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Dynamic Modeling and Experimental Analysis of Magnetic Fluid Dampers

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Abstract

The paper puts forward a magnetic fluid damper with simple structure. A dynamic model of magnetic fluid in the damper is built up based on Newton's second law and the continuity equation. The representation of logarithmic decay rate of damper imposing on the back beam of cantilever beam is acquired through the oscillating energy of an elastic cantilever beam. The designed experiments prove that logarithmic decay rate of beam oscillation is influenced by different radius of permanence magnets in dampers, different gap between casing and permanent magnet and magnetic fluid with different saturation magnetization. It is indicated that theory and experimental results are in agreement under the assumed condition. In particular, it is found that the damping effect of magnetic fluid dampers is improved with the increment of radius of permanent magnet within a certain scope, there is an optimal gap between damper casing and permanent magnet, which can make beam oscillation reach the maximum logarithmic decay rate.

Key words

Magnetic fluid, damper, damping, dynamic model.

1. Introduction

A magnetic liquid is a stable colloidal suspension formed with the dispersion of nano-ferromagnetic or ferrimagnetic particles coated with a surfactant in a liquid carrier liquid. The application of magnetic fluids to damping was proposed at the beginning of the application of magnetic liquids [1]. Then magnetic fluid dampers have been developed for many types. And in 1987, Katsuto Nakatsuka [2] proposed a piston-type magnetic liquid damper, which was used in combination with an air spring to control the vibration of the stage. When electro rheological fluid is mixed with magnetic fluid instead of magnetic liquid in piston damper, Toyohisa Fujita [3-4] found that the improved damper can damp the vibrations in a wider frequency range. Hideaki Fukuda [5-6] further installed an electromagnet outside the piston-type magnetic liquid damper to achieve active control of vibration. So in 1998, Abé Masato [7] proposed a tuned magnetic liquid damper which was subsequently developed by Yasuhiro Ohira [8-9] and was validated by simulation and experiment. In 2002, V. G. Bashtovoi [10] created a magnetic liquid dynamic vibration absorber, which is found to be most suitable for damping small amplitude (less than 1mm) and small frequency (less than 1Hz). In order to reduce the heat generated during the damping process, a magnetic liquid colloid damper has been proposed in recent years [11]. In order to apply the damper to the micro-machine, the porous elastic flaky magnetic liquid damper have been developed [12]. Up to now, some magnetic liquid dampers have been used in mechanical, instrumentation and aerospace fields, and some are still at the theoretical analysis stage, but as a support in damping devices, damping fluid is still the most potential application of magnetic liquid one.

In this paper, aimed at a field that requires a compact and less dissipative energy, we proposed a new kind of magnetic liquid dampers, and gave the dynamic model of the damper, and found out that the effectiveness of the model is verified by experiments.

2. Working Principle of Magnetic Liquid Damper Shock Absorber

As shown in Fig.1, the magnetic liquid damper is composed with a cylindrical nonmagnetic housing, a magnetic liquid filled in the housing, and a cylindrical permanent magnet immersed in the magnetic liquid. If ignoring the gravity of the permanent magnets, we obtained that the permanent magnets will be separated from the inner wall of the housing and suspended in the center of the magnetic liquid by the second type of suspension principle [13]. The use of the shell and the vibration of the object is fixed, the external vibration will cause the permanent magnet and the relative motion between the shells, resulting in a magnetic fluid with a velocity gradient within the flow, so you can rely on the principle of liquid viscosity energy dissipation energy [14].

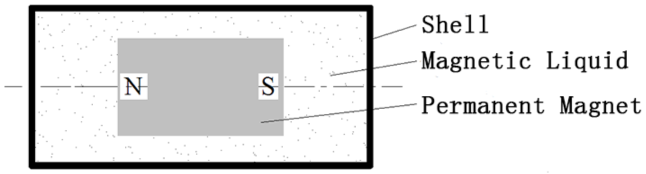

Fig.1. magnetic liquid damper shock absorber structure

In this paper, in order to study simply, with one end fixed, and the other end of the free elastic cantilever beam generates free vibration. The axis of the beam, that is, the straight line connecting the centroids of each section, is taken as the y-axis, and the direction perpendicular to the y-axis is taken as the x-axis. The bending vibration of the beam in the plane of symmetry is independent of the shear deformation and the influence of the rotation of the neutral axis on the bending vibration. The axis of the beam has only the lateral displacement x (y, t), in which t is the time variable. The damper is mounted at the lowest end of the beam and the axis of the cylindrical damper is in the direction of vibration.

