U{}_R$, as shown in Fig. 1(a). For this reason, the pitch difference nut performance can be sufficient only in case of a negligible or small clamping force, e.g. the scissors central shaft.

The present study aimed at developing an anti-loosening nut for a broad range of applications. Such nut should have the following three characteristics: 1) large $T^U{}_R$; 2) stability of $T^U{}_R$ regardless of the applied clamping force F; 3) cost-effective and unsophisticated manufacturing process. The second item is the most important and should achieve the effect shown in Fig. 1(b) as much as possible. For the nut designing and testing we mainly opted for 3D finite element method (FEM) as a time- and cost-efficient approach, compared to the experimental method. A number of studies [10-13] have demonstrated FEM's results reliability and consistency with the experimental findings.

The paper presents a novel anti-loosening nut design, named "SL nut" after the first letters of the authors' surnames. The loosening resistance torque and anti-loosening performance of SL nut were demonstrated via FEM.



(a)



(b)

Fig. 1. (a) The F-T relation of pitch difference nut; (b) the desired of F-T relation. The gray line shows - ordinary nut, green line - anti-loosening nut; T: tightening torque, T_p prevailing torque, T^u untightening torque, $T^u{}_R$ loosening resistance torque.

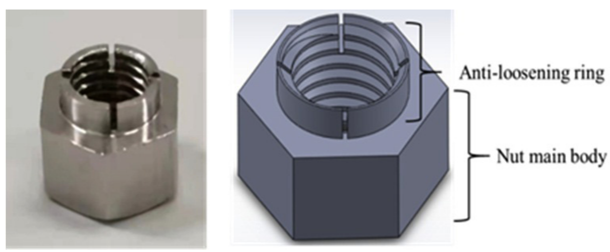
2. The SL nut

2.1 Fundamental design concept of SL nut

The pitch difference nut was described in our previous studies [10-13]. These studies used the JIS standard bolt, but pitch of the nut was larger with the additional length called "pitch difference" (α). As shown in Fig. 1(a), tightening the pitch difference nut requires much larger torque compared to the ordinary nut. However, the prevailing torque T_p (light green area) and $T^U{}_R$ (light yellow area) tends to decrease with an increasing F. Thus, anti-loosening performance of the pitch difference nut tends to decline with increasing F.

We suspect that the reason for the above problem is the inappropriate direction of the $T^u{}_R$ -generating force. In the pitch difference nut, the $T^u{}_R$ -generating force is the bolt axial force between thread mountains F_α [12]; it appears in the Z direction, as shown by the red arrow in Fig. 1(a). In this case, the changing of F will affect the magnitude of $T^u{}_R$ -generating force. So, we hypothesized that applying the $T^u{}_R$ -generating force orthogonally to F (green arrow) could eliminate this issue. In the new desired nut, the $T^u{}_R$ -generating force is a radial force appears in the r-direction, as shown in the red arrow in Fig. 1(b). In this case, the changing of F will not affect the magnitude of $T^u{}_R$ -generating force, and the value of $T^u{}_R$ will be very stable, as shown in Fig. 1(b) light yellow area.

Fig. 2 shows the characteristics of the SL nut. Its lower part is the same as in the ordinary nut. The upper part, hereafter called "anti-loosening ring", was designed to prevent loosening. The latter has four gaps to conform to the bolt. The two parts were made whole from S45C. The JIS standard M12 bolt will used with it. Fig. 3 is a schematic view of a cross-section and



(a) (b)

Fig. 2. The characteristics of the SL nut: (a) is the specimen; (b) shows the 3D-model.

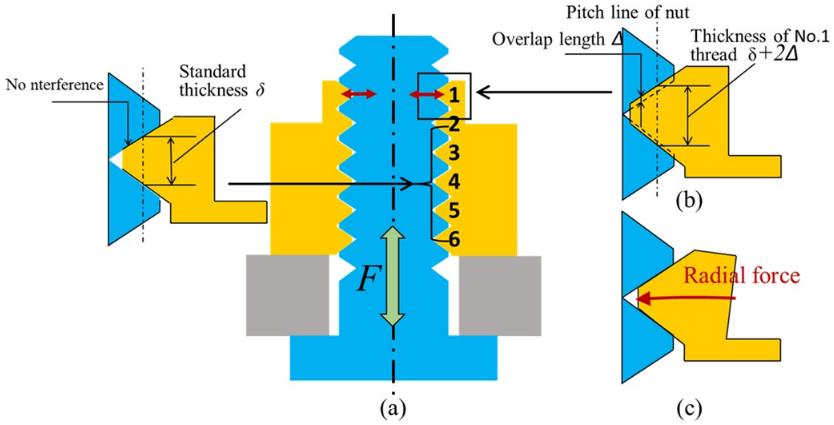


Fig. 3. The forces of the bolt-SL nut: (a) shows a radial force appears in the r-direction; (b) shows the contact status based on the screw pair dimensions; (c) shows the contact status after the screwing.

