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著者	Chen Xin, NODA Nao-Aki, AKAISHI Yu-Ichiro, SANO Yoshikazu, TAKASE Yasushi
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Effect of Pitch Difference on Anti-loosening Performance and Fatigue Strength for High Strength Bolts and Nuts

**Xin Chen[1,*], Nao-Aki NODA[1], Yu-Ichiro AKAISHI[1]
Yoshikazu SANO[1] and Yasushi TAKASE[1]**

[1] Department of Mechanical Engineering, Kyushu Institute of Technology, Fukuoka 804-8550, Japan
* Corresponding author: xin96995@126.com

Abstract A slight pitch difference is considered between the bolt and nut in this study. Here, the pitch of nut is α μm larger than the pitch of bolt. In the first place, the distance δ, of nut screwed onto bolt without any prevailing torque, is experimentally and analytically obtained for each level of α. Then, the relationship between axial force and prevailing torque during tightening process is tested. According to the obtained torque-axial force relationship, for four different levels of α, the loosening experiment based on NAS3350 (National Aerospace Standard) is performed, and the loosening and dropping status of the nuts are investigated. Considering both the anti-loosening ability and the clamping ability of the nut, the desirable range of α is discussed. After that, according to the loosening experiment results, three levels of α are selected to make a further study about the effect of the pitch difference on the fatigue life of bolt. The fatigue experiment is performed and the S-N curves are obtained. Finally, the finite element method is used to make a simulation of the fatigue experiment and the mean stress and stress amplitude at each thread bottom of bolt are analyzed.

Keywords Bolted Joint, Pitch Difference, Anti-loosening Performance, Fatigue Life

1. Introduction

The bolts and nuts are important joining elements disassembled easily for maintenance and widely used because of low cost. To ensure the structure safety, the anti-loosening performance and high fatigue strength have been always requested, and therefore, several special nuts were invented in order to prevent self-loosening [1-3]. The previous experimental results showed that the fatigue strength may be improved under suitable pitch error existing between bolt and nut [4]. The tapered bolt named CD bolt proposed by Nishida, indicated higher fatigue strength experimentally and being widely used in structures [5].

The authors previously analyzed the anti-loosening effect and stress reduction effect for the bolt and nut having slightly pitch difference by applying the finite element method [6-7]. However, the previous studies are limited to theoretical investigations. Therefore, the effect of pitch difference on anti-loosening performance has not been experimentally verified yet, and an experimental analysis for the effect of pitch difference on the fatigue life is also necessary.

In this paper, the M16 (JIS) bolt and nuts are employed and designed with a slight pitch difference. The standard M16 bolt and nut usually have the same pitch of 2000μm. Here, we assume that the nut pitch is equal or slightly larger than the bolt pitch, and we will consider four types of pitch difference, that is, $\alpha=0$, $\alpha=\alpha_{small}$, $\alpha=\alpha_{middle}$ and $\alpha=\alpha_{large}$. The clearance between bolt and nut is assumed as a common dimension of 125μm. At the first place, the position where the prevailing torque appears will be discussed towards four different levels of pitch difference. Then, the relationship between axial force and prevailing torque will be experimentally obtained. To investigate the effect of pitch difference on the anti-loosening performance, the loosening experiment will be performed by using the four groups of bolt joint specimens. Considering the clamping force and loosening status of bolts and nuts during the loosening experiment, the desirable pitch difference will be discussed. Next, according to the loosening experiment results, then the fatigue experiment will be performed to obtain the S-N curves. After that, the finite element method will be applied to simulate the fatigue experiment, and then the endurance limit diagrams will be discussed.

2. Relationship between prevailing torque and axial force

For the normal bolt and nut, the nut can be screwed onto the bolt without torque until touching the clamped member. When a slightly pitch difference α is introduced on each nut pitch, the height of the nut increases accordingly, and therefore, during the screwing process, the contact status will also changes between the bolt threads and nut threads. It can be imagined that the nut threads may get stuck in the bolt threads if the pitch difference α is large enough, and in that case, a prevailing torque is necessary to continue screwing the nut to the requested position. Figure 1 shows a schematic illustration of a screwing status appeared until which the nut becomes immovable without any prevailing torque. Actually, as the nut is screwed onto the bolt, the pitch difference will be accumulated between the bolt threads. Finally, the two ends threads of nut contact with the bolt threads at the same time as shown in Figure 1. Theoretically, when a pitch difference is added on each nut pitch, the distance δ which nut be screwed onto bolt without any torque, can be obtained by formula (1).

