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Force identification of an outboard engine by experimental means of linear structural modeling and equivalent force transformation

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Abstract

This paper presents a study of experimental identification of vibration force generated in a single cylinder outboard engine under a steady-state operation by means of linear structural modeling and equivalent force transformation. The identification is to determine a set of equivalent force vectors that can express actual vibration force generated by firing and inertia force due to dynamic motion of moving components in the engine. The vibration force generated in engines under steady-state operation will be independent of the boundary conditions excluding load condition. Therefore, once such vibration force can be modeled with practical accuracy, the result can be used as a database for CAE simulations of trouble shooting and design improvement analysis for noise and vibration, etc.
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1. Introduction

Accurate modeling and identification of vibration force are essentially important for vibration analysis as well as accurate modeling of mechanical structures including mechanisms and the boundary conditions. Force identification is still an important engineering research topic today. The most basic approach is analytical modeling of inertia force which is generated due to accelerated and decelerated motion of moving components as functions of machine. In this case, mechanical components are normally dealt with as rigid body components, the relative motion of which can create dynamic mechanical function. However, in reality, the identification of vibration force to a flexible mechanical structure cannot be dealt with by such a basic approach. In most cases, the force must be distributed-force with some degrees of uncertainty. In addition, the mass of a structure and the acceleration at the center of mass of the structure cannot work as only two factors

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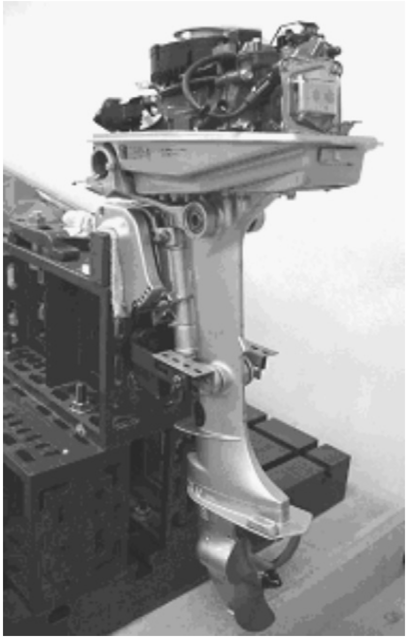

Fig. 1. Tested outboard engine.

to determine vibration force. Therefore, as more practical engineering approach, experimental identification has been studied by many researchers. Many studies were already reported. A few examples of them are listed [1–4], in which more references can be found.

The identification in this paper is carried out in the form of equivalent vibration force representation. The vibration force due to firing and dynamic motion of piston-crank mechanism under a steady-state operational condition will be independent of its mounting boundary condition except for load condition. Therefore, once such vibration force can be modeled appropriately, the resultant data can be used as database for numerical simulations in CAE for noise and vibration design improvement and trouble shooting, etc. For example of systems consisting of outboard engines, suspension systems and boat hulls, the vibration and noise estimation analysis can be realized for any case of changing the combination of outboard engine, suspension system, rubber mounts and boat hull under same load conditions as kept in the database of vibration force.

This paper presents a study of experimental identification of vibration force generated in an outboard engine under a steady-state operational condition by means of linear structural modeling and equivalent force transformation. The outboard engine is shown in Fig. 1. The identification is to determine a set of equivalent force vectors that can express actual vibration force generated by firing and the inertia due to dynamic motion of piston-crank mechanism and others. The result shows that the fundamental harmonic is composed of both translational and rotational force components and that the 1.5 harmonics is dominated by rotational component.

2. Theory and procedure

First, an appropriate number of measurement points are allocated on the whole surface of a single cylinder outboard engine whose maximum output power is 5 HP in order to deal with it as a flexible test structure having multi degrees of freedom. Note that at least three of all the measurement points should be allocated in the part of the cylinder block that can be considered to behave as almost a rigid body in the frequency range of interest, i.e., up to 200 Hz, for the identification. Vibration force is identified at practically reasonable points in the cylinder block as an equivalent force that can simulate actual vibration response of the engine. SIMO frequency response functions at the measurement points are obtained by vibration testing such as hammering test. Modal parameters are obtained for the frequency range using the FRFs by the modal analysis.

