Development of a Dynamic Damper Using Fluids and Rubber*

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Abstract

Dynamic dampers are usually made of rubber and rigid mass elements. They are widely used for cutting the resonance of mechanical structures. In this research, modification was made to a traditional damper by introducing small chambers, channels (orifices) and fluid into the rubber. By using the resonance of the fluid, a two-degree-of-freedom damper was developed with a slight weight increase. Its characteristics and vibration reduction effects were formulated and investigated experimentally. It was demonstrated that the damper works in a broad frequency range as compared to traditional dampers and can cut two resonance peaks simultaneously.

Key words: Damping, Damper, Vibration Control Device, Dynamic Absorber, Fluid, Rubber

1. Introduction

Dynamic dampers are widely used for reducing the vibration of various structures because of their advantages such as a simple structure and low cost. They exhibit a large damping effect at and around the tuning frequency, but less effect at other frequencies. Meanwhile, it is required that machines and automobiles have a reduced weight and a variable operation range to meet the continuous demand for energy conservation; to this end, the number of cases in which multiple-mode resonances are simultaneously excited and the natural frequency of the main system (the target of damping) is varied with the operating conditions has been increased. Therefore, research on dynamic dampers frequently focuses on expanding the frequency range of damping. Researches on active control are also being intensively carried out⁽¹⁾⁽²⁾⁽³⁾, and there have been reports on cases in which the natural frequency of a dynamic damper is changed electromagnetically to follow the resonance of the main system, or the damping coefficient of a damper is controlled by using magnetorheological fluids. However, the applications of active control are limited to special cases because it incurs a high cost and requires an external power source. On the other hand, it is possible to expand the damping frequency range passively by increasing the damping coefficient of a damper, like a Houde damper. However, this results in a decrease in the damping effect at certain frequencies, and in addition, a high weight ratio, i.e., a heavy damper, is generally required.

In general, rubber dynamic dampers that are frequently used in automobiles consist of rubber and rigid mass elements. This kind of dampers forms a damped vibration system with one DOF (Degree Of Freedom). The rubber serves as both a spring element and a damping element. In this study, we modified a conventional dynamic damper as follows. Three chambers were prepared in the rubber section and connected via narrow fluid channels (orifices). A small amount of fluid (liquid) was introduced into the chambers. Using the resonance of the fluid, we realized a 2-DOF dynamic damper with only a slight increase in weight. The weight of the introduced fluid was approximately 12% of the total weight of the damper (the sum of the weights of the rigid mass, rubber, and fluid). We examined the vibration characteristics of the developed damper and carried out experiments to demonstrate its damping effect. It was confirmed that the damping frequency range of the new dynamic damper was wider than that of a conventional one with the same weight. In the reduction of vibration, viscous fluids may be used to enhance damping, and the

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hydraulic engine mounts that are widely employed in automobiles use the resonance of fluids.⁽⁴⁾⁽⁵⁾ However, there have been few dynamic dampers that use the resonance of both fluids and structures.

Fig. 1 Schematic of the proposed 2-DOF dynamic damper

Fig. 2 Three-dimensional view of the fluid channels and chambers

2. Structure and natural frequencies of the proposed damper

2.1 Structure

Figure 1 shows a schematic of the structure of the proposed dynamic damper. This damper is designed so that it can be installed in narrow spaces. It has a planar rigid mass. Three chambers are arranged in the rubber section, and adjacent chambers are connected via fluid channels. In addition, the rubber membrane of each chamber is equipped with a pedestal used to mount the damper onto the main system (the target of damping). If there is a relative displacement between the rigid mass and the mounting pedestals in the z direction, the pressure in the chambers changes. The fluid then flows in the channels because the rate of change in pressure differs between chambers 1 (central chamber) and 2 (side chambers). Figure 2 shows a three-dimensional schematic of the fluid channels and chambers. In Fig. 2, each of the side chambers is connected to the central chamber via a single fluid channel. Even when multiple channels are symmetrically placed to connect the side chambers to the central chamber, the principle and calculation method are similar to those in the case of a single channel; therefore, the case with a single channel is discussed below.

