Torsional Vibration Damping of Diesel Engine with Rubber Damper Pulley*

Katsuhiro WAKABAYASHI**, Yasuhiro HONDA**, Tomoaki KODAMA** and Kunio SHIMONYAMADA**

In this paper, we first describe the characteristics of engine damping that are necessary for the calculation of torsional vibration. Secondly, we describe an experiment in which two crankshaft pulleys with a torsional rubber damper are fitted to a 6-cylinder, high-speed diesel engine. The torsional waveforms of the damper inertia ring and the pulley are measured by means of phase-shift torsigrahphotograph equipment. The measured waveforms are harmonically analyzed and the dynamic characteristics of the stiffness and the damping are investigated from an experimental viewpoint. In addition, torsional waveforms are calculated by means of a method of simulating torsional vibration, in which a transition matrix method is adopted. As a result of comparisons with experimental data, certain dynamic characteristics of pulleys with a torsional rubber damper have been clarified.

Key Words: Damper, Damping, Torsional Vibration, Forced Vibration, Free Vibration, Dynamic Stiffness, Transition Matrix Method

1. Introduction

In recent years, low exhaust gas pollution and superior quality in addition to high performance and low fuel consumption have come to be in greater demand. Two of the important requirements in the design stage are that the vibration and noise levels of automobile engines be reduced and that the timbre be improved further. Torsional vibration dampers of high performance\(^{1(1)-(9)}\), and crankshaft pulleys with a torsional rubber damper\(^{1(11)}\) (hereafter called “damper pulleys”), which can reduce both torsional and bending vibrations, and flywheels with a torsional rubber damper\(^{1(13)}\) have been widely employed in automobile engines, especially in supercharged diesel engines, as measures for vibration and noise reduction. The torsional rubber dampers (hereafter called “dampers”) have been employed most widely in automobile engines.

When designers estimate the amplitude of angular displacement of crankshafts with the above-mentioned dampers in the design process, the following issues are of concern:

(1) the estimation of torsional vibration damping of the engine.

(2) the estimation of dynamic torsional stiffness and damping of the damper.

These dynamic characteristics are indeterminable. Therefore, an experimental equation for determining the values has not existed until now and the quantitative values are inaccurate. Unless these values can be accurately estimated to a certain degree, it is difficult to achieve high-precision computation.

First, we further investigate the experimental results, previously reported only as experimental data in Ref.\(^{(6)}\), of the crankshaft system without a damper by the free vibration method, which has been developed by us, and consider the characteristics of torsional vibration damping of the engine.

Next, we clarify some of the complicated dynamic characteristics of the rubber vibration isolator by measuring the torsional vibrations of the
crankshaft system with a damper pulley.

Finally, we investigate the obtained experimental data from an analytical viewpoint by adopting "a simulation method of the torsional vibration waveform of the crankshaft system by the transition matrix method" [9] - [12].

**Notations**

The main symbols used in this paper are as follows:

- \( b \) : Thickness of the rubber at the representative radius m
- \( C_d \) : Damping coefficient of the rubber part of a rubber damper pulley Nms/rad
- \( I_a \) : Inertia moment of the damper inertia ring kgm²
- \( K_d \) : Dynamic torsional stiffness of the rubber part of a rubber damper pulley Nm/rad
- \( K_d' \) : Complex torsional stiffness of the rubber part of a rubber damper pulley Nm/rad (= \( K_d + jC_d \cdot \omega \cdot \omega \))
- \( |K_d'| \) : Absolute torsional stiffness of the rubber part of a rubber damper pulley Nm/rad (= \( \sqrt{K_d^2 + (C_d \cdot \omega)^2} \))
- \( l \) : Loss factor (= \( C_d \cdot \omega / K_d \))
- \( M \) : Amplitude ratio (= \( \theta_a / \theta_d \))
- \( T_d \) : Period of damped torsional vibration s
- \( W_e \) : Energy expended by excitation torque of torsional vibration Nm
- \( W_h \) : Energy dissipated by hysteresis loss Nm
- \( r_s, r_i \) : Outer and inner diameters of the rubber part, respectively m
- \( \alpha \) : Decay constant s⁻¹
- \( \gamma \) : Strain rate 1/s
- \( \xi \) : Damping ratio
- \( \theta_a \) : Torsional angular displacement of the damper inertia ring rad
- \( \theta_b \) : Torsional angular displacement of the pulley rad
- \( \theta_r \) : Relative torsional angular displacement of the rubber part of a rubber damper pulley rad (= \( \theta_a - \theta_b \))
- \( \Lambda \) : Logarithmic decrement
- \( \rho \) : Representative radius of the rubber part of a rubber damper pulley m
- \( \phi \) : Phase angle between the damper inertia ring and pulley rad
- \( \omega \) : Angular frequency of torsional vibration rad/s
- \( \omega_n \) : Natural angular frequency rad/s