3. Dynamic Modeling of Magnetic Fluid Dampers

3.1 Calculate the Vibration Energy of Elastic Cantilever Beam

The elastic cantilever beam with one end fixed and the other end free is free to have the following expression:

$$x(y,t) = x_0(y)\cos\omega t \tag{1}$$

$$x_0(y) = -A\frac{\begin{bmatrix}(\cos\beta l + \mathrm{ch}\beta l)(\cos\beta y - \mathrm{ch}\beta y) + \\ (\sin\beta l - \mathrm{sh}\beta l)(\sin\beta y - \mathrm{sh}\beta y)\end{bmatrix}}{2\sin\beta l\,\mathrm{sh}\beta l} \tag{2}$$

In which A is the initial offset, β=1.875/l, l is the length of the beam, and is the frequency of the free vibration of the beam.

In the equation (1), we can see that at the time point $t_n=(2n+1)\pi/2\omega$ (n=0, 1, 2…), the beam is in the equilibrium position without force, only the kinetic energy:

$$W=\int_0^l \frac{1}{2}\left(\frac{\partial x(y,t)}{\partial t}\right)^2 \rho S dy=\frac{\rho S\omega^2}{2}\sin^2\omega t\int_0^l\left[x_0(y)\right]^2 dy \tag{3}$$

Where S is the cross-sectional area of the beam and ρ is the density of the beam material. t=t_n, And the expression $x_0(y)$ into the formula (3), and the integral calculation, the vibration energy of the beam can be:

□

$$W=\frac{\rho S\omega^2}{2\beta}\left[\begin{array}{l}\left(\alpha_1^2-\alpha_2^2\right)\left(\frac{1}{2}\sin\beta l-\mathrm{sh}\beta l\right)\cos\beta l+\\ \left(\alpha_1^2+\alpha_2^2\right)\left(\frac{1}{2}\mathrm{sh}\beta l-\sin\beta l\right)\mathrm{ch}\beta l+\\ \alpha_1\alpha_2\left(\sin\beta l-\mathrm{sh}\beta l\right)^2+\alpha_1^2\beta l\end{array}\right] \tag{4}$$

In which,

$\alpha_1=-A\dfrac{\cos\beta l+\mathrm{ch}\beta l}{2\sin\beta l\,\mathrm{sh}\beta l}$, $\alpha_2=-A\dfrac{\sin\beta l-\mathrm{sh}\beta l}{2\sin\beta l\,\mathrm{sh}\beta l}$.

3.2 Calculate the Viscous Energy Dissipation in Magnetic Liquid

By the viscous energy dissipation in the magnetic fluid and the vibration of the beam is restrained. The logarithmic decay rate of the beam vibration $\varLambda$ is used to measure the strength of this inhibition:

$$\varLambda=\ln\left(\frac{A_{i-1}}{A_i}\right) \tag{5}$$

Where A_i is the amplitude of the i-th vibration, and W_i is the vibration energy of the beam. If A_i is the amplitude, W_i is proportional to (1), (2) and (3):

$$\varLambda=\frac{1}{2}\ln\left(\frac{W_{i-1}}{W_i}\right) \tag{6}$$

If $\varDelta W$ is the viscous energy dissipation of the magnetic fluid in the i-th oscillation, $W_{i-1}=W_i+\varDelta W$, the logarithmic decay rate can be expressed as:

$$\varLambda=\frac{1}{2}\ln\left(1+\frac{\varDelta W}{W_i}\right) \tag{7}$$

Next, we will calculate the viscous energy dissipation of the magnetic liquid within one period T of the vibration of the beam. Assumed that the housing is much longer in the axial direction than the permanent magnets therein, it is considered that the permanent magnet vibrates only at the central portion in the housing, and the restoring force exerted by the magnetic liquid in the axial direction of the permanent magnets can be ignored. It is also assumed that the gap between the permanent magnet and the housing is much smaller than the size of the permanent magnet, and since the frequency of vibration of the housing along with the vibration-damping member is small, the flow of the magnetic liquid in the gap between the permanent magnet and the housing can also be considered to be along the axis directional quasi-steady flow.

