ISSN (Print) 1226-1750
ISSN (Online) 2233-6656
Journal of Magnetics 23(3), 405-408 (2018) https://doi.org/10.4283/JMAG.2018.23.3.405

A Ferrofluid-based Planar Damper with Magnetic Spring

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(Received 15 May 2018, Received in final form 11 September 2018, Accepted 11 September 2018)

The ferrofluid-based damper has been a subject of active research recently in depressing low-frequency vibration of rod due to its relative small viscous damping for the vibration pick-up system. In this work, a novel ferrofluid-based planar damper with magnetic spring is firstly proposed. Compared with the traditional ferrofluid-based damper with the elastic restoring force of ferrofluids, the novel damper is designed with steady magnetic spring to obtain a large relative displacement and dissipate external vibration energy efficiently. Experimental results show that the novel damper has a high damping performance. Under the same initial vibration amplitude of beam, the logarithmic decrement of the system with the novel damper is 31.5 times and 5.5 times larger than those of the system without damper and with the traditional damper, respectively. In addition, compared with the system with the traditional damper, the system with the novel damper can reduce damping time by half.

Keywords : ferrofluid, damper, magnetic spring, vibration

1. Introduction

The tuned mass damper (TMD) is usually seen as a passive damper and composed of the inertial mass on the spring with the stiffness and damping. The TMD is a traditional device and absorbs energy of structural vibration through the dynamic response motion of the mass to reduce sway vibration in flexible civil structures, such as bridges, buildings etc. [1, 2]. It is known that the large dissipation energy corresponds to big friction resistance and mass. However, the TMD is seldom used in the low frequency damping vibration because low dissipation energy needs small damping to adjust the response motion of the mass. The big damping, such as mechanical spring damping and the friction damping between two solid interfaces, restrain the response motion of the mass under low frequency vibrations from cantilever rods or tubes.

In order to enhance the performance of the TMD in the low frequency vibration of 1-10 Hz, ferrofluid with nanometer-sized magnetic particles, an intelligent magnetic responsive fluid, is used to improve the damping performance of the TMD, for ferrofluids can obtain magnetostatic force in a nonuniform magnetic field and generate viscous dissipation of the mechanical vibration energy as a passive damper in the role of support and damping [3, 4]. Nowadays, ferrofluids have been used in the low frequency damping vibration of rods, and a type of ferrofluid-based damper is used in the single linear vibration [5, 6]. Though many ferrofluid-based planar dampers have been designed and used in the low frequency damping vibration of rods, the restoring force of them is provided by the ferrofluids elastic support, which is small and instable [7].

In this paper, a novel magnetic spring structure is designed and can produce the magnetic attraction used as the restoring force to develop the ferrofluid-based planar damper, and the damping characteristics of the damper will be deeply studied.

2. Structure and Model of the Novel Damper

2.1. Structure Design

A planar vibration-based electromagnetic generator has been proposed to harvest the low frequency human motion energy in our early work [8, 9]. The ferrofluid-based planar damper has the similar structure compared with the generator, shown in Fig. 1. It is well known that ferrofluids, on the end of PMs with a nonuniform magnetic field, can generate magnetic elastic force acting on the

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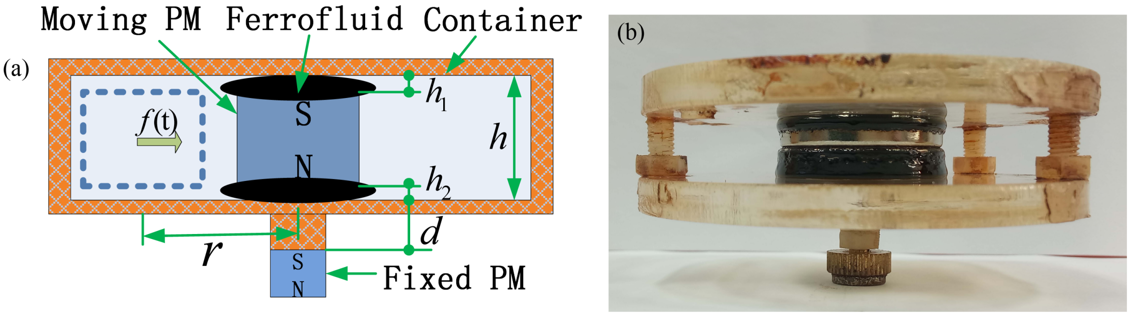

