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Working Load Analysis and Strength Estimation on Bolted Joints during Actual Machine Operation

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Abstract

A working load analysis was carried out on a bolted joint of an axle structure of construction equipment (wheel loader) under actual machine operation. The bolt was set in a final reduction system using a planetary gear mechanism. The working load was measured by a stud with strain gages. From the measurement of the bolt in actual products, the load and stress frequency diagrams were obtained. The estimation technique for the fatigue strength was shown. It was discussed that these techniques can be applied for the simultaneous multi-design.

Key words: Fixing Element, Bolted Joint, Working Load, Load Analysis, Load Frequency, Fatigue Strength, Construction Equipment, Simultaneous Multi-design

1. Introduction

In such heavy duty machinery as industrial and construction machinery, very severe loads are applied to bolted joints which fasten strength members, and therefore full attention must be paid to their strength and self-loosening.

Recently, successive changes must be made in product models to respond to quickly changing market needs, or else many models must be developed at the same time, and efficient product development is necessary. The method of concurrent multiple design is frequently introduced now to many component design processes to cope with this necessity. This is also the case in designing bolted joints used to fasten strength members. This method is carried out as follows: sufficiently detailed measurement, analysis and strength estimation of applied loads are carried out with a prototype machine, general and fundamental technical design data which enable selection of the optimum size are obtained based on the strength profile of the bolts derived from test results, and based on these data, the reliability of bolts in a new model can be economically confirmed in a short time merely by design estimation and simple stress measurement.

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The main types of bolt troubles are fatigue fracture and self-loosening, and their causes are design-based faults and insufficient fastening torque. Much research into the effect of self-loosening prevention measures has been carried out using self-loosening testers. In our first report(1), the authors proposed an absolute estimation method for bolt-loosening during actual machine operation, which enabled prediction, in an early stage of machine development, of the decrease in axial tension after a long period of machine operation, comparing these predictions with measurements in actual machines.

Several design analyses on loads applied to bolted joints have been carried out(2)(3), but there has been no systematically arranged study of measuring methods and analyses of actually applied loads to bolted joints, though this is very important information for realizing the above mentioned concurrent multiple design concept.

In this report, a method for analyzing loads applied to bolted joints was investigated, including the case of multiple loads, and a measuring method using strain gauges and equations for making analysis are proposed. Taking as an example the base carrier of a construction machine, an analysis of loads actually applied to bolted joints during machine operation was carried out, and the effectiveness of the proposed method was proved by showing that accurate fatigue strength estimation was possible using the posited relation between loads and stresses.

2. The analysis on loads applied to bolted joints

In designing bolted joints, the structure is generally so devised that the load condition of bolts is simplified using knock-pins or spigot joints to join with a flange joint, so that either shearing force or tangential force can be neglected. Therefore, in bolted joints to which axial force, bending moment and torsion torque are applied, it is possible to measure and analyze the loads according to the method described below.

Fig.1 shows the positions of strain gauges bonded to a bolt shank for measuring loads as well as the relation of those loads. Each load applied to a bolt joint is assumed to operate along or around the bolt axis. As shown in the figure, the strain gauges at measuring points are arranged so that: 3 orthogonal axis strain gauges are bonded at A and C, a central gauge coincides with the bolt axis, and uni-axial strain gauges bonded at B and D are parallel with the bolt axis. The strains measured