According to the axial symmetry of the shock absorber structure, the problem in the three-dimensional space is reduced to a two-dimensional plane problem. As shown in Fig. 2, only the part above the symmetry axis can be considered. The cylindrical permanent magnet is suspended in the magnetic liquid in the cylindrical shell, and let the inner diameter of the shell be R, the mass of the permanent magnet be m, the length be l_m, the density be m, the radius be r_a, the end surface area be S_m and the outer cylindrical surface be S_a, the density of the magnetic liquid be ρ_f, and the viscosity be η. Ignored the magnetic field of permanent magnets on the magnetic liquid viscosity, it is considered that the permanent magnet is centered in the radial direction in the magnetic liquid and the gap between the permanent magnet and the outer shell is g_a.

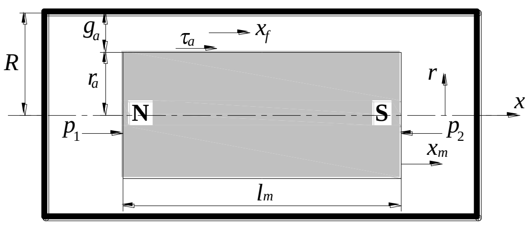

Fig.2. parameters of magnetic liquid damper shock absorber

Assume the displacements of the outer axial direction relative to the initial position is x, and the displacement of the permanent magnet relative to the housing is x_m, the displacement of the magnetic liquid relative to the housing is x_f, and the velocity is v_f. Since neglected the effect of the magnetic field and gravity on the magnetic liquid, we obtain the N-S equation of the magnetic fluid flow in the gap in the r-direction can be simplified as:

$$\frac{\partial p}{\partial r} = 0 \tag{8}$$

By (8), we know that the pressure in the magnetic liquid does not change with the coordinate r. When the pressure of the magnetic liquid on the point on both ends of the permanent magnet is p_1 and p_2 respectively, the pressure of the magnetic liquid on all points on both ends of the permanent magnet is p_1 and p_2, and the pressure inside the magnetic liquid which is also at the same x-position as the both-end faces of the permanent magnets is p_1 and p_2.

Magnetic fluid can be regarded as Newtonian fluid, its internal shear stress can be expressed as

$$\tau = \eta \frac{\partial v_f}{\partial r} \tag{9}$$

Since the magnetic fluid in the gap has a one-dimensional quasi-stationary flow, v_f does not vary with the coordinate x, nor does it change with the coordinate x by equation (9). Therefore, the magnetic liquid layer in the same r-position in the gap is subjected to the same shear and the shear stress of the magnetic liquid on the outer cylindrical surface of the permanent magnet is uniformly expressed as τ_a.

Using the Newton's second law, the cylindrical permanent magnet in the x-direction has

$$m(\ddot{x} + \ddot{x}_m) = S_m(p_1 - p_2) + S_a \tau_a \tag{10}$$

Where $S_m = \pi r_a^2$, $S_a = 2\pi r_a l_m$. Assuming that the acceleration of the permanent magnet relative to the casing is negligible compared to the acceleration of the casing itself, $\ddot{x}_m << \ddot{x}$,equation (10) becomes

$$m\ddot{x} = S_m(p_1 - p_2) + S_a \tau_a \tag{11}$$

In order to determine the value of the right-hand side of the equal sign in (11), it is necessary to determine the velocity distribution of the magnetic liquid in the gap between the casing and the permanent magnet when the casing is vibrated with the vibration-damping member. A magnetic fluid layer having a radius r and a thickness dr of the gap and having the same length as that of the permanent magnet is shown in Fig. 3.

