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Fatigue Life Improvement by Slight Pitch Difference in Bolt-Nut Connections

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Keywords：bolt-nut connections, pitch difference, fatigue strength, finite element method

ABSTRACT

The bolt-nut connections are widely used in various fields. In this paper, a slight pitch difference is introduced between the bolt and nut in order to study the effect on the fatigue performance. Here, we consider that the nut pitch is a few microns larger than the bolt pitch. Fatigue experiments are conducted for three kinds of specimens with different levels of pitch differences. The obtained S-N curves show that the fatigue life is extended to about 1.5 times by introducing a suitable pitch difference. According to the detailed investigation on the fractured specimens, it is found that the fractured positions and the crack configuration are totally different depending on the pitch difference. The mechanism of the fatigue strength improvement is discussed in terms of the stress amplitude and average stress at each bolt thread.

INTRODUCTION

The bolt-nut connections are regarded as an important joining technique widely used in various engineering fields. The fatigue failure and self-loosening sometimes lead to severe accidents. To ensure the connected structure safety, the anti-loosening performance and high fatigue strength have been required. Most previous studies are mainly focusing on developing the anti-loosening performance, and a few studies are contributing toward improving the fatigue strength. This is because high stress concentration K_t=3-5 appearing at the bolt thread cannot be reduced very easily. Usually for special bolt-nut connections the anti-loosening ability sacrifices the fatigue strength and the low price significantly. In other words, anti-loosening bolt-nut connections have not been developed yet without reducing the fatigue strength and without raising the cost.

Maruyama's study indicated that the fatigue strength may be improved depending on the pitch error (1973). The effect of the thread shape on the fatigue life of bolt has also been investigated by Nishida (1994). The tapered bolts, called CD bolts (Critical Design for Fracture), are widely used because they have shown higher fatigue strength experimentally (Nishida, 1980, 2009). In terms of anti-loosening, several special nuts were invented to deal with the self-loosening problems (Minamida Industry, 1999; Hard Lock Industry 2002; Daiki Industry, 2004).

This study aims to realize both fatigue strength improvement and anti-loosening performance by considering one certain innovation on the screws. A slight pitch difference α is introduced between the bolt and nut. Xiao et al. (2011) previously analyzed the anti-loosening effect and stress reduction effect for the bolt and nut having a slight pitch difference by applying the finite element method. The authors' preliminary study showed that the bolt fatigue life could be improved by introducing the pitch difference of α=15 μm (Akaishi et al., 2013). Moreover, in the anti-loosening study, the experimental results showed that α=33 μm is the most desirable pitch difference to realize the anti-loosening performance (Noda et al., 2015). As a further research, in this paper, more detailed fatigue experiment is conducted systematically under a series of cyclic fatigue loads for three types of specimens, i.e. α=0, α=15 μm and α=33 μm, where α=0 represents the standard bolt-nut connections. Then, the S-N curves are obtained and the improved fatigue lives are discussed. To clarify the effect of pitch difference, the Finite Element Method (FEM) is applied to analyze the stress amplitude and average stress at each bolt threads. The mechanism of fatigue life improvement is considered

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by comparing the experimental results to those obtained using the finite element method.

EXPERIMENTAL SET-UP

Material properties

The Japanese Industrial Standard (JIS) M16 bolts and nuts are employed, and the fatigue experiments are conducted by using the 392 kN Servo Fatigue Testing Machine. Table 1 and Table 2 show the JIS standard and the material properties of bolt and nut in this study. The normal M16 bolt and nut have the same pitch dimension as 2000 μm. Herein, the nut pitch is assumed to be (2000+α) μm. Three types of pitch differences, namely α=0, 15 μm and 33 μm, are considered. The clearance between bolt and nut is assumed to have a standard dimension as 125 μm. The schematic diagram of bolt and nut is shown as Figure 1. Figure 2 illustrates the effect of the pitch difference on the contact status between bolt and nut threads when a large pitch difference is introduced.

Fatigue tests

The experimental device used in the fatigue tests is shown in Figure 3. The bolt specimens are subjected to a series of repeated loadings. Table 3 shows the experimental loading conditions and the corresponding stress according to the bottom cross sectional area of the bolt A_R=141 mm^2. The cycling frequency of the loadings is 8 Hz.