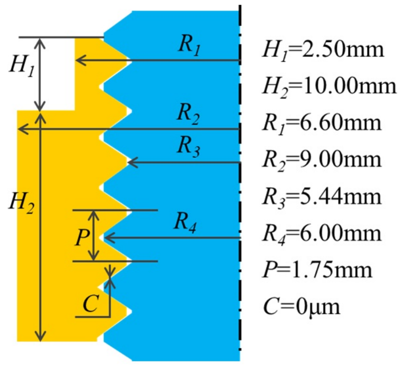


Fig. 4. The SL nut dimensions. H_1 - the anti-loosening ring height, H_2 - the nut body height, R_1 - the anti-loosening ring radius, R_2 - the nut body radius, R_3 - the nut hole radius, R_4 - the bolt radius, P - the pitch, C - the clearance between bolt and nut thread in axial direction.

the expected deformed state of the SL nut. The no. 1 thread's revolution is thicker the no. 2-6. The thickness of no. 2-6 thread's revolutions are standard δ = 0.875 mm, the thickness of no. 1 thread is $\delta+2\Delta$, thicker than others. So, it forms an interference fit with the bolt. The volume of interference (overlap length) is denoted by Δ. The Δ = 0-135 μm are analyzed in this study. When the SL nut is screwed into the bolt, the no. 1 thread revolution squeezes the bolt in the radial direction. The radial force generates T_p and T^u_R.

Fig. 4 shows the SL nut dimensions. The clearance of the standard M12 screw pair is 20-77 μm [10], but the clearance between the same bolt and SL nut is 0 μm. Due to the existence of interference, the SL nut will not be easily screwed into the bolt, so it is not necessary to keep the clearance, and smaller the clearance provides better anti-loosening performance.

We performed FEM analyses to assess the effect of F changes on T^u_R of the SL nut and test the anti-loosening performance.

2.2 FEM analyses

Two analyses were performed in this study.

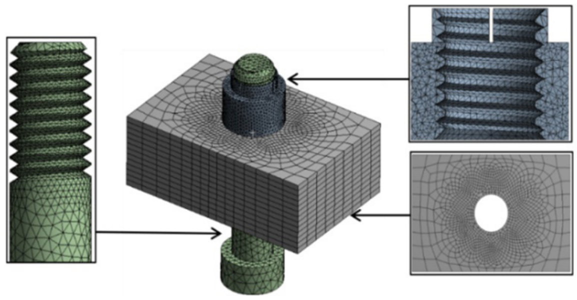


Fig. 5. 3D FEM mesh of SL nut, bolt and clamped plate. The thread surface and bearing surface have been meshed refinement.

1) Assessing the influence of F on T^u_R during tightening-untightening processes.

2) Testing the anti-loosening performance and finding a suitable overlap length Δ in the loosening process.

Fig. 5 shows the 3D model of the bolt, nut, and clamped plate. The hexagons of the bolt and nut were replaced by cylindrical surfaces for simplification. The element size of the thread and the bearing surface was refined. According to our previous unpublished research, acceptable results quality can be obtained with the number of elements exceeding 5×10^4. In this study, 8.6×10^4 elements were used with 15.6×10^4 nodes and the average element side length in the thread equaled 0.6 mm. In this simulation, non-linear material properties and contact were considered. The friction coefficients at the thread surface and bearing surface were set to μ_s = 0.12 and μ_w = 0.15, respectively [10, 12, 24]. The bolt and clamped plate are made of SCM435, and the nut is made of S45C. Table 1 shows the material properties.