$$\frac{\delta\alpha}{P} = \frac{nC_y}{2tan\theta} \quad , \quad C_x = \frac{C_y}{tan\theta} \tag{1}$$

where δ is the distance where the prevailing torque appears, α is the pitch difference, n is the thread number of nut, θ is the thread angle, P is the pitch of bolt, C_x and C_y are the clearance between bolt and nut (see Figure 1). To clarify the mechanism, four sets specimens of bolts and nuts with different levels of α are investigated. The four levels of α has a relationship of $\alpha=0<\alpha_{small}<\alpha_{middle}<\alpha_{large}$, where α=0 represents the normal bolt and nut. As seen in Figure 2 (a), the thread length of bolt is of 42mm and the height of nut is 16mm. The clamped plate, shown in Figure 2 (b), has a width of 34.5mm. It is known that without clamped plate, the normal nut (α=0) can be screwed onto the blot until δ equals 42mm. However, with increasing the pitch difference, the fitting distance δ without any prevailing torque will decrease. The relationship between δ and the pitch difference are investigated. Table 1 shows the measured value of fitting distance δ for four levels of α.

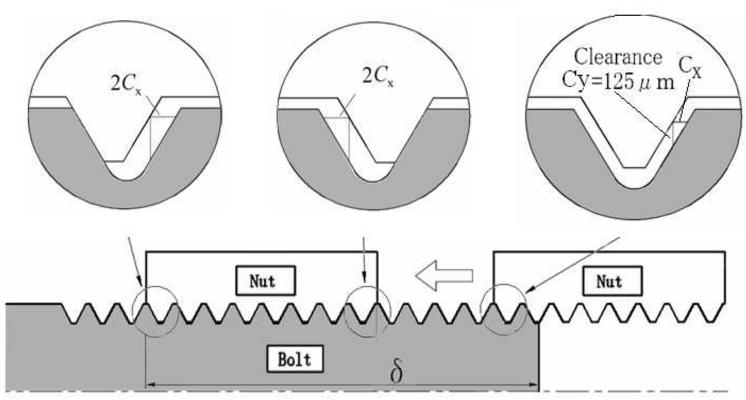
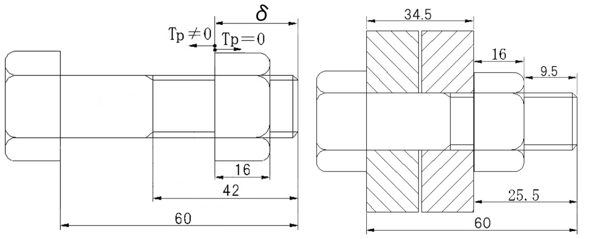


Figure 1. Schematic illustration for bolt and nut which have slight pitch difference

Table 1. Fitting distance δ without any prevailing torque

Pitch difference α (μm)	δ (mm)
0	42
α_{small}	25.5<δ<42
α_{middle}	16<δ<25.5
α_{large}	δ<16



(a) Without clamped plate (b) With clamped plate

Figure 2. Fitting status [mm]

When $\alpha=\alpha_{small}$, the nut can be screwed to the requested position for clamped plate (δ=25.5mm) without any prevailing torque. For the cases of $\alpha=\alpha_{middle}$ and $\alpha=\alpha_{large}$, a prevailing torque is necessary to carry the nut touching the clamped plate.

Next, we will focus on the relationship between axial force and prevailing torque of the bolt in the tightening process. Figure 3 shows the experimentally obtained torque-axial relationship of the bolt during the tightening process until the nut reaches the position in Figure 2(b). Whether the prevailing torque is needed or not, for all the nuts, the final tightening torque is set as the same value of 70 N·m. For the normal bolt and nut, the axial force produces just after the torque is applied on the nut. At the point of torque equals 70 N·m, the axial force, in other words, the clamping force of the bolt, reaches 24 kN. The case of $\alpha=\alpha_{small}$ has the same torque-axial force relationship with α=0. When α increases to α_{middle}, a prevailing torque of 40 N·m is needed before the nut becomes contacting with the clamped plate. Under the same tightening torque of 70 N·m, the axial force of bolt is reduced by 4 kN compared with the normal case. When $\alpha=\alpha_{large}$, the prevailing torque increases accordingly, on the other hand, the axial force decreased significantly to 8 kN.