In the structure shown in Fig. 1, chamber 1 is also equipped with a pedestal to increase the stability of mounting. Chamber 1 has a greater volume and a smaller area for the mounting pedestal than those of chamber 2. It is possible to design the damper without a mounting pedestal in chamber 1, with the aim of increasing the flow rate of the fluid; however, this is a special case of the structure shown in Fig. 1 and is not discussed here.

Since we focus on the vibration with small amplitude, the above description of the fluid flows should be more accurately explained as that the fluid in the channels vibrates (oscillates) as a single body in the x direction. Let us qualitatively explain the vibration reduction mechanism on the basis of the resonance of the fluid as follows. The fluid in the channels works as mass elements (inertial term), and the bulk expansions of the chambers

serve as restoring springs, thus forming a vibration system. This vibration system is similar to the air vibration of resonant silencers.⁽⁶⁾⁽⁷⁾ The vibration of the fluid induces pressure fluctuations in the chambers, which act on the main system (the target of damping) via the mounting pedestals. In this case, the rigid mass serves as a weight that allows the pressure fluctuations in the chambers to be transmitted to the main system. A conventional rubber dynamic damper forms a 1-DOF system, whereas the proposed one shown in Fig. 1 forms a 2-DOF system based on the resonance modes of both the structure and the fluid.

2.2 Natural frequencies

The equation of motion is derived using the Lagrangian equation. A numerical analysis appears to be effective for determining the behavior of the fluid because the system has a complicated shape; however, very simple analysis is generally preferred in the design phase. In this study, we use an approximation model for discussion. Assuming the weight of the rubber to be negligible and the rigid mass to vibrate in the z direction, the kinetic energy of the system is given by

$$T = \frac{1}{2}m\dot{w}^2 + \frac{1}{2}\rho \iiint_{\text{Fluid}} (\dot{v}_x^2 + \dot{v}_y^2 + \dot{v}_z^2) dx dy dz \quad (1)$$

Here, T is the kinetic energy, m is the weight of the rigid mass, w is the displacement of the rigid mass in the z direction, ρ is the density of the fluid, and v_x , v_y , and v_z are the velocities of the fluid in the x , y , and z directions, respectively. The superscript dot ($\dot{}$) denotes derivatives with respect to time. In the narrow fluid channels connecting the chambers, the velocity of the fluid is much greater than in other sites. Because of this, the kinetic energy of the fluid is considered to concentrate in the fluid channels.⁽⁴⁾⁻⁽⁷⁾ Therefore, Eq. (1) can be approximated as follows.

$$T = \frac{1}{2}m\dot{w}^2 + \rho(1 + \alpha)L \int_0^R \int_0^{2\pi} v_x^2 r dr d\theta \quad (2)$$

Here, the compressibility of the fluid is ignored, and the fluid is assumed to flow (vibrate) only in the longitudinal direction (x direction) in the fluid channels. The two fluid channels are symmetrically placed (see Fig. 1) and have a circular cross section with a radius of R and a length of L . For each fluid channel, the symbols r and θ represent the radius and angle, respectively, on a local cylindrical coordinate system with the origin at the center of the channel. In Eq. (2), it is assumed that v_x is not a function of x . This assumption holds inside the fluid channels but generally does not hold at the ends of the fluid channels. Moreover, the fluid in the chambers (outside the fluid channels) has, albeit relatively low, kinetic energy. Therefore, we need a parameter α to correct the kinetic energy expressed by Eq. (2).⁽⁶⁾⁽⁷⁾

Because the fluid in contact with the rubber wall does not move in the fluid channels, the velocity distribution on the cross section of a fluid channel is assumed to be given by Eq. (3) by referring to parallel flow inside a tube.

