STUDY ON VIBRATION CONTROL DEVICE USING POWER GENERATOR

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ABSTRACT
In this paper, the authors propose a vibration control device using power generator in order to develop a small damper which is suitable for structural vibration control in space. The vibration control device consists of a ball screw, piston, ball nut, gear, power generator, and rod ends. The linear motion of the piston is converted into a rotary motion by the ball screw, and then the electric power is generated as dissipation of energy. A very low sinusoidal input displacement is applied to the device, and the characteristics of the damping force are examined with open circuit and short circuit of the generator. Experimental results are compared with theoretical results. Numerical simulation is applied to a flexible structure, by using sinusoidal input motion, and random motion.

1. INTRODUCTION
The authors have recently proposed a vibration device using power generator which has a large damping effect and controllable in damping coefficient, under low frequency ranges such as 0.1 Hz or below [1]. The device will be suitable for vibration control of a flexible structure and a long vibration period, such as in space. On the other hand, the authors have also proposed a cut-off system of vibration control using a liquid inertia mass of water and/or functional fluid [2].

In this paper, an alternative device is studied. The device has a rotating moment of inertia combined with an electric power generator which has a function of damping, instead of liquid inertia mass (hereinafter, referred to as the “V.C.D.”).

Study of a damping device based on electricity generating system was presented by K. Sunakoda, and two others [3]. This device was named as “Mechatro Damper” [3-4]. The “Mechatro Damper” is an effective means for controlling vibration object as a semi-active damper [4-5]. The V.C.D. is similar to the “Mechatro Damper”, but nevertheless is quite different because of utilizing rotating moment of inertia. On the other hand, utilization of auxiliary mass for reducing the response of main mass has a long history. Dynamic damper and tuned mass damper are well-known means. The V.C.D. has a rotating moment of inertia, but it has a function as series inertia mass.

The purpose of this study is to show the reducing characteristics of vibration of the V.C.D., by theory and experiment. Vibration reducing model attached with V.C.D. is also investigated, and numerical simulation results show the V.C.D. is effective for reducing random vibration and/or seismic vibration.

2. STRUCTURE, FUNCTION, AND DAMPING FORCE AND INERTIA FORCE
2.1 Structure of the V.C.D.
The structure of the V.C.D. is shown in Figure 1. It transforms vibrating thrust motion of the piston into a rotational motion using a ball screw and ball nut mechanism. The rotational motion drives the power generator through the speed-increasing gear. The damping force of the V.C.D. is created by consuming the generated electric power, and it depends on the controllable resistance between terminal A and B of the power generator. The rotating disk is attached to the end of ball screw shaft. It has a moment of inertia, and functions as an equivalent mass.
2.2 Damping Force and Inertia Force

The relationship between the displacement $x$ and the angle of rotation $\theta$ can be expressed as:

$$\theta = \frac{2\pi}{l}x$$

(1)

Where $l$ is a lead of ball screw.

Let $\alpha$ be a speed-increasing ratio, then the angular velocity of the power generator shaft is expressed by Eq. (2).

$$\omega = \frac{2\pi\alpha}{l}\dot{x}$$

(2)

Total force $F$ of the V.C.D. consists of the damping force of the power generator, a rotating moment of inertia of the power generator $I_1$, and a rotating moment of inertia $I_2$ of the other rotating parts such as the rotating disk and ball screw shaft. Then the total force can be given by

$$F = \frac{2\pi\alpha}{\eta} \left( \frac{K_T}{R + R_a}\frac{d\theta}{dt} + \frac{2\pi}{\eta l^2} I_2 \frac{d^2\theta}{dt^2} \right)$$

(3)

While the induced electromotive force generated by the power generator, and torque at the input axis can be expressed by following formulas:

$$E = K_E \omega$$

(4)

$$T = K_T I_a$$

(5)

Where, $K_E$, $K_T$, and $I_a$ denote the coefficient of induced electromotive force, the coefficient of generated torque, and the armature current, respectively.

When the resistance $R$ is connected to the terminal of the power generator, the relationship between the armature current and the induces electromotive force is expressed by Eq. (6)

$$I_a = \frac{E}{R + R_a}$$

(6)

Where, $R_a$ is the motor resistance.

Equation (6) shows that the armature current can be controlled, by changing the resistance.

