Analytical and Experimental Study on Anti-Loosening Performance and Fatigue Life Improvement for Bolt-Nut Connections Having Slight Pitch Difference

Analytische en experimentele studie van de weerstand tegen losdraaien
en de verhoging van de vermoeiingslevensduur voor bout-moerverbindingen
met licht verschillende spoed

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Summary

Bolt-nut connections can be regarded as the most important fastening elements used to connect and disconnect mechanical components conveniently with low cost. Surprisingly a large number of bolt-nut connections are used in a wide variety of machines and structures, such as machine tools, construction machinery, steel towers, bridges, transportation equipments, etc. However, self-loosening of bolt–nut connections often occurs when the fastener are subjected to dynamic external loads, such as impact and vibrations. Besides, fatigue failure of bolt is also always of concern, which sometimes leads to severe accidents. To ensure the connected structure's safety, the anti-loosening performance and high fatigue strength are required with low cost.

Most previous studies on special bolt-nut connections 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 appearing at the first bolt thread cannot be reduced very easily. Moreover, usually for special bolt-nut connections the anti-loosening ability affects the fatigue strength and the low price significantly.

The final objective of this thesis is to develop a special bolt-nut connection, which can realize both anti-loosening performance and fatigue strength improvement without raising the cost. Motivated by this conception, a slight pitch difference is introduced between bolt and nut. If the nut pitch is larger than the bolt pitch, the maximum average stress and maximum stress amplitude at the first bolt thread can be reduced because the

contact status of the first thread changes to no contact status after the loading. However, if the nut pitch is smaller than the bolt pitch, the contact status of the first bolt thread before the loading is not changed and the contact force just becomes larger than the contact force of normal bolt-nut connection after the loading. In this study, the methodology includes both experimental approach to realize anti-loosening performance and fatigue strength improvement, and finite element analysis approach to describe the stress state at special bolt threads and explaining the improvement mechanism.

In this study a standard size M16 (JIS) bolt-nut connection is considered where the nut pitch is slightly larger than the bolt pitch. The fatigue experiments as well as the loosening experiments are conducted under different pitch differences. Axisymmetric models are created and analyzed by finite element method to explain the stress state at the contact threads between bolt and nut.

In the first place, in order to investigate the effect of pitch difference on the fatigue life, the fatigue experiments are conducted for the specimens having three types of pitch differences, namely standard bolt-nut connection (no pitch difference), small and very small pitch differences. Then, it is found that the fatigue life of bolt can be extended by introducing suitable pitch differences. To clarify the effect of pitch difference, finite element method is used to analyze the stress state at each bolt thread. The analysis shows that both the average stress and the stress amplitude at the first bolt thread can be reduced by introducing a suitable pitch difference although large stress appears at the seventh and eighth threads instead of the first thread for small pitch difference. Next, to investigate the effect of the clearance between bolt and nut, the commonly used maximum and minimum clearances are considered. Then, the effect of the clearance on the fatigue strength is discussed considering the contact status between the bolt-nut connection threads.

Secondly, in order to clarify the effect of pitch difference on the anti-loosening performance, a series of pitch differences, namely standard bolt-nut connection, small, middle, large and very large, are used in the loosening experiments based on NAS3350. Also, the prevailing torque necessary for the nut rotation is measured experimentally for those pitch differences before the nut touching the clamped body. Then, the bolt axial force is investigated as a function of the tightening torque after the nut touching the clamped body. It is found that the large values of pitch difference may provide large prevailing torque, which results in anti-loosening performance although too large pitch difference may deteriorate the bolt clamping ability. Finally, a middle value of pitch difference is found to be the most desirable pitch difference to obtain the anti-loosening performance without losing the clamping ability. By applying the finite element method to the screwing process, the results show that large plastic deformation happens at nut threads for a very large pitch difference. The mechanism of anti-loosening for bolt-nut having slight pitch difference are discussed.

Thirdly, since it is found that a middle value of pitch difference is the most suitable pitch difference for anti-loosening, the fatigue life mechanism is improved for standard bolt-nut connection, small and middle value of pitch difference under various stress amplitudes. Then, it is found that the fatigue life for a middle value of pitch difference is 1.2 times larger than that of a standard bolt-nut connection. However, the obtained S-N curves show that the fatigue life for a small value of pitch difference is the most desirable, which is 1.5 times larger than that of a standard bolt-nut connection. Detailed investigation is also performed on the crack configuration of the fractured specimens. For the specimens of small and middle values of pitch difference, it is found that the crack initiates and propagates at the fifth and sixth bolt threads at the beginning of the experiment, and after that, new cracks appear at fourth and third threads, toward the first thread until final fracture happens. The fatigue life extension mechanism can be explained in this way. By applying the finite element method, the fatigue life improvement is discussed in terms of the stress amplitude and average stress at each bolt thread.

Furthermore, in order to improve the accuracy of the finite element analysis, the chamfered corners at both nut ends are considered by using a 6-thread-model. It is found that the 6-thread-model is useful for analysing 8-thread-nut contacting bolt threads because nuts always have chamfered threads at both ends. For small and middle values of pitch difference, the finite element analysis shows that high stress amplitude occurs at the sixth and seventh threads, and the results are in good agreement with the experimental results.

At the end of this thesis, main conclusions of this study are summarized for fatigue life improvement and anti-loosening performance. And finally, a method for how to find out a suitable pitch difference to improve both anti-loosening and fatigue life is proposed.

Samenvatting

(Dutch summary)

Bout-moer verbindingen kunnen worden beschouwd als de belangrijkste bevestigingselementen die kunnen gebruikt worden om verbinding te maken van mechanische componenten op een gemakkelijk manier en met lage kosten. Verrassend worden er een groot aantal van de bout-moer verbindingen gebruikt in een breed scala van machines en structuren, zoals mechanische constructies, bouwmachines, stalen torens, bruggen, transportmiddelen, etc. Echter, zelf-loskomen van de bout-moer verbindingen vaak voorkomt wanneer het bevestigingsmiddel blootgesteld wordt aan externe dynamische belastingen, zoals stoten en trillingen. Trouwens, is vermoeiing breuk van de bout ook altijd van belang, wat soms leidt tot ernstige ongevallen. Om de veiligheid van de aangesloten structuur te waarborgen, zijn de anti-loskomen prestaties en hoge vermoeiingssterkte met lage kosten vereist.

De meeste eerdere studies op speciale bout-moer verbindingen zijn vooral gericht op het ontwikkelen van anti-loskomen prestaties, en enkele studies bijdragen naar het verbeteren van de vermoeiingssterkte en levensduur. Dit is omdat een hoge spanningsconcentratie die verschijnt op de eerste boutdraad niet gemakkelijk kan worden verminderd. Bovendien, meestal voor speciale bout-moer verbindingen, beïnvloedt het anti-loskomen mogelijkheid de vermoeidheid kracht en de lage prijs aanzienlijk.

Het uiteindelijke doel van dit proefschrift is de ontwikkeling van een speciale bout-moer-verbinding, die zowel anti-loskomen prestaties en de verbetering van vermoeiingssterkte kan realiseren zonder verhoging van de kosten. Gemotiveerd door dit concept, wordt er een toonhoogteverschil ingevoerd tussen bout en moer. Als de moer toonhoogte groter dan de bout toonhoogte is, wordt de maximale gemiddelde stress en maximale spanning amplitude op de eerste bout draad verlaagd omdat de eerste draad voor belasting in contactstatus is en na de belasting in niet-contactstatus is. Indien de toonhoogte van de moer kleiner is dan de toonhoogte van de bout, wordt de contactstatus van de eerst draad van de bout voor de belasting niet gewijzigd, en de wordt contactkracht net groter dan de contactkracht van standaard bout-moer verbinding na de belasting. In deze studie, omvat de methodologie zowel experimentele benadering om anti-loskomen prestaties en de verbetering van de vermoeiingskracht te realiseren, als eindige elementenmethode analyse aanpak van de spanningstoestand te beschrijven op speciale schroefdraad en verduidelijking van de verbetering van het mechanisme.

In deze studie wordt er een standaard formaat M16 bout-moer verbinding beschouwd, waar de moer toonhoogte iets groter is dan de bout toonhoogte. De vermoeiingsexperimenten en het loskomen experimenten worden uitgevoerd onder een aantal toonhoogteverschillen. Axis-symmetrische modellen worden gemaakt en door eindige elementenmethode geanalyseerd om de spanningstoestand in het contact draden tussen bout en moer te verduidelijken.

In de eerste instantie, om het effect van de toonhoogteverschil op de levensduur te onderzoeken, wordt de vermoeidheid experiment uitgevoerd voor monsters met drie soorten toonhoogteverschillen, namelijk standaard bout-moer verbinding (geen toonhoogteverschil), klein en zeer klein toonhoogteverschillen. Dan blijkt het dat de levensduur van de bout kan worden verlengd door het introduceren van geschikte toonhoogteverschillen, bijvoorbeeld klein en zeer klein waarden. Om het effect van toonhoogteverschil te verduidelijken, wordt de eindige elementenmethode gebruikt om de spanningstoestand te analyseren voor ieder boutdraad. De analyse toont dat zowel de gemiddelde spanning en de spanning amplitude bij de eerste boutdraad kan worden verminderd door het invoeren van een geschikte toonhoogteverschil, hoewel grote spanning verschijnt in de zevende en achtende schroefdraden in plaats van de eerste schroefdraad voor een klein toonhoogteverschil. Vervolgens wordt het effect van de speling tussen de bout en moer onderzocht door de bekende methode van maximale en minimale speling. Vervolgens wordt het effect van de speling op de vermoeiingssterkte besproken gelet op contactstatus tussen de bout-moer verbindingsdraad.

Ten tweede, om het effect van de toonhoogteverschil op de anti-loskomen prestaties te tonen, wordt een reeks toonhoogteverschillen, standaard bout-moer verbindingen, klein, gemiddelde, groot en zeer groot, gebruikt op loskomen experimenten. Ook wordt de draaimoment voor de moer rotatie experimenteel gemeten voor deze toonhoogteverschillen voordat de moer het geklemde lichaam aanraakt. Vervolgens wordt de bout axiale kracht onderzocht als functie van het aandraaimoment nadat de moer het geklemde lichaam aanraakt. Het blijkt dat de grote waarden van toonhoogteverschil grote draaimoment voorziet, hoewel te groot toonhoogteverschil kan leidt tot versleten van de klemmen capaciteit. Tenslotte blijkt het dat een gemiddelde waarde van toonhoogteverschil het meest wenselijke toonhoogteverschil om de anti-losmaken werking te verkrijgen zonder verlies van de klemmen capaciteit. Door toepassing van de eindige elementen analyses over de schroeven proces, tonen de resultaten aan dat grote plastische vervorming gebeurt bij de moerdraad voor zeer groot toonhoogteverschil. Het mechanisme van de anti-loskomen van geschroefde moer die klein toonhoogteverschil wordt besproken.

Ten derde, aangezien het blijkt dat een gemiddelde waarde van toonhoogteverschil het meest geschikt toonhoogteverschil voor anti-losmaken is, is de vermoeiingssterkte levensduur verbeterd voor standaard bout-moer verbinding, klein en gemiddelde toonhoogteverschil onder verschillende spanning amplitudes. Dan blijkt het dat de levensduur van een gemiddelde waarde van toonhoogteverschil is 1,2 keer groter dan die van standaard bout-moer verbinding. De verkregen S-N curves tonen dat de levensduur voor een kleine waarde van toonhoogteverschil is het wenselijk, dat is 1,5 keer groter dan die van standaard bout-moer verbinding. Gedetailleerd onderzoek wordt ook uitgevoerd op de scheur configuratie van de gebroken monsters. Voor de monsters van kleine en gemiddelde waarden van toonhoogteverschil, blijkt het dat de scheur initieert en propageert bij de vijfde en zesde bout draadden in het begin van het experiment, en daarna nieuwe scheuren verschijnen bij de vierde en derde bout draadden, in de richting van de eerste draad tot dat de definitieve breuk gebeurt. De levensduur uittrektechniek te verklaren op deze manier. Door toepassing van de eindige elementenanalyse, wordt de verbetering van levensduur besproken in termen van de gemiddelde spanning en de spanning amplitude bij elke boutdraad.

Bovendien, om de nauwkeurigheid van de eindige elementenanalyse te verbeteren, worden de afgeschuinde hoeken aan beide uiteinden van de moer gemodelleerd met behulp van een 6-draad -model. Het blijkt dat de 6-draad-model nuttig is voor het analyseren van 8-draad-moer in contact omdat moeren altijd draden afgeschuind hebben aan beide uiteinden. Voor kleine en gemiddelde waarden van toonhoogteverschil, de eindige elementenanalyse tonen dat hoge stress amplitude optreedt bij de zesde en de zevende draden, en de resultaten zijn in goede overeenstemming met de experimentele resultaten.

Aan het einde van dit proefschrift worden de belangrijkste conclusies van dit onderzoek samengevat voor verbetering levensduur en anti-loskomen prestaties. En tot slot, de methode hoe om uit te vinden van een geschikte toonhoogteverschil met zowel anti-loskomen als vermoeiing levensduur te verbeteren wordt voorgesteld door te overwegen de aanpassing van de speling tussen bout en moer.

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Nomenclature

FEM	finite element method
JIS	Japanese Industrial Standard
A_R	cross sectional area of the bolt
C_y	clearance between bolt and nut in y direction
C_x	clearance between bolt and nut in x direction
C_{max}	maximum clearance in y direction
C_{min}	minimum clearance in y direction
D_{max}^{nut}	maximum effective diameter of nut
D_{min}^{nut}	minimum effective diameter of nut
d_{max}^{bolt}	maximum effective diameter of bolt
d_{min}^{bolt}	minimum effective diameter of bolt
F	load
$F_{average}$	average load
F_{max}	maximum load
F_{min}	minimum load
F_{α}	bolt axial force between the nut threads
K_t	stress concentration factor
n	nut thread number
n_c	contacted thread number of nut

N	number of cycles to failure
p	pitch of bolt
T	torque
T_p	prevailing torque
α, $\alpha1$, $\alpha2$, $\alpha3$, $\alpha4$, $\alpha5$, $\alpha6$, $\alpha7$, $\alpha8$, $\alpha9$	pitch difference
$\alpha_{verysmall}$, α_{small}, α_{middle}, α_{large}, $\alpha_{verylarge}$	pitch difference
δ_t	distance where the prevailing torque appears
θ	thread angle
σ_a	stress amplitude
σ_m	average stress
σ_{tmax}	maximum tangential stress
σ_n	normal stress
σ_w	fatigue limit
σ_s	yield stress
σ_ψ, σ_θ	stress variations defined by local coordinate at bolt thread
σ_{eq}	equivalent von-Mises stress
ρ	root radius of bolt

Chapter 1

Introduction

1.1 The history of bolt

Bolt-nut connections are one of the most common mechanical elements. In general, bolt-nut connections have the following advantages:

1. They can be easily assembled and disassembled;
2. They can be set while making necessary adjustment, or can be set with high precision with simple fastening tools;
3. As the wedge effect of threads can be utilized, even very thick members can be fastened tightly.

Because of these advantages, surprisingly large numbers of bolt-nut connections are used in a wide variety of machines and structures, such as machine tools, construction machinery, steel towers, bridges, transportation equipment, etc.

While the history of threads [1] can be traced back to 400 BC, the most significant developments of bolt were made during the last 150 years. Initially, screw threads for fasteners were made by hand. Because of a significant increase in demand, it was necessary to speed up the production process. In Britain in 1760, J and W Wyatt introduced a factory process for the mass production of screw threads. However, this led to another problem: each company produced its own threads, nuts and bolts so there was a huge range of different sized screw threads on the market. Until 1841, Joseph Whitworth suggested standardizing the size of the screw threads in Britain. Therefore, someone could make a bolt in England and someone in Glasgow could make the nut and they would both fit together. His proposal was that the angle of the thread flanks was standardized at 55 degrees, and the number of threads per inch, should be defined for various diameters.

In 1864, William Sellers proposed a 60 degree thread and various thread pitches for different diameters. This developed into the American Standard Coarse Series and the Fine Series. One advantage the Americans had over the British was that their thread form had flat root and crests. This made it easier to manufacture than the Whitworth standard, which had rounded roots and crests. It was found that the Whitworth thread performed better in dynamic applications while the rounded root of the Whitworth thread improved fatigue performances.

During World War I and World War II, the lack of consistency between screw threads in different countries became a huge obstacle on the war effort. In 1948, Britain, the USA and Canada agreed on the Unified thread as the standard for all countries that used imperial measurements. It uses a similar profile as the DIN metric thread previously developed in Germany in1919. This was a combination of the best of the Whitworth thread form (the rounded root to improve fatigue performance) and the Sellers thread (60 degree flank angle and flat crests). This led to the ISO (International Standards Organization) metric thread which is used in all industrialized countries today.

Japanese Industrial Standards (JIS) specifies the standards used for industrial activities in Japan. ISO, JIS and DIN standards are based upon the metric system and are closely related. The screw thread specifications based on JIS also apply to ISO and DIN threads.

1.2 Fatigue failure and self-loosening of bolt-nut connection

1.2.1 Fatigue failure of bolt

Fatigue failure of bolt is always of concern, which sometimes leads to severe accidents. Figure 1.1 shows a giant bolt (called tie rod – outside diameter of thread = 478 mm) failed after using 3 years in 1975, Japan. The investigation on the fracture surface reported that the failure is attributable to the fatigue crack which initiated at the thread root and propagated [2].

Figure 1.1: A giant bolt failed because of the fatigue fracture, 1975, Japan [2].

The main cause for the fatigue failure of bolt is the uneven load sharing among the bolt thread, and the first loaded thread carries more load than all the subsequent threads as shown in Figure 1.2. The high stress concentration exists at thread roots.

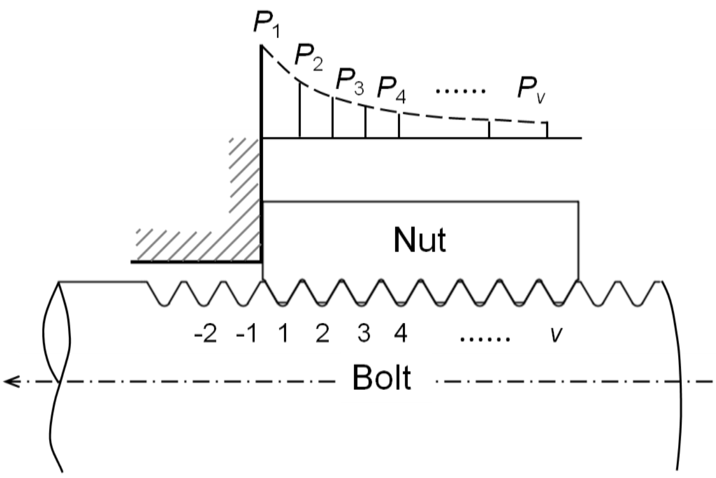


Figure 1.2: Load distribution on bolt threads [2].

1.2.2 Self-loosening

A significant advantage of a bolted joint over other joint types, such as welded and riveted joints, is that they are capable of being dismantled. This feature, however, can cause problems if it unintentionally occurs. Such unintentional loosening, frequently called self-loosening, often occurs when the fastener are subjected to dynamic external loads, such as impact and vibrations, self-loosening results in the failure of engineering products such as railway cars, vehicles, and construction machines.

Figure 1.3 shows a railway accident that a high speed train derailed in Cumbria, UK on Friday 23 February 2007 [3]. This accident was as a result of nuts becoming detached from the bolt allowing the switch rail to be struck by the inner faces of passing train wheels. This caused subsequent failures of other parts of the switch structure and ultimately the derailment of the train.

Figure 1.3: A high speed train derailed, 2007, UK [3].

1.3 Objectives

Fatigue failure of bolts and self-loosening of bolt-nut combinations has been subject of intensive research for many years and several concepts have been developed in an attempt to improve the integrity of the bolt-nut connections. Amongst these concepts, the advantageous influence on fatigue life of modifying the nut pitch so that it's slightly different from the pitch of the bolt threads is already known for some decades.

In this study, the effect of pitch difference on the fatigue life improvement as well as the anti-loosening performance will be discussed systematically. The main objective of this thesis is to develop a type of bolt-nut connection which can realize both anti-loosening performance and fatigue strength improvement without raising the cost. The objectives are summarized as follows:

1) Investigating the effect of pitch difference on the anti-loosening performance
2) To obtain the *S-N* curves for specimen with different pitch differences
3) Discussing the suitable pitch difference which can realize anti-loosening performance and fatigue life improvement
4) Using Finite Element Analysis to describe the effect of pitch difference on the stress state at bolt threads
5) Discussing the practical application of this type of bolt-nut connections

1.4 Thesis scopes and methodology

This thesis is organized as follows:

Chapter 1 gives an introduction on the background of this study, including the brief history of bolt-nut connections and the fatigue failure and self-loosening problems, followed by the objects of this thesis.

Chapter 2 presents a literature review on the relevant works. It consists of the research on fatigue strength improvement of bolt, the research on self-loosening and anti-loosening nut, and the effect of pitch difference on fatigue failure of bolt.

Chapter 3 studies the effect of pitch difference on the fatigue strength of bolt –nut connection. Fatigue fracture is experimentally investigated for three types of pitch differences and FEM is used to analyze and explain the experimental findings. The effect of clearance between bolt and nut of the stress state is also discussed analytically.

Chapter 4 reports the effect of pitch difference on the anti-loosening performance as well as its mechanism. In the first place, with varying the pitch difference, the prevailing torque necessary for the nut rotation before the nut touching the clamped body is measured experimentally. Next, the tightening torque is investigated in relation to the bolt axial force after the nut touching the clamped body. Then, based on the loosening experiment results, the suitable pitch difference is discussed in terms of the anti-loosening ability without losing the clamping ability. Finally, the FEM is applied to analyze the deformation of the contacted threads between the bolt and nut.

Chapter 5 describes the mechanism of fatigue life improvement for bolt-nut having pitch difference. The fatigue experiments are conducted under different values of pitch difference with varying stress amplitude systematically. The fractured specimens are detailed investigated on the crack configuration. To improve the accuracy of the finite element analysis, the chamfered corners at both nut ends are considered. Eventually, the most desirable pitch difference is discussed in terms of improving both anti-loosening and fatigue life.

Chapter 6 outlines the conclusions of this thesis and makes recommendations for future work.

Chapter 2

Literature review on the special bolt-nut connection

2.1 Research on fatigue strength improvement of bolt

During the last few decades, many investigations related to the fatigue failure of bolt-nut connections have been carried out by using the experimental method which is the most basic technique. Yakushev [4] investigated the effect of manufacturing technology on the fatigue strength of thread connections. His work showed that the rolled screw improve the fatigue strength significantly compared with the cut threads and grinded thread. Majzoobi et al [5] studied the thread pitch and found that ISO standard coarse threaded bolts have a higher fatigue life than the fine threaded bolts. Nishida [6] discussed the effect of type of thread on the fatigue life of screws, including triangular thread, trapezoidal thread, positive buttress thread and negative buttress thread. It was found that the traditional triangular thread has an excellent total balance when considering fatigue strength and machinability. Nishida also proposed the tapered bolt, named CD bolt (Critical Design for Fracture), which has been confirmed that the new profile approximately doubles the fatigue strength of bolts as compared to the traditional profiles [7, 8]. Hirai and Uno [9] developed a new super high tension bolt by considering the R-r shape thread, which has two different radii at the bottom of bolt thread, and it was shown that the stress concentration factor could be reduced to the 60% value of the conventional high tension bolt thread. Table 2.1 summarized the general parameters controlling the fatigue strength of bolt [2]. The non-uniform load distribution, stress concentration and non-uniform contact are three factors that lead to the low fatigue strength of bolt. Compared to the other possible countermeasures, CD bolt addresses all the features which lower the fatigue strength of normal bolts. Table 2.2 shows three commonly used techniques for fatigue strength improvement, including the CD-bolt, pre-tensioning of bolt and low strength nut [2].

Table 2.1: Controlling Parameters for Fatigue Strength of Bolt-Nut Connections

	Improving non-uniform load distribution?	Reducing stress concentration?	Improving non-uniform Contact?
Nut Shape Improvement	Yes	?	?
Bolt Shape Improvement	Yes	?	?
Thread Root Improvement	?	Yes	?
Soft Nut	Yes	?	Yes
CD Bolt	Yes	Yes	Yes

Table 2.2 Fatigue strength improvement for different techniques

Method	Fatigue strength improvement
CD bolt	80% up
Pre-tensioning of bolt (Tightening in plastic range)	25% up
Low strength material of nut (SNCM630 → S20C)	17% up

In addition to the factors of bolt itself, some studies also paid attention on the effect of the tightening or loading conditions on the fatigue life of bolted joints. Suzuki et al. and Kawano et al [10, 11] reported the fatigue characteristics of bolted joints tightened in elastic and plastic region. Hobbs et al [12] discussed the effect of eccentric loading on the fatigue performance of high-tensile bolts.

The analytical and numerical methods have also been applied to clarify the stress status at the bolt threads. Kenny and Patterson [13-15] studied the load and stress distribution in a bolt-nut connector by 3-D frozen-stress photo-elastic analysis and compared their results with theoretical and numerical solutions. The axisymmetric finite element analyses have been studied in some literature [16, 17]. Chen [18] compared the load ratio of axisymmetric model with three-dimensional model. The results show the loading ratio is quite similar on each thread between axisymmetric and three-dimensional models, from which, it was concluded that the helical effect does not influence the load distribution and the axisymmetric model can give a good estimation of load distributions. The authors previously analyzed the tapered threads with the finite element method based on the proposal by Nishida, and discussed the stress reduction effect of the tapered thread under several geometrical conditions [19].

2.2 Research on self-loosening and anti-loosening nut

Many experiments have been performed in order to find out the reasons of self-loosening of bolt-nut connections. Jounker and Yamamoto et al [20, 21] reported that the loosening is mainly due to external shear load, which is perpendicular to the axial. Pai et al and Kasei et al [22, 23] studied that slight loosening is caused by the slip of the bolt head. Sase et al [24] pointed out that the loosening is caused by a twist caused by a relative slip between the male screws and the female screws and a slackening of the bolt torsion between the bearing surfaces. Moreover, Izumi et al and Pai et al [25, 26] applied Finite Element Method on these types of research.

To prevent the self-loosening, many special anti-loosening bolts and anti-loosening nuts have been invented and discussed in the past decades [27-31]. Here, several anti-loosening nut will be introduced.