Fig. 1. (Color online) Structure of the ferrofluid-based planar damper with magnetic spring (a) Scheme and (b) Photo.

permanent magnet (PM). The moving PM with diameter of 30 mm and length of 9.5 mm is placed in a non-magnetizable cylindrical container with an inner diameter of 78 mm and an adjustable height. The gaps between the moving PM and inner bottom surfaces of the container are filled with an amount of the kerosene-based ferrofluids (density 1.06×103 kg/m^3 and viscosity 2.51 mPa·s). The moving PM can be supported by the two drops of ferrofluids, which carry out the damping functions. In addition, the moving PM is retained and controlled by the magnetic attraction with the external fixed PM with a diameter of 10 mm and a height of 2 mm. The external fixed PM is located in the external center of the bottom plane of the container.

2.2. Model of the novel Damper

The novel damper is also modeled as a second order spring mass damper system with external excitation [8], and the well known differential equation is

$$m\ddot{r}(t)+c\dot{r}(t)+kr(t)=-m\ddot{s}(t) \tag{1}$$

Here m is the mass of the movable PM, c is the total damping constant, k is the stiffness of the magnetic spring. $r(t)$ is the relative displacement of the vibration mass to the container. $s(t)$ is the displacement of the external excitation vibration on the container.

The moving PM surrounded by ferrofluids makes response to the external vibration. At the same time, the ferrofluids provide the viscous resistance and dissipate the response vibration energy with the damping coefficient c, which can be confirmed by the damping ratio ξ and the logarithmic decrement δ in the following equation.

$$\xi = c_t/(2m\omega_n),\ \delta = 2\pi\xi/\sqrt{1-\xi^2} \tag{2}$$

Under stationary state, the moving PM is in the inner center of the container and is acted by the magnetic attraction of the fixed PM in the axial direction. During movements, the moving PM oscillates in the low bottom of the container and is always subjected to restoring force provided by the parted force of the magnetic attraction in the horizontal direction F [8], which is

$$F = \frac{\mu_0 Q_{mov} Q_{fix} r}{4\pi(d^2+r^2)^{3/2}} \tag{3}$$

It can be seen in equation (3) that the restoring force is affected by the structure parameters of the vertical distance between two PMs d and the displacement of movable PM from central pole r, as well as the vacuum permeability μ_0

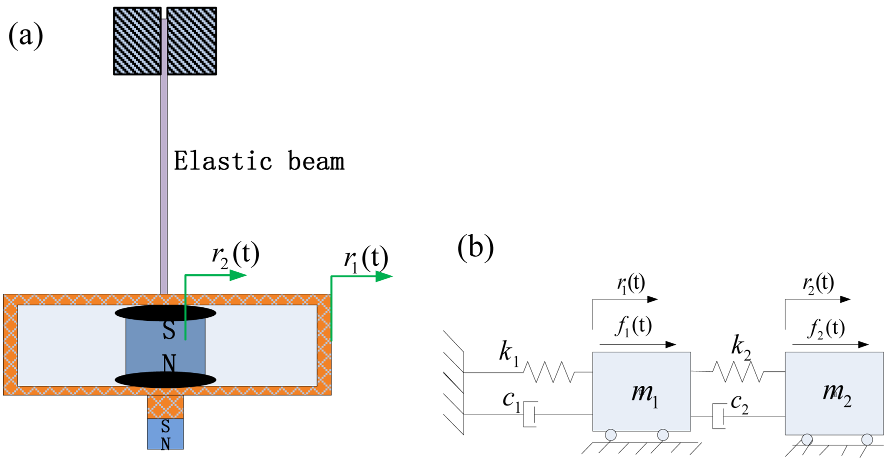

Fig. 2. (Color online) The damping system with the novel damper (a) Scheme and (b) model.

and magnetic field intensity of PMs Q. Based on this, it is known that the stiffness of the magnetic spring $k(=F/r)$ is mainly confirmed by the vertical distance between the movable PM and fixed PM d.

