Vibration Control Using an Impact Damper System*

— Comparison between Translating Impactor and Rotating Impactor, and Improvement of Rotating Impactor —

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Abstract

Impact damper systems are used to reduce the vibration of a lightly damped resonant primary system. The impact damper consists of an impactor and a container. The container is attached to the mass part of the primary system. The container mainly consists of two walls, and the impactor is placed inside the container. And since the impactor is free to move in the container, the impactor collides with the walls. When the impactor collides, some of the vibration energy of the primary system is momentarily converted to kinetic energy of the impactor. That is, impact damper system is damper which utilizes this energy conversion by impact phenomenon. We compared the damping characteristics in an impact damper system between a translating impactor and a rotating impactor and found that the rotating impactor is superior to the translating impactor. Then, with the aim of improving the damping characteristics of the rotating impactor, we proposed that an eccentric mass be added to the rotating impactor and confirmed that the damping effect improves because of the eccentric mass. Next, we calculated the mechanical energy of a vibration system having an impact damper and investigated the mechanism behind this improvement in damping characteristics due to the eccentric mass. In particular, we classified the impacts that occur between the impactor and wall into three types and examined the change in energy of the primary vibration system immediately after such impacts. In this way, we clarified the mechanism behind improved damping effect due to the addition of an eccentric mass.

Key words: Vibration Control, Damping, Impact Damper, Translating Impactor, Rotating Impactor, Mechanical Energy, Eccentric Mass

1. Introduction

One method of controlling structural vibration is to make appropriate use of impact phenomena. Typical of such a method is the impact damper, which is equipment for converting the vibrational energy of a vibration system to the kinetic energy of an impactor by having the impactor collide with a wall built into the vibration system.

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There have been many reports on research related to impact dampers, some going back a number of years. Bapat et al. studied a multi-unit impact damper having multiple damping elements each featuring a single impactor built into a container⁽¹⁾. They examined how the number of damping elements as well as friction characteristics between an impactor and container affected the mechanical characteristics of the impact damper. Nayeri et al., meanwhile, performed studies on multi-unit impact dampers that can actively adjust the clearance between wall and impactor⁽²⁾. Zinjade et al., in turn, proposed the application of impact dampers to equipment as a method of controlling friction-driven oscillations and demonstrated their effect⁽³⁾. Each of these three reports treated impactors as masses undergoing translational movement, and there have been many other reports dealing with such translating impactors⁽⁴⁾⁻⁽⁷⁾.

In addition to the above, Uno et al. performed a study that took the rotating movement of impactors into account. They described the effect of applying impact dampers to the vibration of lighting poles caused by road vibrations and wind⁽⁸⁾. Sato et al. experimentally determined damping characteristics when applying rotating impactors to impact dampers, and demonstrated the validity of calculations on the behavior of rotating impactors by capturing impactor movement using a high-speed camera⁽⁹⁾. In addition, the authors have studied damping characteristics when applying an impact damper having rotating impactors to a system that vibrates in multiple directions⁽¹⁰⁾.

In the above way, research to date has examined the damping characteristics of impact dampers separately for translating impactors and rotating impactors. Reports on comprehensive studies on both translating impactors and rotating impactors have yet to appear.

In this report, we calculate the damping characteristics of an impact damper for the case of a translating impactor and the case of a rotating impactor, and show that the use of a rotating impactor results in better damping characteristics than the use of a translating impactor. We also propose an enhanced rotating impactor with the aim of improving the damping characteristics of the rotating impactor. The enhanced rotating impactor is achieved by adding an eccentric mass to the rotating impactor with the idea of making it easier for impacts to occur between the impactor and wall. We show that the use of this enhanced rotating impactor improves damping characteristics and examine the mechanism of this improvement from the viewpoint of vibrational energy.

2. Equations of Motion for a Vibration System with an Impact Damper

2.1 Equations of Motion. The vibration model when using a rotating impactor is shown in Fig. 1. Here, the primary vibration system (primary system) has one degree of freedom and consists of spring constant K , damping coefficient C , and mass M . The impactor is a rotating impactor of mass m_i , moment of inertia I , and radius r . In the figure, x is input displacement, y is displacement of the primary system, z is displacement of the impactor, and θ is angular displacement. Subscripts h and v for variable z denote displacement in the horizontal and vertical directions, respectively, and Cle denotes the amount of clearance between the wall and impactor.