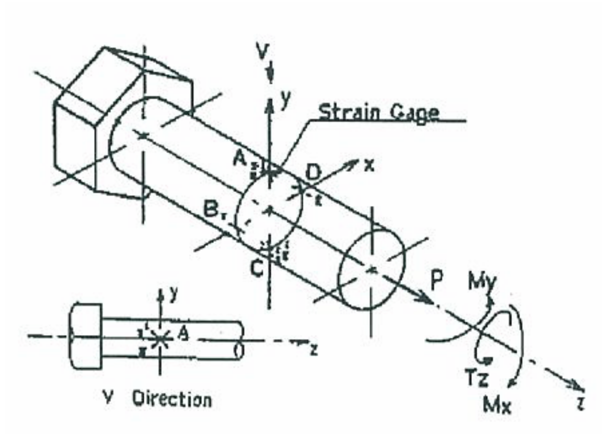

Fig.1 Relation between the measuring locations of strain gauges and loads

by the orthogonal 3 axis strain gauges Ⅰ, Ⅱ, Ⅲ at A for example are denoted as $\varepsilon_{A\mathrm{I}}$, $\varepsilon_{A\mathrm{II}}$, $\varepsilon_{A\mathrm{III}}$ respectively, and the strains measured by uni-axial strain gauges at B and D are denoted as $\varepsilon_{B\mathrm{II}}$, $\varepsilon_{D\mathrm{II}}$ respectively. Then, the loads applied to the bolt-joint are characterized as follows.

$$P = \frac{E \cdot Ar}{2}\left(\varepsilon_{A\mathrm{II}} + \varepsilon_{C\mathrm{II}}\right) \cdots\cdots\cdots (1)$$

$$Mx = \frac{E \cdot Zb}{2}\left(\varepsilon_{A\mathrm{II}} - \varepsilon_{C\mathrm{II}}\right) \cdots\cdots\cdots (2)$$

$$My = \frac{E \cdot Zb}{2}\left(\varepsilon_{B\mathrm{II}} - \varepsilon_{D\mathrm{II}}\right) \cdots\cdots\cdots (3)$$

$$Tz = \frac{G \cdot Zp}{2}\left\{\left(\varepsilon_{A\mathrm{III}} - \varepsilon_{A\mathrm{I}}\right) + \left(\varepsilon_{C\mathrm{III}} - \varepsilon_{C\mathrm{I}}\right)\right\} \cdots (4)$$

Here,

P : Axial force at the measured point (Axial tension)

Mx: Bending moment around x axis at the measured point

My: Bending moment around y axis at the measured point

Tz: Torsion torque around z axis at the measured point

Ar: Sectional area of bolt shank at the measured point

Zb: Section modulus of bolt shank

Zp: Polar modulus of section of bolt shank

E : Modulus of longitudinal elasticity of bolt shank

G : Modulus of transverse elasticity of bolt shank

The composite bending moment of bending moments Mx and My and the direction of the operational axis of this composite moment are shown as below.

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$$Me = \sqrt{Mx^2 + My^2} = \frac{E \cdot Zb}{2}\sqrt{(\epsilon_{A\,II} - \epsilon_{C\,II})^2 + (\epsilon_{B\,II} - \epsilon_{D\,II})^2} \quad \cdots (5)$$

$$\tan \alpha_M = \frac{My}{Mx} = \frac{\epsilon_{B\,II} - \epsilon_{D\,II}}{\epsilon_{A\,II} - \epsilon_{C\,II}} \quad \cdots\cdots\cdots (6)$$

Here,

Me: Composite bending moment

α_M: Angle from x axis to operational axis of composite bending moment (in counter clockwise direction)

3. Experimental verification of methods for load measuring and analysis

It was verified by a simple experiment whether multiple applied loads to a bolt joint could be analyzed and separately evaluated accurately by analytical equations deriving loads from measured strains.

3.1 The case of an axial force and a bending moment

The axial force and bending moment were measured by attaching strain gauges to a stepped stud constructed for load measurement at its center as shown in Fig.1, and putting this in the vertically symmetrical tensile testing machine shown in Fig.2-a). This stud is also used in load measurement during actual operation of a machine as mentioned later. A bending moment is applied by

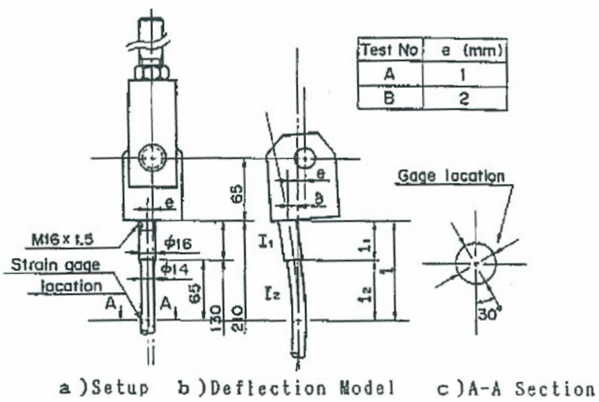


Fig.2 Experimental apparatus for axial tension and bending moment

setting eccentricity “e” between the axes of the tensile testing machine and the stud. This experimental apparatus can be configured as shown in Fig.2-b), where an eccentric tensile load is applied to a rod. Neglecting the influence of the supporting part of the stud, the bending moment Mc can be expressed by the equations below.