Next, the engine runs at a constant rotational speed, and the vibration response at all the measurement points are simultaneously measured in time domain and transformed into frequency domain. The inverse matrix method is then applied to determine a set of equivalent force at all the measurement points. Note here that it is conceptually very important to realize that the force identification is an engineering modeling method to represent real distributed force by the form of some number of concentrated forces. In this study, real distributed force for vibration is considered to be generated by firing. Then, at first, it is assumed that only the measurement points on the cylinder block play a role as the points directly applied the external vibration force. According to the abovementioned modeling, the simultaneous equations of the correlation between measured vibration response in firing and the vibration force at the measurement points are expressed by

$$\mathbf{y}(\omega) = \mathbf{H}(\omega)\mathbf{f}(\omega), \tag{1}$$

where $\mathbf{y}(\omega)$ is the vibration response in vector form in a steady-state operation of the engine, and $\mathbf{H}(\omega)$ is the frequency response function matrix formulated using the modal parameters identified by the experimental modal analysis. The dimension is $\mathbf{H}(\omega) \in R^{(n\times m)}$. Normally $n = 3 \times p$ where p is the number of all the measurement points, and $m = 3 \times q$ where q is the number of the measurement points in the part of the cylinder block. And $\mathbf{f}(\omega)$ is the force vector whose components correspond with all the freedom of the measurement points in the part of the cylinder block. n is then the number of total degrees of freedom of the measurement points where vibration response are measured in both cases of vibration testing for obtaining modal parameters and firing operational vibration. Each measurement point has three degrees of translational freedom, i.e., x-, y-, and z-directional freedom in a Cartesian coordinate system. In general case, n is much larger than m. The least mean square method is applied to solve the equations with respect to the force vector. That is,

$$\mathbf{f}(\omega) = (\mathbf{H}(\omega)^h\mathbf{H}(\omega))^{+}\mathbf{H}(\omega)^h\mathbf{y}(\omega), \tag{2}$$

where $(\mathbf{H}(\omega)^h\mathbf{H}(\omega))^{+}$ is the generalized inverse matrix.

Under the assumption that the part of the cylinder block can behave as a rigid body in the frequency range under the constant running, the vibration force in frequency domain at the actual measurement points in the part of the cylinder block are transformed into vibration force at the intersection between the center line of the crankshaft and the connecting rod equivalently using the geometric data. The method determines a set of equivalent force consisting of the translational force and the moment of force at the intersection point to express the actual vibration force. Equivalent force transformation is easily computed as follows. Let us use an example using Fig. 2. Eq. (2) obtains the equivalent force generating vibration response of the engine in the form of concentrated translational force components distributed at the measurement points on the cylinder

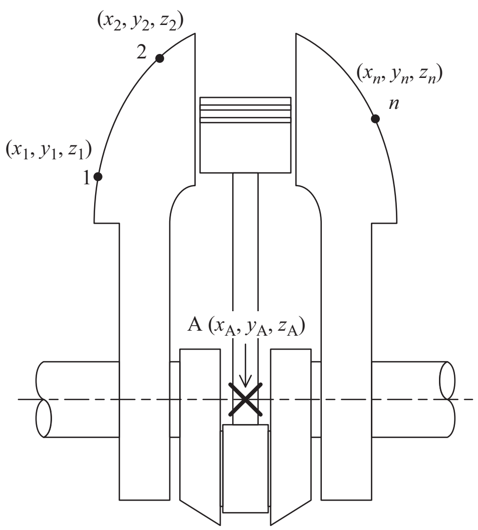

Fig. 2. Schematics for explanation about geometric transformation.

block. Let us denote the coordinates of the measurement points No.1 to No.q by $(x_i, y_i, z_i)(i = 1 \sim q)$ and the coordinate of the point at the intersection between the center line of the crankshaft and the connecting rod by (x_A, y_A, z_A). Under the assumption that the cylinder block oscillates as a rigid body, the geometrical relation can formulate the equation of the force transformation from the translation force components at the actual measurement points into three translational and three moments of force components at the intersection A by the following equation:

$$\begin{pmatrix} F_{Ax} \\ F_{Ay} \\ F_{Az} \\ N_{Ax} \\ N_{Ay} \\ N_{Az} \end{pmatrix} = \begin{bmatrix} 1 & 0 & 0 & \cdots & 1 & 0 & 0 \\ 0 & 1 & 0 & \cdots & 0 & 1 & 0 \\ 0 & 0 & 1 & \cdots & 0 & 0 & 1 \\ 0 & -(z_1 - z_A) & y_1 - y_A & \cdots & 0 & 0 & y_n - y_A \\ z_1 - z_A & 0 & -(x_1 - x_A) & \cdots & z_n - z_A & -(z_n - z_A) & -(x_n - x_A) \\ -(y_1 - y_A) & x_1 - x_A & 0 & \cdots & -(y_n - y_A) & x_n - x_A & 0 \end{bmatrix} \begin{pmatrix} F_{1x} \\ F_{1y} \\ F_{1z} \\ \vdots \\ F_{nx} \\ F_{ny} \\ F_{nz} \end{pmatrix}. \tag{3}$$