Super Lock Nut

The Super Lock Nut (SLN) is developed in order to prevent self-loosening. Figure 2.1 shows the super lock nut. There is a thin walled tube between the upper and lower thread, which can be deformed along the axial direction so that the phase difference of lower and upper threads is produced. This phase difference induces the contrary forces on the surfaces of upper and lower threads, which bring out the anti-loosening performance [32].

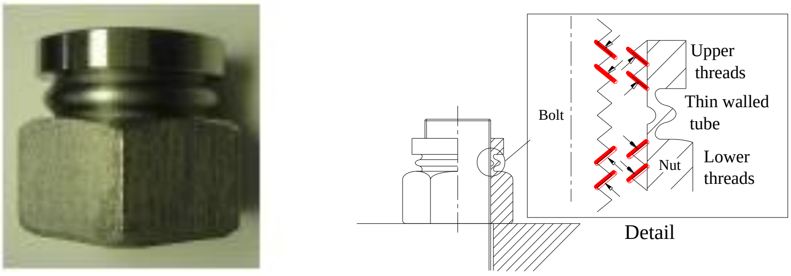


Figure 2.1: Super Lock Nut

Hard Lock Nut

Figure 2.2 shows a schematic diagram of Hard Lock Nut [33]. Hard lock nut have been used successfully in many applications worldwide for more than a decade. Hard lock nut uses a unique wedge principle to create a powerful self-locking force. A small curve in the sliding part of the convex top of the lower nut acts as the wedge. When the concave upper nut is tightened, the effect produced is exactly the same as that produced by a hammer driving in a wedge [34].

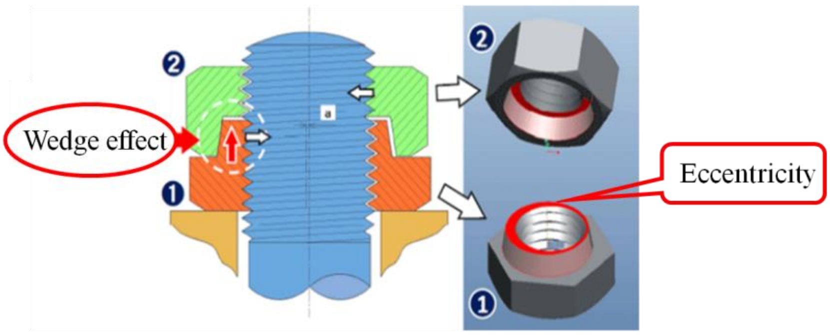


Figure 2.2: Hard lock nut [33]

Double Nut

"Double nut method" is a commonly used countermeasure against loosening. If tightened properly, double-nutting can generate a large rotation resistance torque. Figure 2.3 shows the proper tightening method of double nut applications [35]. First of all, tighten the lower nut to the specified torque. Then, tighten the upper nut to the specified torque. Holding the upper nut in place, turn the lower nut in the loosening direction until the upper and lower nuts are compressed mutually by the thread surfaces. Since it is the upper nut that determines the axial tension, it is important to properly control the torque of the upper nut. Also, because it is the upper nut that supports the axial tension, it is necessary for the upper nut to have a sufficient height.

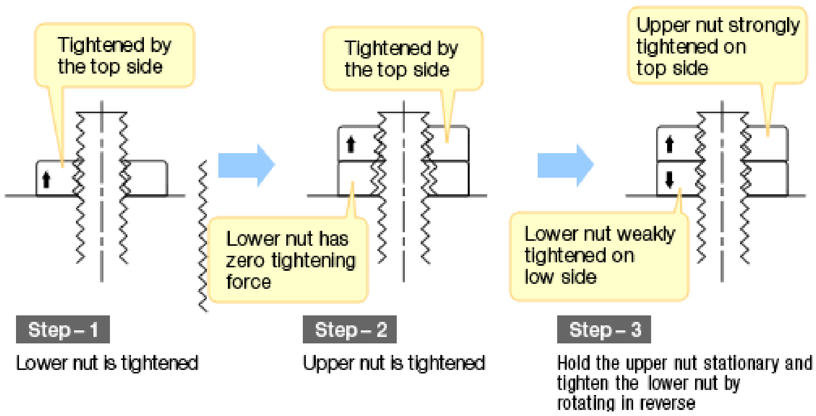


Figure 2.3: Tightening method of double nut [35]

2.3 Effect of pitch difference on fatigue failure of bolt

The concept of differential pitch was first suggested by Stromeyer [36] in 1918. He suggested that the load distribution in a threaded connection thread could be optimized by varying the relative pitches. Then, the theoretical load distribution in bolt-nut has been developed by Sopwith [37], who also used his formula to discuss the load distribution improvement along the bolt threads by varing pitch. He found that a smaller pitch in the bolt than in the nut would improve the load distribution. Sparling [38] found that the fatigue strength of the bolt can be improved by increasing the clearance between the first few engaged threads at the load bearing face of the nut by tapering the nut thread, which produces an effective difference in pitch. This modification was investigated by Kenny and Patterson [39] by applying the frozen stress three-dimensional photoelasticity.

Maruyama [40] analyzed the influence of pitch error and the loaded flank angle error of the bolt thread upon the stress at the root of the bolt thread by copper-electroplating method with the finite element method. It was considered that the pitch adjustment has a larger effect than the flank angle adjustment for improving the fatigue strength of the bolt thread.

Recently, Ward [41] reported that over-pitching of nuts, that is, increasing the pitch slightly compared to the mating component - has also proven successful in improving fatigue properties by distributing the load more evenly over the engaged threads. The same effect can be achieved by de-pitching of bolts. The philosophy behind this is simple. For standard fasteners, as the joint is loaded, the bolt is stretched and the nut is compressed, giving a pitch error that causes a poor distribution of load. In over-pitching the nut thread, the thread pitches can be engineered to match in the loaded condition. The optimal amount of over-pitching depends on the material properties, bolt dimensions and thread form.

However, the previous studies on pitch difference were limited to fatigue strength improvement, and the effect of pitch difference on the anti-loosening performance has not been investigated yet. There is no systematic experimental data are available, e.g. the S-N curves for specimens of different pitch differences have not been obtained.

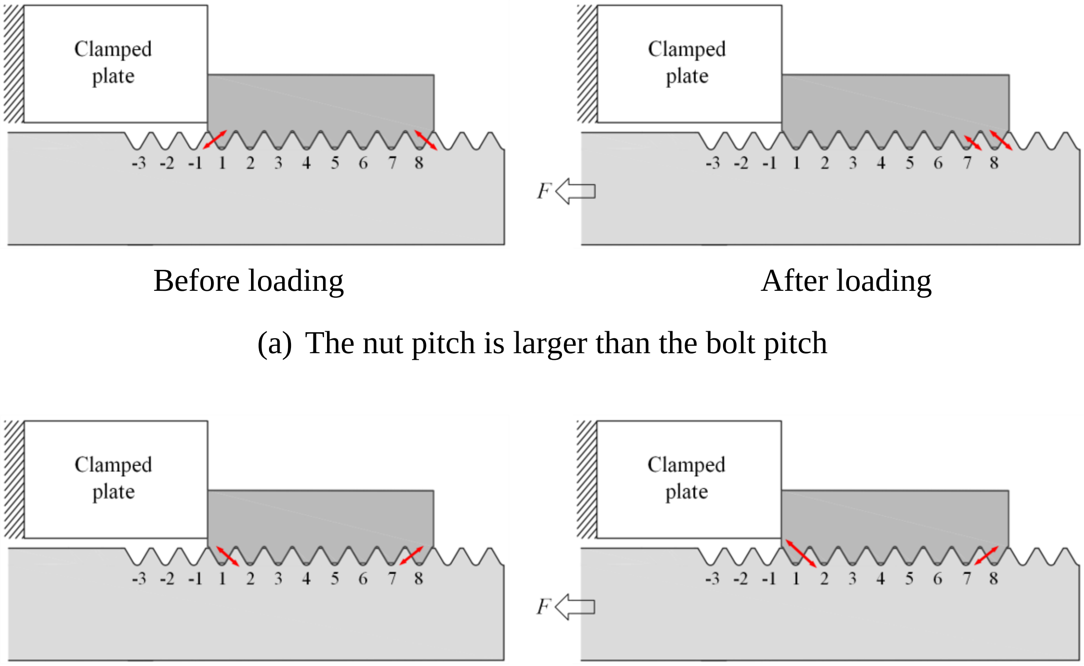
Chapter 3

Effect of pitch difference on the fatigue life improvement

3.1 Overview

As shown in Figure 3.1, if the nut pitch is larger than the bolt pitch, at No.1 thread left side surface contact before the loading is changed to no contact after the loading. However, if the nut pitch is smaller than the bolt pitch, the right side contact surface of No.1 thread before the loading is not changed and the contact force just becomes larger than the contact force of normal bolt-nut connection after the loading. Therefore, the largest stress concentration at No.1 thread can be reduced only by the larger nut pitch.

In this study, a slight pitch difference α is introduced between the bolt and nut. Herein, we consider that the nut pitch is a few microns larger than the bolt pitch. Table 3.1 summarizes the pitch differences reported in this whole text, and the same tables are inserted in Chapter 4 and Chapter 5 for the convenience of readers. In this chapter, in order to study the fatigue failure and fracture in bolt-nut connection, fatigue experiments are conducted for specimens having three types of pitch differences α, i.e. α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$, where α=0 represents the standard bolt-nut connections and they have a relationship of $\alpha=0<\alpha_{verysmall}<\alpha_{small}$. The fatigue life is discussed focusing on the fracture positions of those specimens. To clarify the effect of pitch difference, the axisymmetric model is created by Finite Element Method (FEM) to analyze the contact status and the stresses in threads. The effect of a fitting clearance is also discussed considering the contact status between the real bolt-nut connection threads.



Before loading After loading

(a) The nut pitch is larger than the bolt pitch

Before loading After loading

(b) The nut pitch is smaller than the bolt pitch

Figure 3.1: Contact status between bolt and nut before and after the loading (↘ contact)

Table 3.1: Pitch difference α in each Chapter

		Pitch difference α									
		$0 < \alpha1 < \alpha2 < \alpha3 < \alpha4 < \alpha5 < \alpha6 < \alpha7 < \alpha8 < \alpha9$									
Chapter 3	Fatigue experiment	0	$\alpha_{verysmall}$	α_{small}							
	FE analysis	0	$\alpha_{verysmall}$	α_{small}		$\alpha4$					
Chapter 4	Loosening experiment	0	$\alpha1$ (=$\alpha_{verysmall}$)	$\alpha2$ (=α_{small})	$\alpha3$	$\alpha4$	$\alpha5$	$\alpha6$ (=α_{middle})	$\alpha7$	$\alpha8$ (=α_{large})	$\alpha9$
	FE analysis							α_{middle}			
Chapter 5	Fatigue experiment	0		α_{small}				α_{middle}			
	FE analysis	0		α_{small}				α_{middle}			

(The real values of the pitch difference cannot be open because of the patent application.)

3.2 Experimental set-up

The Japanese Industrial Standard (JIS) M16 bolt-nut connections with strength grade 8.8 are employed. The bolt material is chromium-molybdenum steel SCM435, and the nut material is medium carbon steel S45C quenched and tempered, whose properties are indicated in Table 3.2, and whose stress-strain curves are shown in Figure 3.3. Table 3.3 shows the mechanical properties of bolt with strength grade 8.8 (Table 3.2 and Figure 3.3 show the mechanical properties bolt material, which should be different from the mechanical properties of bolt, e.g., the yield strength of bolt material SCM435 is larger than the minimum yield strength of bolt because of the stress concentration at bolt thread.).

Table 3.2: Mechanical property of bolt and nut materials

	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

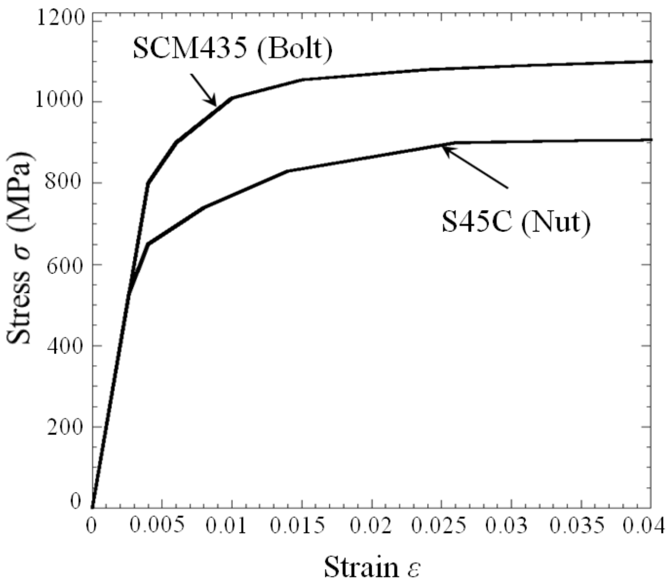


Figure 3.2: Stress strain relation for SCM435 (Bolt) and S45C (Nut)

Table 3.3 Mechanical properties of bolt

	Strength grade	Min. Yield strength (MPa)	Min. Tensile strength (MPa)
Bolt	8.8	>660	>830

Figure 3.3 shows the dimensions of bolt-nut specimen used in fatigue experiment. Figure 3.4 shows the schematic diagram of bolt-nut connection. In the experimental specimen, No.-13 is the starting thread, and in the analytical model, No.-3 bolt thread is the starting thread. In this study, the thread number of Figure 3.4 (b) will be used. From the reference [42], it is known that the stress status at threads No.-2 to No. -12 in Figure 3.4 (a) are same with the case of tension of bolt alone, and the number of thread of this part has almost no effect on the stress status. Therefore, a simplified analytical model in Figure 3.4 (b) is used in this study. Figure 3.5 shows the detailed dimension of bolt and nut threads. The standard M16 bolt-nut connection has the same pitch dimension of 2000 μm, here, the nut pitch is assumed to be equal or slightly larger than the bolt pitch. The clearance between bolt and nut is assumed as a standard dimension, i.e. 125 μm. Three types of pitch differences, namely α=0, α=$\alpha_{verysmall}$ and α=α_{small}, are considered in this study.

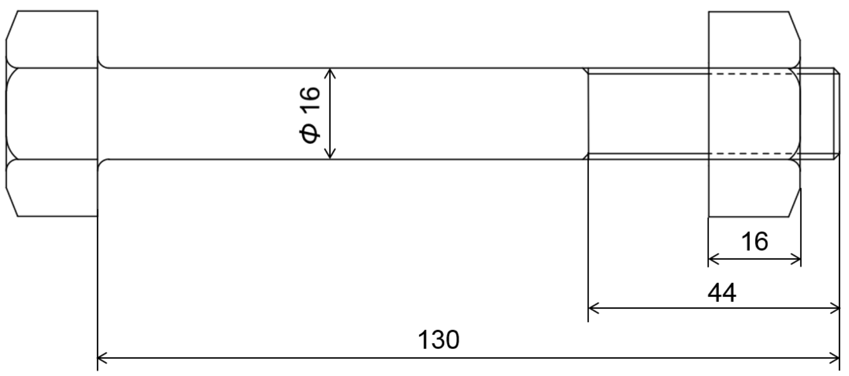
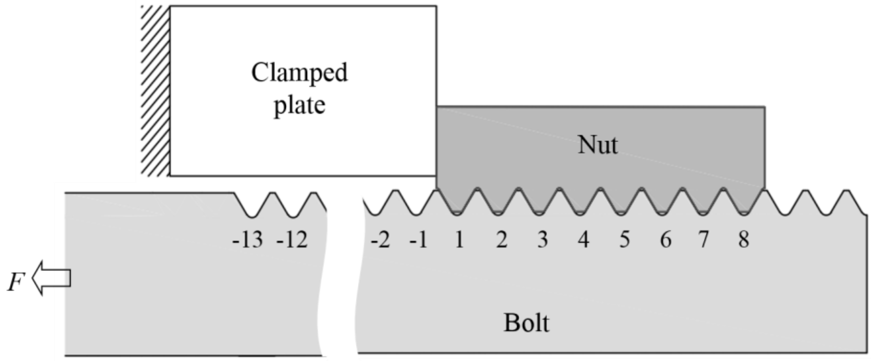
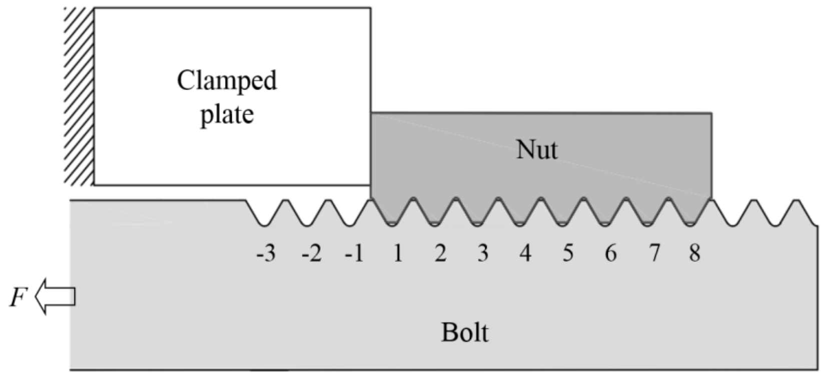


Figure 3.3: Bolt-nut specimen in fatigue experiment (dimensions in mm)



(a) Experimental specimen



(b) Analytical model (No.-3 is used as the starting thread)

Figure 3.4: Schematic diagram of experimental specimen and analytical model

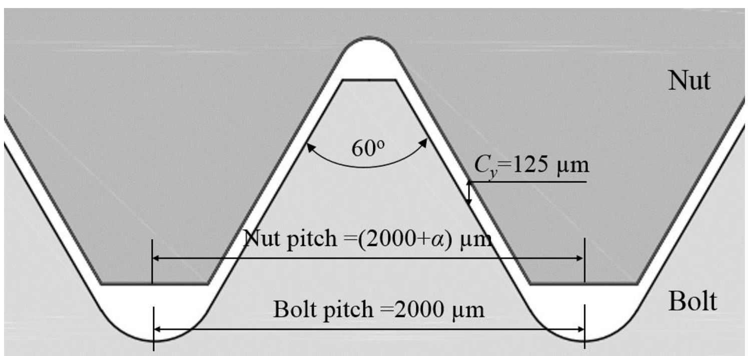
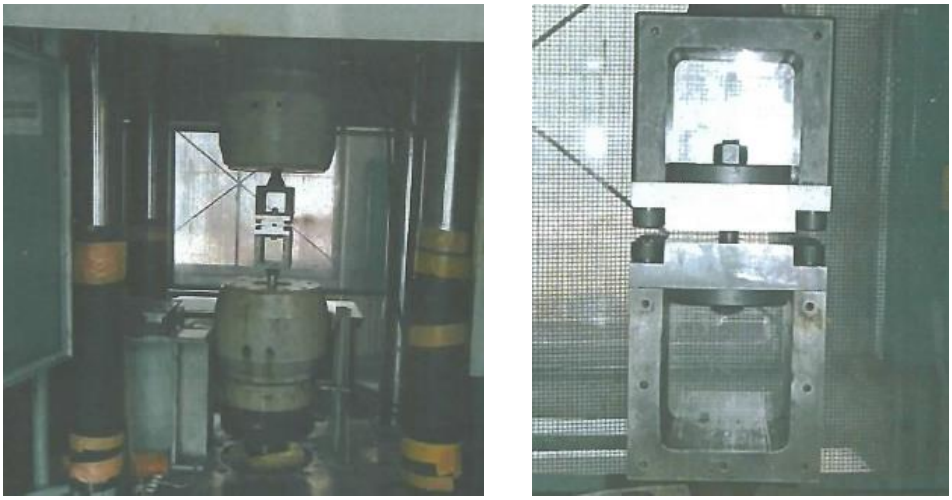


Figure 3.5: Pitch difference and clearance

The 60 ton Servo Fatigue Testing Machine with cycling frequency of 9 Hz is used in this experiment. From the reference [43], it is known that the metal fatigue strength is not affected by the frequency 5 Hz-80 Hz. The fatigue experimental device assembly drawing are shown in Figure 3.6. In the first place, the fatigue experiment is performed for the specimen of $\alpha=\alpha_{small}$, which is subjected to an axial force of F=30±14.1 kN. Since the cross sectional area of the bolt A_R=141 mm^2, the corresponding stress amplitude is 100 MPa. After repeated 1.94×10^5 stress cycles, fracture did not happen, therefore, the applied load was changed to F=30±18.3 kN, for which the corresponding stress amplitude increased to 130 MPa.

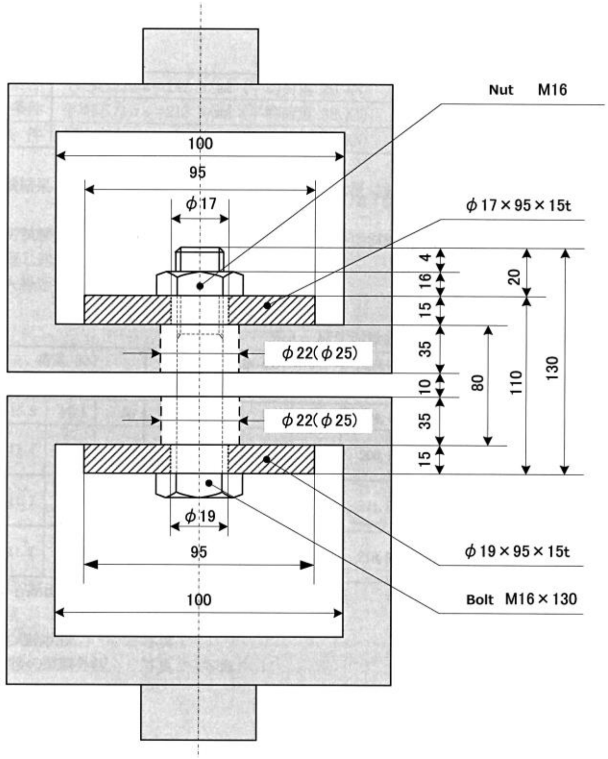
Under this loading, the fatigue experiment continues for another 2×10^5 cycles, at which fracture occurred. In the case of α=0, under the load of F=30±18.3 kN fracture happened at 2.19×10^5 cycles, and for $\alpha=\alpha_{verysmall}$, fracture happened at 2.71×10^5 cycles under the same loading conditions.

Figure 3.7 shows the fracture positions of the three different specimens. For the standard bolt-nut connection (α=0), the fracture happens at the first bolt thread. However, for $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$, fracture happens at No.-3 thread.



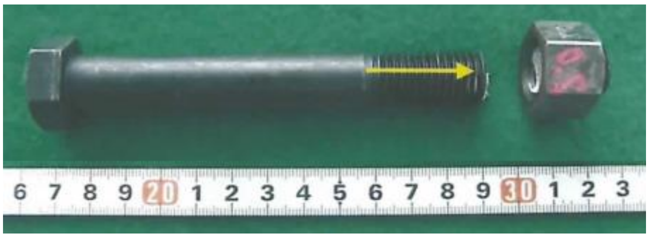
(a) Experimental device

Figure 3.6: Fatigue experimental device (Cont.)

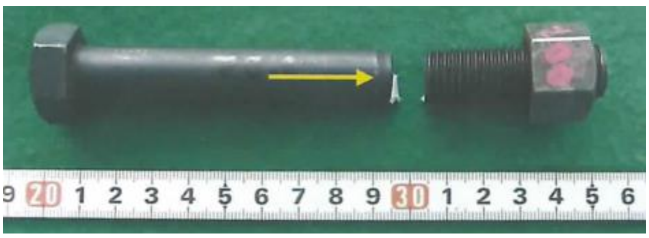


(b) Illustration diagram of experimental device (dimensions in mm)

Figure 3.6: Fatigue experimental device

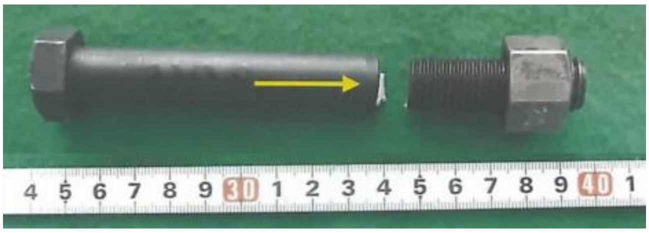


(a) α=0 Position of fracture: No.1 thread



(b) $\alpha=\alpha_{verysmall}$ Position of fracture: No.-13 thread (No.-3 in analytical model)

Figure 3.7: Fractured specimens (Cont.)



(c) $\alpha=\alpha_{small}$ Position of fracture: No.-13 thread (No.-3 in analytical model)

Figure 3.7: Fractured specimens

Utilizing a similar fatigue experimental result [2], the slope of *S*-*N* curves for $\alpha=0$, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$ are depicted in Figure 3.8. Then, Miner's rule is applied to calculate the equivalent fatigue life of $\alpha=\alpha_{small}$ under the load of F=30±18.3 kN, and the result is shown in Table 3.4. It can be seen that the fatigue lives of $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$ are longer than that of $\alpha=0$.

Table 3.4: Results of fatigue experiment

Specimens	$\alpha=0$	$\alpha=\alpha_{verysmall}$	$\alpha=\alpha_{small}$
Axial force F (kN)	30±18.3		
Stress σ (MPa)	213±130		
Number of cycles until fracture happen at No.1-8 threads	2.19×10^5	$>2.71\times10^5$	$>2.49\times10^5$*
Position of fracture	No.1 thread	No.-3 thread	No.-3 thread

*Until the number of cycles=1.94×10^5, F=30±14.1kN

It is known that the reduction of the stress concentration at No.-3 thread is achievable to avoid the fracture at this position, thus, for $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$, a longer fatigue life at which fracture happens between No.1 thread and No.8 thread is expected.