2.3. Model of the rod vibration damping systems with the novel Damper

The ferrofluid-based planar damper can be used to reduce the low frequency of rod in the plane. The low frequency vibration is free oscillations provided by an elastic beam with one end fastened and the other free. The novel damper is fixed on the free end of the elastic beam, which forms the two-degree-of-freedom damping system to realize anti-resonance, as shown in Fig. 2. The damping system can be divided into two parts: the excitation mass and the response mass, and the two masses can move out of phase with each other to transfer the vibration energy. The vibration energy acting on the excitation mass is finally partly dissipated by viscous friction of ferrofluid drops on the moving PM. The model equation of the damping system is

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{bmatrix} \ddot{r}_1(\mathrm{t}) \\ \ddot{r}_2(\mathrm{t}) \end{bmatrix} + \begin{bmatrix} c_1+c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}\begin{bmatrix} \dot{r}_1(\mathrm{t}) \\ \dot{r}_2(\mathrm{t}) \end{bmatrix} + \begin{bmatrix} k_1+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{bmatrix} r_1(\mathrm{t}) \\ r_2(\mathrm{t}) \end{bmatrix} = \begin{bmatrix} f_1(\mathrm{t}) \\ f_2(\mathrm{t}) \end{bmatrix} \quad (4)$$

where r_1(t) and r_2(t) are the absolute transient displacements of the elastic beam and the moving PM, respectively. c_1 and k_1 represent damping coefficient and the stiffness of elastic beam, respectively. f_1(t) and f_2(t) are the external total force, such as gravities, magnetic force etc. In this work, gravities acting on the moving PM and elastic beam can be ignored due to the large length of the elastic beam.

3. Experiments and Analysis

It is known that the ferrofluid-based planar damper dissipates the vibration energy by the working element of ferrofluid drop covering the moving PM. The damping property is affected by the surface shape of ferrofluid drop between the moving PM and the inner walls of the container. The contact surface shape of the ferrofluid drop is developed by magnetostatic force, the gravity and the surface tension. The amount and height of the ferrofluid drop in the PM axial direction causes change of the magnetostatic force. So the damping coefficient c is discussed by changing the ferrofluids mass m_f and adjusting the container inner height h in the axial direction. In the experiment, different ferrofluids mass is added on the moving PM, and the container's inner height is measured, as shown in Table 1. The damping coefficient decreases with the increasing of the ferrofluids mass under the same the container inner's height of 11 mm as would be expected, but decreasing degree is small, varying from 0.1566 to 0.1563 when ferrofluids mass increases from 2.9 g to 3.8 g. The large magnetostatic force of the ferrofluid drops on the moving PM can improve the contact surface between the moving PM and inner wall of the container, and increases with the ferrofluids mass increasing. The ferrofluids drop mass of 2.9 g on the moving PM has attained the saturation value for the maximum magnetostatic force, and a larger ferrofluids drop mass only give samller contribution to the force, for example the ferrofluids drop mass of 3.8 g. In addition, Table 1 shows that the smaller the container's inner height, the greater the damping coefficient. When the ferrofluids drop is squeeze by reducing the container's inner height form from 12 mm to 10 mm with enough ferrofluids of 3.8 g, the contact surface shape of the ferrofluids drop changes largely and make the damping coefficient increase above 2.5 times.