2. **Experimental Apparatus and Method**

This section concerns the main specifications of the test engine and pulley damper, as well as the method of measuring the torsional vibration waveforms. Since the method of experiment used in section 3 is explained in detail in Ref. (6), we describe it only briefly here. In the experiment, the engines are operated under full load to cause steady forced torsional vibration. Under such conditions, the combustion is stopped abruptly either by stopping the inflow of fresh air by closing the air intake manifold or by using a decompression valve to obtain free damped torsional vibration waveforms.

### 2.1 Main specifications of test engine and pulley damper

The test engine is a 6-cylinder, in-line, high-speed diesel engine, and the main specifications are shown in Table 1. Figure 1 shows the dimensions of the test pulley damper and Table 2 shows the parameters. This table also shows the static stiffness and

<table>
<thead>
<tr>
<th>Items</th>
<th>Contents</th>
</tr>
</thead>
<tbody>
<tr>
<td>Designed for</td>
<td>Automobile high speed diesel engine</td>
</tr>
<tr>
<td>Type of engine</td>
<td>4-cycle, direct Injection</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>6 cylinders</td>
</tr>
<tr>
<td>Arrangement</td>
<td>In-line</td>
</tr>
<tr>
<td>Bore and stroke</td>
<td>m 0.105 - 0.125</td>
</tr>
<tr>
<td>Total stroke volume</td>
<td>m³ 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 brake torque Nm / r / min</td>
<td>451 / 1800</td>
</tr>
<tr>
<td>Firing order</td>
<td>1 - 5 - 3 - 6 - 2 - 4</td>
</tr>
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</table>

### Table 2 Parameters of the test rubber damper pulley

<table>
<thead>
<tr>
<th>Items</th>
<th>Damper A</th>
<th>Damper B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured natural frequency of damper Hz</td>
<td>148.0</td>
<td>229.0</td>
</tr>
<tr>
<td>Inertia moment of inertia ring kgm²</td>
<td>0.0369</td>
<td>0.0369</td>
</tr>
<tr>
<td>Inertia moment of pulley kgm²</td>
<td>0.0462</td>
<td>0.0462</td>
</tr>
<tr>
<td>Static torsional stiffness Nm / rad</td>
<td>2.548 x 10⁴</td>
<td>3.332 x 10⁴</td>
</tr>
<tr>
<td>Material of rubber</td>
<td>Natural rubber and nitrile rubber</td>
<td></td>
</tr>
</tbody>
</table>

![Fig. 1 Dimensions of the test rubber damper pulley](image-url)
natural frequency obtained from the static and free vibration tests, respectively. Types A and B of the damper pulleys differ in Shore hardness but their rubber parts have the same shape and dimensions. The materials are similar—quality rubber.

2.2 Method of measuring torsional vibration waveform

An eddy current dynamometer was connected via a universal joint to the flywheel of the engine. The test damper pulley was fitted to the end of the crankshaft. The gears for generating signal pulses were mounted on the damper inertia ring and pulley. The electric frequency signals proportional to engine speed were obtained from the electromagnetic pickup. The measured signals were transmitted to the phase-shift torsigraphe 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 measured torsional waveforms of the damper inertia ring and pulley were harmonically analyzed using the F.F.T. analyzer. The torsional angular displacement was measured under full load from 800 r/min to 3 200 r/min. The indicator diagrams, data from which were necessary for the vibration analysis described later, were measured using the piezotype indicator in the sixth cylinder from the pulley side.

