A Study on the Modeling and Dynamic Characteristics of the Viscous Damper Silicone Fluid Using Vibration Control of Engine Crankshaft Systems

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***Abstract*—As the silicone fluid that is inside torsional viscous friction dampers (hereafter called "viscous dampers") is a non-Newtonian fluid, the effective viscosity in the actuation is different under the operating conditions, and the dynamic characteristics are complex. In this paper, the authors discuss the modeling and dynamic characteristics of torsional damping and springs of viscous dampers through an experimental investigation by adopting a simultaneous vibration measurement method at two points. This method was found to be very effective for having a good grasp of the torsional behaviors of the damper inertia ring and the damper casing in order to clarify the effect of the silicone fluid on the torsional vibration of the engine crankshaft system. In the estimation of the torsional damping coefficient and spring constant of the viscous damper from the experimental results, the damper part is treated as a one-degree-of-freedom equivalent vibration system, and its viscous friction part can be assumed to be replaced approximately with a concentrated equivalent linear damper and spring between the concentrated equivalent masses consisting of the damper inertia ring and the casing, including the pulley part.**

***Index Terms*—Silicone fluid, viscous damper, modeling, measuring method, dynamic characteristics, torsional spring constant, damping coefficient, crankshaft system.**

I. Introduction

Silicone oil (e.g., dimethyl silicone fluid) is a non-Newtonian fluid that has a complex effective viscosity that differs according to the operating conditions [1–9]. When the viscosity of the silicone oil filling increases, there will be increases in not only vibration damping, but also complex damping that combines elasticity and damping. In this research, an engine equipped with viscous dampers is operated, and the torsional vibrations of the inertia ring of the viscous damper and the damper casing are measured simultaneously (hereinafter, the simultaneous two-point measurement method) [4, 5]. Since the silicone fluid inside torsional viscous dampers is a non-Newtonian fluid, the effective viscosity in actuation is different from that under the operating conditions, and the dynamic characteristics are complex. Here, the authors took into account the dynamic characteristics of torsional damping, and the springs of viscous dampers were experimentally investigated in this study using the simultaneous two-point measurement method [10–14]. In this paper, the modeling and dynamic characteristics of the torsional damping and springs of the silicone fluid inside the viscous dampers were first investigated by changing the moments of inertia of the inertia ring, the viscosity of the silicone fluid, and the clearance between the damper inertia ring and the casing.

II. Main Specifications of the Viscous Damper and Test Engine

A. Specifications of the Test Engine

The major specifications of the test engine are shown in **Table I**. The specifications of the viscous damper are referred to in the following section.

B. Shape Dimension and Specifications of the Viscous Damper

In addition to considering the damper shape that is used in current high-speed diesel engines, the standard shape and dimensions are established, as shown in **Fig. 1**. A transparent acrylic resin pervious to light is used for a section of the viscous damper casing in order to carry out simultaneous two-point measurements on the damper casing and the inertia ring. The shape and dimensions of the test viscous damper are based on the aforementioned design damper shape and dimensions in order to investigate the vibration characteristics of the viscous

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damper. Six circumferential direction clearance dimensions determined by the casing's internal dimensions and the inertia ring's outer diameter, and six axial direction clearance dimensions determined by the casing's side thickness and the inertia ring's thickness are used for a total of eleven inertia rings created by combining these dimensions.

The specifications of these inertia rings are shown in **Table II**. Furthermore, the kinematic viscosity of the silicone oil in the test viscous damper is 0.10 m^2/s as a standard, and lower and higher values of 0.05 m^2/s and 0.30 m^2/s, respectively, are used to give a total of three viscosities. **Table III** shows the specifications of the silicone oil.

TABLE I. MAIN SPECIFICATIONS OF THE TEST ENGINE.