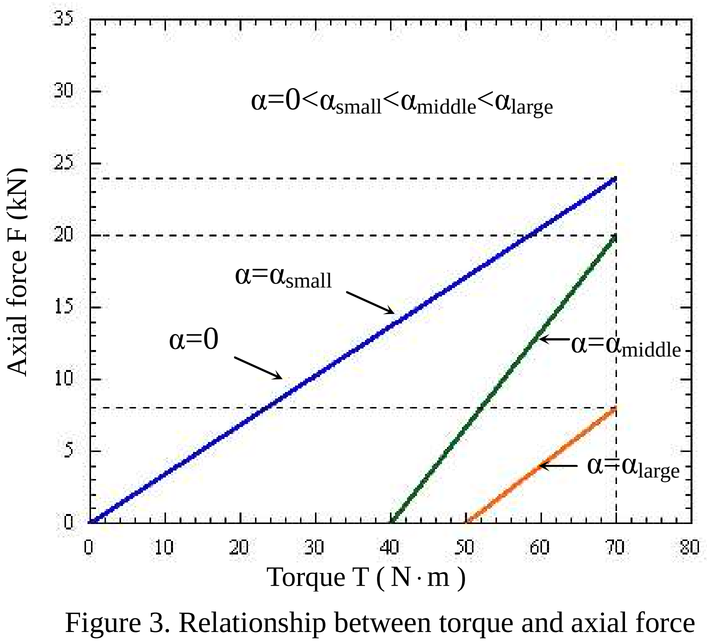


Figure 3. Relationship between torque and axial force

3. Effect of pitch difference on anti-loosening performance

Based on the torque-axial force relationship obtained above, the loosening experiment is performed to investigate pitch difference effect on the anti-loosening performance. The same specimens are used in the torque-axial force and loosening experiments for $\alpha=0$, $\alpha=\alpha_{small}$, $\alpha=\alpha_{middle}$ and $\alpha=\alpha_{large}$. Two specimens are investigated together for the loosening experiment. The experimental device is impact-vibration testing machine based on NAS3350 (National Aerospace Standard) whose vibration frequency is 1,800 cycles per minute, and vibration acceleration is 20G. If the vibration cycles are over 30000, we may judge the anti-loosening performance is enough. The specimens are fixed on the experimental device under the same torque of 70 N·m before the vibration starts. The experiment device is shown in Figure 4. When the device starts vibrating, a counter connected with the experiment device shows the cycles of vibrations. Table 2 shows the cycles of vibrations when dropping happens and when loosening happens.

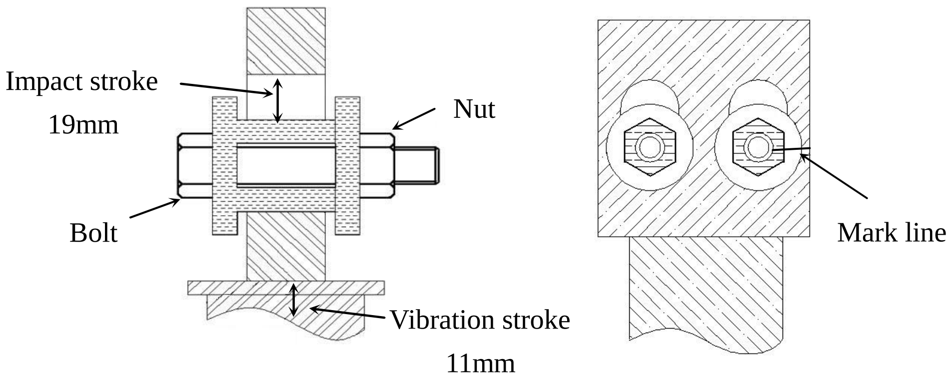


Figure 4. Loosening experimental device based on NAS3350

For $\alpha=0$ and $\alpha=\alpha_{small}$, the nuts drops when the vibration number is nearly 1000 cycles. For $\alpha=\alpha_{middle}$, the nuts does not drop before 30,000 cycles of vibration, but the loosening is observed for one specimen. Moreover, the axial force decreases to 20 kN in this case. For $\alpha=\alpha_{large}$, neither dropping nor loosening happens, however, the clamping ability of the bolt and nut decreases significantly to F=8 kN, which is not acceptable for actual application. It is found that, on the one hand, when α is