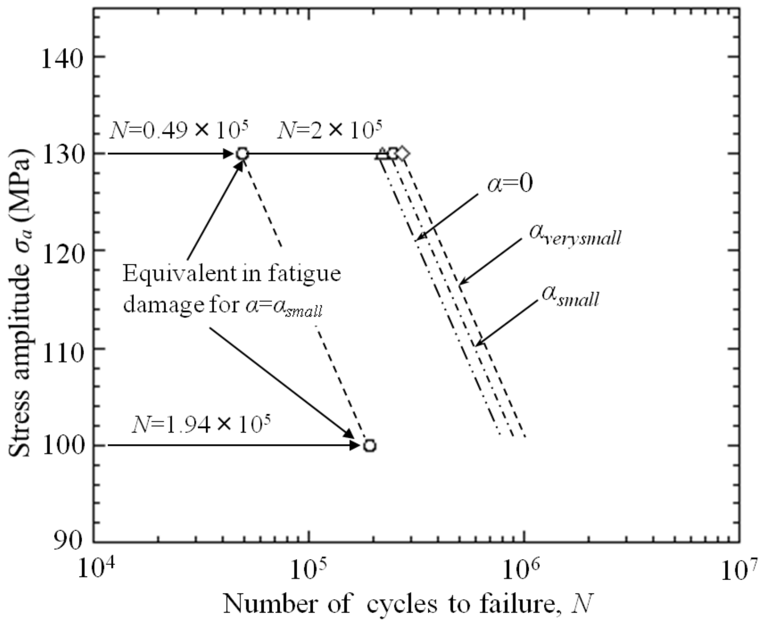


Figure 3.8: *S-N* curve

3.3 Finite element analysis

Figure 3.9 shows the axisymmetric model of the bolt-nut connection created by using FEM code MSC.Marc/Mentat 2012. The fixed component is a cylindrical clamped plate with an inner diameter of 17.5 mm, outer diameter of 50 mm and thickness of 35 mm. The Young's modulus is 206 GPa and the Poisson's ratio is 0.3 for all the materials of bolt, nut and clamped plate. The bolt, nut and clamped body are modeled as three contact bodies. Friction coefficient of 0.3 with Coulomb friction is used for the analysis. The clamped body is fixed in the horizontal direction, and load F is applied on the bolt head. A fine mesh is created at the root of bolt thread with the size of 0.01 mm×0.01 mm, and 4-noded, axisymmetric solid, full integration element is used. The number of elements for bolt, nut and clamped body are 18250, 8160 and 92 respectively. At the first place, in order to investigate the effect of friction, the stress concentration factor is calculated for α=0 by setting three different coefficient of friction, i.e. μ=0, 0.15 and 0.3, under an axial force of 30 kN. It is found that the friction effect is very small. For α_{small}, the maximum relative difference of K_t is less than 10% when μ=0 and μ=0.3 are considered. In this study, therefore, the coefficient of friction is put equal to μ=0.3. As the first step, elastic analysis is performed.

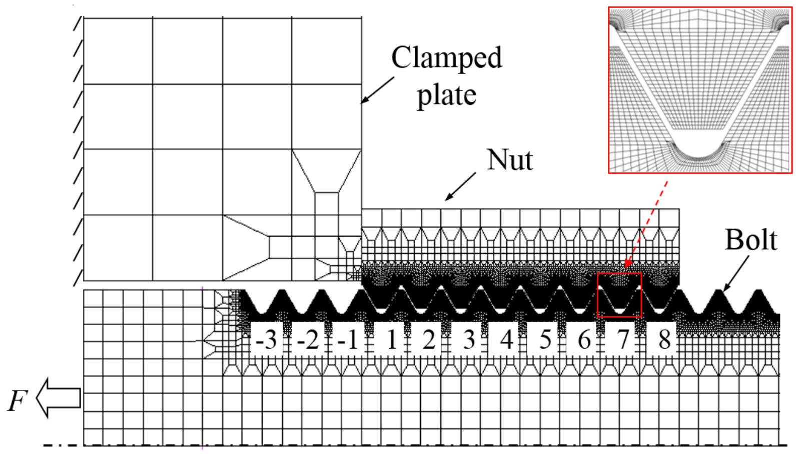


Figure 3.9: Axi-symmetric finite element model

3.3.1 Stress concentration factor

The stress concentration at the root of bolt thread is evaluated by using the stress concentration factor K_t defined as the following Equation,

$$K_t = \frac{\sigma_{t\max}}{\sigma_n}, \sigma_n = \frac{F}{A} \tag{3.1}$$

where σ_{tmax} is the maximum tangential stress appearing at each bolt root, and σ_n is equal to the total bolt axial force F divided by the cross section A as shown in Figure 3.10.

The K_t of each bolt root is indicated in Figure 3.11 under the minimum load F_{min}=30-18.3=11.7 kN, average load $F_{average}$=30 kN and maximum load F_{max}=30+18.3=48.3 kN.

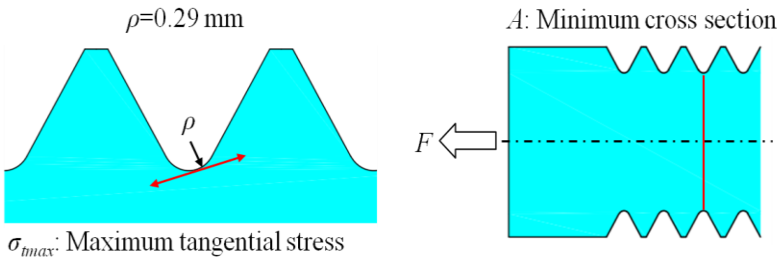
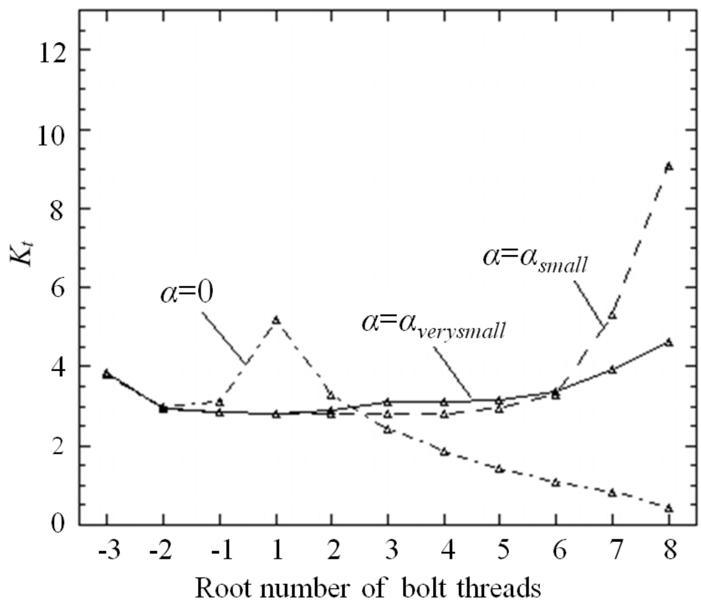


Figure 3.10: Definition of K_t

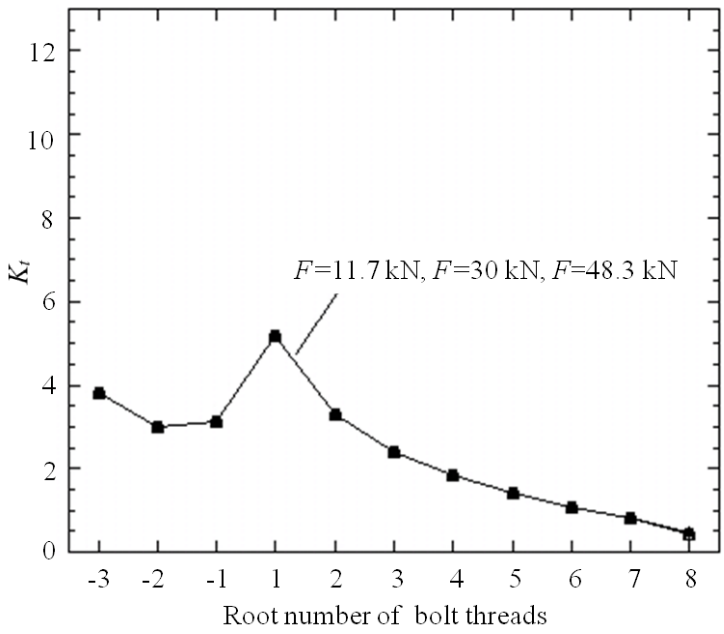
Figure 3.11 (a) shows the comparison of the stress concentration factors K_t for α=0, $\alpha_{verysmall}$ and α_{small} under the same load of F=30 kN. It is found that when $\alpha=\alpha_{verysmall}$ is introduced, the stress concentration at No.1 thread reduces significantly. However, the stress concentration at No.7 thread and No.8 thread increases largely when $\alpha=\alpha_{small}$.

Figure 3.11 (b)-(d) show the stress concentration factors of each bolt root under different loads. For the standard bolt-nut connection, with increasing the load the stress concentration factor, K_t, at each root does not change. In the case of $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$, however, with increasing the load the stress concentration at No.8 thread decreases sharply.

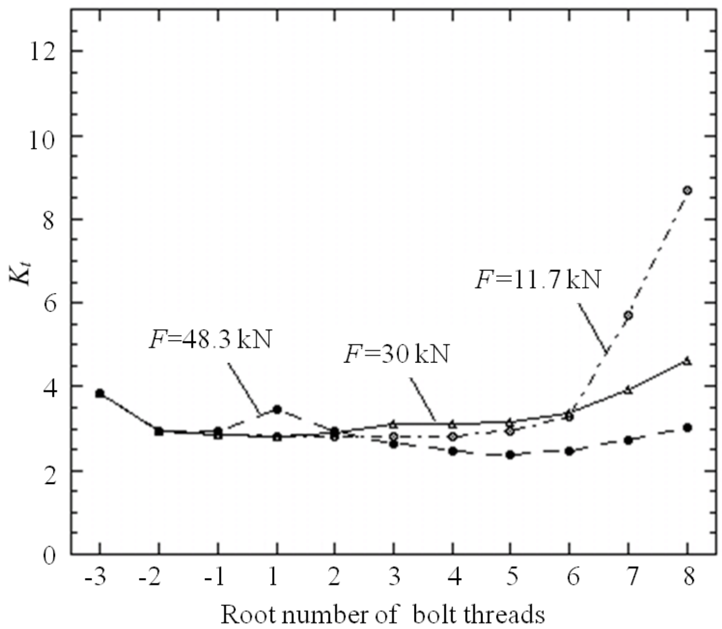


(a) α=0, $\alpha_{verysmall}$ and α_{small} under F=30 kN

Figure 3.11: Stress concentration factor K_t at the root of bolt thread (Cont.)

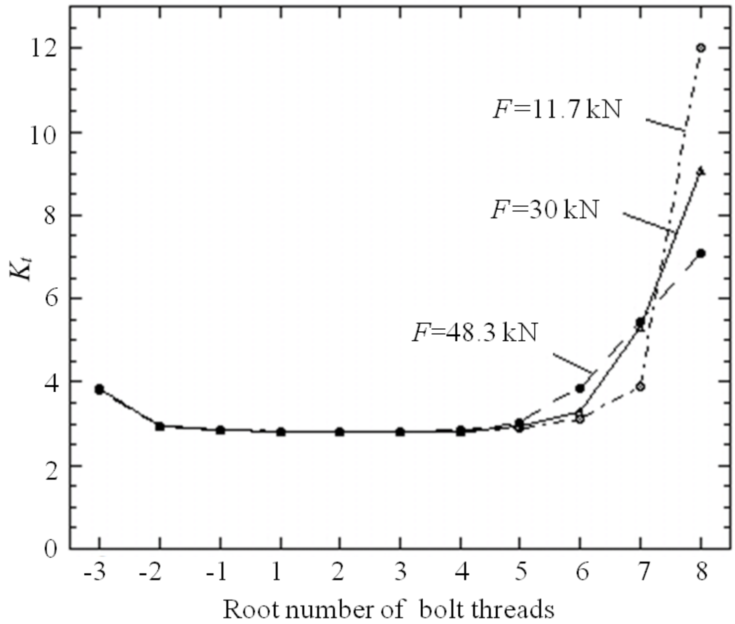


(b) α=0



(c) $\alpha=\alpha_{verysmall}$

Figure 3.11: Stress concentration factor K_t at the root of bolt thread (Cont.)



(d) $\alpha=\alpha_{small}$

Figure 3.11: Stress concentration factor K_t at the root of bolt thread

It can be imagined that when the pitch difference is introduced the contact status between bolt threads and nut threads varies depending on the applied load. To make this point clear, the contact status of bolt and nut threads will be analyzed in the next section.

3.3.2 Contact status of between bolt and nut threads

The experimental load of F=30±18.3 kN is applied to the models of α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$. Before analyzing the stress state, the effect of pitch difference on the contact status of bolt and nut threads is investigated. Figure 3.12 shows the total number of contact threads between bolt and nut with increasing the load from F_{min}=11.7 kN to F_{max}=48.3 kN. As shown in Figure 3.12, for the standard bolt-nut connection (α=0), all the nut threads are in contact with bolt threads independent of the magnitude of the load.

However, for $\alpha=\alpha_{verysmall}$, only three bolt threads, i.e., No. 6, No. 7 and No. 8, are in contact with nut threads under $F=F_{min}$, although with increasing the load the contact thread number increases. When $F=F_{max}$, the contact status becomes similar to the case of the standard bolt-nut connection. For $\alpha=\alpha_{small}$ under $F=F_{min}$, only No. 7 and No. 8 bolt threads are in contact with nut threads, and even under $F=F_{max}$, only No. 6, No.7 and No. 8 bolt threads are in contact with nut threads.

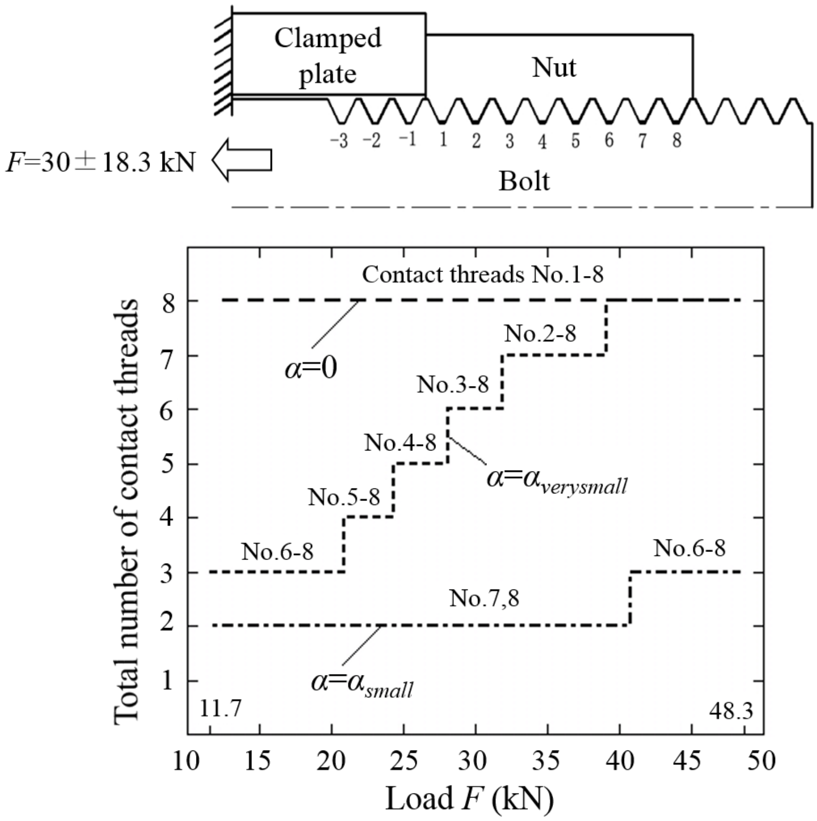


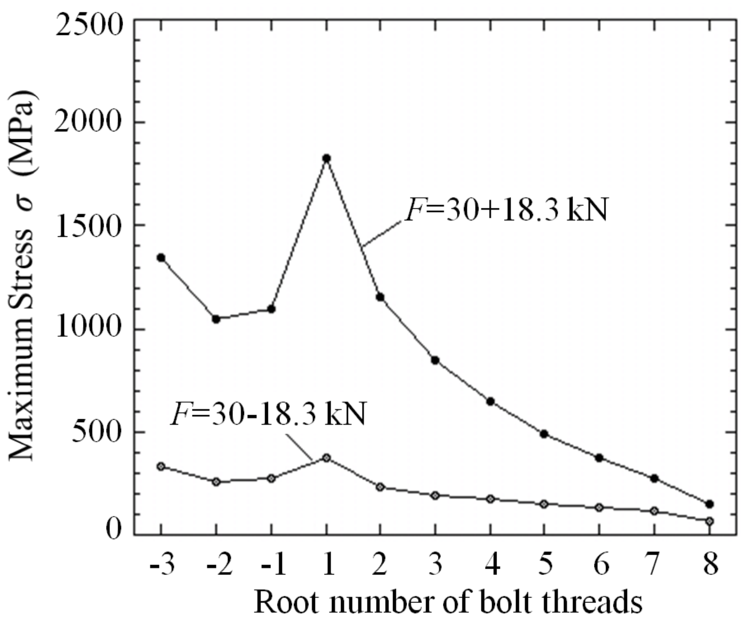
Figure 3.12: Number of contact threads between bolt and nut

3.3.3 Mean stress and stress amplitude at the root of bolt threads

Figure 3.13 shows the maximum stress, σ, at each root of bolt thread under different loads, i.e. F_{min}=30-18.3 kN and F_{max}=30+18.3 kN. The endurance limit diagrams are obtained as shown in Figure 3.14 based on the results of Figure 3.13. Herein, the average stress σ_m and stress amplitude σ_a are defined using the following Equation

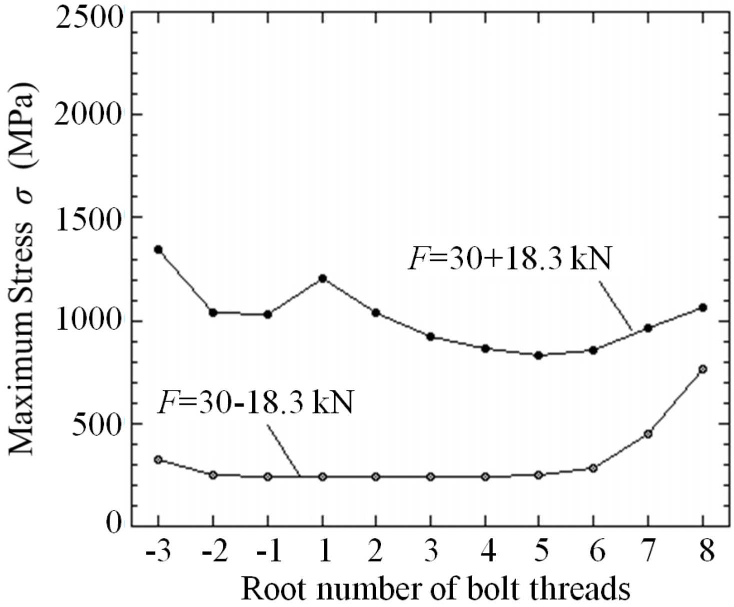
$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}, \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{3.2}$$

where σ_{max} is the maximum stress of each thread under the maximum load F=30+18.3 kN, and σ_{min} is the maximum stress of each thread under the minimum load F=30-18.3 kN. As shown in Figure 3.14, the fatigue limit σ_w of the material SCM435 (JIS) is 420 MPa, and the yield stress σ_s is 800 MPa.

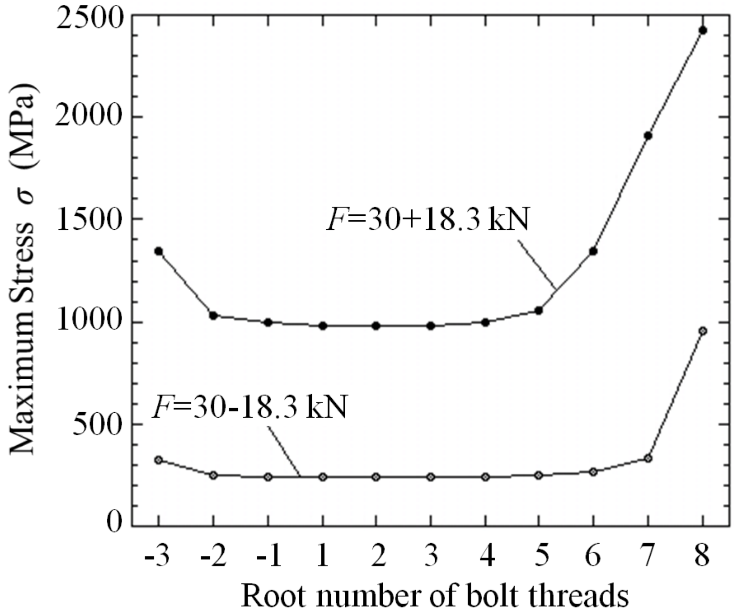


(a) α=0

Figure 3.13: Maximum stress σ at the root of bolt thread under different loads (Cont.)



(b) $\alpha=\alpha_{verysmall}$

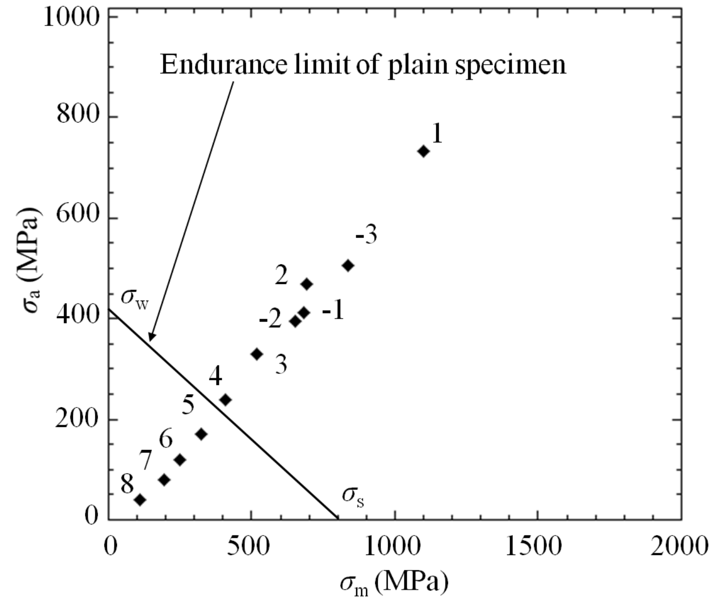


(c) $\alpha=\alpha_{small}$

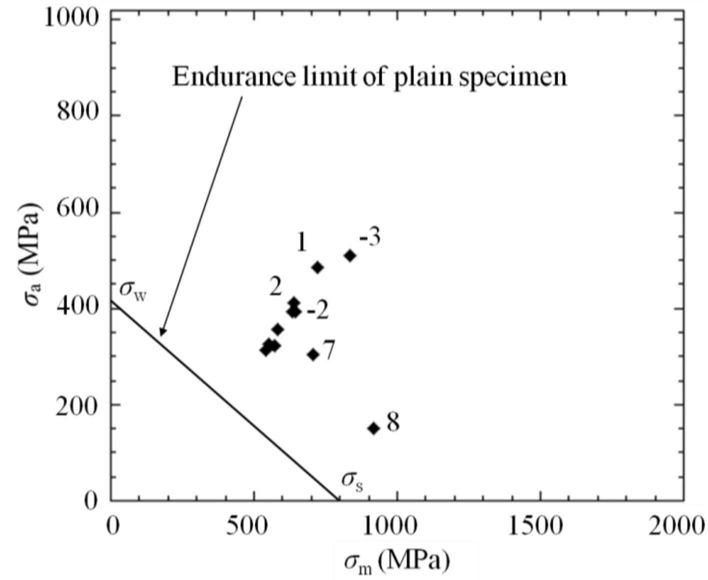
Figure 3.13: Maximum stress σ at the root of bolt thread under different loads

For the standard bolt-nut connection, the bolt thread No. 1 has the maximum stress amplitude as shown in Figure 3.14 (a). On the other hand, for $\alpha=\alpha_{verysmall}$ in Figure 3.14

(b), it is seen that the stress amplitude as well as the mean stress at thread No. 1 decreases significantly. For $\alpha=\alpha_{small}$ in Figure 3.14 (c), large stress appears at threads No. 7 and No. 8 instead of thread No. 1.

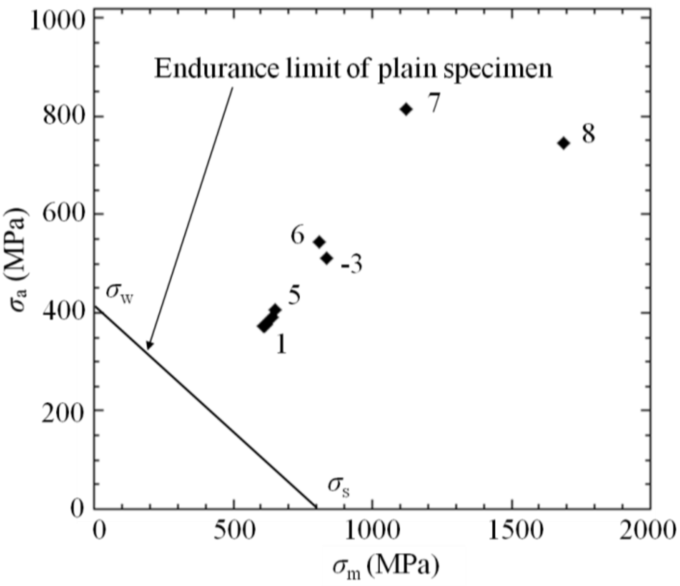


(a) $\alpha=0$



(b) $\alpha=\alpha_{verysmall}$

Figure 3.14: Endurance limit diagrams for $\alpha=0$, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$ (Cont.)

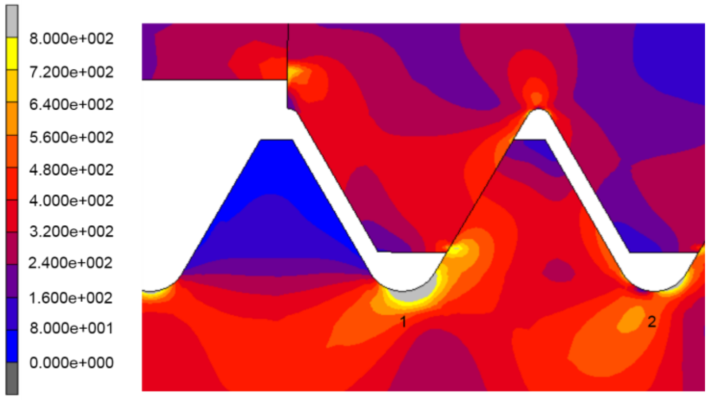


(c) $\alpha=\alpha_{small}$

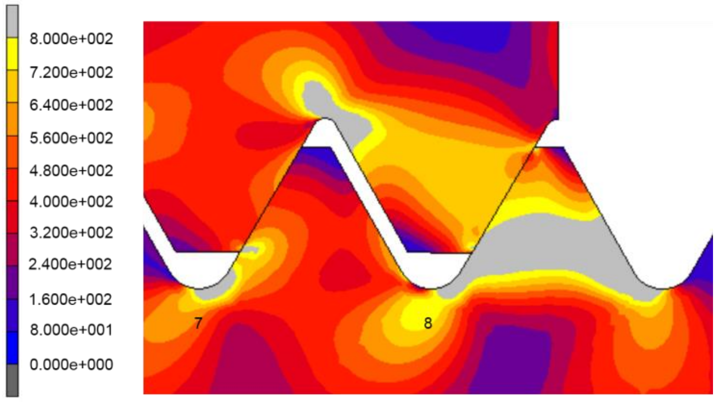
Figure 3.14: Endurance limit diagrams for α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$

Since the results of elastic analysis show that the maximum stress is far over the yield stress 800 MPa of the bolt material SCM435 (JIS), the elastic-plastic analysis is also performed under the same load of F=30±18.3 kN. Here, the same material of SCM435 is considered for bolt, nut and clamped body in the elastic-plastic analysis. The changes of stress status at bolt threads from elastic analysis to elastic-plastic analysis are investigated, although the analysis with considering different materials should has higher accuracy. In Chapter 4 and Chapter 5, different materials (SCM435 for bolt and clamped body, S45C for nut) are considered in the analysis in order to match the experiments.