In this model, as explained later, the frictional force between the base surface of the primary system and the impactor is calculated from the amount of deformation in the impactor. It is for this reason why impactor movement in the vertical direction is taken into account.

The equation of motion of the primary system is given by Eq. (1), and the equations of motion of the impactor in the horizontal and vertical directions are given by Eqs. (2) and (3), respectively. The equation of motion describing the rotation of the impactor is given by Eq. (4).

Fig. 1 Impact damper

$$\begin{aligned} M\ddot{y} = & -K(y-x) - C(\dot{y}-\dot{x}) \\ & -P_{lw} + P_{rw} \\ & + \mu_b \operatorname{sgn}(\dot{z}_h - \dot{y} - r\dot{\theta})P_b \end{aligned} \quad (1)$$

$$\begin{aligned} m_i\ddot{z}_h = & P_{lw} - P_{rw} \\ & - \mu_b \operatorname{sgn}(\dot{z}_h - \dot{y} - r\dot{\theta})P_b \end{aligned} \quad (2)$$

$$\begin{aligned} m_i\ddot{z}_v = & -\mu_{lw} \operatorname{sgn}(\dot{z}_v + r\dot{\theta})P_{lw} \\ & - \mu_{rw} \operatorname{sgn}(\dot{z}_v - r\dot{\theta})P_{rw} \\ & + P_b \\ & - m_i g \end{aligned} \quad (3)$$

$$\begin{aligned} I\ddot{\theta} = & \left[-\mu_{lw} \operatorname{sgn}(\dot{z}_v + r\dot{\theta})P_{lw} \right. \\ & + \mu_{rw} \operatorname{sgn}(\dot{z}_v - r\dot{\theta})P_{rw} \\ & \left. + \mu_b \operatorname{sgn}(\dot{z}_h - \dot{y} - r\dot{\theta})P_b \right] r \end{aligned} \quad (4)$$

As for the subscripts in these equations, *lw* denotes the left wall, *rw* the right wall, and *b* the base surface, while μ indicates coefficient of friction and *g* the acceleration of gravity. In addition, *sgn* is the slipping judgment constant; it is assigned a value of 1 if the expression within the parentheses is positive, -1 if negative, and 0 if 0. The symbol *P* represents the contact force between the impactor and a wall or between the impactor and the base as given by the following equations:

$$\begin{aligned} P_{lw} = & N_{lw} \left\{ k_{nh} (-z_h + y)^{3/2} \right. \\ & \left. + c_{nh} (-\dot{z}_h + \dot{y}) \right\} \end{aligned} \quad (5)$$

$$\begin{aligned} P_{rw} = & N_{rw} \left\{ k_{nh} \{z_h - (y + Cle)\}^{3/2} \right. \\ & \left. + c_{nh} (\dot{z}_h - \dot{y}) \right\} \end{aligned} \quad (6)$$

$$\begin{aligned} P_b = & N_b \left\{ k_{nv} (-z_v)^{3/2} \right. \\ & \left. + c_{nv} (-\dot{z}_v) \right\} \end{aligned} \quad (7)$$

Here, *N* is the contact judgment constant; it is assigned a value of 1 if contact is being made between the impactor and a wall or between the impactor and the base and a value of 0 otherwise.

The equation of motion related to rotation is unnecessary when the impactor is of the translating type. Thus, when using a translating impactor, only Eqs. (1) to (3) without θ term are used as equations of motion.

2.2 Contact Force. We used Hertzian contact theory to compute the contact force P at the time of impact between the impactor and wall or between the impactor and the base of the primary system. The impact analysis model is shown in Fig. 2. Contact force is calculated in terms of a spring (spring constant k_n) and damper (damping coefficient c_n)⁽¹¹⁾. In Eqs. (5) to (7), subscripts h and v denote the horizontal and vertical directions and constants k_n (k_{nh} , k_{nv}) are given by Eq. (8). In that equation, E denotes Young's modulus and σ Poisson's ratio, and subscripts i and w denote the impactor and wall of the primary system, respectively (b in the case of the base surface). Damping coefficient c_n (c_{nh} , c_{nv}) is given by Eq. (9), where δ represents the amount of impactor deformation and e the coefficient of restitution between the impactor and primary system.