Table 2. Experimental result for anti-loosening performance

Pitch difference α	Sample	Nut drop	Cycles for dropping	Cycles for start loosening	Prevailing torque (N·m)	Axial force (kN)
0	No.1	Yes	751	-	0	24
	No.2		876	-		
α_{small}	No.3		813	-	0	24
	No.4		1528	-		
α_{middle}	No.5	No	30000	21000	30	20
	No.6		30000	30000		
α_{large}	No.7		30000	30000	67	8
	No.8		30000	30000	57	

too small, the expected anti-loosening performance cannot be realized. On the other hand, when α is too large, the clamping ability will decrease significantly because large deformation takes place on the bolt thread. Therefore, considering both the anti-loosening ability and the clamping ability, $\alpha=\alpha_{middle}$ can be selected as the most suitable pitch difference.

From the loosening experiment results we can see that, a proper pitch difference has a positive effect on the anti-loosening performance of bolt and nut. Next, we will focus on the effect of pitch difference on fatigue strength of bolt, considering three types of pitch difference of $\alpha=0$, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$.

4. Effect of pitch difference on fatigue strength of bolt

4.1. Fatigue experiment

The M16 (JIS) blots and nuts, strength grade 8.8, are employed in the fatigue experiment. The specimens are subjected to a mean force of 30 kN. Since the cross sectional area A_R of the bolt is 141 mm^2, the corresponding mean stress of the specimens is 213 MPa. For normal M16 bolt, it is known that the fatigue strength is about 50 MPa. Here, the stress amplitude is ranged from 50 MPa to 160 MPa in this experiment. The 392kN (40tonf) Servo Fatigue Testing Machine is used with cycling frequency 8Hz. The assembly drawing is illustrated in Figure 5 and the experimental device is shown in Figure 6. Three groups of bolt joint specimens of $\alpha=0$, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$ are investigated.

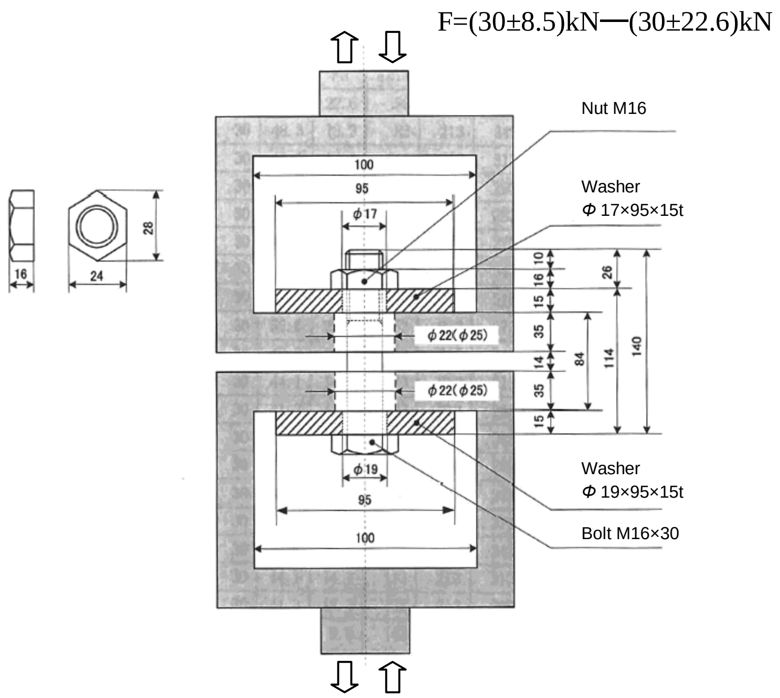


Figure 5. Schematic illustration of fatigue experiment [mm]

Figure 6. Fatigue experimental device

4.2. S-N curves

The S-N curves with fatigue limit at $N=2\times10^6$ are obtained as shown in Figure 7. It is found that the fatigue lives are different clearly depending on the three levels of pitch difference. Table 3 shows the comparison between the fatigue life normalized by the results of $\alpha=0$. When the stress amplitude is above 80 MPa, the fatigue life for $\alpha=\alpha_{small}$ is about 1.4 times larger and the fatigue life for $\alpha=\alpha_{middle}$ is about 1.2 times larger than the one of the normal bolt and nut of $\alpha=0$.