$$v_x = V_x \left(1 - (r/R)^2\right) \quad (3)$$

Here, V_x is the velocity of the fluid at the center of a fluid channel. Then, the following relationship holds between V_x and the average velocity in a fluid channel $\bar{v}_x$.

$$\bar{v}_x = \frac{1}{\pi R^2} \int_0^R \int_0^{2\pi} v_x r dr d\theta = \frac{1}{2} V_x \quad (4)$$

By substituting Eqs. (3) and (4) into Eq. (2), we obtain

$$T = \frac{1}{2}m\dot{w}^2 + \frac{4}{3}(1 + \alpha)\rho L s \bar{v}_x^2 \quad (5)$$

Here, s is the cross-sectional area of a fluid channel.

The motion of the fluid in the fluid channels causes a change in the volumes of the chambers. Since this study focuses on small vibrations, the change in volume is much smaller than the original volume of the chambers. Assuming the bulk spring constant of each chamber to be unchanged even when the volume experiences a small change, the potential energy of the system is expressed by

$$U = \frac{1}{2} K_1 (A_1 w - 2s\bar{u}_x)^2 + 2 \left\{ \frac{1}{2} K_2 (A_2 w + s\bar{u}_x)^2 \right\} \quad (6)$$

$$= \frac{1}{2} K_1 (A_1 w - 2s\bar{u}_x)^2 + K_2 (A_2 w + s\bar{u}_x)^2$$

Here, U is the potential energy, and the bulk spring constants K_1 and K_2 represent the pressures required to change the volumes of chambers 1 and 2 by one unit, respectively, under the conditions that the fluid channels between the chambers are completely blocked and the rigid mass and the mounting pedestals are fixed. The unit of K_1 and K_2 is N/m^5 . A_1 and A_2 are the equivalent areas of the mounting pedestals used to calculate the changes in the volumes of chambers 1 and 2, respectively, with respect to the relative displacement in the z direction between the rigid mass and the mounting pedestals. $\bar{u}_x$ is the average particle displacement of the fluid in the x direction in the fluid channels and has the following relationship with $\bar{v}_x$.

$$\bar{v}_x = \dot{\bar{u}}_x \quad (7)$$

From Eqs. (5)-(7), the Lagrangian function is given by

$$T - U = \frac{1}{2} m \dot{w}^2 + \frac{4}{3} (1 + \alpha) \rho L s \dot{\bar{u}}_x^2 - \frac{1}{2} k_b w^2 + 2k_d \bar{u}_x w - k_t \bar{u}_x^2 \quad (8)$$

Here, parameters k_b , k_t , and k_d are given as follows.

$$k_b = K_1 A_1^2 + 2K_2 A_2^2 \quad (9)$$

$$k_t = (2K_1 + K_2) s^2 \quad (10)$$

$$k_d = (K_1 A_1 - K_2 A_2) s \quad (11)$$

The energy dissipation as a function of the damping coefficient due to the deformation of the rubber, c_R , and that due to the viscosity of the fluid, c_F , is introduced as

$$\phi = \frac{1}{2} c_R \dot{w}^2 + 2 \left(\frac{1}{2} c_F \dot{\bar{u}}_x^2 \right) = \frac{1}{2} c_R \dot{w}^2 + c_F \dot{\bar{u}}_x^2 \quad (12)$$

From the Lagrangian equation

$$\begin{cases} \frac{d}{dt} \left(\frac{\partial(T-U)}{\partial \dot{w}} \right) - \frac{\partial(T-U)}{\partial w} + \frac{\partial \phi}{\partial \dot{w}} = 0 \\ \frac{d}{dt} \left(\frac{\partial(T-U)}{\partial \dot{\bar{u}}_x} \right) - \frac{\partial(T-U)}{\partial \bar{u}_x} + \frac{\partial \phi}{\partial \dot{\bar{u}}_x} = 0 \end{cases} \quad (13)$$

the equation of motion of free vibration is given by

$$\begin{cases} m\ddot{w} + c_R\dot{w} + k_b w - 2k_d \bar{u}_x = 0 \\ 4(1+\alpha)\rho L s \ddot{\bar{u}}_x + 3c_F \dot{\bar{u}}_x - 3k_d w + 3k_t \bar{u}_x = 0 \end{cases} \quad (14)$$