$$Mc = P(e - \delta) \cdots\cdots\cdots\cdots\cdots (7)$$

$$\delta = \frac{\alpha_1 . e}{\alpha_1 \cos\alpha_1 l_1 . \cos \alpha_2 l_2 - \alpha_2 \sin\alpha_1 l_1 \cdot \sin \alpha_2 l_2} - e \cdots (8)$$

Here,

$$\alpha_1 = \frac{P}{EI_1}, \quad \alpha_2 = \frac{P}{EI_2}$$

where
I_1 is the geometrical moment of inertia of the supporting part of the stud, and
I_2 is the geometrical moment of inertia of the measuring part of the stud.

Fig.3 compares the composite bending moment Me, calculated with Equation (5) using the values calculated by the above equations based on the strain gauge measurements, to the results calculated with Equation (7). They agree each with other in overall trend, but there is a rather large difference between particular values. This is considered to be due to inaccuracy in the finishing of the test piece and the back-lash of the threads.

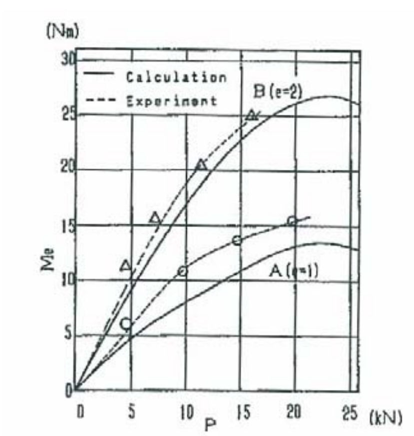


Fig.3 Calculated and Experimental results of bending moment

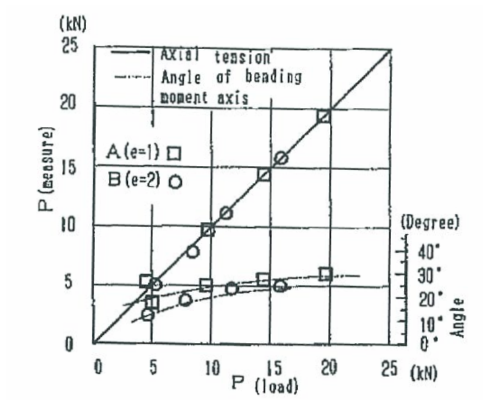


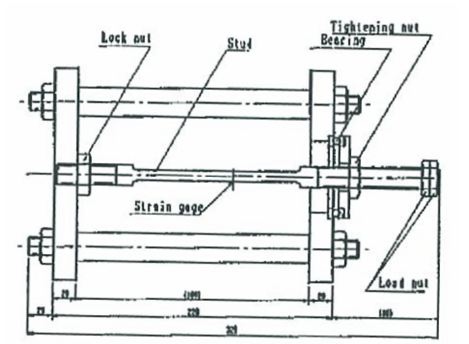
Fig.4 Axial tension and angle of bending moment axis

Fig.4 shows the measured axial tension and the angle of the bending moment axis. It is shown that the applied load can be measured with high accuracy by measuring axial tension. The accuracy of measurement of angle of bending moment axis α_M improves as the applied load is increased. This experiment was carried out for a stud set 30 degrees from the axis of the bonded strain gauges, as shown in Fig.2-c).

The above mentioned results show that this measurement method enables accurate analysis, separately evaluating axial tension and bending moment in the case where both are applied at the same time.