3. Characteristics of Torsional Vibration Damping of Engine

3.1 Definition of decay constant and relationship between decay constant and damping ratio

The decay constant and damping ratio can be calculated from the damped free torsional vibration waveforms. The value of \( \alpha \), which can be obtained directly from the decay torsional vibration record, is defined by the following equation and is called the decay constant.

\[
\alpha = \frac{A}{T_d}
\]

Since the logarithmic decrement and period of damped torsional vibration can be obtained from the decay torsional vibration curve, the value of \( \alpha \) is determined by Eq. (1).

Next, the following equation is given among decay constant \( \alpha \), damping ratio \( \xi \) and natural angular frequency \( \omega_n \).

\[
\alpha = \xi \cdot \omega_n
\]

If the values of \( \alpha \) and \( \omega_n \) are known, the damping ratio can be determined.

<table>
<thead>
<tr>
<th>Number</th>
<th>Decay constant</th>
<th>I / s</th>
</tr>
</thead>
<tbody>
<tr>
<td>All rings installed</td>
<td>4.3</td>
<td>6.0</td>
</tr>
<tr>
<td>No.4 ring removed</td>
<td>3.7</td>
<td>0.0</td>
</tr>
<tr>
<td>No.2 and No.3 rings removed</td>
<td>3.5</td>
<td>8.0</td>
</tr>
<tr>
<td>No.2, No.3 and No.4 rings removed</td>
<td>2.8</td>
<td>0.0</td>
</tr>
</tbody>
</table>

3.2 Characteristics of torsional vibration damping of engine

In order to investigate the characteristics of torsional vibration damping of engines, the free vibration experiments were conducted under the condition that the number of piston rings was changed for all the cylinders, which entails an investigation of the characteristics of vibration damping, such as damping ratio, on the basis of Eq. (2) to investigate the decay constant of the same engine. The engine chosen as the object of this investigation (total stroke volume: 10.2 L, maximum brake power: 143 kW/2,400 r/min, maximum brake torque: 667 Nm/1,600 r/min) had six cylinders, each fitted with three compression rings and two oil-control rings. We conducted free vibration experiments in which the number of rings was changed for all the cylinders, as described below: (1) all the piston rings were used, (2) No. 4 oil-control ring was removed, (3) No. 2 and No. 3 compression rings were removed, and (4) No. 2, No. 3 and No. 4 rings were removed. The results of measured decay constant are shown in Table 3. The piston rings were called No. 1, No. 2, ..., No. 5 from the cylinder head. The experimental data have already been reported in Ref. (6), as mentioned above, but we added our consideration concerning the experimental results from a new standpoint.

From the experimental results shown in Table 3, the piston rings seem to greatly influence torsional vibration damping, particularly when oil-control rings are removed. Thus, we investigated the amount of energy dissipated by the dampings of the compression and oil-control rings. If it is assumed that the energies dissipated by the dampings of the compression rings, the oil-control rings and the other parts are constant, the algebraic equations with the dissipated energy are obtained from four experimental values shown in Table 3. The experiments are called experiment 1, 2, 3 and 4 from the top to the bottom. The difference in the results between experiments 1 and 3 shows the dissipated damping energy upon the removal of two compression rings. The difference in the results between experiments 1 and 3, or experiments 3 and 4 is the decay constant equivalent to the
damping energy dissipated by one oil-control ring. Since two values (nearly equal values) of the decay constant, for one compression ring and one oil-control ring, were obtained by solving the algebraic equations, the value of the decay constant for one compression ring and one oil-control ring was determined by calculating the average of the two values. The value of the decay constant for one compression ring is 4.2 1/s and that of the decay constant for one oil-control ring is 7.2 1/s. Hence the dissipated damping energy for one oil-control ring is nearly 1.7 times as large as that for one compression ring. All of the piston rings share 60 percent of the total dissipated damping energy.

Next, the breakdown of the remaining 40 percent of the total dissipated damping energy, which was shared by parts other than the piston rings, was investigated. Table 4 shows the values of damping energy dissipated by hysteresis loss $W_h$, which were obtained using the highly reliable Lewis formula(1), and those of the ratio of $W_h$ to $W_e$ ($W_e$: work input by excitation torque). Judging from the calculated results shown in Table 4, it seems that the damping energy dissipated by hysteresis loss is less than 10 percent of the total damping energy, and that the other parts, such as bearing parts, share 30 percent.

