Fatigue Characteristics of Bolted Joints under Transverse Vibration

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Abstract.
Loosening-fatigue tests under transverse vibration have been performed to reveal the effect of bolt size in a bolted joint subjected to transverse vibration. Commercial hexagon head bolts in three sizes; M8, M10 and M12, were used in the experiment. Size effect has been verified by means of estimating the nominal stresses at the first thread root using test bolts with attached strain gages. Differences in fatigue characteristics between a bolted joint and a bolt/nut assembly have also been investigated. The results showed that the actual fatigue limit decreased with an increase in bolt size although the larger size bolt has a higher apparent fatigue limit, the threshold of the amplitude of transverse vibration force. The decreasing ratio was larger than the decreasing ratio in the axial fatigue of a bolted joint. It was confirmed that the fatigue characteristic under transverse vibration shows a marked size effect. The results also showed that the threshold of the amplitude of transverse vibration corresponding to the apparent fatigue limit of the bolt/nut assembly was higher than that of the bolted joint.

Introduction

Accidents due to fatigue failure and self-loosening of bolted joints are still occurring. The recent fatigue failure of wheel bolts on trucks and trains is an example. Fatigue failure and self-loosening of bolted joints are the two most important problems for vehicles that are subjected to vibration loading.

Fatigue and self-loosening characteristics depend greatly on the loading condition to which a bolted joint is subjected. Hence these must be investigated for each loading condition respectively. Fatigue and self-loosening characteristics of a bolted joint due to axial vibration have been already discussed extensively [1-3]. However fatigue failure due to transverse vibration has not been sufficiently discussed as yet although the characteristics of loosening have been fully discussed [4-6]. In our previous study, loosening-fatigue tests of bolted joints under small transverse vibration were implemented to reveal loosening-fatigue mechanisms in the high cycle region [7,8]. The results showed that if a bolt has loosened within 10^3~10^4 vibration cycles, damage such as crack nucleation at the root of the first thread is not observed, and loosening is due to bolt rotation. However, if loosening does not occur until approximately 10^5~10^6 cycles, a crack is observed at the root of the first thread in all the experiments. Loosening occurs due to loss of bolt stiffness caused by crack nucleation and propagation which then leads to bolt rotation. The results also showed that the fatigue life of a bolted joint under transverse vibration depends on the amplitude of the transverse vibration force and does not depend on the initial clamping force.

Bolts and fasteners are typical notched specimens. And the bolted joint under transverse vibration is subjected to cyclic bending moment. Therefore it is considered that the fatigue characteristic of a bolted joint under transverse vibration shows a marked size effect. In this study, loosening-fatigue tests under transverse vibration have been performed to reveal the size effect of bolted joints under transverse vibration. Commercial hexagon head bolts in three sizes; M8, M10 and M12, were used. Size effect has been verified by means of estimating the nominal stresses at the first thread root using test bolts with attached strain gages. Also differences in fatigue characteristics between a bolted joint

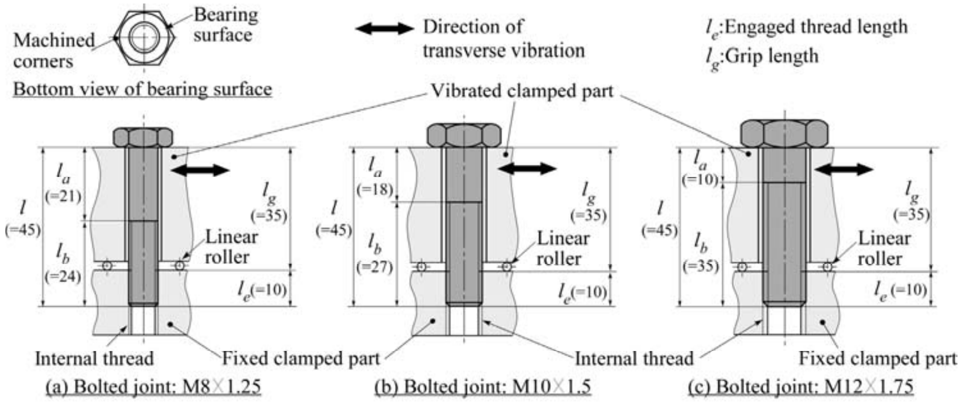

Fig.1 Specifications and tightening situation of the test bolts for investigation of size effect

and a bolt/nut assembly have been investigated using a commercial hexagon head bolt, M10. In this study, “bolted joint” indicates a joint tightened by a bolt and an internal thread, and “bolt/nut assembly” indicates a joint tightened by a bolt and a nut.