Fig. 2 Impact analysis model

$$k_n = \frac{4\sqrt{r}}{3} \left(\frac{1-\sigma_i^2}{E_i} + \frac{1-\sigma_w^2}{E_w} \right)^{-1} \quad (8)$$

$$c_n = 2\alpha \sqrt{\beta \times m_i \times k_n} |\dot{\delta}|^{1/4} \quad (9)$$

$$\alpha = \frac{1}{2\pi} \ln \frac{1}{e^2}$$

$$\beta = 1.235$$

3. Comparison between Translating Impactor and Rotating Impactor

We previously computed the equations of motion described above and found that simulation results agreed well with experimental results^{(10), (12)}. In this report, we perform calculations to compare the damping characteristics achieved by a translating impactor with those by a rotating impactor

3.1 Calculation Conditions. We continue trying to reduce the vibration of the pantograph support system by applying an impact damper. The lowest natural frequency of the pantograph support system is 2 Hz⁽¹³⁾. On this account we made the simple cantilevered experimental equipment⁽¹⁰⁾ which modeled the pantograph support system.

The values listed in Table 1 are tailored to the experimental equipment used in our research. Based on the model's natural frequency of 2.03 Hz, the excitation frequency was varied from 1.8 to 2.1 Hz with a sweep time of 400 s. Clearance in the horizontal direction was arbitrarily set and input acceleration was set to 0.06 m/s². The coefficients of friction μ_{rw} and μ_{lw} were fixed at 0.2 and μ_b was varied from 0.2 to 1. In addition, it is confirmed that this frequency-sweep condition is enough to evaluate the vibration amplitude of primary system experimentally.

We numerically solved the above equations of motion using the *Runge-Kutta* method and determined the time-history response of the primary system and impactor.

Table 1 Conditions of calculation

Mass of primary system M	6.76 kg	Moment of inertia I	$3.31 \times 10^{-4} \text{ kg}\cdot\text{m}^2$
Mass of impactor m_i	0.55 kg	Radius r	35 mm
Spring constant K	1103 N/m	Young's modulus E_w, E_i	205 Gpa
Damping coefficient C	0.22 N·s/m	Poisson's ratio σ_w, σ_i	0.3
Natural frequency	2.03 Hz	Coefficient of restitution e	0.8
Excitation frequency	1.8 ~ 2.1 Hz	Coefficient of friction μ_{hw}, μ_{rv}	0.2
Sweep time	400 s	Coefficient of friction μ_b	0.2, 0.6, 1
Foundation's acceleration	0.06 m/s ²		

3.2 Results of Displacement Calculations. The displacement of the primary system when using a translating impactor is shown in Fig. 3. Here, the results given in Figs. 3 (a) and (b) correspond to a coefficient of friction μ_b of 0.2 and 1, respectively. The horizontal axis represents excitation frequency and the vertical axis the displacement. Clearance was set to 14 mm.

These results show that the maximum value of displacement increases as the coefficient of friction μ_b becomes larger. As described in more detail below (section 3.3), a larger coefficient of friction prevents both movements and collisions of the translating impactor.

Displacement results when using a rotating impactor are shown in Fig. 4. In this case, peak displacement of the primary system hardly changes when varying the coefficient of friction.

Fig. 3 Displacement of primary system with translating impactor (Clearance 14 mm)

Fig. 4 Displacement of primary system with rotating impactor (Clearance 14 mm)

The effects of coefficient of friction μ_b on peak displacement are summarized in Fig. 5. The horizontal axis represents clearance and the vertical axis the peak displacement of the primary system. The results for the translating impactor and rotating impactor are shown in Figs. 5(a) and (b), respectively.

We first examine the results for the translating impactor. For any coefficient of friction μ_b , peak displacement initially decreases as clearance increases, and after reaching a minimum value remains constant. In this report, the clearance corresponding to the minimum value of peak displacement is called the optimum clearance. In these results, optimum clearance for coefficients of friction 0.2, 0.6, and 1 is 11 mm, 7 mm, and 2 mm, respectively. It can be seen that the magnitude of displacement at optimum clearance decreases as the coefficient of friction decreases. Thus, to achieve a favorable damping effect with a translating impactor, coefficient of friction μ_b between the impactor and base surface must be made small. As a practical measure, a rolling member can be installed between the impactor and base surface⁽¹³⁾.