The number of piston rings in the latest, high-speed diesel engines has been decreasing, so engines generally have two pistons fitted with two compression rings and one oil-control ring. In this case, the above-mentioned proportion of dissipated damping energy is changed and the amount related to piston ring parts is naturally reduced. The following was clarified after recalculation of the proportion of dissipated damping energy for piston ring parts on the basis of the previously obtained results: (1) all piston rings share 45 percent of the total dissipated damping energy, (2) parts other than piston rings share 55 percent of the total damping energy, and (3) energy dissipated by hysteresis loss is less than 15 percent of the total damping energy.

4. Dynamic Characteristics of Rubber Parts of Pulley Dampers

4.1 Measured torsional vibration waveforms

Figure 2 illustrates the measured torsional vibration waveforms in the vicinity of the 6th-order resonant engine speed of 2635 r/min as an example of the results of measurement at the damper inertia ring and pulley of the crankshaft system with the test damper pulley. In order to further study the contents of both waveforms, they were harmonically analyzed and compared with each other at each main order, as shown in Fig. 2.

The measured waveform of the pulley is the waveform in which the 6th-order amplitude is amplified due to the measurement at the 6th-order resonant point. On the other hand, the measured waveform of the damper inertia ring via the rubber part shows a slightly different tendency, in which the 6th-order amplitude is not particularly amplified and, in addition, the 3rd- and 4.5th-order amplitudes are large.
4.2 Measured amplitude curves of angular displacement

Figure 3 illustrates the measured amplitude curves of angular displacement at the pulley end of the crankshaft without a damper pulley. Figure 4 shows the measured amplitude curves of angular displacements at the pulley and the damper inertia ring of the crankshaft with the B-type damper pulley. These figures show merely the main-order amplitude curves of angular displacements. The large 6th-order resonant point occurs at 2,524 r/min and its resonant amplitude is $14.9 \times 10^{-3}$ rad in Fig. 3 in the case of installing no damper pulley. Since the tuning ratio of the B-type damper pulley is not optimum, the 6th-order amplitude is not greatly reduced.

4.3 Calculation of dynamic characteristic values from measured waveform

The values of dynamic torsional stiffness and damping coefficient of damper rubber parts can be obtained from the harmonically analyzed results of waveforms measured at the damper inertia ring and the pulley. The equation of motion at the damper inertia ring is

$$I_a \cdot \dot{\theta}_d + C_d (\dot{\theta}_d - \dot{\theta}_p) + K_d (\theta_d - \theta_p) = 0, \quad (3)$$

where $\dot{\theta}_d = \theta_d \cdot e^{i\omega t}$ and $\dot{\theta}_p = \theta_p \cdot e^{i\omega t}$.

The following Eq. (4) can be obtained by rearranging the expression of Eq. (3) by substituting the above-mentioned relational expression into Eq. (3) and rearranging:

$$K_d = -\frac{I_a \cdot \omega^2 \cdot M \cdot (M - \cos \phi)}{M^2 + 2 \cdot M \cdot \cos \phi}$$

$$C_d = -\frac{I_a \cdot \omega^2 \cdot M \cdot \sin \phi}{M^2 + 2 \cdot M \cdot \cos \phi}$$

where $M = \theta_d \cdot \theta_p$.

In Eq. (4), the values of amplitude ratio $M$ and phase angle $\phi$ can be obtained by analyzing harmonically the waveforms measured at the damper inertia ring and the pulley, where the values of $I_a$, $\omega$ are known. Therefore, the values of dynamic torsional stiffness $K_d$ and damping coefficient $C_d$ can be determined from Eq. (4). Table 5 shows the values of $K_d$ and $C_d$, and those of absolute torsional stiffness $|K|_1$ and loss factor $l$ which can be calculated using

![Fig. 3 Measured amplitude curves of angular displacement (without rubber damper pulley)](image)

![Fig. 4 Measured amplitude curves of angular displacement (with B-type rubber damper pulley)](image)