Experiment

Test Bolt. Figure 1 shows schematic illustrations of the tightening situations of the test bolts for investigation of size effect. These bolts were commercial hexagon head bolts; M8, M10 and M12; property class 8.8. The nominal length of the test bolt was l=45mm. The complete thread length for M8 was l_b=24mm, the M10 bolt was l_b=27mm and the M12 bolt was l_b=35mm. The grip length of the bolted joints, i.e. the length between the bearing surface and the first engaging thread point, was l_g=35mm for each bolt. The engaging thread length, i.e. the thread length between the first engaging thread point and the bolt tip, was l_e=10mm for each bolt. The corners of the bearing surface of all bolts were machined with a lathe to create a flat contact surface as shown in Fig.1.

Figure 2 shows schematic illustrations of the tightening situations of test bolts for investigation of fatigue characteristics of a bolt/nut assembly and a bolted joint. These bolts were commercial hexagon head bolt, M10. The bolt property class was 10.9. The nominal length of the test bolt for the bolt/nut assembly was l=80mm. The nominal length of the test bolt for the bolted joint was l=65mm. This test bolt for the bolted joint was made by means of cutting the test bolt for the bolt/nut assembly with a lathe. The complete thread length of the bolt for the bolt/nut assembly was l_b=28mm, and the complete thread length of the bolt for the bolted joint was l_b=13mm. The grip length of both the bolted joint and the bolt/nut assembly was l_g=55 mm. The engaging thread length of the bolt/nut assembly was l_e=8 mm because of the nut height. In order to make the situation for the bolt/nut assembly correspond as closely as possible, the

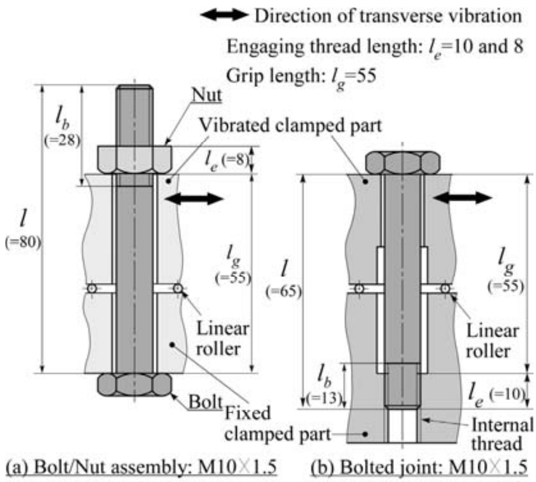

Fig.2 Specifications and tightening situation of a Bolt/Nut assembly and a bolted joint

engaging thread length of the bolted joint was determined at le=10mm. The corners of the bearing surface of all the bolts were machined with a lathe to create a flat contact surface.

Experimental Apparatus. Figure 3 shows the schematic illustrations of the apparatus for the loosening-fatigue tests to determine size effect. The apparatus was designed to simulate a two-plate structure for the loosening-fatigue test. In Fig.3, the test bolts were tightened into the internal thread adaptor, with its rotation fixed on the base plate, through a bearing surface part and a load cell for measuring the clamping force F located in the center of the apparatus. The details of the tightening situation of the bolted joint are shown in Fig.1. The lower clamped part including the load cell for clamping force measurements was fixed on the base plate and the upper clamped part was vibrated with an air vibrator. This transverse vibration force $\Delta P_t/2$ was controlled by air pressure applied to the air vibrator, with a constant amplitude load. The frequency of the vibration depended on the air pressure, and varied from 50 Hz to 60 Hz.

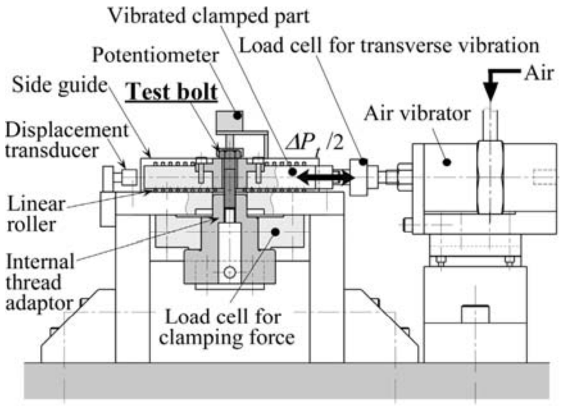

Fig.3 The experimental apparatus for loosening and fatigue test

The contact surfaces between the two clamped parts were hardened and two lubricated linear rollers were placed between these parts to reduce the friction. Frictional losses were measured to be less than 1% of the transverse vibration force and were ignored in this study. The displacement δ of the vibrated clamped part was measured by a non-contact displacement transducer, and bolt rotation θ was measured by a potentiometer during the loosening-fatigue test. The internal thread adaptor plate part was made of a medium carbon steel JIS S55C and manufactured by tapping. The bearing surface part was also made of a medium carbon steel JIS S55C.