Turning to the results for the rotating impactor shown in Fig. 5(b), it can be seen that the magnitude of displacement at optimum clearance hardly changes for different values of the coefficient of friction. Thus, in contrast to a translating impactor, there is no need to minimize friction between the impactor and base surface when using a rotating impactor. Furthermore, for the coefficient of friction of 0.2 and greater, the optimum clearance hardly changes. This means that optimum clearance will not be affected even if the coefficient of friction between the impactor and base surface is somewhat inaccurate, which is a beneficial feature in actual design work.

Fig. 5 Peak values of displacement in primary system

3.3 Number of Impacts. The damping effect of an impact damper is obtained from the impacts that occur between the impactor and the walls of the primary system. As described later, there are “good impacts” and “bad impacts,” but the “number of impacts” is thought to be a good index for assessing the damping effect. We therefore calculated the number of impacts when using a translating impactor and rotating impactor. Results are shown in Fig. 6 for different coefficients of friction μ_b . The shaded bars represent the results for the translating impactor and the grey bars the results for the rotating impactor. These results were obtained when setting the clearance value to the optimum clearance in question.

For the translating impactor, these results show that the number of impacts decreases as the coefficient of friction increases. This tells us that the impactor is not impacting the wall effectively even when using optimum clearance, which means that a sufficient damping effect cannot be obtained.

In the case of the rotating impactor, however, the coefficient of friction has no significant effect on the number of impacts. This behavior can be readily understood if we consider the rotating movement of the impactor.

Fig. 6 Number of impacts

To summarize, an impact damper using a rotating impactor is insensitive to the friction between the impactor and base surface. As a result, there is no need to be attentive to the friction between the impactor and base surface when configuring the impact damper. This is a key point when setting out to design an impact damper. A rotating impactor is superior by virtue of its ability to obtain good damping effects regardless of the coefficient of friction.

4. Exchange of Energy between Primary System and Impactor on Impact and Associated Problem

The damping effect of an impact damper is governed by the exchange of energy between the primary system (wall) and impactor at the time of impact. Accordingly, examining the impact-damper damping characteristics that we have discussed up to now from the viewpoint of energy transfer at the time of impact is important for understanding the physical meaning of these characteristics. In the following discussions, we investigate damping only for the rotating impactor, which we have concluded to have good damping characteristics.

4.1 Types of Impacts and Energy Exchange. Impacts between an impactor and wall can be broadly divided into three types as summarized in Table 2. For each case, we consider the kind of energy exchange that takes place between the primary system and impactor.

Impact *A* corresponds to a head-on collision between the impactor and wall. Since the mass of the primary system is generally larger than that of the impactor, the energy of the impactor will, for the most part, increase because of the impact. Conversely, the energy of the primary system decreases.

Impact *B* corresponds to a rear-end collision by the wall on the impactor. In this case, the velocity of the primary system (wall), which is initially greater than the velocity of the impactor, decreases on impact. That is to say, the energy of the primary system decreases on impact.

Impact *C*, on the hand, corresponds to a rear-end collision by the impactor on the wall. Here, the relationship between the primary system and impactor in impact *B* is reversed, and as a result, the energy of the primary system increases on impact.

In summary, impacts *A* and *B* are good impacts in terms of damping effect while impact *C* is a bad impact. For an impact-damper vibration system using a rotating impactor, we performed calculations to determine what kind of impacts are taking place and how the mechanical energy of the primary system changes as a result of those impacts. Results of these calculations are shown in Fig. 7. The horizontal axis represents the velocity of the primary system and the vertical axis the velocity of the impactor. The size (diameter) of the circles in the figure indicates the amount of change in the energy of the primary system E_c as given by the following equation:

$$E_c = \left[\frac{1}{2} M \dot{y}_2 + \frac{1}{2} K (y_2 - x_2)^2 \right] - \left[\frac{1}{2} M \dot{y}_1 + \frac{1}{2} K (y_1 - x_1)^2 \right] \quad (10)$$

As for the subscripts in this equation, 2 denotes the just after impact, 1 denotes the just before impact. In the calculations, clearance and coefficient of friction μ_b were set to 7 mm and 0.2, respectively, and the results of all impacts occurring in a frequency sweep were plotted.

