

Enhancement of Vibration Analysis Accuracy of Engine Structure by Considering Nonlinear Properties between Components

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ABSTRACT

A high-accuracy and generally applicable method of predicting the frequency response functions of engine structures up to the radiated noise range has been developed.

Because the joints between engine parts possess nonlinear properties, as exemplified by the head gasket, analytic models that are linearized in relation to the state of the joints are essential in conducting frequency response analyses. The analysis discussed in this paper was divided into two stages. In the first stage, pressure data for the joint sections was obtained by means of nonlinear contact analyses. In the frequency response analyses conducted in the second stage, accuracy was increased by defining the stiffness and damping of the joint sections as linear properties based on the pressure data, and formulating vibration analysis models that considered strain due to bolt axial force.

Some properties of the joint sections cannot be determined in materials tests, and pressure conversion tables assumed to express properties dependent on pressure were therefore defined. Uncertain values were identified on the basis of test results for the base engine. The application of this method to another engine demonstrated that the frequency response analysis achieved a level of accuracy that represented no impediment to practical use, and indicated that the method possessed general applicability for frequency response analyses of engine structures.

1. Introduction

Today, demands for the realization of enhanced environmental performance from automobiles are increasing. In order to respond to these demands, automakers are seeking to realize weight-savings at the level of engine parts, in addition to proceeding with the development of combustion control that involves turbocharging. At the same time, demands for the achievement of greater commercial appeal in terms of noise and vibration performance are also increasing.

In many cases, the reduction of the weight of engine structures increases vibration response sensitivity, and turbocharging and the realization of high-speed combustion increase exciting force. Because of this, balancing the achievement of increased commercial appeal through enhanced noise and vibration performance with environmental performance is becoming increasingly challenging.

To further complicate the situation, demands for the reduction of the engine development period are also

increasing, and the formulation of technologies for applying the finite element method (FEM), which makes it possible to analyze frequency response with a high level of accuracy using the CAD models only, is essential in order to help enable the development of engine configurations able to realize a low vibration response at a reduced weight.

In order to analyze the frequency response of engine assemblies composed of a large number of parts with a high degree of accuracy, it is necessary to correctly model the stiffness and the damping of the joints between the parts. Because the head gasket and other parts display nonlinear elastic properties of the joints between the parts, the frequency response functions (FRF) of engine assemblies change depending on the bolt axial force. Figure 1 shows changes in the FRF during a test in which the cylinder head and the block were tightened with different bolt axial forces. The surface of the external wall of the cylinder block was the exciting point, and the surface of the cylinder head combustion chamber was the response point.

In the high-frequency range, the differences in FRF due to the bolt axial force become conspicuous, demonstrating

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the importance of considering nonlinear elastic properties in relation to the bolt axial force.

It has also been known that even in areas of direct contact between metal surfaces, the effective joint region changes due to the bolt axial force, and differences occur in the peak frequencies and peak levels of FRF. Therefore, accurate modeling of the stiffness and the damping of the joints between parts is essential.

Reports have previously been published concerning analyses that take the effective joint region of the bolt joints into consideration^{(1), (2)}. However, there has been insufficient discussion of the damping occurring in the joints, and in many cases the models employ modal damping for the entire structure. As an example, Fig. 2 shows the results of an analysis of an engine assembly. The same models as were employed in the development method discussed in this paper were used for the main structural parts model and the bolt model, and past empirical values were used for the modal damping of the entire structure. The camshaft drive chain-side bearing was the exciting point and the surface of the external wall of the lower block was the response point.

Given the significant deviation in the peak frequency close to 2 kHz and the significant deviation in the response level due to the fact that there is excessive modal damping, it cannot be claimed that this method achieves sufficient analytic accuracy.