3.2 The case of an axial force and a torsion moment

It was investigated whether axial tension and torsion torque can be separately measured with accuracy in the case where both are applied at the same time, by measuring stress changes where axial tension is initially applied to a stud to which torsion torque is being



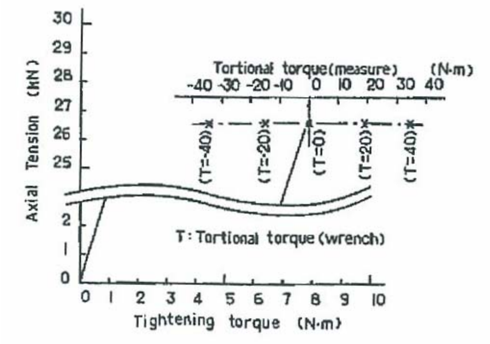


Fig.5 Experimental apparatus for axial tension and torsion torque

Fig.6 Experimental results of axial tension and torsion torque

applied, using the experimental apparatus shown in Fig.5. The experiment was done by adding torsion torque with a load nut after twisting a tightening nut with torque of 78.5 Nm.
（＊fig.5 では load nut とありますが、loading nut とすべきでは？）

Strain was measured at the central part of the stud in the same manner as the case of Fig.2. Fig.6 shows the experimental results; axial tension change with addition of torsion torque after tightening is scarcely observed at all. This means that this measurement method also enables analysis separating these two factors with enough accuracy in the case where axial tension and torsion torque are both applied. The same should hold for the case of a bending moment and a torsion torque applied at the same time, considering the relation between axial strain and shearing strain. The reason why the measured torsion torque from strain gauges in the stud is slightly lower than it should be, 85% of the actual loaded torsion torque, is likely that there were losses due to friction at the bearing.

4. Application to load measurement in an actual machine

The fundamental conception of measurement of a load applied to bolt joints has been described. It was confirmed by simple experiments that this method enables analysis of multiple separate applied loads with accuracy. In this chapter, the results applying this method to load measurement in an actual machine are described.

4.1 Test pieces and experimental conditions

Fig.7-a) shows an axle of a building construction machine (a wheel loader). The bolts here analyzed were setting bolts to fix a hub to a planetary gear type reduction gear central axis. As the hub is a rotating part, the bolts are considered to be applied with multiple loads such as torsion torque and axial tension. Fig.7-b) shows the dimensions of the setting bolts. In order to analyze loads applied to the bolts thus structured in a simple way, measurement was made of a stud with a cross section as shown in fig.8-a). Fig.8-b) shows the dimensions of the measuring stud.

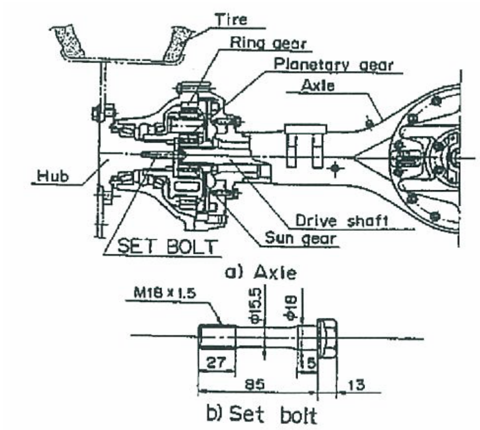


Fig.7 Tested structure and a set bolt

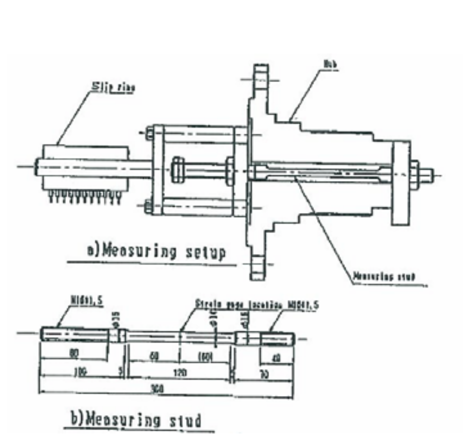


Fig.8 Load measuring apparatus and a measuring stud

Operation load measurement was carried out during a cycle of shoveling and dumping operations as shown in Fig.9, which is a typical working pattern for a wheel loader. The graph shows the measured pattern of load applied to tires etc. in one operation cycle. It is necessary to do analysis of fatigue strength to estimate data over a long period, and therefore loads during repeated cycles were measured for 30 minutes.

