論文 **Original Paper**

An Experimental Study on Dynamic Characteristics of Torsional Stiffness and Torsional Damping Coefficient of Viscous-Friction Dampers

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Abstract: The dynamic characteristics of torsional stiffness and torsional damping coefficient of viscous-friction dampers are investigated in this paper by adopting simultaneous vibration measurement method at two points. The vibration displacements of the damper casing and the damper inertia ring can be simultaneously measured in this method. It has become possible that the more detailed dynamic characteristics of the viscous-friction damper can be grasped by the method. Especially, it is an effective method to grasp the behaviors of the damper inertia ring and the damper casing for clarifying the effect of the silicone fluid on the torsional vibration of engine crankshaft system. The damper casing was made of acrylic resin in order to measure the behavior of the damper inertia ring on engine operation. It is possible to measure the torsional vibration displacements waveforms by the optical signals from pulse tapes stuck on both peripheral sides of the damper casing and the damper inertia ring. And, the dynamic characteristics of torsional stiffness and torsional damping coefficient of the silicone fluid were investigated by changing the viscosity of the silicone fluid and the clearance between the damper casing and the damper inertia ring. As the experimental results, it is shown in detail that the torsional vibration of the engine crankshaft system is greatly influenced by the peripheral gap, the lateral gap and the viscosity of silicone fluid of the viscous-friction damper, and that the values of the peripheral and lateral gaps between the outside of the annular seismic mass and the inside of the damper casing given by the widely-adopted BICERA's empirical formula are not always optimum on the every viscosity of the filled silicone fluid of the viscous-friction damper.

Keywords: Forced, Damping, Damper, Torsional Vibration, Diesel Engine, Dynamic Characteristics, Viscous-friction Damper, Experiment

1. Introduction

Typical torsional viscous-friction dampers (hereafter called "viscous dampers") consist of an annular seismic mass enclosed in a damper casing. The peripheral and lateral gaps between these two members are filled with a viscous fluid, e.g. dimethyl silicone fluid, and the thin layer is formed. As the silicone fluid is non-Newtonian fluid[1]-[5], the effective viscosity in the actuation is different from that in the operating condition and the complicated characteristics are shown[6]-[8]. The damping not only is improved with the increase of the viscosity of filled silicone fluid, but also the complex damping combined with the elasticity increases. In addition, the velocity gradient arises in the silicone fluid layer between the damper casing and the damper inertia ring, when torsional vibration is generated, and the vibration energy is dissipated as the heat energy by the shear resistance[8],[9]-[13]. Therefore, the clearance dimension between the inside of the damper casing and the outside of the damper inertia ring becomes an important subject in the viscous damper design[14]. In addition, these factors have a great effect on the torsional vibration characteristics of the engine crankshaft system[15]-[22]. Then, the experiments of torsional vibration characteristics were carried out under these conditions of the changes of the viscosity and the gap dimension of the viscous damper in an automotive high-speed diesel engine[23]-[26]. The torsional vibration angular displacement waveforms were simultaneously measured in two points of the damper casing and the damper inertia ring. It is the purpose to calculate directly the dynamic values from the measured waveforms and to make their characteristics clear.

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Table 1 Main Specifications of Experimental Engine

Particulars		Contents
Designed for		High-Speed Diesel Engine
Type for		4-Stroke Cycle, Direct Injection
Number of Cylinders		6-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 Brake Torque	$Nm/r/min$	451/1800
Firing Order		1-5-3-6-2-4

2. Specifications of Experimental Engine

The test engine used for the measurement of torsional vibration angular displacement waveform is the 6-cylinders, in-line, automotive high-speed diesel engine. The main specifications of the test engine are shown in **Table 1**[23]-[26].

3. Geometrical Dimension of Viscous-Friction Damper

3.1 Geometrical Dimension of Standrad Design Viscous-Fluid Damper

The engine crankshaft system with a viscous damper is replaced with the equivalent vibration system. The effective mass moment of inertia I_e ($=0.011$ kgm^2) and the natural angular velocity ω_e of the engine crankshaft system can be obtained by Holzer's method[12],[14],[15]. Then, the inertia moment of damper inertia ring I_d ($=0.330\times 10^{-2}$ kgm^2) can easily be calculated on the assumption that the damper inertia ratio $R(=I_d/I_e)$ is equal to 0.3[25],[26]. The clearance δ of the standard viscous damper is mostly determined from the BICERA's empirical formula[14] shown in the following.

$$\delta = 0.010 + 0.010 \cdot \sqrt{\left(\frac{D_0}{10}\right)} \quad \text{[inch]} \qquad (1)$$

Where, D_0: outer diameter of damper inertia ring. The clearance is determined to be approximately 0.0197 inch = 0.50 mm by substituting D_0 ($=241.00$ mm) $=9.49$ inch into equation (1). Then, the inside diameter of the damper casing becomes 242.00 mm. The dimension and the shape of the standard viscous damper shown in **Figure 1** were decided by considering those of the dampers widely adopted in the high-speed diesel engines until the present.

3.2 Geometries of Various Viscous-Friction Dampers

As shown in **Tables 2(a), 2(b)** and **3**, the dimensions of the damper inertia rings and the viscosities of the silicone fluids, respectively, are diversely varied in this experiment,

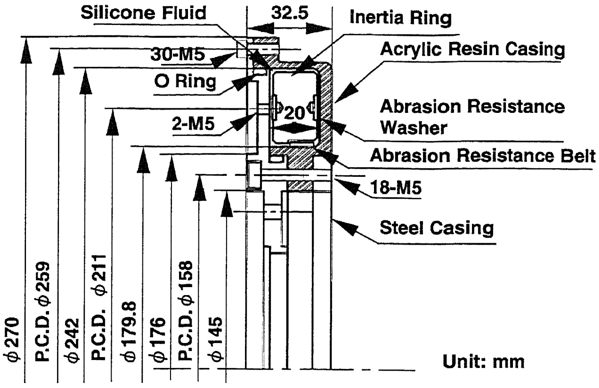

Fig. 1 Geometry of Viscous-Fluid Damper

Table 2(a) Dimensions of Damper Inertia Ring (Change of Lateral Gap)

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

* Designed Standard Viscous Damper

Table 2(b) Dimensions of Damper Inertia Ring (Change of Peripheral Gap)

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-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

* Designed Standard Viscous Damper

Table 3 Kinematic Viscosity of Silicone Fluid

Number of Silicone Fluid	Kinematic Viscosity m^2/s
No. 01	5.0×10^{-2}
No. 02	1.0×10^{-1}
No. 03	3.0×10^{-1}