Table 1. The dependence of the damping coefficient c on the container height h and the mass of ferrofluids m_f.

m_f \ h	10 (mm)	11 (mm)	12 (mm)
1.99 (g)	/	0.2393	/
2.90 (g)	/	0.1566	/
3.80 (g)	0.3766	0.1563	0.1531

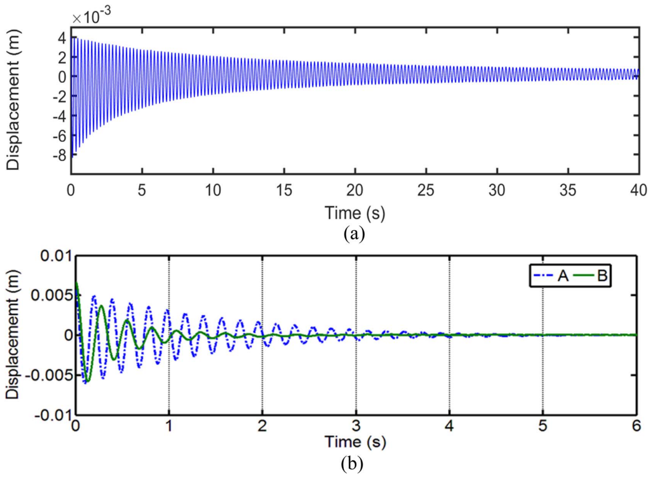

Fig. 3. (Color online) Experimental results of the system oscillations with the transient displacement for the cases (a) without ferrofluids damper and (b) with it. A and B refer to transient displacement of the system including the ferrofluids damper without external fixed PM and with it, respectively.

An experiment is developed to confirm the magnetic spring constant by using spiral test frame which is assembled with calibrated scale and digital push-pull gauge. It is known that the magnetic attraction of the magnetic spring is determined by the vertical distance of the two PMs d. When the vertical distance is 17 mm, the stiffness of the magnetic spring is 32 N/m and the resonance frequency is 4 Hz.

In the experiment, the system oscillations are conducted by a steel rule used as the elastic beam with a size of $214 \times 30 \times 1$ mm. The oscillation frequency and initial amplitude of the system are about 4 Hz and 6 mm, respectively. The excitation mass and the response mass are 87.76 g and 54.45 g, respectively. The dissipating element of the system is the damper with the ferrofluids drop of mass 3.8 g and the container's inner height of 11 mm.

Three cases are chosen to study the vibration damping effect of the system. The first is that the moving PM is blocked, which is to say the moving PM oscillates with the container together. The damping structure works as a normal single-degree-of-freedom system with the mechanical damping and stiffness of the steel rule. In another case, the moving PM is released and the damping structure shows two-degree-of-freedom system. The traditional ferrofluid-based damper without magnetic spring [5, 7] is used to be compared with the novel damper provided in this paper. Fig. 3. shows the experimental results of the system oscillations without damper and with different dampers. It can be seen that dampers provide an effective damping for the elastic beam oscillations and restrain the system vibration. By comparison, the logarithmic decrement is 0.019 without damper, 0.11 with traditional damper, and 0.6 with the novel damper. In addition, compared with the system without the damper, the system with the damper oscillates and delays more shortly. It takes about 4 seconds and 2 seconds for the traditional damper and the novel damper to dissipate the vibration energy, respectively, for the traditional damper with weak and disordered restoring force makes the relative displacement of the response mass mitigate and decrease its damping effect. However, the novel damper with magnetic spring can provide constant stiffness and obtain the maximum relative displacement under the resonance condition with high damping effect.

4. Conclusions

In summary, a novel ferrofluid-based planar damper with magnetic spring was introduced in this study, which is an extension of a traditional ferrofluid-based damper. Unlike traditional ferrofluids dampers whose restoring force is provide by the ferrofluids elastic force between the moving PM and the inner wall of the container, the novel ferrofluids damper works and uses magnetic attraction as the restoring force to obtain large relative displacement. This property makes the novel ferrofluids damper a particularly effective damping for dissipating the low frequency vibration energy. A prototype device was fabricated and the damping characters and behaviors were experimentally studied. According to experiments, it is known that the novel damper shows a high damping performance with a larger logarithmic decrement. Under the same initial vibration amplitude of beam, the system with the novel damper only needs a half time taken by the system with traditional damper to dissipate the vibration energy.

Acknowledgements

This work has been supported by the Fundamental Research Funds for the Central Universities (2016RC010), China.

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