This paper discusses a method of increasing the accuracy

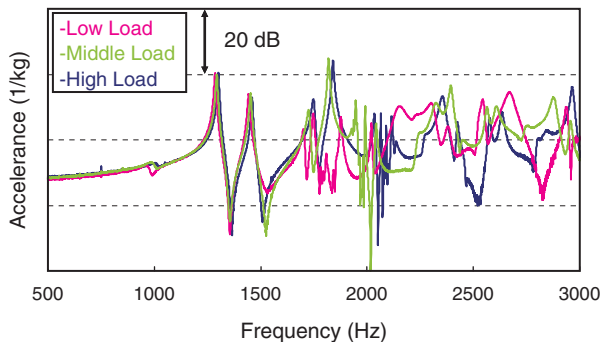


Fig. 1 Experimental FRF under different bolt axial force conditions

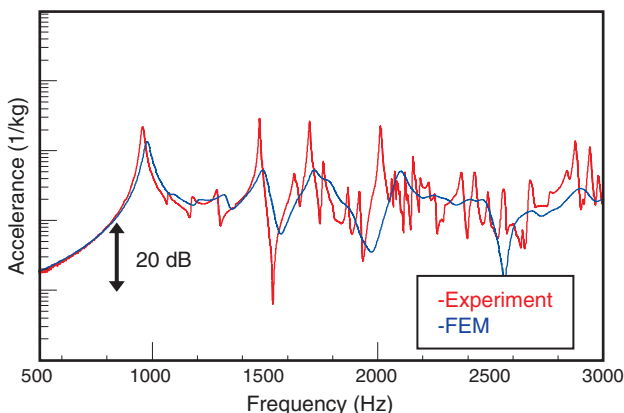


Fig. 2 FRF calculated by previous method

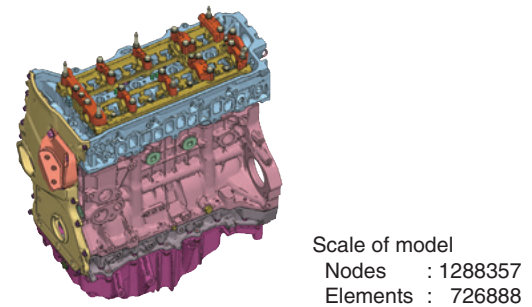
of prediction of FRF by modeling the stiffness and damping of the joint sections based on pressure data for the sections in contact obtained from nonlinear contact analyses. Taking radiated noise into consideration, the range up to 3 kHz in an assembled state was set as the frequency range for the analysis.

2. Development Method

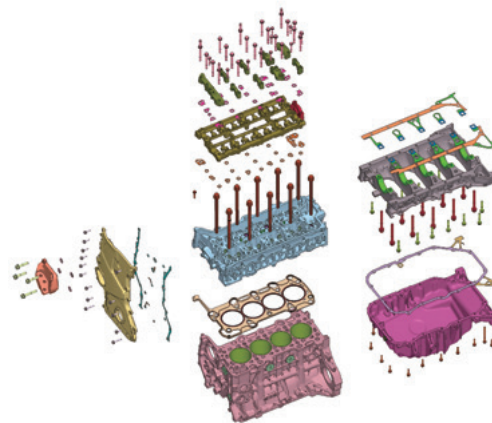
The analysis included the cylinder block, the lower block, and the cylinder heads, representing the main structure of the engine, and the oil pan and chain case, the parts that radiate noise.

In the engine configuration studied, joint sections with different nonlinear properties exist between the parts (head gaskets, metal contact sections, and formed in place gaskets (FIPG) sections). Consequently, joint elements were set between the parts and their nonlinear elastic properties were defined. Figure 3 shows the composition of models.

Regarding the structure in an assembled condition as displaying linear behavior and conducting a frequency response analysis is a practical method of determining the FRF within the minute amplitude range of engine noise and vibration phenomena. In addition, the modal method represents an advantage over the direct method in terms of calculation efficiency in determining the FRF up to the high-frequency range, in which response peaks of multiple



(a) Assembled condition



(b) Disconnected condition

Fig. 3 Composition of FE models

eigen frequencies are observed. In order to apply the modal method, it is necessary to define the coupling state between parts as stiffness rather than force.

Given these conditions, a two-stage analytic method was employed, in which a nonlinear contact analysis was conducted, following which a modal frequency response analysis was conducted with the stiffness of the joint elements due to contact pressure and the damping occurring at the contact boundaries defined as linear properties of the materials.

2.1. Hypothetical Stiffness and Damping Properties of Joint Elements

Because some properties cannot be determined in unit tests using test pieces and it is not possible to unambiguously define the joint elements as displaying linear properties, the following method was proposed.

The Young's modulus of the joint elements is a property that can basically be calculated from the nonlinear elastic properties determined in unit tests using test pieces and the thickness of the elements. However, it is believed that in a state of low contact pressure, nonlinear properties are observed even with vibrations of minute amplitudes.

The shear modulus and the damping coefficient are properties that cannot be determined in unit tests.

The conversion tables shown in Fig. 4 were introduced to define the properties of the joint elements, including properties displaying this type of uncertainty, as linear properties in response to contact pressure. The blue solid lines show certain values for properties as determined in unit tests, while the red broken lines show uncertain values for properties.

(1) Young's modulus

The conversion table shown in Fig. 4(a) was defined to show the Young's modulus of the head gasket in a clamped state, calculated from the nonlinear elastic properties and the thickness of the elements, as a linear property in relation to contact pressure.

It is believed that in a state of high contact pressure, the metal contact sections and the FIPG sections display the same Young's moduli as the base metal materials, but that in a state of low contact pressure, they are nonlinear and display lower Young's moduli. The conversion table in Fig. 4(b) was defined to linearize their properties. In addition, the metal contact sections and the FIPG sections were separately defined because the latter display differing properties due to the thickness of the FIPG film.