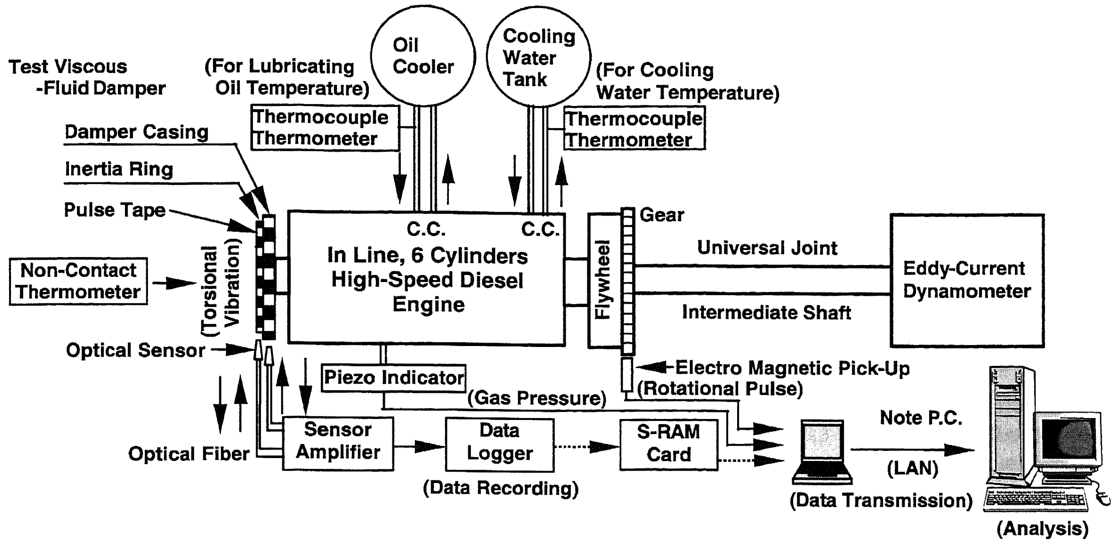

Fig. 2 Schematic Diagram of Simultaneously Measuring System of Torsional Vibration Waveforms at Two Points

in order to investigate the dynamic characteristics of torsional stiffness and totsional damping coefficient of the viscous dampers[23]-[26]. The number of the damper inertia ring of the standard viscous damper is No. 02–S* in **Tables 2(a)** and **(b)**. The dimension of the gap of the standard damper is determined from the BICERA's empirical formula[14]. The peripheral and lateral gaps between the damper inertia ring and the damper casing of the other dampers are changed on the basis of the gap of the standard damper.

4. Simultaneous Measurement of Torsional Angular Displacement Waveform

The test engine was equipped with the torsional viscous-friction damper. An eddy-current dynamometer was connected to the crankshaft of the engine via an intermediate shaft and a universal joint. The torsional vibration waveforms were simultaneously measured at the outsides of the damper inertia ring and the damper casing. Transparent acrylic resin suitable for penetrating light was adopted as the material of the damper casing part. The tapes, in which white and black parts were arranged alternately for generating signal pulses, were stuck on the outsides of the damper casing and the damper inertia ring. The floodlight from the light emission division of the photo sensor and the reflected light were detected by the light-receiver. The electric frequency signals proportional to engine speed were obtained from the photo pickup. The measured signals were transmitted to the phase-shift torsiograph equipment via the adapter which calculated the average of angular velocity (the center frequency). The torsional vibration waveforms could be obtained from the torsional angles, which were calculated using the relationship between the measured and center frequencies. The signals were recorded by the data logger via the amplifier. The measured torsional waveforms of the damper inertia ring and the damper casing were harmonically analyzed using the personal computer. The schematic drawing of the experimental system shown in **Figure 2**. The torsional vibration waveforms were measured under full load from 1000 to 3000 r/min. The temperatures of the cooling water and the lubricating oil of the engine kept constant, and also the surface temperature of the viscous damper was retained at the approximately fixed 333 K during the experiments[23]-[26].

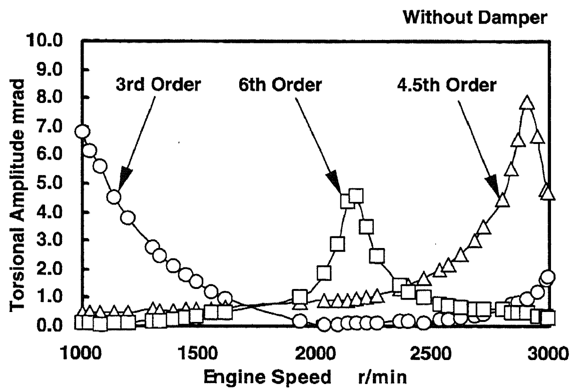

Fig. 3 Amplitude Curves of Torsional Angular Displacements at Pulley End (Without Damper)

5. Measurement Results of Amplitude of Torsional Angular Displacement

Figure 3 illustrates the measured amplitude curves of torsional angular displacement at the pulley end of the engine crankshaft system without a viscous damper. The 4.5th and 6th order components of the 2nd node resonant torsional vibrations appear mainly at 2870 and 2170 r/min, respectively. These orders are major critical ones of a 6-cylinders, in-line engine. **Figures 4(a)** and **(b)** illustrate

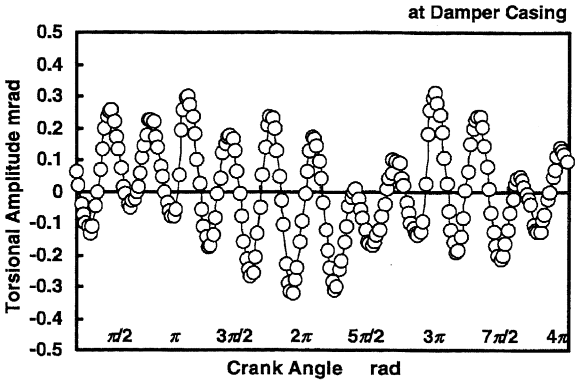

Fig. 4(a) Waveform of Torsional Angular Displacement at Damper Casing (Inertia Ring No. 02–S*, 6th Order Resonant Engine Speed: 2170 r/min)