Table 5 Dynamic torsional spring constant, damping coefficient and loss factor

(a) part I

<table>
<thead>
<tr>
<th>Name of damper</th>
<th>Order vibration</th>
<th>Natural frequency</th>
<th>Strain rate $H_z$</th>
<th>Torsional spring constant $K_d$ Nm/rad</th>
<th>Damping coefficient $C_d$ Nms/rad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damper A</td>
<td>4.5th</td>
<td>132.8</td>
<td>575.27</td>
<td>4.55 x 10^4</td>
<td>27.56</td>
</tr>
<tr>
<td></td>
<td>6th</td>
<td>143.6</td>
<td>382.89</td>
<td>9.67 x 10^4</td>
<td>17.73</td>
</tr>
<tr>
<td></td>
<td>6th(l)</td>
<td>259.5</td>
<td>920.00</td>
<td>1.99 x 10^5</td>
<td>10.73</td>
</tr>
</tbody>
</table>

(b) part II

<table>
<thead>
<tr>
<th>Name of damper</th>
<th>Order vibration</th>
<th>Absolute torsional spring constant $k_a$ Nm/rad</th>
<th>Loss factor $l$</th>
<th>Imaginary part of complex torsional spring constant $C_d$ Nms/rad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Damper A</td>
<td>4.5th</td>
<td>5.01 x 10^4</td>
<td>2.30 x 10^4</td>
<td>3.01 x 10^4</td>
</tr>
<tr>
<td></td>
<td>6th</td>
<td>9.90 x 10^4</td>
<td>1.63 x 10^4</td>
<td>6.00 x 10^4</td>
</tr>
<tr>
<td></td>
<td>6th(l)</td>
<td>2.00 x 10^5</td>
<td>7.75 x 10^4</td>
<td>1.75 x 10^4</td>
</tr>
<tr>
<td>Damper B</td>
<td>4.5th</td>
<td>1.50 x 10^4</td>
<td>5.60 x 10^4</td>
<td>3.73 x 10^4</td>
</tr>
<tr>
<td></td>
<td>6th</td>
<td>1.02 x 10^4</td>
<td>4.02 x 10^4</td>
<td>1.40 x 10^4</td>
</tr>
<tr>
<td></td>
<td>6th(l)</td>
<td>2.30 x 10^4</td>
<td>4.10 x 10^4</td>
<td>8.95 x 10^4</td>
</tr>
</tbody>
</table>

I: 1st node vibration, II: 2nd node vibration

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the values of $K_d$ and $C_d$.

The dynamic characteristics of the vibration isolator rubber depend generally on (1) temperature, (2) frequency, and (3) average strain and strain amplitude effects on the basis of same shape and material. To reveal the effect of one factor among the above-mentioned factors on the dynamic characteristics, it is necessary to keep the values of the other factors constant in the experiment. In our experiments, the surface temperature of the rubber part was kept at 313 K, but the other factors could not be controlled due to the vibration characteristics of the crankshaft. In full consideration of these conditions of the other factors, strain rate $\gamma$, which is the product of strain amplitude and angular frequency, is defined by the following equation and the experimental results in Table 5 are arranged on the basis of strain rate.

$$\theta = \theta + \omega \cdot \rho / b$$  \hspace{1cm} (5)

\[ \rho = \frac{3(r^2 - r_f^2)}{4(r^2 - r_t^2)} \]  \hspace{1cm} (6)

Equation (6) is obtained for the uniform, hollow, circular cross section under the condition that the torsional moment (in such a case that the shearing stress in the representative radius $\rho$ is uniformly distributed over the entire cross section) is equal to that (in the case that shearing stress distribution $\tau = G \cdot \rho \cdot C_d$) caused by the simple shearing deformation in the circumferential direction.