Table 1 JIS Standards of Bolt and Nut

	Strength grade	Yield strength (MPa)	Tensile strength (MPa)
Bolt	8.8	660	830
Nut	8	-	-

Table 2 Material Properties of Bolt and Nut

	Young's modulus (GPa)	Poison's ratio	Yield strength (MPa)	Tensile strength (MPa)
SCM435 (Bolt)	206	0.3	800	1200
S45C (Nut)	206	0.3	530	980

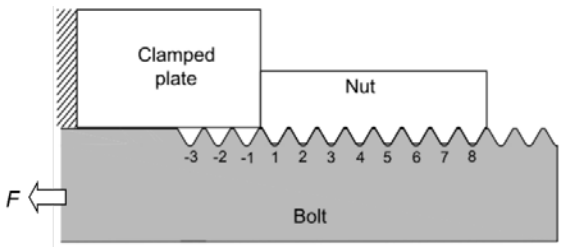


Fig. 1. Schematic diagram of bolt joint

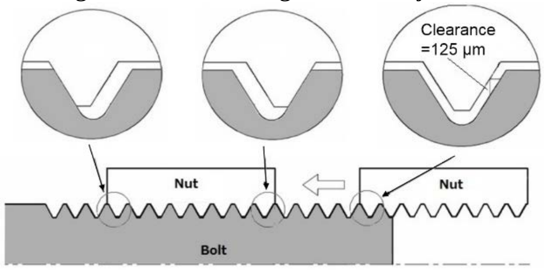


Fig. 2. Contact status between bolt and nut considering pitch difference

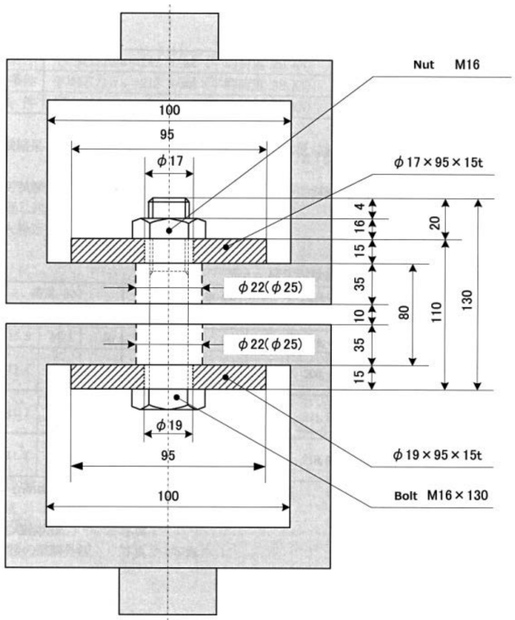


Fig. 3. Experimental device (dimensions in mm)

Table 3 Experimental loading conditions

Load (kN)		Stress (MPa)	
Mean load	Load amplitude	Mean stress	Stress amplitude
30	22.6	213	160
30	18.3	213	130
30	14.1	213	100
30	11.3	213	80
30	9.9	213	70
30	8.5	213	60

EXPERIMENTAL RESULTS

Figures 4-6 show the fractured specimens subjected the stress amplitude σ_a=100 MPa. For α=0, it is confirmed that the fracture always occurs at No.1 thread as shown in Figure 4. For α=15 μm and α=33 μm, the final fractured positions are between thread No.1 and thread No.3. The fractured surfaces of α=15 μm and α=33 μm are different from the one of α=0 because the surface is not flat.

Fig. 4. Specimen (α=0, σ_a=100 MPa)

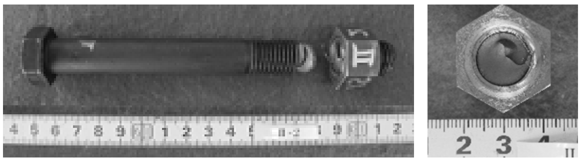


Fig. 5. Specimen (α=15 μm, σ_a=100 MPa)

Fig. 6. Specimen (α=33 μm, σ_a=100 MPa)

The S-N curves with fatigue limit at N=2×10^6 are obtained as shown in Figure 7. The fatigue limit is defined as the stress amplitude under which the specimen sustains N=2×10^6 stress cycles. It is found that the fatigue lives are depending on the three levels of pitch differences. Table 4 shows the fatigue life normalized by the results of α=0. When the stress amplitude is above 80 MPa, the fatigue life for α=15 μm is about 1.5 times and the fatigue life for α=33 μm is about 1.2 times of the standard bolt-nut connections (α=0). However, near the fatigue limit, the fatigue lives of the three types of specimens are similar, and the fatigue limits remain at the same value of 60 MPa.

