

Reducing Vibration of A 2-Cylinder Outboard Motor by the Pendulum Motion Mount Method

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ABSTRACT

First-order vibration, which an engine generates, has been isolated by applying the pendulum motion mount method to a 4-stroke 2-cylinder outboard motor. First, the vibration of the outboard motor was theoretically analyzed using a two-dimensional kinetic model. Through this analysis, the theory of isolating vibration using the pendulum motion mount method was established by finding the conditions which isolate vibration effectively. Next, this method was applied to the design of an outboard motor. The mount location was optimized in consideration of both the function and the vibration, and the mount structure was designed in consideration of endurance. The body of the outboard motor was designed by optimizing the vibratory characteristics in order to avoid the resonance. The experimental verification using the newly developed 4-stroke 2-cylinder small outboard motor revealed that the effectiveness of vibration reduction was equal to or better than that of an outboard motor equipped with a first-order balancer.

1. Introduction

Outboard motors are mounted as power sources on various types of small boats, and span a wide range of sizes from 2 PS class to over 200 PS. Engines also vary according to the boat size and application from single cylinder types to large scale V-6 type engines. Of these, 2-cylinder engines are mainly used in the 5 PS to 20 PS class, and are mounted on small fishing boats or pleasure boats, etc. Most outboard motors use 2-stroke engines, but the change to 4-stroke engines which have better emissions characteristics and fuel economy is spreading rapidly. 4-stroke 2-cylinder engines normally use a 360° crank which has one explosion per crank revolution.

In this case the exciting force (forces which induce vibration) generated by the engine is dominated by the first-order inertia force and driving torque reaction. Technologies used to reduce vibration include setting a soft mount rubber spring constant to improve the vibration isolating effects, or attaching a reciprocating balancer and rotating balancer to reduce the exciting force itself. The former method generally reduces the support stiffness of outboard motors, so simultaneously ensuring steerability and stability and isolating vibration becomes an issue. The latter method makes it possible to simultaneously isolate vibration and ensure steerability and stability, but adding balancers creates the new issues of increased weight, size, cost and mechanical noise.

In order to solve these problems of 4-stroke 2-cylinder outboard motors, vibration reduction technology for outboard motors was constructed using the pendulum motion mount method based on the theory previously reported in “Isolating Vibration through the Use of the Traveling Center of Engine Oscillation”⁽¹⁾. This paper presents the theory of reducing vibration by the pendulum motion mount method, mount and outboard motor body structure design methods, testing methods and results using the newly developed 4-stroke 2-cylinder small outboard motor BF9.9 as an example.

2. Vibration Isolating Methods and Aim

The outboard motor structure is shown in Fig. 1. The engine is integrally linked to the extension and the gear case, which is the drive portion of the system, to comprise the outboard motor. This outboard motor body is supported by upper and lower mounts which are attached to a swivel shaft (also called the mount frame). Fig. 2 shows the vibration transmission routes of outboard motors. The exciting force of 4-stroke 2-cylinder engines is dominated by the first-order inertia force and driving torque reaction as shown in this figure, and these forces become the vibration sources for boats equipped with outboard motors. The vibration of the excited outboard motor body is transmitted to the swivel shaft via the upper and lower mounts. A handle is integrated into this swivel shaft, and is supported by the swivel case in a manner that allows rotation. Vibration is transmitted to

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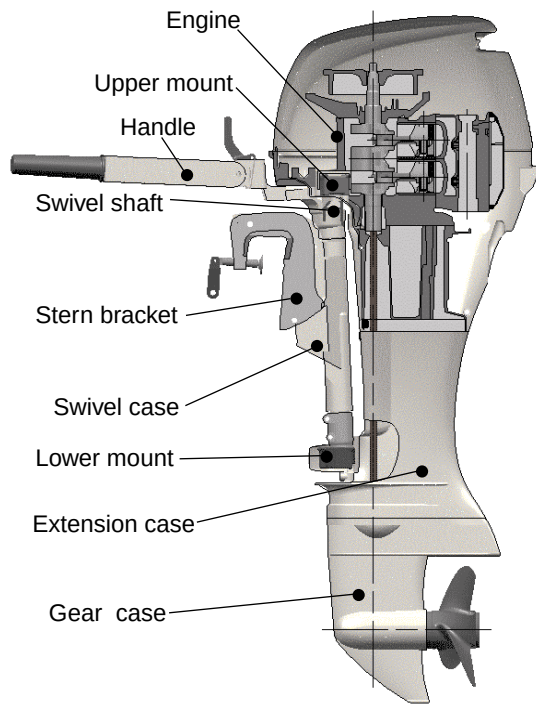


Fig.1 Cutaway view of outboard motor

the hull from the stern bracket to which this swivel case is fixed.