(2) Shear modulus

It is believed that in a state of high contact pressure, shear moduli are identical to those of the base metal material, but in a state of low contact pressure, the shear moduli are lower than those of the base metal. In addition, it is believed that infinitesimal slippages occur in a state of low contact pressure, and because of this the shear moduli are underestimated compared to the Young's moduli. Given this, the conversion table shown in Fig. 4(c) was defined, showing the shear moduli as exponential functions in relation to pressure.

(3) Damping coefficient

Most damping between parts is believed to originate in minute relative displacements between the parts when they vibrate. It is believed that in a state of low contact pressure the relative displacement will be high and a high level of damping will be produced, while in a state of high contact pressure the level of damping will be low due to the low

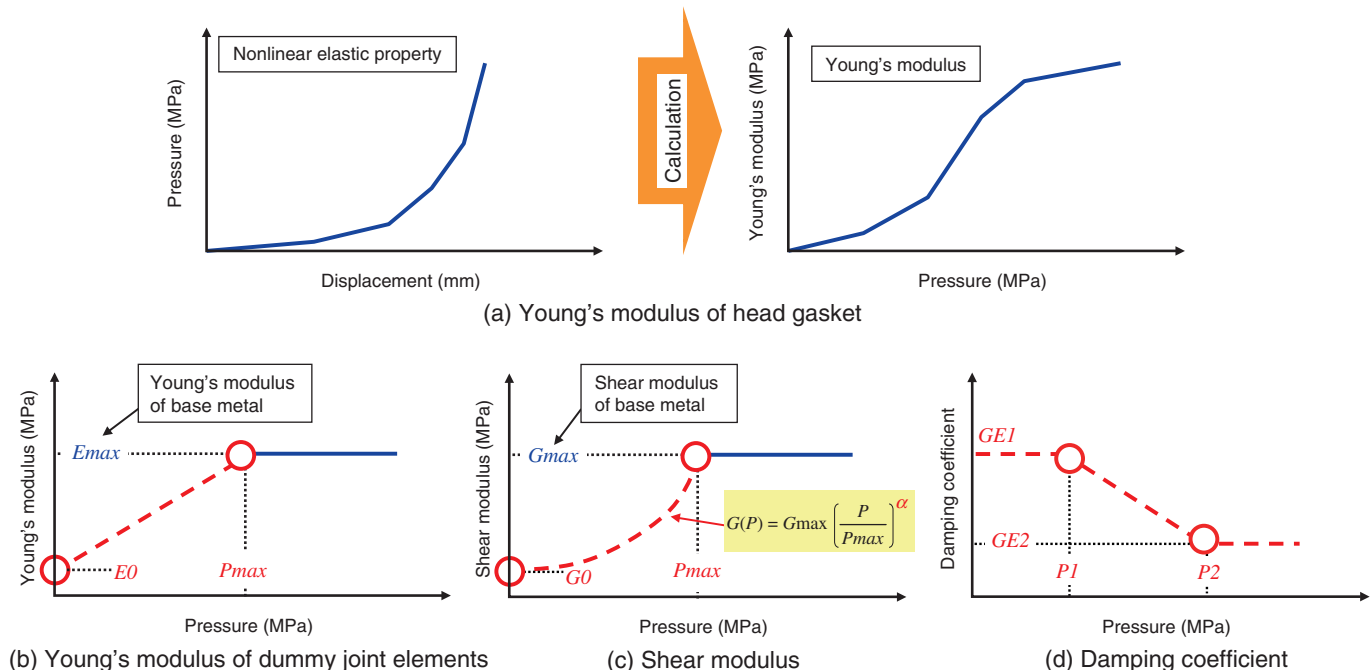


Fig. 4 Conversion tables of joint elements

relative displacement between the parts. The conversion table shown in Fig. 4(d) was defined on this basis.

2.2. Identification of Hypothesized Properties of Joint Elements

The uncertain parameters were identified so that the level of deviation from FRF as determined in excitation tests on multiple engine assemblies was minimized.

3. Nonlinear Contact Analysis

This chapter will discuss the nonlinear contact analysis (MD/NASTRAN SOL600) conducted in the first stage of the analysis.

3.1. Composition of Models

The analytic methodology was formulated with the diesel engine employed in the 2004 model ACCORD as the subject of the analysis. Here, the details of the previously mentioned structural model (Fig. 3) will be discussed.

The nonlinear contact analysis model is composed of a model of the main structural parts, such as the cylinder block; a model of the joint elements, such as the head gasket; and a bolt model.

3.1.1. Model of main structural parts

The accuracy of the modeling of individual parts is dependent on factors including shape, material properties and element size. In order to create models equivalent to parts for which figures had been verified in actual tests, the shapes of the tested parts were faithfully reproduced, and material properties measured using test pieces cut from the tested parts were employed⁽³⁾. Tetrahedral secondary elements were employed. Accuracy increases as the element size is reduced, but this results in an increased calculation load. The element size was therefore optimized by determining convergence.