With respect to any rigid body motion as same as about force transfer formulation expressed by Eq. (3), Eq. (4) can be formulated to express the relationship of infinitesimal displacement between the translational components at arbitrary points from No.1 to No.n and six degrees of freedom of a particular point A.

$$\begin{pmatrix} \delta_{1x} \\ \delta_{1y} \\ \delta_{1z} \\ \vdots \\ \delta_{nx} \\ \delta_{ny} \\ \delta_{nz} \end{pmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & z_1 - z_A & -(y_1 - y_A) \\ 0 & 1 & 0 & -(z_1 - z_A) & 0 & x_1 - x_A \\ 0 & 0 & 1 & y_1 - y_A & -(x_1 - x_A) & 0 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\ 1 & 0 & 0 & 0 & z_n - z_A & -(y_n - y_A) \\ 0 & 1 & 0 & -(z_n - z_A) & 0 & x_n - x_A \\ 0 & 0 & 1 & y_n - y_A & -(x_n - x_A) & 0 \end{bmatrix} \begin{pmatrix} \delta_{Ax} \\ \delta_{Ay} \\ \delta_{Az} \\ \theta_{Ax} \\ \theta_{Ay} \\ \theta_{Az} \end{pmatrix}, \tag{4}$$

where δ and θ denote translational and rotational displacement, respectively.

Then, a new point can be created as a pseudo measurement point computationally. In this paper, using the measurement of vibration response at three points on the cylinder block in the outboard engine, the vibration response at the intersection between the center line of the crankshaft and the connecting rod is created computationally using Eq. (4). Identified equivalent force vectors at the three points on the cylinder block are also transfered to the intersection by Eq. (3). The modeling of the outboard engine is then expressed in the form of modal model with respect to nine actual measurement points and one computational measurement point at the intersection. Since six measurement points are placed in the flexibly behaving part of the outboard engine in the frequency range up to 200 Hz, it will be valuable to investigate whether the transformations are available for the force identification and the simulation modeling of vibration. This paper demonstrates the successful application result about the outboard engine to show the availability.

3. Result and discussion

Fig. 3 is a schematics of measurement point distribution on the tested engine. The number of measurement points is nine. Three points are placed on the part of the cylinder block, and other six points are on the cover structure from the cylinder part till the end of the extensional drive shaft part. Equivalent force vectors are determined at three measurement points on the cylinder block. Then, the resultant force is transformed into the translation force component and the moment of force component at the intersection between the center line of the crankshaft and the connecting rod.

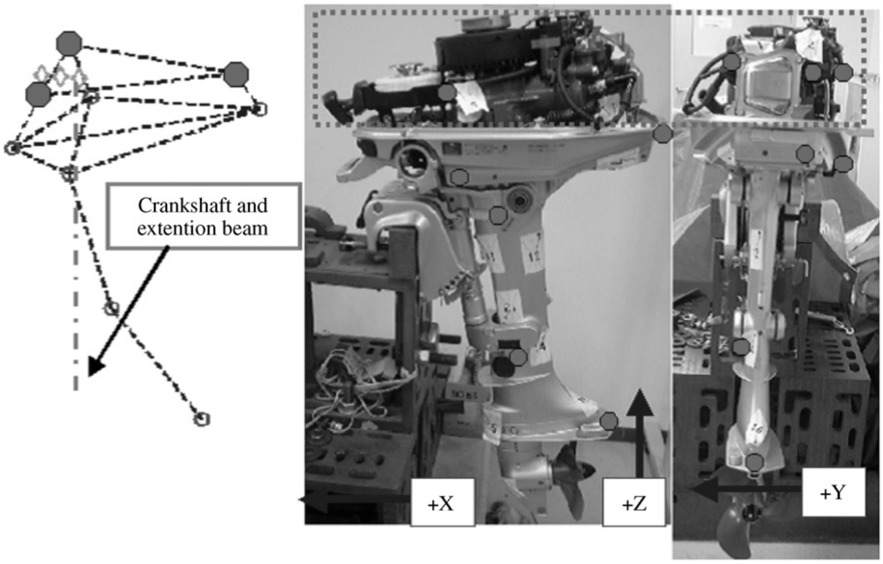

Fig. 3. Schematics of measurement point location.