CRACK OBSERVATION

Figure 8 shows the observed trajectory of cracks along the longitudinal cross section of the specimens at the fatigue stress amplitudes σ_a=60 MPa, 70 MPa, 100 MPa and 160 MPa. For α=0, small cracks occur at thread No.1 and thread No.2. For α=15 μm and α=33 μm, large cracks occur between thread No.2 and thread No.7. Moreover, with increasing the stress amplitude, the cracks show different shapes indicating changes in mode mixity.

It can be seen in Figure 8 that for the standard bolt-nut connections α=0, the crack occurs at thread No.1 causing finial fracture. However, for the specimens of α=15 μm and α=33 μm, the initial cracks start at thread No.5 or thread No.6, extending toward thread No.1 and finally fracture happen nearby thread No.1. From the S-N curves in Figure 7 and the observations of crack trajectories in Figure 8, we can conclude that the fatigue life of the bolt-nut connections may be extended by introducing a pitch difference because the changes in crack propagation trajectory may take place.

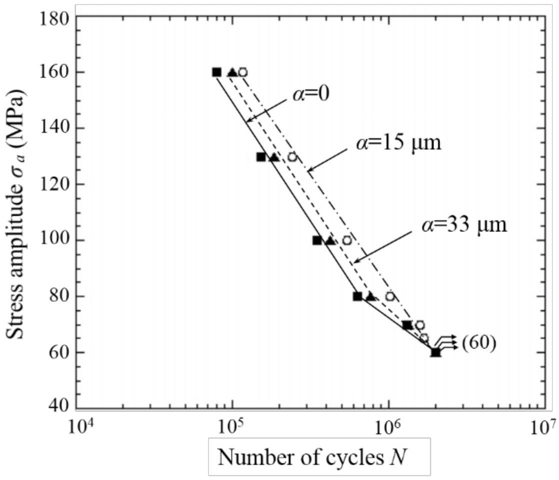


Fig. 7. S-N curves

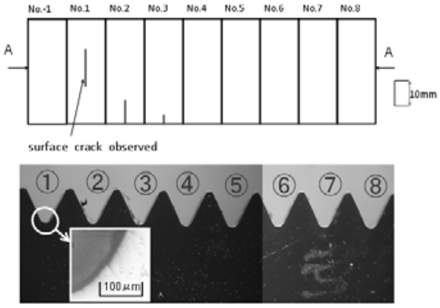
Table 4 The fatigue life improvement due to α (N: cycles of failure)

	α (μm)	Stress amplitude σ_a (MPa)				
		160	130	100	80	70
$\frac{N}{N_{\alpha=0}}$	0	1	1	1	1	1
	15	1.49	1.60	1.53	1.61	1.21
	33	1.26	1.22	1.20	1.21	1.02
N	0	79,030	151,860	350,760	636,490	1,307,860
	15	117,550	242,810	536,690	1,025,370	1,586,980
	33	99,680	184,770	422,500	770,870	1,327,860

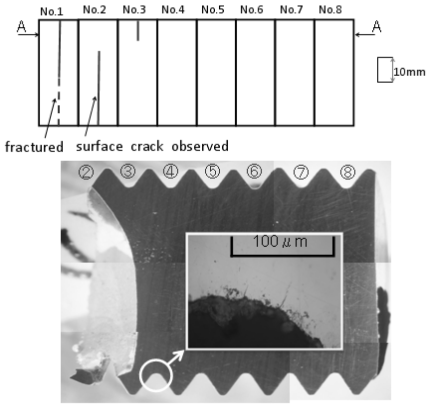
FINITE ELEMENT ANALYSIS

Stress analysis at bottom of bolt thread

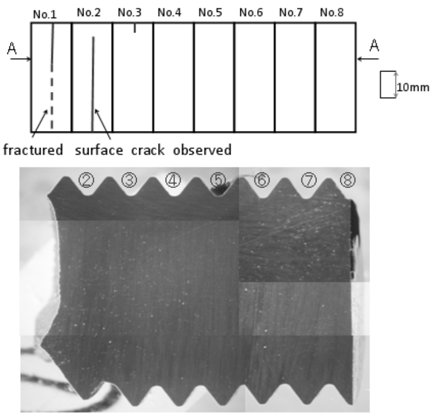
To analyze the stress states at the bottom of the bolt threads, finite element models are created by using FEM with MSC.Marc/Mentat 2007. Three models have different pitch differences, i.e. α=0, α=15 μm and α=33 μm, in accordance with the experimental configurations of the test specimens. Figure 9 shows the axisymmetric model of the bolt-nut connection and the clamped plate. Figure 10 shows the local coordinate at the bottom of bolt thread. An elastic-plastic analysis is performed for three models under the same load, i.e. F=30±14.1 kN. The material properties listed in Table 2 are used in the calculation.