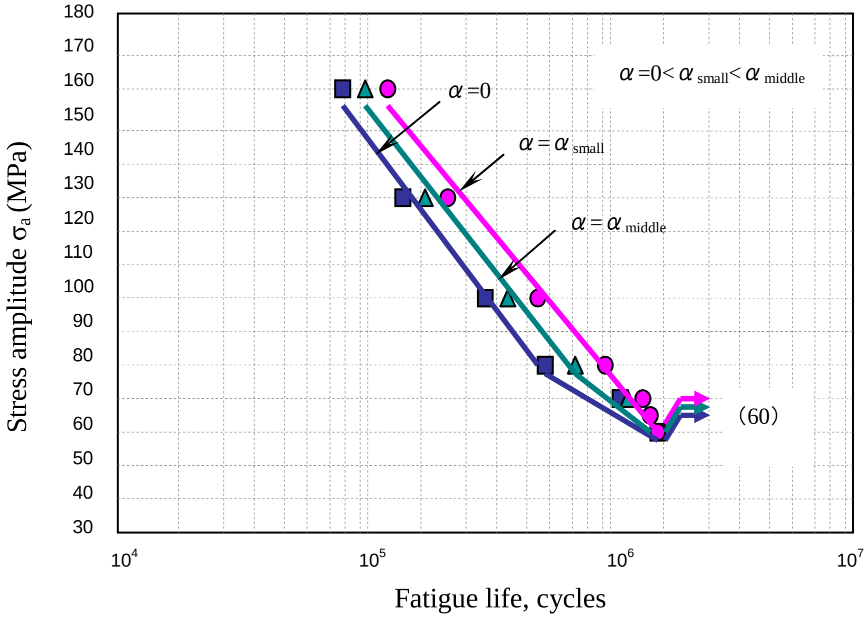


Figure 7. S-N curves

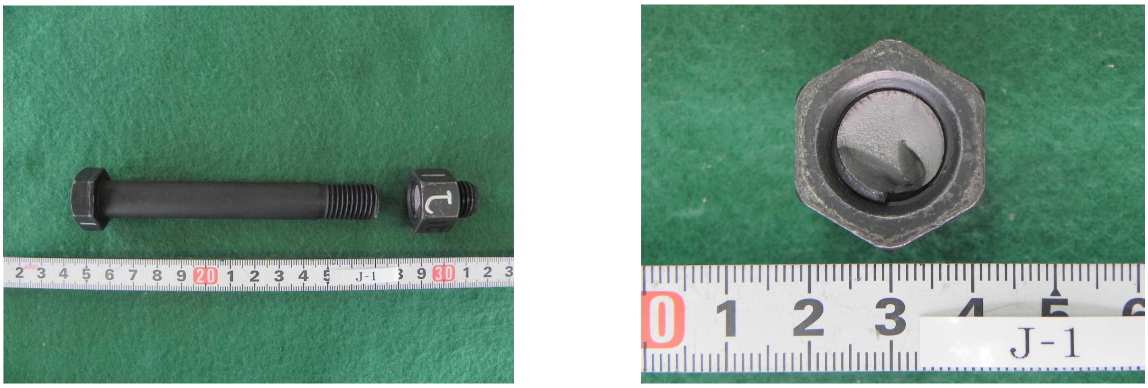
Table 3. Comparison between the fatigue life for $\alpha=0$, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$ (Mean stress σ_m=213MPa)

Pitch difference α	Stress amplitude σ_a (MPa)				
	160	130	100	80	70
0	1	1	1	1	1
α_{small}	1.49	1.60	1.53	1.61	1.21
α_{middle}	1.26	1.22	1.20	1.21	1.02