Previously the authors have investigated pitch difference nut systematically by using the FEM analysis where the pitch difference is treated as a kind of overlap as an initial boundary condition [8, 10-13, 24]. It was confirmed that if such initial overlap length Δ is not too large, no analysis errors happened. In a similar way, in this study as shown in Fig. 6 the overlap Δ is treated as an initial boundary condition. Then, a prior analysis is conducted showing that if the overlap Δ ≤ 135 μm no

Table 1. Material properties of the bolt and nut.

	Young's modulus E [GPa]	Poisson's ratio v	Yield strength σ_y [MPa]	Tensile strength σ_B [MPa]	Tangent modulus E_t [MPa]
SCM435	206	0.3	800	1200	2000
S45C	206	0.3	530	980	2250

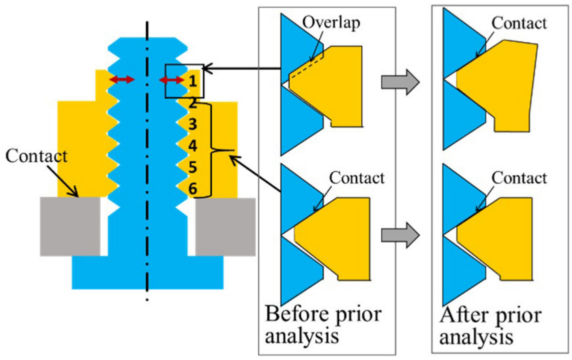
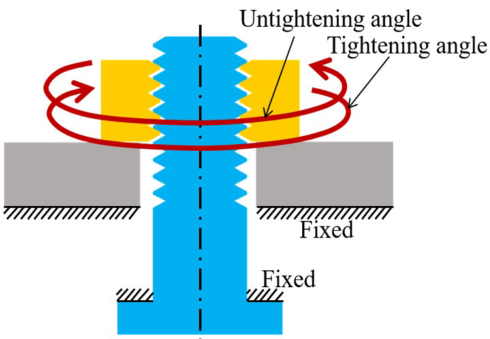
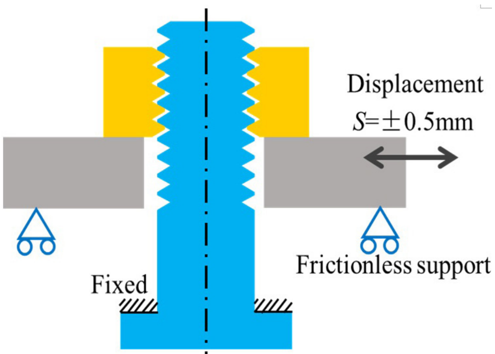


Fig. 6. Assembly method of bolt, nut and clamped body in FEM model.



(a) Boundary conditions of tightening-untightening processes



(b) Boundary conditions of loosening process

Fig. 7. Boundary conditions.

analysis errors happened.

The boundary conditions used in the analyses were as following.

i. Tightening-untightening process (Fig. 7(a)): The bolt head and the bottom surface of the clamped plate were fixed, the rotation angles $+\theta$ and $-\theta$ were applied on the nut for tightening and untightening the nut, respectively. To test the SL nut's anti-loosening performance under a large clamping force, F was set to 47 kN. This value of F ($F_{70\,\%}$) ensures that the bolt nominal stress reaches 70 % of the yield stress.

ii. Loosening process (Fig. 7(b)): The Junker type vibration test (standard DIN25201) was simulated to evaluate the anti-loosening performance. The bolt head was fixed. The initial rotation angle $+\theta$ was preset to tighten the nut to $F_{70\,\%}$. The clamped plate moved only in ±X direction with the maximum displacement S = ±0.5 mm [9, 18, 21]. 20 vibration cycles were simulated in this study to allow the nut to self-loose under the transverse force. The anti-loosening performance was assessed by the loss of clamping force.

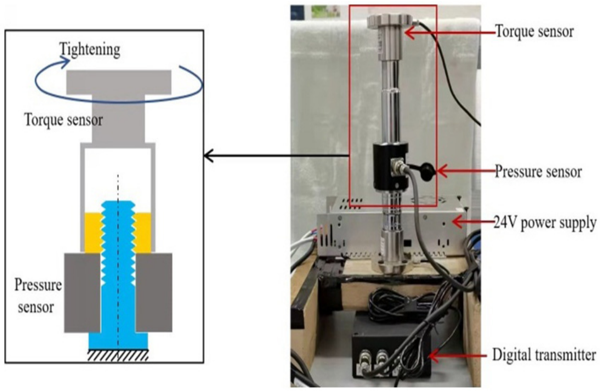


Fig. 8. Experiment device and experimental condition.