Table 2 Types of impacts

Type	Velocity	Damping effect	Symbol
A	$V_w \cdot V_i < 0$	Good	$\square$
B	$V_w \cdot V_i \geq 0$ $ V_w \geq V_i $	Good	$\diamond$
C	$V_w \cdot V_i \geq 0$ $ V_w < V_i $	Bad	$\times$

Fig. 7 Primary system's changing value of energy

In quadrants 2 and 4, the velocities of the primary system and impactor are reversed ($V_w \cdot V_i < 0$), which means that impact A occurs in these quadrants. Here, we can see that the circles representing change in energy of the primary system are colored “white” indicating that the energy of the primary system decreases (becomes negative). In other words, the energy of the primary system decreases on the occurrence of impact A.

Next, we look at quadrants 1 and 3. In these quadrants, the primary system and impactor are moving in the same direction and the velocity of the primary system is either larger or smaller than that of the impactor. The 45-degree supplementary line in the figure acts as a border line separating the region in which the velocity of the primary system is greater from the region in which the opposite is true.

The region enclosed by the supplementary line and the X-axis is the region in which the velocity of the primary system is greater than that of the impactor. This is the region corresponding to impact B, the rear-end collision by the primary system on the impactor. In this region, we can see that the circles are colored white indicating that the energy of the primary system decreases.

Now, the region enclosed by the supplementary line and the Y-axis is the region corresponding to a rear-end collision by the impactor on the primary system, that is, impact C. In this region, the circles are colored grey indicating that the energy of the primary system increases.

The above underscores the description at the beginning of this section that impacts A and B are “good impacts” while impact C is a “bad impact.” It also shows how decrease or increase in the energy of the primary system can be determined according to the type of impact (impact A, impact B, or impact C).

4.2 Problem with Rotating Impactor. We here examine the time-history response of an impact-damper vibration system using a rotating impactor from the viewpoint of impact types and energy exchange as discussed above.

Energy of the primary system for a coefficient of friction μ_b of 0.2 and clearance of 16 mm is shown in Fig. 8. Here, the 16-mm clearance is an optimum clearance as can be seen from Fig. 5(b). This result (Fig. 8) shows that the energy of the primary system alternately increases and decreases with time and becomes maximum at about 170 s.

A blow-up of these results at about 170 s is shown in Fig. 9(a) and results for a

clearance of 20 mm, which is a value slightly larger than the optimum clearance, are shown in Fig. 9(b). The energy of the impactor is also shown here, and the occurrence of impact types *A*, *B*, and *C* are superposed on the impactor-energy results using square, diamond, and cross symbols. That is to say, the occurrence of “good impacts *A/B*” and “bad impact *C*” are also noted in the figure.

We first examine the results for the 16-mm optimum clearance in Fig. 9(a). Up to about 170 s, the energy of the primary system decreases progressively due to good *A*-type impacts. This decrease in the energy of the primary system up to 170 s is accompanied by a decrease in the energy flowing into the impactor. However, decrease in the energy of the primary system and impactor makes for little vibration in both the primary system and impactor resulting in no impacts. This phenomenon continues up to 173 s, and during this no-impact period from about 170 s to 173 s (about 3 s), the energy of the primary system increases as none of its energy is flowing into the impactor. At the same time, the vibration of the primary system gradually becomes larger, and *A*-type impacts start to occur again from 173 s on. In other words, the energy of the primary system reverts back to a decreasing pattern.

Fig. 8 Energy of primary system (Clearance 16 mm, $\mu_b = 0.2$)

From the same point of view, we now examine the results in Fig. 9(b) for clearance slightly increased to 20 mm. Here, the no-impact time period, that is, the time period in which the energy of the primary system increases, lasts for about 5 s from about 238 s to 243 s. Thus, as clearance becomes larger than the optimum clearance, the period in which no impacts occur lengthens and the energy of the primary system increases by that much. In addition, the energy of the primary system becomes maximum in a time slot different than that of the 16-mm optimum clearance, which is why the time slot shown in Fig. 9(b) differs from that of Fig. 9(a).

Thus, even when using a rotating impactor having the feature of “rotation,” or in other words, “easy movement,” impacts between the primary system and impactor cease to occur as the energy/vibration of the primary system becomes small. This lack of impacts causes vibration in the primary system to once again increase. It would be beneficial if the rotating impactor could be improved to prevent this increase in primary system vibration.

