Adaptive Tuned Vibration Absorber based on Magnetorheological Elastomer

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ABSTRACT: This study presents an adaptive tuned vibration absorber (ATVA) which is based on magnetorheological elastomer (MRE). Traditional dynamic absorber has limited its application and vibration absorption capacity for its narrow working frequency bandwidth. MRE is a kind of smart material whose modulus can be controlled by applied magnetic field. Based on MREs, an ATVA which works on shear mode is proposed in this study. After the vibration mode shapes of the ATVA are analyzed, the mechanical structure of the ATVA is brought forward. Then the magnetic circuit of the ATVA is identified by ANSYS software. By using a modified dipole model, the shift-frequency properties of the ATVA versus magnetic field and strains are theoretically analyzed and simulated. Furthermore, by employing a beam with two ends supported, its shift-frequency property and vibration absorption capacity are experimentally justified. The experimental results demonstrate that the designed ATVA has better performance than traditional passive absorber in terms of frequency-shift property and vibration absorption capacity.

Key Words: magnetorheological elastomer, vibration absorber, semi-active.

INTRODUCTION

MOST of the mechanical, civil, and construction systems encounter undesirable vibrations. To avoid damage, fatigue failure, and to increase the system lifetime, many researches have been done to attenuate the unwanted vibration. Dynamic vibration absorber (DVA) invented by Frahm (1909) is a classic solution to suppress vibrations of machines and structures. Traditional DVA is typically composed of an oscillator, a spring element, and a damping element. However, this kind of passive absorber theoretically brings the object base to rest at a single excitation frequency, the resonance frequency of the DVA. For many practical systems, the working condition may change over time and cause the absorber to become inefficient and potentially aggravate the base vibration. Adaptive tuned vibration absorber (ATVA) is developed to solve this problem, which is a device similar to DVA but holds adaptive element to trace the working condition. Generally, ATVA can vary its natural frequency to track uncertain or time-varying excitation frequencies by employing the adaptive spring element which is able to alter its stiffness. According to the method of changing the stiffness, the ATVAs can be sorted into two major groups: one is to use variable geometries (Walsh and Lamancusa, 1992; Li et al., 2005) and the other is to use smart materials (Flatau et al., 1998; Williams et al., 1999; Davis and Lesieutre, 2000).

Magnetorheological elastomers (MREs) are smart materials where polarizable particles are suspended in a non-magnetic solid or gel-like matrix. Typically, magnetic fields are applied to the polymer composite during cross-linking so that chainlike structures can be formed and fixed in the matrix after curing. The unique characteristic of MRE is that its shear modulus can be controlled by the external magnetic field rapidly, continuously, and reversibly (Shen et al., 2004; Gong et al., 2005). Such properties make MREs promising in many applications, such as ATVAs, stiffness tunable mounts and suspensions, and variable impedance surfaces. Watson (1997) applied a patent using MREs for a suspension bushing. Later, Stewart et al. (1998) and Ginder et al. (2000) constructed and tested tunable automotive mounts and bushings based on MR elastomers. Ginder et al. (2001) did pioneer work on the development of an adaptive tunable vibration absorber (ATVA) using MREs. Koo et al. (2003) at Virginia Tech University used magnetorheological dampers in semi-active tuned vibration absorbers to

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1045-389X/07/12 1205–6 $10.00/0 DOI: 10.1177/1045389X07083128

Los Angeles, London, New Delhi and Singapore

control structural vibrations. Lerner and Cunefare (2007) examined MREs placed in three different absorber configurations: shear, longitudinal, and squeeze modes and found the squeeze mode device exhibited the largest frequency shift of 507%.

The objective of this study is to investigate the application of MREs to adaptive tuned vibration absorber. The vibration mode shapes have been analyzed to ensure that the ATVA works by shear mode. Another problem to be confirmed is the magnetic circuit; a closed C-shape structure has been identified by ANSYS software. Furthermore, the shift-frequency properties of the ATVA versus magnetic field and strains are analyzed and simulated by using the modified magnetic dipole model. It is also experimentally verified together with its attenuation ability.