The aim of the pendulum motion mount method is to simultaneously achieve the same effective vibration and high support stiffness as when balancers are added. The basic concept is to generate lateral engine vibration which is perpendicular to the direction in which the thrust and self-weight act. Steerability, stability and reliability are assured by supporting the motor with high stiffness in the directions of the thrust, self-weight and steering effort, and sufficient vibration isolating effects are obtained by supporting the motor with low stiffness in the direction of the vibration. This concept is shown in Fig. 3.

Generating vibration in the lateral direction may adversely affect the hull and handle vibration, so suitable mount placement locations and characteristics must be set. In particular, these engines use a bar handle (hereafter "handle") which is directly supported by the operator's hand, so careful consideration must be given to vibration. In addition, the natural frequency of the outboard motor body in the lateral direction is generally near the normal engine speed, so countermeasures must be taken to eliminate the effects of resonance.

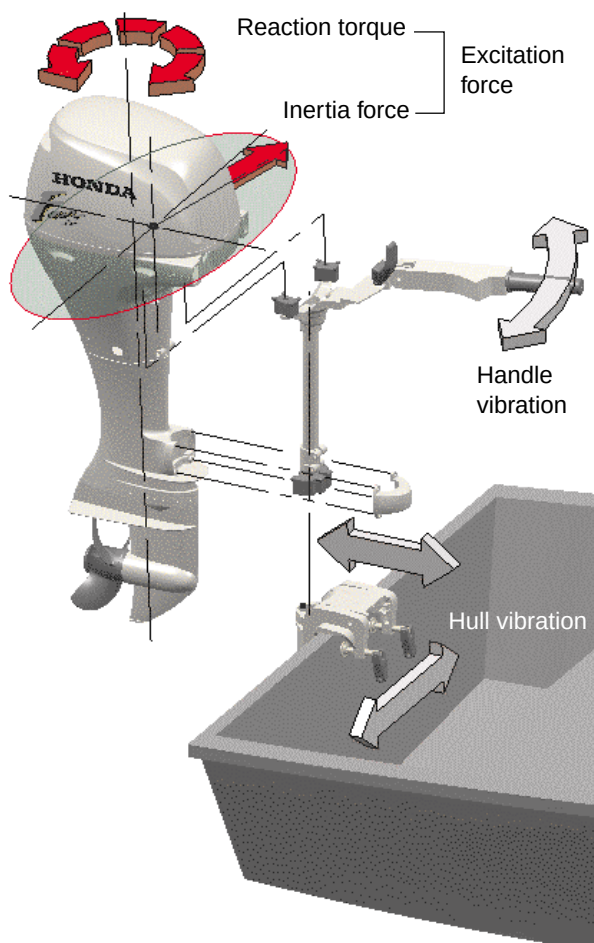


Fig.2 Vibration transmission to hull and handle

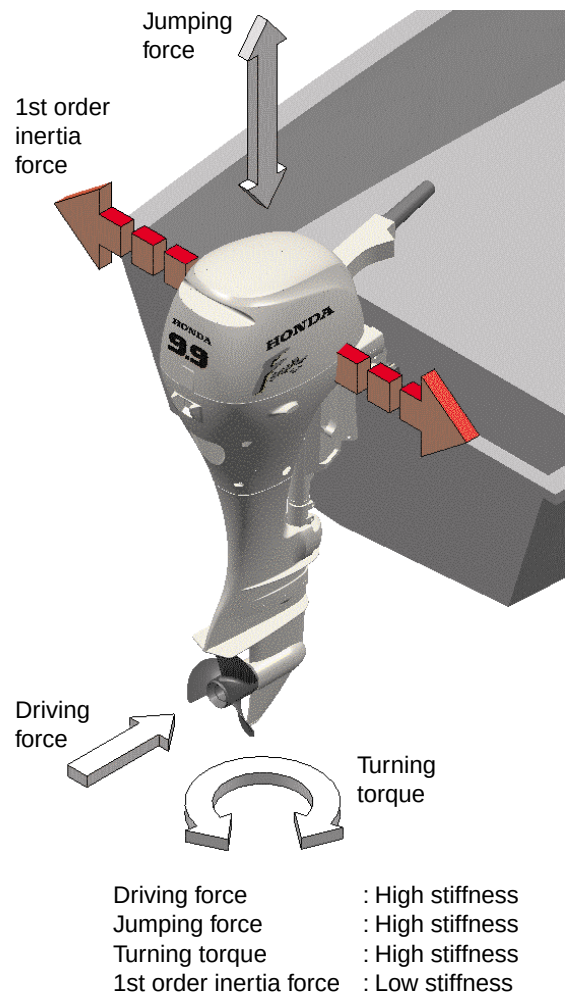


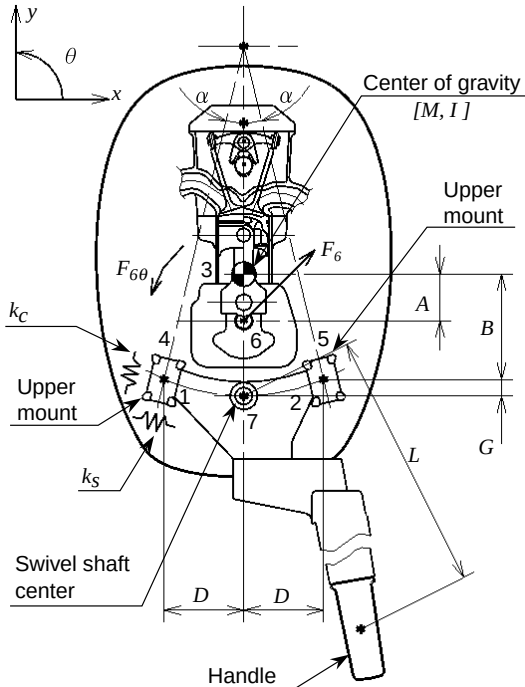
Fig.3 Concept of reducing vibration

3. Theory of Vibration Reduction by the Pendulum Motion Mount Method

The pendulum motion mount method has been reported to provide extremely high vibration isolating effects by generating the first-order inertia force of 4-stroke 2-cylinder engines in a single direction perpendicular to the cylinder axis, and supporting the engine by a mount having a small angle of inclination which satisfies the uncoupled condition.⁽²⁾ In this case, the first-order inertia force and driving torque reaction are dominant and also are approximately the same phase, therefore the engine vibration has a center of rotation (hereafter “center of vibration”) and the engine moves like a pendulum. Vibration is isolated utilizing this motion, so this method is named the pendulum motion mount method. This center of vibration moves according to the load and engine speed.

Fig. 4 shows an analysis model of an outboard motor as viewed from above, and the symbols used in this analysis are as follows.

- U_{ij} : Displacement of each nodal point
(i: Nodal point number, j: Direction)
- F_{ij} : Force acting on each nodal point
(i: Nodal point number, j: Direction)