In the loosening-fatigue tests to investigate fatigue characteristics of a bolt/nut assembly and a bolted joint, the same apparatus was used except for the internal thread adaptor. In the experiment, a new bearing surface adaptor which is similar to shape of the internal thread adaptor was used instead of the internal thread adaptor because the test bolt was inserted from the bottom of the apparatus through the load cell. Then the bolt was tightened by the nut from the top side as shown in Fig.2(a). The bolt rotation was fixed by attaching a socket wrench on the bolt head.

Experimental Procedure. In the loosening-fatigue tests, the test was started after the test bolt was tightened with a wrench to F_i=15kN. It was stopped when the clamping force F reached zero or the loading cycle exceeded 1×10^7 cycles. In all experiments, the thread surface and bearing surface were lubricated with MoS_2 grease. A new test bolt was used in each experiment. Before each experiment, a tap was inserted into the internal thread or the nut. If an abnormality of the internal thread was detected, the adaptor was replaced with a new one. Otherwise, the adaptor was used repeatedly in the experiments. The bearing surface contacting the bolt was polished with #600 sand paper before each experiment. A washer was not used in the experiments. For the experiment of the bolt/nut assembly, a new test bolt and a new test nut were used in each experiment.

In the loosening-fatigue test, the loosening-fatigue life N_f of the bolted joint depends on the amplitude $\Delta P_t/2$ of the transverse vibration force [7]. Hence there is a threshold of $\Delta P_t/2$ at which the bolt does not loosen or come apart due to fatigue failure. The threshold $\Delta P_{tw}/2$ was found by gradually reducing $\Delta P_t/2$ from 0.65 kN in each experiment. The loosening-fatigue limits σ_{tw} at each $\Delta P_{tw}/2$ were measured using the strain gages attached to the test bolt. Fig.4 shows schematic illustrations of the test bolts with strain gages. Fig.4 (a) shows an M8 bolt measured for stress. Fig.4 (b) shows an

M10 bolt and Fig.4 (c) shows an M12 bolt. These bolts were fitted with strain gages in order to measure the bending moment M applied to the bolts due to transverse vibration. In these bolts, the bolt threads which do not engage with the internal threads were removed by a lathe in order to attach the strain gages. The wires of the strain gages were threaded out through the inside holes.

When the bolted joint or the bolt/nut assembly is subjected to transverse vibration, the bending moment which the bolt receives is expressed as follows.

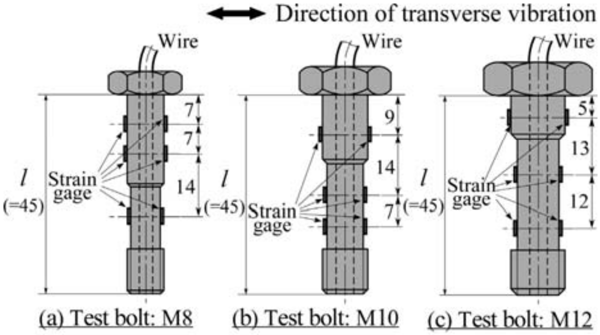

Fig.4 Schematic illustration of test bolts which were attached strain gages to measure the bending moment due to the transverse vibration force

$$M = P_t \cdot (l_g - x) - M_B \tag{1}$$

where P_t is the transverse force and l_g is the grip length. x is the position on the bolt axis from the bearing surface of the bolt. M_B is the moment to constrain the engaging bolt thread.

The nominal stress at the root of the first thread can be easily calculated using the following equation.

$$\sigma_t = \frac{M_B}{Z} \tag{2}$$

where Z is the section modulus at the root of the first thread of the bolt.

After the bending moment distribution was measured for each $\Delta P_t/2$, the nominal stresses σ_t at the root of the first thread was calculated using Eq.(2).