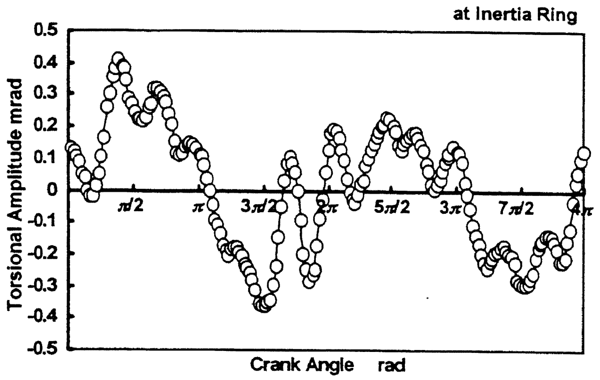

Fig. 4(b) Waveform of Torsional Angular Displacement at Damper Inertia Ring (Inertia Ring No. 02–S*, 6th Order Resonant Engine Speed: 2170 r/min)

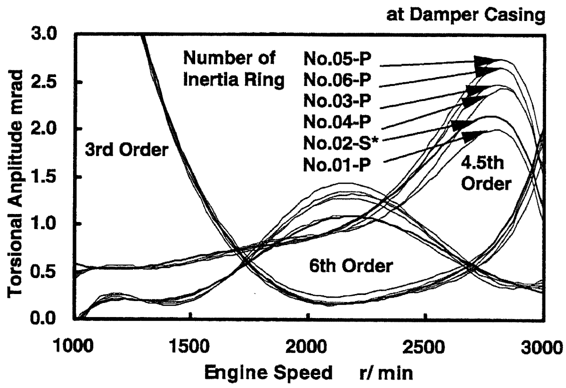

Fig. 5(a) Torsional Amplitude Curves of Damper Casing on Conditions of Peripheral Gap Changes (Lateral Gap: Constant, Kinematic Viscosity: 0.05 m^2/s)

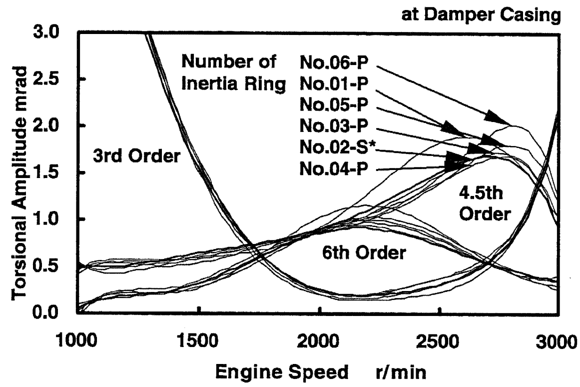

Fig. 5(b) Torsional Amplitude Curves of Damper Casing on Conditions of Peripheral Gap Changes (Lateral Gap: Constant, Kinematic Viscosity: 0.10 m^2/s)

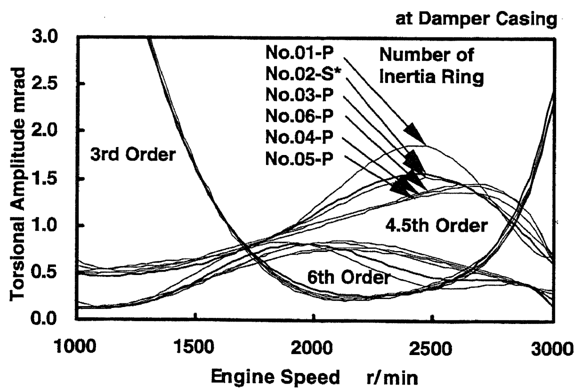

Fig. 5(c) Torsional Amplitude Curves of Damper Casing on Conditions of Peripheral Gap Changes (Lateral Gap: Constant, Kinematic Viscosity: 0.30 m^2/s)

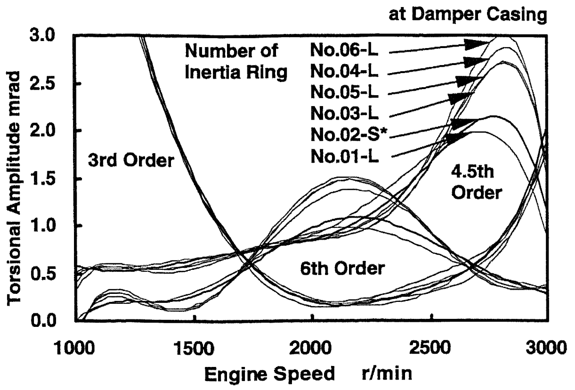

Fig. 6(a) Torsional Amplitude Curves of Damper Casing on Conditions of Lateral Gap Changes (Peripheral Gap: Constant, Kinematic Viscosity: 0.05 m^2/s)

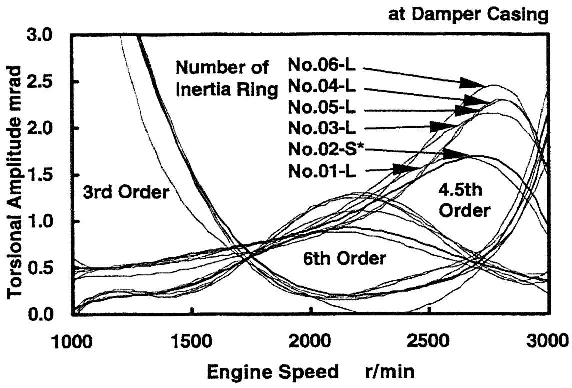

Fig. 6(b) Torsional Amplitude Curves of Damper Casing on Conditions of Lateral Gap Changes (Peripheral Gap: Constant, Kinematic Viscosity: 0.10 m^2/s)

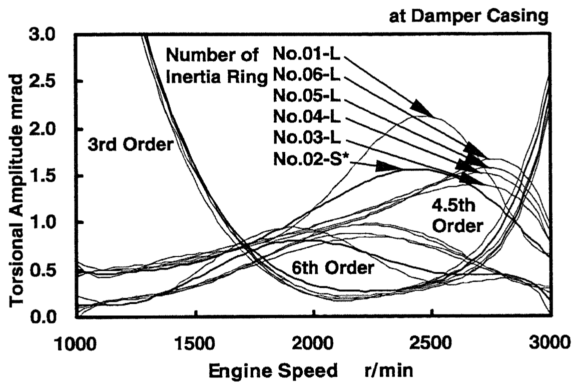

Fig. 6(c) Torsional Amplitude Curves of Damper Casing on Conditions of Lateral Gap Changes (Peripheral Gap: Constant, Kinematic Viscosity: 0.30 m^2/s)