- M : Engine mass
- I : Engine moment of inertia
- k_s : Spring constant in the shear direction
- k_c : Spring constant in the compressive direction
- α : Mount angle of inclination
- ω : Angular velocity



- Node number
- 1, 2 : Mount point at swivel shaft
- 4, 5 : Mount point at engine
- 3 : Center of engine gravity
- 6 : Rotating center of crankshaft
- 7 : Rotating center of swivel shaft

Fig.4 Two dimensional vibration analysis model

When the pendulum motion mount method is applied to outboard motors, the first-order inertia force produced by the engine is generated in the x direction in Fig. 4, so the outboard motor body should be supported from the swivel shaft by an engine mount having a small angle of inclination which satisfies the uncoupled condition. The swivel shaft is generally combined with the handle and attached to the stern bracket in a manner which allows rotation. The swivel case has a bolt for adjusting this friction, and increasing the friction constrains the rotating direction vibration of the swivel shaft. The following analysis was performed under these conditions, and handle vibration was evaluated by the constraining torque.

3.1. Exciting Force on the Hull

This section investigates the effects of the inertia force and driving torque reaction produced by the engine on the outboard motor and hull vibration. Structurally, the outboard motor has a small moment of inertia around the vertical axis compared to the moment of inertia around the other two axes, and exciting force is not generated in the vertical direction. Therefore, the outboard motor body was assumed to be a 2-dimensional rigid body, and its vibratory properties were investigated theoretically. The equation of motion without damping for the vibration system shown in Fig. 4 is as follows.

$$\begin{bmatrix} M & & \\ & M & \\ & & I \end{bmatrix} \begin{bmatrix} \ddot{U}_{3x} \\ \ddot{U}_{3y} \\ \ddot{U}_{3\theta} \end{bmatrix} + \begin{bmatrix} 2k_x & 0 & 2Bk_x - 2Dk_{xy} \\ & 2k_y & 0 \\ SYMM. & & 2B^2k_x - 4BDk_{xy} + 2D^2k_y \end{bmatrix} \begin{bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{bmatrix} = \begin{bmatrix} F_{6x} \\ F_{6y} \\ AF_{6x} + F_{6\theta} \end{bmatrix} \quad (1)$$

Where,

$$\begin{bmatrix} k_x \\ k_y \\ k_{xy} \end{bmatrix} = \begin{bmatrix} \cos^2 \alpha & \sin^2 \alpha \\ \sin^2 \alpha & \cos^2 \alpha \\ -\sin \alpha \cos \alpha & \sin \alpha \cos \alpha \end{bmatrix} \begin{bmatrix} k_s \\ k_c \end{bmatrix} \quad (2)$$

The uncoupled condition is shown in equation (3).