Particulars	Contents
Designed for	High-speed diesel engines
Type for	Four–stroke cycle, direct injection
Number of cylinders	Six–cylinders
Arrangement	In–line
Bore and stroke (m)	0.105–0.125
Total piston displacement (m^3)	0.006469
Compression ratio	17.0
Maximum brake output (kW/r/min)	230/3200
Maximum torque (Nm/r/min)	451/1800
Firing order	1–5–3–6–2–4

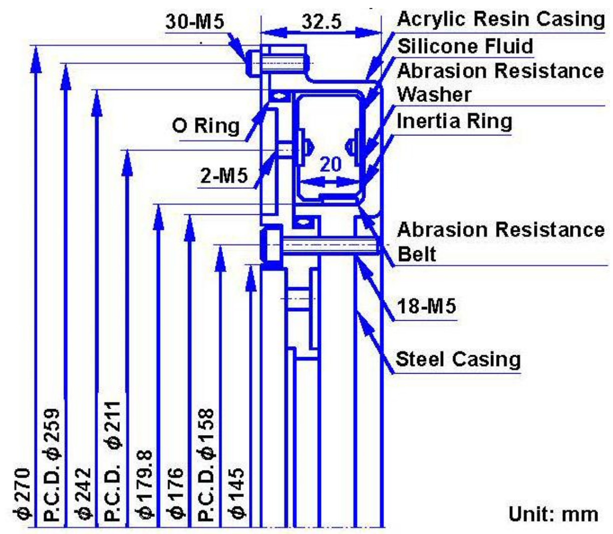

Figure 1. Shape and dimensions of the viscous damper.

III. MODELING OF THE VISCOUS DAMPER SILICONE FLUID

A two degree-of-freedom torsional vibration system of an engine crankshaft system is replaced with a viscous damper with an equivalent torsional vibration model, as shown in **Fig. 2**. The vibration modeling of the viscous damper consists of an inertia ring part, a silicone oil part, and a damper casing part, as shown in Fig. 2. The inertia ring part and the damper casing part are assumed to be rigid elements. The viscous damper silicone fluid part consists of dashpot and an equivalent spring considering spring effect.

TABLE II. DIMENSIONS OF THE DAMPER INERTIA RING.

Number of inertia ring	Moment of inertia $\times 10^{-2}$ (kgm^2)	Thickness $\times 10^{-2}$ (m)	Outside radius $\times 10^{-1}$ (m)	Inside radius $\times 10^{-2}$ (m)
No. 01-L	3.256	1.940	1.205	9.040
No. 02-S*	3.200	1.900	1.205	9.040
No. 03-L	3.029	1.800	1.205	9.040
No. 04-L	2.882	1.700	1.205	9.040
No. 05-L	2.723	1.600	1.205	9.040
No. 06-L	2.564	1.500	1.205	9.040
No. 01-P	3.229	1.900	1.207	9.040
No. 02-S*	3.200	1.900	1.205	9.040
No. 03-P	3.115	1.900	1.200	9.040
No. 04-P	3.031	1.900	1.195	9.040
No. 05-P	2.960	1.900	1.190	9.040
No. 06-P	2.890	1.900	1.185	9.040

L: change of lateral gap, peripheral gap: constant; P: change of peripheral gap, lateral gap: constant. *Designed standard viscous damper.

TABLE III. NUMBER OF SILICONE FLUID AND KINEMATIC VISCOSITY OF SILICONE FLUID.

Number of silicone fluid	Kinematic viscosity (m^2/s)
No. 0 1	5.0×10^{-2}
No. 0 2	1.0×10^{-1}
No. 0 3	3.0×10^{-1}

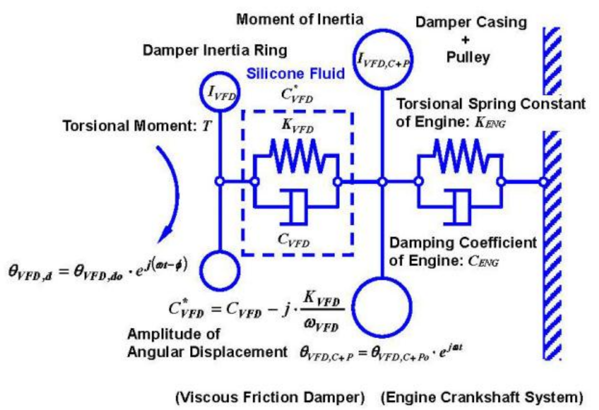

Figure 2. A two-degree-of-freedom equivalent torsional vibration system of an engine crankshaft system with a viscous damper.