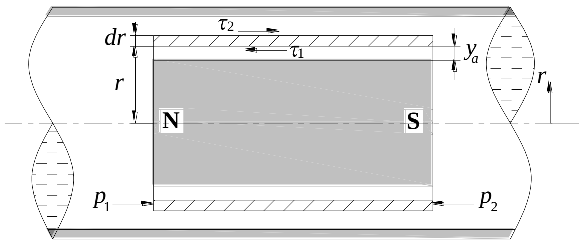

Fig.3. the permanent magnet and the gap between the shell magnetic fluid force

Apply the Newton's second law to the magnetic liquid layer

Let the radial distance between the layer and the outer surface of the permanent magnet be y_a, which is a variable. Based on the above analysis, it is assumed that the shear stress of the magnetic liquid on the inner and outer sides of the magnetic liquid layer is τ_1 and τ_2, respectively, and the compressive stresses of the magnetic liquid on the layers at the left and right ends are p_1 and p_2, respectively.

$$2\pi r dr l\rho_f(\ddot{x}_f+\ddot{x}) = p_1 2\pi r dr - p_2 2\pi r dr + \tau_2 2\pi r l_m - \tau_1 2\pi r l_m \tag{12}$$

Where in the force generated by the shearing stress outside the liquid layer can be obtained by using the inner shearing stress according to the rate of change of the force in the r-direction:

$$\tau_2 2\pi r l_m = \tau_1 2\pi r l_m + \frac{\partial}{\partial r}(\tau_1 2\pi r l_m)dr \tag{13}$$

Take (9), (13) into equation (12) and assume that the acceleration of the magnetic liquid relative to the shell can be ignored, we have

$$-\frac{p_1-p_2}{l_m}+\rho_f\ddot{x} = \frac{\eta}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v_f}{\partial r}\right) \tag{14}$$

In addition, for the flow of the magnetic fluid in the shell and the permanent magnet gap, there are boundary conditions

When r=ra, $v_f = \dot{x}_m$; When r=R, vf=0 (15)

Solve equation (14), and finally by the boundary conditions, we obtain

$$v_f = \left(1-\frac{y_a}{g_a}\right)\dot{x}_m - \frac{Fg_a^2}{4\eta}\frac{y_a}{g_a}\left(1-\frac{y_a}{g_a}\right) \tag{16}$$

In which,

$$F = -\frac{p_1-p_2}{l_m}+\rho_f\ddot{x} \tag{17}$$

According to the equation (16), we can calculate the gap between the shell and permanent magnets for magnetic fluid flow

$$Q = \int_0^{g_a} 2\pi r v_f dy_a = \pi R g_a \dot{x}_m - \frac{\pi R g_a^3}{12\eta} F \tag{18}$$

And acting on the shear stress of the cylindrical permanent magnet surface

$$\tau_a = \eta \left(\frac{\partial v_f}{\partial y_a} \right)_{y_a = 0} = -\frac{\eta}{g_a} \dot{x}_m - \frac{g_a}{4} F$$

(19)

By all of the magnetic fluid inside the shell of conservation of mass, we have

$$\dot{x}_m S_m + Q = 0 \tag{20}$$

S_m with equation (18) into equation (20), we simplify and obtain

$$F = \frac{12\eta}{R g_a^3} \left[R g_a + (R - g_a)^2 \right] \dot{x}_m \tag{21}$$

Take equation (21), respectively, into the substitution equation (19) and equation (17)

$$\tau_a = -\frac{\eta}{R g_a^2} \left(3R^2 - 2R g_a + 3 g_a^2 \right) \dot{x}_m \tag{22}$$

$$p_1 - p_2 = \rho_f l_m \ddot{x} - \frac{12 l_m \eta}{R g_a^3} \left(R^2 - R g_a + g_a^2 \right) \dot{x}_m \tag{23}$$