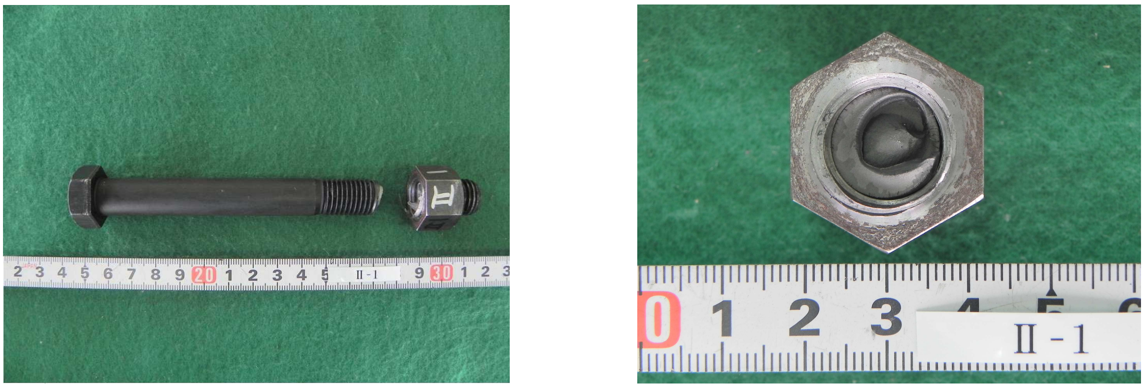
However, near the fatigue limit, the fatigue life of three group specimens is not very different, and the fatigue limit remains the same value of 60 MPa for three cases of pitch difference.

4.3. Investigation of the fractured specimens

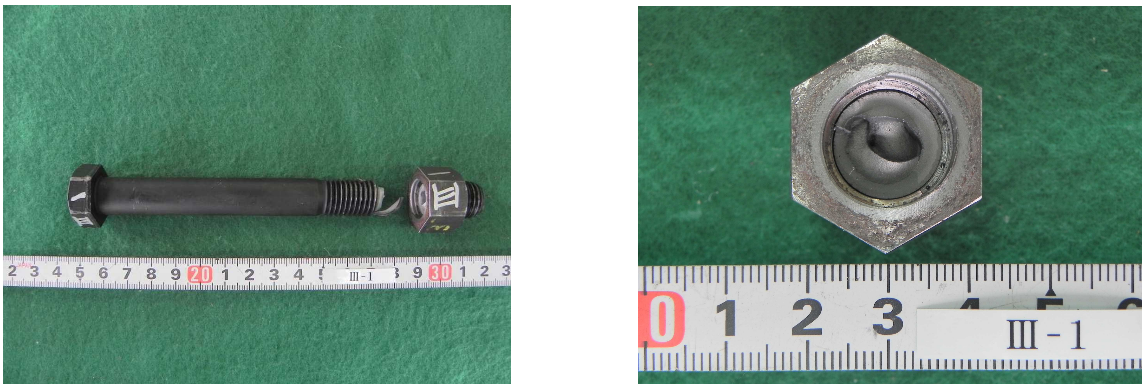
Figure 8 shows the examples of fractured specimens (σ_a=130MPa). For the normal bolts and nuts, it is confirmed that the fracture is always happened at the first thread bottom as shown in Figure 8(a). For the specimens of $\alpha=\alpha_{small}$, the fracture position is between the No.1 thread and No.3 thread of bolt. It should be noted that the fracture surfaces have the different characteristic compared with the



(a) α=0 (fatigue life 151,860 cycles)



(b) $\alpha=\alpha_{small}$ (fatigue life 242,810 cycles)



(c) $\alpha=\alpha_{middle}$ (fatigue life 184,770 cycles)

Figure 8. Fractured specimens (σ_a=130MPa)

case of α=0. For the specimens of $\alpha=\alpha_{middle}$, the fracture position of bolt and the fracture surface are nearly same with the case of $\alpha=\alpha_{small}$.

4.4. FEM analysis

For the normal bolt and nut, it is known that the largest stress concentration appears at the No.1 thread of bolt. However, when α is introduced, the contact status between bolt and nut is changed, and therefore, the stress distribution along the bolt threads should be reconsidered. To analyze the stress status of the bolt threads, the finite element models are created by using FEM code MSC.Marc/Mentat 2007. Three models have the different pitch difference of α=0, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$ in according with the experiment. As shown in Figure 9, the axisymmetric model of bolt and nut is created. The fixed component is assumed as a cylindrical clamped plate with an inner diameter 17.5mm, outer diameter 50mm and thickness 35mm. The Young's modulus is 206GPa and the Poisson's ratio is 0.3 for all the materials for bolt, nut and clamped plate. In the experiment, the bolt axial force F=(30±8.5)kN—(30±22.6)kN was studied. In the analysis, the bolt axial force F=30 ±14.1kN is considered as an example. The assumed amplitude Fa=14.1kN is corresponding to the stress amplitude σ_a=100 MPa at the minimum section of the bolt. The elastic calculation is performed as the first step, but the results show that the stress is over the yield stress σ_s=800 MPa of the material SCM435 for the wide region. Then, the elastic-plastic calculation is performed again. Based on the mean stress and the stress amplitude at each root of bolt thread, the endurance limit diagrams are obtained as shown in Figure 10 to Figure12. The fatigue limit σ_N of the material SCM435 is 420 MPa. The mean stress σ_m and stress amplitude σ_a are explained as following:

