

Research on Engine Mount Characteristics Identification of Outboard Motor Using Experimental Modal Analysis

Takeshi OKADA*

Kazuyuki SHIOMI*

Tetsuro IKENO*

ABSTRACT

Research was conducted on a method which can directly determine the engine mount dynamic stiffness in an outboard motor to gain a better understanding of the engine mount vibration characteristics of an outboard motor.

First, a method was established to identify the dynamic stiffness of the engine mount in a physical coordinate system using modal parameters acquired from experimental modal analysis. Using the calculation results obtained by the finite element method applied to the outboard motor model, it was revealed that this method has the ability to identify dynamic stiffness with high accuracy.

Next, the method was applied to two outboard motor models, the small BF5 (3.7 kW) and the large BF225 (165 kW). The dynamic stiffness of the engine mount was identified by the results of experimental modal analysis and it was compared to the result obtained by the conventional method using the dynamic stiffness measurement equipment. The results show that this method can identify the engine mount dynamic stiffness more adequately than the conventional method, even when the engine mounts are subjected to loads corresponding to thrust force or even in the case that the stiffness of the parts supporting an outboard motor is low.

1. Introduction

Outboard motor engines range from small single-cylinder 2 kW class engines to large V-type six cylinder engines in excess of 180 kW, and are mounted on various types of boats that differ in weight, shape and structure around the world. The excitation force produced by these engines varies widely for each engine type, while boats also have diverse vibration systems and acoustic fields, so the vibration and noise generated by the compound system of the outboard motor and the boat are almost infinite in variety. This is an important difference from the issue of noise and vibration in products such as motorcycles and cars, where the engine and chassis

are developed as a set, and that difference makes analysis and countermeasure development difficult.

On the other hand, while the absolute value of vibration and noise must be reduced as in motorcycles and cars, with boats there are also strong demands to realize more comfortable noise and vibration. This makes it important not only to reduce the vibration and noise of the outboard motor, but also to study the way of installing the outboard motor for better noise and vibration. Among efforts to reduce outboard motor vibration and optimize the engine mounts, the reduction of outboard motor vibration using a pendulum motion mount method has been reported⁽¹⁾. However, analysis and countermeasures for vibration and noise that include the boat must

* Asakahigashi R&D Center

be conducted individually for each case due to the reasons mentioned above. This makes systematic research difficult, and as a result there are few reported cases.

Transfer path analysis is known as a technique for analyzing vibration and noise in complex structural systems such as those in boats^{(2),(3)}. This technique analyzes how force is transferred within an actual structure such as a car. Typical methods for identifying the transmitted force include the dynamic stiffness method, which uses a dynamic spring rate⁽⁴⁾, and the inverse matrix method, which uses an inverse matrix of the transfer function. The dynamic stiffness method directly uses the dynamic stiffness of the physical coordinate system so the effects of changing the structural characteristics and geometry can be easily handled. However, this method is also easily influenced by the stiffness of the mounting frame and other parts that support the outboard motor (outboard motor support parts) and nonlinear characteristics, so it is difficult to achieve accuracy. In contrast, the inverse matrix method tends to be more accurate, but has difficulty handling the effects of changing the structural characteristics and geometry.

A highly accurate transfer path analysis that can easily handle various combinations of outboard motors and boats is required to analyze and implement measures for the vibration and noise of compound systems consisting of outboard motors and boats. For this reason, an attempt was made to experimentally clarify the dynamic characteristics of outboard motor engine mounts⁽⁵⁾. This method consists of performing experimental modal analysis on an outboard motor in the assembled condition, and directly calculating the dynamic stiffness of the physical coordinate system. Establishing this method would make it possible to obtain the dynamic stiffness of engine mounts when the outboard motor's weight and thrust force are applied. This would in turn provide a more accurate understanding of the forces transmitted from the engine to the boat, which would allow more accurate vibration and noise analysis.

2. Engine Mount Dynamic Stiffness Identification Method

Figure 1 shows an outline of the engine mount dynamic stiffness

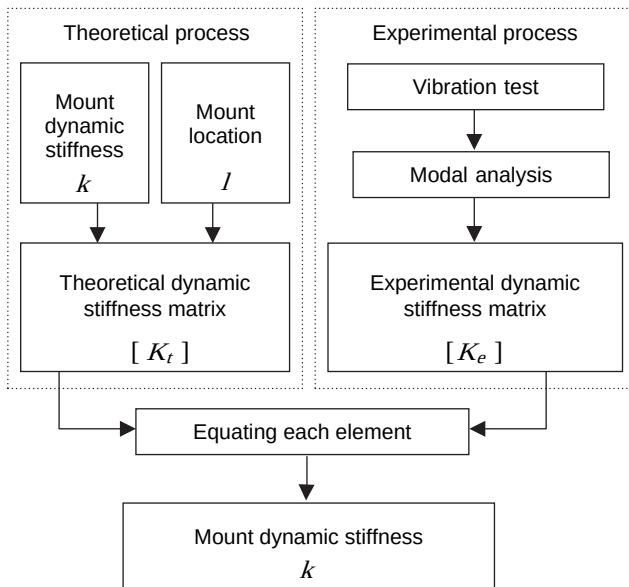


Fig. 1 Outline of engine mount dynamic stiffness identification method

identification method. Here, the dynamic stiffness matrix of the vibration system is calculated both from theoretical equations and through experiments. The dynamic stiffness of each engine mount is then identified by solving the simultaneous equations in which both matrix components are equivalent.