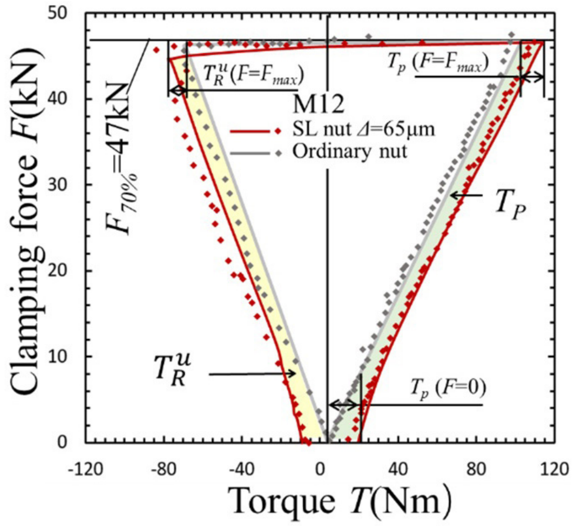
2.3 Experimental condition

In order to compare the analytical T_P T^U_R with experimental results, the F-T relation is measured in this study. The experiment conditions are shown in Fig. 8. The bolt head is fixed, the nut is tightened manually until the clamping fore reaches $F_{70\,\%}$, then untighten the nut until F = 0. The clamping force can be measured by the pressure sensor, and the T_P T^U_R can be measured by torque sensor. The molybdenum disulfide grease spray is only used on thread surface as lubricating oil. The nut specimen with Δ = 65 and 105 μm are tested.

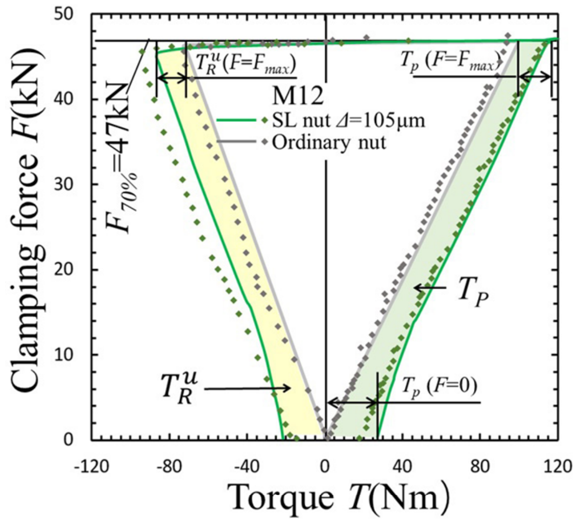
3. Results

3.1 Tightening-untightening process of SL nut

Fig. 9 shows the F-T relation of SL nut when (a) Δ = 65 μm and (b) Δ = 105 μm. The solid line shows the FEM result, the dotted points show the experimental result. In the tightening process, when (a) Δ = 65 μm and (b) Δ = 105 μm, the analytical results coincide with the experimental results within a few percent error. In untightening process, however, the experimental results slightly larger than analytical results. This is because the lubricating oil film breaks and friction coefficient increases. The friction coefficient increases more due to a small amount of metal debris caused by wear embedded in the



(a) Δ = 65 µm



(b) Δ = 105 µm

Fig. 9. *F-T* relationship of the ordinary nut and SL nut with Δ = 65 and 105 µm.

clearance portion. The friction increase contributes to the larger anti-loosening torque and better anti-loosening performance. Note that FEM cannot express the friction increase.

In Fig. 9, two conditions of the prevailing torque T_p are considered for the SL nut by FEM. One is before tightened T_p when $F = 0$ and the other is T_p under fully tightening when $F = F_{max}$. In addition, loosening resistance torque T^U_R when $F = F_{max}$ is also considered. Although the plastic deformation during the tightening affects these three torques, the three kinds of torques obtained by the FEM are nearly the same as can be expressed $T_p(F = 0) \approx T_p(F = F_{max}) \approx T^U_R(F = F_{max})$. As shown in the *F-T* relations of the SL nut it should be noted that the magnitude of those torques remains nearly constant irrespective of *F* in both tightening and untightening process. In other words, the *F-T* relation of the SL nut has the desired performance indicated in Fig. 1(b).

