Design of a Magnetic Negative-stiffness-damper for Vibration Isolation

Hangfei Zhou*, Shaorong Xieb and Yi Sun*

School of Mechatronic Engineering and Automation, Shanghai University, Shanghai, China

*Corresponding author e-mail: yisun@shu.edu.cn, 1HangfeiZhou@163.com,

bSrxie@shu.edu.cn

Abstract. This paper presents a novel low frequency vibration isolator. The isolator is mainly composed of a coil spring, two permanent magnets and a number of copper plates of different thickness. The negative stiffness provided by the two permanent magnets is offset by the positive stiffness of the spring, thus reducing the natural frequency of the system and broadening the isolating bandwidth. When the magnet moves, it creates eddy current damping on the copper plate. The movement of the permanent magnet and copper form a viscous damper. This makes up for the disadvantage that the isolator cannot dissipate the vibration energy. Principle of the novel vibration isolator is introduced. The optimum configuration of the vibration isolator was obtained by experiments.

1. Introduction

Vibration has been paid more and more attention in the field of industry. There are also many researches on vibration isolator, among which how to improve the performance of isolator is an eternal topic. The working condition of the vibration isolator is \( \omega/\omega_n > \sqrt{2} \), which means the vibration isolator works only when the vibration frequency is more than \( \sqrt{2} \) times of the natural frequency of the system. Based on the natural frequency of the system \( \omega_n = \sqrt{k/m} \), \( \omega_n \) can be reduced by reduce the stiffness \( k \) or increase the mass \( m \). However, excessive mass \( m \) will be dangerous. Therefore, reducing stiffness \( k \) is a reasonable way to reduce the natural frequency. In order to reduce the stiffness \( k \) of the vibration isolator, it can be realized by parallel connection of positive and negative stiffness mechanisms.

There have been many studies on negative stiffness mechanism in the past. Liu X et al. proposed a low-frequency vibration isolator composed of three springs [1-3], which uses two oblique springs to provide negative stiffness to offset the positive stiffness of the vertical spring. Carrella et al. proposed a negative stiffness mechanism composed of three permanent magnets [4, 5].

The vibration isolators mentioned above can only serve as vibration isolators base on the damping of the system itself is very small. When the damping is very small, the resonance peak value of the system is very large, and the isolators can’t dissipate the energy of vibration. When the vibration frequency is equal to the natural frequency of the system, dangerous resonance will occur. The instrument may be damaged seriously. Therefore, it is necessary to introduce an energy dissipation device.
Compared with other dampers, magnetic dampers have the advantages of no contact, no wear, no power supply and high reliability. Therefore, a magnetic eddy current damping device is added to the vibration isolator in this paper. Bissal proposed several single degree of freedom magnetic dampers [6-8], studied the damping effect of different topological structure of the magnet combination when moving in the copper tube. Choi proposed a radial electromagnetic damper with adjustable damping [9]. Babak et al. proposed a vibration isolator which mainly consists of two ring permanent magnets and a conductor plate [10].

In this paper, a novel low frequency vibration isolator is studied according to the principle of magnetic negative stiffness and the principle of magnetic damping. The best configuration of the vibration isolator is obtained by studying the influence of copper plate thickness and copper plate position on the damping performance.