Substitution of Eqs. (4) and (6) into Eq. (5) yields,

$$T = \frac{K_T K_E \omega}{R + R_a}$$

(7)

Substituting Eqs. (1), (2), and (7) into Eq. (3), the total force $F$ of the V.C.D. can be expressed as:

$$F = \frac{1}{\eta} \left( \frac{2\pi\alpha}{l} \left( \frac{K_T}{R + R_a} \dot{x} + I_2 \ddot{x} \right) + \frac{1}{\eta l^2} \frac{2\pi}{\eta} \dot{x} \right) I_2 \ddot{x}$$

(8)

In Equation (8), damping coefficient $c$ can be written as:

$$c = \frac{1}{\eta} \left( \frac{2\pi\alpha}{l} \left( \frac{K_T}{R + R_a} \right) I_2 \right)$$

(9)

Therefore the damping coefficient of the V.C.D. varied as a function of the terminal resistance, and the terminal is short-circuit (that means $R = 0$), the maximum damping coefficient is obtained, the terminal is open-circuit (that means $R = \infty$), the damping coefficient become zero in theoretically.

3. VIBRATION TEST

3.1 Outline of Vibration Test

The V.C.D. was designed and manufactured. Table 1 shows the design parameters. Figure 2 shows the schematic diagram of the vibration test. In testing, the hysteresis loops were obtained by fixing one end of the specimen to a reaction wall with a load cell and applying sinusoidal forced displacement (frequency range: $0.04$ Hz ~ $2.0$ Hz, input displacement: $\pm 5$ mm ~ $\pm 25$ mm) to the other end.

<table>
<thead>
<tr>
<th>Table 1 Design parameters of V.C.D.</th>
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<tbody>
<tr>
<td>Rotary conversion efficiency $\eta$</td>
</tr>
<tr>
<td>Gear’s speed-increasing ratio $\alpha$</td>
</tr>
<tr>
<td>Ball screw’s lead $l$</td>
</tr>
<tr>
<td>Torque constant $K_T$</td>
</tr>
<tr>
<td>Constant of electromotive force $K_E$</td>
</tr>
<tr>
<td>Motor resistance $R_a$</td>
</tr>
<tr>
<td>Motor’s inertia $I_1$</td>
</tr>
</tbody>
</table>

Fig.1 Structure of the V.C.D.

Fig.2 Schematic diagram of the vibration test
3.2 Test Results

The relationship between force and frequency are shown in Figure 3, and the hysteresis loops of the force and displacement are shown in Figure 4.

In Figure 3, theoretical damping values are compared with the experimental values by compensating for the effect of the rotating moment of inertia of the power generator \( I_1 \), and a rotating moment of inertia \( I_2 \) of the other rotating parts and the effect of the friction which occurred at the sliding surface of the piston. Namely, the experimental damping force is determined by eliminating other forces. The results of the experiment give good agreement with the theoretical values in low speed-increasing ratio (4:1), and the experimental values are larger than theoretical values in the case of high speed-increasing ratio. Although there are some differences between theory and experiment, it seems the damping force is proportional to the frequency (velocity).

Figure 4 shows the total forces of the V.C.D., when the speed-increasing ratio is 23 times. Solid line is plotted by using Eq. (8). The hysteresis loop is similar to an ellipse and the axis of abscissa is inclined toward second and forth quadrants in all cases, both in theory and experiment. This inclination results from the effect of rotating moment of inertia.

Judging from Figure 3 and Figure 4, the discrepancies between the theoretical forces and the experimental forces become larger with increasing the speed-increasing ratio of gear. In all cases, the experimental forces are larger than theoretical results. And the efficiencies of the rotating of the speed-increasing gear mechanism which consists of planetary gear system are estimated. The efficiencies are determined as 91% in 4:1 ratio, 81% in 23:1 ratio, and 62% in 107:1 ratio, respectively.

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**Fig. 3** Relation between Force and Frequency

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**Fig. 4** Damping force characteristics

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(c) Speed-increasing ratio 23:1 Input amplitude ±25mm

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(d) Speed-increasing ratio 23:1 Input amplitude ±15mm

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(e) Speed-increasing ratio 23:1 Input amplitude ±5mm
4. VIBRATION REDUCING MODEL AND NUMERICAL SIMULATION

4.1 Vibration Reducing Model

Figure 5 shows a schematic drawing of vibration reducing model by using V.C.D.. The V.C.D. is attached between main mass \( m \) and ground. From Eq. (1), Eq. (2), and Eq. (3), the equivalent mass \( M_e \) can be expressed as:

\[
M_e = \frac{2\pi}{\epsilon^2} \left( I_1 + \alpha^2 I_2 \right)
\]  

(10)

Kinetic energy \( K.E. \), potential energy \( P.E. \), and dissipation energy \( D.E. \) become as:

\[
K.E. = \frac{1}{2} m \dot{x}_0^2 + \frac{1}{2} M_e (\dot{x}_1 - \dot{x}_0)^2
\]  