the measured torsional vibration waveforms in the vicinity of the 6th order resonant engine speed of 2170 r/min, as an example of the results of simultaneous measurement at the damper casing and the damper inertia ring of the engine crankshaft system with the standard viscous-friction damper. **Figures 5(a), (b)** and **(c)** illustrate the measured torsional amplitude curves on condition that the peripheral gap is changed and the viscosity of the silicone fluid is also varied. **Figures 6(a), (b)** and **(c)** illustrate the measured torsional amplitude curves on condition that the lateral gap is changed, and the viscosity of the silicone fluid is also varied. Judging from these measured amplitude curves, the peripheral gap, the lateral gap and the viscosity of the silicone fluid of the viscous damper are greatly influenced on the torsional vibration of the engine crankshaft system. The resonant amplitudes of the engine crankshaft system with the damper (inertia ring number No. 02–s*) of the gap clearance of 0.5 mm, which is obtained by BICERA's empirical formula (refer to equation (1)), are not always the minimum values in comparison with other dampers. The resonant amplitudes of the 4.5th and 6th orders tend to reduce with the increase of the viscosity of the filled silicone fluid. The amplitudes of the 3rd order torsonal vibration, in which the resonance does not appear within the operating engine speed, are not influenced by the viscosity. The influence of the peripheral gap on the reduction of the resonant amplitude is greater than that of the lateral gap.

6. Formula for Computation of Dynamic Characteristics Value in Silicone Fluid Part of Viscous-Fricrion Damper

The viscous damper consists of an annular seismic mass enclosed in a damper casing filled with silicone fluid. The equation of the complex torsional damping coefficient is derived in order to investigate the dynamic characteristics of the silicone fluid parts of the viscous damper on the basis of the measured data obtained from the experiments. The complex coefficient of torsional viscosity of silicone fluid μ^* is defined by the following equation[1]-[8], [23]-[26].

$$\mu^* = \mu' + \frac{G'}{j\cdot\omega} = \mu' - j\cdot\frac{G'}{\omega} \tag{2}$$

Where, μ': real part of complex coefficient of torsional viscosity of silicone fluid, G'/ω: imaginary part of complex coefficient of torsional viscosity of silicone fluid, $j=\sqrt{-1}$, ω: circular frequency of i-th order vibration.

In addition, the damper coefficient K is defined by the following equation[18]-[21],[23]-[26].

$$K=\frac{\pi}{h_s}\cdot(R_0^4-R_i^4)+2\cdot\pi\cdot b\cdot\left(\frac{R_i^3}{h_i}+\frac{R_0^3}{h_0}\right) \tag{3}$$

Where, h_s, h_o, h_i: lateral gap, outer-peripheral gap, inner-peripheral gap between damper casing and damper inertia ring, respectively, R_o, R_i: outer radius, inner radius of damper inertia ring, respectively, b: width of damper inertia ring.

The complex coefficient of torsional viscous damping C_d^* is determined from the product of the damper coefficient and equation (2) as follows[1]-[5],[7],[11][16]-[20],[23]-[26]:

$$C_d^*=K\cdot\mu^*=K\cdot\left(\mu'-j\cdot\frac{G'}{\omega}\right)=C_d-j\cdot\frac{K_d}{\omega} \tag{4}$$

The dynamic characteristics of the complex coefficient of torsional viscous damping are investigated by the examination of torsional damping coefficient C_d and torsional stiffness K_d composed of the real part and the imaginary part of the complex coefficient, respectively. The absolute value of the complex coefficient of torsional viscous damping $|C_d^*|$ can be determined by the following equation[23]-[26].

$$|C_d^*|=\sqrt{C_d^2+(K_d/\omega)^2} \tag{5}$$

The values of dynamic torsional stiffness and torsional damping coefficient of viscous friction parts can be obtained from the harmonically analyzed results of waveforms measured at the damper inertia ring and the damper casing. The equation of motion at the damper inertia ring is as follows[23]-[26]:

$$I_d\cdot\ddot{\theta}_d-C_d^*\cdot(\dot{\theta}_p-\dot{\theta}_d)=0 \tag{6}$$

Where,

$$\theta_d=\theta_{d0}\cdot e^{j(\omega t-\phi)},\ \theta_p=\theta_{p0}\cdot e^{j\omega t}\ \text{and}\ C_d^*=C_d-j\cdot\frac{K_d}{\omega} \tag{7}$$

θ_d: torsional angular displacement of damper inertia ring, θ_p: torsional angular displacement of damper casing.

The following equations (8) and (9) can be obtained by rearranging the expression of equation (6), into which the above-mentioned relational expressions (7) is substituted;

$$K_d=\frac{I_d\cdot\omega^2\cdot M\cdot(M-\cos\phi)}{M^2+1-2\cdot M\cdot\cos\phi} \tag{8}$$

$$C_d=\frac{I_d\cdot\omega\cdot M\cdot\sin\phi}{M^2+1-2\cdot M\cdot\cos\phi} \tag{9}$$

Where, $M=\theta_{do}/\theta_{po}$.

In equations (8) and (9), the values of amplitude ratio M and phase angle of between damper inertia ring and damp-

Table 4(a) Experimental Results of Natural Frequency and Resonant Torsional Amplitudes (Change of Lateral Gap)

Number of Inertia Ring	Kinematic Viscosity m²/s	4.5th Order Vibration		6th Order Vibration	
		Natural Frequency Hz	Resonant Torsional Amplitude mrad	Natural Frequency Hz	Resonant Torsional Amplitude mrad
No. 01-L	5.0×10^{-2}	209.25	2.025	218.00	1.096
	1.0×10^{-1}	196.50	1.902	217.00	0.958
	3.0×10^{-1}	183.00	1.868	188.00	0.833
No. 02-S*	5.0×10^{-2}	207.75	2.147	218.00	1.086
	1.0×10^{-1}	203.25	1.684	217.00	0.916
	3.0×10^{-1}	183.00	1.555	196.00	0.812
No. 03-L	5.0×10^{-2}	210.75	2.461	219.00	1.292
	1.0×10^{-1}	205.50	1.719	222.00	0.972
	3.0×10^{-1}	185.25	1.513	208.00	0.834
No. 04-L	5.0×10^{-2}	212.25	2.429	220.00	1.318
	1.0×10^{-1}	207.00	1.684	223.00	1.005
	3.0×10^{-1}	199.50	1.415	217.00	0.777
No. 05-L	5.0×10^{-2}	212.25	2.758	217.00	1.449
	1.0×10^{-1}	209.25	1.815	222.00	1.049
	3.0×10^{-1}	196.50	1.384	213.00	0.754
No. 06-L	5.0×10^{-2}	210.00	2.688	218.00	1.361
	1.0×10^{-1}	210.75	2.007	221.00	1.152
	3.0×10^{-1}	200.25	1.459	213.00	0.843

* Designed Standard Viscous Damper

er casing ϕ can be obtained by analyzing harmonically the waveforms measured at the damper inertia ring and the damper casing. In addition, the values of I_d, ω are known. Therefore, the values of dynamic torsional stiffness K_d and torsional damping coefficient C_d can be determined from equations (8) and (9), respectively.