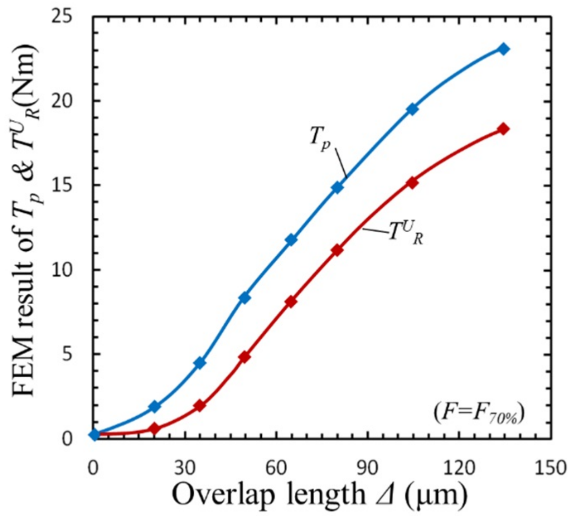


Fig. 10. The effect of overlap length Δ on T^u_R, obtained by FEM.

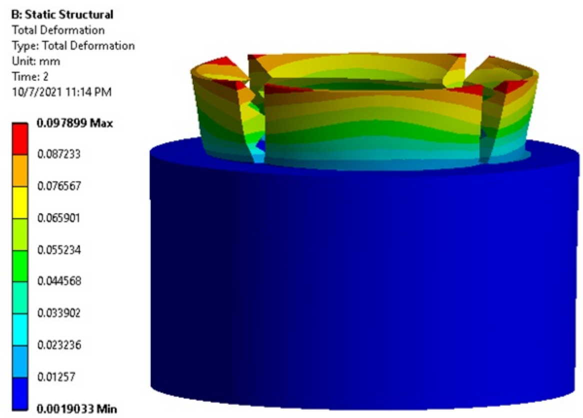


Fig. 11. Deformation of the Δ = 105 µm SL nut.

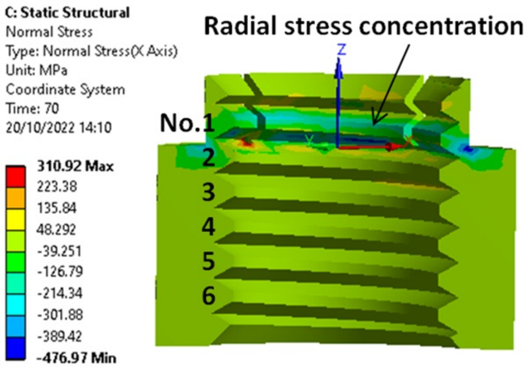
Fig. 10 shows the effect of Δ on T^u_R when $F = F_{70\%}$. As Δ increases from 0 to 135 µm, T^u_R increases from 0 to about 18.6 Nm. An optimal T^u_R can be selected from this range by adjusting Δ. However, the choice is limited by Δ. Too large Δ can reduce the fatigue strength of the anti-loosening ring or, alternatively, prevent the nut from screwing. An optimal Δ can be found in the loosening experiment. In addition, because the existing of plastic deformation, for each Δ, the prevailing torque T_p is always slightly larger than it's loosening resistance torque T_R^u.

Fig. 11 shows the total deformation of the Δ = 105 µm SL nut, when the nut is screwed into the bolt without any clamping force. The amount of deformation is magnified by 20 times. As expected (Fig. 3(c)), the anti-loosening ring expands and deforms outwards. This can provide the radial force. It generates T_R^u and T_p to prevent the nut from loosening.

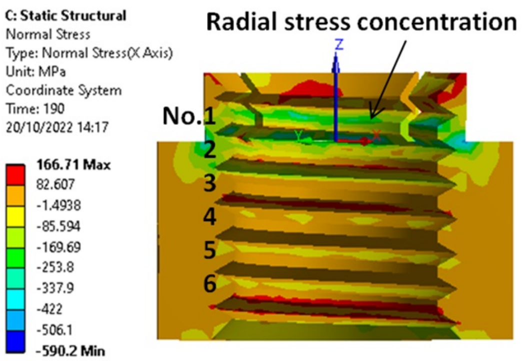
Fig. 12 shows the radial stress σ_r distribution of the SL nut in cylindrical coordinate. The green areas shows the region where a larger compressive stress appears as $\sigma_r \geq$ 200 MPa. As shown in Fig. 12(a), the no. 1 circle of thread has stress concen-

Table 2. The torques of special nuts, the results is obtained by experiment.