2.1. Calculation of Theoretical Dynamic Stiffness Matrix

The rubber mounts were modeled as shown in Fig. 2. Each rubber piece was divided into the dynamic stiffness for four locations, and the translational stiffness was defined in three directions (k_x, k_y, k_z) for each location. This made it possible to express the rotational stiffness of the mount as translational stiffness.

In addition, both the outboard motor and the mounting frame parts that support the outboard motor are assumed to have a higher stiffness than the rubber mounts. By modeling the parts other than the rubber mounts as a rigid body, the outboard motor supported by the rubber mounts becomes a system with six degrees of freedom. Therefore, the static equilibrium equations can be written as shown in Eq. (1).

$$[K_t] \{u_o\} = \{f\} \quad (1)$$

Where,

$$\{u_o\} = \{x_o \ y_o \ z_o \ \theta_x \ \theta_y \ \theta_z\}^T$$

$$\{f\} = \{f_x \ f_y \ f_z \ T_x \ T_y \ T_z\}^T$$

u_o is the displacement and f is the force.

In addition, the theoretical dynamic stiffness matrix $[K_t]$ can be expressed as shown in Eq. (2).

$$[K_t] = \begin{bmatrix} K_{11} & & & & & \\ 0 & K_{22} & & & & \\ 0 & 0 & K_{33} & & & \\ 0 & K_{42} & K_{43} & K_{44} & & \\ K_{51} & 0 & K_{53} & K_{54} & K_{55} & \\ K_{61} & K_{62} & 0 & K_{64} & K_{65} & K_{66} \end{bmatrix} \quad \text{SYM.} \quad (2)$$

Where,

$$K_{11} = \sum_{i=1}^n k_{xi}, \quad K_{22} = \sum_{i=1}^n k_{yi}, \quad K_{33} = \sum_{i=1}^n k_{zi}$$

$$K_{42} = - \sum_{i=1}^n l_{zi} k_{yi}, \quad K_{43} = \sum_{i=1}^n l_{yi} k_{zi}$$

$$K_{51} = \sum_{i=1}^n l_{zi} k_{xi}, \quad K_{53} = - \sum_{i=1}^n l_{xi} k_{zi}$$

$$K_{61} = - \sum_{i=1}^n l_{yi} k_{xi}, \quad K_{62} = \sum_{i=1}^n l_{xi} k_{yi}$$

$$K_{44} = \sum_{i=1}^n (l_{zi}^2 k_{yi} + l_{yi}^2 k_{zi}), \quad K_{54} = - \sum_{i=1}^n l_{xi} l_{yi} k_{zi}$$

$$K_{55} = \sum_{i=1}^n (l_{zi}^2 k_{xi} + l_{xi}^2 k_{zi}), \quad K_{64} = - \sum_{i=1}^n l_{xi} l_{zi} k_{yi}$$

$$K_{65} = - \sum_{i=1}^n l_{yi} l_{zi} k_{xi}, \quad K_{66} = \sum_{i=1}^n (l_{yi}^2 k_{xi} + l_{xi}^2 k_{yi})$$

Here, K is the dynamic stiffness of the rubber mount, l is the mount position, n is the number of mounts, and the subscript i indicates the mount number. In other words, the theoretical dynamic stiffness matrix $[K_t]$ can be formulated as shown in Eq. (2) by using the rubber mount stiffness k , which is an unknown.

2.2. Calculation of Experimental Dynamic Stiffness Matrix

Next the method for calculating the stiffness matrix $[K_e]$ from the vibration test results is described below.

First, an outboard motor fixed as shown in Fig. 3 is excited, and the natural frequencies and mode shape vectors are obtained for the initial six modes in which the outboard motor vibrates more or less as a rigid body. Assuming the outboard motor to be a rigid body, the mode shape vector for a certain mode can be expressed by three translational components and three rotational components for a total of six degrees of freedom components. Equation (3) shows the relationship between the normalized mode shape vector $\{\phi_i\}$ at the measurement point i and the six degree of freedom components $\{\phi_o\}$ that represent the rigid body movement.

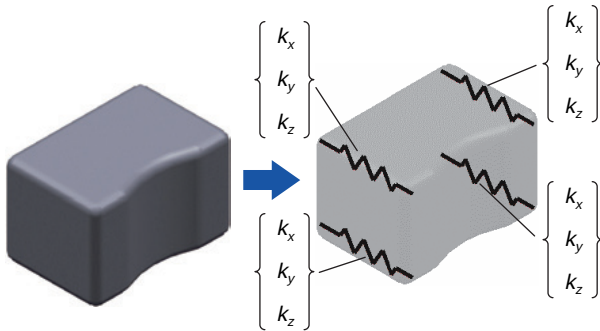


Fig. 2 Modeling of rubber mount



Fig. 3 Outboard motor BF5

$$\{\phi_i\} = [T_i] \{\phi_o\} \quad (3)$$

Where,

$$\{\phi_i\} = \{\phi_{xi} \quad \phi_{yi} \quad \phi_{zi}\}^T$$

$$[T_i] = \begin{bmatrix} 1 & 0 & 0 & 0 & z_i & -y_i \\ 0 & 1 & 0 & -z_i & 0 & x_i \\ 0 & 0 & 1 & y_i & -x_i & 0 \end{bmatrix}$$

$$\{\phi_o\} = \{\phi_{xo} \quad \phi_{yo} \quad \phi_{zo} \quad \phi_{\theta_x} \quad \phi_{\theta_y} \quad \phi_{\theta_z}\}^T$$

Here, $[T_i]$ is the coordinate transformation matrix. x , y and z are the measurement point positions, and i indicates the measurement point number. Equation (3) can be integrated to Eq. (4) for all measurement points. Here, P is the number of measurement points.

$$\{\phi\}_{3P \times 1} = [T]_{3P \times 6} \{\phi_o\}_{6 \times 1} \quad (4)$$

This equation is then transformed to calculate the rigid body mode shape vector $\{\phi_o\}$ as follows.

$$\{\phi_o\} = ([T]^T [T])^{-1} [T]^T \{\phi\} \quad (5)$$

Performing the operations in Eq. (5) for all six modes yields the 6×6 rigid body mode shape vector $[\Phi_o]$.