MATERIAL PREPARATION

The MRE materials consist of natural rubber as a matrix, additives, and carbonyl iron particles with a size of 3–5 μ. At the first step of the fabricating process, all ingredients are thoroughly mixed by mixing roll and then the mixture is compressed into a mold and placed in a self-developed magnet-heat coupled device. During the pre-cured stage, the particles are magnetized and form chains aligned along the magnetic field direction. Thirty minutes later, the magnetic field is turned off and the temperature is raised to 153°C to finish the sulfuration. The MRE properties under various magnetic fields are evaluated by a modified dynamic mechanical analyzer (DMA) from the Triton Co. It can be seen from Figure 1 that the shear modulus shows an increasing trend with magnetic field intensity. However, the increasing slope decreases with the increment of magnetic fields, which is due to the magnetic saturation. In addition, the magneto-induced modulus also shows an increasing trend with the decreasing strain while the zero-field modulus is nearly the same with the strain change.

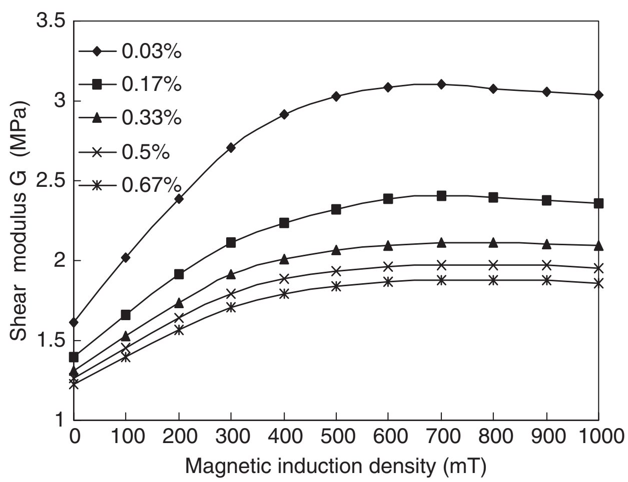

Figure 1. *Shear modulus with magnetic field under different strains load.*

DESIGN OF ATVA

Simulation of Mode Shapes

This study focuses on the ATVA which works by shear mode. So the first step of the design is to analyze the vibration mode shape of the assumed ATVA and make sure the work mode of ATVA is shear. To this end, ANSYS software is employed and a popular sandwich structure is simulated. In this structure, dynamic mass is at the center and through MREs connecting the static mass. The results of vibration mode shapes analyses are shown in Figure 2. Figure 2(a) shows that the first mode shape of ATVA is swing while Figure 2(b) shows the second mode is shear in plumb direction. It means that the sandwich structure with this shape would easily swing instead of shear. To avoid this happening, some structures to make the ATVA work by shear mode should be

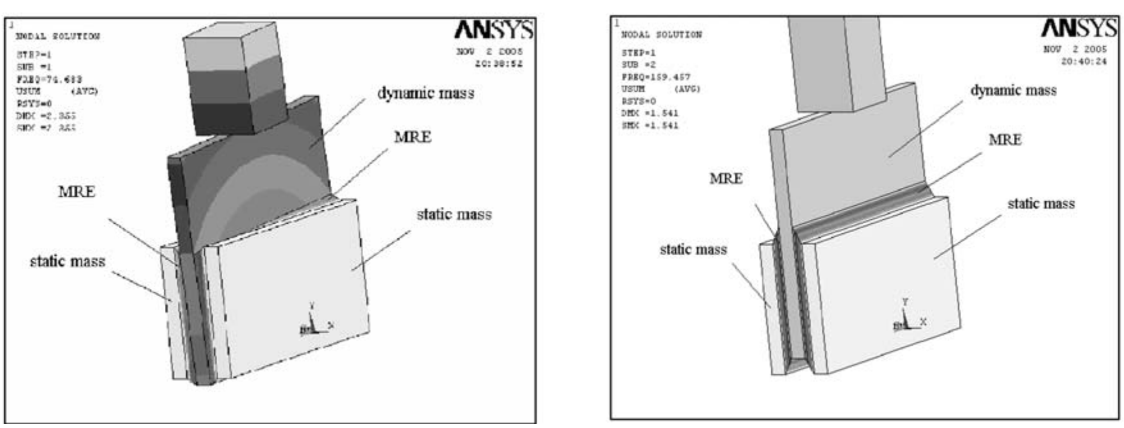

Figure 2. *Mode shapes of sandwich structure.*

leaded in, such as guiderod, sliding chute, and linear bearing.