$$\sigma_m=\frac{\sigma_{max}+\sigma_{min}}{2}, \quad \sigma_a=\frac{\sigma_{max}-\sigma_{min}}{2} \tag{2}$$

where σ_{max} is the maximum stress of each thread bottom under the maximum load of F=30+14.1KN and σ_{min} is the maximum stress of each thread bottom under the minimum load of F=30-14.1KN. The endurance limit diagrams show the dangerous level of each bolt thread. From Figure 10, we can see that the No.1 thread bottom has the high stress amplitude and mean stress, which is corresponding to the fracture position in the experiment as shown in the section 4.3. In Figure 11, when a pitch difference of α_{small} is introduced, the stress amplitude decreases at the No.1 thread bottom. On the other hand, plastic deformation happens mainly at the bottoms of No.7 and No.8 threads. However, the difference among the stress status at each thread is not very large compared

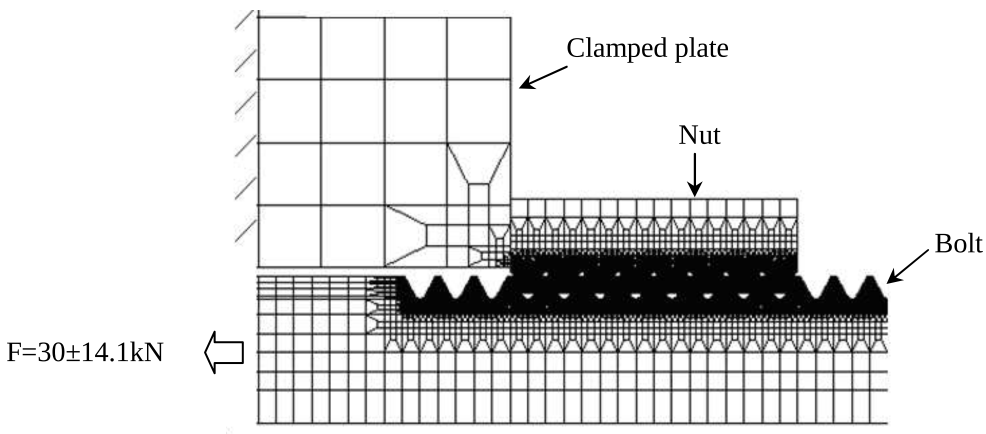


Figure 9. Axisymmetric finite element model

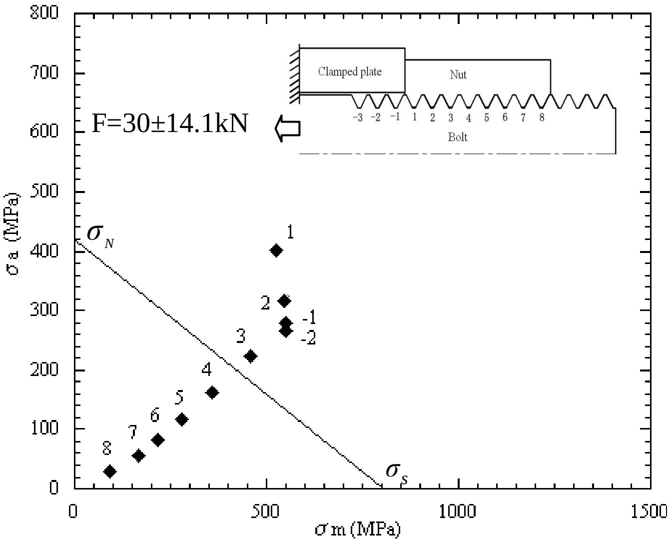


Figure 10. Endurance limit diagram for α=0

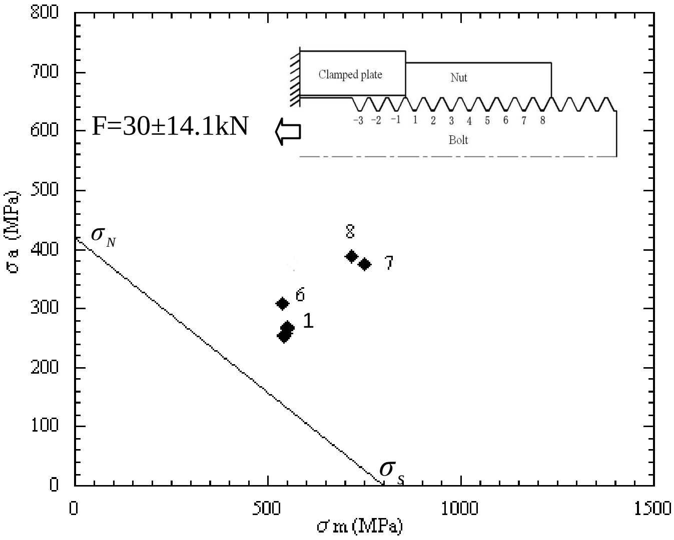


Figure11. Endurance limit diagram for $\alpha=\alpha_{small}$

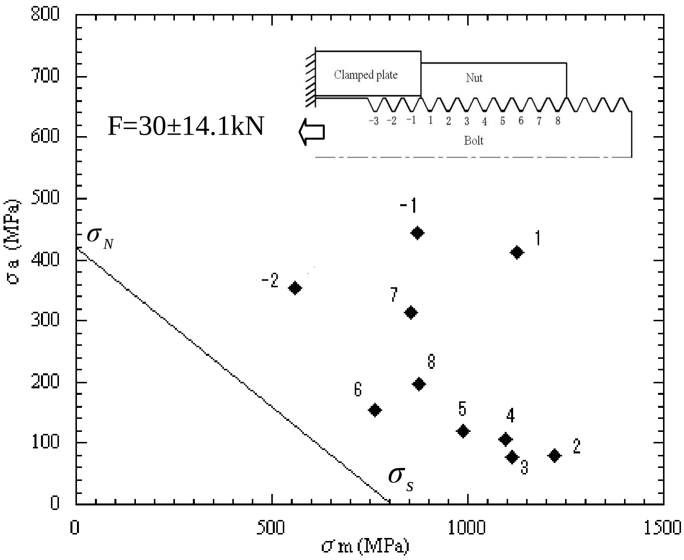


Figure 12. Endurance limit diagram for $\alpha=\alpha_{middle}$

to the normal bolt. For the case of $\alpha=\alpha_{middle}$, as Figure 12 shows, because of the large pitch difference, it is seen that the mean stress values for most threads is over the yield stress the material.

Conclusions