Finally, the experimental dynamic stiffness matrix $[K_e]$ is calculated from the rigid body mode shape vector $[\Phi_o]$ and the matrix $[\Lambda]$, which is a diagonal matrix of the square of the undamped natural angular frequencies.

$$[K_e] = [\Phi_o^T]^{-1} [\Lambda] [\Phi_o]^{-1} \quad (6)$$

2.3. Calculation of Engine Mount Dynamic Stiffness

Fifteen equations can be obtained by making the non-zero components of the theoretical dynamic stiffness matrix $[K_t]$ in Eq. (2) and the experimental dynamic stiffness matrix $[K_e]$ component in Eq. (6) equal. Rewriting these in the form of simultaneous first-order equations related to the rubber mount dynamic stiffness, which is an unknown, results in the following.

$$[L] \{k\} = \{b\} \quad (7)$$

Where,

$$\{k\} = \{k_{x1} \quad k_{y1} \quad k_{z1} \dots\}^T$$

$$\{b\} = \{K_{e11} \quad K_{e22} \quad K_{e33} \dots\}^T$$

Labeling the number of unknowns as m , the matrix $[L]$ is a $15 \times m$ matrix comprised of the rubber mount position coordinates, $\{k\}$ is a $m \times 1$ row vector comprised of the unknown rubber mount dynamic stiffness, and $\{b\}$ is a 15×1 vector comprised of the Eq. (6) component. In Eq. (7), if m is equal to 15 and $[L]$ has full ranks, the rubber mount dynamic stiffness k can be uniquely calculated.

However, even if m is tentatively set as 15, this matrix $[L]$ may not have full rank in the actual mount layout. In this case, the rubber mount dynamic stiffness is obtained by setting some unknown rubber mount stiffness the same as other ones.

3. Simulations Using an FEM Model

Before applying the newly developed identification method to an actual outboard motor, the validity was checked by applying it to a finite element method (FEM) model consisting of the outboard motor and the engine mounts.

Figure 4 shows the FEM model shape. The outboard motor shape is roughly the same as that of the BF5 outboard motor used for actual motor verification in section 4.1. The rubber mounts support the outboard motor in four upper, lower, right and left locations, and the right and left rubber mounts have the same dynamic stiffness. Eigenvalue analysis and frequency response analysis were performed with the outer sides of the rubber mounts restrained, and the dynamic stiffness of the rubber mounts was identified using the natural frequencies, mode shape vectors and the frequency response obtained at optional points as a result of these analyses.

3.1. Rubber Mount Modeling Method

Each rubber mount is identified as having dynamic stiffness in the X, Y and Z directions, but as described in section 2.3., the matrix $[L]$ may not have full rank depending on the rubber mount layout, so the equations are solved assuming some of the rubber mounts to have the same dynamic stiffness. In this FEM model, this corresponds to the dynamic stiffness of the upper and lower rubber mounts in the Z direction. The upper and lower rubber mount layouts use the same left-right and front-back positions, so there is no relative displacement between the upper and lower rubber mounts in the Z direction. This means that the dynamic stiffness cannot be calculated independently, so the solution was obtained assuming the upper and lower rubber mounts to have the same dynamic stiffness. The same thing could also be said for the dynamic stiffness of the right and left rubber mounts in the Y direction, but the right and left rubber mounts have the same dynamic stiffness in the FEM model. Therefore, there are five rubber mount variables: k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} , and k_{Ly} .

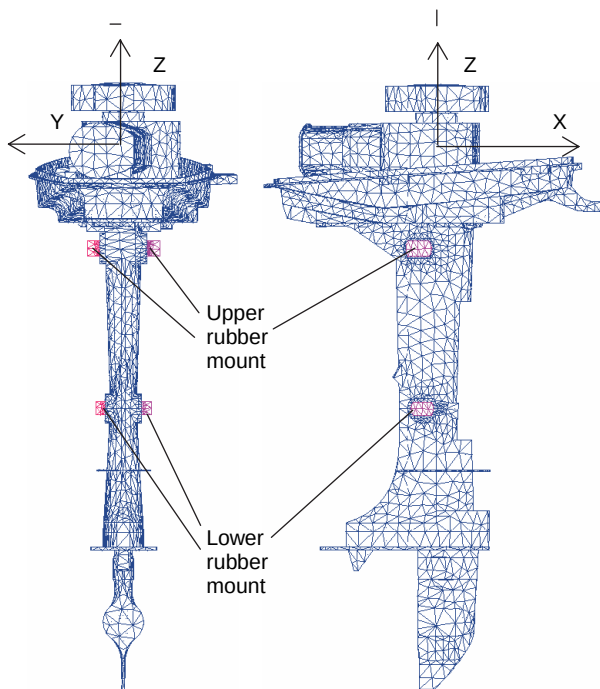


Fig. 4 Schematics of simulation model

Table 1 Results of mount stiffness in simulation study

		Identified	True
Upper mount	k_x	54	54
	k_y	249	261
	k_z	58	49
Lower mount	k_x	72	73
	k_y	351	363
	k_z	58	67

$\times 10^3$ [N/m]

3.2. Identification Results

Table 1 shows the dynamic stiffness of the rubber mounts identified using this method. When evaluated as the total value including the Z direction dynamic stiffness which is assumed to be the same for both the upper and lower rubber mounts, the identified values match closely with the true values, thus indicating that the identification results have sufficient accuracy.