Strictly, Eq. (14) cannot be reduced to an eigenvalue problem because the elastic behavior of rubber depends on frequency and amplitude. However, approximation analysis is generally carried out by ignoring the dependences on frequency and amplitude when small vibration at and around a certain frequency is discussed. The rubber material used in the proposed dynamic damper has little dependence on frequency and amplitude in the frequency range between the two peaks in Fig. 4, as explained later. Hence, we consider Eq. (14) to be a linear problem.

From Eq. (14), the equation of motion of the system undergoing undamped free vibration is given by

$$\begin{bmatrix} m & 0 \\ 0 & 4(1+\alpha)\rho L s \end{bmatrix} \begin{Bmatrix} \ddot{w} \\ \ddot{\bar{u}}_x \end{Bmatrix} + \begin{bmatrix} k_b & -2k_d \\ -3k_d & 3k_t \end{bmatrix} \begin{Bmatrix} w \\ \bar{u}_x \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \quad (15)$$

By solving Eq. (16), the natural frequencies of the system can be obtained.

$$\det \begin{bmatrix} k_b - m\Omega^2 & -2k_d \\ -3k_d & 3k_t - 4(1+\alpha)\rho L s \Omega^2 \end{bmatrix} = 0 \quad (16)$$

Equation (16) has two solutions given by

$$\Omega_1 = \sqrt{\frac{1}{2} \left(\frac{k_b}{m} + \frac{3k_t}{4(1+\alpha)\rho L s} \right) - \frac{1}{2} \sqrt{\left(\frac{k_b}{m} - \frac{3k_t}{4(1+\alpha)\rho L s} \right)^2 + \frac{6k_d^2}{(1+\alpha)m\rho L s}}} \quad (17)$$

$$\Omega_2 = \sqrt{\frac{1}{2} \left(\frac{k_b}{m} + \frac{3k_t}{4(1+\alpha)\rho L s} \right) + \frac{1}{2} \sqrt{\left(\frac{k_b}{m} - \frac{3k_t}{4(1+\alpha)\rho L s} \right)^2 + \frac{6k_d^2}{(1+\alpha)m\rho L s}}} \quad (18)$$

where Ω_1 and Ω_2 are the natural frequencies of the primary and secondary modes, respectively. According to Eqs. (17) and (18), the natural frequencies of the proposed damper are determined by the length and cross-sectional area of the channels, the weight of the rigid mass, and the bulk spring constants of the chambers. Equations (17) and (18) are general solutions. As an exception, when the fluid channels are blocked, i.e., $s = 0$, the natural frequencies are given by

$$\begin{cases} \Omega_1 = 0 \\ \Omega_2 = \sqrt{k_b/m} \end{cases} \quad (19)$$

The system substantially becomes a 1-DOF system. Oppositely, the inertial force of the fluid decreases when the cross-sectional area of a fluid channel s is large. If this can be ignored, the DOF of the system also becomes one, and its natural frequency is given by Eq. (20), similarly to the case of dynamic dampers using only the viscous damping of fluids.

$$\begin{cases} \Omega_1 = \sqrt{k_s/m} \\ \Omega_2 = \infty \end{cases} \quad (20)$$

Here, k_s is the spring constant when the inertial force of the fluid is ignored, and is given in Eq. (23). When the damping term and the inertial term of the fluid in Eq. (14) are ignored,

we obtain

$$\begin{cases} m\ddot{w} + k_b w - 2k_d \bar{u}_x = 0 \\ -3k_d w + 3k_t \bar{u}_x = 0 \end{cases} \quad (21)$$