Incidentally, in the experiments to investigate the fatigue characteristics of the bolt/nut assembly and the bolted joint, the loosening-fatigue limits σ_{tw} were not estimated because it was very difficult to make a hole in the bolt. Hence $\Delta P_{tw}/2$ was used as the apparent loosening-fatigue limit instead of σ_{tw}.

Experimental Result and Discussion

Size Effect of Bolted Joint under Transverse Vibration. Figure 5 shows examples of behaviors of the clamping force F in the loosening- fatigue tests for M8. In Fig. 5, the abscissa is the number of cycles N, the ordinate is the clamping force F. Four behaviors of the clamping force F of the bolted joints were subjected to each amplitude $\Delta P_t/2$ of transverse vibration force.

It can be seen from Fig.5 that the loosening-fatigue life N_f depends on the amplitude $\Delta P_t/2$ of transverse vibration force, and all the bolts except for the one subjected to $\Delta P_t/2$=0.26 kN were loosened before N=1$\times$10^7 cycles. Therefore there is a threshold $\Delta P_{tw}/2$ between 0.26 kN and 0.30 kN. Two fatigue cracks from the opposite side were observed at the root of the first thread in the test bolt which loosened by N=1$\times$10^7 cycles. As soon as the clamping force F started decreasing, it went down sharply. It can be seen from these results that the fatigue cracks propagate drastically after the cracks have nucleated.

Figure 6 shows the relationships between $\Delta P_t/2$ and the loosening-fatigue life N_f for each bolt size. In Fig.6, the abscissa is the loosening-fatigue life N_f, the ordinate is the amplitude of the transverse vibration force $\Delta P_t/2$. The symbols ●, ■ and ◆ show the result of a loosened bolt with crack nucleation at the root of the first thread for bolts sized M8, M10 and M12 respectively. The symbols △→, □→ and ○→ show run-out data for the each bolt size. The highest amplitudes of transverse vibration force $\Delta P_t/2$ of the run-out data become the threshold $\Delta P_{tw}/2$ of amplitude of transverse vibration force corresponding to the apparent loosening-fatigue limits.

It can be seen from Fig.6 that loosening-fatigue life N_f significantly depends on $\Delta P_t/2$ for all bolt sizes. $\Delta P_{tw}/2$ and N_f is drastically different according to bolt size. Undoubtedly, the larger size bolt has a markedly higher $\Delta P_{tw}/2$. Next, we will discuss size effect in fatigue under transverse vibration using the loosening- fatigue limit σ_{tw}.

Table 1 shows the fatigue characteristics for each bolt size. In Table 1, β is the notch factor in axial fatigue of the bolted joint. σ_{aw} is the calculated axial fatigue limit for each bolt size. The axial fatigue limits σ_{aw} were estimated using a following equation proposed by Yoshimoto et al [9].

$$\sigma_{aw} = \frac{\sigma_{w0}}{\beta} \cdot \frac{(\sigma_T - \sigma_{0.2})}{(\sigma_T - \sigma_{w0})} \tag{3}$$

where σ_{w0} is the fatigue strength of a smooth specimen of a bolt material, σ_T is the true ultimate strength and $\sigma_{0.2}$ is the proof stress of a bolt material. This equation can be obtained from the Goodman Diagram of a bolted joint. For a property class 8.8 bolt, the parameters in Eq.(3) were σ_{w0}=290, σ_T=1370, $\sigma_{0.2}$=640 [10]. The transverse loosening-fatigue limit σ_{tw} in Table 1 was estimated using the test bolt attached the strain gages from $\Delta P_{tw}/2$.

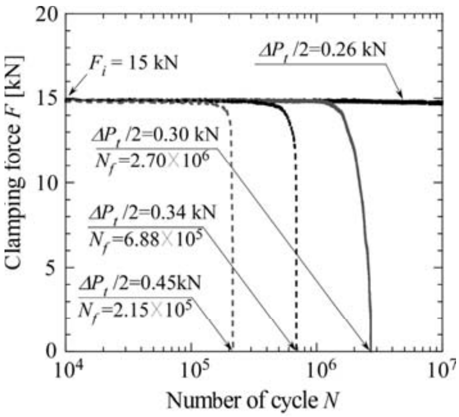

Fig.5 Variations of clamping force of bolted joints under transverse vibration (M8)