Fig. 9 Energy of primary system and impactor ($\mu_b = 0.2$)

5. Improvement of Damping Characteristics by a Rotating Impactor with an Eccentric Mass

In the previous section, we described a problem that occurs with a rotating impactor. This problem can be summarized by saying that no impacts will occur between the primary system and impactor if the rotating impactor does not move well. In other words, this problem can be solved if the rotating impactor can be given a structure that makes movement easier. With this in mind, we wondered whether giving the rotating impactor an unbalanced structure, i.e., adding an eccentric mass to it, just might make it easier for the impactor to move about.

5.1 Range of Clearance for Improving Characteristics. As described above, the damping characteristics of an impact damper are governed by the amount of clearance. Knowing this, we considered the benefits of improving the rotating impactor within a range of clearance for which favorable damping characteristics can be obtained.

In this regard, we looked again at the results of Fig. 5(b). In the previous section, we investigated damping characteristics for a rotating impactor with coefficient of friction μ_b set to 0.2. The results in Fig. 5(b) for $\mu_b = 0.2$ indicate that optimum clearance is 16 mm. This is why clearance is commonly set to 16 mm to obtain the best damping characteristics. However, considering the variation and uncertainty in mechanical and vibration characteristics, it would be natural for an engineer to select an amount of clearance having some margin instead of narrowing it down to one point at an optimum value of 16 mm. In this case, the appropriate selection would be a region in which the magnitude of vibration hardly changes with respect to change in clearance, that is, the range of clearance for which damping characteristics are insensitive to fluctuation in clearance. This range, as shown by the results of Fig. 5(b), corresponds to the region in which clearance is somewhat larger than the optimum clearance.

Thus, for the region that includes the optimum clearance plus slightly larger values of clearance, we investigated the best approach to take for achieving a rotating impactor with enhanced characteristics.

5.2 Vibration Model with a Rotating Impactor Adding an Eccentric Mass. We consider a rotating impactor to which an eccentric mass has been added. The model of an impact-damper vibration system with this type of impactor is shown in Fig. 10. Here, eccentric mass m_e is attached to the impactor at its periphery, and mass m_i of the impactor is total mass including eccentric mass m_e .

The equations of motion of this vibration system consist of Eqs. (1) to (3) in section 2 plus Eq. (11) below. Equation (11) is achieved by adding the moment corresponding to the eccentric mass to Eq. (4) in section 2.

Fig. 10 The vibration model with a rotating impactor which added an eccentric mass

$$I\ddot{\theta} = \left[-\mu_{lw} \operatorname{sgn}(\dot{z}_v + r\dot{\theta})P_{lw} + \mu_{rw} \operatorname{sgn}(\dot{z}_v - r\dot{\theta})P_{rw} + \mu_b \operatorname{sgn}(\dot{z}_h - \dot{y} - r\dot{\theta})P_b - m_e g \sin(\theta) \right] r \quad (11)$$

5.3 Results of Displacement Calculations. We calculated the time-history response of this vibration system while varying the eccentric mass from 0 kg to 0.3 kg. In these calculations, the total mass of the impactor and moment of inertia remained unchanged despite the change in eccentric mass. Coefficient of friction μ_b was set to 0.2.

Peak displacement of the primary system at optimum clearance for a range of eccentric-mass values is shown in Fig. 11. The horizontal axis represents eccentric mass, and the 1st and 2nd vertical axes represent peak displacement and optimum clearance, respectively. As shown, displacement decreases as eccentric mass increases resulting in a maximum damping effect from 0.13 to 0.26 kg. As the eccentric mass becomes larger, however, the vibration of the primary system starts to increase. This indicates that an appropriate value exists for the eccentric mass. Optimum clearance likewise increases as the eccentric mass becomes larger. The above results demonstrate the effectiveness of the eccentric mass at optimum clearance.

We also studied the magnitude of vibration of the primary system at clearances larger than the optimum clearance. Results are shown in Fig. 12 for an eccentric mass of 0.13 kg that was previous shown to produce good characteristics at optimum clearance. These results are shown together with those for no eccentric mass (0 kg) for comparison purposes. The horizontal axis represents clearance and the vertical axis peak displacement of the primary system. Optimum clearance was 16 mm for both sets of results. It can be seen from these results that vibration of the primary system decreases by about half by adding this 0.13-kg eccentric mass. It can also be seen that adding an eccentric mass can reduce vibration for even clearance values larger than the optimum clearance, that is, for clearances up to 30 mm.