ATVA Structure

The efficient component for ATVA to absorb vibration is its oscillator or dynamic mass. As shown in Figure 3, the developed ATVA consists of three main parts: dynamic mass, static mass, and smart spring element with MREs. The electromagnets and magnetic conductors form a closed C-shape magnetic circuit, which are assembled at the mounting shell. These components together construct the dynamic mass. This development makes the ATVA more compact and more efficient because no additional oscillator and nearly most components can be considered as dynamic mass. The static mass consists of the shear plate and the base. Through the shear plate, the smart spring elements (MREs) connect the dynamic mass and the static mass. Furthermore, as analyzed above, a guiderod and a linear bearing are employed to ensure the ATVA works by the shear mode.

Magnetic Circuit Analyses

Magnetic conductor which forms a magnetic circuit with MREs is the vital component of the ATVA based on MREs. As shown in Figure 4, for shear mode, the direction of magnetic field is parallel to the direction of particle chains while the shear force is perpendicular to the particle chains. The purpose of analyzing the magnetic circuit is to optimize the turn number of coils, the circuit, and the distribution of magnetic lines of flux, etc. The avail tool of the magnetic circuit analyses is ANSYS software. Figure 5 is a result of the designed ATVA. For a coil with 2200 rounds, when the applied current is 0.5A, the magnetic field through MRE can reach 0.9 T, which is enough to utilize as shown in Figure 1.

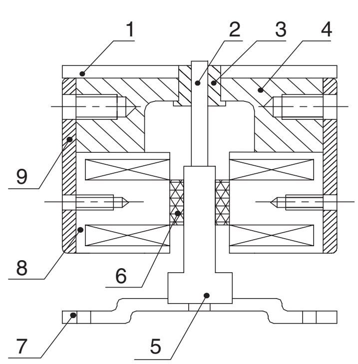

Figure 3. *The scheme of designed ATVA. 1. Cover; 2. Guiderod; 3. Linear bearing; 4. Magnetic conductor; 5. Shear plate; 6. MREs; 7. Base; 8. Electromagnet; 9. Mounting shell.*

THEORETICAL ANALYSIS OF SHIFT-FREQUENCY PROPERTY

The MRE's shear modulus G consists of two terms, zero-field modulus G_0 and magneto-induced modulus ΔG_d:

$$G = G_0 + \Delta G_d. \tag{1}$$

The natural frequency of the ATVA can be expressed as:

$$f = f_0 + \Delta f \tag{2}$$

where f_0 is the initial natural frequency and Δf is the magneto-induced frequency:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{G_0 A}{mh}} \tag{3}$$

$$\Delta f = \frac{1}{2\pi}\sqrt{\frac{G_0 \times A}{m \times h}} \bullet \left(\sqrt{1 + \frac{\Delta G_d}{G_0}} - 1\right). \tag{4}$$

From Equation (3), the initial resonant frequency of the ATVA can be designed to match the primary system. Equation (4) reveals that the frequency-shift capacity is not only related to the MR effect but also related to the initial shear modulus. Large initial modulus with the same MR effect will cause wide frequency-shift bandwidth. To theoretically analyze the frequency-shift property with magnetic field, a modified magnetic dipoles model considering the interaction between all particles at the same chain is employed. The model of a single chain is shown in Figure 6, where d is the distance

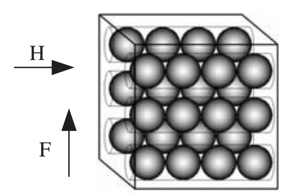

Figure 4. *The directions of magnetic field and shear force.*

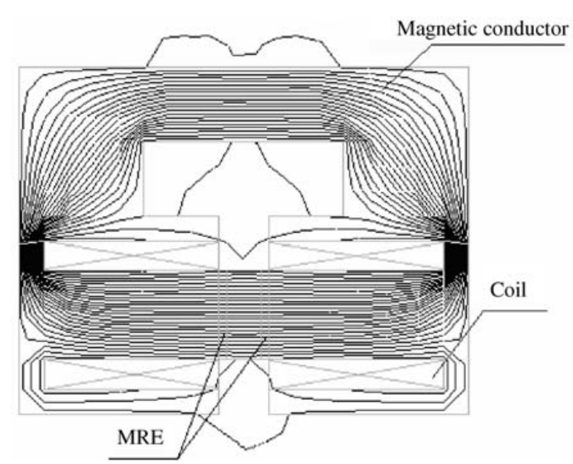

Figure 5. *The magnetic circuit of a C-shape structure.*

between two neighbor particles before deformation, and θ is the shear angle. For small deformation, $\theta \to 0$, the magneto-induced modulus can be expressed as:

$$\Delta G = 3\phi\mu_m\mu_0\beta^2 H_0^2\left(\frac{R}{d}\right)^3\zeta\left(\left(\frac{10}{A^2}+\frac{2}{B^2}\right)+\frac{48\beta\zeta}{A^3}\left(\frac{R}{d}\right)^3\right) \tag{5}$$

where 3β is the effective susceptibility and it approaches 3 for ferromagnetic particles, μ_0, μ_m correspond to the permeability in vacuum and the relative one of the matrix respectively, φ is the volume fraction of particles and R is the radius of the particle, $A = 1 - 4\beta\cos^3\theta(R/d)^3\zeta$, $B = 1 + 2\beta\cos^3\theta(R/d)^3\zeta$ and $\zeta = \sum_{k=1}^{\infty} 1/k^3 \approx 1.202$. Defining the shear strain to be ε, the magneto-induced modulus can be represented as:

$$\Delta G = 3\phi\mu_m\mu_0\beta^2 H_0^2\left(\frac{R}{d}\right)^3 \times \zeta\left(\frac{12a^7(1+2ba^3+14b^2a^6-8b^3a^9)}{(1-4ba^3)^3(1+2ba^3)^2}\right) \tag{6}$$

where $a = (1+\varepsilon^2)^{-1/2}, b = \beta\zeta(R/d)^3$. Obtained from Equation (5), Figure 7 shows that the natural frequency of ATVA has increasing trend with magnetic field. In the initial stage, natural frequency accelerates and then the rate of slope drives to stabilization before saturation. From Equation (6), the influence of shear strain to the shift-frequency property can be obtained and is shown in Figure 8 where the magnetic field is fixed as 800 KA/m. The frequency decreases while the strain increases and gets stabilized when shear strain is 100%.

It can be seen that strain influences the shift-frequency property so hugely that the amplitude of the ATVA must be evaluated. The model of ATVA coupled with primary system can be simplified as Figure 9, where m_1 is the mass of primary system, k_1 is the stiffness of primary system, ω_1 is the natural frequency of primary system, and x_2 is the vibration amplitude of primary system, m_2 is the mass of ATVA, k_2 is the stiffness of ATVA, ω_2 is the natural frequency of ATVA, and x_2 is the vibration amplitude of ATVA. When external excitation is $p_1 sin\omega t$, the complex amplitude $\bar{x}_2$ can be