7. Experimental Results and Considerations

7.1 Experimental Results of Resonant Amplitudes

Tables 4(a) and **(b)** show the measured natural frequencies and resonant torsional amplitudes obtained from the experiments by the changes of the dimensions of the damper inertia rings and the viscosity of silicone fluid indicated in **Tables 2(a), 2(b)** and **3**. There are more effective dampers in the reduction of resonant amplitudes than the standard design damper No. 02–S* which have the clearance of 0.5 mm determined from BICERA's experimental equation[14], by judging from the results shown in **Tables 4(a)** and **(b)** on condition that the inertia moment of damper inertia ring of each damper is approximately equal to one another. As some viscous dampers are more effective in resonant amplitude reduction than the standard design damper No. 02–S*, the clearance value given by BICERA's empirical formula[14] is not always optimum on the every viscosity of the filled silicone fluid of the viscous damper.

7.2 Relationship between Resonant Torsional Amplitude and Natural Frequency by Change of Viscosity of Silicone Fluid

Figure 7(a), (b), which draws full lines in accordance with the equal values of the inertia moments of the damper inertia rings, show the relationship between resonant torsional amplitude of the 4.5th order vibration and the natural frequency by the change of the viscosity of silicone fluid, respectively. The effect of the moment of inertia on the resonant torsional amplitude can be disregarded in the high viscosity, but it cannot be disregarded in the low viscosity. It is possible to investigate the dynamic characteristics of the torsional vibration of the engine crankshaft system with the viscous damper under the experimental condition of changing the viscosity of the silicone fluid. It is a general tendency on the low viscosity that the resonant amplitude are greatly changed and that the natural frequencies are scarcely changed. On the other hand, it is also a general tendency on the high viscosity that the natural frequencies are greatly changed and that the resonant amplitudes are scarcely changed.

Table 4(b) Experimental Results of Natural Frequency and Resonant Torsional Amplitude (Change of Peripheral Gap)

Number of Inertia Ring	Kinematic Viscosity m^2/s	4.5th Order Vibration		6th Order Vibration	
		Natural Frequency Hz	Resonant Torsional Amplitude mrad	Natural Frequency Hz	Resonant Torsional Amplitude mrad
No. 01–P	5.0×10^{-2}	202.50	1.990	214.00	0.983
	1.0×10^{-1}	197.25	1.667	211.00	0.878
	3.0×10^{-1}	185.25	2.146	192.00	0.953
No. 02–S*	5.0×10^{-2}	207.75	2.147	218.00	1.086
	1.0×10^{-1}	203.25	1.684	217.00	0.916
	3.0×10^{-1}	183.00	1.555	196.00	0.812
No. 03–P	5.0×10^{-2}	212.25	2.723	217.00	1.484
	1.0×10^{-1}	207.00	2.147	221.00	1.112
	3.0×10^{-1}	201.00	1.401	225.00	0.846
No. 04–P	5.0×10^{-2}	213.00	2.862	216.00	1.480
	1.0×10^{-1}	211.50	2.304	220.00	1.243
	3.0×10^{-1}	204.00	1.513	218.00	0.894
No. 05–P	5.0×10^{-2}	212.25	2.723	218.00	1.384
	1.0×10^{-1}	211.50	2.286	223.00	1.286
	3.0×10^{-1}	206.25	1.571	221.00	0.983
No. 06–P	5.0×10^{-2}	211.50	3.001	216.00	1.522
	1.0×10^{-1}	208.50	2.461	219.00	1.276
	3.0×10^{-1}	208.50	1.676	225.00	0.983

* Designed Standard Viscous Damper

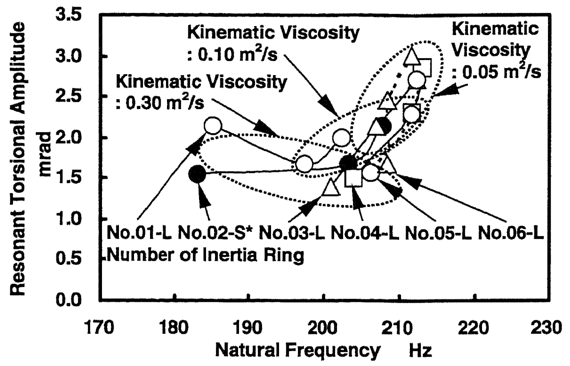

Fig. 7(a) Relationship between Resonant Torsional Amplitude and Natural Frequency by Change of Kinematic Viscosity (Lateral Gap: Change, Peripheral Gap: Constant, 4.5th Order Vibration)

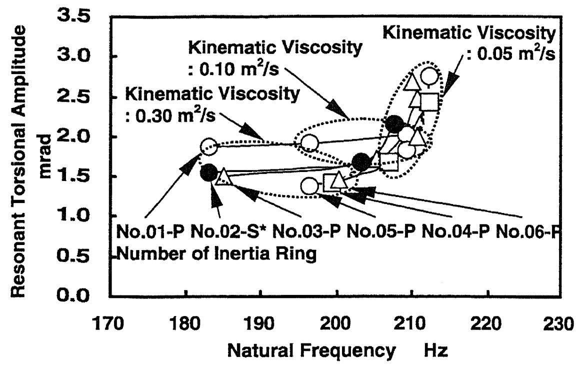

Fig. 7(b) Relationship between Resonant Torsional Amplitude and Natural Frequency by Change of Kinematic Viscosity (Peripheral Gap: Change, Lateral Gap: Constant, 4.5th Order Vibration)

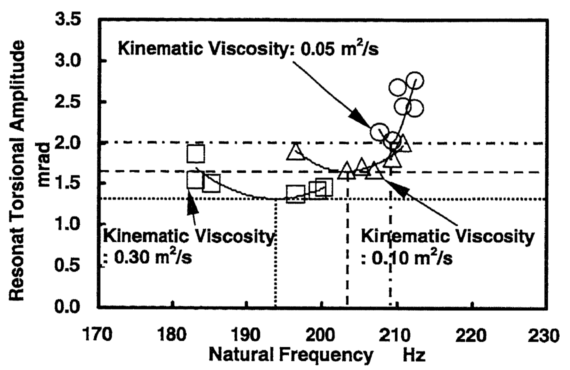

Fig. 8(a) Relationship between Resonant Torsional Amplitude and Natural Frequency by Gap Change of Clearance in Every Kinematic Viscosity (Peripheral Gap: Change, Lateral Gap: Constant, 4.5th Order Vibration)