And take equation (22) and (23) into equation (11), we have

$$\dot{x}_m = \frac{(R - g_a)(\rho_f - \rho_m) R g_a^3}{2\xi\eta} \ddot{x} \tag{24}$$

Among them

$$\xi = 6R^3 - 9R^2 g_a + 10 R g_a^2 - 3 g_a^3$$

Take equation (21) into equation (16), we have

$$v_f = \frac{1}{R g_a^3} \begin{bmatrix} 3\left(R^2 - R g_a + g_a^2 \right) y_a^2 - \\ g_a \left(3R^2 - 2R g_a + 3 g_a^2 \right) y_a + R g_a^3 \end{bmatrix} \dot{x}_m \tag{25}$$

Combined equation (24) with equation (25), vibration acceleration of y position point on the beam is introduced by equation (1) , we get the beam during vibration damper in magnetic liquid flow velocity is as follows:

$$v_f = -\omega^2 \frac{(R - g_a)(\rho_f - \rho_m)}{2\xi\eta} x_0(y) \cos \omega t \cdot \left[3\left(R^2 - R g_a + g_a^2 \right) y_a^2 - g_a \left(3R^2 - 2R g_a + 3 g_a^2 \right) y_a + R g_a^3 \right] \tag{26}$$

Since the time of incompressible fluid in a unit of energy dissipation can be expressed as

$$\Delta W' = \frac{1}{2}\eta\int\left(\frac{\partial v_i}{\partial x_k}+\frac{\partial v_k}{\partial x_i}\right)^2 dV$$

There into, v_i and v_k are components of fluid movement speed on the coordinates x_i and x_k respectively, and V is the volume of fluid. This article only considers the flow of the magnetic fluid in the permanent magnet and the shell gap so the energy consumption also occurs only in the flow, and ignores the influence of the magnetic field of magnetic fluid viscosity so its energy dissipation can be simplified as

$$\Delta W' = \eta\int\left(\frac{\partial v_f}{\partial y_a}\right)^2 dV \tag{27}$$

Take equation (26) in the equation, and due to the $dV = 2\pi l_m(r_a + y_a)dy_a$, we have

$$\Delta W' = \frac{\pi}{2}\omega^4\zeta\frac{(\rho_m-\rho_f)^2}{\eta}[x_0(y)]^2\cos^2\omega t \tag{28}$$

Among them

$$\zeta = l_m r_a^2 g_a^3\frac{3r_a^5+\frac{15}{2}r_a^4 g_a+12r_a^3 g_a^2+11r_a^2 g_a^3+6r_a g_a^4+g_a^5}{(6r_a^3+9r_a^2 g_a+10r_a g_a^2+4g_a^3)^2} \tag{29}$$

In the vibration of the beam within the period T the viscosity of magnetic fluid for consumption

$$\Delta W = \int_0^T \Delta W' dt = \frac{\pi^2}{2}\omega^3\zeta\frac{(\rho_m-\rho_f)^2}{\eta}[x_0(y)]^2 \tag{30}$$

3.3 Calculate Suspension Dynamic Logarithmic Decrement

By equation (7), (4) and (30), and can be equipped with magnetic liquid damping of shock absorber suspension elastic dynamic expression of the logarithmic decrement

$$\Lambda = \frac{1}{2}\ln\left\{1+\frac{\pi^2}{2}\omega^3\zeta\frac{(\rho_m-\rho_f)^2}{\eta}[x_0(y)]^2\frac{1}{W}\right\} \tag{31}$$

Which ζ and $x_0(y)$ respectively by equation (29) and (2) is given.

4 Experiments

As shown in the experiment in the fig.4 (fig.5 for its corresponding physical photo), we study on the device that elastic cantilever beam selection of brass, size is 1200 * 50 * 5 mm, weighing 2.55 kg. Its one end is fixed and the other end installed with magnetic fluid shock absorber, beam free vibration frequency ω is 1.74 Hz, the initial amplitude at the moment of experiment is 15mm. The semiconductor acceleration sensor fixed at the free end of a beam can get voltage signal which forms quantitative relationship with vibration acceleration, the signal can be read by the data collector (DI-710), converted by modulus and input into the computer which is connected with it and the sampling frequency of data collector is 200 Hz. The signal input into the computer is first converted to acceleration data and beam vibration speed and displacement data are acquired by acceleration data.