$$Bk_x = Dk_{xy} \quad (3)$$

At this time, the equation of motion (1) becomes as follows.

$$\begin{bmatrix} M & & \\ & M & \\ & & I \end{bmatrix} \begin{bmatrix} \ddot{U}_{3x} \\ \ddot{U}_{3y} \\ \ddot{U}_{3\theta} \end{bmatrix} + \begin{bmatrix} K_x & 0 & 0 \\ & K_y & 0 \\ SYMM. & & K_\theta \end{bmatrix} \begin{bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{bmatrix} = \begin{bmatrix} F_{6x} \\ F_{6y} \\ AF_{6x} + F_{6\theta} \end{bmatrix} \quad (4)$$

Where,

$$\begin{bmatrix} K_x \\ K_y \\ K_\theta \end{bmatrix} = \begin{bmatrix} 2k_x \\ 2k_y \\ -2B^2k_x + 2D^2k_y \end{bmatrix} \quad (5)$$

The frequency response in equation (5) is obtained as follows.

$$\begin{Bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{Bmatrix} = \begin{Bmatrix} \frac{F_{6x}}{-M\omega^2 + K_x} \\ \frac{F_{6y}}{-M\omega^2 + K_y} \\ \frac{AF_{6x} + F_{6\theta}}{-I\omega^2 + K_\theta} \end{Bmatrix} \quad (6)$$

On the other hand, the fulcrum reaction force in the swivel shaft constrained condition is expressed by the following equation.

$$\begin{Bmatrix} F_{7x} \\ F_{7y} \\ F_{7\theta} \end{Bmatrix} = \begin{bmatrix} -K_x & 0 & 0 \\ 0 & -K_y & 0 \\ (B+C)K_x & 0 & -K_\theta \end{bmatrix} \begin{Bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{Bmatrix} \quad (7)$$

In this study, the first-order inertia force produced by the engine is generated only in the x direction, so F_{7y} in equation (7) is 0. This shows that the exciting force F_{7x} on the hull becomes smaller as K_x becomes smaller. Therefore, extremely high vibration reduction effects with respect to hull vibration can be obtained by using a mount with a small angle of inclination.

3.2. Moment Exciting the Handle

The moment $F_{7\theta}$ which excites the handle is shown in equation (8).

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x)\omega^2 + 1} - \frac{A}{-(I/K_\theta)\omega^2 + 1} \right) F_{6x} - \frac{1}{-(I/K_\theta)\omega^2 + 1} F_{6\theta} \quad (8)$$

Furthermore, the properties of the moment $F_{7\theta}$ which excites the handle were investigated in detail by dividing equation (8) into the moment at low and high engine speeds as follows.

3.2.1. Moment Exciting the Handle at Low Engine Speeds

$AF_{6x} < F_{6\theta}$ at low engine speeds, so F_{6x} in equation (8) is assumed to be 0.

$$F_{7\theta} = -\frac{F_{6\theta}}{-(I/K_\theta)\omega^2 + 1} \quad (9)$$

In other words, this shows that as the rotating stiffness K_θ approaches 0, the exciting force on the handle also approaches 0. The following relationship is obtained from the uncoupled condition equation (3) and the condition where the rotating stiffness $K_\theta = 0$

$$k_x k_y = k_{xy}^2 \quad (10)$$

Equation (11) is obtained by substituting equation (2) into equation (10).

$$k_s k_c = 0 \quad (11)$$

The spring constant k_c in the compressive direction is generally a larger value than the shearing spring constant k_s , indicating that mount rubber with a small shearing spring constant is necessary. Setting $k_s = 0$ and substituting equation (2) into the uncoupled condition equation (3) results in the following relational equation.

$$(B \sin \alpha - D \cos \alpha) \sin \alpha = 0 \quad (12)$$

The above equation indicates the following two conditions.

(Condition I) $\tan \alpha = D/B$

(Condition II) $\alpha = 0, D = 0$

Condition I is the conventional inclined mount. Condition II is the mount with a small angle of inclination, and corresponds to the ultimate form when $k_c \gg k_s$. When using an engine mount with a small angle of inclination, the mount span D should be set as small as possible within the range that does not impair steerability and stability. Furthermore, in 4-stroke 2-cylinder engines $AF_{6x} = F_{6\theta}$ is true near the idle range, which means $U_{3\theta} = 0$. Therefore, it is predicted that handle vibration at low engine speeds can be vastly reduced by using a pendulum motion mount.

3.2.2. Moment Exciting the Handle at High Engine Speeds

$AF_{6x} \gg F_{6\theta}$ at high engine speeds, so $F_{6\theta}$ in equation (8) is assumed to be 0.

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x)\omega^2 + 1} - \frac{A}{-(I/K_\theta)\omega^2 + 1} \right) F_{6x} \quad (13)$$

Equation (13) can be transformed as follows.