Figure 3.15 indicates the equivalent stress at bolt threads where the high stress appears for α=0 and $\alpha=\alpha_{small}$. For α=0, the plastic strain zone only occurs at the root of the No.1 bolt thread. On the other hand, for $\alpha=\alpha_{small}$, the plastic strain appears at the root of No. 7 thread and the wide region of No. 8 thread.



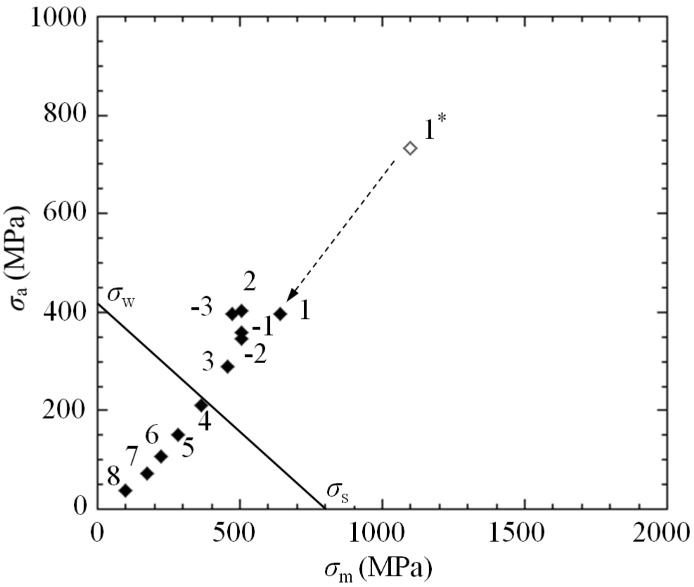
(a) α=0



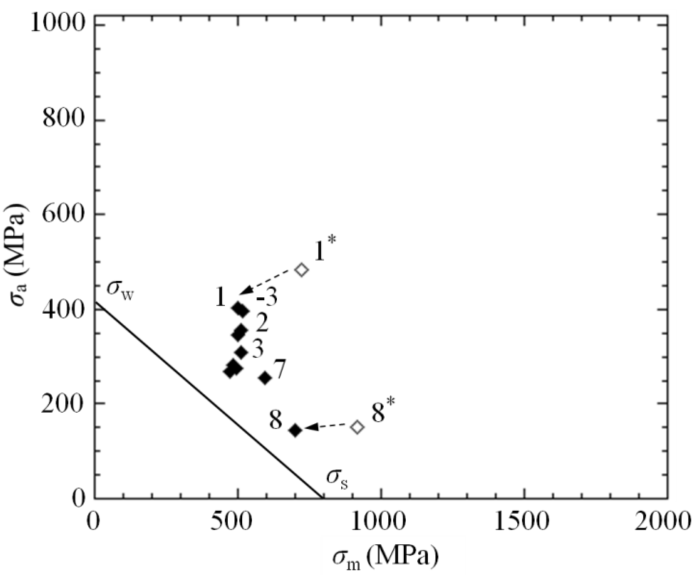
(b) $\alpha=\alpha_{small}$

Figure 3.15: The equivalent stress in MPa under F=30+18.3 kN

Figure 3.16 presents the endurance limit diagrams based on the elastic-plastic analysis considering the von-Mises stress at each bolt thread. For α=0, the stress decreases significantly at thread No. 1 compared with the elastic analysis result. Similarly, for $\alpha=\alpha_{small}$, the stress at No. 7 and No. 8 threads decrease significantly. It should be noted that compared with α=0, the severity at each thread are almost the same for $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$.

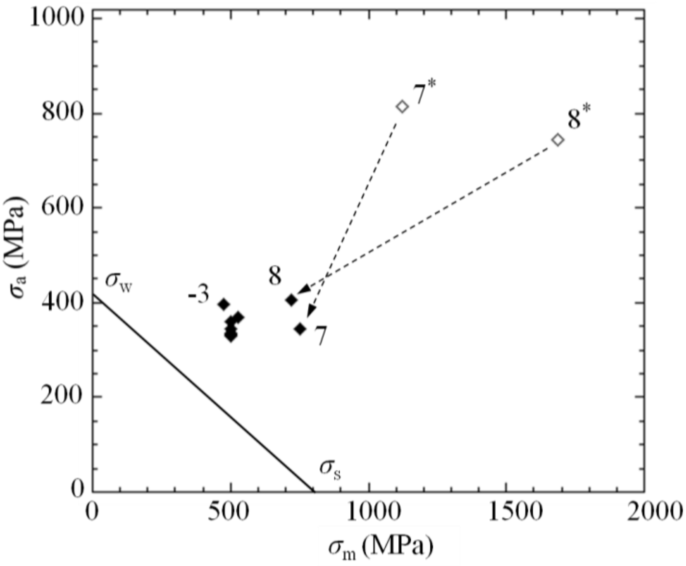


(a) α=0



(b) $\alpha=\alpha_{verysmall}$

Figure 3.16: Endurance limit diagrams based on elastic-plastic analysis (∗: data in elastic analysis) (Cont.)



(c) α=α_{small}

Figure 3.16: Endurance limit diagrams based on elastic-plastic analysis (*: data in elastic analysis)

3.4 Effect of the clearance on the stress state at bolt threads

For α=0 and α=$\alpha_{verysmall}$, the most dangerous root of bolt thread appeared in the FEM results agrees well with the experimental failure position of the bolt. However, for α=α_{small} such an agreement between FEM and the experimental results has not been obtained. As a further research, the effect of the clearance between bolt and nut on the stress state of bolt threads is investigated.

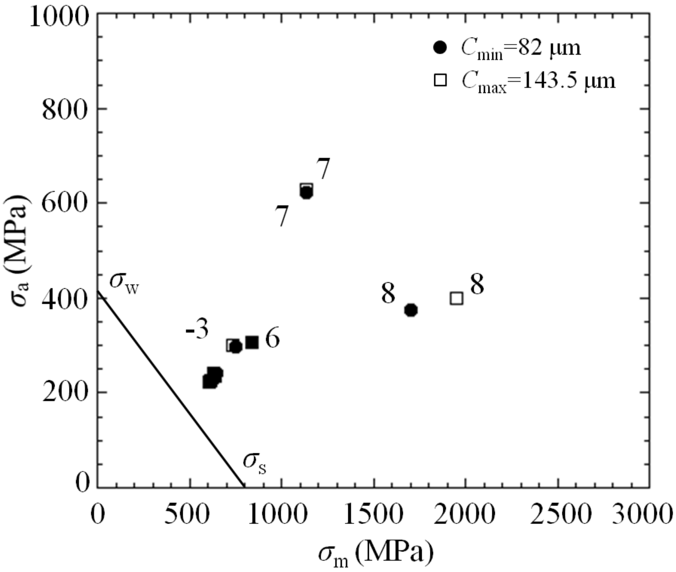
In the above analysis, the clearance between the bolt and nut is assumed as a standard value, i.e. 125 μm. The maximum clearance C_{max} and the minimum clearance C_{min} can be defined by Equation (3.3) based on JIS:

$$C_{max} = \frac{1}{2}(D_{max}^{nut} - d_{min}^{bolt}), \quad C_{min} = \frac{1}{2}(D_{min}^{nut} - d_{max}^{bolt}) \tag{3.3}$$

where D_{max}^{nut} and D_{min}^{nut} denote the maximum and minimum effective diameter of nut, respectively and d_{max}^{bolt} and d_{min}^{bolt} denote the maximum and minimum effective diameter of bolt, respectively. From Equation (3.3), for the M16 bolt-nut connection, the clearance ranges from 19 μm to 205 μm. However, the actual clearance can be determined by multiplying the maximum clearance by a factor ranged from 0.4 to 0.7. Thus, for M16 bolt-nut connections, the actual minimum and maximum clearance are C_{min}=205 μm×0.4=82 μm and C_{max}=205 μm×0.7=143.5 μm, respectively.

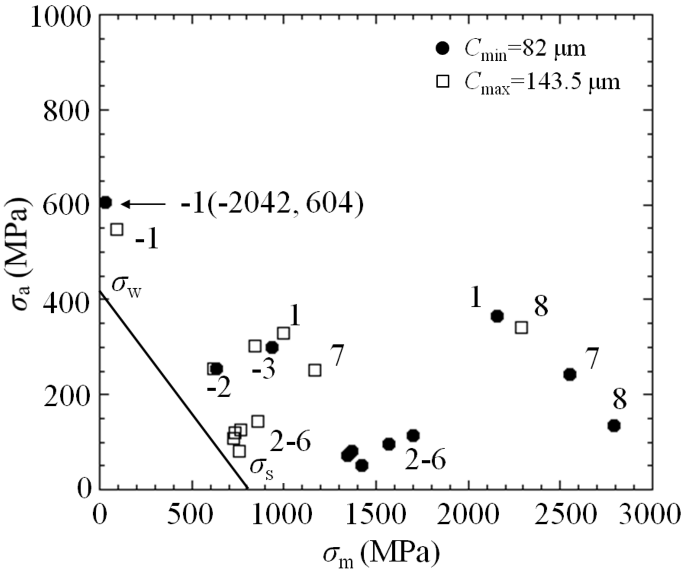
For $\alpha=\alpha_{small}$ and another larger pitch difference $\alpha=\alpha_4$, the elastic analysis is performed considering C_{min}=82 μm and C_{max}=143.5 μm. Here, the relatively easy way elastic analysis is considered to investigate the trends of stress status at bolt threads with different clearances, also the elastic-plastic simulation should has higher accuracy. The load condition is F=30±11 kN. Figure 3.17 shows the endurance limit diagrams for $\alpha=\alpha_{small}$ and $\alpha=\alpha_4$ considering the minimum and maximum clearances.

In Figure 3.17 (a), when the clearance is changed from C_{min} to C_{max} for $\alpha=\alpha_{small}$, the stress status at No.8 thread changes slightly. In Figure 3.17 (b), for $\alpha=\alpha_4$, with increasing the clearance from C_{min} to C_{max}, the average stress decreases at No. 1 and No.7 threads, and the stress amplitude at No.8 thread increases slightly.



(a) $\alpha=\alpha_{small}$

Figure 3.17: Endurance limit diagrams considering different clearance for $\alpha=\alpha_{small}$ and $\alpha=\alpha_4$ (Cont.)



(b) α=α_4

Figure 3.17: Endurance limit diagrams considering different clearance for α=α_{small} and α=α_4

Figure 3.18 and Figure 3.19 show the contact status between bolt and nut for α=α_{small} and α=α_4 considering the minimum and maximum clearance. The contact threads are marked by red arrows.

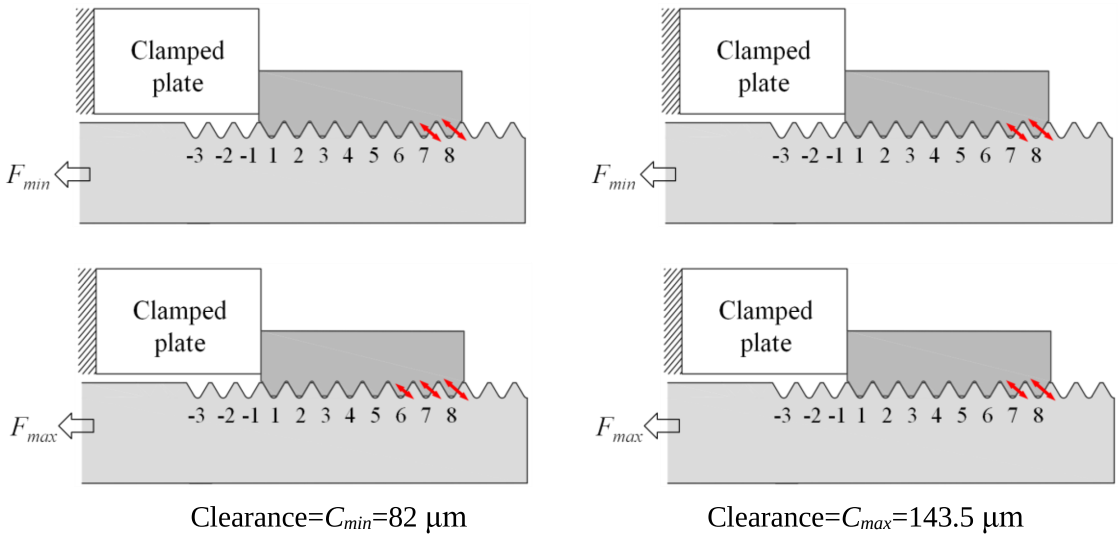


Figure 3.18: Contact status for α=α_{small}

For $\alpha=\alpha_{small}$, with increasing the clearance from $C_{\min}$ to $C_{\max}$, the contact status between bolt and nut show nearly no difference under the same load.

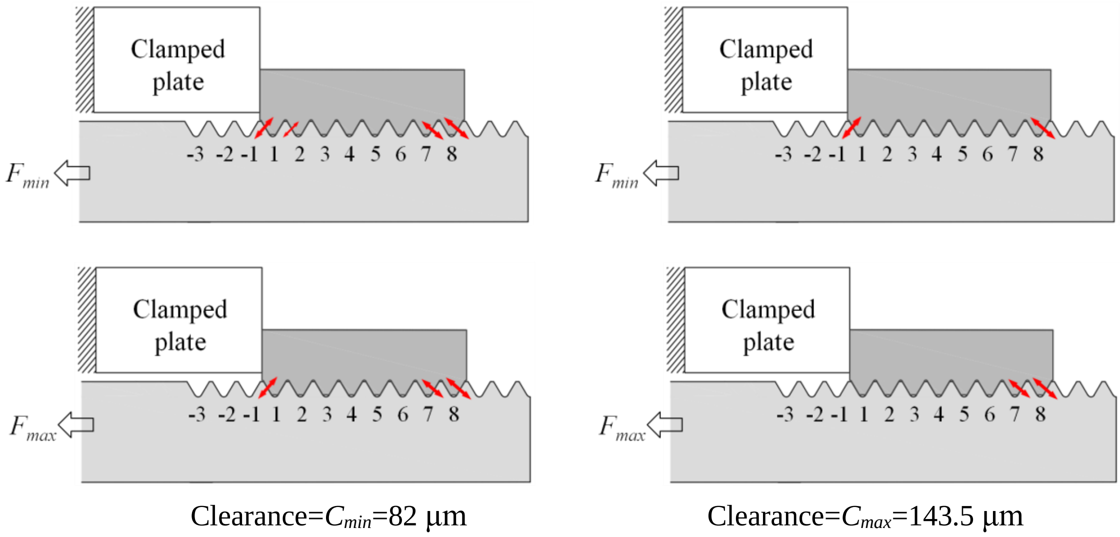


Figure 3.19: Contact status for $\alpha=\alpha_4$

For $\alpha=\alpha_4$, it can be seen that the contact status are quite different when clearance changed from $C_{\min}$ to $C_{\max}$ especially under the load F_{max}. From the comparison between $\alpha=\alpha_{small}$ and $\alpha=\alpha_4$, it can be found that the clearance does not affect very much for $\alpha=\alpha_{small}$ but affect largely for $\alpha=\alpha_4$ ($\alpha_{small}<\alpha_4$).

3.5 Conclusions

In this chapter, fatigue failure and fracture of bolt-nut connection having a slight pitch difference have been analyzed using experimental techniques as well as FEM. The fatigue experiment was conducted for three specimens with different types of pitch differences. According to the FEM results, the stress states and the contact status at each root of bolt threads was presented and discussed. The conclusions can be summarized as follows:

(1) For the standard bolt-nut connection ($\alpha=0$), the fatigue fracture happens at No.1

thread, while it happens at No.-3 thread for $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$. Since the stress concentration can be reduced at No.-3 thread to avoid the fracture at this position, it is found that the fatigue life of bolt can be extended by introducing a suitable pitch difference.

(2) The FE analysis shows that both the average stress and stress amplitude at root No.1 of bolt threads can be reduced by introducing a suitable pitch difference. For $\alpha=\alpha_{small}$, large stress appears at No. 7 thread and No. 8 instead of No. 1 thread. The FE analysis explains the experimental results.

(3) When the pitch difference is small, usually only No. 7 and No. 8 bolt threads contact with nut threads even the clearance changes. On the other hand, when the pitch difference is large, the contact status of No. 1 bolt thread may change from left side contact to no contact. Therefore, with increasing the pitch difference, the clearance between bolt and nut affects the contact status more significantly.

Chapter 4

Effect of pitch difference on the anti-loosening performance

4.1 Overview

The previous chapter clarified that the fatigue life of bolt is improved by introducing suitable pitch difference under a certain level of stress amplitude. 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. Moreover 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 until now without reducing the fatigue strength and without raising the cost.

This Chapter therefore focuses on the effect of pitch difference between bolt-nut connections upon the anti-loosing performance. In the first place, with varying the pitch difference α, the prevailing torque necessary for the nut rotation before the nut touching the clamped body is measured experimentally. Next, the tightening torque is investigated in relation to the bolt axial force after the nut touching the clamped body. Furthermore, the effect of pitch difference on the anti-loosening performance is studied experimentally, and the most desirable pitch difference is discussed considering the clamping ability with the bolt axial force. By applying the finite element analyses on the screwing process, the mechanism of anti-loosening for bolt-nut having pitch difference are discussed. Table 4.1 summarizes the pitch differences reported in this whole text.

Table 4.1: Pitch difference α in each Chapter

		Pitch difference α									
		$0 < \alpha 1 < \alpha 2 < \alpha 3 < \alpha 4 < \alpha 5 < \alpha 6 < \alpha 7 < \alpha 8 < \alpha 9$									
Chapter 3	Fatigue experiment	0	$\alpha_{verysmall}$	α_{small}							
	FE analysis	0	$\alpha_{verysmall}$	α_{small}		$\alpha 4$					
Chapter 4	Loosening experiment	0	$\alpha 1$ (=$\alpha_{verysmall}$)	$\alpha 2$ (=α_{small})	$\alpha 3$	$\alpha 4$	$\alpha 5$	$\alpha 6$ (=α_{middle})	$\alpha 7$	$\alpha 8$ (=α_{large})	$\alpha 9$
	FE analysis							α_{middle}			
Chapter 5	Fatigue experiment	0		α_{small}				α_{middle}			
	FE analysis	0		α_{small}				α_{middle}			

(The real values of the pitch difference cannot be open because of the patent application.)

4.2 Effect of the pitch difference α on the nut rotation

4.2.1 Experimental set-up

Japanese Industrial Standard (JIS) M16 bolt-nut connections are employed to study the slight pitch difference. Figure 4.1 shows the dimensions of bolt-nut specimen used in the loosening experiment. Figure 4.2 shows the schematic illustration of bolt-nut connection having pitch difference.

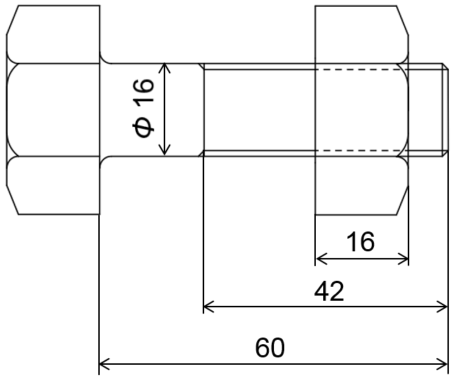
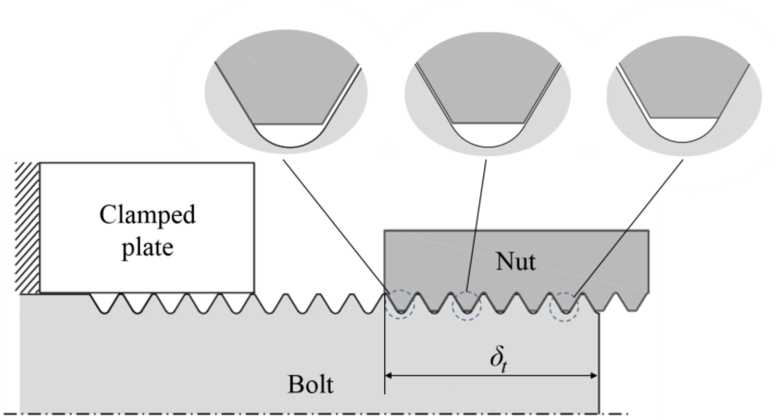
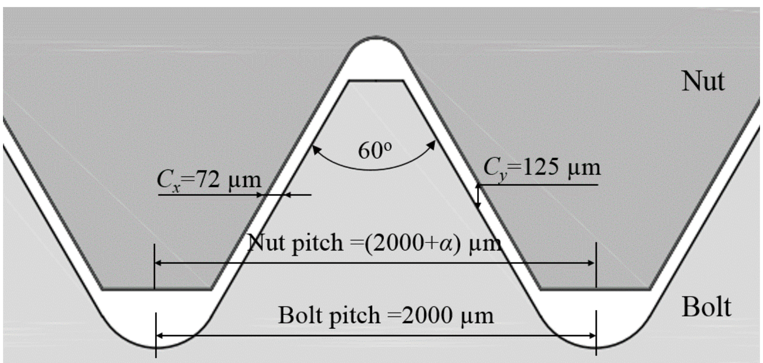


Figure 4.1: Bolt-nut specimen in loosening experiment (dimensions in mm)



(a) Contact status between bolt and nut when the nut pitch is slightly larger than the bolt pitch (δ_t: The distance where the prevailing torque appears)



(b) Pitch difference and clearance between bolt and nut

Figure 4.2: Schematic illustration of bolt-nut connection having pitch difference

Figure 4.2 (a) also shows the contact status between bolt and nut threads during the screwing process. As the nut is screwed onto the bolt, the pitch difference α is accumulated. Then finally, both the nut threads number n=1 and n_c=6 become contact with the bolt threads as shown in Figure 4.2 (a). The distance δ_t where the contact appears can be obtained geometrically as shown in Equation (4.1) and Equation (4.2). The nut rotation does not need torque before the distance δ_t but does need torque after δ_t,

$$n_c \alpha = 2C_x, C_x = \frac{C_y}{\tan\theta'} \tag{4.1}$$

$$\delta_t = n_c p \tag{4.2}$$

where p is the pitch of bolt (2 mm), α is the pitch difference, n_c is the contacted threads number of nut except for n=1, θ is the thread angle (=60°), $\theta' = (\pi - \theta)/2$, C_x and C_y are the clearance between bolt and nut. The specimens in this study have five different levels of pitch difference α, which have a relationship of $\alpha=0<\alpha_{small}<\alpha_{middle}<\alpha_{large}<\alpha_{verylarge}$, where α=0 represents the normal bolt-nut connections. Here, it should be noted that the nut has 8 threads and therefore Equation (4.1) is valid when n_c is less than 8 threads. Table 2 shows the distance δ_t where the thread contact appears and the nut thread number n_c obtained from Equation (4.1) and Equation (4.2). The distance δ_t can be predicted for α_{middle}, $\alpha=\alpha_{large}$ and $\alpha=\alpha_{verylarge}$, although no thread contact may be expected for $\alpha=\alpha_{small}$ because the required contacted threads number n_c is more than the total threads number 8 of the employed nut.

4.2.2 Prevailing torque necessary for the nut rotation before the nut touching the clamped body

After the nut threads contacted over distance δ_t as shown in Figure 4.2 (a), so-called prevailing torque is necessary for the nut rotation even though the nut does not touch the clamped body yet. Table 4.2 indicates prevailing torque T_p experimentally in comparison with contacted length δ_t and contacted nut number n_c obtained from Equations (4.1) and (4.2).

For $\alpha=\alpha_{small}$, the value n_c is larger than 8, and therefore the thread contact does not appear and the prevailing torque is zero experimentally. For $\alpha=\alpha_{middle}$, since value n_c is smaller than 8, the thread contact appears experimentally and prevailing torque is T_p=25 N·m. For $\alpha=\alpha_{large}$ prevailing torque T_p=50 N·m appears, and for $\alpha=\alpha_{verylarge}$ the threads deformed too largely and the nut is fixed during the rotation before touching the clamped body.

Table 4.2: Position where prevailing torque appears δ_t and number of nut threads contacted n_c

Pitch difference α	Theoretically obtained δ_t (mm)	The number of nut threads contacted n_c	Prevailing torque T_p (N·m)
0	-	-	No
α_{small}	19.2	9.6 (>8)	No
α_{middel}	8.8	4.4 (<8)	25
α_{large}	7.4	3.7 (<8)	50
$\alpha_{verylarge}$	5.8	2.9 (<8)	Fixed

4.2.3 Relationship between the prevailing torque and clamping force after the nut touching the clamped body

Since the bolt and nut are used for connecting members, the clamping ability is essential. In this sense, after the nut touching the clamped body, the relationship between the tightening torque and the clamping force is therefore investigated. Note that tightening torque T is different from prevailing torque T_p, which is defined only before the nut touching the clamped body. The tightening torque was controlled by using an electric torque wrench, and the clamping force was measured by using the strain gauge attached to the clamped body surface as shown in Figure 4.3 (a). The uniaxial strain gauge with a length of 2 mm KFG-2 (Kyowa Electronic Instruments Co., Ltd.) was used in this measurement. Before the experiments, calibration tests were performed by compressing the clamped body to obtain the relationship between the clamping force and strain. Similar tests were performed to calibrate the torque wrench as shown in Figure 4.3 (b).