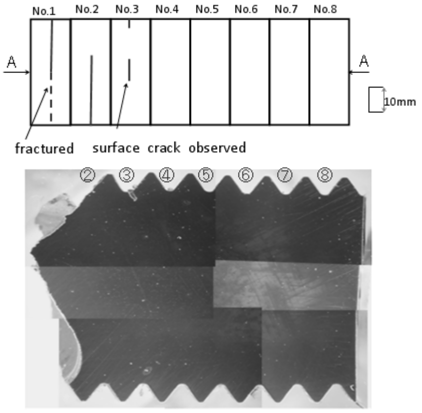
(α=0, σ_a=60 MPa)



(α=0, σ_a=70 MPa)

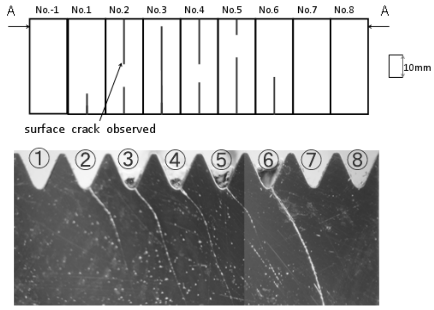


(α=0, σ_a=100 MPa)

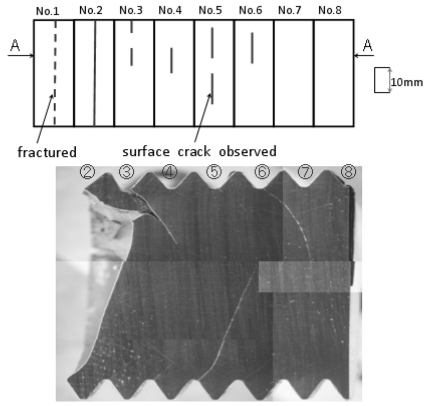


(α=0, σ_a=160 MPa)

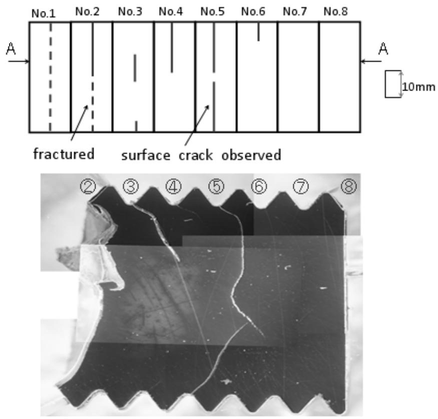
Fig. 8 (a). Observation of crack trajectories



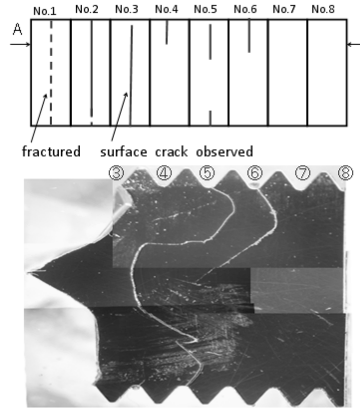
(α=15 μm, σ_a=60 MPa)



(α=15 μm, σ_a=70 MPa)

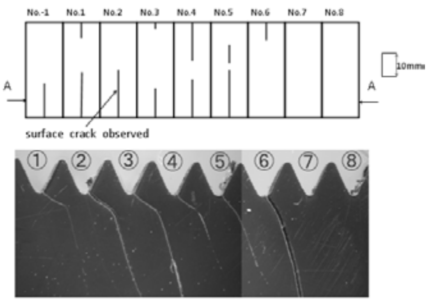


(α=15 μm, σ_a=100 MPa)

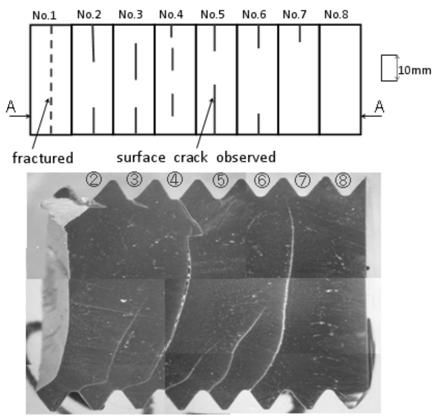


(α=15 µm, σ_a=160 MPa)