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x) + 1/\omega^2} - \frac{A}{-(I/K_\theta) + 1/\omega^2} \right) \frac{F_{6x}}{\omega^2} \quad (14)$$

Assuming that $F_{7\theta} = 0$ in equation (14) is the condition required to cancel $F_{7\theta}$ at high engine speeds, the following equation is obtained.

$$\frac{K_x}{M} = \frac{A}{B+C} \frac{K_\theta}{I} \quad (15)$$

In other words, if the natural frequency in the translational direction can be matched with the frequency obtained by multiplying the natural frequency in the rotational direction by $\sqrt{A/(B+C)}$, the handle vibration at high rotation speeds can also be canceled.

3.3. Summary of Theoretical Analysis Results

Theoretical analysis showed that the pendulum motion mount method can reduce both handle vibration and hull vibration at all engine speeds. These results are summarized below.

- (1) The pendulum motion mount method can effectively reduce vibration caused by inertia force transmitted to the hull.
- (2) Vibration caused by the driving torque reaction at low engine speeds can be isolated equally by applying the conventional inclined mount or the pendulum motion mount method. In addition, with the pendulum motion mount method, the exciting force is reduced by having the driving torque reaction and inertia force cancel each other out near the idle range, thus vastly reducing vibration.

- (3) Handle vibration and the moment of excitation to the hull caused by the inertia force at high engine speeds can be canceled by applying the pendulum motion mount method.

4. Applied Design for Outboard Motors

4.1. Structural Design of the Engine Mount

As stated above, the BF9.9 design aimed to maximize vibration isolating performance while ensuring support stiffness with respect to self-weight, thrust and steering effort. Support stiffness with respect to thrust generally tends to be proportional to the product of the mount spring constant and the distance between the upper and lower mounts. On the other hand, with the pendulum motion mount method, the vibration isolating performance tends to increase as the lateral spring constant decreases. Therefore, the structure was provided with a long distance between the upper and lower mounts by lowering the lower mount within the range where problems with water resistance or spray do not occur. As a result, the partial load acting on each mount decreased. This makes it possible to set low mount spring constants while still maintaining the same support stiffness.

The lower mount shown in Fig. 5 was designed as a one-piece mount in consideration of water resistance and splashing conditions. The upper mount had a somewhat large mount span in consideration of steerability and stability, and used an angle of inclination α such that the mount rubber faced roughly toward the position of the center of percussion I/AM.

The mount rubber was selected to give the upper and lower mounts nonlinear spring characteristics to further reduce vibration at low speeds. Fig. 6 shows the upper mount structure. Here, rough-textured rubber is encased in a baked housing using a ribbed mandrel. Fig. 7 shows the nonlinear characteristics obtained by this structure.

4.2. Structural Design of the Outboard Motor

The theory of vibration isolating support described in Chapter 3 assumes that the outboard motor body, that is to say the sprung portion, is a rigid body. An outboard motor body has long vertical structures, so the natural frequency of the outboard motor body easily approaches the normal engine speed. If the engine speed approaches the resonance point and the outboard motor body oscillates as an elastic body instead of a rigid body, the above-mentioned pendulum motion mount method theory is not applicable. Furthermore, this vibration is transmitted via the mounts to the handle and the hull. This vibration causes a deterioration of the vibration feeling. Therefore, the authors focused on the fact that natural frequencies and natural vibration modes greatly affect outboard motor body vibration, and designed the structure to prevent the outboard motor body from resonating.

4.2.1. Natural Vibration Mode of the Outboard Motor Body

Methods for minimizing the effects of resonance were investigated. A general multi-degree of freedom equation of motion without damping is shown below.