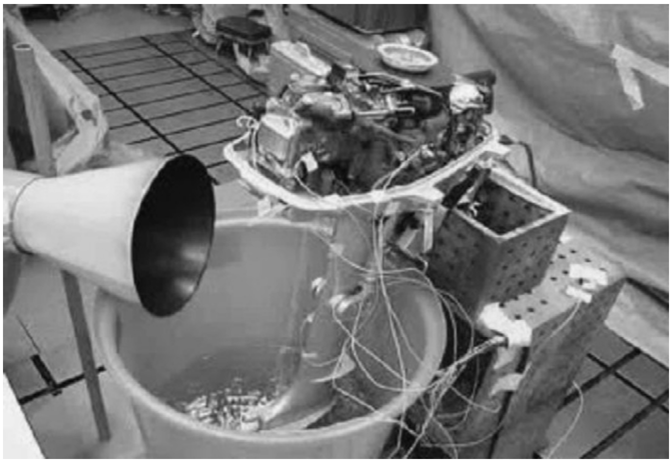

Fig. 4. Vibration measurement of tested engine under a constant speed at 2600 rev/min.

Fig. 4 shows an appearance of vibration measurement under a constant running at 2600 rev/min. Fig. 5 shows the resultant force identified at the intersection with respect to the constant engine operation. The upper graph displays three translational force components with respect to x-, y- and z-direction. The lower graph does three moments of force components about x-, y- and z-axis. The biggest force spectrum appears at 44 Hz, which is the frequency of the primary fundamental. The following features can be found in the figures. The first harmonic spectrum consists of both translational and the moment of force. The one-half order sub-harmonic spectrum consists of almost only the moment of force, i.e., the moment of force about z-axis. The second harmonic dominantly consists of translational force. The resultant feature qualitatively accords with one derived based on the theory of engine dynamics. Fig. 6 shows the comparison of vibration responses at three of nine measurement points between actually measured data and the computed result using the identified force.

The solid lines denote experimental data and the dotted lines denote computational result using the identified equivalent force at the intersection. The computational result fits very well with the experimental data. Note that very good fitting of vibration responses are obtained about all other measurement points as

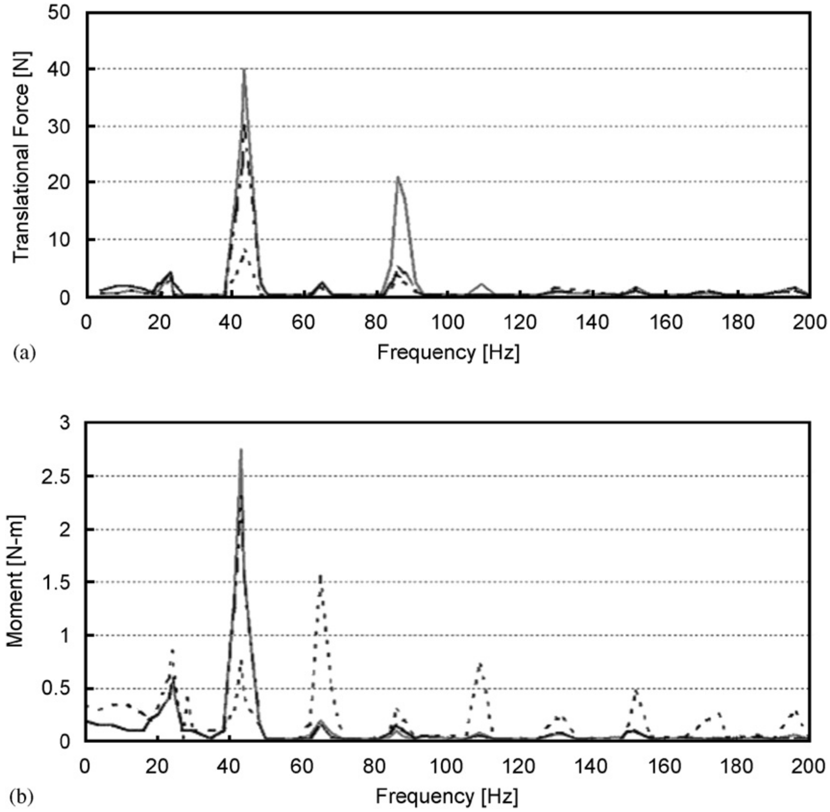

Fig. 5. Identified equivalent force at a point of interest under a constant speed at 2600 rev/min: (a) translational force, (b) moment, —— x-direction, -.-.- y-direction, ----- z-direction.

well as ones shown in Fig. 6. It is verified that the vibration behavior of the outboard engine due to firing force can be accurately simulated using the identified force at the intersection. Although no doubt the equivalent force transfer expressed by Eq. (3) is true within a rigid body, it is not guaranteed that the vibration response is accurately simulated for any point of a flexible structure having a part behaving as a rigid body by using the equivalent force transfer and the infinitesimal displacement transfer formulation expressed by Eq. (3). Because Eq. (3) can be applied only within a rigid body. The abovementioned result of this application demonstrates that the simulation is available.