(11)

\[
P.E. = \frac{1}{2} k(x_1 - x_0)^2
\]  

(12)

\[
D.E. = \frac{1}{2} c(\dot{x}_1 - \dot{x}_0)^2
\]  

(13)

Substitution of Eqs. (11), (12), and (13) into Lagrange's equation of motion, yields:

\[
\dot{m}x_0 + M_e (\ddot{x}_1 - \ddot{x}_0) + c(\dot{x}_1 - \dot{x}_0) + k(x_1 - x_0) = 0
\]  

(14)

\[
M_e (\ddot{x}_1 - \ddot{x}_0) + c(\dot{x}_1 - \dot{x}_0) + k(x_1 - x_0) = 0
\]  

(15)

Substituting the change of coordinate \( y = x_1 - x_0 \), into Eq. (14) and (15), we obtain:

\[
(m + M_e) \ddot{y} + c\dot{y} + ky = -m\ddot{x}_0
\]  

(16)

\[
M_e \ddot{y} + c\dot{y} + ky = 0
\]  

(17)

Where, \( y \) denotes relative displacement of the mass with respect to ground. In Eq. (17), it is clear that a disturbing force doesn’t apply to the equivalent mass \( M_e \). Substituting \( x_0 = X_0 \sin(\omega t) \) into Eq. (14), (15) we obtain the absolute displacement \( x \) of the main mass and the phase angle:

\[
x = X_0 \sqrt{\frac{k - (M_e \omega^2) + (c\omega)^2}{k - \omega^2(m + M_e)}} \sin(\omega t - \theta)
\]  

(18)

\[
\theta = \tan^{-1} \frac{c\omega}{k - M_e \omega^2 + (c\omega)^2}
\]  

(19)

And transmissibility \( T \) can be written as:

\[
T = \sqrt{\frac{(1 - 2\eta^2)^2 + (2\zeta\eta)^2}{(1 + \eta^2)(1 + 2\zeta^2)}}
\]  

(20)

Where

\[
\zeta = \frac{M_e}{m}, \quad \eta = \frac{\omega}{\omega_n}, \quad \omega_n = \sqrt{\frac{k}{m}}, \quad \zeta = \frac{c}{2\sqrt{mk}}
\]


4.2 Numerical Simulation

1) Transmissibility

The transmissibility of the displacement of the main mass with different damping ratio, and with different mass ratio are shown in Figure 6(a) ~ (c). In Figure 6(a). We can see a clear cut-off frequency (transmissibility = zero) when the damping ratio equals zero. And the peak values of the transmissibility in Figure 6(b), (c) are reduced with increasing mass ratio of \( \zeta = M_e / m \) and with increasing damping ratio. This fact shows that the resonance frequency of the main mass can be shifted toward the low frequency domain, by introducing a large rotating moment of inertia to the V.C.D.. It also suggests that the transmissibility can be controlled by changing the terminal resistance of the V.C.D.. The value of \( \zeta = 0.13 \) shows that the main mass \( m \) is 73kg and equivalent mass \( M_e \) is 9.5kg (which only consists of rotating ball screw shaft, without a rotating disk).
2) Time History Analysis

Figure 7(a) ~ (d) shows the time history response of the displacement. A random vibration of the displacement is used. These figures also show that the response of the displacement is reduced with increasing mass ratio of \( \varepsilon = M_e / m \) and with increasing damping ratio. When \( \varepsilon = M_e / m = 5 \), the displacement of the main mass is similar to the input displacement. This fact shows the V.C.D. changes its stiffness for high rigidity, with increasing mass ratio. By selecting an appropriate rotating moment of inertia, and by adjusting a damping coefficient, suitable reduction of the response will be obtained.

5. CONCLUSION

The vibration control device which has a rotating moment of inertia combined with the electric power generator is proposed. From experimental and numerical simulation study, following results are obtained.

1) The damping force is proportional to the frequency (velocity).

2) The damping forces give good agreement with the theoretical values in low speed-increasing ratio (4:1), however, these are larger than theoretical values in the case of high speed-increasing ratio. These discrepancies will be caused by the efficiency of the speed-increasing ratio of gear and friction.

3) The peak values of the transmissibility are reduced with increasing mass ratio of \( \varepsilon = M_e / m \).

4) The resonance frequency of the main mass can be shifted toward the low frequency domain.

5) Suitable reduction of the response will be obtained by selecting an appropriate rotating moment of inertia, and by adjusting a damping coefficient.

REFERENCES


Fig.7 Input and response wave