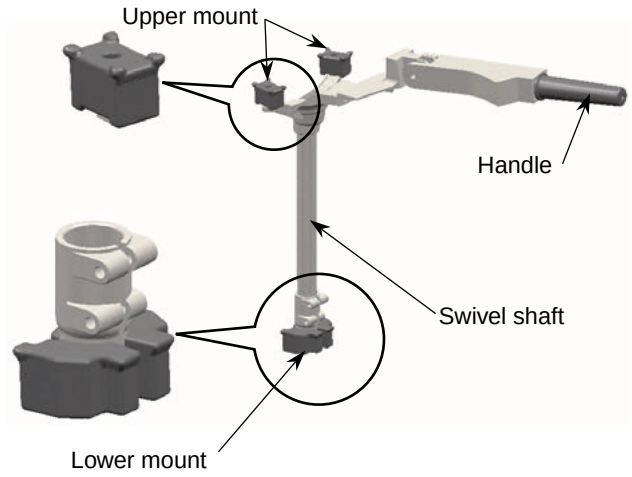


Fig.5 Structure around mount

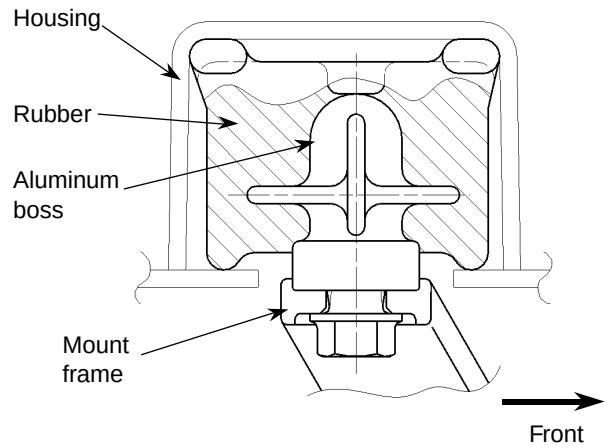


Fig.6 Section of upper mount

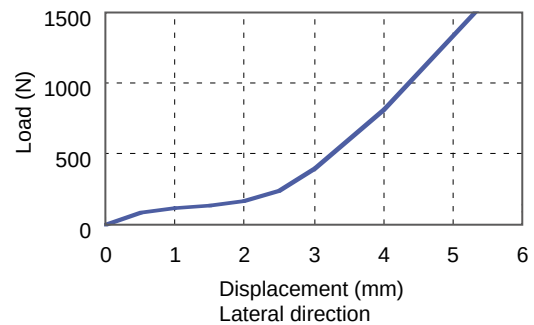


Fig.7 Spring rate of upper mount

$$[M]\{\ddot{U}\} + [K]\{U\} = \{F\} \quad (16)$$

Here, $[M]$ is the mass matrix, $[K]$ is the stiffness matrix, $\{U\}$ is the displacement vector, and $\{F\}$ is the exciting force vector.

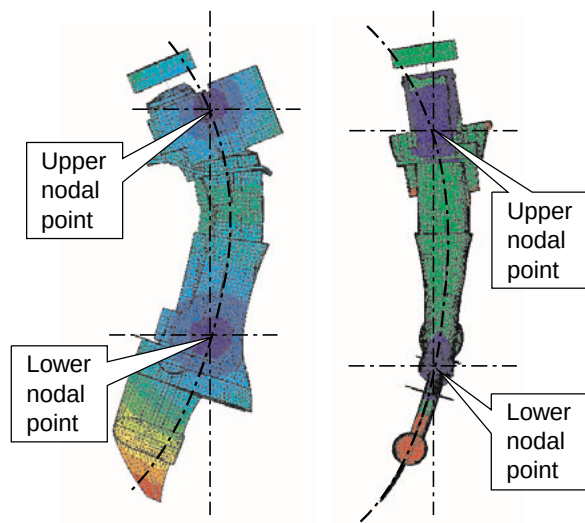
The above equation was converted to an equation of motion with modal coordinates to focus on the resonance of first-order lateral bending. The following is a generally known equation.⁽³⁾

$$[m_r]\{\ddot{\xi}_r\} + [k_r]\{\xi_r\} = [\phi]^{-1}\{F\} \quad (17)$$

Here, $[m_r]$ is the modal mass matrix, $[k_r]$ is the modal stiffness matrix, $\{\xi_r\}$ is the modal displacement vector, and $\{\phi\}$ is the natural vibration mode vector. The right side of equation (17) shows that the amplitude of vibration for that natural vibration mode is 0 when the product of the transposition of a certain natural vibration mode vector and the exciting force vector is 0. This is the principle for canceling resonance.

Fig. 8 shows the BF9.9 sprung bending first-order natural vibration mode calculated by the finite element method. The colors in the figure indicate the amplitude contour of these modes. The centers of the blue circles indicate the node positions for that mode. An outboard motor body is a continuous body, so there are an infinite number of natural vibration modes. However, the natural vibration mode shown in Fig. 8 with the lowest frequency has the greatest influence on vibration feeling. In case of the pendulum motion mount method, the exciting force is generated only in the lateral direction, so the authors focused on the lateral bending natural vibration mode shown in Fig. 8(b). This natural vibration mode has nodal points in the engine and gear case portions. The exciting force acts in the same phase at the center of the two cylinders, so according to the above mentioned resonance canceling principle, resonance should not occur if these nodal points are located approximately in the center of the two cylinders. This enables ideal elastic support with the outboard motor body moving approximately as a rigid body.

While developing the BF9.9, efforts were made to locate these nodal points close to the center of the two cylinders. For the final prototype specifications, eigenvalue analysis was performed using the finite element method for the S, L and UL specifications each possessing different extension case lengths. Table 1 shows the distance between the upper nodal point of the lateral bending mode and the center of the two cylinders. All three specifications placed the upper nodal point for lateral bending slightly above the center point of



(a) Longitudinal bending mode (b) Lateral bending mode

Fig.8 Shape of fundamental modes of the eigenvector

Table 1 Distance between upper nodal point and center of two cylinders

Type	Distance (m)
S (Short)	5.3×10^{-2}
L (Long)	4.5×10^{-2}
UL (Ultra long)	3.3×10^{-2}

the two cylinders which is the exciting force center, but prospects for achieving relatively small values were obtained.