Fig. 11 Displacement of primary system and optimum clearance

Fig. 12 Displacement of primary system using rotating impactor with eccentric mass

We can therefore say that a rotating impactor with good characteristics can be realized by adding an eccentric mass within a region of clearance values that includes the optimum clearance plus larger clearance values.

5.4 Mechanism behind Improvement in Damping Characteristics by Adding an Eccentric Mass. We here consider the physical reasons as to why damping effect improves by adding an eccentric mass to the rotating impactor.

The energy of the primary system and impactor versus time for an impactor with an eccentric mass of 0.13 kg is shown in Fig. 13. The clearance here is an optimum clearance of 16 mm. From these results, we see that no impacts occur near the 168-s and 174-s time points and that these periods of no impacts are short, each lasting about one second. But we have already seen from Fig. 9(a) for the case of no eccentric mass (0 kg) that the no-impact period lasted about 3 seconds. Thus, adding an eccentric mass has the effect of shortening the period of no impacts thereby suppressing increase in the energy of the primary system.

To determine why the addition of an eccentric mass results in shortening of the no-impact period, we focused our attention on the angular displacement of the impactor. The results of calculating the angular displacement of the impactor is shown in Fig. 14. Angular displacement for no eccentric mass is shown in Fig. 14(a) and that for an eccentric mass of 0.13 kg is shown in Fig. 14(b). The horizontal axis represents the same time slot as that in Fig. 9(a) and Fig. 13.

For an eccentric mass of 0 kg, the occurrence of a C-type impact (cross symbol in the figure) near 170 s is followed by angular displacement with small amplitude for five cycles. But from 173 s on, angular displacement again falls into a nearly steady state. Thus, for the 3-second period from 170 to 173 s, no impacts with the primary system occur resulting in a time period corresponding to an increase in the vibration of the primary system.

Next, for an eccentric mass of 0.13 kg, periods of no impact occur near 168 s and 174 s. These periods are short and are immediately followed by a return to steady-state behavior. This is because the impactor tends to behave like an inverted pendulum due to the addition of the eccentric mass.

Fig. 13 Energy of primary system and impactor (Eccentric mass 0.13 kg, clearance 16 mm)

(a) Eccentric mass 0 kg, clearance 16 mm

(b) Eccentric mass 0.13 kg, clearance 16 mm

Fig. 14 Angular displacement of impactor

In short, the reason why addition of an eccentric mass can suppress increase in the energy of the primary system is that the eccentric mass induces rotational movement in the impactor thereby promoting impacts between the impactor and primary system.

We next investigated the reason why the addition of an eccentric mass of 0.27 kg or larger has the effect of reducing the damping effect. The displacement of the primary system and impactor for an eccentric mass of 0.3 kg is shown in Fig. 15. Here, clearance was set to the value of optimum clearance (38 mm). The circles in the figure denote the occurrence of any type of impact without distinguishing between *A*, *B*, and *C* types of impacts. It can be seen from these results that impacts only occur at the initial phase and the final phase and that no impacts occur in the time period from about 50 s to 300 s. As shown in Fig. 11, the value of optimum clearance increases with increase in eccentric mass. In the case of an eccentric mass of 0.3 kg, optimum clearance turns out to be 38 mm and a phenomenon occurs in which the impactor is unable to move from one wall to the other, which means no impacts. This is why the addition of an eccentric mass of 0.27 kg or larger makes impacts difficult to occur and reduces the damping effect.

Fig. 15 Displacement of vibration system and impactor (Eccentric mass 0.3 kg, clearance 38 mm)

5.5 Influence of Initial Conditions. The calculation results presented so far are for the case in which the displacement, velocity, and acceleration of both the primary system and impactor are all set to 0 with the impactor in contact with the left wall as initial conditions. We decided to investigate how a change in initial conditions might influence damping characteristics.

We set up 10 sets of initial conditions as listed in Table 3 and performed calculations for no eccentric mass (0 kg) and an eccentric mass of 0.13 kg with coefficient of friction μ_b set to 0.2. When condition 1 of Table 3 is used for the calculation, the primary system and the impactor show stable motion after a sweep start for 50 seconds, regardless of having eccentric mass or not. We set the value of velocity and angular velocity at this time and the double values of those as the initial conditions.