Figure 5 shows a comparison of test and analysis results for the cylinder block and the oil pan as an example of the verification of the accuracy of FRF in the models of individual parts. 3 kHz was assumed in an assembled state, and the frequency range up to 5 kHz was set as the target for the analysis.

The results demonstrate that it was possible to analyze FRF with a high accuracy.

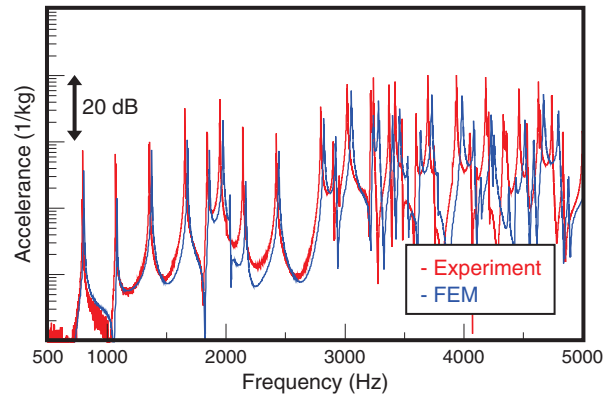
3.1.2. Joint element model

Gasket elements making it possible to define nonlinear elastic properties on one-layered hexagonal elements were employed in SOL600, one of the MD/NASTRAN nonlinear contact analyses. Contact was judged between the nodal points of the gasket elements and the element surfaces of the parts in contact, and the reaction force was calculated in relation to the distance between the nodal points. Because of this, there was no need to consider consistency in the positioning of the nodal points, and the elements also

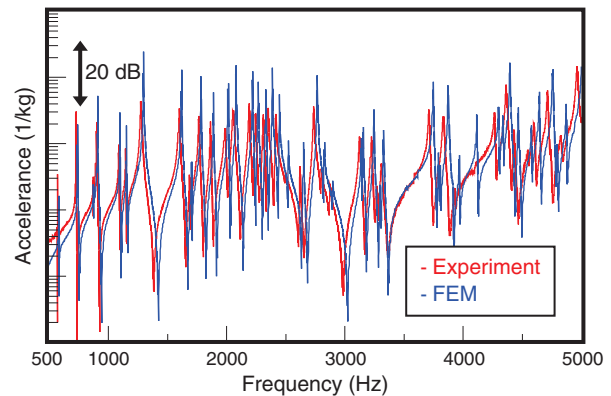
increased convenience in terms of creation and amendment of the models.

The elements employed were approximately one-third the size of the elements used in the model of the main structural parts in order to make it possible to reproduce the pressure distribution on the contact surfaces.

Figure 6 shows the head gasket model. Each part of the head gasket displays differing nonlinear elastic properties due to the bead shapes and the thickness of the structural sheets, and property values were therefore defined on the basis of values obtained from unit tests on test pieces for



(a) FRF of cylinder block



(b) FRF of oilpan

Fig. 5 FRF of component

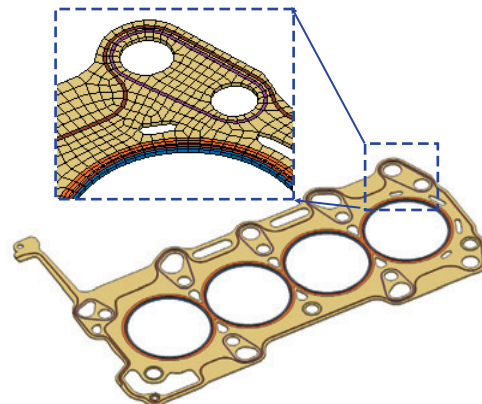


Fig. 6 Head gasket model

each region. As an example of this process, Fig. 7 shows the definition of nonlinear elastic properties for seven regions.

To make it possible to treat the metal contact sections and the FIPG sections in the same way as the head gasket, the hypothetical thin joint elements shown in Fig. 8 were introduced, and defined as gasket elements.

The nonlinear elastic properties were defined on the basis of values calculated from the Young's moduli of the metals in the contacting sections. Because the reactive force of the FIPG due to viscoelasticity was much lower in comparison to the joint stiffness of the main structural parts, it was considered possible to ignore this parameter. As examples, Fig. 9 shows the properties of the cast iron,