IV. NUMERICAL CALCULATION METHOD OF THE DYNAMIC CHARACTERISTICS FOR THE VISCOUS DAMPER

A. Numerical Calculation of the Complex Damping Coefficient

In this section, an equivalent vibration model is constructed for the damper unit by considering the inertia ring, silicone oil, and damper casing separately (refer to Fig. 2). Considering the inertia ring part and deriving an equation of motion yield the following equation.

$$I_{VFD} \cdot \frac{d^2\theta_{VFD,d}}{dt^2} + C^*_{VFD} \cdot \left(\frac{d\theta_{VFD,d}}{dt} - \frac{d\theta_{VFD,p}}{dt} \right) = I_{VFD} \cdot \ddot{\theta}_{VFD,d} + C^*_{VFD,d} \cdot \left(\dot{\theta}_{VFD,d} - \dot{\theta}_{VFD,p} \right) = 0 \quad (1)$$

Here, $\theta_{VFD,do}$ is the amplitude of the torsional vibration angular displacement of the damper inertia ring, $\theta_{VFD,po}$ is the amplitude of the torsional vibration angular displacement of the damper casing, and I_{VFD} is the moment of inertia of the damper inertia ring. The torsional vibration angular displacement of the damper pulley is $\theta_{VFD,p} = \theta_{VFD,po} \cdot e^{j\omega t}$, and the amplitude of the torsional vibration angular displacement of the damper inertia ring is $\theta_{VFD,d} = \theta_{VFD,do} \cdot e^{j(\omega t - \phi)}$. The complex torsional damping coefficient C^*_{VFD} is determined from the product of the damper constant K_{VFD} and the complex viscosity coefficient: μ^*_{VFD}.

$$C^*_{VFD} = K_{VFD} \cdot \mu^*_{VFD} = K_{VFD} \cdot \left(\mu'_{VFD} - j \cdot \frac{G'_{VFD}}{\omega_{VFD}} \right) = C_{VFD} - j \cdot \frac{K_{VFD}}{\omega_{VFD}} \quad (2)$$

$$K_{VFD} = \frac{\pi}{h_s} \cdot \left(R_o^4 - R_i^5 \right) + 2 \cdot \pi \cdot b \cdot \left(\frac{R_o^3}{h_o} - \frac{R_i^3}{h_i} \right) \quad (3)$$

and the absolute value of the complex torsional damping coefficient $\left| C^*_{VFD} \right|$ becomes

$$\left| C^*_{VFD} \right| = \sqrt{C_{VFD}^2 + \left(\frac{K_{VFD}}{\omega_{VFD}} \right)^2} \quad (4)$$

where K_{VFD} is the damper coefficient; μ'_{VFD} is a real part of the complex coefficient of torsional viscosity of silicone oil; G'_{VFD}/ω_{VFD} is an imaginary part of the complex coefficient of torsional viscosity of silicone oil; C_{VFD} is the torsional damping coefficient; K_{VFD} is the torsional spring constant; h_s, h_o, and h_i are the lateral gap, outer-peripheral gap, and inner-peripheral gap, respectively, between the damper casing and the inertia ting; R_o and R_i are the outer radius and inner radius, respectively, of the damper inertia ring; b is the width of the damper inertia ring.

B. Relationship among the Complex Damping Coefficient, Damping Coefficient, and Spring Constant

The numerical calculation methods for the torsional damping coefficient C_{VFD} and the torsional spring constant K_{VFD} are described on the basis of data obtained from the installation test in order to investigate the dynamic characteristics of the silicone oil of the viscosity damper [refer to Eqs. (5) and (6)]. The complex damping coefficient of the silicone oil was determined from the product of the damper constant and the complex viscosity coefficient. Accordingly, by investigating the characteristics of the torsional spring constant K_{VFD} and the damping coefficient C_{VFD}, which are composed, respectively, of the real and imaginary parts of the complex damping coefficient of the silicone oil as defined by Eq. (2), these were equal to investigating the characteristics of the complex damping coefficient.

C. Introduction of the Damping Coefficient and Spring Constant of the Viscous Damper Silicone Fluid

It is possible to find the amplitude ratio and phase difference for the damper casing and the inertia ring from a harmonic analysis of the torsional vibration wave-forms using the simultaneous two-point measurements. Using these values, a formula that can numerically calculate the kinetic torsional spring constant and the damping constant of the damper silicone oil is introduced. By substituting the torsional vibration angular displacement of the damper inertia ring, $\theta_{VFD,d} = \theta_{VFD,do} \cdot e^{j(\omega t - \phi)}$, and the torsional vibration angular displacement of the damper casing, $\theta_{VFD,p} = \theta_{VFD,po} \cdot e^{j\omega t}$, into the equation of motion [Eq. (1)] and solving for the torsional spring constant K_{VFD} and torsional damping coefficient C_{VFD}, the following equations are obtained.