Next, the frequency response function (FRF) at the measurement point was calculated using the obtained dynamic stiffness and the mass characteristics of the outboard motor used in the FEM model to verify the validity of the identification results. Figure 5 shows the FRF at the lower Y direction excitation point of the outboard motor. In this study the upper and lower rubber mounts were identified as having the same dynamic stiffness in the Z direction, and this figure shows that the FRF obtained using these identification values matches the true values closely. This indicates that the dynamic stiffness of the rubber mounts can be obtained using the newly developed method.

4. Application to a Small Outboard Motor

4.1. Structure and Modeling

The newly developed method was first used to identify the dynamic stiffness of the rubber mounts for a small outboard motor (BF5) with a weight of approximately 20 kg.

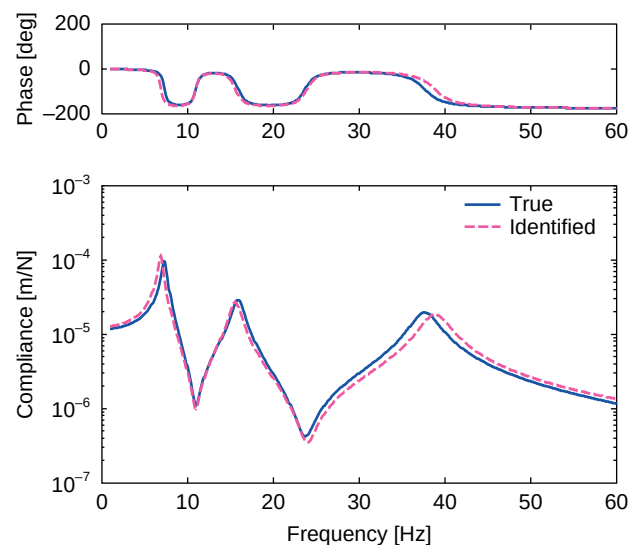


Fig. 5 Fitting of FRFs between identification and FEM model

Figure 3 shows the appearance of this outboard motor, and Fig. 6 shows the rubber mount layout. As seen here, there are two upper rubber mounts and three lower rubber mounts, and the right and left rubber mount layouts are the same. The difference from the FEM model in section 3 above is the addition of the #5 rubber mount in the lower position. This #5 rubber mount is located close to the #3 and #4 rubber mounts, so if the stiffness is obtained independently, the matrix [L] becomes unique, and is thought to result in a large error. Therefore, the #5 rubber mount was included in the #3 and #4 rubber mounts. Like the FEM model, the dynamic stiffness in the Z direction was treated as the same for the upper and lower rubber mounts. Therefore, the five dynamic stiffness values k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} and k_{Ly} were obtained.

4.2. Identification Results

The outboard motor was fixed as shown in Fig. 3, a vibration by hammering test was performed, and the FRF was measured. Figure 7 shows the natural frequencies for the initial six modes obtained from the measurement results, and here the maximum frequency is approximately 50 Hz. In contrast, the 1st elastic mode of the outboard motor is approximately 130 Hz, so the outboard motor is thought to move more or less as a rigid body in the initial six modes. Therefore, it was judged that the dynamic stiffness of the rubber mounts could be calculated using the newly developed method which assumes the outboard motor to be a rigid body.

Table 2 compares the rubber mount dynamic stiffness identification results obtained using the newly developed method and the discrete rubber mount dynamic stiffness measurement results obtained by the conventional method using dynamic stiffness measurement equipment. Next the FRF at the measurement points was calculated from the dynamic stiffness in the same manner as in section 3.2. to verify the identification accuracy. Figure 8 shows the experimental FRF, the FRF obtained from the identification results and the FRF obtained from the discrete dynamic stiffness measured using the conventional method at the lower excitation point of the outboard motor. This shows that the results obtained using both the newly developed method and the conventional method match fairly closely with the experimental results.

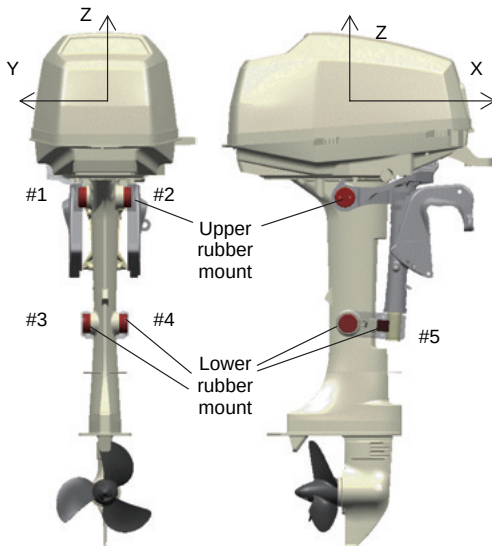


Fig. 6 Schematics of tested outboard motor BF5

Table 2 Results of mount stiffness BF5

		Thrust force		
		0[N]		500[N]
		Identified	Conventional method	Identified
Upper (#1,2)	k_x	52	106	103
	k_y	138	263	181
	k_z	153	101	174
Lower (#3,4)	k_x	414	351	805
	k_y	312	359	432
	k_z	155	211	175

