A broadband frequency-tunable dynamic absorber for the vibration control of structures

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Abstract. A passive-type dynamic vibration absorber (DVA) is basically a mass-spring system that suppresses the vibration of a structure at a particular frequency. Since the natural frequency of the DVA is usually tuned to a frequency of particular excitation, the DVA is especially effective when the excitation frequency is close to the natural frequency of the structure. Fixing the physical properties of the DVA limits the application to a narrowband, harmonically excited vibration problem. A frequency-tunable DVA that can modulate its stiffness provides adaptability to the vibration control device against non-stationary disturbances. In this paper, we suggest a broadband frequency-tunable DVA whose natural frequency can be extended by 300% to the nominal value using the magnetorheological elastomers (MREs). The frequency adjustability of the proposed absorber is first shown. The real-time vibration control performance of the frequency-tunable absorber for an acoustically excited plate having multiple resonant peaks is then evaluated. Investigations show that the vibration of the structure can be effectively reduced with an improved performance by the DVA in comparison to the conventional passive-type absorber.

1. Introduction

In mechanical systems, vibrations should be reduced to avoid structural failure and to provide safety and comfort to humans [Mohan, 2003]. Moreover, the induced vibrations are transmitted to other parts of a structure and cause unexpected noise problems. A dynamic vibration absorber (DVA) can be used for reducing vibrations in structures. A passive-type DVA is especially effective when the excitation frequency is close to the natural frequency of the absorber. However, fixing the physical properties of the DVA, e.g., the mass and spring constant, may limit the application of the device to narrowband, harmonically excited problems. Any mechanical component whose stiffness can be modulated can provide adaptability to the DVA against non-stationary disturbances [1–5], but the implementation of such adaptability might be a complex process.

Studies have been conducted on the development of smart systems employing functional materials in association with the semi-active control schemes. By changing system parameters such as the spring constant and the damping coefficient of the material itself, the dynamic response of the system can be changed. Since this strategy is based on the passive constitution of devices, the system is basically stable. Similar control performance can be achieved in the case of semi-active control. Typical examples of semi-active materials include magnetorheological fluids (MRFs), in which micrometer-sized ferromagnetic particles are dispersed within the carrier fluid. By applying an external magnetic field,
the yield stress in the shear direction can be changed so that the resultant apparent viscous property is also changed. The fluid property changes within a few milliseconds of exposure to a magnetic field of relatively low strength, and hence, the development of mechanical devices such as dampers, brakes, and clutches has gained considerable attention [6].

The magnetorheological elastomer (MRE), which is an elastomeric composite incorporating ferrous particles, was developed recently for overcoming problems such as sedimentation and aggregation of particles associated with MRF. Under the influence of a magnetic field, magnetically polarized particles form chain-like structures according to the inter-particle attractive forces, resulting in changes in the apparent viscoelasticity of the elastomer. The fundamental features of MREs are well reported, and attempts have also been made to enhance the changes in their viscoelasticity [7–14]. Other studies dealt with the application of MREs directly to a system as a controllable element for semi-active frequency-tunable DVAs [15–18]. However, these investigations appear to be limited to the fundamental properties of the absorbers alone, such as the tunable frequency bandwidth, and there is a lack of extensive studies on their use and on the damping performance when combined with the primary structure.

In the present study, we suggest a broadband frequency-tunable DVA whose property can be extended over a wide frequency range using the MREs. The frequency adjustability of the proposed absorber is first shown. The real-time vibration control performance of the frequency-tunable absorber for an acoustically excited plate having multiple resonant peaks is then evaluated. Investigations show that the vibration of the structure can be effectively reduced with an improved performance by the DVA in comparison to the conventional passive-type absorber.

### 2. An MRE-based Variable-stiffness DVA

#### 2.1 Specifications of the variable-stiffness DVA

A frequency-tunable DVA was developed using the MREs as variable stiffness elements. The schematic and the picture of this DVA are presented in Fig. 1. The damper consisted of a set of steel cores, a brass, a magnetic coil, two MREs, and a duralumin housing. A wire of diameter 0.7 mm was used to wind the coil in 600 turns. By applying an electric current to the coil, the steel cores formed a closed magnetic path for the generated flux. In addition to the steel core located at the center of a moving mass, concentrically-located brass ring worked as an auxiliary mass. Two ring-shaped MREs are located in between two concentric steel rings at upper and lower parts of the DVA, and their stiffness was variable owing to the effect of magnetic flux penetration.

![Fig.1 Schematic and a picture of the variable stiffness DVA](image-url)
2.2 Measurement of the frequency-shift property of the absorber

The variation characteristics of the natural frequency and the damping ratio against the applied magnetic field for the tunable dynamic absorber were evaluated. Schematic of the experimental setup is shown in Fig 2. A swept-sine signal whose frequency varied from 0 to 800 Hz was used to excite the absorber. Accelerations at the base and at the absorber mass were measured by two accelerometers, and a transfer function was calculated using a fast Fourier transform (FFT) analyser for each applied current varying from 0 to 3.2 A by 0.2 A. The natural frequency and the damping ratio were then read from the response curve. The damping ratio $\zeta$ was estimated according to the half power bandwidth method, for which the ratio was calculated as follows.

$$\zeta = \frac{\Delta f}{2f_0}$$  \hspace{1cm} (1)

In Eq. (1), $f_0$ represents a peak frequency of the transfer function, and $\Delta f$ a bandwidth between two points on the curve at which the response magnitude dropped -3dB from the peak value.
The natural frequency shift property and the damping ratio measurement results are shown in Fig. 3. From the figure, it is shown that the baseline frequency of 60 Hz in the absence of the magnetic field was extended to 250 Hz for a 3 A current. The magnification of frequency change in this case was more than 300%. On the other hand, the observed damping ratio tended to decrease with the applied current. However, we should note that the estimated damping ratio values are not accurate, since a reliable damping ratio value is thought to be less than 0.1 in the half power bandwidth method. Such a high damping value is attributed to the viscous property of the host elastomer. Realization of the consistent lower damping value is whatsoever preferable for the dynamic absorber to work effectively for mitigating vibration of the primary structure.