	SL nut Δ = 0-135 μm	Pitch difference nut α = 0-50 μm	Super slit nut	U-nut	Outer cap nut
T_p	0-24.8 Nm	0-29.9 Nm	13.5 Nm	1.5 Nm	1.2 Nm
T^U_R ($F = F_{25\,\%}$)	0-26.5 Nm	0-15.0 Nm	≤ 13.5 Nm	≤ 1.5 Nm	≤ 1.2 Nm
T^U_R ($F = F_{70\,\%}$)	0-27.5 Nm	≈ 0	≤ 13.5 Nm	≤ 1.5 Nm	≤ 1.2 Nm



(a) Radial stress σ_r, when F = 0



(b) Radial stress, when $F = F_{70\,\%}$

Fig. 12. Radial stress distribution of the Δ = 105 μm SL nut.

tration before the fastener is tightened, which indicates the T^u_R can be generated by radial force. As shown in Fig. 12(b), after the fastener is tightened to $F = F_{70\,\%}$, the No.1 circle of thread appears stress concentration, which indicates the T^u_R also can be generated by radial force. In addition, due to the existence of clamping force, tensile radial stress $\sigma_r > 0$ occurs in most areas of the nut. It can be seen from the figures that whether the clamping force exists or not, the SL nut will squeeze the bolt radially, so that the SL fastener can generates T^u_R.

3.2 Loosening process

According to the DIN25201 standard, the anti-loosening performance is sufficient if the residual clamping force exceeds 80 % of the initial clamping force. Although the experiment is conducted until 1500 vibration cycles, only 20 vibration cycles can be simulated by using FEM due to the limitation of computational time.

Fig. 13 shows FEM simulation results. The clamping force

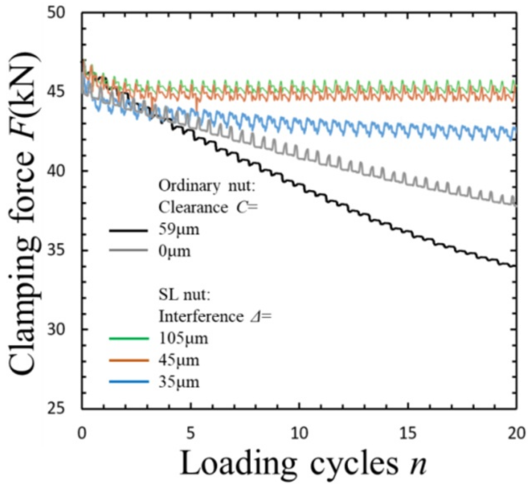


Fig. 13. Relation between loading cycles and clamping force of the SL and ordinary fasteners in the loosening process, obtained by FEM.

decreases largely for the ordinary nut when the clearance C = 59 μm although the anti-loosening performance can be improved when C = 0. By introducing the overlap length Δ = 35 μm in the SL nut, the anti-loosening performance is improved more. When Δ = 45 μm and Δ = 105 μm, the clamping force F does not decrease anymore when the loading cycle n = 2~20. The initial drop of F can be explained by the two reasons:

1) During the tightening process, the bolt accumulates a certain elastic torsion angle. When the clamping plate starts to move, the elastic torsion angle unwinds, causing a drop in F.

2) The tightening creates a small plastic deformation of the bolt's thread resulting in an increase of the distance between bolt head and nut.

In the analysis, we tried several values of Δ. Increasing Δ from 45 μm to 105 μm did not significantly affect the anti-loosening performance. Given the above-mentioned considerations on the Δ choice, we opted for 45 μm as the minimum value at which a sufficient anti-loosening performance can be achieved.

4. Discussion

Investigation of the anti-loosening mechanism of the pitch difference nut allowed us to identify an important requirement to achieve stable T_p and T^U_R under a wide range of F. The proposed design is based on this new approach. The SL nut has the anti-loosening ring, where the first thread's revolution creates an interference fit with the bolt. Thus, the ring squeezes the bolt in the direction perpendicular to the bolt axis and the clamping force. Such forces orientation ensures stable torques irrespective of F magnitude. Owing to the gaps in the ring, the nut can be screwed to the bolt even with a significant overlap length.