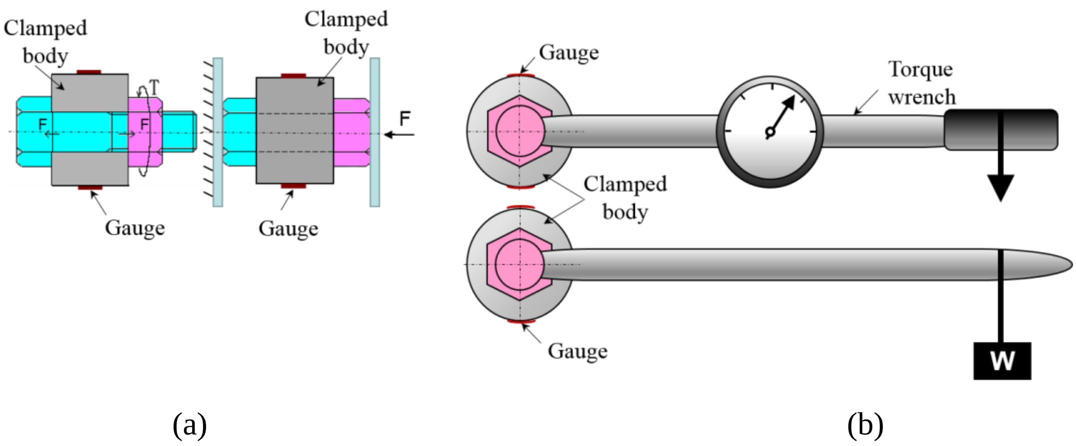


Figure 4.3: (a) Calibration method for bolt axial force measurement and (b) Calibration method for torque wrench

In order to compare anti-loosening performance for different pitch differences, the same tightening torque should be applied. When the tightening torque T=70 N·m is applied to the standard bolt-nut α=0, the bolt-axial force becomes F=24 kN. The value F=24 kN corresponds to the bolt-axial stress 160 MPa, which is 20% of the yield stress 800 MPa of SCM435. The value F=24 kN is smaller compared to the normal bolt-axial force used in many cases. For example, 70% of yield stress is recommended as a standard tightening torque [44]. However, if larger bolt- axial force is used, the effect of α on the anti-loosening performance cannot be clearly discussed. In fact, when T=150 N·m was applied in our preliminary experiment, bolt-nut seizure was sometimes observed even for $\alpha=0$ and $\alpha=\alpha_{small}$. Therefore, in this study, the smaller tightening torque T =70 N·m is used to compare the anti-loosening ability conveniently. In this study, turning is used for manufacturing nuts which leads to the seizure occurring more easily than tapping which is usually used for manufacturing nuts.

Figure 4.4 shows the tightening torque vs. clamping force relationship experimentally obtained. When $\alpha=\alpha_{small}$, the torque-clamping force relationship is equal to the one of α=0. When $\alpha=\alpha_{middle}$, the prevailing torque of 25 N·m is required before the nut touching the clamped plate. Under the same tightening torque T=70 N·m, the clamping force is reduced to F=20 kN. When $\alpha=\alpha_{large}$, under T=70 N·m the axial force decreases significantly to F=8 kN, which is only 1/3 of the axial force of α=0.

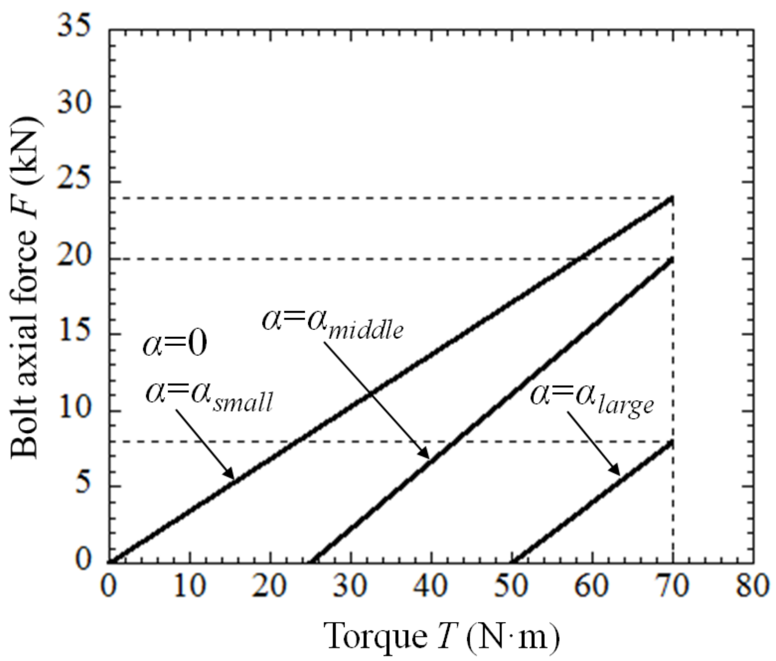


Figure 4.4: Relationship between torque and clamping force

In this study, both the clamping force F and the tightening torque T were directly obtained in the experiments. The effect of pitch difference on the F-T relation was focused on and discussed, keeping the other variables constant for all the specimens, including the same thread surface condition as well as the same clamped body.

It is known that the relation between clamping force and torque is affected significantly by the friction. For the standard bolt-nut connections, the torque T and clamping force F has a relation as follows [44]:

$$T = K \cdot d \cdot F \tag{4.3}$$

$$K = \frac{1}{2d}\left\{ d_2 \left(\frac{\mu}{\cos\theta'} + \tan\beta \right) + \mu_n \cdot d_n \right\} \tag{4.4}$$

Here, K is the torque coefficient; d is the nominal diameter of bolt; d_2 is the effective diameter; d_n is the effective diameter bearing surface; μ is the friction coefficient of threaded portion; μ_n is the friction coefficient of bearing portion; θ' is the half angle of screw thread (JIS screw 30°), and β is the lead angle.

To know the friction condition in this study, by applying Equation (4.3) and (4.4) to the standard bolt-nut connection (α=0), the torque coefficient K and friction coefficient μ can be obtained as shown in Table 4.3. The obtained friction coefficient μ= 0.14 is close to the average value of normal lubrication case 0.15.

Table 4.3 Friction coefficient between bolt nut threads

Pitch difference	Clamping force F (kN)	Tightening torque T (N·m)	Torque coefficient K	Friction coefficient of threaded portion μ
α=0	24	70	0.182	0.14

(d=16 mm, d_2=14.701 mm, d_n=20.56 mm, $\tan\beta$=0.0433, μ=μ_n)

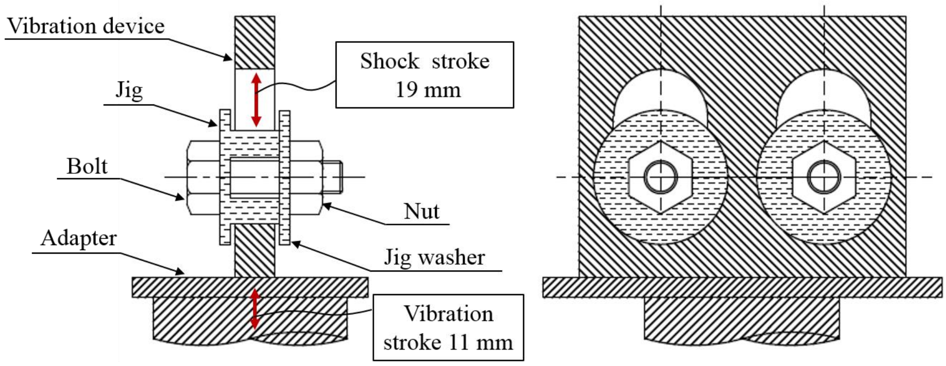
4.3 Loosening experiment

4.3.1 Experimental device to investigate anti-Loosing performance

Based on the torque-axial force relationship obtained above, the loosening experiment is performed to investigate the effect of pitch difference on the anti-loosening performance. For each pitch difference α, two specimens are tested together in the loosening experiment. As shown in Figure 4.5, the experimental device is an impact-vibration testing machine based on NAS3350 (National Aerospace Standard) whose vibration frequency is 1,800 cycles per minute, and vibration acceleration is 20 G. The maximum vibration cycle of NAS3350 is 30,000, therefore, if the vibration cycles are over 30,000, we may judge the anti-loosening performance is enough. A counter connected with the experimental device shows the number of cycles of vibrations. For all the specimens, the nuts are tightened under the same torque of 70 N·m.

(a) Loosening experimental device

Figure 4.5: Loosening experimental device based on NAS3350 (Cont.)



(b) Illustration diagram of vibration device

Figure 4.5: Loosening experimental device based on NAS3350

4.3.2 Experimental results for anti-loosing performance

Table 4.4 shows the vibration number when loosening happens. Table 4.4 also indicates the prevailing torque measured in the loosening experiment and the bolt axial forces estimated from Figure 4.4. For α=0 and $\alpha=\alpha_{small}$, the nuts dropped at about 1,000 vibrations. For $\alpha=\alpha_{middle}$, the nuts did not drop until 30,000 vibrations, but the loosening was observed for one specimen. For $\alpha=\alpha_{large}$, no loosening is observed until 30,000 vibrations although the axial force is estimated only 8 kN. It may be concluded that if α is too small, the anti-loosening performance cannot be expected and if α is too large, the clamping ability is not enough. Considering both the anti-loosening and clamping abilities, $\alpha=\alpha_{middle}$ can be selected as the most suitable pitch difference. It should be noted that the most desirable pitch difference of $\alpha=\alpha_{middle}$ is obtained under clearance C_y=125 µm.

Table 4.3 shows that the prevailing torque increases as the pitch difference increases. The effect of pitch difference on the screwing process will be discussed in the next section in order to clarify the mechanism of the anti-loosening. Figure 4.6 shows the results for all of the specimens in the loosening experiment, from which it is seen that the anti-loosening performance can be realized when the pitch difference is equal or larger than $\alpha 6$.

Table 4.4: 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	
$\alpha_{verylarge}$	No.9	-	-	-	＞70	-

(*Axial force is estimated from Figure 4.4)

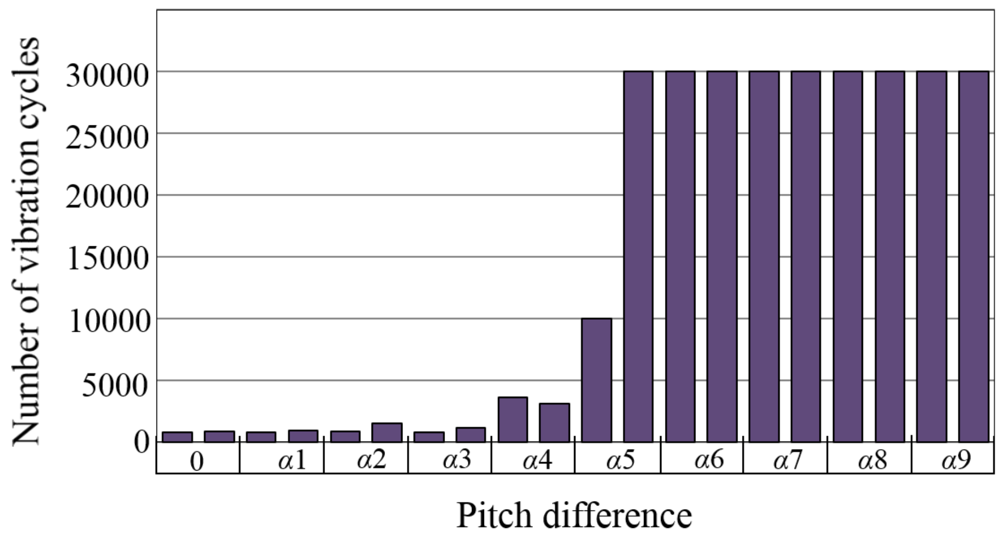


Figure 4.6: Loosening experiment results

The anti-loosening performance can also be studied by applying a same clamping force to all specimen with different tightening torques. In that case, it can be predicted that larger tightening torques will appear for specimen having larger pitch differences. One advantage of using a same torque is that the tightening torque can be directly controlled by using the torque wrench. If a same clamping force is used, we have to refer the relation between torque and clamping force as obtained in Figure 4.4, then, apply a corresponding tightening torque to each specimen. Therefore, in this study, a same torque was used to study the anti-loosening performance.

4.4 Finite element analyses to investigate bolt axial force between the nut threads

The previous discussion shows that $\alpha=\alpha_{large}$ has a good anti-loosening performance but insufficient clamping ability. This is due to the large deformation of the threads during the tightening process. To confirm this, an axisymmetric model of the bolt-nut connection is constructed by using the FEM code MSC. Marc/Mentat 2012. The material of the bolt is SCM435 and the material of the nut is S45C to match the experimental conditions. Herein, bolt, nut and clamped body are modeled as three bodies in contact. In the tightening process, the accumulated pitch difference causes the axial force between the bolt threads engaged with the nut thread. In this modelling, the tightening process is expressed by shifting the nut thread position discontinuously, one by one, at the thread interval. As the nut is moving towards the bolt head, the accumulation of the pitch difference leads to a slight overlap between the bolt threads and the nut threads. The direct constraints method is invoked in the detection of contact in MSC. Marc. Then, the nut is compressed while the engaged part of bolt is stretched in the simulation. In this way, the axial force between the bolt threads can be investigated step by step as the nut is shifted onto the bolt. It should be noted that this axisymmetric simulation may include some numerical errors but the real axial force between the bolt threads is difficult to be measured experimentally because the nut is engaged at this position. The isotropic hardening law was assumed with von Mises yield criterion. Friction coefficient of 0.3 was assumed and Coulomb friction was used. In the next sub-section, the results for $\alpha=\alpha_{middle}$ and $\alpha=\alpha_{verylarge}$ will be compared.

4.4.1 Bolt axial force between the nut threads F_α before the nut touching the clamped body

Since the nut pitch is larger than the bolt pitch, bolt axial force F_α in tension appears between the nut threads. F_α corresponds to prevailing torque T_p. It should be noted that F_α is different from the bolt axial force (clamping force) obtained in Figure 4.4. Here, the axial force F_α between bolt threads arising from the accumulation of pitch difference in the tightening process. The real axial force between the bolt threads is difficult to be measured experimentally because the nut is engaged at this position. Figure 4.7 indicates F_α for $\alpha=\alpha_{middle}$ before the nut touching the clamped body from Position A to nut Position G. Position A is where the prevailing torque appears, and Position B is where the nut shifted at the pitch interval from Position A and so on. Finally, Position G is where the nut starts contacting the clamped body. From Position A to Positions B, C, the whole nut is being shifted onto the bolt, and therefore the accumulated pitch difference affects the results. From Position C to Positions D, E, F, G, the pitch difference is not accumulated since the whole nut is already on the bolt.

Figure 4.8 shows F_α for $\alpha=\alpha_{verylarge}$ from Position A to Position H. Position A is where the prevailing torque appears, and Position H is where the nut starts contacting the clamped body. Different from $\alpha=\alpha_{middle}$, as the nut is being shifted onto the bolt, the bolt axial forces corresponding to No. 1 and No. 8 nut threads become smaller than the middle part of the engaging bolt threads. This result is due to No.2, 7 nut threads also contact as well as No.1, 8 nut threads under $\alpha=\alpha_{verylarge}$. On the other hand, under $\alpha=\alpha_{middle}$ only No.1, 8 nut threads contact to bolt threads.

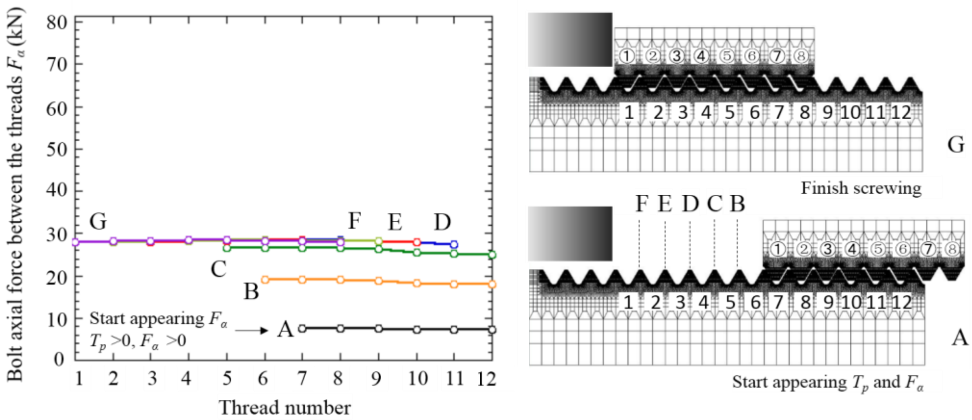


Figure 4.7: Bolt axial force for α=α_{middle} for the screwing process from Position A to Position G

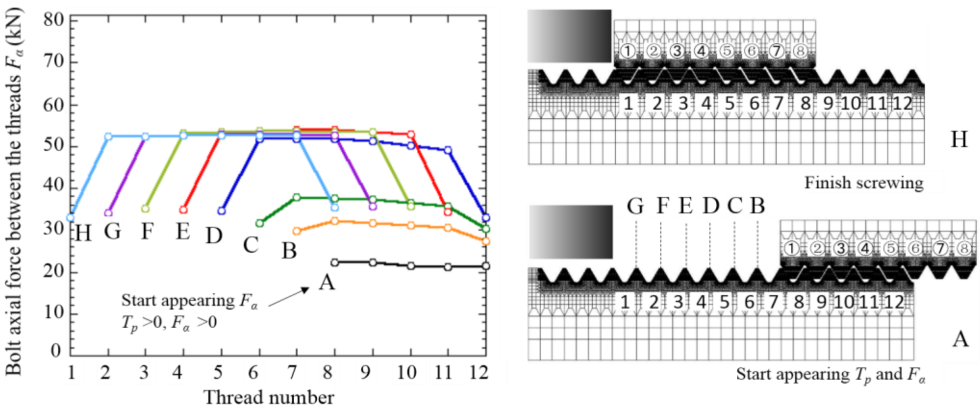


Figure 4.8: Bolt axial force for α=$\alpha_{verylarge}$ for the screwing process from Position A to Position H

4.4.2 Plastic deformation appearing at the bolt threads

Figure 4.9 shows the equivalent plastic strain of threads for α=α_{middle} at Position G where the nut touches the clamped body in the experiment. Similarly, Figure 4.10 shows the equivalent plastic strain of threads for α=$\alpha_{verylarge}$ at Position H. It may be concluded that too large pitch difference α=$\alpha_{verylarge}$ may cause large deformation at nut threads resulting in deterioration of bolt clamping ability. On the other hand, suitable pitch difference may cause suitable deformation keeping clamping ability.

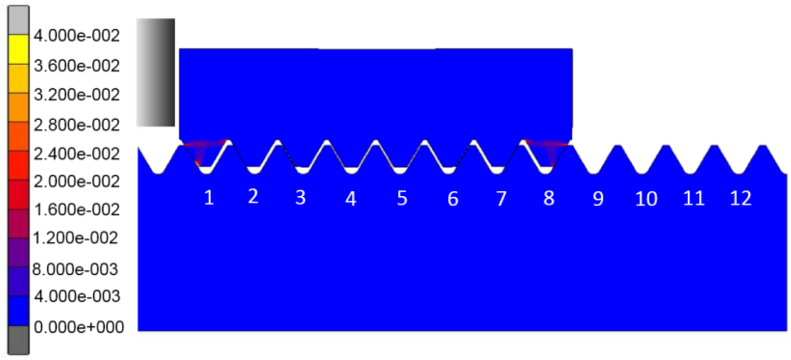


Figure 4.9: Equivalent plastic strain for $\alpha=\alpha_{midlle}$ at Position G

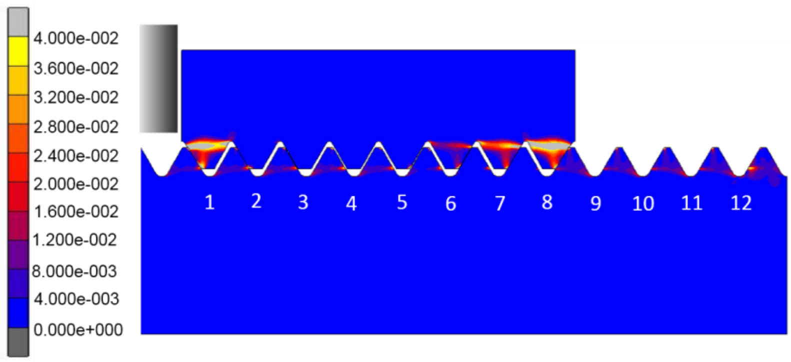


Figure 4.10: Equivalent plastic strain for $\alpha=\alpha_{verylarge}$ at Position H

4.5 Conclusions

In this Chapter, with varying the pitch difference α, the prevailing torque necessary for the nut rotation before the nut touching the clamped body was measured experimentally. Next, the bolt axial force was investigated in relation to the prevailing torque. The loosening experiment was conducted under a series of pitch differences. The finite element analyses were applied to investigate the bolt axial force between nut threads as well as the deformation at the bolt and nut threads. The conclusions can be summarized as follows:

(1) It is found that the large value of α may provide large prevailing torque that causes anti-loosening performance although too large α may deteriorate the bolt clamping ability.

(2) Considering both the anti-loosening performance and the clamping ability, $\alpha=\alpha_{middle}$ is found to be the most desirable pitch difference. This is because that the nuts did not drop for $\alpha=\alpha_{middle}$ without losing clamping ability.

(3) The anti-loosening experiment shows the nuts did not drop for $\alpha=\alpha_{large}$ also but clamping ability is deteriorated. The FEM analyses show that for $\alpha=\alpha_{verylarge}$ the large plastic deformation happens at nut threads.

Chapter 5

The mechanism of fatigue life improvement

5.1 Overview

Chapter 3 reported that fatigue life could be improved by introducing the pitch difference of $\alpha=\alpha_{small}$. Moreover, in Chapter 4, the experimental results showed that $\alpha=\alpha_{middle}$ is the most desirable pitch difference to realize the anti-loosening performance. As a further research, in this Chapter, more detailed fatigue experiment is conducted systematically under a series of cyclic fatigue loads for three types of specimens, i.e. $\alpha=0$, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, where $\alpha=0$ 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 FEM is applied to analyze the stress amplitude and average stress at each bolt threads. The mechanism of fatigue life improvement is considered by comparing the experimental results to those obtained using the finite element method. Table 5.1 summarizes the pitch differences reported in this whole text.

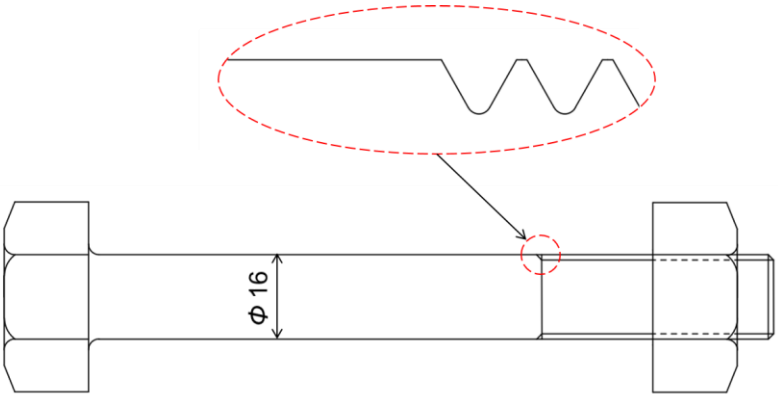
Table 5.1: Pitch difference α in each Chapter

		Pitch difference α									
		$0 < \alpha1 < \alpha2 < \alpha3 < \alpha4 < \alpha5 < \alpha6 < \alpha7 < \alpha8 < \alpha9$									
Chapter 3	Fatigue experiment	0	$\alpha_{verysmall}$	α_{small}							
	FE analysis	0	$\alpha_{verysmall}$	α_{small}		$\alpha4$					
Chapter 4	Loosening experiment	0	$\alpha1$ (=$\alpha_{verysmall}$)	$\alpha2$ (=α_{small})	$\alpha3$	$\alpha4$	$\alpha5$	$\alpha6$ (=α_{middle})	$\alpha7$	$\alpha8$ (=α_{large})	$\alpha9$
	FE analysis							α_{middle}			
Chapter 5	Fatigue experiment	0		α_{small}				α_{middle}			
	FE analysis	0		α_{small}				α_{middle}			

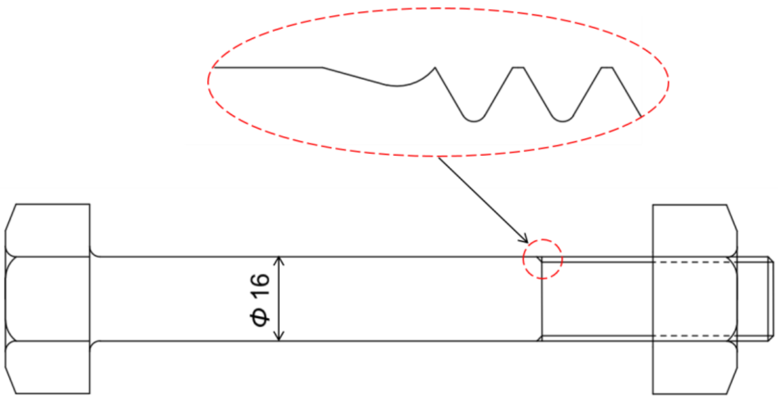
5.2 Fatigue experiment

5.2.1 Specimens and experimental conditions

In Chapter 3, fracture happened at starting thread portion for $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$ since higher stress concentrations appear at this position. In Chapter 5, therefore, the geometry at the starting thread was slightly changed as shown in Figure 5.1 in order to avoid the fracture. The experimental device used in the fatigue tests is shown in Figure 5.2. The bolt specimens are subjected to a series of repeated loadings. Table 5.2 shows the experimental loading conditions and the corresponding stress according to the bottom cross sectional area of the bolt A_R=141 mm^2.



(a) Bolt specimens used in Chapter 3



(b) Bolt specimens used in Chapter 5

Figure 5.1: Change of geometric shape at the starting thread portion in bolt specimens

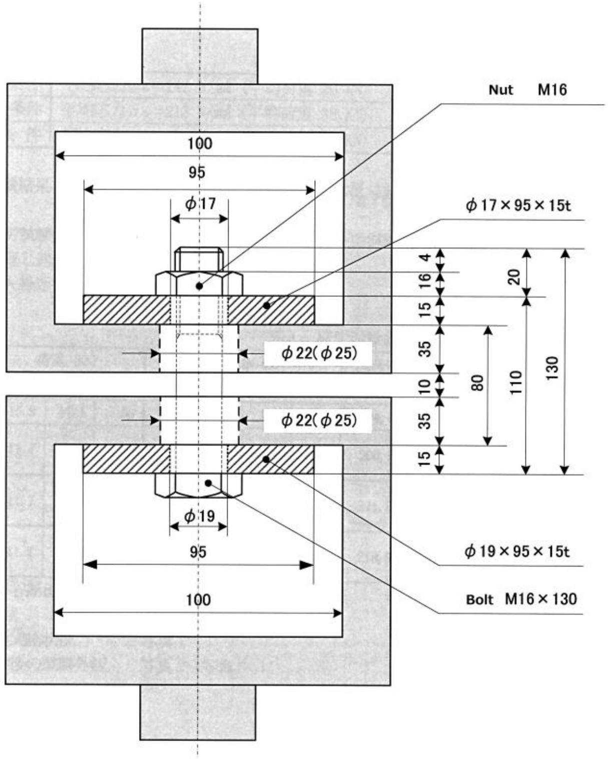


Figure 5.2: Experimental device (dimensions in mm)

Table 5.2: 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

5.2.2 Experimental results

Figures 5.3 shows 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 5.3 (a). For α=α_{small} and α=α_{middle}, the final fractured positions are between No.1 thread and No.3 thread. The fractured surfaces of α=α_{small} and α=α_{middle} are different from the one of α=0 because the surface is not flat.