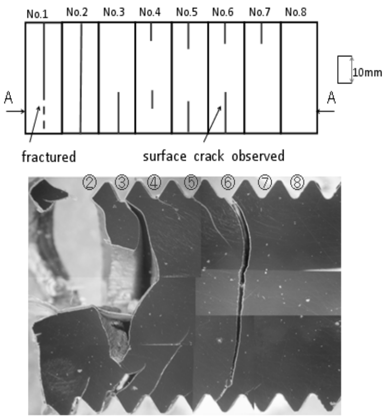
Fig. 8 (b). Observation of crack trajectories



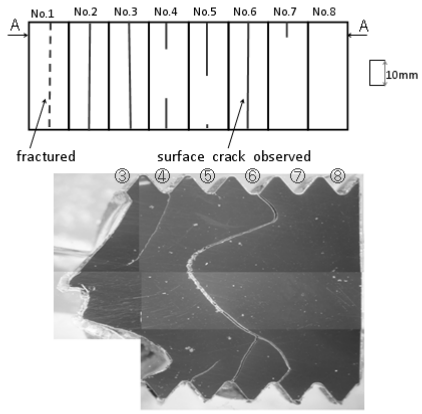
(α=33 µm, σ_a=60 MPa)



(α=33 µm, σ_a=70 MPa)



(α=33 µm, σ_a=100 MPa)



(α=33 µm, σ_a=160 MPa)

Fig. 8 (c). Observation of crack trajectories

Figure 11 illustrates the stress strain relation for SCM435 and S45C. The coefficient of friction between bolt and nut is 0.3. Figure 12 shows the stress variations for σ_ψ, σ_θ and the equivalent von-Mises stress σ_{eq} at each bolt thread from thread No.5 to thread No.8. Herein, the stress variation σ_ψ is taken into account. The position of the maximum stress amplitude is marked as shown in Figure 12. As indicated in Figure 12, at each bolt thread from thread No.-3 to thread No.8, the maximum stress amplitude and the mean stress are investigated at the point where the maximum stress amplitude appears.