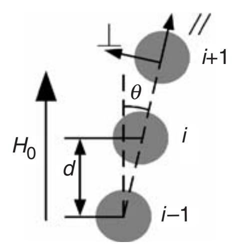

Figure 6. *The model of particle chain.*

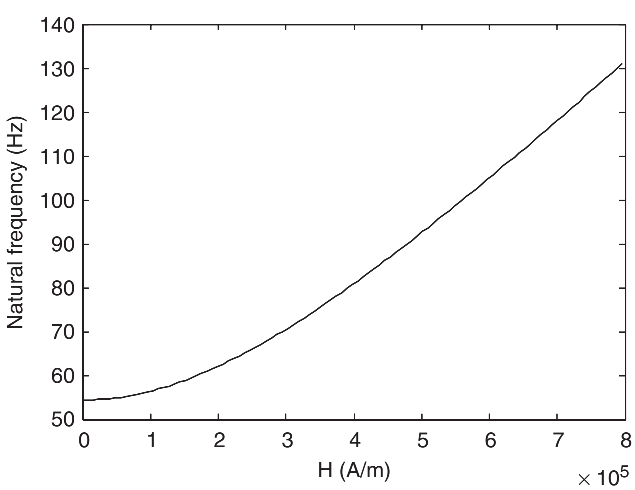

Figure 7. *Natural frequencies versus magnetic field.*

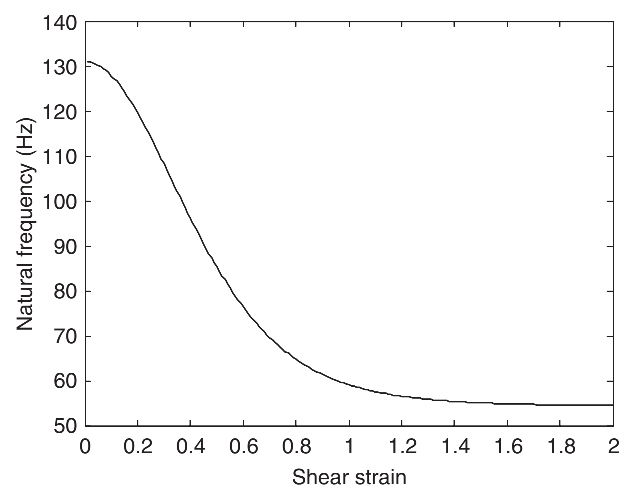

Figure 8. *Natural frequencies versus shear strain.*

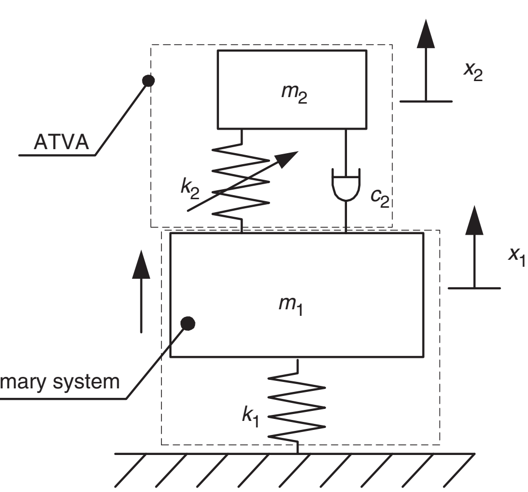

Figure 9. *Model of ATVA with primary system.*

expressed as:

$$\bar{x}_2 = \frac{p_1(k_2 + ic\omega)}{-k_2^2 + ic\omega(m_1\omega_1^2 - (m_1/m_2)k_2 - k_2))} \quad (7)$$

The maximum shear displacement B_2 is:

$$B_2 = \delta_{st} + |x_2| = \frac{g}{\omega^2} + |x_2| \quad (8)$$

where δ_{st} is the static deflection of ATVA. Figure 10 is a classic example of calculated shear displacement with and without static deflection. The shear displacement without static deflection is relatively small compared to the displacement with static deflection. It means that the static deflection has vast contribution to the shear displacement especially in the low frequencies. So limiting the static deflection is a method to enhance the tuned frequency band. It also can be seen that the shear displacement decreases sharply with the increasing excitation frequency. When the excitation frequency is higher than 40 Hz in this example, the shear displacement trends to a stable level and has little influence on tuned frequency of ATVA. Thus, ATVA working with MRE should work at relatively high frequency band, because the stability of the shift-frequency property would have problems at low frequency band.

EXPERIMENTAL EVALUATION OF ATVA

Frequency-shift Property

To investigate the frequency-shift capability of the MREs a beam with both ends supported is employed. The developed ATVA is placed at the center of the beam. White noise excitation applied to the beam is generated by an exciter. Two accelerometers are placed on the oscillator and the base beam to measure their responses, respectively. The measured signals are sent to the dynamic signal analyzer. The transfer function can be obtained by using FFT analysis and the testing results are shown in Figure 11. It shows that the curves of transfer function move rightward with the increment of the magnetic field. The resonance frequency can be obtained by reading the peak-to-peak value. It is shown in Figure 12. Its trend agrees well with the theoretical results. The resonance frequency increases from 41 Hz at 0 A to 63.75 Hz at 0.6 A and its relative frequency change is as high as 155%.

Vibration Attenuation

Compared with the frequency-shift experiment, the difference of system to evaluate vibration absorption is that an impedance head is induced. The impedance head connects with the beam and the exciter. The exciter provides sinusoidal excitation to the beam with a linear frequency scan. The base point impedance with various magnetic fields can be measured by the dynamic signal analyzer. To evaluate the vibration absorption capacity, a ratio γ of base point admittance H_A with TVA and that H_o without TVA is employed to reflect the effect of vibration absorption:

$$\gamma = 20\lg(|H_A/H_O|) \quad (9)$$

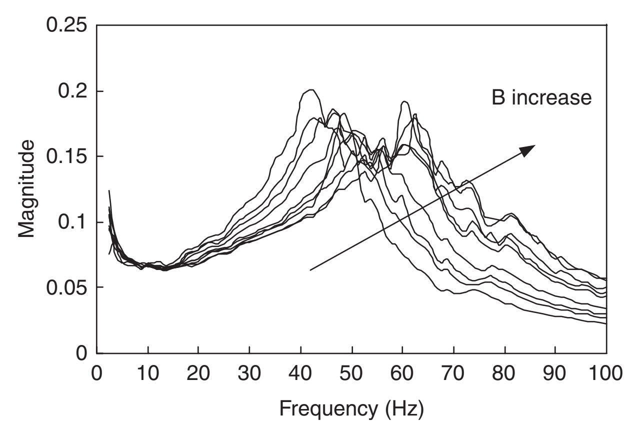

Figure 11. *The transfer function at various magnetic fields.*

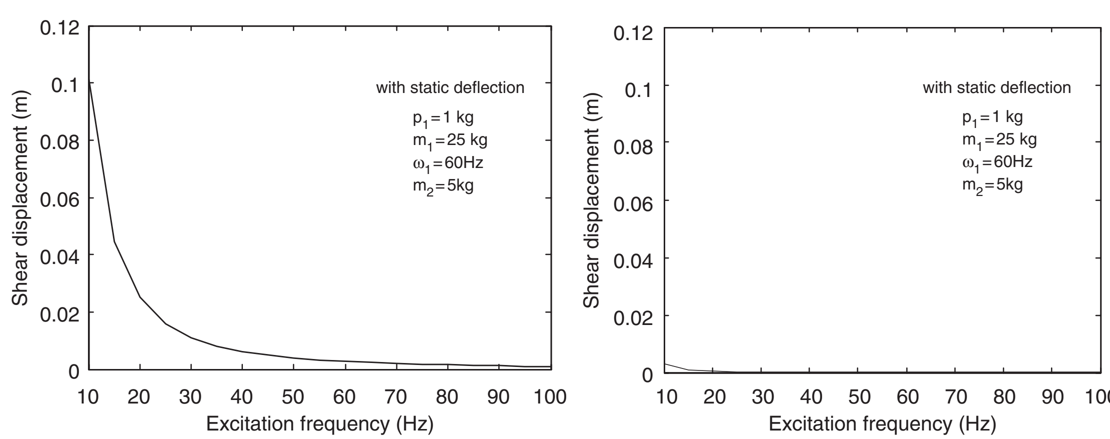

Figure 10. *The maximum shear displacement with and without static deflection.*

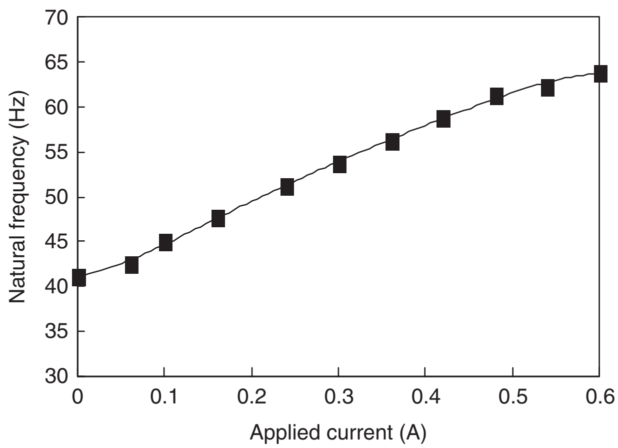

Figure 12. *The relationship between applied current and resonance frequency.*

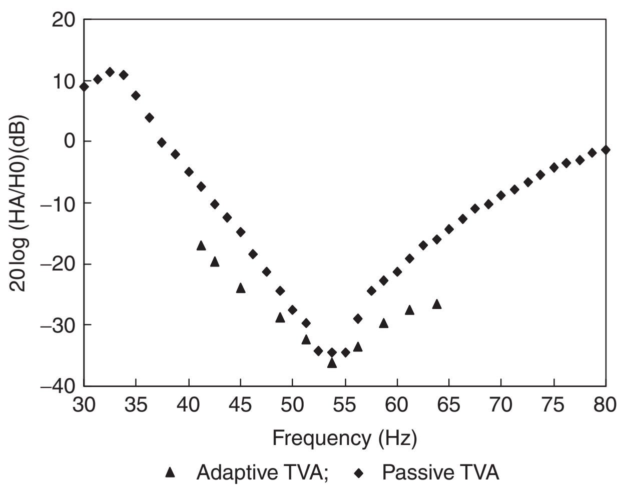

Figure 13. *Comparison of vibration attenuation between ATVA and TVA.*

The comparison of vibration absorption capacity between the ATVA and the passive DVA is shown in Figure 13. The upper data are the results of passive DVA which is realized by fixing the current of ATVA and the lower data are the results of ATVA whose frequency is tuned to trace the excitation frequency. For the passive DVA, the best vibration attenuation efficiency occurs at the natural frequency of the primary system. The effect becomes worse sharply while the excitation frequency is apart from beam natural frequency and a new resonance hump occurs at 34 Hz. For ATVA, in tunable frequency band, it has better absorbing effects than passive DVA.