By substituting the second equation of Eq. (21) into the first equation to remove $\bar{u}_x$, we obtain

$$m\ddot{w} + \left(k_b - \frac{2k_d^2}{k_t} \right) w = 0 \quad (22)$$

Denoting the term corresponding to the spring constant as k_s and substituting Eqs. (9)-(11) into Eq. (22), we obtain

$$k_s = k_b - \frac{2k_d^2}{k_t} = \frac{K_1 K_2 (A_1 + 2A_2)^2}{2K_1 + K_2} \quad (23)$$

For weak coupling between the vibration mode of the structure and that of the fluid, *e.g.*, when the rigid mass m is very large, the natural frequency of the resonance mode of the fluid (the secondary mode of the damper) is approximated, using Eq. (18), as

$$\Omega_2 = \sqrt{\frac{3k_t}{4(1+\alpha)\rho Ls}} \quad (24)$$

This is the case of a hydraulic engine mount.⁽⁴⁾⁽⁵⁾

2.3 Static stiffness

The static spring constant between the rigid mass and the mounting pedestals is not directly related to the damping performance of the damper. However, it is an important parameter when a damper is sandwiched between two components as a support, with the aim of reducing the relative vibration between the two components, similarly to the case of engine mounts.

In Eq. (22), the coefficient of the restoring force is the spring constant between the rigid mass and the pedestals, which is given by Eq. (23) and is independent of the dimensions of the fluid channels. Equation (23) also represents the relationship between the static spring constant and the static bulk spring constants of the chambers; however, the static and dynamic spring constants are different, even though they are expressed by the same equation, because the elastic properties of rubber are different between static and dynamic cases. In this study, the statically measured spring constant, k_s , was used to calculate K_1 and K_2 , as explained in section 3.

3. Measurement of vibration characteristics of the damper

A rigid mass (see Fig. 1) was fabricated from a metal plate, and the rubber (see Fig. 2) that form the fluid channels and chambers were molded using a die. A dynamic damper was made by gluing them together, with fluid being enclosed inside. Antifreeze solution, which is frequently used in hydraulic engine mounts, was used as the fluid. Table 1 summarizes the dimensions and weights of each section. In the schematic shown in Fig. 2, the chamber on each side is connected to the central chamber via a single fluid channel; however, because of the dimensional limitations, the proposed dynamic damper has two fluid channels with the same cross-sectional area on each rubber wall between the chambers. Although the analytical equations in section 2 were derived under the assumption of a single fluid channel in each rubber wall, these equations can also be used without modification in the case of multiple fluid channels by adopting the sum of the cross-sectional areas of all fluid channels.⁽⁶⁾ The cross-sectional area s in Table 1 is the sum of the cross-sectional areas of the two fluid channels on each rubber wall. The ratio of

the bulk spring constant of chamber 1 to that of chamber 2, K_1/K_2 , was used as a design parameter and the size of the chambers was determined by finite element analysis such that K_1/K_2 would be 0.35.

We examined the characteristics of the damper alone prior to examining its damping effect. Figure 3 shows the setup used for measurement. The damper was fixed on a vibration table that was excited harmonically (sine wave) with a constant acceleration amplitude by an electromagnetic exciter. The exciting frequency was swept slowly from 20 Hz to 200 Hz. The response of the rigid mass was measured using an acceleration pickup. In the measurement setup shown in Fig. 3, the damper undergoes foundation excitation. In this case, the equation of motion is given by Eq. (25), obtained from Eq. (14).

$$\begin{cases} m\ddot{w} + c_R\dot{w} + k_b w - 2k_d \bar{u}_x = k_b w_g + c_R \dot{w}_g \\ 4(1+\alpha)\rho L s \ddot{\bar{u}}_x + 3c_F \dot{\bar{u}}_x - 3k_d w + 3k_t \bar{u}_x = -3k_d w_g \end{cases} \quad (25)$$