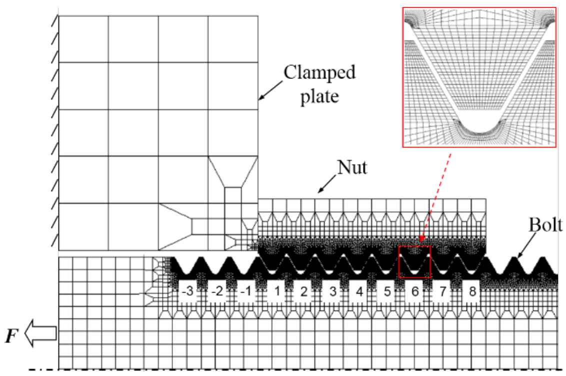


Fig. 9. Axisymmetric finite element model of bolt-nut connections

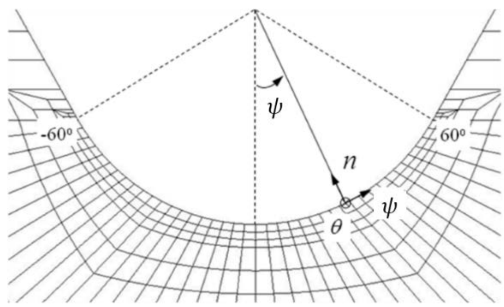


Fig. 10. Local coordinate at bottom of bolt thread

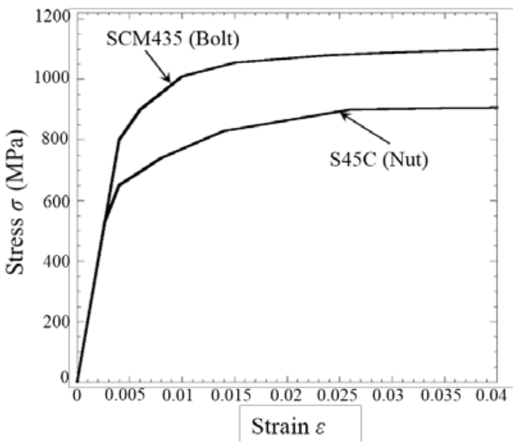
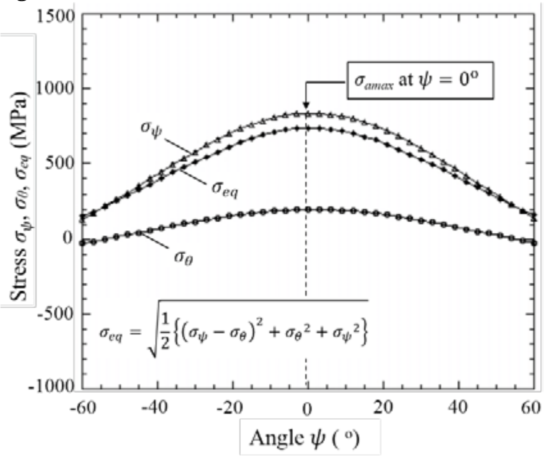
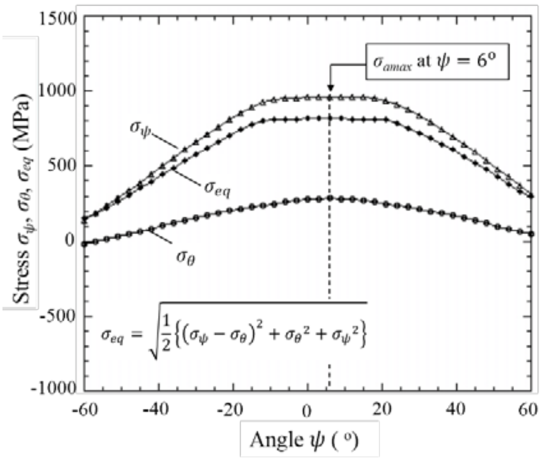


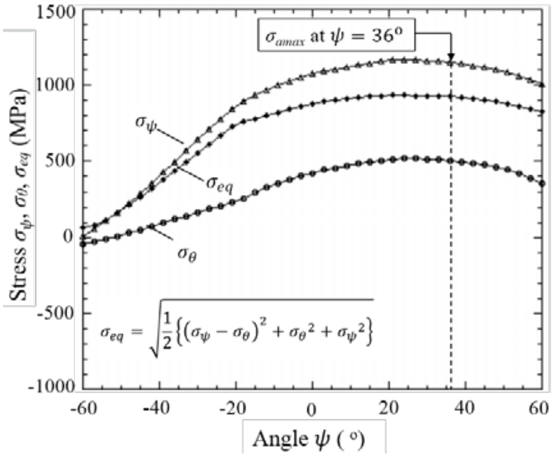
Fig. 11. Stress strain relation for SCM435 and S45C



(a) Stress at bottom of thread No.5

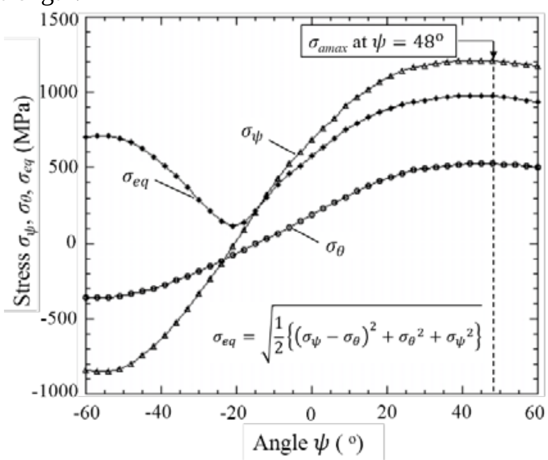


(b) Stress at bottom of thread No.6



(c) Stress at bottom of thread No.7

The endurance limit diagrams are obtained as shown in Figure 13. In the endurance limit diagram, the Soderberg line (Black et al., 1968) is plotted. Herein, the point σ_w represents the fatigue strength corresponding to the case of complete reversal (σ_m =0), and the point σ_{sl} corresponds to the yield strength.