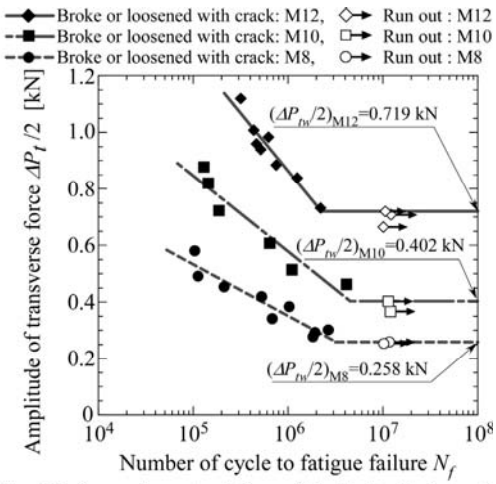

Fig.6 Fatigue characteristics of the bolted joints of each bolt size under transverse vibration

Table 1 Fatigue limits for each bolt size

Bolt size	Notch factor in axial fatigue β	Calculated axial fatigue limit σ_{aw} [MPa]	Apparent transverse loosening-fatigue limit $\Delta P_{tw}/2$ [kN]	Transverse loosening-fatigue limit σ_{tw} [MPa]
M8	3.23	60.7	0.258	142
M10	3.44	57.0	0.402	90.5
M12	3.58	54.8	0.719	66.8

It can be seen in Table 1 that both the axial fatigue limits σ_{aw} and the transverse fatigue limits σ_{tw} increase with a decrease in bolt size. Size effect under transverse vibration can be confirmed from these results. When comparing σ_{aw} and σ_{tw}, the increase ratio of σ_{tw} with a decrease in bolt size is larger than that of σ_{aw}. It is considered that this is caused by the stress gradient at the root of the first thread. In case of the axial fatigue of bolted joint, the stress gradient at the root of the first thread is generated by only the stress concentration. The stress gradient for the transverse fatigue of a bolted joint is generated by both the stress concentration and the cyclic bending condition. Therefore, it is considered that the increase ratio of σ_{tw} with a decrease in bolt size was larger than that of σ_{aw}.

Difference between Bolt/Nut Assembly and Bolted Joint. Figure 7 shows the relationships between $\Delta P_t/2$ and the loosening-fatigue life N_f of the bolt/nut assembly and the bolted joints. In Fig.7, the abscissa is the loosening-fatigue life N_f, and the ordinate is $\Delta P_t/2$. The symbol ● shows a loosened or broken bolt with crack nucleation of the bolt/nut assembly. The symbol ○➜ shows run-out data of the bolt/nut assembly. The symbol ■ shows a loosened or broken bolt with crack nucleation of the bolted joint. The symbol □➜ shows run-out data of the bolted joint.

It can be seen from Fig.7 that the threshold $\Delta P_{tw}/2$ of the amplitude of transverse vibration corresponding to the apparent loosening-fatigue limit of the bolt/nut assembly is slightly higher than that of the bolted joint. The loosening-fatigue lives N_f are almost the same for both cases. It is considered that the difference in $\Delta P_{tw}/2$ was caused by a difference in constraint strength at the engaging threads. Fig.8 shows the schematic illustrations and the bending moment diagrams of the deformed bolt/nut assembly and bolted joint due to transverse vibration. When the bolted joint and the bolt/nut assembly are exposed to

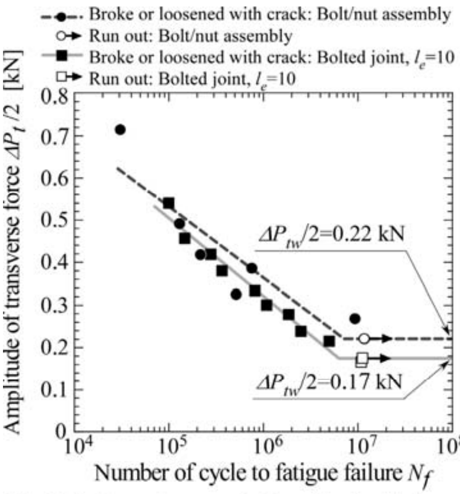

Fig.7 Fatigue characteristics of bolted joints and bolt/nut assembly under transverse vibration