Figure 5 shows the relationship between absolute torsional stiffness and strain rate. Figure 6 illustrates the relationship between loss factor and strain rate. Both Figs. 5 and 6 are aimed at expressing the absolute torsional stiffness and the loss factor as a function of the strain rate on the condition that the nonlinear dynamic characteristic values of the damper rubber part are assumed to be approximately expressed by linear equations. The values of the dynamic torsional stiffness $K_d$ and the damping coefficient $C_d$ (or $C_{d\omega}$), which are necessary as input data for the simulation analysis in section 5, are arranged in the same way as shown in Figs. 5 and 7, respectively. The value of absolute torsional stiffness can be determined by $|K_d| = \sqrt{(K_d)^2 + (C_{d\omega})^2}$, but the following expression applies under the condition that the value of $C_{d\omega}$ is fairly small in comparison with the value of $K_d$; $|K_d| = K_d$. Therefore, the values of $|K_d|$ and $K_d$ approximately agree in the figure. The curves obtained by the curve-fitting method are illustrated in Figs. 5, 6 and 7. The values of the correlation coefficients against the curves of Dampers A and B are in the regions of 0.82-0.89 and 0.86-0.94, respectively.

5. Analytical Investigation of Torsional Vibration Characteristics of Crankshaft with Damper Pulley

5.1 Method of simulating torsional vibration waveforms

The method of simulating the torsional vibration waveform by the transition matrix method, which has been developed by us, is a step-by-step numerical calculation method, which makes indirect use of Taylor series. Ref. (6) contains the details of the derivation of the equations related to this simulation method. The equation, considering up to the fourth derived function, at the final stage necessary for the calculation is

$$\begin{bmatrix} \theta \\ \dot{\theta} \end{bmatrix}_{s+1} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \begin{bmatrix} \theta \\ \dot{\theta} \end{bmatrix}_s + \begin{bmatrix} B_{11} & B_{12} \\ B_{21} & B_{22} \end{bmatrix} \begin{bmatrix} \ddot{\theta} \\ \dot{\theta} \end{bmatrix}_s + \begin{bmatrix} C_{11} & 0 \\ 0 & C_{12} \end{bmatrix} \begin{bmatrix} \dddot{\theta} \\ \ddot{\theta} \end{bmatrix}_s$$  \hspace{1cm} (7)

where $A_{ij}$, $B_{ij}$, and $C_{ij}(i=1,2, j=1,2)$, which

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constitute the transition matrix, are partial $n \times n$ matrices. The partial matrices are determined from step size and various factors constituting the equivalent vibration systems. The excitation torque $F_{a}$ and the time differentials $\dot{F}_{a}$ and $\ddot{F}_{a}$ can be determined by adopting the measured indicator diagram and the specifications of reciprocating mass parts, and $\theta_{a1}$ and $\theta_{a2}$ can be calculated by giving the initial values of $\theta_{a}$ and $\dot{\theta}_{a}$. This calculation can be repeated continuously. Since no steady waveform was obtained for the first two or three periods, the subsequent periods were adopted as the final results.

Since the values of the exciting torque are equal to zero in the case of free vibrations, only the first term in Eq. (7) need be considered.

5.2 Input data necessary for calculation of torsional vibration waveforms

Figure 8 shows the equivalent torsional vibration system of a crankshaft with a rubber damper pulley replaced according to the analytical method described in section 5.1. The rubber part of the damper pulley is replaced with a Voight model. Table 6 shows the list of data, including the numerical values of the equivalent vibration systems, which are used for the numerical calculation.

Figure 9 illustrates the relationships between the decay constant $a$, the natural circular frequency $\omega_{n}$ and the damping ratio $\zeta$ obtained by the vibration experiments with six types of in-line, high-speed diesel engines (cylinder bore: 83-120 mm, total stroke volume: 2-10 L) (see Ref. (6)). The relationship between the engine damping coefficient $C_{e}$ and the decay constant $a$ for each cylinder can be obtained by the analysis of free vibration of the damped multimass torsional vibration system proposed by Kanda (10). Therefore, the value of $\omega_{n}$ can be calculated by Holzer's method and that of $a$ can be determined by referring to Fig. 9. Lastly, the value of $C_{e}$ ($C_{e} = C_{i}$ in Fig. 8) can be determined from the

Table 6 Numerical values of the equivalent vibration systems (with a rubber damper pulley)