$$K_{VFD} = \frac{I_{VFD} \cdot \omega_{VFD}^2 \cdot M \cdot \left(M - \cos\phi \right)}{M^2 + 1 - 2 \cdot M \cdot \cos\phi} \quad (5)$$

$$C_{VFD} = \frac{I_{VFD} \cdot \omega_{VFD} \cdot M \cdot \sin\phi}{M^2 + 1 - 2 \cdot M \cdot \cos\phi} \quad (6)$$

Here, the amplitude ratio (kinetic scale factor) is $M = \theta_{VFD,do} / \theta_{VFD,po}$. Since the angular frequency and moment of inertia of the damper inertia ring are already known, the kinetic torsional spring constant and torsional damping coefficient can be calculated by substituting the value of the amplitude ratio M and the value of the phase difference ϕ of the damper casing and inertia ring into Eqs. (5) and (6).

V. Simultaneous Two-Point Measurements Method, Measurement Results, and Numerical Calculation Method for Torsional Vibration Angular Displacement

Using a universal joint as an intermediary, the test engine is connected to an eddy-current dynamometer. The test viscous damper is mounted on the free end of the engine crankshaft. A tape for pulse generation, which creates a signal from black and white sections lined up alternately, is attached to the casing of the test viscous damper and the inertia ring. Lighting is produced by the emitting part of an optical sensor, and a voltage signal is detected by a light receiver. A frequency signal in proportion to the change in rotation is determined from the detected signal. The obtained signal is amplified and then recorded by a data recorder. Furthermore, the average angular velocity (central frequency) is operated by this voltage signal, the torsional vibration angular displacement is numerically calculated from the difference between the detected frequency and the central frequency, and a torsional vibration waveform is acquired. This torsional vibration is harmonically analyzed on a personal computer. The torsional vibration angular displacement is measured by driving the engine speed at full load between 1,000 and 3,000 r/min. During the experiment, the engine coolant, lubricating oil, and viscous damper surface were held at a constant temperature of 333 K. In previous research, estimations were made from the equivalent vibration model when identifying the kinetic characteristic value of the silicone oil; however, the experimental method developed in the present work allows the kinetic characteristic value of the silicone oil for immediate operating conditions to be determined. In this section, the forced torsional vibration numerical calculation technique using the transfer matrix method is summarized for an engine crankshaft system with a mounted viscous damper. Furthermore, the validity of the equivalent torsional vibration model for the viscous damper unit referred to in this article is considered [3] (refer to Fig. 2). Figs. 3 and 4 shows the torsional vibration angular displacement amplitude curves under the condition of constant kinematic viscosity (0.10 m^2/s). Judging from these figures, the torsional vibration angular displacement amplitude decreases with the increase of the kinematic viscosity of the filling silicone fluid. In addition, the torsional amplitude is affected more greatly under the condition of the change of the lateral gap compared with that of the change of the peripheral gap. The latter tendency depends on the damper shape illustrated in Fig. 1.

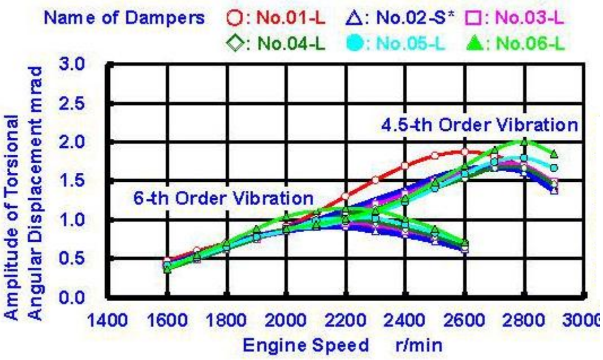

Figure 3. Amplitude curves of torsional angular displacement values at the damper casing under the conditions of peripheral gap change (lateral gap: constant, kinematic viscosity: 0.10 m^2/s).