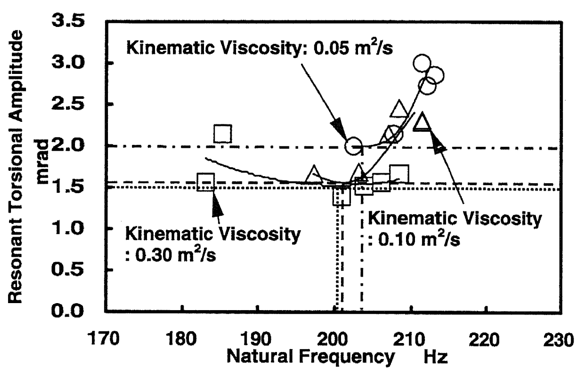

Fig. 8(b) Relationship between Resonant Torsional Amplitude and Natural Frequency by Gap Change of Clearance in Every Kinematic Viscosity (Lateral Gap: Change, Peripheral Gap: Constant, 4.5th Order Vibration)

7.3 Relationship between Resonat Torsional Amplitude and Natural Frequency on Condition of Constant Viscosity of Silicone Fluid

Figures 8(a) and **(b)** show the relationship between the resonant torsional amplitude and the natural frequencies of the 4.5th vibration, respectively, obtained by changing the clearance gap at every viscosity. The optimum clearance which minimizes the resonant amplitude exists at every viscosity, judging from the these experimental results. The optimum clearance between the outside of the damper inertia ring and the inside of the damper casing have influence on the viscosity of the silicone fluid.

7.4 Effect of Clearance Gap on Resonant Torsional Amplitude

Figures 9(a) and **(b)** illustrate the relationship between the resonant torsional amplitude and the clearance gap on condition of the dimensional change of (a) the only lateral gap or (b) the only peripheral gap, respectively. As shown in **Table 3**, the viscosity of the silicone fluid is diversely varied in this experiment. The clearance gap of less than 0.5 mm in the range of the low viscosity of the silicone fluid is more effective on the reduction of the resonant torsional amplitude than the clearance gap of 0.5 mm given by BICERA's empirical formula[14]. But the resonant torsional amplitude in the clearance gap of more than 0.5 mm becomes smaller in the range of the high viscosity of the filled silicone fluid than that in the clearance gap of 0.5 mm. And, the optimum clearance gap of the damper filled with the 0.10 m^2/s high viscosity of the silicone fluid is nearly 0.5 mm.

7.5 Relationship between Torsional Stiffness, Torsional Damping Coefficient and Narural Frequency by Viscosity of Silicone fluid

Figure 10(a), (b) illustrates the relationship between torsional stiffness, torsional damping coefficient and natural frequency of the damper with damper inertia ring No. 04–P, in which the peripheral clearance gap is 1.5 mm. The clearance gap is constant and the viscosity is changed in

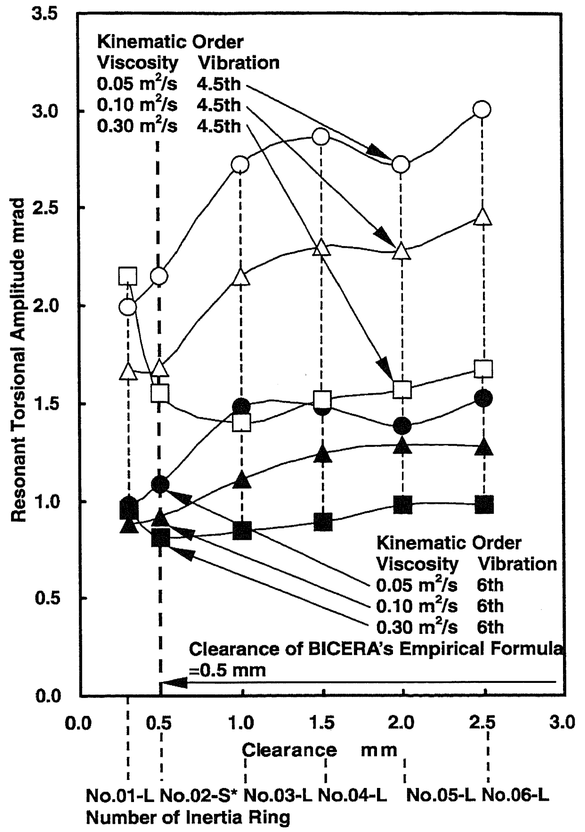

Fig. 9(a) Relationship between Resonant Torsional Amplitude and Clearance Gap (Gap Change of Only Lateral Clearance)

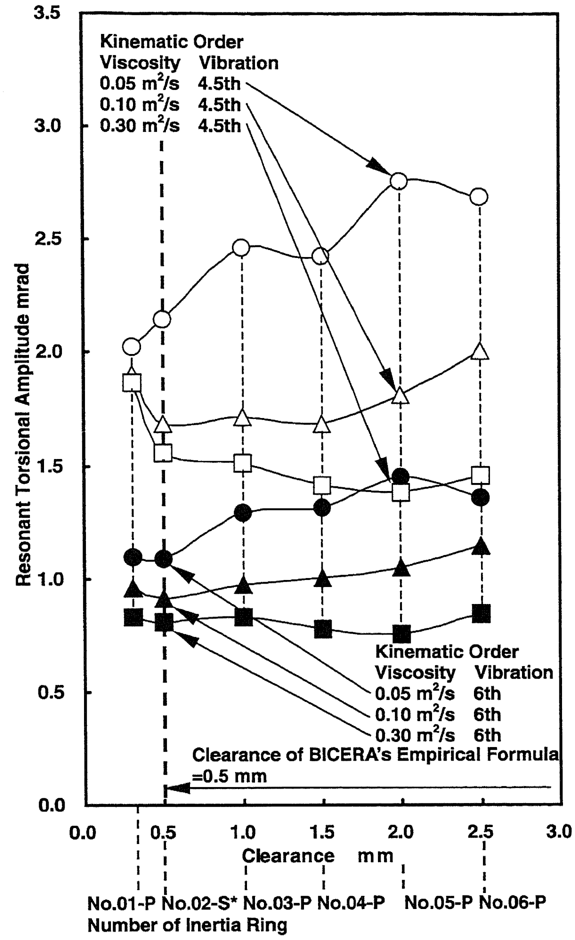

Fig. 9(b) Relationship between Resonant Torsional Amplitude and Clearance Gap (Gap Change of Only Peripheral Clearance)