In this study, a slightly pitch difference is designed between bolt and nut. Four different levels of pitch difference $\alpha=0$, $\alpha=\alpha_{small}$, $\alpha=\alpha_{middle}$ and $\alpha=\alpha_{large}$ are experimentally considered. The relationship between prevailing torque and axial force for each case of α has been analyzed as the first step. Next, the loosening experiment and the fatigue experiment have been performed successively to study the anti-loosening performance and the fatigue strength of bolt. Finally, by using the finite element method, the effect of pitch difference on the stress status of the bolt thread bottom has been numerically analyzed. The conclusions can be summarized as follows:

(1) It is found that $\alpha=\alpha_{middle}$ is the most desirable pitch difference to realize both the anti-loosening and clamping abilities.

(2) It is found that $\alpha=\alpha_{small}$ is the most desirable pitch difference to extend the fatigue life of the bolt and nut. Compared with the normal bolt and nut, the fatigue life for $\alpha=\alpha_{small}$ can be extended by about 1.4 times and the fatigue life for $\alpha=\alpha_{middle}$ can be extend ed by about 1.2 times.

(3) According to the FEM analysis, it is found that the stress amplitude at the No.1 thread bottom decreases significantly for $\alpha=\alpha_{small}$, and due to the large plastic deformation at No.7 and No.8 thread bottoms, the stress amplitude here is almost the same as other thread bottoms.

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