Here, w_g is the displacement of the foundation excitation. Since the excitation is harmonic, we can express the relationship between the displacement and amplitude as Eqs. (26)-(28), and then we have Eqs. (29) and (30).

$$w = W e^{j\omega t} \quad (26)$$

$$w_g = W_g e^{j\omega t} \quad (27)$$

$$\bar{u}_x = \bar{U}_x e^{j\omega t} \quad (28)$$

$$\bar{U}_x = \frac{3k_d W - 3k_d W_g}{d_k} \quad (29)$$

$$W = \frac{(k_b + j\omega c_R)d_k - 6k_d^2}{(-\omega^2 m + j\omega c_R + k_b)d_k - 6k_d^2} W_g \quad (30)$$

Here, W , W_g , and $\bar{U}_x$ are the displacement amplitude of the rigid mass in the z direction, that of the foundation excitation, and the average displacement amplitude of the fluid in the fluid channels, respectively. All of them are complex numbers. Moreover, j is the imaginary unit, t is time, and ω is the angular frequency. d_k in Eqs. (29) and (30) is given by

$$d_k = -4(1+\alpha)\omega^2 \rho L s + j3\omega c_F + 3k_t \quad (31)$$

Figure 4 shows the frequency response of the rigid mass obtained from the analysis and experiment. In the figure, the bold and thin lines represent the experimental and analytical results, respectively. The left and right vertical axes indicate the absolute ratio of W to W_g and the phase difference between them, respectively, and the horizontal axis indicates frequency.

In the analysis, K_1 and K_2 were obtained by substituting their ratio ($K_1/K_2 = 0.35$) and k_s into Eq. (23). Here, k_s is the spring constant between the rigid mass and the mounting pedestals when the inertia and viscous effect of the fluid are ignored, and was obtained as follows. We first measured the static spring constant between the rigid mass and the mounting pedestals by gradually applying force to the rigid mass in the z direction and measuring the displacement in the same direction. Then we multiply the obtained static spring constant by the dynamic correction coefficient of the rubber material to obtain k_s , which was found to be approximately 2×10^4 N/m. The fluid channel's length was corrected in a similar way to the case of resonant silencers (the correction value for length is $1.6R$, where R is the radius of the orifice with circular cross section).⁽⁷⁾ For the specifications in Table 1, the correction coefficient α is 0.6. Regarding the validity of this correction, we confirmed that our prototype dampers show good agreement in terms of natural frequency between the analytical and experimental results.

Table 1 Specifications of the damper

Weight of rigid mass	$m = 90$ g
Density of fluid	$\rho = 1000$ kg/m ³
Weight of fluid	14 g
Weight of rubber	11 g
Cross sectional area of fluid channel	$s = 30$ mm ²
Length of fluid channel	$L = 6$ mm
Area of middle pedestal	100 mm ²
Area of side pedestals	250 mm ²
Width of the damper	60 mm
Length of the damper	90 mm

Fig. 3 Schematic of the damper under foundation excitation

Fig. 4 Frequency responses of the mass excited by foundation excitation

Fig. 5 Reaction force of the damper applied to the vibration table

Fig.6 Schematic of a spring-mass-damping system under foundation excitation

Fig.7 Schematic of a traditional dynamic damper consisting of rubber and a rigid mass

For the damping coefficients in the analysis, the measured damping ratios of the primary and secondary modes ($\zeta_1 = 0.25$ and $\zeta_2 = 0.14$) were used. Because the analysis was carried out using the actually measured spring constant and damping ratios, the analytical and experimental results are in good agreement in Fig. 4. In general, it is difficult to obtain such high accuracy when measurement data are unavailable. However, the above analysis is considered to be effective for obtaining guidelines for tuning, even if the high accuracy in Fig. 4 cannot actually be achieved.