$\times 10^3$ [N/m]

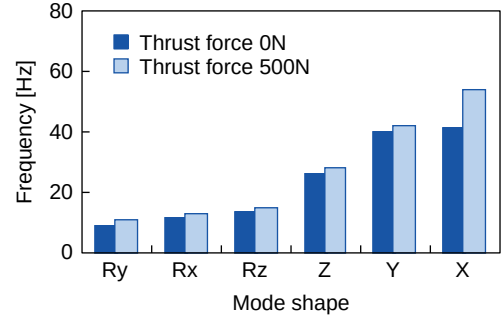


Fig. 7 Natural frequencies

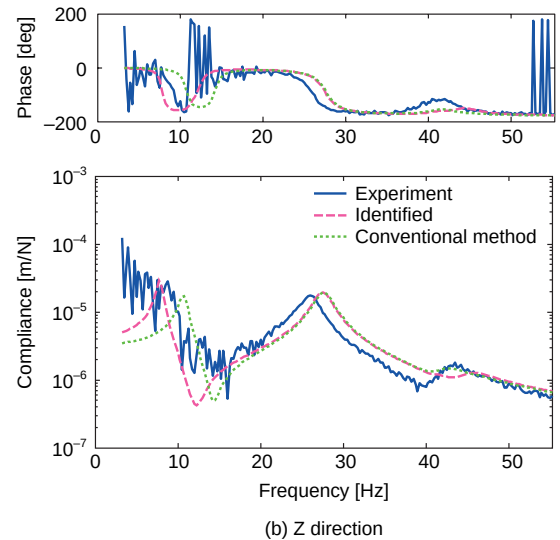
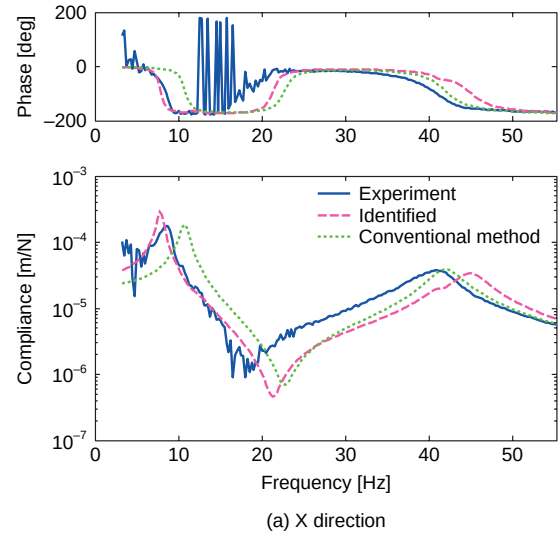


Fig. 8 Comparison of FRFs between identification and experiment

4.3. Change in Rubber Mount Dynamic Stiffness Due to the Application of Thrust Force

As shown above, the newly developed method can identify the dynamic stiffness of the rubber mounts in the condition assembled to an actual outboard motor. This feature can be used to understand changes in the dynamic stiffness of the rubber mounts caused by propeller thrust force, which was formerly not possible. Therefore, excitation tests were performed by applying a static load equivalent to the maximum thrust force (500 N) to the propeller part via a soft rubber piece, and the dynamic stiffness of the rubber mounts was identified.

As shown in Fig. 7, the natural frequency increases somewhat when thrust force is applied. In addition, the dynamic stiffness of the rubber mounts also increases as shown in Table 2. In particular, the dynamic stiffness of the rubber mounts in the X direction changes greatly to approximately double that without thrust force. Next, the FRF was calculated using the identified dynamic stiffness and compared with the experimental values. As shown in Fig. 9, the identification results closely match the experimental values, indicating that changes in the dynamic stiffness due to the application of thrust force are accurately represented.