(d) Stress at bottom of thread No.8

Fig. 12. Stress at bottom of bolt thread (α=15 μm, F=30+14.1 kN)

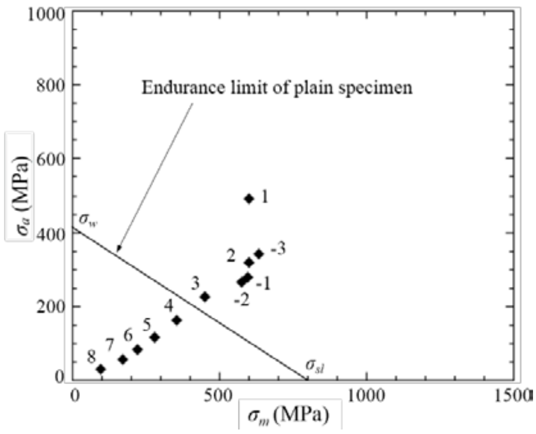
It should be noted that because of the stress gradient, the maximum stress amplitude for fracture of notched specimens is always larger than that of the plain specimens. Therefore, the stress data plotted beyond the line ($(\sigma_m/\sigma_{sl})+(\sigma_a/\sigma_w)>1$) does not represent the real fracture at the bolt thread. From Figure 13 (a), it can be seen that for the standard bolt-nut connections, the bottom of thread No.1 has the highest stress amplitude, which corresponds to the fracture position in the fatigue experiment as illustrated previously. In Figure 13 (b), when a pitch difference of α=15 μm is introduced, on one hand the stress amplitude decreases at the bottom of thread

No.1 and on the other hand, the stress amplitude at threads No.6, No.7 and No.8 increases significantly. For α=33 µm, the severe stress state occurs nearby threads No.1 and No.7 as shown in Figure 13 (c).

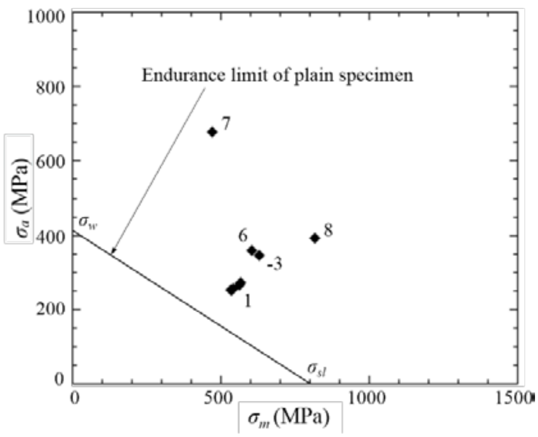
Effects of incomplete thread and clearance

The analysis shows that by introducing α=15 µm, high stress appears at threads No.7 and No.8. However, the crack observation shows that the initial crack occurs at thread No.5 instead of threads No.7 and No.8. In order to further investigate these results, the effects of the incomplete thread and the clearance between bolt and nut on the stress state are discussed. Figure 14 shows the FE model for the incomplete thread bolt.

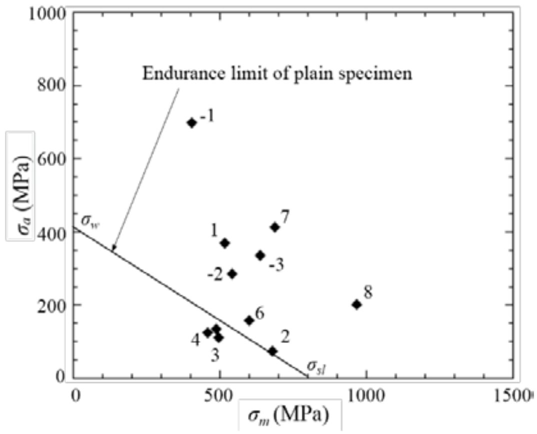
Figure 15 (a) shows a comparison between the results for the incomplete and complete threads. It is found that the stress amplitude at threads No.7 and No.8 decreases and the stress amplitude at thread No.6 increases by considering incomplete thread. In Figure 15 (b), the effect of clearance (see Figure 2) is indicated by comparing the results for clearance of 125 µm and 250 µm. In this experiment, the clearance between the bolt and nut is controlled at 125 µm, however, the nut may be inclined and the clearance become larger, i.e. 250 µm.



(a) α=0, σ_a=100 MPa



(b) α=15 µm, σ_a=100 MPa



(c) α=33 µm, σ_a=100 MPa

Fig. 13. Endurance limit diagrams

Figure 15 (b) shows that with increasing the clearance the stress amplitude at threads No.7 and No.8 decreases and the stress amplitude at thread No.6 increases. This may explain the reason why no crack is observed at threads No.7 and No.8 and crack initiates around thread No.5.

