SIMPLE MODELING FOR STIFFNESS EVALUATION OF BOLTED JOINTS USING INTERFACIAL ELEMENT

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Abstract. This study has developed a novel finite element named interfacial element which simulates the contact between microscale asperities at contact surfaces of bolted joints. In this element, the contact is assumed to be the Hertzian contact of elastic asperities whose peak heights obey the Gaussian distribution. Based on this assumption, the stiffness of the interfacial element is derived from the compressive stress and the surface texture of the interfaces. On the other hand, it is necessary for large-scale simulations that target the entire vehicle body to reduce the number of nodes and elements in the finite element models. This study has further proposed simple modeling for stiffness evaluation of bolted joints using the interfacial element. Finite element simulations by simplified models in which heads, axes and holes of bolts were ignored were conducted and compared with detailed models and hammering tests. The results revealed that the mean value of the natural frequency of the simplified models had good agreement with that of the detailed models and the hammering tests though the calculation accuracy of the simplified models were lower than the detailed models. The bolt heads and the nuts could be ignored by increasing the density of the bolt axes to be equal to the total weight.

1 INTRODUCTION

Bolted joints are one of major joining methods of dissimilar materials required to construct multi-material structures [1–4]. Bolted joints have high joint strength because the external load is supported by bolt axes. However, the stiffness of bolted joints easily changes depending on the surface texture at the contact surface of jointed members and the clamping force of the bolt and nut. This is caused by the contact state between the microscale asperities at the contact surfaces which affects the transmission of the force between the members. In particular, the effect of the clamping force is remarkable. The interfacial stiffness and the natural frequency of the jointed part decrease with the decrease of the clamping force. The previous study developed a novel finite element named interfacial element which simulates the contact between microscale asperities at contact surfaces of bolted joints [5,6]. In this element, the contact is assumed to be the Hertzian contact of elastic asperities whose peak heights obey the Gaussian distribution. Based on this assumption, the stiffness of the interfacial element is derived from the compressive stress and the surface texture of the interfaces. The surface texture parameters are the number of asperity peaks per unit area, mean radius of curvature at asperity
peaks and standard deviation of asperity peak heights those represent the randomness of the contour profile of the contact surface.

On the other hand, it is necessary for large-scale simulations that target the entire vehicle body to reduce the number of nodes and elements in the finite element (FE) models. Even when one hole is created in the FE model, the number of nodes and elements easily increases. This study has further proposed simple modeling for stiffness evaluation of bolted joints using the interfacial element. Finite element simulations by simplified models in which heads, axes and holes of bolts were ignored were conducted and compared with calculation results of detailed models and measurement results of hammering tests.

2 INTERFACIAL ELEMENT [5,6]

Figure 1 shows the contact model of nominally flat surfaces. As shown in the left of Fig. 1, the pieces 1 and 2 are joined by the compressive stress $p_z$. $z$-axis is defined along the normal direction of the interfaces of the pieces. $x$-axis and $y$-axis are defined to be normal to each other as shown in Fig. 1. The contact model describes the contacts of the spherical asperities shown in the middle of Fig. 1. The orthotropic model shown in the right of Fig. 1 simulates the interfacial stiffness. The total height of the asperities is $L$, and the gap between the mean planes of the asperity peak heights is $d$. The elastic modulus $E_z$, $G_{yz}$ and $G_{zx}$ are derived by the following process, and the elastic modulus on the other directions is set to zero.

Assuming that the asperity peak heights obey the Gaussian distribution and the contact of asperities is the Hertzian contact [7,8], the relationship between the nominal compressive stress $p_z$ and the gap $d$ is described by using the homogenized elastic modulus $C$ as follows [9].

$$p_z = C \cdot F_{5/2}(d)$$  \hspace{1cm} (1)

where

$$F_m(d) = \frac{1}{\sqrt{2\pi}} \int_0^\infty \zeta^m \exp \left[ -\frac{1}{2} \left( \zeta + \frac{d}{\sigma} \right)^2 \right] d\zeta$$  \hspace{1cm} (2)

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$$  \hspace{1cm} (3)

Using the subscript $i$ as the identifier for the piece $i = 1, 2$, $\sigma_i$ is the standard deviation of the asperity peak heights of the piece $i$. The homogenized elastic modulus $C$ is described as follows.
\[ C = \frac{64\pi^2\eta_1\eta_2}{15} \left( \frac{1}{\beta_1} + \frac{1}{\beta_2} \right)^\frac{3}{2} \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-\frac{1}{2}} \]  

where \( \eta_i \) is the number of the asperity peaks per unit area, and \( \beta_i \) is the mean radius of the curvature at asperity peaks. \( E_i \) and \( \nu_i \) are the Young’s modulus and the Poisson’s ratio of the base material of the piece \( i \), respectively.