CONCLUSIONS

In this study, a shear mode ATVA based on MRE was designed, simulated, and evaluated. After the vibration mode shapes of the ATVA are analyzed and the magnetic circuit of the ATVA is identified by ANSYS software, a compact and efficient ATVA is proposed. Furthermore, the shift-frequency properties of the ATVA versus magnetic field and strains are analyzed and simulated by using the vibration theory and the modified magnetic dipole model. It is also experimentally verified together with its attenuation ability. The experimental results demonstrate that the designed ATVA has better performance than traditional passive absorber in terms of frequency-shift property and vibration absorption capacity.

ACKNOWLEDGMENTS

Financial support from NSFC (Grant No. 10672154) and SRFDP of China (Project No. 20050358010) is gratefully acknowledged. Scholarship BRJH funding of Chinese Academy of Sciences is also appreciated.

REFERENCES

Davis, C.L. and Lesieutre, G.A. 2000. "An Actively Tuned Solid-State Vibration Absorber Using Capacitive Shunting of Piezoelectric Stiffness," *Journal of Sound and Vibration*, 232(3):601–617.

Flatau, A.B., Dapino, M.J. and Calkins, F.T. 1998. "High Bandwidth Tenability in a Smart Vibration Absorber," *Proceedings of SPIE*, 3327:463–473.

Frahm, H. 1909. "Device for Damping Vibrations of Bodies," US Patent, No. 989958.

Ginder, J.M. and Nichols, M.E., Elie, L.D. and Clark, S.M., 2000. "Controllable-Stiffness Components Based on Magnetorheological Elastomers," *Proceedings of SPIE*, 3985:418–425.

Ginder, J.M., Schlotter, W.F. and Nichols, M.E. 2001. "Magnetorheological Elastomers in Tunable Vibration Absorbers," *Proceedings of SPIE*, 4331:103–110.

Gong, X.L., Zhang, X.Z. and Zhang, P.Q. 2005. "Fabrication and Characterization of Isotropic Magnetorheological Elastomers," *Polymer Testing*, 24(5):669–676.

Koo, J.-H., Ahmadian, M. and Setareh, M. 2003. "Experimental Evaluation of Magnetorheological Dampers for Semi-active Tuned Vibration Absorbers," *Proceedings of SPIE*, 5052:83–91.

Lerner, A.A. and Cunefare, K.A. "Performance of MRE-based Vibration Absorbers," *Journal of Intelligent Material Systems and Structures* OnlineFirst, 05 May 2007 (doi:10.1177/1045389X07077850).

Li, J.F., Gong, X.L., Zhang, X.Z., Xu, Z.B. and Zhang, P.Q. 2005. "The Research of Dynamic Properties of Adaptive Tuned Vibration Absorber," *Chinese Journal of Experimental Mechanics*, 20(4):507–514.

Shen, Y., Golnaraghi, M. F. and Heppler, G. R. 2004. "Experimental Research and Modeling of Magnetorheological Elastomers," *Journal of Intelligent Material Systems and Structures*, 15(1):27–35.

Stewart, W.M., Ginder, J.M., Elie, L.D. and Nichols, M.E. 1998. "Method and Apparatus for Reducing Brake Shudder," US Patent, No.5,816,587.

Walsh, P.L. and Lamancusa, J.S. 1992. "A Variable Stiffness Vibration Absorber for Minimization of Transient Vibrations," *Journal of Sound and Vibration*, 158(2):195–211.

Watson J.R. 1997. "Method and Apparatus for Varying the Stiffness of a Suspension Bushing," US Patent, No.5,609,353.

Williams, K.A., Chiu, G.T. and Bernhard, R.J. 1999. "Passive-adaptive Vibration Absorbers Using Shape Memory Alloys," *Proceedings of SPIE*, 3668:630–641.