A. *Comparison of Numerical Calculation and Experimental Results*

Comparisons of the numerical calculation and experimental results for the amplitude of the torsional

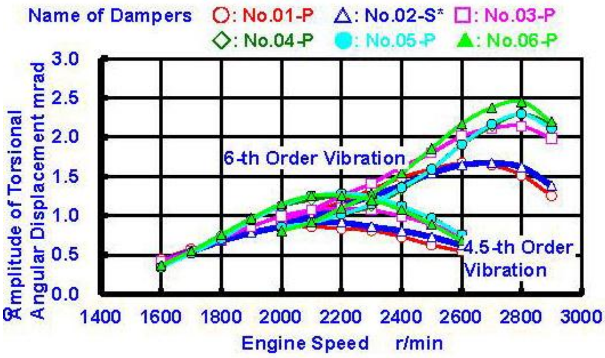

Figure 4. Amplitude curves of torsional angular displacement values at the damper casing under the conditions of lateral gap change (peripheral gap: constant, kinematic viscosity: 0.10 m^2/s).

vibration angular displacement of a viscous damper mounted on a crankshaft system are shown in Figs. 3 and 4. The plots in the diagrams show the experimental value, and the lines show the numerical calculation results. The torsional vibration characteristics show a good agreement between the numerical calculation and the measurement results. Therefore, the equivalent vibration model of a viscous damper unit can also be applied to the equivalent vibration model of the whole engine system. Moreover, the validity of the equivalent vibration model for the silicone oil is shown.

VI. Spring Constant and Damping Coefficient of Viscous Dampers Near-Natural Frequency

Figs. 5 and 6 illustrate, respectively, the values of torsional spring constant and damping coefficient obtained from the torsional vibration angular displacement at the inertia ring and the casing by the simultaneous two-point measurement method. The dynamic characteristic values are nearly the same in every kinematic viscosity regardless of the order. The values of torsional spring constant and damping coefficient have a tendency to increase with the increase of the kinematic viscosity of the filling silicone fluid.

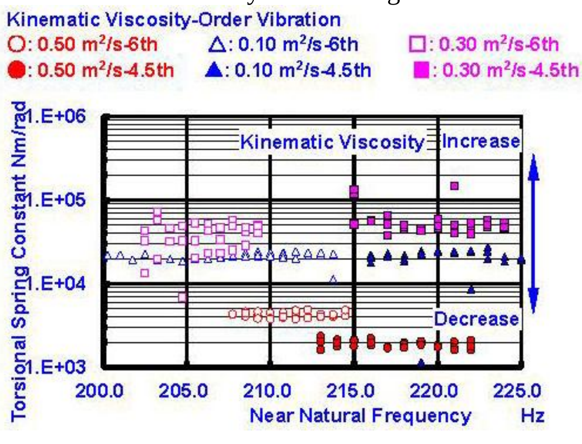

Figure 5. Relationship between the torsional spring constant and near-natural frequency by simultaneous measurement at two points (number of inertia ring: No. 03-L, in-line six-cylinder engine).

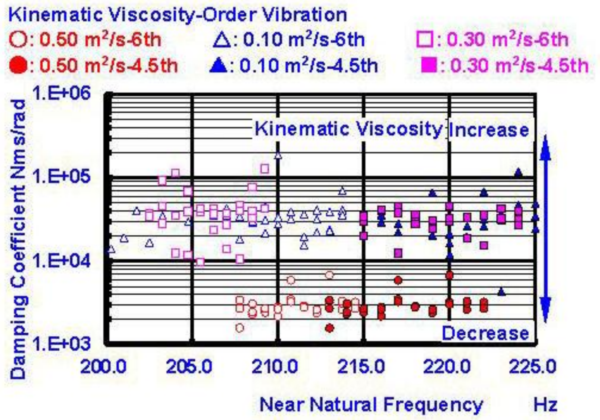

Figure 6. Relationship between the torsional damping coefficient and kinematic viscosity under conditions of peripheral gap change (lateral gap: constant, in-line six-cylinder engine).

VII. Relationship between the Kinematic Viscosity and Spring Constant of Silicone Oil Filling and Damping Coefficient (Consideration of the Spring Effect of the Silicone Oil Filling)