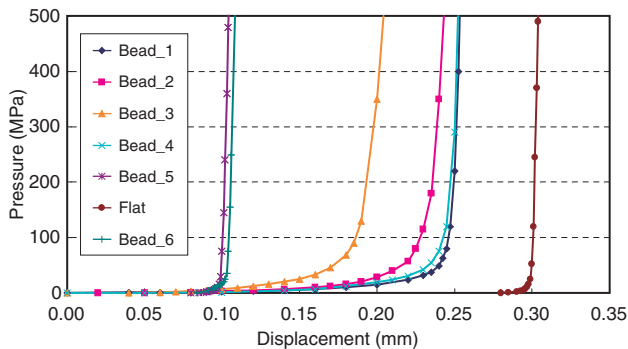


Fig. 7 Elastic properties of head gasket

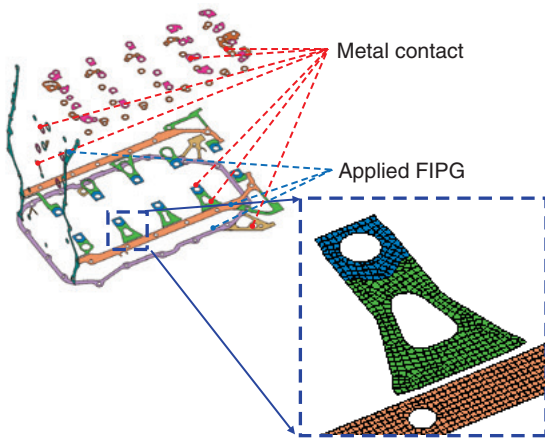


Fig. 8 Joint elements model

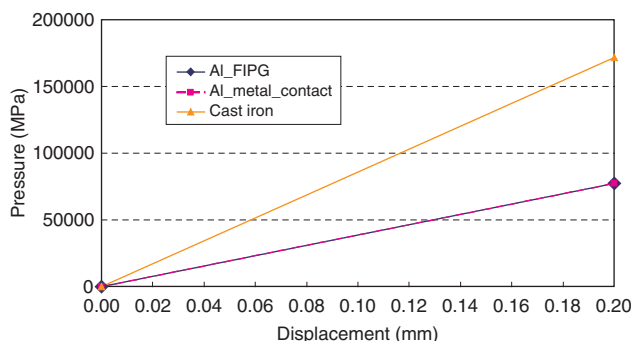


Fig. 9 Elastic properties of joint elements

aluminum, and FIPG sections of the lower block. Because the Young's moduli of the metals are employed, the properties display linear gradients.

3.1.3. Bolt model

In order to correctly reproduce the distribution of mass and help to enable an accurate analysis of the distribution of pressure in the joints, the bolt model employed hexagonal elements that reproduced the shapes of the heads of the bolts (Fig. 10). The not-in-contact sections of the bolt shafts were divided at the middle, and drawing direction values for bolt axial force were imparted to the separated sections.

3.2. Engine Configurations for Nonlinear Contact Analysis

In order to verify the ability of the analytic method to respond to a variety of parts structures, analyses were conducted of the four engine assembly configurations with differing composition of parts shown in Fig. 11.

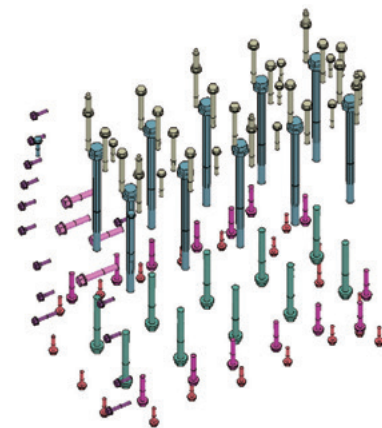


Fig. 10 Bolts model

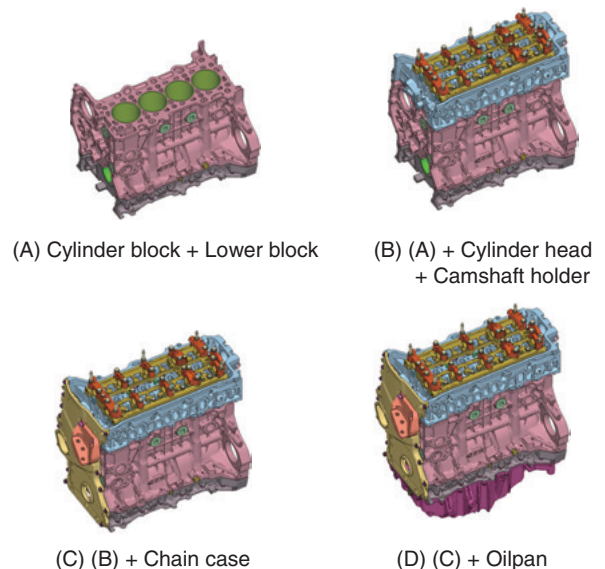


Fig. 11 Assembled models

3.3. Results of Nonlinear Contact Analysis

Figure 12 shows (a) the Mises stress distribution and (b) the contact pressure distribution in the joint elements for assembly (D) as an example of the results obtained from the nonlinear contact analysis.