The experimental based equivalent force identification method is simply composed of the inverse matrix method and the geometric transformation method. Nevertheless, it can realize equivalent force identification at points of interest, in cylinder block although the real vibration cannot be measured at them. This paper shows that the experimental method is simple and capable for many practical applications.

In order to make a database of such vibration forces in practice, the abovementioned procedure of identification is carried out under an appropriate number of different engine rotational speed and thrust load conditions at screw in water. Vibration and structural-borne noise of cruisers having outboard engines are dominantly due to engine vibration as well as other types of cruisers. The main structural parts will be thought as three, i.e., a boat hull, suspension rubber mounts and an outboard engine. The combination of these structural elements can dominate the vibration characteristics of total structural system as "cruiser". Using the database together with appropriate computational models of these structural elements, computational prediction of vibration and noise can be done. It will be used to assist engineering design and customer decision to find favorable combination of cruiser main structural elements from the view point of noise and vibration quality.

Strictly, since the outboard engine is running as a moving mechanics powered by firing force, the modeling must become very complex with nonlinear characteristics, etc. Nevertheless, the simple linear model

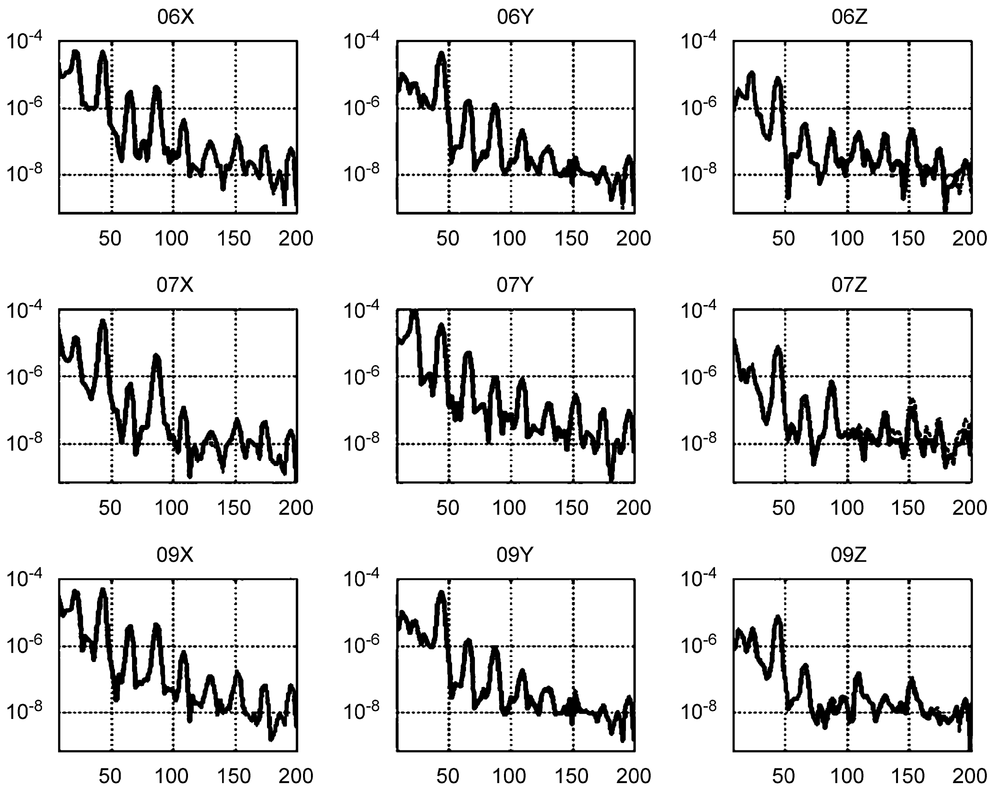

Fig. 6. Comparision of vibration responses between experiment and computational result under a constant speed at 2600 rev/min, ······ experiment, —— computational.

approximation in this paper produces the force identification with practically acceptable accuracy. This paper shows only a piece of relatively basic application result. However the authors think it will inform the availability of this kind of simple technique for practical use.

4. Conclusions

In this paper, the authors presented an experimental based force identification method, and showed a test result at 2600 rev/min of a constant running operation of a single cylinder outboard engine whose maximum output power is 5 HP. Real vibration force was identified in the form of the equivalent force consisting of translational force and the moment of force at the intersection between the center line of the crankshaft and the connecting rod. The identified force can re-generate the vibration response of the same running condition of the engine in the frequency range up to 200 Hz under dealing with as a flexible structure.

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