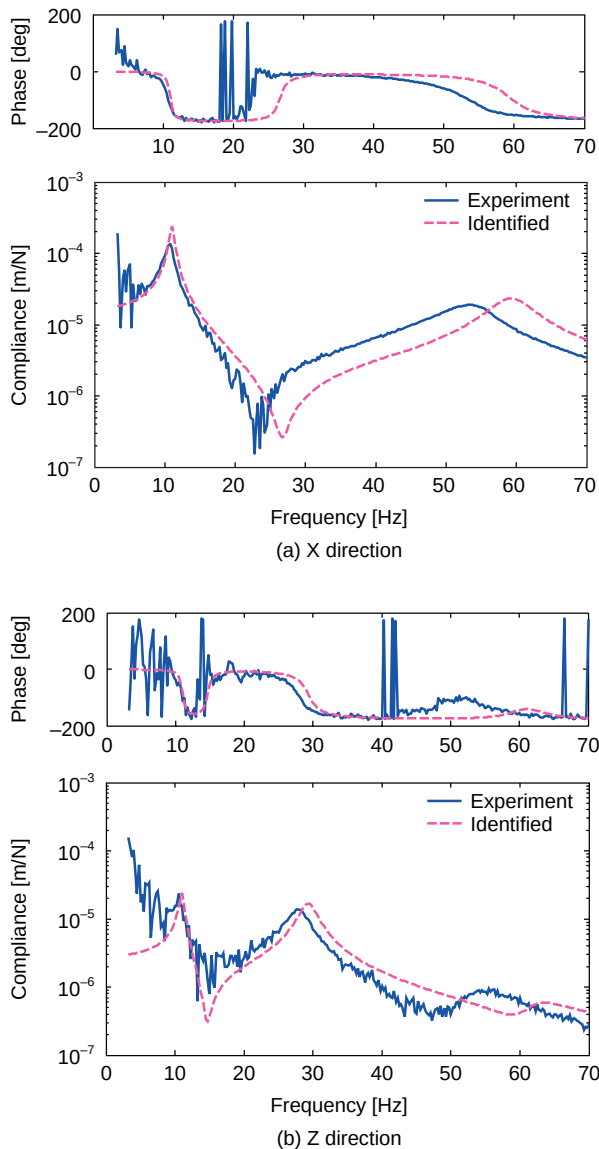


Fig. 9 Comparison of FRFs between identification and experiment

5. Application to a Large Outboard Motor

5.1. Rubber Mount Structure and Modeling

Next, the newly developed method was applied to a large outboard motor with a different size and rubber mount structure to verify the validity once again. The outboard motor used was the BF225, which weighs approximately 270 kg.

Figure 10 shows the outboard motor and the structure of the rubber mount supports used in the experiment. The outboard motor is supported by two upper rubber mounts and two lower rubber mounts that are attached to the mounting frame.

Figure 11 shows the structure of the lower rubber mount. The lower rubber mount deforms as the thrust force increases, and ultimately makes contact with the outboard motor at part A. Therefore, the dynamic stiffness of the lower rubber mount is

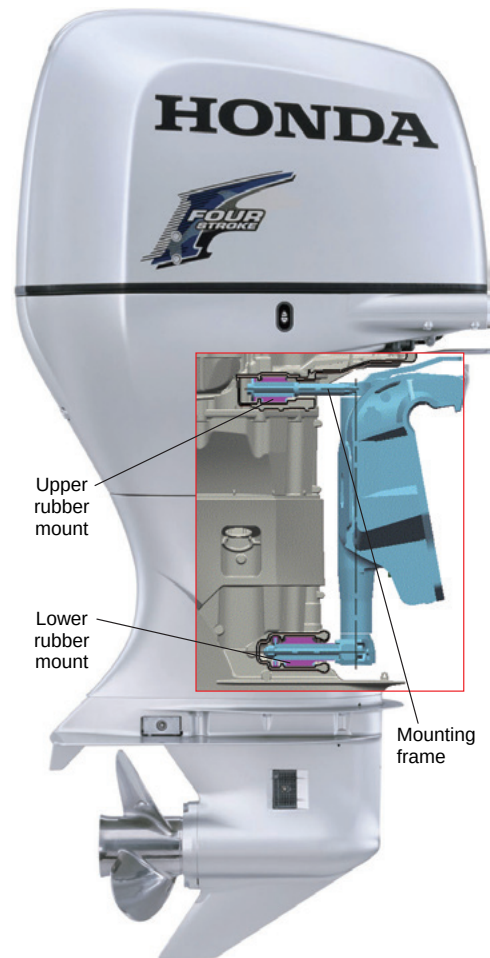


Fig. 10 BF225 side view

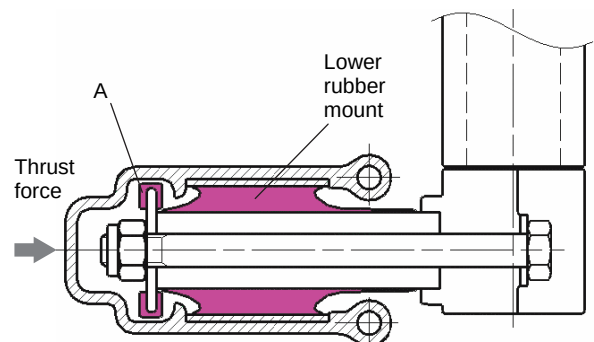


Fig. 11 Section of lower rubber mount

Table 3 Results of mount stiffness BF225

		Thrust force		
		0[N]		8000[N]
		Identified	Conventional method	Identified
Upper (#1,2)	k_x	1550	2280	2940
	k_y	680	14700	850
	k_z	1880	14700	3190
Lower (#3,4)	k_x	1540	1490	15200
	k_y	1100	10930	1690
	k_z	1880	10930	3190

$\times 10^3$ [N/m]

thought to change greatly at the maximum thrust force.

Like the small outboard motor, the dynamic stiffness in the Z direction was treated as the same for the upper and lower rubber mounts. Therefore, the five dynamic stiffness values k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} , and k_{Ly} were identified.

5.2. Identification Results

Table 3 compares the identification results obtained using the newly developed method and the discrete dynamic stiffness obtained by the conventional method using dynamic stiffness measurement equipment. In this case the identification results obtained by the newly developed method greatly underestimated the dynamic stiffness in the Y and Z directions for both the upper and lower rubber mounts. Next, the FRF at the measurement points were calculated from the dynamic stiffness to verify the identification accuracy, and these results are shown in Fig. 12. In addition, Fig. 13 shows the natural frequencies for the initial six modes in which the outboard motor vibrates more or less as a rigid body. The FRF in the X and Z directions obtained using the conventional method both differ greatly from the experimental results, and the natural frequencies for the X, Y and Rx modes are also completely different. In contrast, the FRF calculated from the identification results are more accurate than the conventional method, and there are no significant differences in the natural frequencies for all modes. Therefore, the dynamic stiffness of the rubber mounts obtained by this identification method is thought to be suited for evaluating and analyzing the effects of vibration on the boat body.