The contact pressure displays different values in each part due to the effect of factors including the bolt axial force, the stiffness of the part, and the properties of the gasket beads. This demonstrates that nonlinear contact analysis was essential in obtaining this data.

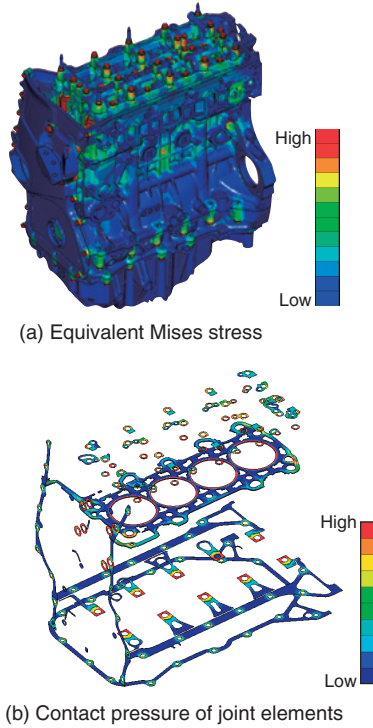


Fig. 12 Nonlinear contact analysis results

4. Frequency Response Analysis

This section will discuss the frequency response analysis (MD/NASTRAN SOL111) conducted in the second stage of the analysis.

The states of connection of the main structural parts and the joint elements and bolts in the nonlinear contact analysis are linked by the force between the nodal points judged to be in contact. In order to maintain the same linkage, the nodal points judged to be in contact were connected by means of multi-point constraints (MPC), and a frequency response analysis was conducted.

Because the eigen frequencies of the bolt shafts depend on the strain due to the axial forces and therefore vary, axial forces were imparted to the bolts in the analysis.

4.1. Identification of Hypothetical Property Values

Based on the concepts discussed in Chapter 2, the uncertain parameters were identified using the four assembly configurations with different parts structures (see Section 3.2) and employing optimization methods, design of experiments, and other mechanisms to minimize deviation between the values for FRF determined in the analysis and the values determined in excitation tests.

The number of parameters which it was necessary to identify is shown as X^* in Fig. 13. The number of parameters to be identified was minimized by considering the values for the head gasket sections, the metal contact sections, and the FIPG sections to be identical.

4.2. Joint Element Properties

Figure 14 shows results for the distribution of property values for the joint elements defined in accordance with the identified conversion tables. Figure 14(a) shows the distribution of the Young's moduli, Fig. 14(b) the

	Young's modulus		Shear modulus			Damping coefficient			
	$E0$	P_{max}	$G0$	P_{max}	a	$GE1$	$P1$	$GE2$	$P2$
Head gasket	—	—	X1	X2	X3	X4	X5	X6	X7
Metal contact	X8	X9	X10	X9	X11	X12	X13	X14	X15
FIPG	X16	X9	X17	X9	X11	X18	X13	X14	X15