As mentioned above, the validity of the equivalent vibration model for the viscous damper was verified. Using this model, the numerical calculation results for the torsional spring constant and torsional damping coefficient of the silicone oil are considered. **Fig. 7** shows the relationship between the kinematic viscosity and the torsional spring constant for each inertia ring. The value of the torsional spring constant increases in proportion to the kinematic viscosity. A transition of the values of the torsional spring constant can be seen at 2.0 to 3.0 $\times 10^4$ Nmrad^{-1} for low viscosity and at 2.5 to 3.6 $\times 10^4$ Nmrad^{-1} for high viscosity. Thus, the spring effect also increases in proportion to the viscosity of the silicone oil. If a high-viscosity silicone oil is used when designing a viscous damper, it is necessary to consider this spring effect. Fig. 8 shows the relationship between the torsional damping coefficient and the kinematic viscosity for each inertia ring. The value of the torsional damping coefficient increases in proportion to the kinematic viscosity. A transition of the values of the torsional damping coefficient occurs at 1.0 to 2.0 $\times 10^3$ Nmsrad^{-1} for low viscosity and at 1.1 to 4.0 $\times 10^3$ Nmsrad^{-1} for high viscosity. The effect of the kinetic characteristic value due to the change of the inertia ring dimensions is small.

VIII. Summary

In this research, a high-speed diesel engine crankshaft system with a mounted viscous torsional vibration damper was operated, and the torsional vibration angular displacement values of the inertia ring and the damper casing of the viscous damper were measured by the simultaneous two-point measurements. The moment of the damper inertia ring, clearance gap dimensions between the damper inertia ring and the

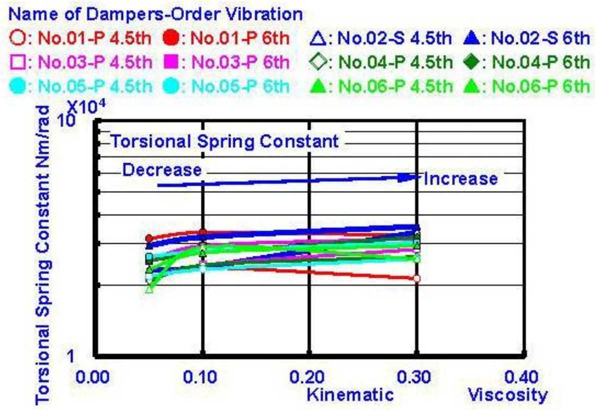

Figure 7. Relationship between the torsional spring constant and kinematic viscosity under conditions of peripheral gap change (lateral gap: constant, in-line six-cylinder engine).

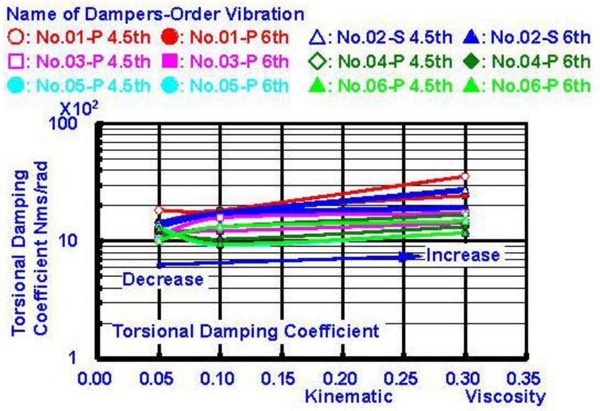

Figure 8. Relationship between the torsional damping coefficient and kinematic viscosity under conditions of peripheral gap change (lateral gap: constant, in-line six-cylinder engine).

casing (damper inertia ring dimensions), and the kinematic viscosity of the silicone oil in the viscous damper were varied. Furthermore, an equivalent torsional vibration model for the viscous damper was constructed and applied to the equivalent vibration model for the entire crank system using the transfer matrix method, and numerical calculations were performed. Through a comparison of these measurement results and numerical calculations, the kinematic characteristics of viscous dampers were investigated. Our key findings are summarized below.

(1) By focusing on only the viscous damper and introducing an integrated particle equivalent vibration model that considers the spring effect, a formula that gives the torsional damping coefficient C_{VFD} and the torsional spring constant K_{VFD} was found. The values of C_{VFD} and K_{VFD} can be determined by substituting the measured values (amplitude and phase difference) that can be acquired by the simultaneous two-point measurement method.

(2) Substituting the values of C_{VFD} and K_{VFD} acquired using the above-mentioned method for the damper part of the torsional vibration model that replaced the viscous damper mounted on the crankshaft system in the equivalent vibration system gives a torsional vibration

numerical calculation result that agrees well with the experimental results. This proves the validity of the values of $\boldsymbol{C_{VFD}}$ and $\boldsymbol{K_{VFD}}$ proposed in this research.

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