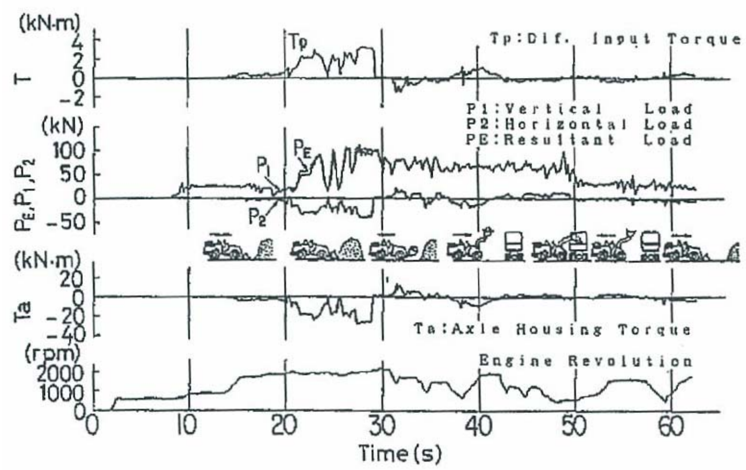


Fig.9 Load pattern (1 cycle) of the axle part during shoveling and dumping work

4.2 Load measurement during machine operation and load peak frequency of occurrence analysis

Load estimation is done with the equations (1)-(4) given in chapter 2, where shearing force and tangential force are assumed to be negligible. Measured loads were analyzed by a data processing device with a mini-computer.

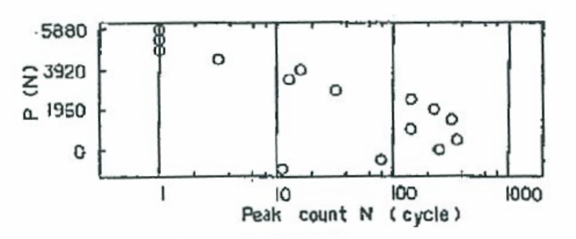


Fig.10 Load distribution (Axial tension)

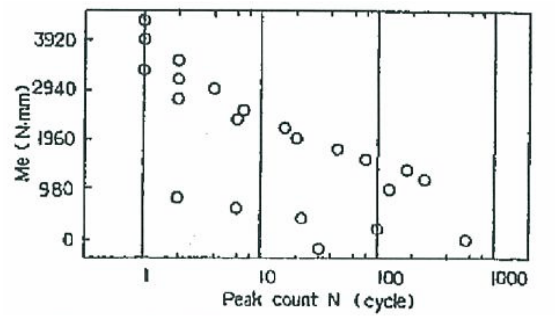


Fig.11 Load distribution (Bending moment)

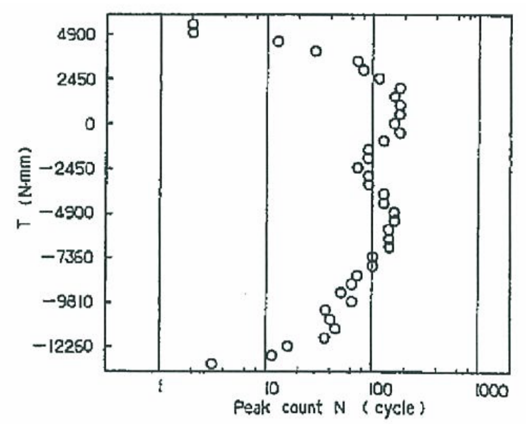


Fig.12 Load distribution (Torsion torque)

Figs.10, 11 and 12 give numbers of occurrences of load peaks of axial tension, bending moment and torsion torque, respectively, applied to the bolt joint. Both the load peak distributions of bending moment and of torsion torque seem to have two peaks. These are considered to result from superposition of respective load peaks during shoveling and traveling work and in forward and reverse travel.

