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

Tomoaki Kodama  
Department of Science and Engineering, School of Science and Engineering,  
Kokushikan University, Tokyo, Japan  
Email: kodama@kokushikan.ac.jp

Yasuhiro Honda  
Department of Science and Engineering, School of Science and Engineering,  
Kokushikan University, Tokyo, Japan  
Email: honda@kokushikan.ac.jp

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 fluid.
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 m²/s as a standard, and lower and higher values of 0.05 m²/s and 0.30 m²/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

<table>
<thead>
<tr>
<th>Particulars</th>
<th>Contents</th>
</tr>
</thead>
<tbody>
<tr>
<td>Designed for</td>
<td>High-speed diesel engines</td>
</tr>
<tr>
<td>Type for</td>
<td>Four-stroke cycle, direct injection</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>Six-cylinders</td>
</tr>
<tr>
<td>Arrangement</td>
<td>In-line</td>
</tr>
<tr>
<td>Bore and stroke (m)</td>
<td>0.105–0.125</td>
</tr>
<tr>
<td>Total piston displacement (m³)</td>
<td>0.006469</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.0</td>
</tr>
<tr>
<td>Maximum brake output (kW/r/min)</td>
<td>230/3200</td>
</tr>
<tr>
<td>Maximum torque (Nm/r/min)</td>
<td>451/1800</td>
</tr>
<tr>
<td>Firing order</td>
<td>1–5–3–6–2–4</td>
</tr>
</tbody>
</table>

### Table II. Dimensions of the Damper Inertia Ring

<table>
<thead>
<tr>
<th>Number of inertia ring</th>
<th>Moment of inertia x10^2 (kgm²)</th>
<th>Thickness x10^-2 (m)</th>
<th>Outside radius x10^-1 (m)</th>
<th>Inside radius x10^-1 (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 01-L</td>
<td>3.256</td>
<td>1.940</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 02-S</td>
<td>3.200</td>
<td>1.900</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 03-L</td>
<td>3.029</td>
<td>1.800</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 04-L</td>
<td>2.882</td>
<td>1.700</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 05-L</td>
<td>2.723</td>
<td>1.600</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 06-L</td>
<td>2.564</td>
<td>1.500</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 01-P</td>
<td>3.229</td>
<td>1.900</td>
<td>1.207</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 02-S</td>
<td>3.200</td>
<td>1.900</td>
<td>1.205</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 03-P</td>
<td>3.115</td>
<td>1.900</td>
<td>1.195</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 04-P</td>
<td>3.031</td>
<td>1.900</td>
<td>1.190</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 05-P</td>
<td>2.950</td>
<td>1.900</td>
<td>1.185</td>
<td>9.040</td>
</tr>
<tr>
<td>No. 06-P</td>
<td>2.890</td>
<td>1.900</td>
<td>1.185</td>
<td>9.040</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Number of silicone fluid</th>
<th>Kinematic viscosity (m²/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 0 1</td>
<td>5.0 x 10^-2</td>
</tr>
<tr>
<td>No. 0 2</td>
<td>1.0 x 10^-1</td>
</tr>
<tr>
<td>No. 0 3</td>
<td>3.0 x 10^-1</td>
</tr>
</tbody>
</table>

Figure 2. A two-degree-of-freedom equivalent torsional vibration system of an engine crankshaft system with a viscous damper.
\[
I_{VFD} \frac{d^2 \theta_{VFD,d}}{dt^2} + C_{VFD}^* \left( \frac{d \theta_{VFD,d}}{dt} - \frac{d \theta_{VFD,p}}{dt} \right) = I_{VFD} \cdot \dot{\theta}_{VFD,d} + C_{VFD}^* \left( \dot{\theta}_{VFD,d} - \dot{\theta}_{VFD,p} \right) = 0
\]