2. Configuration of Vibration isolator

Fig. 1 exhibits the general assembly drawing of the vibration isolator in this paper. Its main components are: 1) load mass block, 2) linear bearing, 3) sleeve, 4) copper plate, 5) copper plate tray, 6) upper magnet, 7) coil spring, 8) lower magnet. The sleeve in this design has a grid structure, so the copper plate tray can be fixed in different positions of the sleeve to adjust the position of the copper plate in the system. In this paper, the thickness of copper plate can be changed by placing different amounts of copper plate in the middle of the upper and lower copper sheet tray. The upper magnet and the lower magnet are the same magnets, they are arranged in the form of mutual attraction and isolate by the coil spring. The vibration isolation function of the vibration isolator is realized by the magnets and the coil spring. The positive stiffness of the coil spring is offset by the negative stiffness provided by the two permanent magnets so that the low-frequency vibration isolation can be realized. The lower magnet is fixed at the bottom of the sleeve, and the upper magnet moves up and down with the load mass. When the conductor and the magnetic field move relative to each other, there will be eddy currents in the conductor, which will form an induced magnetic field. Based on Lenz's law, this induced magnetic field will generate a damping force to hinder the relative motion. Since there is a non-zero resistance on the conductor, based on Ohm's law, the relative motion energy will be dissipated in the form of Joule heat, thus dissipating the vibration energy and enhancing the damping effect. Since the magnetic field is not uniformly distributed in space, the eddy current is composed of two parts: a changing magnetic field that generates the eddy current and a conductor that moves in a static magnetic field that generates the eddy current.

![Figure 1. Assembly drawing of vibration isolator.](image)
The physical properties of the vibration isolator are shown in Table 1.

**Table 1.** Physical properties of the vibration isolator.

| Property                                      | Value          |
|-----------------------------------------------|----------------|
| Inside diameter of copper sheet               | 32 (mm)       |
| Outside diameter of copper sheet              | 70 (mm)       |
| Inside diameter of permanent magnet           | 10 (mm)       |
| Outside diameter of permanent magnet          | 30 (mm)       |
| Thickness of permanent magnet                 | 8 (mm)        |
| Magnetization of permanent magnet             | 1.07e6 (A/m)  |
| Stiffness of spiral spring                    | 2 (N/mm)      |
| Load quality                                  | 500 (g)       |
| Copper electrical conductivity                | 5.998e7 (S/m) |

Fig. 2 shows the simulation of the flux density distribution of the magnets. It can be seen that when the radius is greater than 30, the magnetic flux density becomes very small, and it is not significant to continue to increase the conductor radius. Therefore, the outer diameter of the copper plate designed in this paper is 35 mm.

![Figure 2. Magnetic flux density distribution of the magnets.](image)

Fig. 3 shows the force-gap curve of the two magnets, it can be seen that the slope of the curve is non-linear. Which means the magnetic negative stiffness is non-linear, the closer two magnets are, the greater the magnetic negative stiffness. However, the stiffness of the coil spring is constant value 2N/mm. In order to make the positive and negative stiffness offset to get a positive small stiffness, but not to make the negative stiffness too much cause the total stiffness of the system is always negative. When designing the device, we choose the magnetic gap of 20mm as the static equilibrium position. The original length of the coil spring is 25mm. Under the action of load, magnetic force and helical spring force, the magnetic spacing is 20mm in static balance. In this case, the stiffness of the system can be approximated as a small positive constant within the amplitude of ±4mm.

![Figure 3. Force-Gap curve of two magnets.](image)
3. Experiments

Fig. 4 shows the transmissability curves of isolators with different thickness of copper plates, and the curves 1-5 in the figure respectively represent the thickness of copper plates of 0, 8, 16, 24 and 32mm. When research the influence of copper plate thickness on the vibration reduction effect, the axial center position of the copper plate is parallel to the static equilibrium position of the upper magnet, as shown in Fig. 5 (1). The research results show that the natural frequency of the vibration isolator is 6.65Hz, which can attenuate any vibration greater than 10Hz. And the vibration suppression can be reduced to -27dB at 100Hz. As the thickness of the copper plate increased from 0 to 24mm, the resonant peak value decreased from 16dB to 5dB, which had a significant damping effect. However, when the thickness of copper plate reaches 24mm, the vibration reduction effect tends to be constant. Therefore, considering the economic factors and the factors of damping effect, the optimum thickness of the copper plate of the shock isolator designed in this paper is 24mm.

Figure 4. Transmissibility curves of isolators based on copper plates of different thickness.

Figure 5. Copper plate with thickness of 24 mm in different positions.