(a) α=0

(b) α=α_{small}

(c) α=α_{middle}

Figure 5.3: Fractured specimens (σ_a=100 MPa)

The S-N curves with fatigue limit at N=2×10^6 are obtained as shown in Figure 5.4. 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 5.3 shows the fatigue life normalized by the results of α=0. When the stress amplitude is above 80 MPa, the fatigue life for α=α_{small} is about 1.5

times and the fatigue life for α=α_{middle} 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.

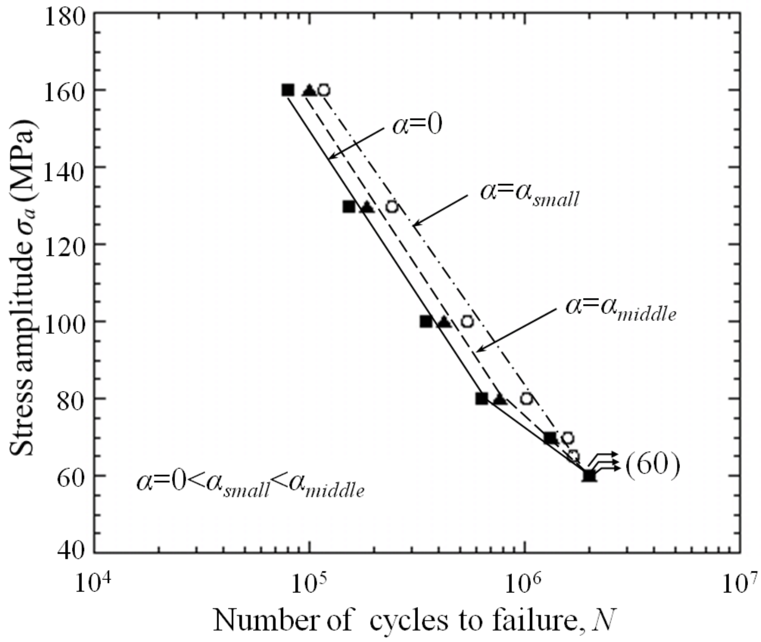


Figure 5.4: *S*-*N* curves

Table 5.3: 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
	α_{small}	1.49	1.60	1.53	1.61	1.21
	α_{middle}	1.26	1.22	1.20	1.21	1.02
N	0	79,030	151,860	350,760	636,490	1,307,860
	α_{small}	117,550	242,810	536,690	1,025,370	1,586,980
	α_{middle}	99,680	184,770	422,500	770,870	1,327,860

It is found that a remarkable difference of fatigue life appeared for α=0 in the first fatigue experiment (Chapter 3) and second fatigue experiment (Chapter 5) with the same stress amplitude 130 MPa. This scatter is considered to be caused by the different rolling lots. It is known that the specimens from different rolling lots have large variation. Since the used specimens in Chapter 3 and Chapter 5 come from different rolling lots, the fatigue life was somewhat different for the specimens having similar configuration. However, for the specimens from same rolling lot, the variation was small. As shown in Figure 5.4, the *S-N* curves distinct depend on the pitch difference.

5.3 FEA on the stress state at bolt threads

To analyze the stress states at the bottom of the bolt threads, finite element models are created by using FEM with MSC.Marc/Mentat 2012. Three models have different pitch differences, i.e. α=0, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, in accordance with the experimental configurations of the test specimens. Figure 5.5 shows the axisymmetric model of the bolt-nut connection and the clamped plate. Figure 5.6 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 3.2 are used in the calculation.

Figure 5.7 shows the stress variations for σ_ψ, σ_θ and the equivalent von-Mises stress σ_{eq} at each bolt thread from No.5 thread to No.8 thread. Herein, the stress variation σ_ψ is taken into account. The position of the maximum stress amplitude is marked as shown in Figure 5.7. At each bolt thread from No.-3 thread to No.8 thread, the maximum stress amplitude and the average stress are investigated at the point where the maximum stress amplitude appears.

In the FE analysis $\sigma_{\psi\max}$ is the stress σ_ψ at each thread under the maximum load, and $\sigma_{\psi\min}$ is the stress σ_ψ at each thread under the minimum load. The maximum stress amplitude and average stress are investigated at the same angle ψ where the maximum stress amplitude appears.

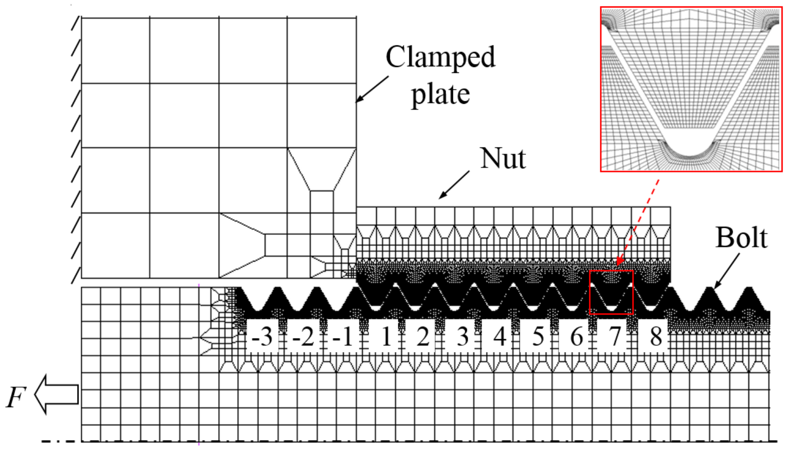


Figure 5.5: Axisymmetric finite element model of bolt-nut connections

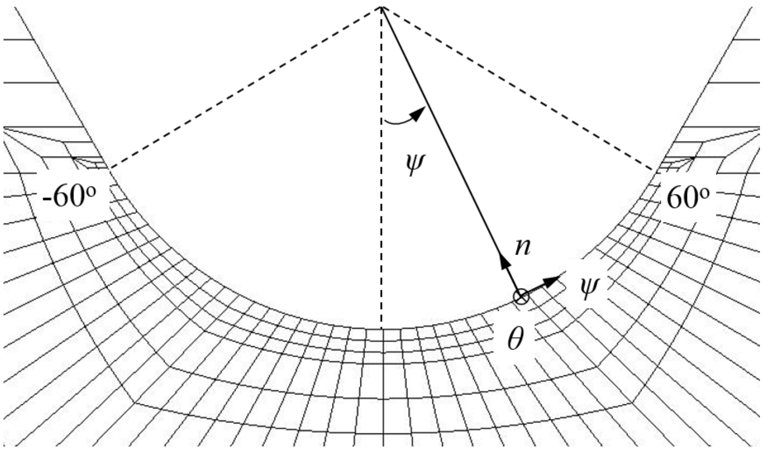
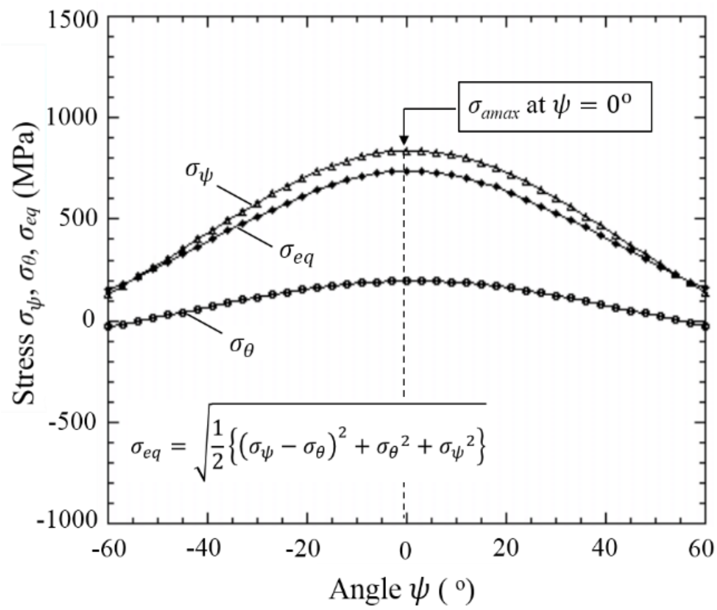
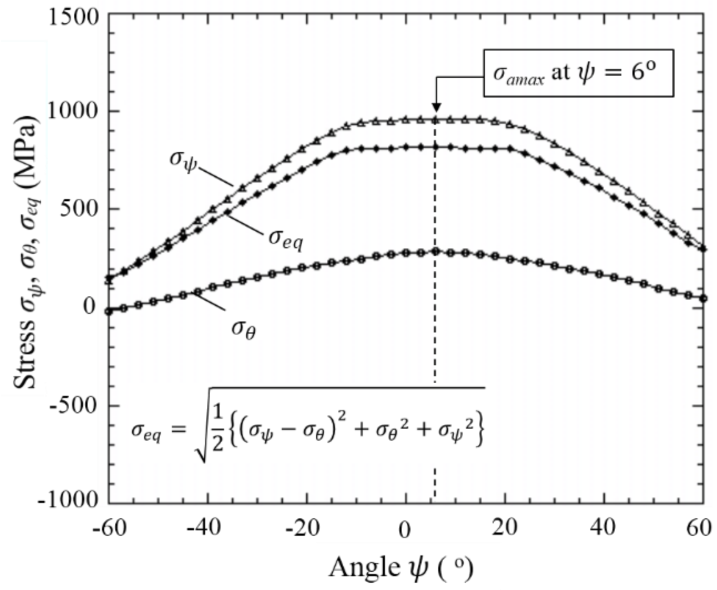


Figure 5.6: Local coordinate at bottom of bolt thread

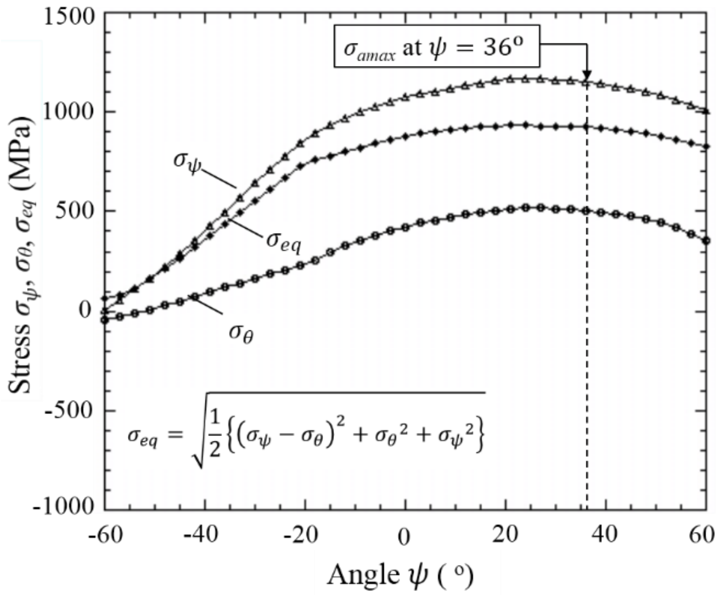


(a) Stress at bottom of thread No.5

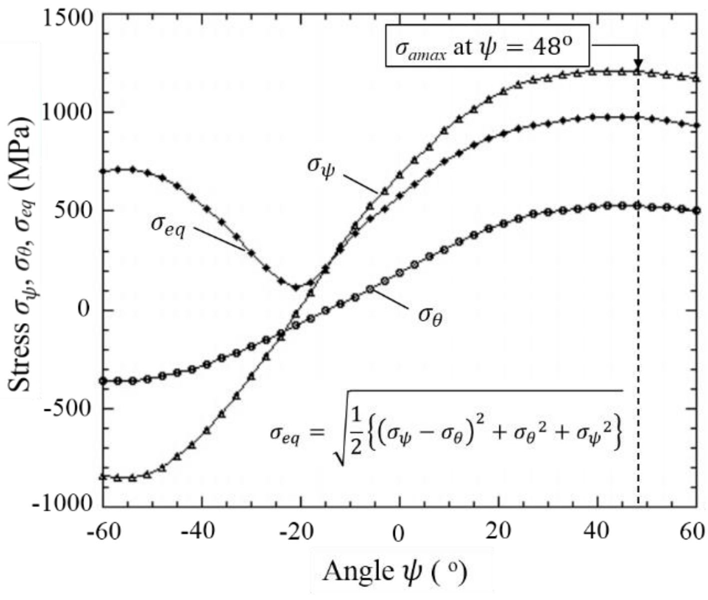


(b) Stress at bottom of thread No.6

Figure 5.7: Stress at bottom of bolt thread (α=α_{middle}, F=30+14.1 kN) (Cont.)



(c) Stress at bottom of thread No.7

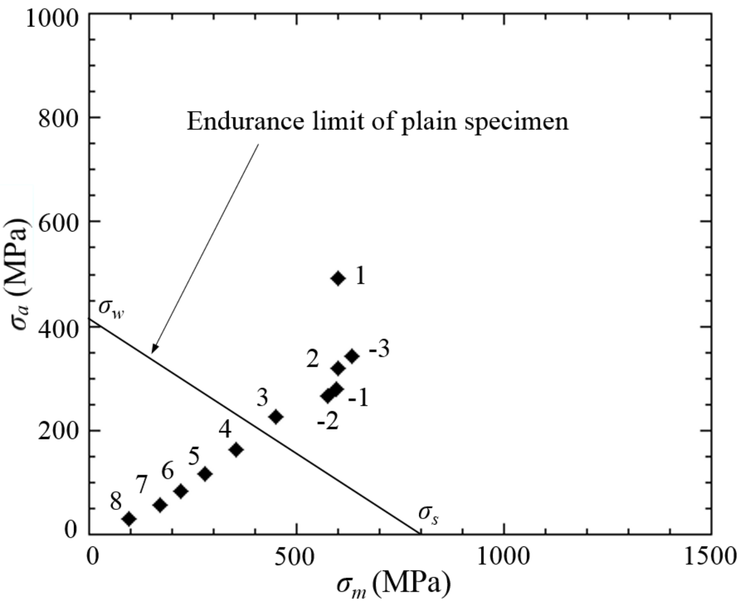


(d) Stress at bottom of thread No.8

Figure 5.7: Stress at bottom of bolt thread (α=α_{middle}, F=30+14.1 kN)

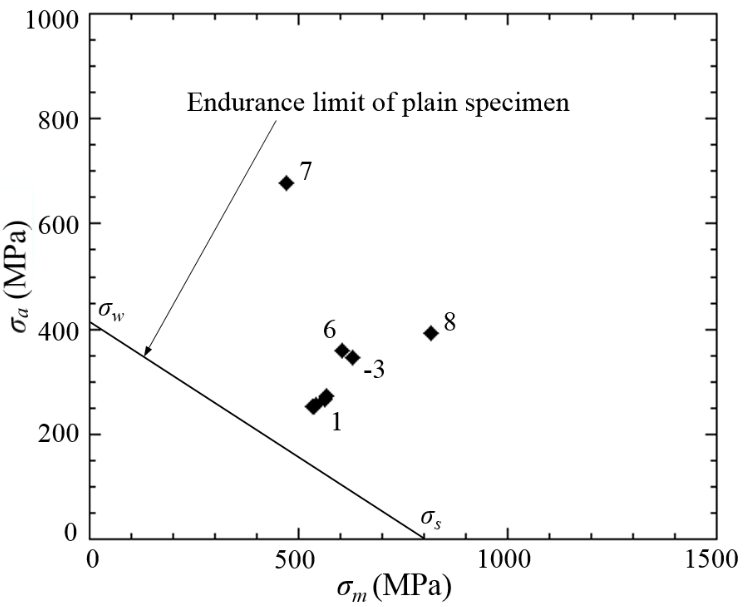
The endurance limit diagrams are obtained as shown in Figure 5.8. In the endurance limit diagram, the Soderberg line [45] is plotted. Herein, the point σ_w represents the fatigue strength corresponding to the case of complete reversal (σ_m =0), and the point σ_{sl} corresponds to the yield strength.

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. The Soderberg line indicates the endurance limit diagram for plain specimen. Therefore, the stress data plotted above the Soderberg line does not represent the real fracture at the bolt thread. The usage of endurance limit diagram with Soderberg line is intended to make a comparison of the relative severity of each bolt threads. From Figure 5.8 (a), it can be seen that for the standard bolt-nut connections, No.1 thread has the highest stress amplitude, which corresponds to the fracture position in the fatigue experiment as illustrated previously. In Figure 5.8 (b), when a pitch difference of α=α_{small} is introduced, on one hand the stress amplitude decreases at No.1 thread and on the other hand, the stress amplitude at No.6, No.7 and No.8 threads increases significantly. For α=α_{middle}, the severe stress state occurs nearby No.1 and No.7 threads as shown in Figure 5.8 (c).

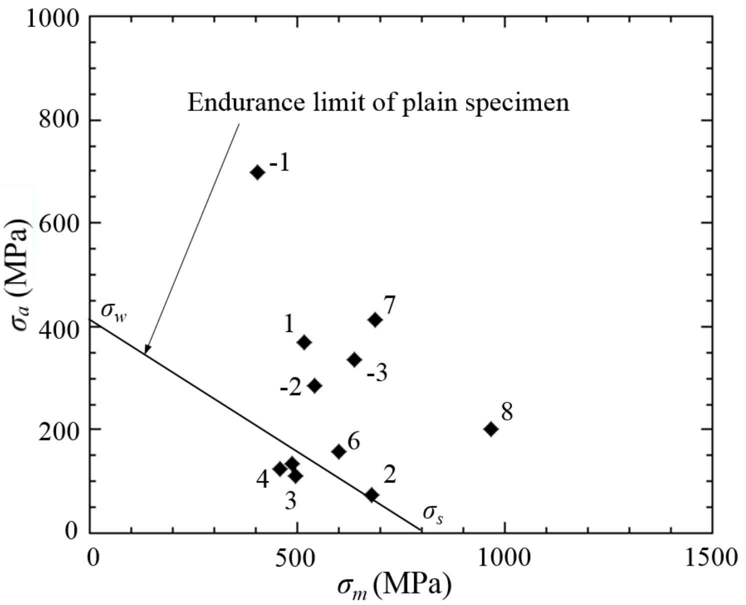


(a) α=0

Figure 5.8: Endurance limit diagrams (σ_a=100 MPa) (Cont.)



(b) α=α_{small}



(c) α=α_{middle}

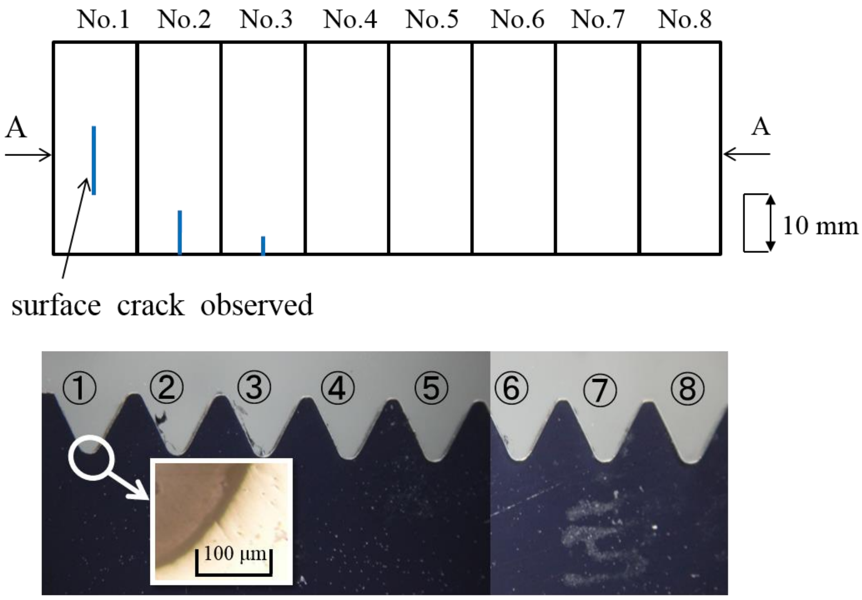
Figure 5.8: Endurance limit diagrams (σ_a=100 MPa)

5.4 Crack observation

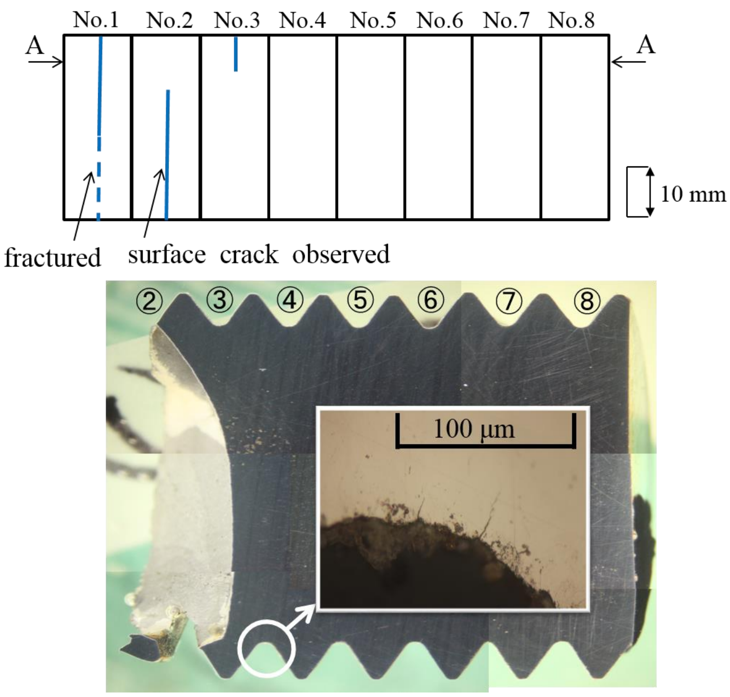
Figure 5.9 shows the observed trajectory of cracks along the longitudinal cross section of the specimens at the fatigue stress amplitudes σ_a = 60 MPa, 70 MPa, 100 MPa and 160 MPa. For α=0, small cracks occur at No.1 thread and No.2 thread. For $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, large cracks occur between No.2 thread and No.7 thread. Moreover, with increasing the stress amplitude, the cracks show different shapes indicating changes in mode mixity.

It can be seen in Figure 5.9 that for the standard bolt-nut connections α=0, the crack occurs at thread No.1 causing finial fracture. However, for the specimens of $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, the initial cracks start at No.5 thread or No.6 thread, extending toward No.1 thread and finally fracture happen nearby No.1 thread. From the S-N curves and the observations of crack trajectories in Figure 5.9, 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.

According to the crack observation at the fatigue limit σ_a=60 MPa, for α=0, some non-propagating cracks were observed. However, for $\alpha=\alpha_{small}$, and $\alpha=\alpha_{middel}$, some propagating cracks occurred. Therefore, it can be concluded that the actual fatigue limit of $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middel}$ should be lower than 60 MPa. This decrease in fatigue limit was caused by the easily happened cracks at No.5-6 threads when a pitch difference was introduced, although the pitch difference changed the fracture mechanism of bolt and contributed to the fatigue life improvement.

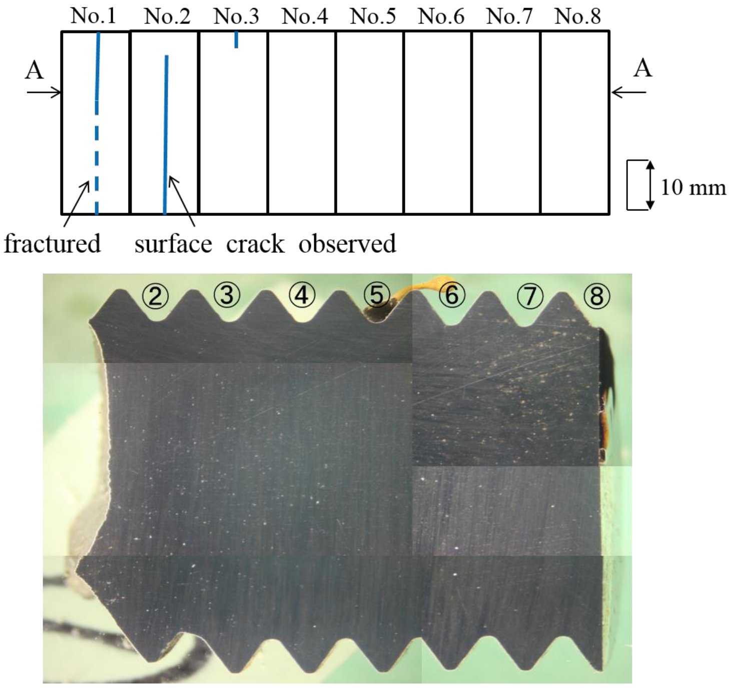


(α=0, σ_a=60 MPa)

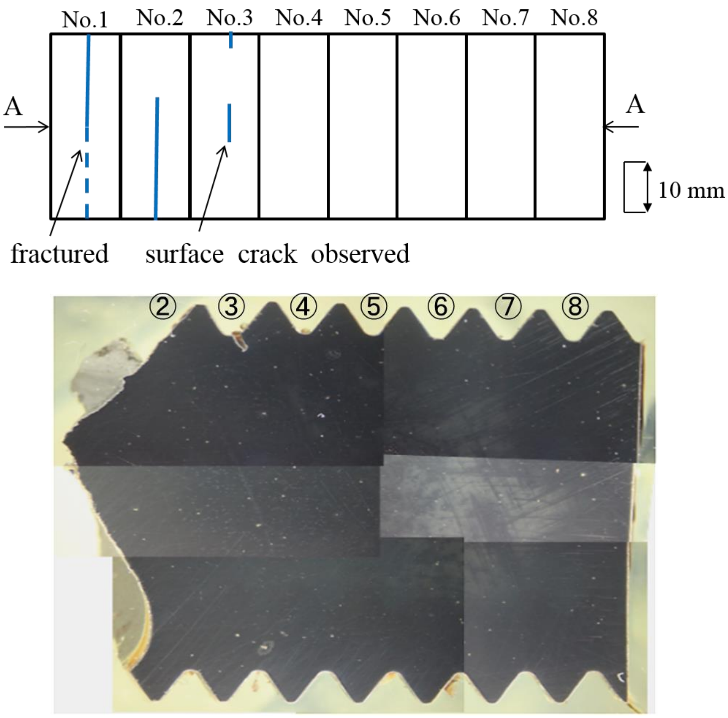


(α=0, σ_a=70 MPa)

Figure 5.9: Observation of crack trajectories (Cont.)