4.2.2. Natural Frequency of the Outboard Motor Body

An additional method for eliminating the effects of outboard motor body resonance was to raise the natural frequency to a level that is much higher than the normal engine speed. This can be achieved by increasing the stiffness and reducing the mass of the outboard motor body.

In conventional outboard motors, the mount is often located near the center of gravity, but with the pendulum motion mount method the mount can be positioned much farther forward. This made it possible to use a structure whereby the secondary-moment of the extension case lower mount was improved a significant 2.5 times over the previous BF9.9 model. This structure is shown in Fig. 9. In addition, eliminating the balancers reduced the weight by approximately 2 kg which also contributed to increasing the natural frequency.

As a result, the lateral bending first-order natural frequency of the outboard motor body was increased to 218 Hz, which is over 1.5 times the 139 Hz of the conventional model.

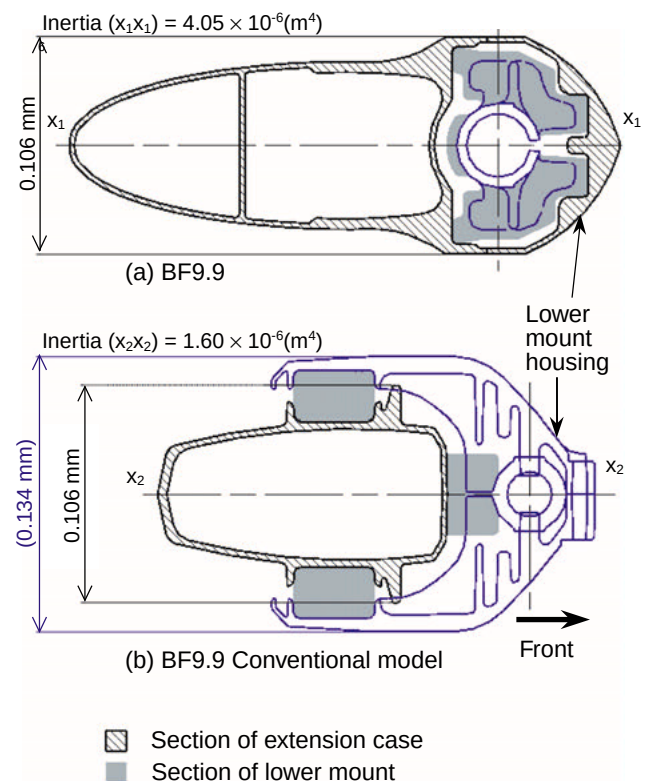


Fig.9 Comparison of section around lower mount

4.2.3. Evaluation of Outboard Motor Body Vibration

Frequency response analysis by MSC/NASTRAN was performed to investigate the degree to which natural vibration mode influences outboard motor body vibration. Outboard motor body vibration is a combination of rigid body vibration and natural vibration mode vibration as shown in Fig. 10.

The lateral acceleration value was divided into the vibration components of the rigid body vibration and natural vibration mode, and the contribution of each component at the normal engine speed was calculated. Table 2 shows the lateral acceleration at the mount positions for the S, L and UL specifications. At the upper mount, the vibration component of the natural vibration mode was kept to 5% or less of the total. This confirmed that the effects of the lateral bending first-order natural vibration mode had fallen to a level which could be largely ignored, and that the outboard motor body vibration had approached that of a rigid body.

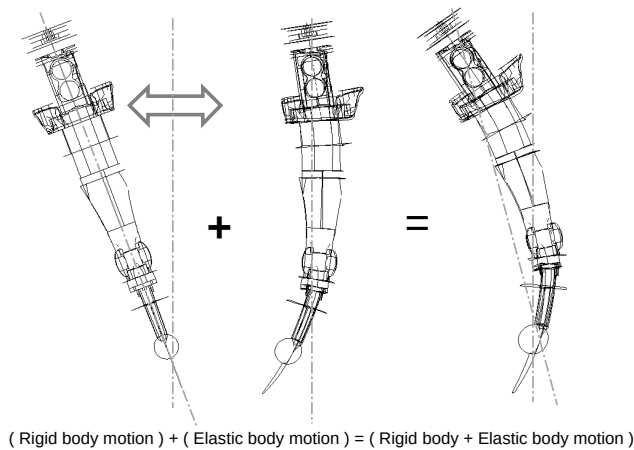


Fig.10 Lateral vibration of outboard motor

Table 2 Influence of elastic body motion lateral acceleration at mount points of S, L, and UL types (m/sec²)