3. Experimental investigation of the damping performance of the MRE-based DVA

3.1 Experimental setup of the plate vibration system with DVA

The performance of the proposed frequency-tunable DVA is investigated for the vibration control of a multi degree-of-freedom system. A schematic of the experimental setup is shown in Fig. 4. An acrylic plate having dimensions of 390 mm × 490 mm × 8 mm was clamped at the top of a rigid box. The box was made of duralumin plates with thickness 20 mm, and the inner dimension of the box was 350 mm × 450 mm × 500 mm. The plate was excited acoustically by a loudspeaker placed inside the box; the swept-sine excitation signal was generated by a function generator. Accelerations of both the plate and the dynamic absorber were measured by accelerometers. The signals obtained through these sensors were processed by a fast Fourier transform (FFT) analyser as frequency-dependent acceleration.

The system responses of the swept-sine excitation of frequency varying from 50 Hz to 500 Hz at the speed of 4 Hz/s were first obtained for the cases in which the stiffness properties were fixed for applied current between 0 A and 3 A. Furthermore, the response was also measured when the natural frequency of the absorber was changed automatically in real time according to the detected instantaneous disturbance frequency.

![Fig. 4 Experimental setup of a multi degree-of-freedom system with the DVA.](image-url)
3.2 A controller design for a dynamic absorber tuned in real time

As shown in Fig. 5, the natural frequency of the adjacent system is tuned adaptively to match the frequency of a harmonic external force within the variable range, whereas the damper property is fixed respectively for the lower and upper outer bounds at the minimum or the maximum stiffness value. The switching rule for the stiffness value $k_{\text{MRE}}$ is defined as follows [19].

$$
k_{\text{MRE}} = \begin{cases} 
  k_{\text{min}} & f < f_{\text{min}} \\
  k_{\text{var}} & f_{\text{min}} \leq f \leq f_{\text{max}} \\
  k_{\text{max}} & f_{\text{max}} < f 
\end{cases}
$$

In Eq. (5), $k_{\text{min}}$ and $k_{\text{max}}$ signify the minimum and the maximum stiffness values of an MRE, $f_{\text{min}}$ and $f_{\text{max}}$ the lower and the upper bounds of tunable frequency range, and $f$ the detected disturbance frequency.

3.3 Experimental results and discussions

Frequency response of the primary system measured when the absorber property was fixed are shown in Fig. 6. Curves correspond to the cases in which the applied electric current were fixed, and the symbols indicate the amplitudes at the anti-resonance node; these amplitudes were extracted from the property-fixed curves within the variable frequency range. Below and above this range, the plots were extracted directly from the response curves when the applied current values were fixed at 0 A and 3 A, assuming that the system response could not be changed. Fig. 6 shows that both the amplitude and frequency of the peaks in the curve vary with the changes in elastomer stiffness. The anti-resonance
node can be observed as a local minimum, which moves toward the higher frequency with the increase in current. The plots indicate that the vibration response can be minimized consistently for a wide frequency range, by automatically tuning the natural frequency of the absorber to the disturbance frequency.

Under the same swept-sine excitation, force transmission was mitigated by real-time tuning of the absorber. Based on the acceleration signal measured by the accelerometer attached to the primary system, the instantaneous disturbance frequency was predicted by the controller. According to the stiffness switching rule shown in Fig. 5, the generated control signal was then fed into the DC power supply so that appropriate electric current corresponding to the target natural frequency could be produced.

Fig. 7 shows the acceleration response of the primary system corresponding to the real-time tuning of the absorber in frequency domain. In the figure, the response of the primary system without the absorber, the responses when the absorber property was fixed at the stiffness corresponding to the applied current of 0 A and 3 A, and also the predicted response plot when the absorber is tuned at each harmonic disturbance frequency are shown. The figure shows that the primary system response obtained by the absorber tuned in real-time is similar to the curve and the minima that are extracted manually from each property-fixed response curve. The experiment verified that the vibration can be minimized by the frequency-tunable absorber within the variable range. The fact is also supported by the comparison of the time histories of the primary system displacement shown in Fig. 8, where an increase of the amplitude near the resonance is constantly suppressed by the tuned absorber.
4. Conclusions

In this study, a stiffness-controllable elastomer, known as MRE, was used in a DVA whose natural frequency could be tuned by means of the applied magnetic field strength. The DVA was used to mitigate the vibration of the structure in conjunction with a real-time stiffness-switching algorithm. Frequency-shift measurement showed that the absorber had the large frequency variation, where the nominal frequency of 60 Hz was extended to 300 Hz by applying an electric current of 3 A. The damping ratio tended to decrease with the increase of the magnetic field. The damping performance the absorber for an acoustically-excited plate was then evaluated experimentally. The swept-sine excitation results of the system demonstrated that the plate vibration could be reduced effectively using real-time, frequency-tunable dynamic absorbers rather than using passive-type absorbers for a wide frequency range. The proposed tunable DVA with MREs can be used for reducing vibrations by a simple alteration of the stiffness element in the conventional passive-type absorber. The improvement of a material damping property of the MRE itself could further enhance the mitigation performance of the absorber.

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