Assuming that the small displacement occurs in the vibration of the specimen, the \( z \)-direction Young’s modulus of the interfaces can be described as follows.

\[ E_z = -\frac{\partial p_z}{\sigma(d/L)} = \frac{F_{7/2}(d)}{F_{5/2}(d)} p_z \cdot L/\sigma \]  

Similarly, Björklund derived the relationship between the shear stress and the displacement tangential to the interfaces under the compressive stress \( p_z \) based on the Mindlin’s theory [10]. Assuming that the stiffness in the tangential direction to the interfaces is isotropic, the shear modulus is described as follows.

\[ G_{yz} = G_{zx} = 4 \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \left( \frac{2 - \nu_1}{G_1} + \frac{2 - \nu_2}{G_2} \right)^{-1} E_z \]  

where \( G_{yz} \) is the shear modulus at the \( y \)-surface in the \( z \)-direction, and \( G_{zx} \) is the shear modulus at the \( x \)-surface in the \( z \)-direction. \( G_i \) \( (i = 1, 2) \) is the shear modulus of the base material of the piece \( i \). In the finite element method (FEM) using the above model, the finite element whose elastic modulus are given by Eq. (5) and Eq. (6) is inserted into the interfaces. The finite element has been named as the interfacial element in this study.

3 INFLUENCE OF THREAD AND SEATING SURFACES

3.1 FE model

Before the investigation of the simple modeling, the influence of thread and seating surfaces of bolt and nut were investigated by the FEM of single bolted joint. Figure 2 shows the FE model of the single bolted joint (9190 nodes and 6687 elements). The constituent material of the bolted joint was set to steel (Young’s modulus \( E_i = 200 \) GPa, Poisson’s ratio \( \nu_i = 0.3, \) Density = 7800 \( \text{kg/m}^3 \)). The surface texture parameters were set to reference values as shown in Table 1. The interfacial element was applied on the thread and the seating surfaces. The fixing element is connected to the main body in order to stably compute the rigid body mode. The length of the bolt axis was given as three patterns (\( L = 20, 50, 105 \) mm).

<table>
<thead>
<tr>
<th></th>
<th>Number of asperity peaks per unit area</th>
<th>Mean radius of curvature at asperity peaks</th>
<th>Standard deviation of asperity peak heights</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bolt</td>
<td>24221</td>
<td>9.750</td>
<td>0.413</td>
</tr>
<tr>
<td>Nut</td>
<td>13348</td>
<td>5.604</td>
<td>2.862</td>
</tr>
<tr>
<td>Cylindrical collar vs. Bolt</td>
<td>4841</td>
<td>5.977</td>
<td>1.468</td>
</tr>
<tr>
<td>Cylindrical collar vs. Nut</td>
<td>8180</td>
<td>7.303</td>
<td>0.636</td>
</tr>
</tbody>
</table>

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3.2 Results and discussion

Figures 3 and 4 show the calculation results of the natural frequency of the single bolted joint. The natural frequency of the rigid body modes is constant subjected to the clamping force because the interfacial elements have no deformation in these rigid body modes as shown in Figures 3 and 4.

Figure 2: Schematic and finite element model of single bolted joint

**Figure 3:** Natural frequency of rigid body modes (L = 20 mm)

![Figure 3: Natural frequency of rigid body modes (L = 20 mm)](image)

**Figure 4:** Natural frequency of 7th mode with different length of bolt axis

![Figure 4: Natural frequency of 7th mode with different length of bolt axis](image)

- Fixing element
  - Young’s modulus = 0.1 GPa
  - Poisson’s ratio = 0.3
  - Density = 1 kg/m³

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Figure 5: Vibration shapes of single bolted joint (L = 20 mm)

(a) 1st mode  
(b) 2nd mode  
(c) 3rd mode  
(d) 4th mode  
(e) 5th mode  
(f) 6th mode

Figure 6: 7th vibration shape of single bolted joint with different length of bolt axis

(a) 7th mode, L = 20 mm  
(b) 7th mode, L = 50 mm  
(c) 7th mode, L = 105 mm

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Fig. 5. The natural frequency of the 7th mode which is the lowest mode except the rigid body modes is affected by the interfacial stiffness of the thread and the seating surfaces and increases with the increase of the clamping force. The vibration shape is the 1st bending deformation as shown in Fig. 6. Moreover, the natural frequency increases with the decrease of bolt axis. The natural frequency will be higher than 20 ~ 70 kHz when the bolt axis is shorter than 20 mm. The influence of the thread and seating surfaces will be small on the natural frequency under 20 kHz and the bolt axis under 20 mm.

4 INFLUENCE OF BOLT HEADS, AXES AND HOLES

4.1 FE model and hammering test

The thick plates joint and the thin plates joint were prepared as shown in Figs 7 and 8. These joints consist of steel (Young’s modulus $E_t = 200$ GPa, Poisson’s ratio $\nu_t = 0.3$, Density = $7800$ kg/m$^3$). In the hammering tests, the natural frequency was obtained from the time history of

![Figure 7: Thick plates joint and its finite element models](image1)

![Figure 8: Thin plates joint and its finite element models](image2)
the acceleration just after the impact by the impulse hammer. The clamping force was measured by inserting the pressure measurement films (Prescale made by Fujifilm Corporation) into the seating surfaces of the bolts. The monolithic specimen whose shape is same as the thick plates joint was also tested. The monolithic specimen is the T-shaped thick plate clamped by 8 sets of bolt and nut.