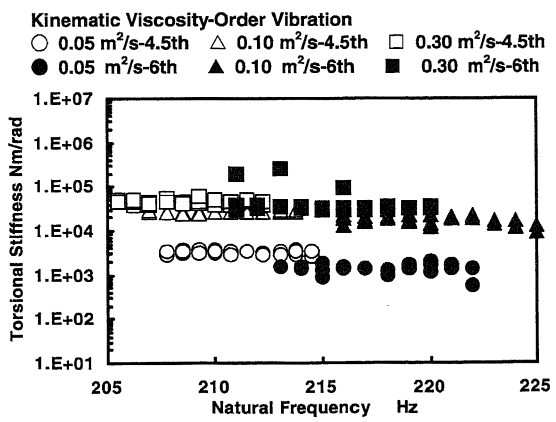

Fig. 10(a) Relationship between Torsional Stiffness and Natural Frequency (Number of Inertia Ring: No. 04–P)

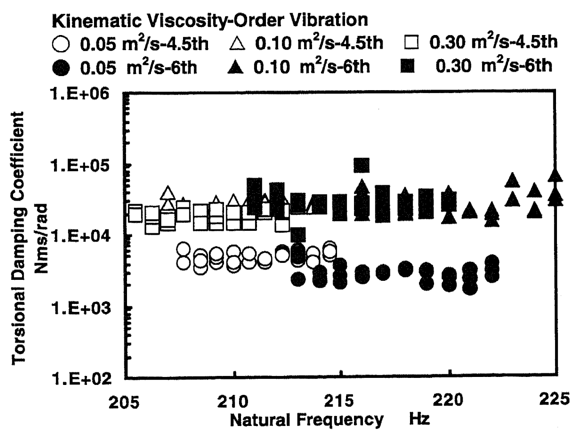

Fig. 10(b) Relationship between Torsional Damping Coefficient and Natural Frequency (Number of Inertia Ring: No. 04–P)

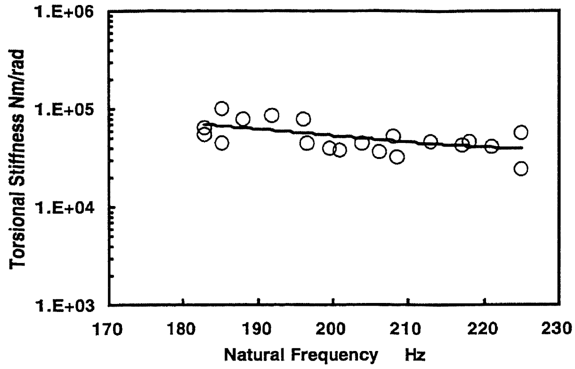

Fig. 11(a) Relationship between Torsional Stiffness and Natural Frequency (Kinematic Viscosity: 0.30 m^2/s)

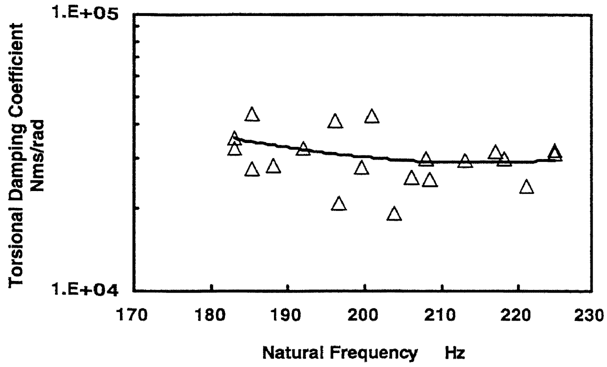

Fig. 11(b) Relationship between Torsional Damping Coefficient and Natural Frequency (Kinematic Viscosity: 0.30 m^2/s)

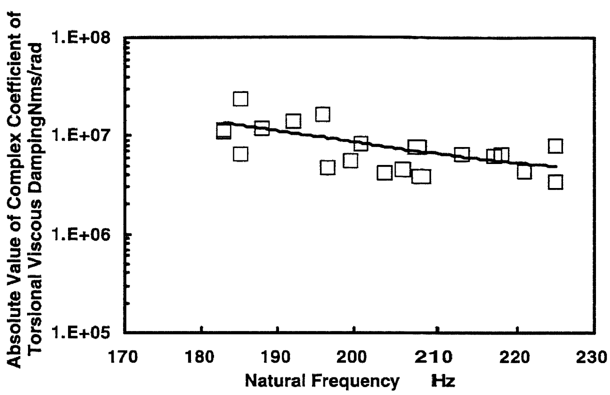

Fig. 11(c) Relationship between Absolute Value of Complex Coefficient of Torsional Viscous Damping and Natural Frequency (Kinematic Viscosity: 0.30 m^2/s)

this experiment. The values of the torsional stiffness and the torsional damping coefficient become larger with the increase of the viscosity. Judging from the results, considered in the design process of the viscous-friction damper.

7.6 Relationship between Torsional Stiffness, Torsional Damping Coefficient, Absolute Valus of Complex Coefficient of Torsional Viscous Damping and Natural Frequncy

Figure 11(a), (b), (c) illustrate the relationship between torsional stiffness (refer to equation (8)), torsional damping coefficient (refer to equation (9)) or absolute value of complex coefficient of torsional viscous damping (refer to equation (5) and natural frequency, respectively, on the kinematic viscosity of 0.30 m^2/s. The main purpose of the experiment is to investigate especially the dynamic characteristics of the viscous damper filled with high viscosity fluid which has a great effect on the natural frequency of the torsional vibration of the engine crankshaft system. The absolute value of the complex coefficient of torsional viscous damping has approximately the tendency to decrease with the increase of natural frequency. The dynamic torsional stiffness, the torsional damping coefficient and the absolute value of the complex coefficient of the torsional viscous damping have a tendency to decrease slightly with the increase of the natural frequency. They depend on the natural frequency.

7.7 Relationship between Torsional Stiffness, Torsional Damping Coefficient, Absolute Valus of Complex Coefficient of Torsional Viscous Damping and Damper Coefficient

Figure 12(a), (b), (c) illustrate the relationship between torsional stiffness, torsional damping coefficient, or absolute value of complex coefficient of torsional viscous damping and damper coefficient (refer to equation (3)), respectively, on the kinematic viscosity of 0.30 m^2/s. The dynamic torsional stiffness, the torsional damping coefficient and the absolute value of the complex coefficient of the torsional viscous damping have a tendency to increase with the increase of the damper coefficient.