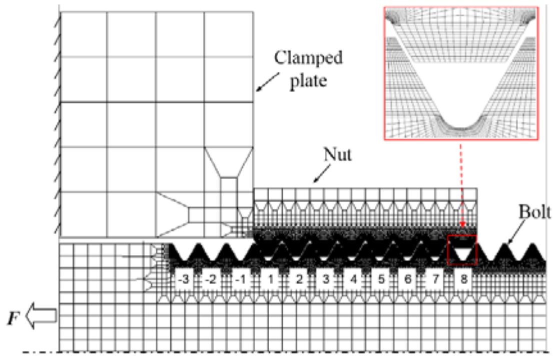
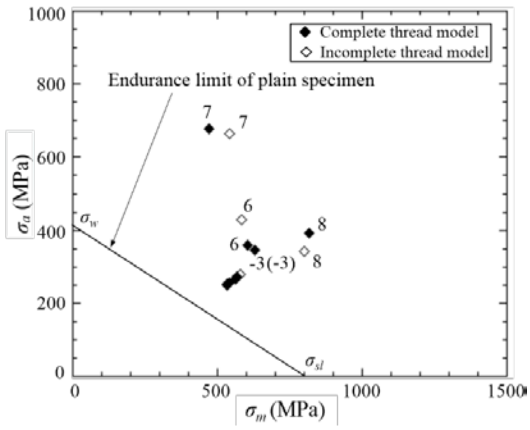
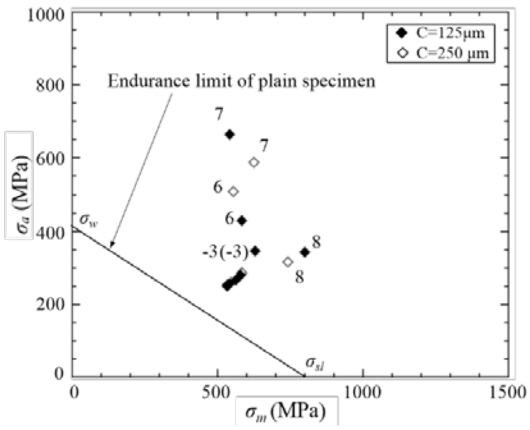


Fig. 14. Axisymmetric finite element model for incomplete thread



(a) Incomplete thread model vs complete thread model



(b) Clearance=125 μm vs clearance=250 μm

Fig. 15. Endurance limit diagram (α=15 μm, σ_a=100 MPa)

CONCLUSIONS

In order to realize both the fatigue life improvement and anti-loosening performance of bolt-nut connections, a slight pitch difference is introduced between M16 bolt and nut. In this paper, based on the obtained results in the previous anti-loosening study, the fatigue experiment was conducted for three levels of pitch difference, i.e. α=0, α=15μm, and α=33μm. The effect of the pitch difference on the stress state at bolt threads was numerically analyzed using the finite element method. The conclusions can be summarized as follows:

(1) Compared with the standard bolt-nut connection of α=0, the fatigue life for α=15 μm can be extended to about 1.5 times (see Fig. 7).

(2) It is found that the stress amplitude at thread No.1 decreases significantly when a pitch difference is introduced (see Fig. 13 (a) and (b)). For α=15 μm, the FE results shows that high stress amplitude occurs at threads No.6, No.7 and No.8, which almost corresponds to the experimental observations (see Fig. 13 (b) and Fig. 8).

(3) For the specimens with α=15 μm, it is found that the crack occurs at thread No.5 in the first place, then extends toward thread No.1 until final fracture happens nearby thread No.1. Therefore, the fatigue life of the bolt is extended compared with the standard bolt-nut connection (see Fig. 8 and Fig. 7).

(4) In FE analysis high stresses appear at threads No.7 and No.8, although in the experimental observation the cracks happen at thread No.5. The experimental results can be explained more clearly by considering the incomplete threads in FE simulation (see Fig. 15).

The M16 bolt-nut connections (JIS) are employed here as the study objective. The extended study will discuss more connections of different sizes in a further research.

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微小螺距差對螺栓螺母聯接件的疲勞壽命的影響

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摘 要

本文旨在探討微小螺距差對螺栓螺母聯接件的疲勞壽命的影響. 通過增大螺母的螺距使得螺紋之間的接觸狀況發生變化. 疲勞試驗結果顯示合理的螺距差可以大幅度提高螺栓的疲勞壽命, 螺栓也呈現出不同的疲勞破壞形態. 通過對試件的裂紋觀察,發現不同的螺距差導致螺栓在疲勞破話過程中產生不同形態的裂紋.