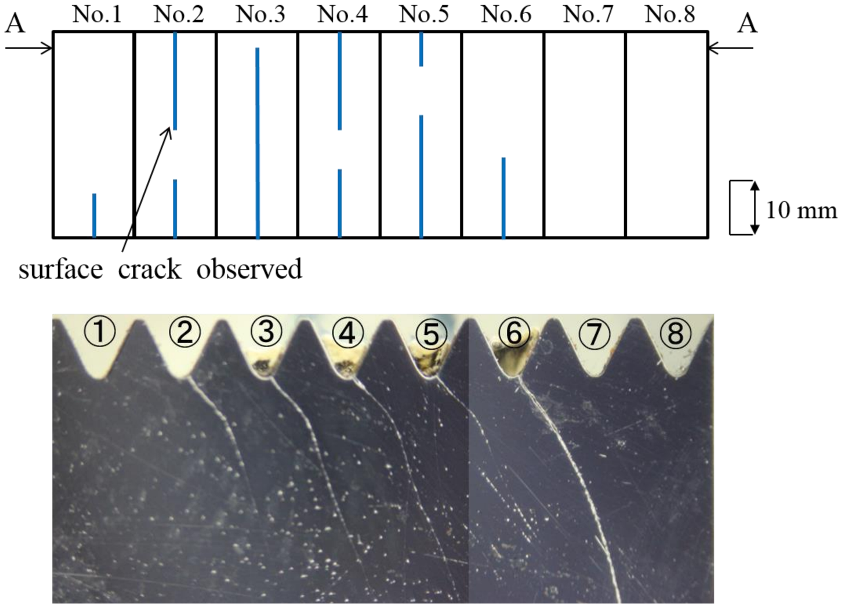


(α=0, σ_a=100 MPa)

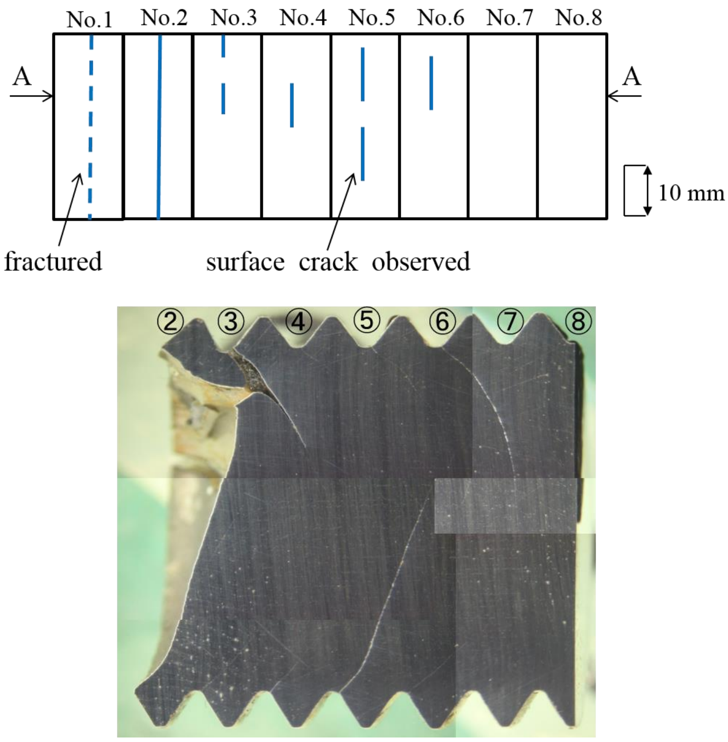


(α=0, σ_a=160 MPa)

Figure 5.9: Observation of crack trajectories (Cont.)

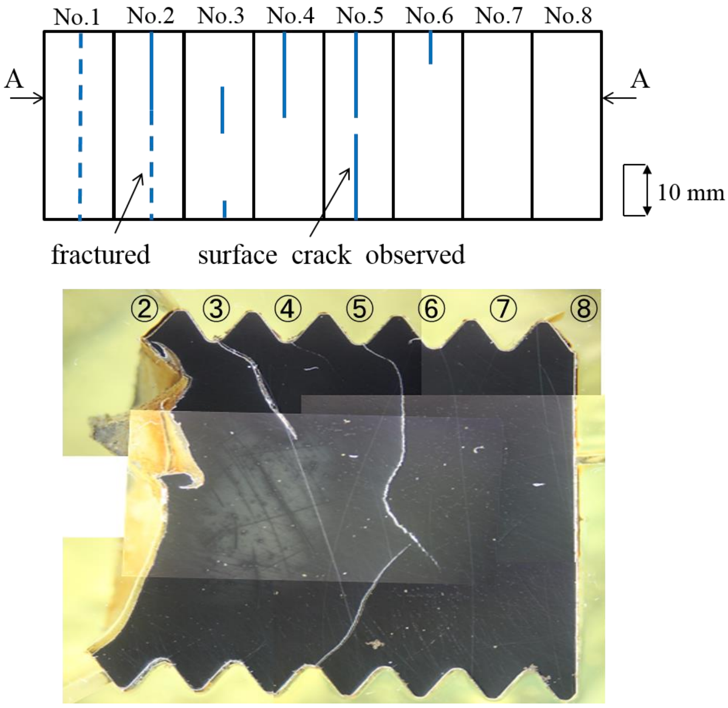


(α=α_{small}, σ_a=60 MPa)

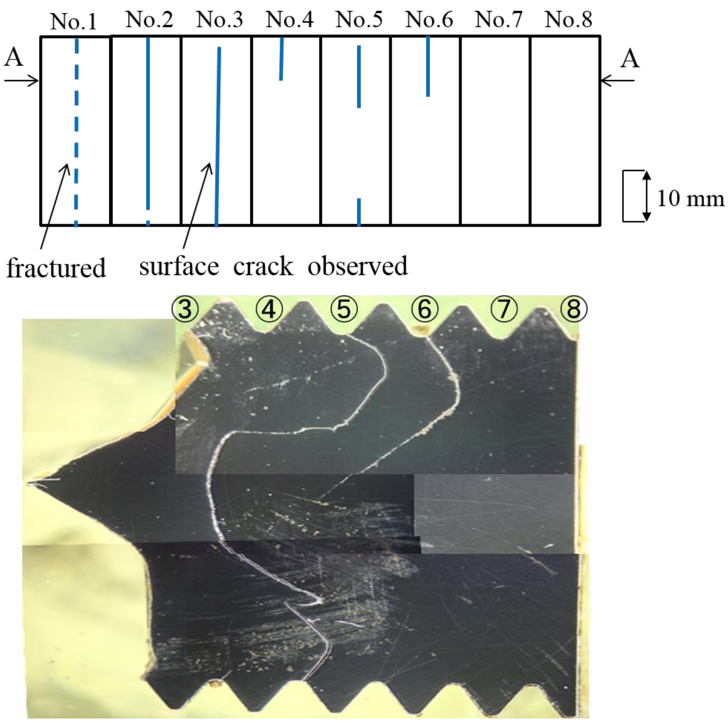


(α=α_{small}, σ_a=70 MPa)

Figure 5.9: Observation of crack trajectories (Cont.)

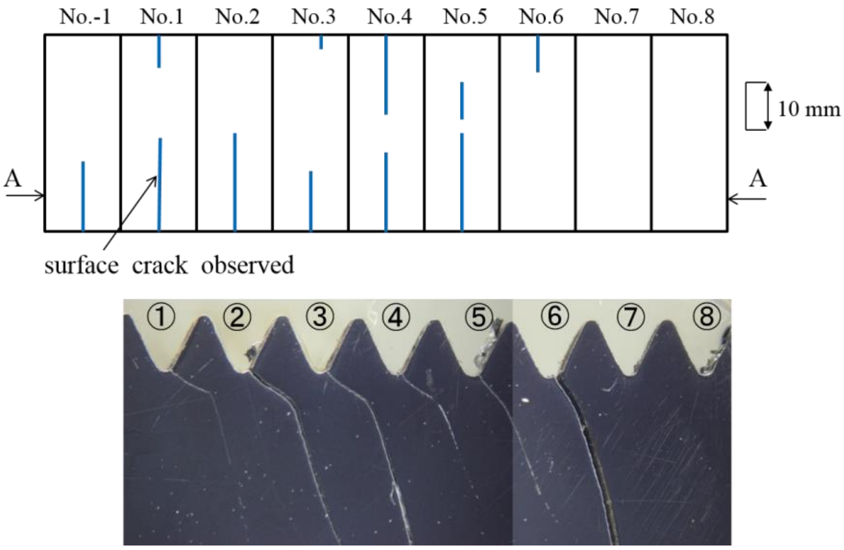


(α=α_{small}, σ_a=100 MPa)

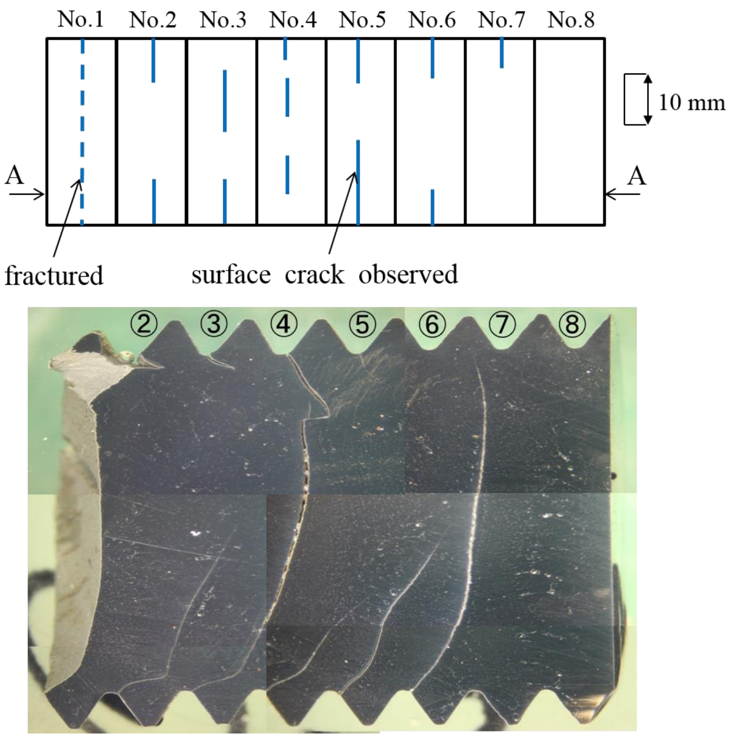


(α=α_{small}, σ_a=160 MPa)

Figure 5.9: Observation of crack trajectories (Cont.)

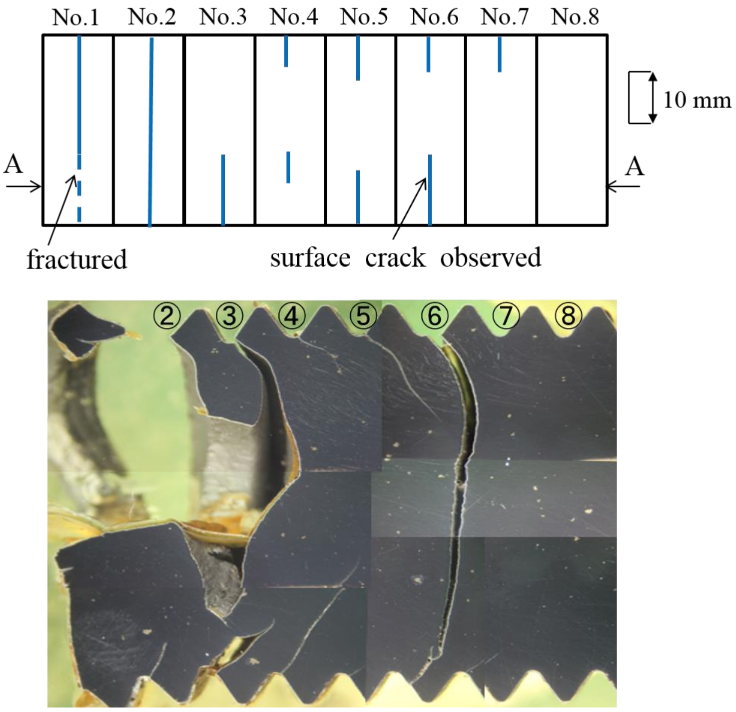


(α=α_{middle}, σ_a=60 MPa)

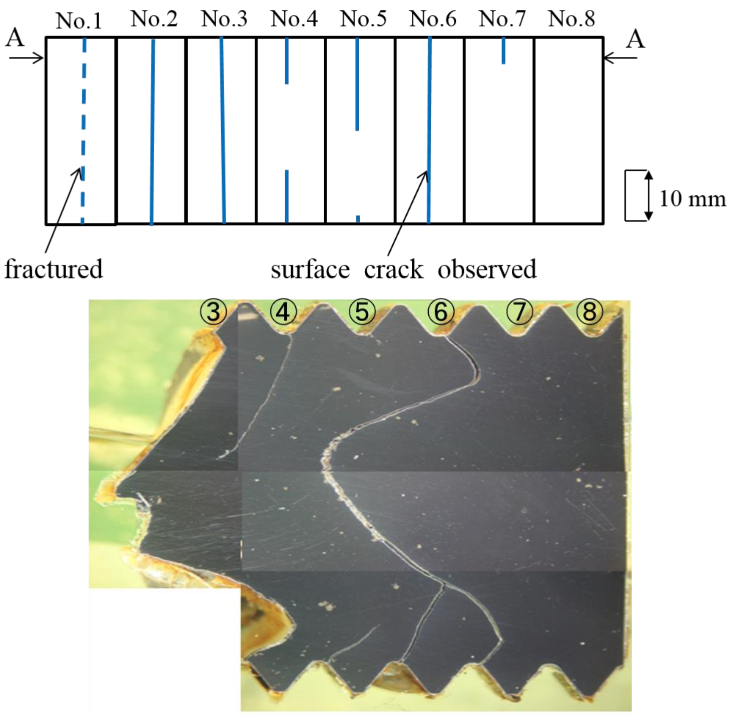


(α=α_{middle}, σ_a=70 MPa)

Figure 5.9: Observation of crack trajectories (Cont.)



($\alpha = \alpha_{middle}$, σ_a=100 MPa)



($\alpha = \alpha_{middle}$, σ_a=160 MPa)

Figure 5.9: Observation of crack trajectories

5.5 Effects of incomplete nut thread

In the above discussion, the complete thread model of 8-thread-nuts were considered by FE analyses, but usually as shown in Figure 5.10 both ends of nuts have chamfered corners to make bolt inserting smoothly, and those nuts were used in this fatigue experiment.

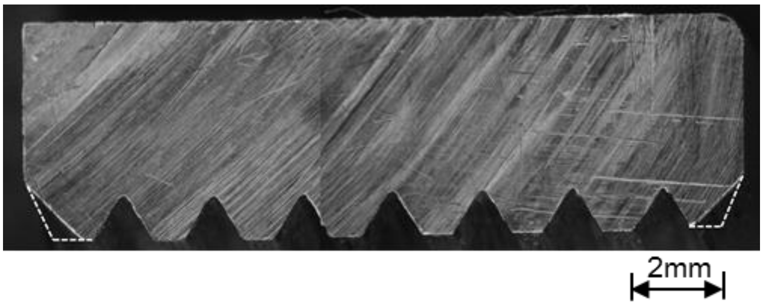


Figure 5.10: Incomplete threads at nut ends by cut away

In the first place, therefore, the chamfered corner is modeled by an incomplete thread model A as shown in Figure 5.11. Figure 5.12 shows FE mesh for model A and Figure 5.13 shows the endurance limit diagram when α=α_{small} and σ_a=100 MPa. From Figure 5.13, it is seen that the No.8 thread stress decreases and the No.6 thread stress increases. However, the No.6 thread stress is not most dangerous because No.8 thread still contacts to the bolt thread.

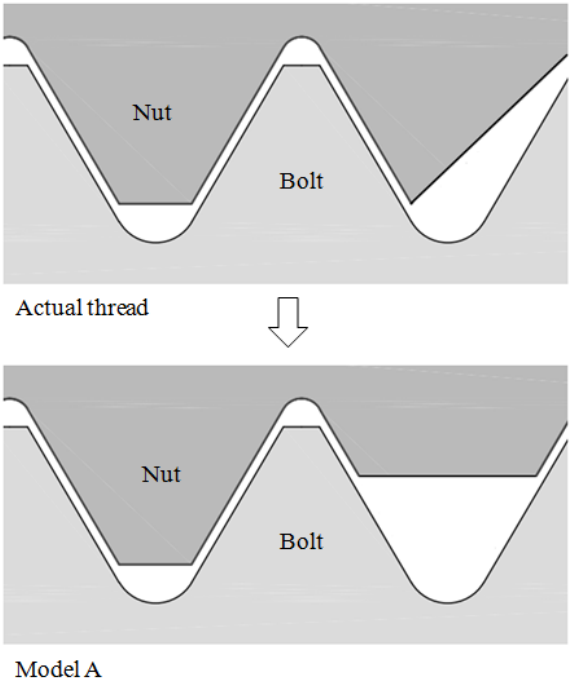


Figure 5.11: Incomplete thread model A

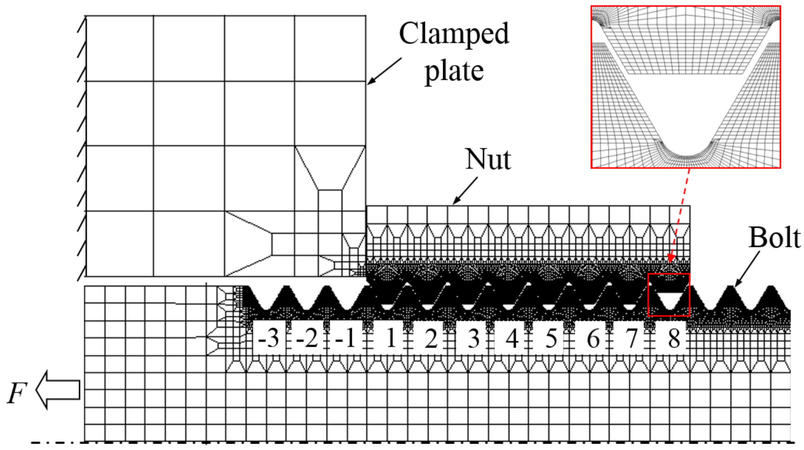


Figure 5.12: Axisymmetric finite element mesh for model A considering incomplete thread

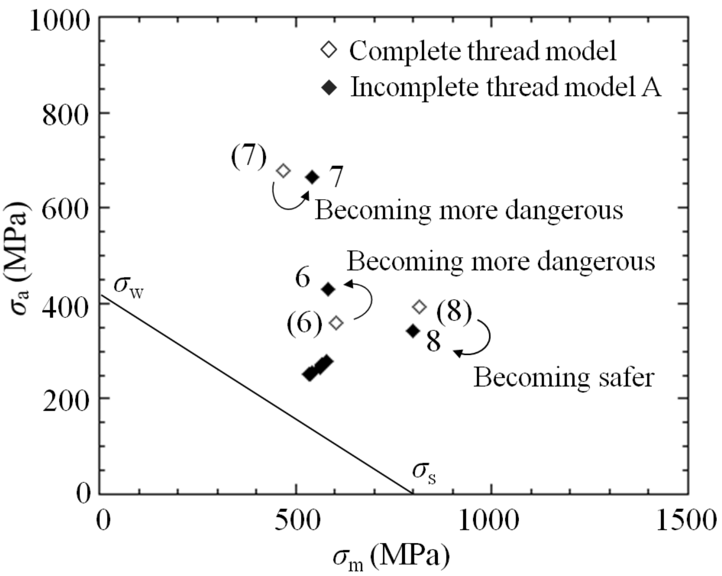


Figure 5.13: Endurance limit diagram for $\alpha=\alpha_{small}$ when σ_a=100MPa, incomplete thread model A vs. complete thread model

Therefore, thread model B as shown in Figure 5.14 is considered in the next where the incomplete nut thread does not contact bolt thread anymore because both ends of chamfered nut ends do not contact bolt thread. Figure 5.15 shows FE mesh for model B. Figure 5.16 shows each thread stress when the maximum and minimum load F=30±14.1 kN are applied. For α=0, the maximum stress amplitude appears at No.2 thread as shown in Figure 5.16 (a). Therefore the analytical result coincides with the experimental result in Figure 5.9. When $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, the maximum stress amplitude appearing at No.6 thread, which is close to the crack location in Figure 5.9.

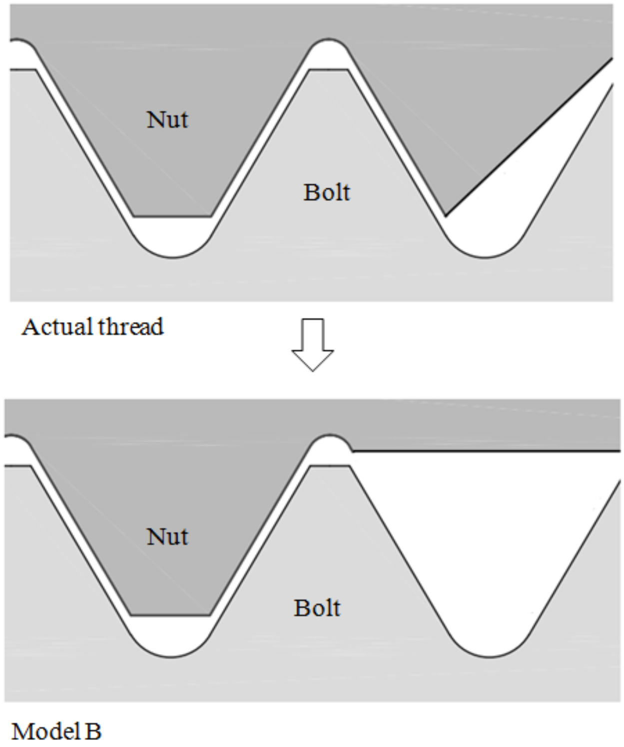


Figure 5.14: Incomplete thread model B

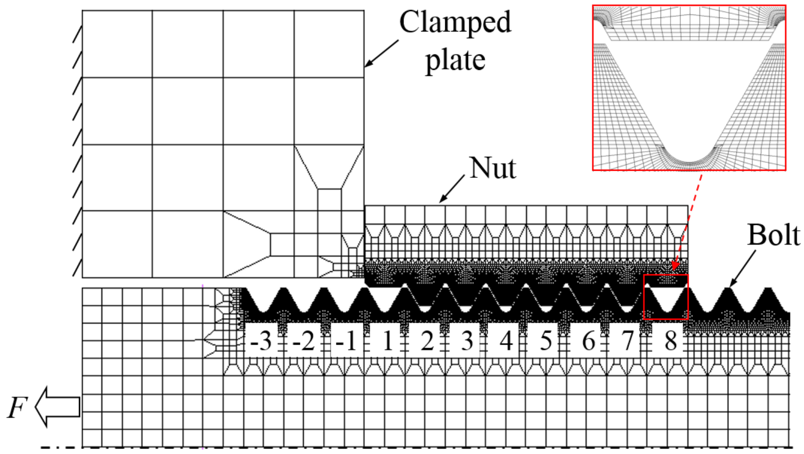
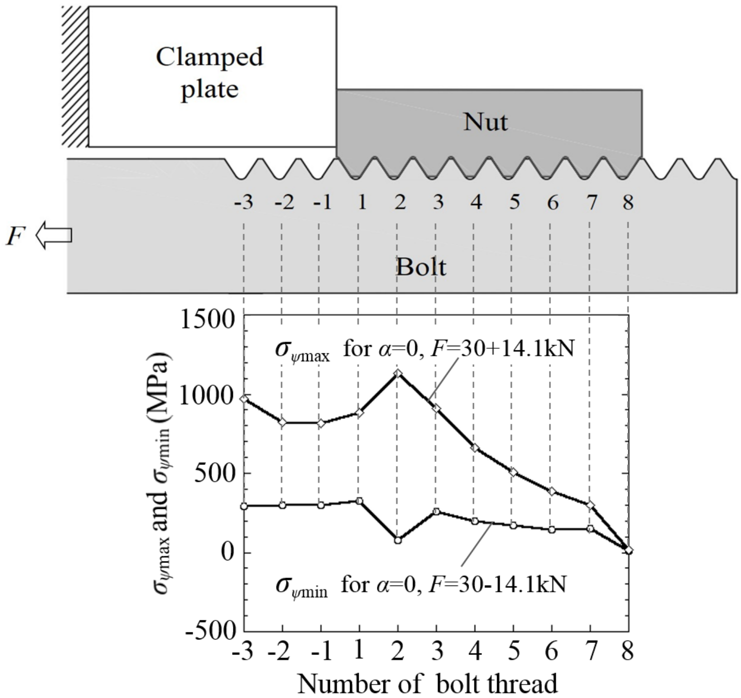
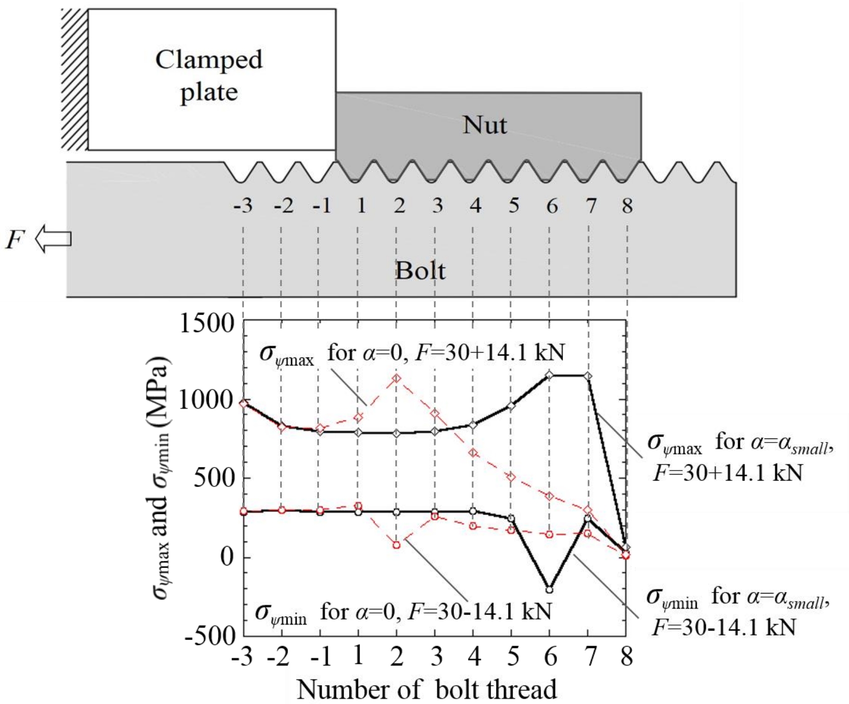