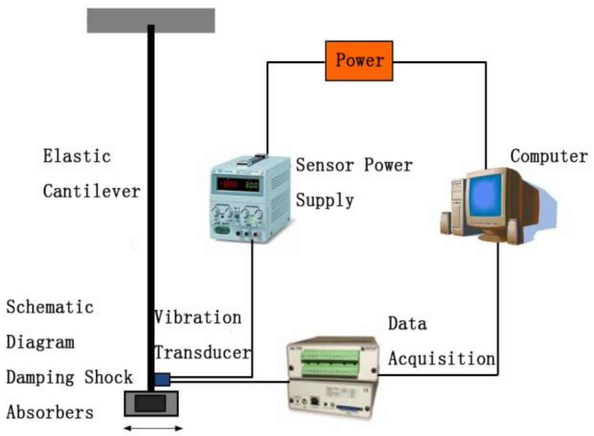

Fig.4. experiment device schematic diagram

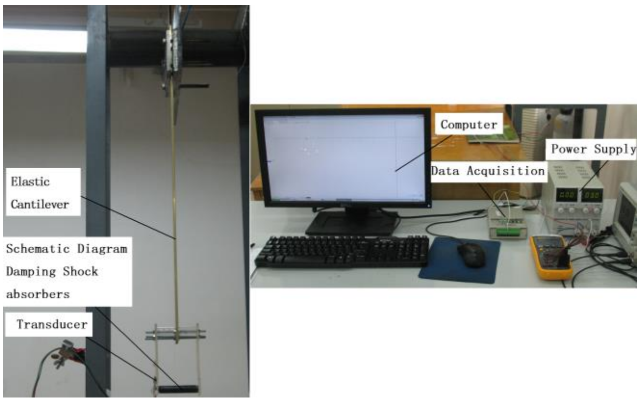

Fig.5. experimental device object graph

In order to verify the correctness of the theory, we validate r_a, permanent magnet radius, g_a, the gap between permanent magnet and the shell, and the influence from the magnetic liquid with different saturation magnetization on the logarithmic decrement rate of cantilever. The structure parameters of damping shock absorber and the physical parameters of magnetic fluid are given in Table 1. The material for permanent magnet all is Nd-Fe-B, its length is 12 mm, and the base load fluid of magnetic liquid is kerosene, the dispersed are magnetic particles of Fe_3O_4. The assembled shock absorber weighs 19.93 g at most.

Table 1. Experimental parameters

<table>
<tr><th>Experimental Purpose</th><th>Inner Diameter of Casing 2×R/ mm</th><th>Diameter of Permanent Magnet 2×r_a/ mm</th><th>Saturation Magnetization of Magnetic Liquid M_s/ kA/m</th></tr>
<tr><td rowspan="5">Impact from r_a on Vibration Logarithmic Decrement Rate</td><td>4.5</td><td>2.5</td><td rowspan="5">27.01</td></tr>
<tr><td>8.5</td><td>6.5</td></tr>
<tr><td>12</td><td>10</td></tr>
<tr><td>15.5</td><td>13.5</td></tr>
<tr><td>19</td><td>17</td></tr>
<tr><td rowspan="8">Impact from g_a on Vibration Logarithmic Decrement Rate</td><td rowspan="8">12</td><td>2.96</td><td rowspan="8">27.01</td></tr>
<tr><td>3.96</td></tr>
<tr><td>4.96</td></tr>
<tr><td>5.96</td></tr>
<tr><td>7.96</td></tr>
<tr><td>8.46</td></tr>
<tr><td>9.16</td></tr>
<tr><td>10</td></tr>
<tr><td rowspan="5">Impact from different magnetic fluid on Vibration Logarithmic Decrement Rate</td><td rowspan="5">12</td><td rowspan="5">10</td><td>11.94</td></tr>
<tr><td>24.32</td></tr>
<tr><td>27.01</td></tr>
<tr><td>36.46</td></tr>
<tr><td>41.66</td></tr>
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Under the above-mentioned condition, a kind of vibration displacement image is as shown in Fig. 6.