Type	Mount	Rigid body motion	Elastic body motion	Total
S	Upper	120	-3	117
	Lower	64	-8	56
L	Upper	121	-4	117
	Lower	51	-10	42
UL	Upper	121	-4	118
	Lower	53	-14	39

5. Testing Methods and Results

Fig. 11 shows a photograph of the newly developed 4-stroke 2-cylinder outboard motor BF9.9 to which the pendulum motion mount method is applied. Vibration was compared using the conventional 4-stroke 2-cylinder outboard motor BF9.9. The conventional BF9.9 cancels the first-order inertia force generated by the engine and converts

it into a first-order inertia couple by adding a dual axis balancer to the top of the crankcase. Vibration was also measured without this dual axis balancer. Table 3 shows the basic specifications for the new and conventional BF9.9 models.

Vibration was measured using a 16ft. FRP Japanese boat. The conditions were gradual acceleration while cruising in a straight line on calm water; and, acceleration pickups were attached to the handle and seat for measurements. Fig. 12 shows the results for the first-order component obtained by performing vibration level analysis relative to engine speeds on the measurement data.

Handle vibration was almost the same as the outboard motor with the balancer at medium to low speeds, but was reduced by 50% at high speeds. In addition, significant vibration reducing effects were obtained at all operating speeds compared to the boat without the balancer. Hull vibration was also about the same as for the boat with the balancer, but significant vibration reducing effects were obtained compared to the boat without the balancer. Thus, the pendulum motion mount method was verified to significantly reduce both hull and handle vibration.



Fig.11 BF9.9

Table 3 Specification

Model	BF9.9 New model	BF9.9 Conventional model
Bore × Stroke (mm)	58 × 42	58 × 53
Displacement (cm ³)	222	280
Maximum engine output (kW/r/min)	7.4/6000	7.4/5500
Maximum Torque (N · m/r/min)	14.9/4000	16.6/3000
Reciprocating mass (kg)	0.386	0.349
Sprung mass (kg)	36	39

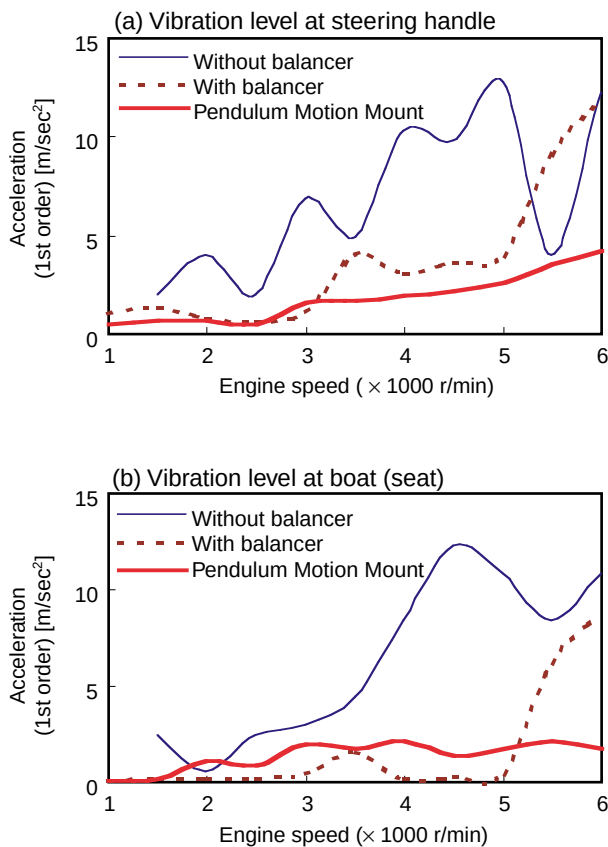


Fig.12 Effect of vibration reduction by Pendulum Motion Mount

6. Conclusions

Vibration reduction using the pendulum motion mount method was applied to a 4-stroke 2-cylinder outboard motor. The theory of isolating vibration was clarified, an actual outboard motor was manufactured, and the following results were obtained.

- (1) Hull and handle vibration when inclined mounts are applied to the outboard motor was clarified through 2-dimensional theoretical analysis.
- (2) Using a pendulum motion mount was theoretically shown to effectively isolate 4-stroke 2-cylinder outboard motor vibration at all engine speeds.
- (3) The conditions for canceling handle vibration caused by inertia force were clarified.
- (4) Outboard motor body and mount structures capable of realizing the pendulum motion mount method were presented.
- (5) An outboard motor based on this technology was newly developed, and vibration reduction effects exceeding those for a conventional outboard motor with a balancer were achieved.

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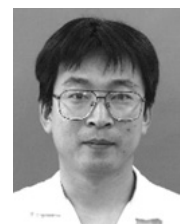
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