Fig. 13 Parameters of identification

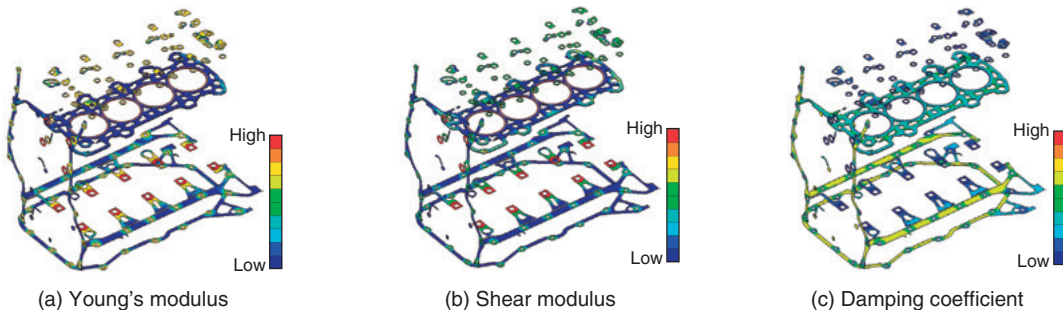


Fig. 14 Properties of joint elements for frequency response analysis

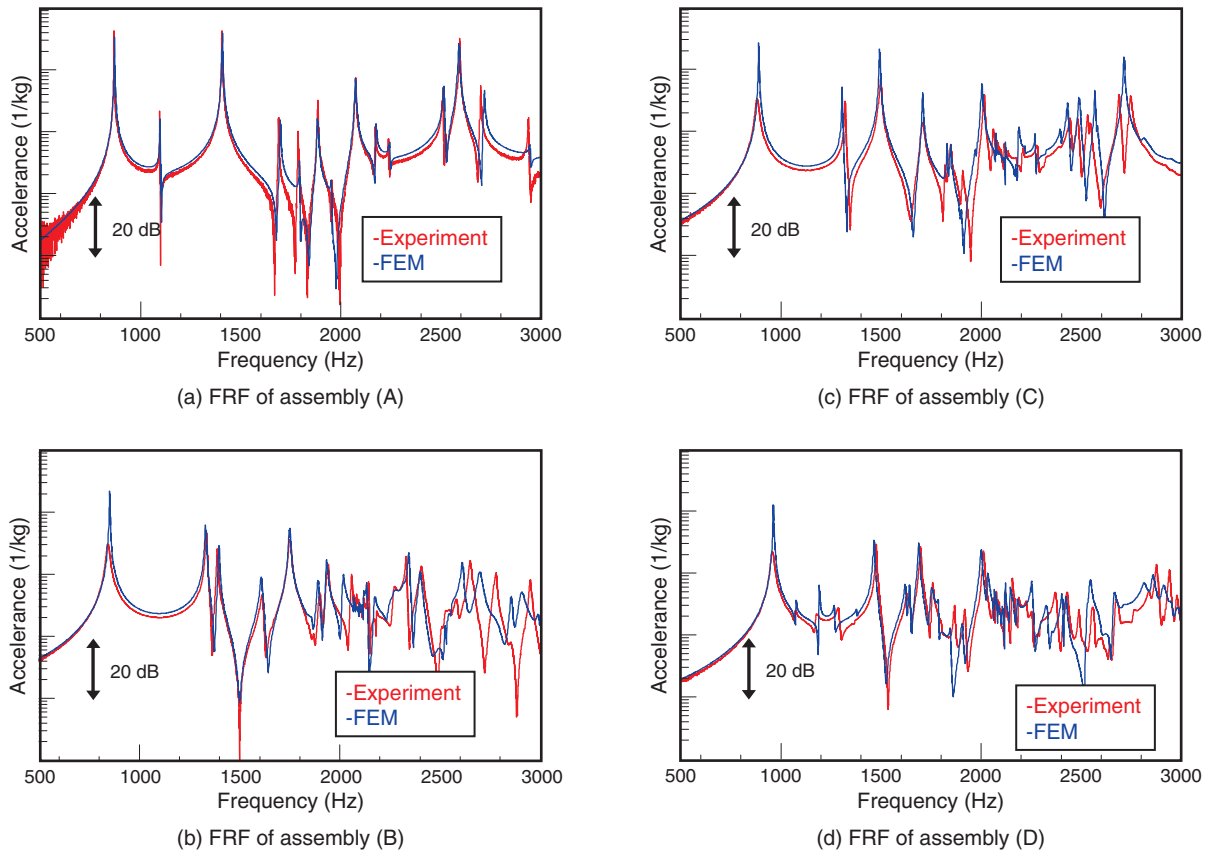


Fig. 15 FRF of assembled models

distribution of the shear moduli, and Fig. 14(c) the distribution of the damping coefficients.

4.3. Results of Frequency Response Analysis

Figure 15 shows a comparison of the results of frequency response analyses and tests for the four assembly compositions. The camshaft drive chain-side bearing was the exciting point, and the surface of the external wall of the lower block was the response point.

FRF has been determined with a high accuracy for each of the assembly compositions, indicating that the identified stiffness and damping of the joint elements were accurately modeled.

5. Verification of Method

5.1. Engine used in Verification

Tests were conducted using a different engine (the diesel engine employed in the 2008 model year ACCORD), as shown in Fig. 16, in order to verify the validity of the developed method. Because the analysis was to be conducted using only CAD data, the FEM model was formulated using 3D-CAD data, and deviations in the shapes of the parts in relation to the shapes of the parts used in tests were not corrected. The same material properties as were employed in the model discussed in Chapter 3 were used.

Figures 17(a) and 17(b) respectively show results for the cylinder block and the lower block as examples of verification for individual parts.

In the case of the lower block, deviations are observed which are thought to originate in factors including deviations of shape in relation to the parts used in tests⁽³⁾, but this corresponds to the eigen frequency deviation of around 3%, indicating sufficient accuracy for practical use for a model formulated from CAD data.

5.2. Results of Frequency Response Analysis

Figure 18 shows an example of the results of frequency response analyses conducted using the pressure conversion tables for identified joint element properties discussed in Section 4.1. The transmission-side bearing was the exciting