The number of nodes and elements of each model is shown in Table 2. The interfacial element was applied only on the interfaces of the plates. The surface texture parameters were

<table>
<thead>
<tr>
<th>Model</th>
<th>Thick plates joint</th>
<th>Thin plates joint</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Nodes</td>
<td>Elements</td>
</tr>
<tr>
<td>Detailed model</td>
<td>13002</td>
<td>10521</td>
</tr>
<tr>
<td>Monolithic model</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Simplified model A</td>
<td>12186</td>
<td>10329</td>
</tr>
<tr>
<td>Simplified model B, C</td>
<td>10104</td>
<td>8482</td>
</tr>
</tbody>
</table>

Table 3: Surface texture parameters of plates

<table>
<thead>
<tr>
<th>Material</th>
<th>Number of asperity peaks per unit area $\eta_i$ [mm$^{-2}$]</th>
<th>Mean radius of curvature at asperity peaks $\beta_i$ [μm]</th>
<th>Standard deviation of asperity peak heights $\sigma_i$ [μm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Short thick plate</td>
<td>11346</td>
<td>3.247</td>
<td>2.189</td>
</tr>
<tr>
<td>Long thick plate</td>
<td>9311</td>
<td>6.012</td>
<td>2.493</td>
</tr>
<tr>
<td>Thin plate</td>
<td>25328</td>
<td>27.75</td>
<td>0.151</td>
</tr>
</tbody>
</table>

Figure 9: Simplification of bolt heads, axes and holes
measurement values as shown in Table 3. The FE model of the thick plates joint is the 1/4 model. The monolithic models in which the material constants of the interfacial element were set to those of steel was also prepared in order to investigate the influence of the interfacial stiffness between the plates. Figure 9 shows the schematic of the simplification of the bolt heads, axes and holes. Taking account of the results of the chapter 3, the contacts of the thread and the seating surfaces were ignored. The bolt axes were formed as solid shafts. The simplified model A ignored the bolt holes. The interfaces between the bolt axes and the plates were formed by common nodes. The clamping force would be transmitted to the interfacial elements as compressive force through the interfaces between the bolt axis and the plate as shearing force. The simplified model B and C ignored the bolt heads and the nuts. The transmission process of the clamping force is same as the simplified model A. Because the total weight of the bolted joints would reduce by the amount of the bolt heads and the nuts in the simplified model B, the density of the bolt axes in the simplified model C were increased to be equal to the simplified model A (Density = 18000 kg/m³ in the thick plates joint, 62000 kg/m³ in the thin plates joint).

4.2 Results and discussion

Figures 10 and 11 show the natural frequency of the thick plates joint and the thin plates joint, respectively. The natural frequency is the 1st bending mode which is the lowest natural frequency.
frequency except the rigid body modes as shown in Figs 12 and 13. In the hammering tests, the natural frequency of the thick plates joint and the thin plates joint increases with the increase of the clamping force. On the other hand, the natural frequency of the monolithic specimen is constant even if the clamping force is increased. Therefore, the natural frequency is affected by the interface between the plates, and the influence of the thread and the seating surfaces is small because the natural frequency of the monolithic specimen is constant.

In Fig. 10, the calculation result of the detailed model agrees with the hammering test. Hence, the interfacial element successfully simulates the relationship between the natural frequency and the clamping force. The simplified models agree with each other. It seems that the influence of the bolt heads and the nuts is small because the bolt heads and the nuts are relatively smaller than the plates. The mean values of the natural frequency of the simplified models have good agreement with the detailed model and the hammering test. The reduction of the calculation accuracy depends on the presence or absence of the bolt holes.

In Fig. 11, the natural frequency of the detailed model is closer to the hammering test than the monolithic model. The simplified models A and C agree with the mean value of the detailed model. The simplified model B is higher than the hammering test, the detailed model and the simplified models A, C. It is difficult to just ignore the influence of the bolt heads and the nuts in the thin plates joint. In this situation, the bolt head and the nut could be ignored by increasing the density of the bolt axes to be equal to the total weight in the simplified model.

5 CONCLUSIONS

This study has proposed simple modeling for stiffness evaluation of bolted joints using the interfacial element based on the investigation of the influence of thread surfaces, seating
surfaces, bolt heads, bolt axes and holes. From the results of the numerical calculations and the hammering tests, the natural frequency of the detailed models and the simplified models was closer to the values of the hammering tests though the calculation accuracy of the simplified models were lower than the detailed models. The bolt heads and the nuts could be ignored by increasing the density of the bolt axes to be equal to the total weight in the simplified models. This work was supported by JSPS KAKENHI Grant Number JP18K03849 and JP21K03776.

REFERENCES