4.3 The relation between load and stress

In this experiment, a load measuring stud was used instead of an actual bolt, therefore it is necessary to estimate the load and stress of an actual bolt from the measurements of the stud.

When an external force Wa is applied to the bolt-joint part of the load measuring stud, the relation between axial tension generated in the stud P_A and the external force is as follows.

$$P_A = \Phi_T \cdot Wa \quad \cdots\cdots\cdots\cdots\cdots (9)$$

Here,

Φ_T : Stiffness constant factor of the bolt-joint part of the load measuring stud

When the same external force Wa is applied to the bolt-joint part of the actual bolt, the axial tension of the actual bolt P can be expressed in the same manner, and therefore the conversion equation of axial tension is as follows.

$$P = \frac{\Phi_A}{\Phi_T} P_A \cdot \cdots\cdots\cdots\cdots\cdots (10)$$

Here,

Φ_A : Stiffness constant factor of the bolt-joint part of the actual bolt

Table 1 shows the calculated and experimentally determined values of stiffness constants and stiffness constant factors of bolt-joint parts of a load measuring stud and an actual bolt of prototype machine. As reported by Shibahara, Oda[5] and Sawa, Kumano[6], the stiffness constants that were experimentally determined are lower than those calculated with the standard equation.

Table 1 Calculated and experimental data of stiffness constant and stiffness constant factor

Term			Calculation	Experiment
Stud joint for measuring instrument	Stiffness constant	Stud (N/μm)	129.7	90.2
		Member (N/μm)	1280.7	1225.8
	Stiffness constant factor Φ_T		0.092	0.066
Bolted joint for prototype machine	Stiffness constant	Bolt (N/μm)	598.0	350.1
		Member (N/μm)	2370.3	2187.9
	Stiffness constant factor Φ_A		0.201	0.138
Stiffness constant ratio Φ_A / Φ_T			2.18	2.09

Bending moment of the bolts in a prototype machine Me' was also empirically characterized by the equation below for easier calculation.

$$Me = \frac{\Phi_A}{\Phi_T} Me' \cdot \cdots\cdots\cdots\cdots\cdots (11)$$

The load applied to the measured part as derived by the above mentioned method must in some cases be converted to the load applied to the weakest part of the bolt, but here the load thus introduced approximates the load applied to the weakest part of the bolt.

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The maximum stresses on the weakest part of the bolt are calculated from those loads as follows.

$$\sigma_{max} = \frac{P}{A_2} + \frac{Me}{Z_2} \cdots\cdots (12)$$

$$\tau_{max} = \frac{T_z \cdot d_2}{2 I_p} \cdots\cdots (13)$$

Here,

σ_{max} : Maximum axial stress at the section of weakest part of a bolt

τ_{max} : Maximum shear stress at the section of weakest part of bolt

A_2 : The area of the section of weakest part of bolt

Z_2 : Section modulus of weakest part of bolt

I_p : Polar moment of inertia of weakest part of a bolt

d_2 : Effective diameter of weakest part of bolt

Bolt fracture is generally characterized in accordance with shearing strain energy criteria. [(7)-(10)] In this report, when fatigue fracture of a bolt is investigated, the maximum stress on a bolt generated by actual loads applied to a bolt joint during machine operation is also considered to follow the shearing strain energy theory and to be calculated by the equation below.

$$\sigma_e = \sqrt{\sigma_{max}^2 + 3 \cdot \tau_{max}^2} \cdots\cdots (14)$$

Here,

σ_e : Equivalent stress

4.4 Strength estimation

The load frequency charts shown in section 4.2 are related to loads applied at an arbitrary section area along the axis of a stud or a bolt. Generally speaking, since the load condition varies at different positions along the axis of a stud or a bolt, it is not easy to predict the load condition at certain section area, the weakest section area for instance, from the analyzed result at some other section area.

However, as already mentioned, the structures of most critical bolts of a commercial machine are generally so devised and designed that their load condition is simplified and made uniform. Therefore, in many cases, the load condition at the weakest section area can be estimated by analysis of load at a certain arbitrary section area. The weakest point of a bolt is generally near the first thread ridge meshing with internal thread, [(11),(12)] and the measured load data here is of practical use for approximation of overall load. In this report as well, it is considered that the weakest point of a bolt can be derived from the load measured at the point where measurement is practically feasible.