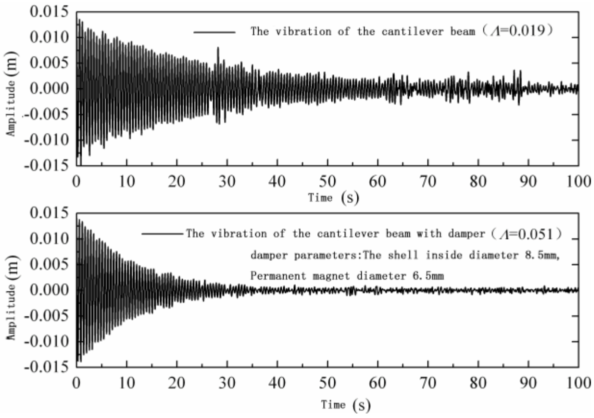
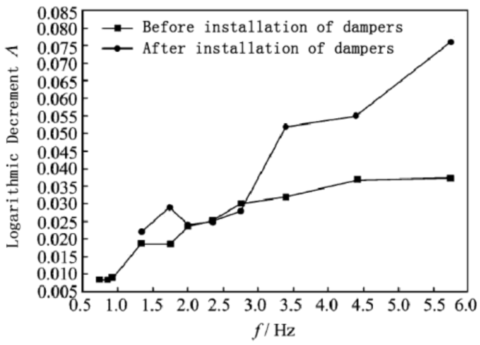


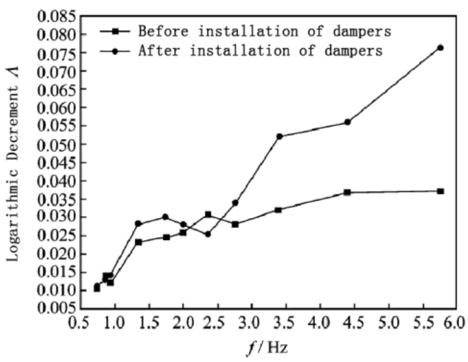
Fig.6. amplitude change curves of free end of a cantilever

5 Analysis on Experiment Findings

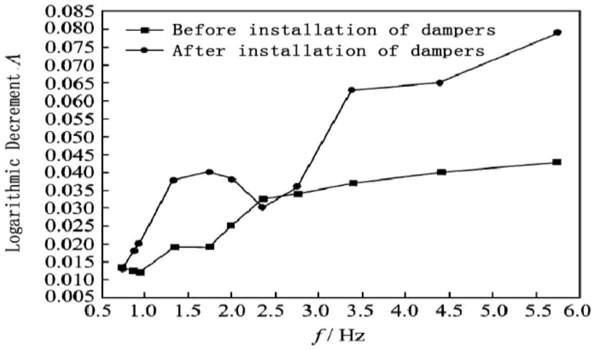
Fig.7 indicates that different initial amplitude is offered at the free end of a cantilever beam before and after installation of dampers, logarithmic decrement rate of cantilever vibration changes along with the change of frequency curve. Fig.8(a)~8(c) show that the three kinds of different initial amplitude of vibration, in the frequency of the experimental area, after installation of dampers, logarithmic decrement rate of cantilever vibration first increases as the frequency increases, decreases as the frequency continues increasing, and then increases again as the frequency increases till its peak 5.75 Hz.