<table>
<thead>
<tr>
<th>Number of mass</th>
<th>Name of mass</th>
<th>Inertia moment $G \cdot I_{d} / K_{a}$ $km^{2}$</th>
<th>Equivalent length $m$</th>
<th>Damping coefficient of engine Nms/$rad$</th>
<th>Damping coefficient of damper Nms/$rad$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Inertia ring</td>
<td>0.0360</td>
<td>9.38655 x 10^{-7}</td>
<td>0.000</td>
<td>0.000</td>
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<tr>
<td>2</td>
<td>Damper pulley</td>
<td>0.0462</td>
<td>21.3259</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>3</td>
<td>No.1 cylinder</td>
<td>0.0328</td>
<td>6.6196</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>4</td>
<td>No.2 cylinder</td>
<td>0.0225</td>
<td>6.6196</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>5</td>
<td>No.3 cylinder</td>
<td>0.0328</td>
<td>6.6196</td>
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<td>0.000</td>
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<tr>
<td>6</td>
<td>No.4 cylinder</td>
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<td>6.6196</td>
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<td>0.000</td>
</tr>
<tr>
<td>7</td>
<td>No.5 cylinder</td>
<td>0.0225</td>
<td>6.6196</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>8</td>
<td>No.6 cylinder</td>
<td>0.0328</td>
<td>4.0626</td>
<td>0.000</td>
<td>0.000</td>
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<td>9</td>
<td>Flywheel</td>
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<td>500.0000</td>
<td>0.000</td>
<td>0.000</td>
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<tr>
<td>10</td>
<td>Dynamometer</td>
<td>4.5869</td>
<td></td>
<td>0.000</td>
<td>0.000</td>
</tr>
</tbody>
</table>

($G \cdot I_{d} = 9.38655 \times 10^{-7} \text{ Nm}^{2}, K_{a} = \text{Nms}/\text{rad}$)

Fig. 9 Relationship between decay constant, natural angular frequency and damping ratio

Fig. 8 Equivalent torsional vibration system of an engine system with a rubber damper pulley

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relationship between \( \alpha \) and \( \omega_n \) by the Kanda method. The damping coefficient of the engine is 6.86 Nms/rad (constant), as shown in Table 6.

Since the simulation program can yield the torsional vibration waveform for a given engine speed, the value of the strain rate is calculated using the measured dominant-order amplitude of angular displacement at a given engine speed. Then, the values of the dynamic torsional stiffness \( K_a \) and damping coefficient \( C_a \) of the rubber part can be obtained by referring to Figs. 5 and 7, respectively. These obtained values of \( K_a \) and \( C_a \) are used as the input data for the calculation. Since the curve-fitted curves in Figs. 5 and 7 are representative of all the experimental data, the obtained values of \( K_a \) and \( C_a \) can be regarded as the mean values.

5.3 Calculated results and considerations

Figure 10 shows the main-order amplitude curves of the angular displacements obtained by analyzing harmonically the calculated torsional vibration waveforms of the crankshaft without a damper pulley. The calculated amplitude curves in this figure correspond to the measured amplitude curves of the angular displacements in Fig. 3. The measured amplitude curves and the calculated amplitude curves are generally very similar to one another in the magnitude of resonant amplitude and resonant engine speed. From this finding, it has been verified that the value of the damping coefficient of the engine is reasonable.

Figure 11 shows the main-order amplitude curves of the angular displacements at the damper inertia ring and the pulley obtained by analyzing harmonically the calculated torsional vibration waveforms of the crankshaft with the rubber damper pulley. The calculated amplitude curves in this figure correspond to the measured amplitude curves of the angular displacements in Fig. 4. As compared with the experimental results, these calculated results contain slight error, but this degree of error is allowable in practice.

6. Conclusion

Accurate estimation of torsional vibration damping of the engine part and the dynamic characteristics of the rubber part of the damper pulley are problems in the calculation of torsional vibration of a crankshaft with a rubber damper pulley. The results of investigation of these characteristics from experimental and analytical viewpoints are as follows:

1. In recent years, small high-speed diesel engines have been constructed with reduced numbers of piston rings. Therefore, the value of the ratio of the damping energy dissipated by the piston ring to the total damping energy has been reduced, but it is still large.

2. We expressed the absolute torsional stiffness and damping of the rubber part of the damper pulley as a function of the strain rate under the conditions of the same kind rubber material, constant temperature and invariable shape factor.

3. It has been shown that the simulation of torsional vibration waveforms by the transition matrix method can be used to analyze the torsional vibration of a crankshaft with a rubber damper pulley.

References