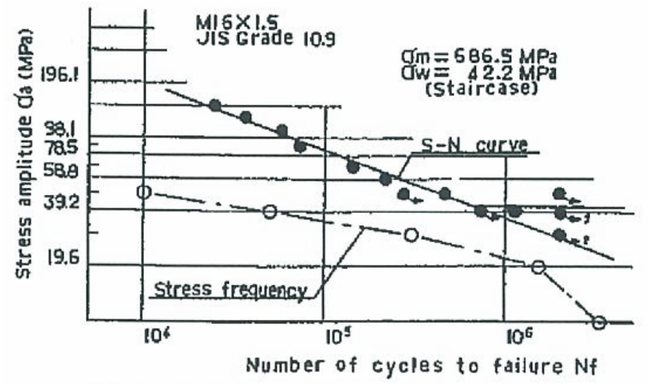

Fig.13 Stress level occurrence frequency of bolts and S-N curve

Fig.13 is a diagram showing frequency of occurrence of levels of stress and the S-N curve of a bolt in a tested machine. In order to estimate the bolt strength, the lifetime under operational load is calculated from these results using a fatigue life estimation method. The stress occurrence frequency data in the figure are the results of counting occurrences of levels, based on the same method as used in calculating load peak occurrence frequency, calculating the equivalent stress with equation (14) using the time series data of load peak values. The S-N curve was derived by testing an equivalent bolt used in a commercial machine using an electro-hydraulic servo fatigue testing equipment. Various stress amplitudes are here plotted in the figure against the number of their occurrences in 5000 hours to make the comparison with the S-N curve easier.

The stress applied to bolts during machine operation is usually designed to be under the fatigue limit, but in such a prototype machine as this example, service stress occasionally exceeds the fatigue limit as in the case of Fig.13. In such a case, safety is achieved by setting a limited lifetime using a modification of Miner’s law, or the machine is redesigned to reduce the maximum stress below the fatigue limit. In this example, the bolts shown in Fig.7 were also adopted for commercial machines, and strength reliability in actual commercial operation has been confirmed, as there have been no troubles with those machines which have passed the expected period of endurance.

The accuracy of the thus-acquired data of load occurrence frequency and stress occurrence frequency is not sufficient due to the inaccuracy of stiffness constants and measuring positions, but these can be improved to an industrially acceptable level by standardizing these data based on actual commercial results and simple stress measurement using bolt gauges. Especially, higher accurate analysis of stiffness constants may be possible using such detailed equations as in VD12230[2] or the stiffness constant equations by Shibahara and Oda[5] and by Sawa and Kumano.[6]

5. Conclusions

In order to make appropriate estimation of bolt-joint strength, it is important to coordinate the results of measured and analyzed loads applied to bolt-joints into standard technical information for design which enables choice of appropriate size, strength classification and appropriate configuration of bolts. This will make possible fundamental technology for concurrent multiple design which will be especially effective in promoting efficient product development.

In light of the above, a measuring method using strain gauges and equations for analysis which derives loads applied to bolt-joints was proposed. Measurement, analysis, and strength estimation were executed with regard to operational loads applied to the fitting bolts of the base carrier members of a construction machine. The results can be summarized as follows.

(1) Analytical equations for measurements using strain gauges, which calculate loads applied to bolt joints were proposed, and it was experimentally verified that axial tension, bending moment and torsion torque applied to bolts could be separately evaluated with accuracy.

(2) This method was applied to the bolts used to fit the base carrier members of a construction machine, and load peak distribution diagrams were made by analyzing the frequency of occurrence of load peaks of various levels applied to the machine in operation.

(3) A stress distribution diagram was acquired from time serial data of load peaks, and comparing this with the S-N curve of an equivalent bolt used in a commercial machine, strength estimation of bolts during machine operation was shown to be possible by a fatigue lifetime estimation method such as a modification of Miner’s law.

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