(a) Amplitude is 5 mm

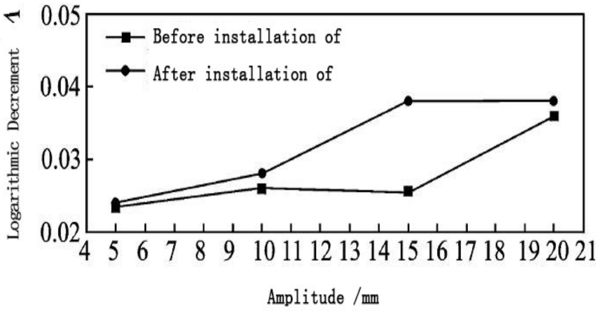


(b) Amplitude is 10 mm

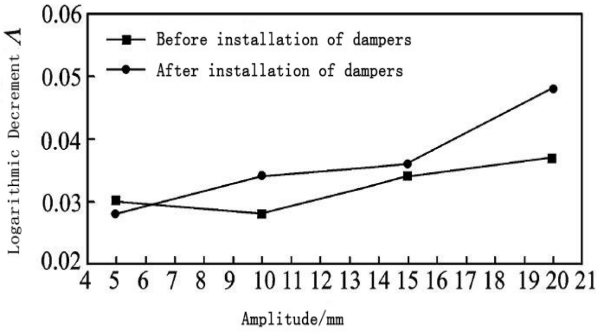


(c) Amplitude is 15 mm

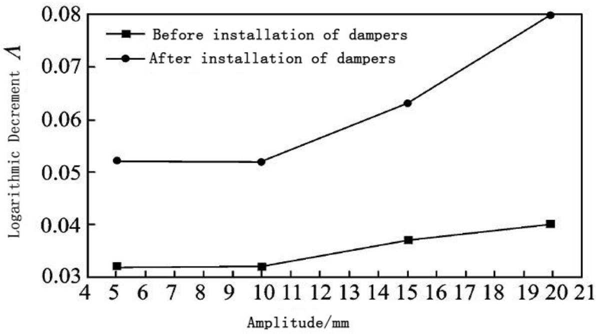
Fig.7 The logarithmic decrement rate under different initial amplitude along with the change of frequency curve



(a) Frequency is 2 Hz (the cantilever length is 1.5m accordingly)



(b) Frequency is 2 Hz (the cantilever length is 1.0m accordingly)



(c) Frequency is 3.39 Hz (the cantilever length is 0.9m accordingly)

Fig.8 The logarithmic decrement rate changing with the amplitude under different frequency curve

Fig.8(a)~8(c) indicate that the cantilever beam with different length before and after installation of dampers, logarithmic decrement curve changes along with the change of amplitude.

Three kinds of different length of copper plate have been chosen and analyzed, after installation of dampers in the amplitude of the experimental area of the cantilever beam, logarithmic decrement rate is greater than the value before the installation, showing that the magnetic fluid damper is the vibration of the cantilever beam at the free end under different initial amplitude and has damping effect, and after installation of dampers, cantilever logarithmic decrement rate is increased with the increment of amplitude, showing that the amplitude becomes greater and the suspension dynamic energy attenuation becomes faster.

To sum up, the logarithmic decrement rate measured by experiments and the length of the cantilever beam has good correlation between amplitude and regularity and the study of magnetic fluid dampers finds out that preliminary applicable length of cantilever beam and amplitude range, which lays a foundation for theoretical and experimental researches.

6 Conclusion

(1) This paper has proposed a kind of magnetic fluid shock absorber with a simple structure, and has established a dynamic modeling of this shock absorber, the vibration logarithmic decrement rate has been acquired through the vibration energy of the elastic cantilever beam, which the shock absorber imposes on back rest of elastic cantilever beam. The experiments show that under the hypothetical conditions, theoretical and experimental results are in a good consistency.

(2) After installation of dampers under experimental conditions, the logarithmic decrement rate of cantilever changes with the frequency and first increases, then decreases, and finally increases again; after installation of dampers, logarithmic decrement rate of cantilever vibration increases with the increment of amplitude.

(3) When the other conditions remain the same, if the radius of permanent magnet becomes less, the damping effect of magnetic fluid shock absorber increases with the increment of the radius of permanent magnet. There is a best gap between shock absorber shell and the permanent magnet to achieve the maximum logarithmic decrement rate of cantilever vibration.

(4) Under the experimental conditions for magnetic fluid damper, different length of cantilever beam vibration plays a very good role. When the length of the cantilever beam is 0.7m, the damper achieves the best damping effect.

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