5.3. Identification when Thrust Force is Applied

Figure 14 shows the change in the natural frequencies when the maximum thrust force (8000 N) is applied. The natural frequencies increase overall, becoming approximately double in X mode. Table 3 shows the identified rubber mount dynamic stiffness. Here as well the dynamic stiffness of the rubber mounts increases when thrust force is applied. In particular, the dynamic stiffness of the lower rubber mount in the X direction increases by a factor of approximately ten. This is thought to be due to contact with the outboard motor at part A of the rubber mount as described in section 5.1., and confirms that the rubber mount characteristics change greatly when thrust force is applied.

Next, Fig. 15 shows the results of calculating the FRF at the excitation points using the identified dynamic stiffness. Whereas the FRFs based on the discrete dynamic stiffness values measured using the conventional method differ greatly from the experimental values, the FRFs based on the identification results show slight differences in the natural frequencies, while the FRF tendencies match relatively well overall. In other words, when application of

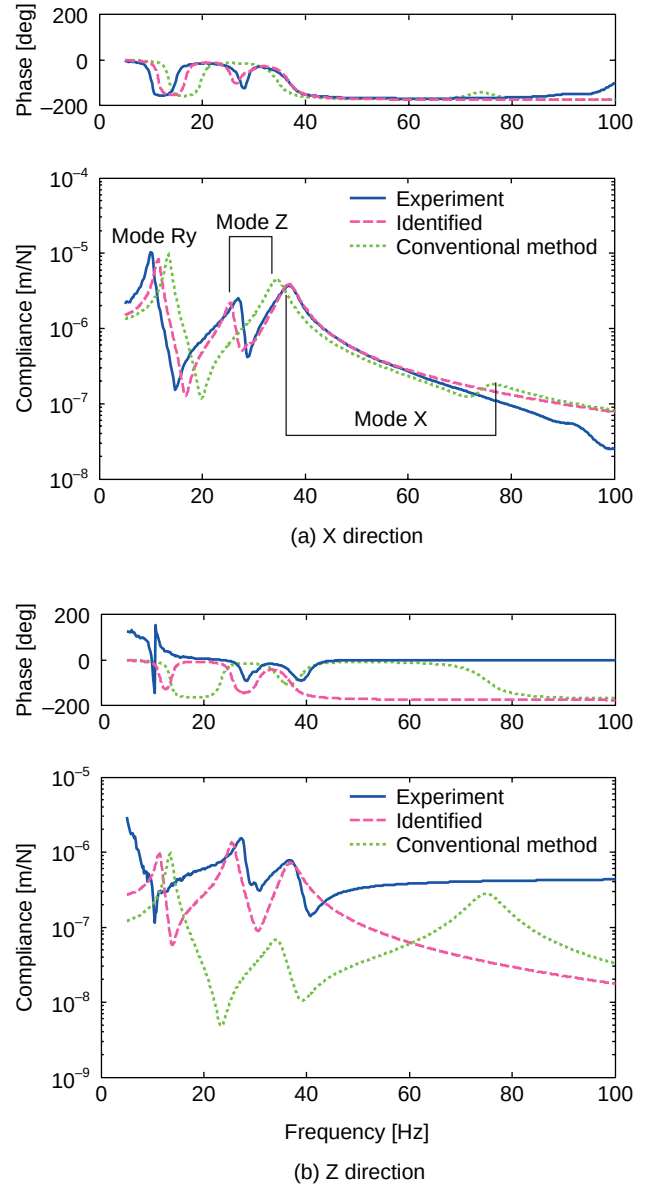


Fig. 12 Comparison of FRFs between identification and experiment

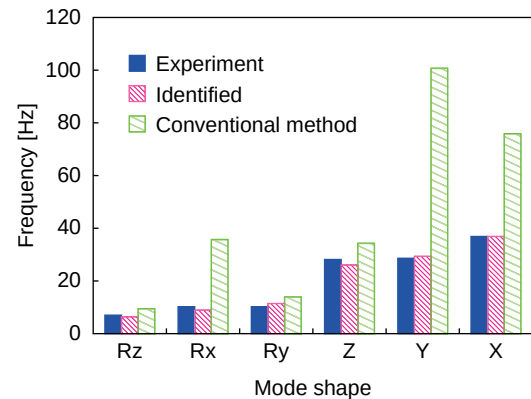


Fig. 13 Natural frequencies

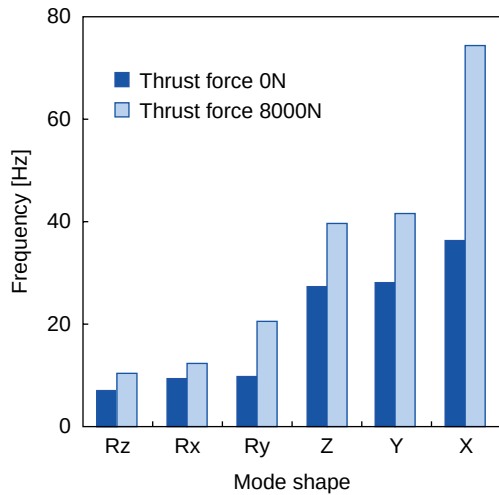


Fig. 14 Natural Frequencies

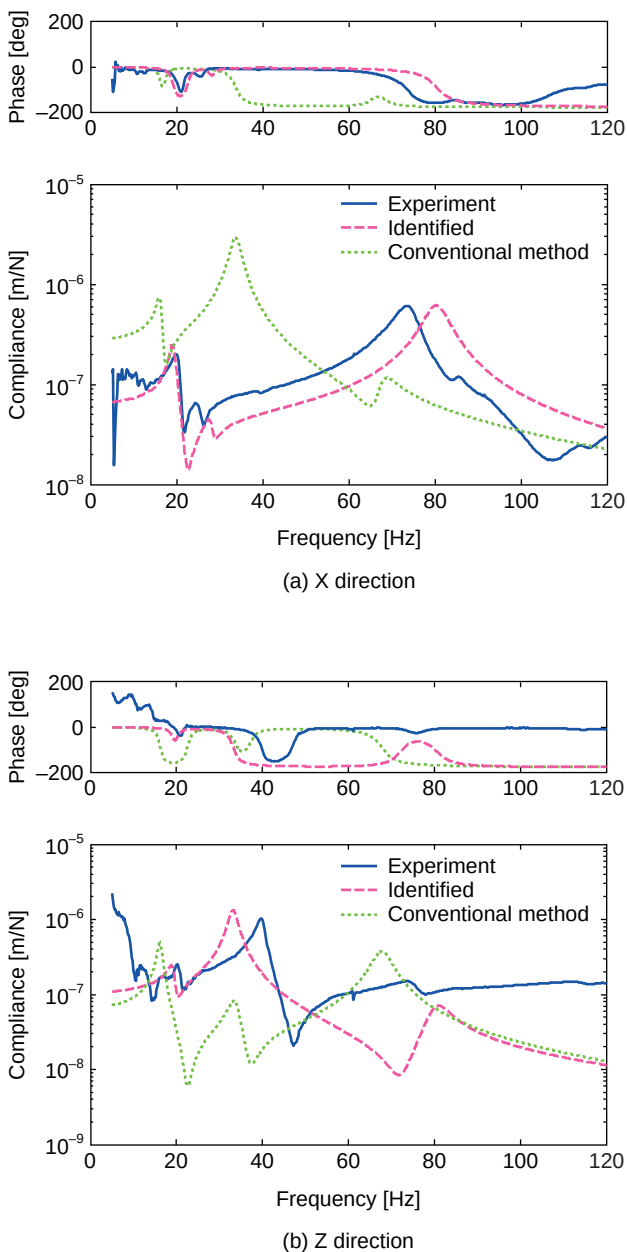


Fig. 15 Comparison of FRFs between identification and experiment

a load produces significant changes in the rubber mount characteristics, it is considered impossible to accurately analyze vibration using the dynamic stiffness obtained by the conventional method.

5.4. Investigation of Differences from the Conventional Method

The BF225 rubber mounts are cylindrical in shape, so they should be soft in the X direction and hard in other directions as shown by the results of the dynamic stiffness tests. However, the identification results obtained by the newly developed method predict an extremely soft Y direction dynamic stiffness, with a stiffness of approximately 1/20 the actual value for the upper rubber mount in particular. It is difficult to think that this great a difference could result simply from assembly onto an actual outboard motor, so the reasons for the difference were investigated.

This identification method assumes that the outboard motor and the outboard motor support parts have sufficient stiffness and act as rigid bodies in the rubber mount resonance range. The natural frequencies of the initial six modes for the rubber mounts are 80 Hz or less even in the condition with thrust force applied, and the 1st elastic mode of the outboard motor is 200 Hz or more, so this is thought to have little effect. However, the stiffness of the outboard motor support parts had not been confirmed prior to the experiment, so confirmation was carried out after the experiment.