Figure 5.15: Axisymmetric finite element mesh for model B considering incomplete threads at both ends of nut

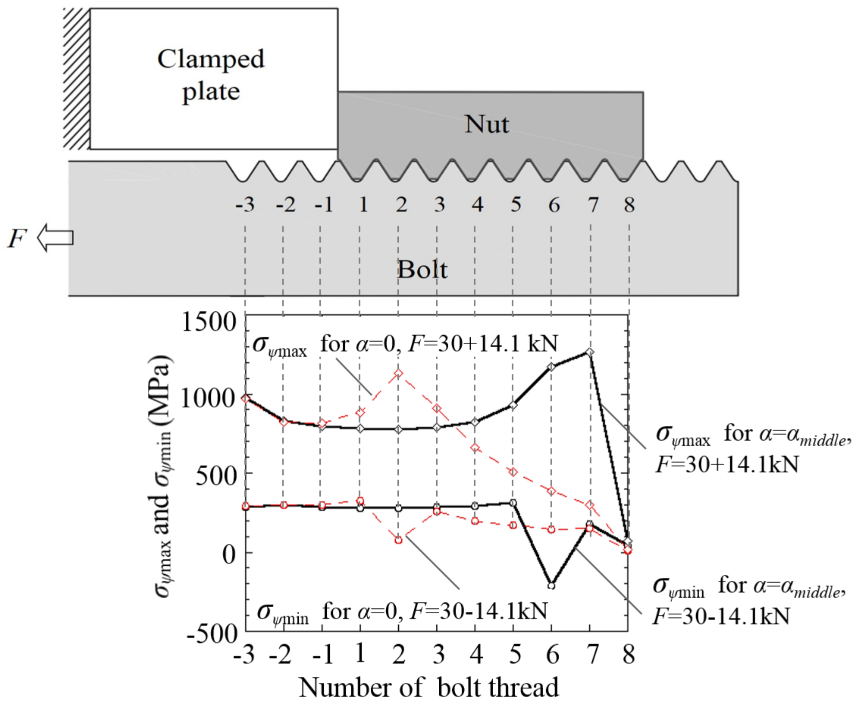


(a) α=0



(b) α=α_{small}

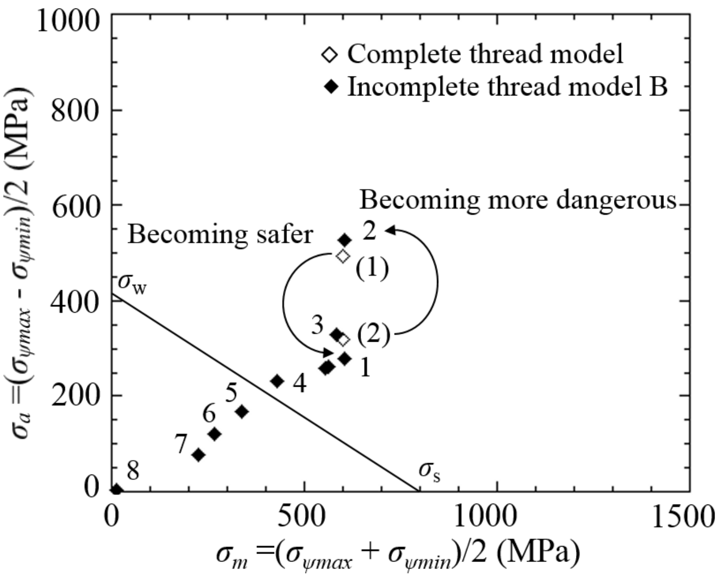
Figure 5.16: Maximum stress $\sigma_{\psi max}$ and minimum stress $\sigma_{\psi min}$ at each thread for model B (Cont.)



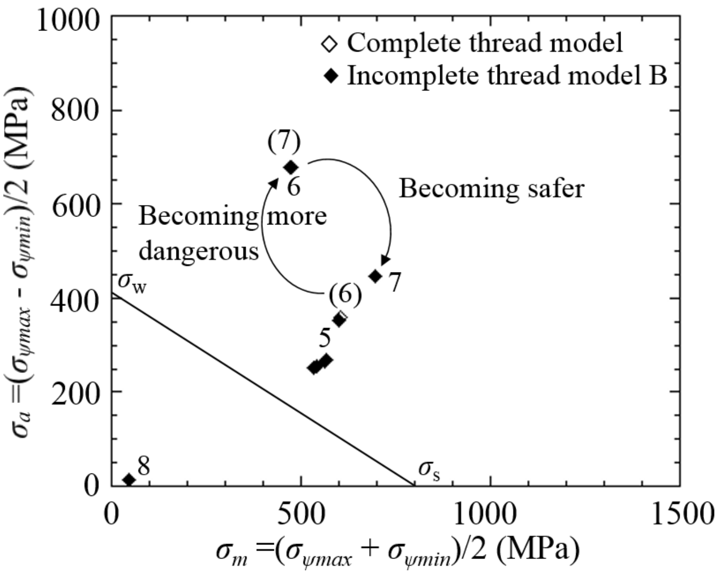
(c) $\alpha=\alpha_{middle}$

Figure 5.16: Maximum stress $\sigma_{\psi max}$ and minimum stress $\sigma_{\psi min}$ at each thread for model B

Figure 5.17 shows the endurance limit diagrams for α=0, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$. By changing 8-thread-model to 6-thread-model B, the most dangerous thread for α=0 is changed from No.1 to No.2. For $\alpha=\alpha_{small}$, No.6 thread becomes the most dangerous, similar to the crack observation results. For $\alpha=\alpha_{middle}$, the most dangerous No.6 thread has a good agreement with the position where the long crack appeared in Figure 5.9. It is seen that 6-thread-model B is useful for considering chamfered nut threads at both ends to explain the experimental results.

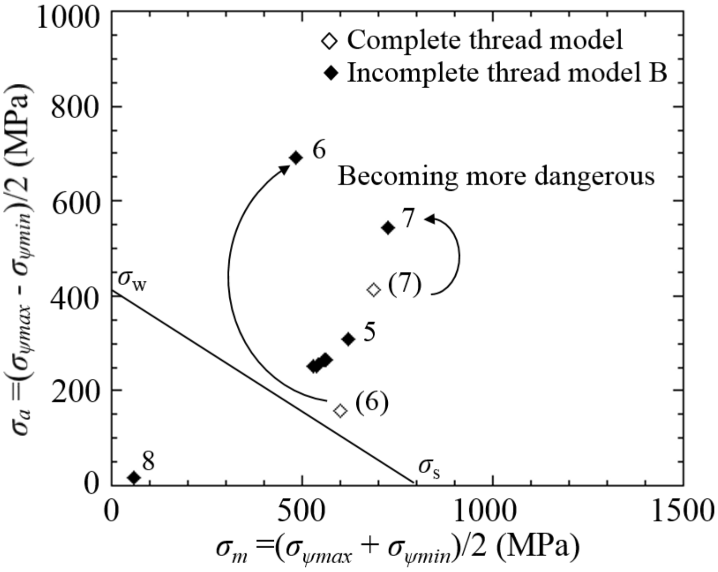


(a) α=0



(b) $\alpha = \alpha_{small}$

Figure 5.17: Endurance limit diagram for α=0, $\alpha = \alpha_{small}$ and $\alpha = \alpha_{middle}$ when σ_a=100 MPa, incomplete thread model B vs. complete thread model (Cont.)



(c) $\alpha=\alpha_{middle}$

Figure 5.17: Endurance limit diagram for α=0, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$ when σ_a=100 MPa, incomplete thread model B vs. complete thread model

One may think that replicating the actual geometry of chamfered threads in Figure 5.10 should be used in the modeling. However, the chamfered angle is not always the same. And the difference between the results for model B and the chamfered model is not very large for $\alpha=\alpha_{small}$ because of no contact at No.1 and No.8 threads. Only the largest difference appears at No. 1 thread for $\alpha=0$ because of no contact for model B and large contact stress for the chamfered model in Figure 5.10. In Figure 5.17, the results of model B shows that No. 1 thread contact for $\alpha=0$ can be realized at No. 2 thread contact in model B. In this study, therefore, simple incomplete thread model B has been used because the main target is to analyze the pitch difference α. The results of the chamfered model for standard bolt-nut $\alpha=0$ are indicated in the appendix A.

5.6 Conclusions

In this Chapter, based on the obtained results in the previous study, the fatigue experiment was conducted systematically for three levels of pitch difference, i.e. α=0, $\alpha=\alpha_{small}$, and $\alpha=\alpha_{middle}$. 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) It is found that $\alpha=\alpha_{small}$ is the most desirable pitch difference to extend the fatigue life of the bolt-nut connection. Compared with the standard bolt-nut connection, the fatigue life for $\alpha=\alpha_{small}$ can be extended to about 1.5 times.

(2) It is found that the stress amplitude at No.1 thread decreases significantly when a pitch difference is introduced. For $\alpha=\alpha_{small}$, the FE results shows that high stress amplitude occurs at No.6, No.7 and No.8 threads, which almost corresponds to the experimental observations.

(3) For the specimens with $\alpha=\alpha_{small}$, 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.

(4) The 6-thread-model is useful for analyzing 8-thread-nut contacting bolt threads because nuts always have chamfered threads at both ends. Then, the results are in good agreement with the experimental results.

Chapter 6

Conclusions and future research

6.1 Overview

The bolt-nut connections are important joining elements widely used to connect and disconnect members conveniently with low cost. To ensure the connected structure safety, the anti-loosening performance and high fatigue strength have been required. This paper therefore focuses on the effect of pitch difference between bolt-nut connection upon the fatigue life improvement and anti-loosing performance. In this study, a slight pitch difference α was considered for the M16 (JIS) bolt-nut connections. The fatigue experiment as well as the loosening experiment was conducted under different pitch differences. The finite element analyses were applied to investigate the stress state at the bolt threads.

6.2 Main conclusions

6.2.1 Conclusions for fatigue life improvement

The preliminary fatigue experiment was carried out under a certain level of stress amplitude by considering the pitch differences of α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$. Then, the fatigue experiment was conducted for α=0, $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$ with varying stress amplitude systematically. Furthermore, detailed investigation was performed on the fracture position as well as the crack configuration of the fractured specimens. By applying finite element analysis, the mechanism of fatigue life improvement was discussed in terms of the stress amplitude and average stress at each bolt thread. Finally, to improve the accuracy of the FE analysis, the chamfered corners at both nut ends are considered. The conclusions can be summarized as follows:

(1) By considering α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$, it is found that the fatigue life of bolt can be extended by introducing a suitable pitch difference. The FE analysis results show that both the average stress and stress amplitude at No.1 bolt thread can be reduced by introducing a pitch difference.

(2) When the pitch difference is small, usually only No. 7 and No. 8 bolt threads contact with nut threads even the clearance changes. On the other hand, when the pitch difference is large, the contact status of No. 1 bolt thread may change from left side contact to no contact. Therefore, with increasing the pitch difference, the clearance between bolt and nut affects the contact status more significantly.

(3) It is found that $\alpha=\alpha_{small}$ is the most desirable pitch difference to extend the fatigue life of the bolt-nut connection. Compared with the standard bolt-nut connection, the fatigue life for $\alpha=\alpha_{small}$ can be extended to about 1.5 times.

(4) For the specimens with $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, it is found that the crack occurs at No.5 or No. 6 thread in the first place, then extends toward thread No.1 until final fracture happens nearby No.1 thread. Therefore, the fatigue life of the bolt is extended compared with the standard bolt-nut connection.

(5) The 6-thread-model is useful for analysing 8-thread-nut contacting bolt threads because nuts always have chamfered threads at both ends. By using the 6-thread-model, the FE analysis shows that high stress amplitude occurs at No.6 thread for $\alpha=\alpha_{small}$ and $\alpha=\alpha_{middle}$, and the results are in good agreement with the experimental observations.

6.2.2 Conclusions for anti-loosening performance

In this study, with varying the pitch difference α, the prevailing torque necessary for the nut rotation before the nut touching the clamped body was measured experimentally. The bolt axial force was investigated in relation to the prevailing torque. The loosening experiment was conducted under a series of pitch differences. Then, the finite element analyses were applied to investigate the bolt axial force between nut threads as well as the deformation at the bolt and nut threads. The conclusions can be summarized as follows:

(1) It is found that the large value of α may provide large prevailing torque that causes anti-loosening performance although too large α may deteriorate the bolt clamping ability.

(2) Considering both the anti-loosening performance and the clamping ability, $\alpha=\alpha_{middle}$ is found to be the most desirable pitch difference. This is because that the nuts did not drop for $\alpha=\alpha_{middle}$ without losing clamping ability.

(3) The anti-loosening experiment shows the nuts did not drop for $\alpha=\alpha_{large}$ also but clamping ability is deteriorated. The FEM analyses show that for $\alpha=\alpha_{verylarge}$ the large plastic deformation happens at nut threads.

For bolt-nut connections having pitch difference, beside the anti-loosening performance and fatigue life improvement, the low cost is also an advantage. Most of the special bolt-nuts have either more components or very special geometry, leading to a complex manufacture process and a high cost which is usually more than 3 times of the normal bolt-nut. The suggested nut in this study can be manufactured as the same way as the normal nut, and the cost is predicted to be about 1.5 times of the normal nut considering the modification of thread tap as well as the checking procedure on the pitch difference.

6.3 Suggestions for future work

The main goal of this study is to find out a suitable pitch difference in order to improve both anti-loosening effect and fatigue life. Figure 6.1 shows a schematic illustration of the fatigue life improvement and anti-loosening improvement by varying the pitch difference when the results of α=0 are regarded as the reference level. On one hand, to improve the fatigue life, the most desirable pitch difference may be close to α_{small}. On the other hand, to improve the anti-loosening performance, the most desirable pitch difference should be larger than α_{middle} and close to α_{large}, although the nut locking phenomenon may happen if α is over $\alpha_{\text{verylarge}}$. Therefore, a suitable range for α can be indicated as shown in Figure 6.1.

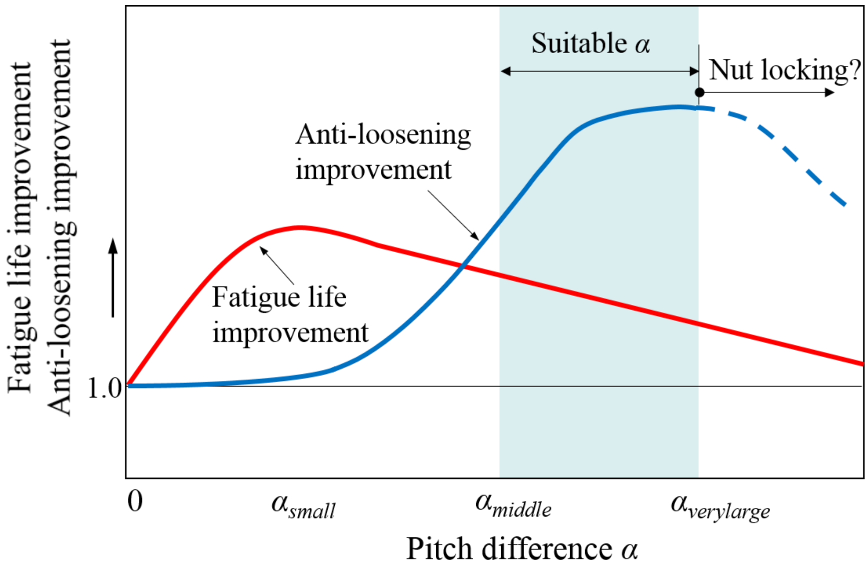


Figure 6.1: Schematic illustration of the fatigue life improvement and anti-loosening improvement

The present results have found that the bolt-nut connections having pitch difference can realize the fatigue life improvement and the anti-loosening performance successfully. Respect to the fatigue life improvement alone, the present results are not as remarkable as some other techniques such as pre-tensioning bolt, which can improve the fatigue limit by 25%. In this study, since the fatigue limit has not been improved yet, as shown in Figure 5.4, the fatigue strength improvement of this type of bolt-nut remains to be studied in a further step.

Appendix A: The results for chamfered model

Figure A1 shows the chamfered model replicating the actual geometry in Figure 5.10. Figure A2 shows the results of the chamfered model in Figure 5.10 in comparison with the results of the complete thread model. It is seen that because of no contact at No.8 thread in the chamfered model, average stress σ_m and stress amplitude σ_a increase in the chamfered model except at No.1 thread. Since the rigidity of the No.1 thread decreases in the chamfered model, the stress at No.1 thread does not change very much.

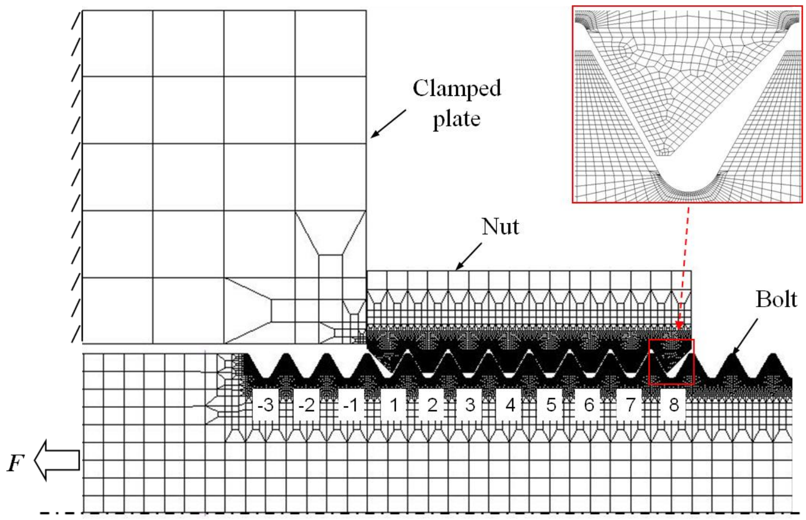


Figure A1: Axisymmetric finite element mesh for chamfered thread model

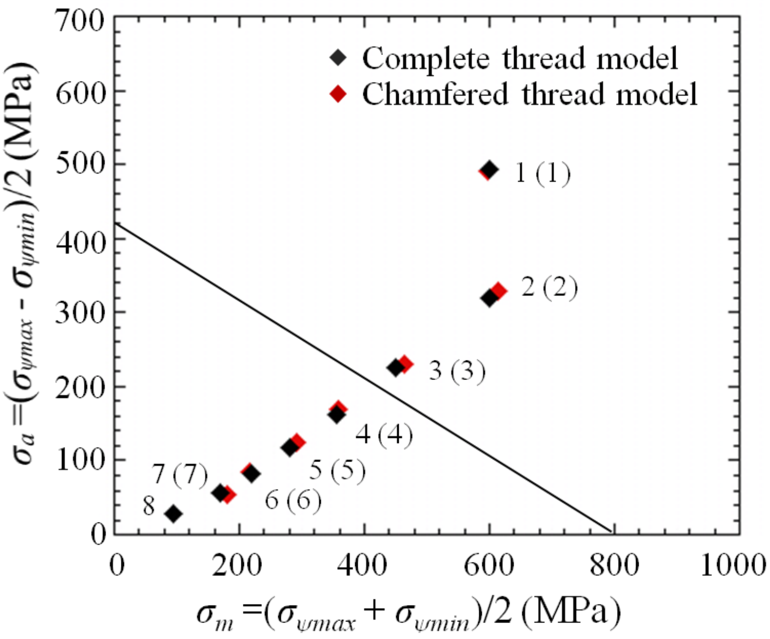


Figure A2: Comparison between the results of chamfered thread model and complete thread model when α=0 and σ_a=100 MPa

Appendix B: Effect of surface roughness on the fatigue strength of bolt

Table B1 [46] shows the effect of processing method and surface roughness on the fatigue strength of bolt. It is found that for the same processing method, the effect of surface roughness on the fatigue strength is small. With the same surface roughness conditions, the rolling thread has higher fatigue strength than the grinding thread.

Table B1 Effect of processing method and surface roughness on the fatigue strength of bolt

<table>
<tr><th rowspan="3">Bolt</th><th colspan="2" rowspan="2">Processing method</th><th colspan="4">Bolt materials</th></tr>
<tr><th colspan="2">Alloy steel</th><th colspan="2">Carbon steel</th></tr>
<tr><th>Method</th><th>Condition</th><th>Surface Roughness Ra in μm</th><th>Fatigue limit N/mm²</th><th>Surface Roughness Ra in μm</th><th>Fatigue limit N/mm²</th></tr>
<tr><td rowspan="4">M12× 1.5</td><td rowspan="2">Grinding</td><td>0.05mm</td><td>0.06-0.1</td><td>90</td><td>0.06-0.1</td><td>70</td></tr>
<tr><td>0.4mm</td><td>0.2-1.6</td><td>70</td><td>0.2-1.6</td><td>60</td></tr>
<tr><td rowspan="2">Rolling</td><td>Type 1</td><td>0.025-0.06</td><td>130</td><td>0.025-0.6</td><td>95</td></tr>
<tr><td>Type 2</td><td>0.06-0.1</td><td>120</td><td>0.2-1.6</td><td>95</td></tr>
</table>

Appendix C: Predicted *S-N* curves for the first fatigue experiment

In Chapter 3, a similar fatigue experimental result [2] was referred and the the *S-N* curves for α=0, $\alpha=\alpha_{verysmall}$ and $\alpha=\alpha_{small}$ were predicted. Here, by using the data obtained in Chapter 5, the slope of the *S-N* curves is reconsidered. Figure C1 shows the predicted *S-N* curves for the first fatigue experiment in Chapter 3 with considering the data obtained in Chapter 5. The Miner's rule is also applied to calculate the equivalent fatigue life for $\alpha=\alpha_{small}$.

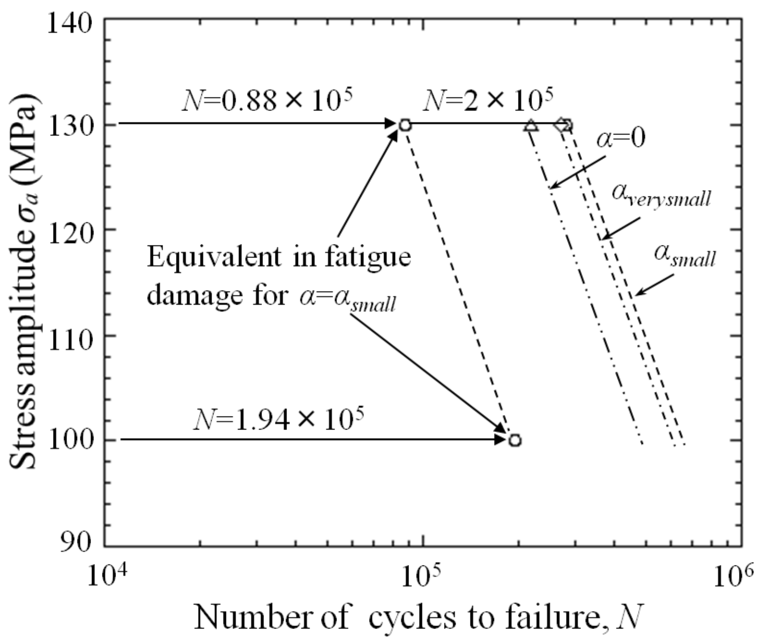


Figure C1: Predicted *S-N* curves in Chapter 3 by referring the data in Chapter 5

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List of publications

A.1 Peer reviewed journal publications included in Science Citation Index

1. **Chen, X.**, Noda, N. A., Wahab, M. A., Sano, Y., Maruyama, H., Wang, H., Fujisawa, R. and Takase, Y., (2015). Fatigue Life Improvement by Slight Pitch Difference in Bolt-Nut Connections. *Journal of the Chinese Society of Mechanical Engineers* (Accepted)

2. Noda, N. A., **Chen, X.**, Sano, Y., Wahab, M. A., Maruyama, H., Fujisawa, R. and Takase, Y., (2015). Effect of Pitch Difference between the Bolt-Nut Connections upon the Anti-Loosening Performance and Fatigue Life. *The Journal of Strain Analysis for Engineering Design* (Under review)

3. **Chen, X.**, Noda, N. A., Wahab, M. A., Akaishi, Y. I., Sano, Y., Takase, Y. and Fekete, G., (2015). Fatigue Failure Analysis in Bolt-Nut Connection Having Slight Pitch Difference Using Experiments and Finite Element Method. *Acta Polytechnica Hungarica* (Under review)

A.2 Peer reviewed journal publications not included in Science Citation Index

1. Akaishi, Y. I., **Chen, X.**, Yu, Y., Tamasaki, H., Noda, N. A., Sano, Y. and Takase, Y., (2013). Fatigue Strength Analysis for Bolts and Nuts Which Have Slightly Different Pitches Considering Clearance, *Transactions of Society of Automotive Engineers of Japan*, 44(4): 1111-1117 (In Japanese)

2. Noda, N. A., Sano, Y., Takase, Y., **Chen, X.**, Maruyama, H., Wang, H. and Fujisawa, R., (2015). Anti-Loosening Performance of Special Bolts and Nuts Having Enhanced Fatigue Life by Introducing Pitch Difference, *Transactions of Society of Automotive Engineers of Japan*, 46(1): 121-126 (In Japanese)

3. Noda, N. A., Sano, Y., **Chen, X.**, Maruyama, H., Wang, H., Fujisawa, R. and Takase, Y., (2015). Fatigue strength for bolts and nuts having slight pitch difference considering incomplete threads of nut. *Transactions of the JSME*, 81(831): 1-13 (In Japanese)

C.1 Publications in conference proceedings

1. **Chen, X.**, Akaishi, Y. I., Yu, Y., Tamasaki, H., Noda, N. A., Sano, Y. and Takase, Y., (2012). Fatigue Strength of Bolts and Nuts Which Have Slightly Different Pitches. *Proceedings of Asia Pacific Conference on Fracture and Strength-Mechanics and Materials 2012*. Busan, Korea

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