Fig. 6 shows the transmissibility curves of the copper plate at different positions when the thickness of copper plate is 24mm. The curves 1-4 respectively represent the positions of the copper plates in Fig. 5(1)-(4). According to the experimental results, the damping effect is obviously stronger when the alignment area between the copper plate and the upper magnet is larger. According to [8], when the magnet and the copper plate generate axial relative motion, the magnetic flux density of the magnet in
the axial direction does not contribute to the damping force, because the cross product of the magnet and the velocity is zero. Therefore, only the radial magnetic flux density works on the damping force. As shown in Fig. 2, the closer to the upper and lower outer edge of the magnet, the greater the radial flux density. This is why the greater the alignment area of the magnet and the copper plate, the greater the damping force. However, due to the limitation of its magnetic field range, the damping force won’t increase indefinitely with the thickness of the copper. In conclusion, the best position of the copper plate isolator is that the axial center position of the copper plate parallel to the static equilibrium position of the upper magnet.

![Figure 6. Transmissibility curves of the copper plate at different positions when the thickness of copper plate is 24mm.](image)

4. Conclusion
A novel low frequency vibration isolator is presented in this paper. The magnetic negative stiffness offset the positive stiffness of the helical spring, thus reducing the stiffness of the whole system. Lower stiffness means lower natural frequency, which can isolate lower frequency vibrations. The natural frequency of the vibration isolator can reach 6.65Hz. On the basis of the vibration isolator, the magnetic damping device is added to the novel isolator. The vibration energy is converted into Joule heat dissipation by using the eddy current damping principle, so as to enhance the damping effect and weaken the resonance peak. In this study, the influence of different thickness copper plates and their positions on the vibration reduction effect was studied by experimental method.

Acknowledgments
This work was financially supported by Shanghai Natural Science Foundation Project [grant numbers 17ZR1410200].

References
[1] Liu X, Huang X, Hua H, On the characteristics of a quasi-zero stiffness isolator using Euler buckled beam as negative stiffness corrector, J. Journal of Sound and Vibration. 332.14 (2013) 3359-3376.
[2] Carrella A, Brennan M J, Waters T P, Static analysis of a passive vibration isolator with quasi-zero-stiffness characteristic, J. Journal of Sound and Vibration. 301.3-5 (2007) 678-689.
[3] Xu D, Yu Q, Zhou J, et al, Theoretical and experimental analyses of a nonlinear magnetic vibration isolator with quasi-zero-stiffness characteristic, J. Journal of Sound and Vibration.
332.14 (2013) 3377-3389.

[4] Carrella A, Brennan M J, Waters T P, et al, On the design of a high-static–low-dynamic stiffness isolator using linear mechanical springs and magnets, J. Journal of Sound and Vibration. 315.3 (2008) 712-720.

[5] Dong G, Zhang X, Xie S, et al, Simulated and experimental studies on a high-static-low-dynamic stiffness isolator using magnetic negative stiffness spring, J. Mechanical Systems and Signal Processing. 86 (2017) 188-203.

[6] Bissal A, Salinas E, Magnusson J, et al, On the Design of a Linear Composite Magnetic Damper, J. IEEE Transactions on Magnetics. 51.11 (2015) 1-5.

[7] Ebrahimi B, Khamesee M B, Golnaraghi F, Permanent magnet configuration in design of an eddy current damper, J. Microsystem Technologies. 16.1-2 (2010) 19-24.

[8] Bae J S, Hwang J H, Park J S, et al, Modeling and experiments on eddy current damping caused by a permanent magnet in a conductive tube, J. Journal of Mechanical Science and Technology. 23.11 (2009) 3024-3035.

[9] Choi J Y, Jang S M, Analytical magnetic torque calculations and experimental testing of radial flux permanent magnet-type eddy current brakes, J. JOURNAL OF APPLIED PHYSICS. 111.7 (2012) 2147483647-0.

[10] Babak E, Mir B K, M. F G, Design and modeling of a magnetic shock absorber based on eddy current damping effect, J. Journal of Sound and Vibration. 315 (2008) 875–889.