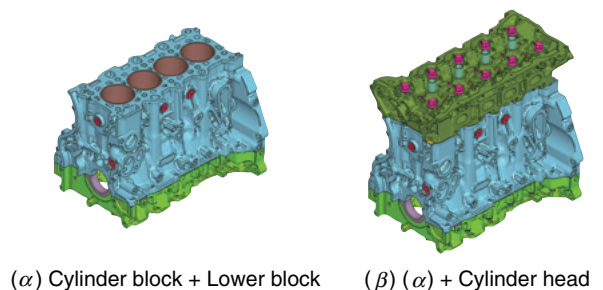
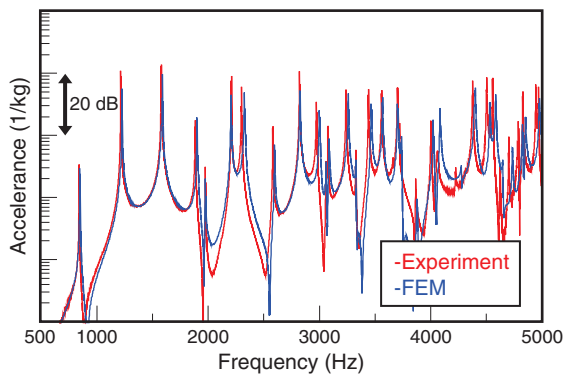
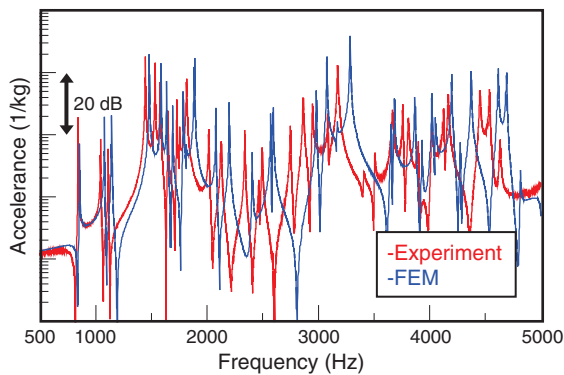


Fig. 16 Models for verification



(a) FRF of cylinder block



(b) FRF of lower block

Fig. 17 FRF of component for verification

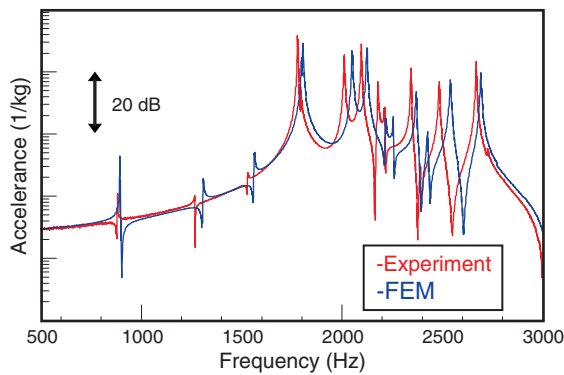
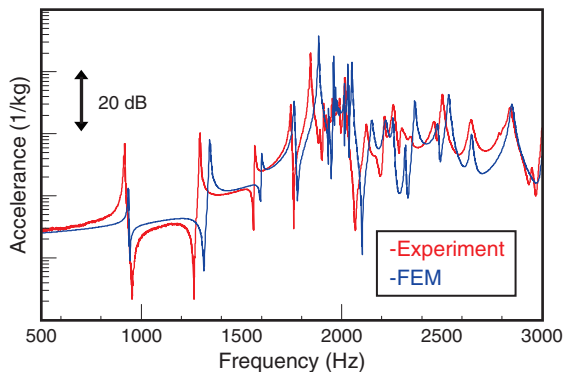
(a) FRF of assembly (α)(b) FRF of assembly (β)

Fig. 18 FRF of assembled models for verification

point, and the surface of the external wall of the cylinder block was the response point.

The level of deviation is basically identical to that for the individual parts, demonstrating the achievement of a level of accuracy suitable for practical use, and verifying the validity and general applicability of the developed method.

6. Discussion

The stiffness and damping of the joint elements used in the frequency response analyses discussed here were identified in order to correspond to test FRF values on the basis of conversion tables assumed to express properties dependent on pressure when parts are clamped. They were not physically verified as local property values. However, the results of employing the same method and the same conversion tables to analyze another engine indicated the achievement of sufficient accuracy for practical use. These results indicate that the method is basically valid and generally applicable as a method of expressing the stiffness and damping of the joints in engine structure vibration analysis models.

The method can be increased in accuracy and made more generally applicable in future by accumulating further examples of its application and making minor corrections in the conversion tables, which contain uncertain elements. It is also believed that it will be possible to expand the application of the method to joints offering different conditions, for example rubber seals.

7. Conclusion

A high-accuracy and generally applicable method of predicting frequency response functions of engine structures up to the radiated noise range was developed.

The method makes it possible to conduct highly accurate analyses using only 3D-CAD data and material properties for the main structural parts, the nonlinear elastic properties of the head gasket and other parts, and the bolt axial force.

Because the developed method makes it possible to conduct high-accuracy analyses at the drawing stage, it is expected that its use in future engine development will contribute to the reduction of the development period, the reduction of prototype numbers, and the realization of engine structures displaying low vibration response characteristics at reduced weights.

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