Table 3 Initial conditions

Condition number	1	2	3	4	5	6	7	8	9	10
<i>Primary system</i>										
Displacement mm	0	0	0	5	10	5	10	10	0	5
Velocity mm/s	0	0	0	50	100	50	100	100	0	50
<i>Impactor</i>										
Displacement mm	0	clearance/2	0	5	10	5	10	10+clearance	clearance/2	0
Velocity mm/s	0	0	50	50	100	50	100	100	0	0
Angular displacement rad	0	0	0	0	0	0	0	0	$\pi/2$	π
Angular velocity rad/s	0	0	0	0	0	5	10	0	0	0

Calculation results are shown in Fig. 16. Results for no eccentric mass are shown in Fig. 16(a) and those for an eccentric mass of 0.13 kg are shown in Fig. 16(b). For each clearance value in the range of clearance values shown, we applied the ten different sets of initial conditions and computed the time-history response of the primary system under each set of initial conditions. Then, from the peak displacements (10 values) obtained for each clearance value under the 10 sets of initial conditions, we determined their mean value and standard deviation. The white and black circles in the figure indicate mean values and the vertical bars the standard deviations.

We compare the results for eccentric masses of 0 and 0.13 kg in Figs. 16 (a) and (b). For an eccentric mass of 0 kg, peak displacement (mean value) is less than 10 mm only within a limited region near a clearance of 15 mm. For an eccentric mass of 0.13 kg, however, peak displacement is less than 10 mm within a broad region of clearances from 16 to 27 mm.

These results demonstrate that the conclusion arrived at for the special set of initial conditions described above (set '1' in Table 3) is generally valid for other initial conditions too. That is, there is no change—even when varying initial conditions—to the conclusion that adding an eccentric mass produces good damping characteristics within a region of clearances that includes the optimum clearance and slightly larger clearance values.

Fig. 16 Influence of initial conditions

At the same time, it can be seen from these results that the magnitude of peak displacement varies according to initial conditions. What this tells us is that determining the optimum clearance for a particular set of initial conditions and configuring an impact damper for that clearance does not guarantee the most favorable damping characteristics. For example, optimum clearance for an eccentric mass of 0.13 kg under the special set of initial conditions described earlier (set '1' in Table 3) was found to be 16 mm (see Fig. 11). However, if we examine the calculation results in Fig. 16(b) for multiple sets of initial conditions, we see that clearance values of 18 and 19 mm are more appropriate than the optimum clearance of 16 mm. Thus, when setting out to select a specific amount of clearance, it would be best to calculate response under multiple sets of initial conditions and to then select a clearance value giving due consideration to how that response changes with change in initial conditions.

6. Conclusions

In this paper, we investigated the damping characteristics of an impact damper system when using a translating impactor and rotating impactor, and proposed an enhanced rotating impactor with the aim of improving the damping characteristics of rotating impactors. We also examined the damping characteristics of an impact damper with a rotating impactor from the viewpoint of vibration energy and clarified the physical meaning of an eccentric mass added to a rotating impactor for improving damping characteristics. These studies

produced the following findings.

(1) Damping effects when using a translating impactor are heavily dependent on the friction between the impactor and base surface of the primary system. The coefficient of friction here must be made as small as possible to obtain favorable damping effects with a translating impactor.

(2) The damping characteristics of an impact damper using a rotating impactor are insensitive to the friction between the impactor and base surface negating the need to give special attention to this friction in impact-damper design. In other words, a rotating impactor, which can provide a good damping effect regardless of the magnitude of the coefficient of friction, is superior to a translating impactor.

(3) Even if using a rotating impactor, there will be a period in which no impacts occur between the primary system and impactor once vibration in the primary system becomes small. During this period, the energy of the primary system (magnitude of vibration) will once again increase presenting a problem.

(4) Adding an eccentric mass to the rotating impactor promotes rotational movement in the rotating impactor facilitating impacts with the primary system and producing a favorable damping effect. The addition of an eccentric mass can therefore solve the problem with the rotating impactor described in (3) above.

A structure vibrates in the different direction at the same time by random excitation force etc. We want to apply the impact damper system for such vibration problem.

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