The vibration of the main system (the target of damping) is reduced by the reaction force applied to it by the damper. Therefore, in the design phase, the damping effect is frequently evaluated by calculating the reaction force with respect to foundation excitation. The reaction force applied to the vibration table from the damper via the mounting pedestals, f_r , is given by

$$f_r = K_1 A_1 (A_1 (w - w_g) - 2s\bar{u}_x) + 2K_2 A_2 (A_2 (w - w_g) + s\bar{u}_x) \quad (32)$$

Assuming that

$$f_r = F_r e^{j\omega t} \quad (33)$$

the amplitude of the reaction force F_r is given by

$$F_r = \frac{(k_b d_k - 6k_d^2) \omega^2 m}{(-\omega^2 m + j\omega c_R + k_b) d_k - 6k_d^2} W_g \quad (34)$$

Figure 5 is obtained by plotting the values of F_r from Eq. (34) using the parameters in Table 1. This reaction force is similar to the superposition of the reaction forces from two traditional dynamic dampers installed to the main system in parallel.

4. Damping effect and discussion

To evaluate the damping effect of the proposed dynamic damper, we carried out experiments using a 1-DOF spring-mass-damping system (the main system, i.e., the target of damping) under foundation excitation (see Fig. 6). The damping effects on the resonances with different frequencies were examined when the resonance frequency of the main system was varied as 58 Hz, 84 Hz, 110 Hz, and 157 Hz by changing the spring but not the mass. In the experiments, the main system was fixed on the vibration table, which

Fig. 8 Damping effect of the proposed damper and that of a traditional dynamic damper to resonances with different frequencies

was excited by an electromagnetic exciter with sine waves. The acceleration amplitude was controlled at a constant value, and the frequency was swept gradually. The response of the mass of the main system was measured using a small acceleration pickup. First, the response of the main system alone without the damper was measured. Next, a similar measurement was carried out by mounting the proposed damper onto the main system, and the result was compared with that of the former case. The ratio of the weight of the damper to that of the mass of the main system was 12.5%. Figure 8 shows the damping effect of the proposed dynamic damper for resonances with different frequencies. For comparison, the results for a conventional rubber dynamic damper with the same weight are also plotted in the figure. The conventional dynamic damper comprises a rubber section and a rigid mass (the loss coefficient of the rubber is approximately 0.15) and is tuned to the 110 Hz resonance of the main system (see Fig. 7).

As shown in Fig. 4, the damper developed in this study has two resonance peaks at 53 Hz and 126 Hz. Therefore, the damping effect shown in Fig. 8 was large for resonances with the frequencies of 58 Hz, 110 Hz, and 157 Hz but relatively small for the resonance with a frequency of 84 Hz, which corresponds to a valley in the frequency responses shown in Fig. 4. The damping effect of the conventional dynamic damper tuned to 110 Hz is large at and around the tuning frequency but small at other frequencies.

5. Conclusions

We improved a rubber dynamic damper by preparing three chambers connected via narrow fluid channels (orifices) in the rubber section and introducing a small amount of fluid into the chambers. Using the resonance of the fluid, we realized a 2-DOF dynamic damper with only a slight increase in weight. The weight of the introduced fluid was approximately 12% of the total weight of the damper developed in this study (the sum of the weights of the rigid mass, rubber, and fluid).

We formulated the motion of the proposed damper for its design, and demonstrated that its vibration characteristics can be estimated by analysis, and that it can be tuned to the main system (the target of damping) by changing the length, the cross-sectional area of the fluid channels, and the bulk spring constants of the chambers. From the experiments on the damping effects to the main system with different resonance frequencies, it was verified that

the proposed damper can reduce vibrations over a wider frequency range than a conventional rubber dynamic damper with the same weight, and can simultaneously cut two resonance peaks.

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