Although most of the previous studies on anti-loosening nuts have not considered T^u_R [5-7, 9, 12], the authors' recent study on the pitch difference nut indicated that the loosening resistance torque T^u_R appears at $F = 0$ with the magnitude T^u_R at $F = 0 \leq T_p$ at $F = 0$. In this previous study, if the tightening force F is relatively smaller, the prevailing torque T_p also appear at $F = F_{max}$ as well as T_p at $F = 0$. The magnitude of those torques have the following relations (1) and (2).

$$T_p(F=0) \geq T_p(F=F_{\max}) \geq T^U{}_R(F=F_{\max}) \tag{1}$$

$$T_p(F=0) > T^u{}_R(F=F_{25\%}) > T^u{}_R(F=F_{70\%}) . \tag{2}$$

Instead, as shown in Table 2, the SL nut have the following totally different relations (3) and (4).

$$T_p(F=0) \; < \; T_p(F=F_{max}) \; < \; T^u{}_R(F=F_{max}) \tag{3}$$

$$T_p(F=0) \; \leq T^u{}_R(F=F_{25\%}) \; \leq T^u{}_R(F=F_{70\%}) . \tag{4}$$

Also, it should be noted that the SL nut can produce much larger T^u_R ($F = F_{70\,\%}$) than the others. Then, the magnitude of T^u_R is insensitive to the clamping force at $F = F_{max}$. Therefore, this SL nut may provide better anti-loosening performance than others.

The FEM analysis in Fig. 10 shows that the SL nut under the constant friction coefficient in the tightening and untightening processes, the value of T_p and the value of T^u_R are almost constant independent of the clamping force F, and T_p is slightly larger than T^u_R.

Based on the FEM analysis in Fig. 10, Eq. (3) in the experiment can be explained from the following two reasons. Compared to the tightening process, the lubricating oil film between threads in the untightening process becomes thinner, which makes the friction coefficient slightly larger. In the tightening process, the metal debris generated by wear is stuck between the thread clearance, which also causes the friction coefficient to increase in the untightening process. Eq. (4) can be explained from the increase of the friction coefficient when the larger tightening force when F increases from $F = F_{25\,\%}$ to $F = F_{70\,\%}$ during the untightening process. The increase of the friction leads to $T^u_R(F = F_{25\,\%}) < T^u_R(F = F_{70\,\%})$.

5. Conclusions

In order to solve the problem that threaded fasteners are prone to loosening failure, a novel threaded fastener called SL-nut is studied. This new type could become a cost-effective solution for a variety of industrial applications. The main results are shown in follows:

1) The SL nut form interference fit with the bolt and squeeze the bolt in the radial direction, the loosening resistance torque T^u_R is generated by the radial force. The magnitude of T^u_R can be controlled by adjusting the overlap length Δ of the nut thread.

2) Compared with other threaded fasteners with prevailing torque T_p, the SL bolt-nut fastener can produce larger T_p and T^u_R, which can provide good anti-loosening performance.

3) The T_p and T^u_R provided by SL bolt-nut fasteners are almost independent of the clamping force F, because the radial force generating those torques is insensitive to the clamping force.

4) The loosening experiment simulation shows that the larger overlap length Δ provides better anti-loosening performance of the SL nut. When Δ is larger than 45 μm, the M12 SL bolt-nut fastener does not loosen. The value Δ = 45 μm can be used as the threshold anti-loosening overlap length of M12.

5) The FEM analysis shows that the SL nut under the constant friction coefficient in the tightening-untightening process T_p and T^u_R are almost constant independent of the clamping force F, and the T_p is slightly larger than T^u_R. Due to increasing the friction coefficient in the untightening process, the following relationship can be confirmed $T_p(F = 0) \leq T^u_R(F = F_{25\,\%}) \leq T^u_R(F = F_{70\,\%})$.

Nomenclature

F	: Clamping force
T	: Tightening torque
T_p	: Prevailing torque
T^u	: Untightening torque
T^u_R	: Loosening resistance torque
Δ	: Overlap length
μ_s	: Friction coefficients at the thread surface
μ_w	: Friction coefficients at the bearing surface
P	: Pitch
σ_r	: Radial stress
C	: Clearance
θ	: Rotation angles of nut
S	: Displacement of clamped plate

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