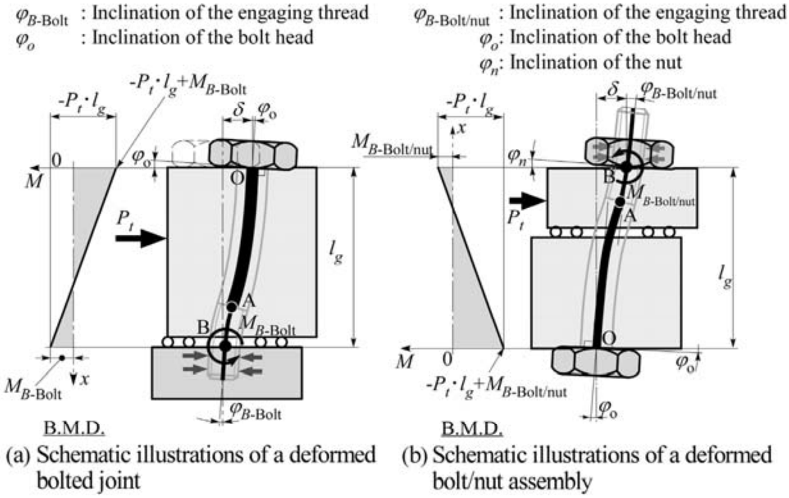

Fig.8 Schematic illustrations of a deformed bolted joint and a deformed bolt/nut joint

transverse vibration, these bolts deform as shown in Fig.8 because the bolts are bent by transverse force and simultaneously constrained by the engaging thread. Accordingly, each bending moment diagram is deformed as shown in Fig.8, and the bending moment is expressed by Eq.(1).

It can be seen from Eq.(1) that the bending moment M is directly proportional to the grip length. From Eq.(2), the nominal stress σ_t at the root of the first thread, the fatigue fracture origin, is determined by M_B.

When the bolt/nut assembly receives transverse vibration, the bolt at the engaging threads portion can slip slightly at the engaging thread surfaces and incline due to elastic deformation of the threads. Furthermore, the bolt at the engaging threads portion can incline still more because the nut inclines simultaneously due to elastic deformation of the engaging threads and the bearing surfaces. But in the case of the bolted joint, the inclination of the bolt at the engaging threads portion occurs due only to the slippage at the engaging thread surfaces and elastic deformation of the threads because the internal thread part cannot incline like the nut. Hence the inclination of the engaging thread portion of the bolt of the bolt/nut assembly is larger than that of the bolted joint. As the result, the constrained moment at the engaging threads portion of the bolt/nut assembly is smaller than that of the bolted joint, and the nominal stress is small.

Using this information, we can discuss whether a bolted joint or a bolt/nut assembly should be used. The advantages of a bolted joint are different from that of a bolt/nut assembly. So we should choose them according to their most effective usage. In the loosening-fatigue test under transverse vibration, the bolt/nut assembly performed slightly better than the bolted joint. These tests were achieved with the same grip length. Although a factor which directly influences the nominal stress at the root of the first thread is the bending moment M_B at the root of the first thread, M_B increases if the grip length is large. Therefore a shorter grip length is better for fatigue under transverse vibration. For instance, if a two-plate structure is fixed with a bolted joint, an internal thread is made by tapping in one side plate instead of using a nut. In this case, the grip length is the same as the thickness the other plate. If the bolt/nut assembly is used the grip length becomes the thickness of the two plates. Hence the grip length of the bolt/nut assembly is greater than that of the bolted joint in most cases. Consequently, if clamped parts are subjected to transverse vibration, bolted joints are more advantageous than bolt/nut assemblies.

Conclusion

Loosening-fatigue tests under transverse vibration were performed to reveal the size effect of a bolted joint under transverse vibration. Also differences in fatigue characteristics between a bolted joint and a bolt/nut assembly have been investigated.

The main conclusions obtained in this study are summarized as follows.

1. In loosening-fatigue characteristics of a bolted joint under transverse vibration, the threshold of the amplitude of transverse vibration corresponding to the apparent loosening-fatigue limit increases with an increase in bolt size. However the actual loosening-fatigue limits decrease with an increase in bolt size. It was confirmed that the fatigue characteristics under transverse vibration showed a marked size effect.
2. The increase ratio of the loosening-fatigue limit with a decrease in bolt size was larger than that of the calculated axial fatigue limit. The reason is that the stress gradient at the root of the first thread is generated by both the stress concentration and the cyclic bending condition due to the transverse fatigue of the bolted joint.
3. In comparing the fatigue characteristics of the bolt/nut assembly and the bolted joint with the same tightening condition, it was discovered that the threshold of the amplitude of transverse vibration corresponding to the apparent fatigue limit of the bolt/nut assembly was higher than that of the bolted joint.

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