8. Conclusions

The experiments, in which the clearance dimensions of the peripheral and lateral gaps filled with viscous fluid and also the kinematic viscosity of its silicone fluid are diverse-

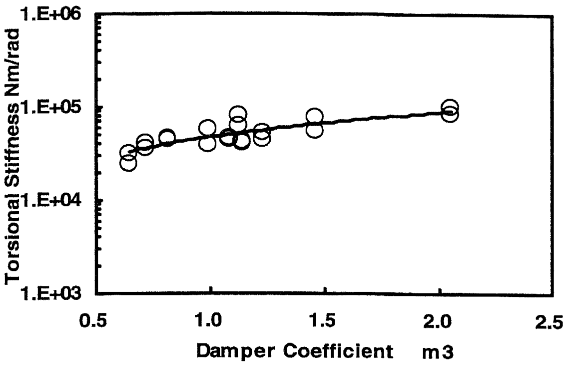

Fig. 12(a) Relationship between Torsional Stiffness and Damper Coefficient (Kinematic Viscosity: 0.30 m^2/s)

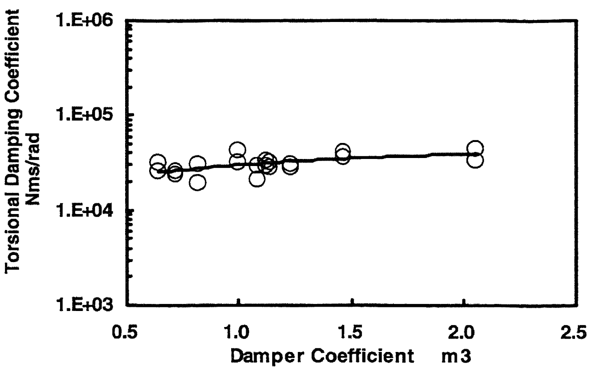

Fig. 12(b) Relationship between Torsional Damping Coefficient and Damper Coefficient (Kinematic Viscosity: 0.30 m^2/s)

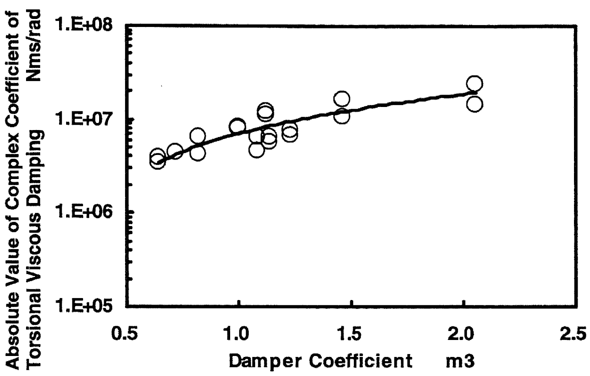

Fig. 12(c) Relationship between Absolute Value of Complex Coefficient of Torsional Viscous Damping and Damper Coefficient (Kinematic Viscosity: 0.30 m^2/s)

ly varied, were been carried out and the angular displacements of torsional vibration at the damper inertia ring and the damper casing were measured simultaneously. As the results of the experiments on the dynamic properties of the viscous-friction dampers, the following conclusions are obtained;

[1] The value of the clearance gap determined from BICERA's empirical formula is not always optimum on the every viscosity of the filled silicone fluid of the viscous-friction dampers.

[2] The optimum viscosity is different by the dimensions of the peripheral and lateral gaps between the outside of the damper inertia ring and the inside of the damper casing.

[3] If the viscosity of the filled silicone fluid is originally decided, the optimum clearance exists in relation to its viscosity.

[4] The clearance gap of less than 0.5 mm in the range of the low viscosity of the silicone fluid is more effective on the reduction of the resonant torsional amplitude than the clearance gap of 0.5 mm obtained by BICERA's empirical formula. But the resonant torsional amplitude in the clearance gap of more than 0.5 mm becomes smaller in the range of the high viscosity of the silicone fluid than that in the clearance gap of 0.5 mm. And the optimum clearance gap of the damper filled with the viscosity of the 0.10 m^2/s is nearly 0.5 mm.

[5] The torsional stiffness, torsional damping coefficient and the absolute value of the complex coefficient of torsional viscous damping has approximately the tendency to decrease with the increase of the natural frequency.

[6] The torsional stiffness, torsional damping coefficient and absolute value of the complex coefficient of torsional viscous damping has approximately the tendency to increase with the decrease of the damper coefficient.

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Definitions

The main symbols used in this paper are as follows.

b: Width of damper inertia ring [mm]

C_d: Real part of complex coefficient of torsional viscous damping (or torsional damping coefficient) [Nms/rad]

C_d^*: Complex coefficient of torsional viscous damping (or complex torsional damping coefficient of torsional viscous-friction damper) [Nms/rad]

$|C_d|$: Absolute value of complex coefficient of torsional viscous damping [Nms/rad]

D_o: Outer diameter of damper inertia ring [mm]

G'/ω: Imaginary part of complex coefficient of torsional viscosity of silicone fluid [Ns/m^{2}rad]

h_i: Inner-peripheral gap between damper casing and damper inertia ring [mm]

h_o: Outer-peripheral gap between damper casing and damper inertia ring [mm]

h_s: Lateral gap between damper casing and damper inertia ring [mm]

I_d: Inertia moment of damper inertia ring [kgm^2]

I_e: Inertia moment of engine crankshaft system [kgm^2]

j: $(=\sqrt{-1})$

K: Torsional damper coefficient [mm^3]

K_d: Torsional stiffness of torsional viscous-friction damper [Nm/rad]

M: Amplitude ratio

R_i: Inner radius of damper inertia ring [mm]

R_o: Outer radius of damper inertia ring [mm]

t: Time [sec]

δ: Clearance of standard viscous-friction damper [mm]

θ_d: Torsional angular displacement of damper inertia ring [rad]

θ_p: Torsional angular displacement of damper casing [rad]

μ^*: Complex coefficient of torsional viscosity of silicone fluid [Ns/m^{2}rad]

μ': Real part of complex coefficient of torsional viscosity of silicone fluid [Ns/m^{2}rad]

ϕ: Phase angle of between damper inertia ring and damper casing [rad]

ω: Circular frequency of i-th order vibration [rad/s]