Here, \( \theta_{VFD,d} \) is the amplitude of the torsional vibration angular displacement of the damper inertia ring, \( \theta_{VFD,p} \) 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 \( \theta_{VFD,p} = \theta_{VFD,p} \cdot e^{j\phi} \), and the amplitude of the torsional vibration angular displacement of the damper inertia ring is \( \theta_{VFD,d} = \theta_{VFD,d} \cdot e^{j(\phi-\delta)} \). The complex torsional damping coefficient \( C_{VFD} \) is determined from the product of the damper constant \( K_{VFD} \) and the complex viscosity coefficient: 

\[
C_{VFD} = K_{VFD} \cdot \mu_{VFD} = K_{VFD} \left( \mu_{VFD} - j \cdot \frac{C_{VFD}}{\omega_{VFD}} \right)
\]

\[
= C_{VFD} - j \frac{K_{VFD}}{\omega_{VFD}}
\]

\[
K_{VFD} = \frac{\pi}{h_i} \left( R_i^e - R_i^i \right) + 2 \cdot \pi \cdot b \cdot \left( \frac{R_i^e}{h_o} - \frac{R_i^i}{h_i} \right)
\]

and the absolute value of the complex torsional damping coefficient \( |C_{VFD}| \) becomes

\[
|C_{VFD}| = \left( C_{VFD}^* + \frac{K_{VFD}}{\omega_{VFD}} \right)^2
\]

where \( K_{VFD} \) is the damper coefficient; \( \mu_{VFD} \) is a real part of the complex coefficient of torsional viscosity of silicone oil; \( C_{VFD} / \omega_{VFD} \) 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; \( h_o \), \( h_i \), and \( h_l \) are the lateral gap, outer-peripheral gap, and inner-peripheral gap, respectively, between the damper casing and the inertia ring; \( R_i^e \) and \( R_i^i \) 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, \( \theta_{VFD,d} = \theta_{VFD,d} \cdot e^{j(\phi-\delta)} \), and the torsional vibration angular displacement of the damper casing, \( \theta_{VFD,p} = \theta_{VFD,p} \cdot e^{j\phi} \), 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.

\[
K_{VFD} = \frac{I_{VFD} \cdot \omega_{VFD} \cdot M \cdot (M - \cos \phi)}{M^2 + 1 - 2 \cdot M \cdot \cos \phi}
\]

\[
C_{VFD} = \frac{I_{VFD} \cdot \omega_{VFD} \cdot M \cdot \sin \phi}{M^2 + 1 - 2 \cdot M \cdot \cos \phi}
\]

Here, the amplitude ratio (kinetic scale factor) is \( M = \theta_{VFD,d} / \theta_{VFD,p} \). 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 \( \phi \) 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 m²/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.

### VI. SPRING CONSTANT AND DAMPING COEFFICIENT OF VISCOS 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.
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 $3.0 \times 10^4$ Nm rad$^{-1}$ for low viscosity and at $2.5$ to $3.6 \times 10^4$ Nm rad$^{-1}$ 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 $2.0 \times 10^3$ Nms rad$^{-1}$ for low viscosity and at $1.1$ to $4.0 \times 10^3$ Nms rad$^{-1}$ 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 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 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 $C_{\text{VFD}}$ and the torsional spring constant $K_{\text{VFD}}$ was found. The values of $C_{\text{VFD}}$ and $K_{\text{VFD}}$ 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 $C_{\text{VFD}}$ and $K_{\text{VFD}}$ 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
This proves the validity of the values of $C_{VFD}$ and $K_{VFD}$ proposed in this research.

REFERENCES


Tomoaki Kodama has a Ph.D. degree in engineering, with the Course of Mechanical Engineering, Department of Science and Engineering, School of Science and Engineering, Kokushikan University, Tokyo, Japan. His research interests are NVH of internal combustion engines (including automotive and marine engines), engineering education, experiments in mechanical engineering, and mechanical design. He can be reached by e-mail at kodama@kokushikan.ac.jp.

Yasuhiro Honda has a Ph.D. degree, with the Course of Mechanical Engineering, Department of Science and Engineering, School of Science and Engineering, Kokushikan University, Tokyo, Japan. His research interests are NVH of internal combustion engines, engineering education, vehicle kinematics, mechanics, and automotive design. He can be reached by e-mail at honda@kokushikan.ac.jp.