Hammering tests were performed on the outboard motor support parts without the outboard motor and the rubber mounts, and the 1st natural frequency was found to be around 100 Hz. However, in actual operation the weight of the outboard motor is applied, so the natural frequency is even lower. In addition, when the outboard motor support parts were also measured during the hammering tests to identify the dynamic stiffness of the rubber mounts, the outboard motor support parts were found to undergo elastic deformation.

In other words, the stiffness of outboard motor support parts is low compared with the rubber mount stiffness, so the identified dynamic stiffness is thought to be not just the dynamic stiffness of the rubber mounts, but the value also including the stiffness of the outboard motor support parts. A future theme is to calculate the dynamic stiffness of only the rubber mounts without the stiffness of the outboard motor support parts, but as described in section 5.3., the vibration of the outboard motor can be expressed using the identified dynamic stiffness. That is to say, the newly developed method is able to identify the overall stiffness of the rubber mounts and the outboard motor support parts, so it is thought to be more practical than the conventional method.

6. Conclusion

A new method was established by identifying the dynamic stiffness of the engine mounts of outboard motors from the results of experimental modal analysis.

- (1) The validity of the newly developed method was confirmed through simulations using an FEM model.
- (2) Vibration tests were conducted using actual outboard motors, and the dynamic stiffness of the mounts was identified. The frequency response functions (FRF) calculated from the identification results were shown to correlate fairly well with the experimental FRF.
- (3) When the outboard motor support parts have a low stiffness, the conventional method that uses the results of dynamic stiffness tests is unable to accurately estimate the vibration characteristics.

- (4) The newly developed method is superior to the conventional measurement method using dynamic stiffness measurement equipment in the following ways.
- Identification is performed using the results of vibration tests conducted on outboard motors in the assembled condition, which makes it possible to identify the dynamic stiffness of the mounts when the outboard motor's weight and thrust force equivalent to actual operation are applied.
 - The dynamic stiffness is identified for the overall outboard motor support parts including the engine mounts, so identification is possible even when the outboard motor support parts have low stiffness.

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■ Authors ■



Takeshi OKADA



Kazuyuki SHIOMI



Tetsuro IKENO