A SEMI-ACTIVE DYNAMIC VIBRATION ABSORBER BASED ON MAGNETO-RHEOLOGICAL ELASTOMER FOR SHIP EQUIPMENT: EXPERIMENT AND ANALYSIS

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Magnetorheological materials are a class of smart materials whose rheological properties may be rapidly varied by a prescribed magnetic field. In this paper, a new shear-mode semi-active dynamic vibration absorber (SDVA) is designed based on the stiffness variable characteristics of magnetorheological elastomers. Semi-active vibration absorber is applied for a marine gearbox and the natural frequency-shift performance of the absorber is tested. The natural frequencies of the SDVA under different magnetic fields and temperatures are evaluated by an experiment test rig. Experimental results show that the natural frequency of SDVA has a decreasing trend as the temperature is increased. However, it has an increasing trend with an increase in the control current at the same temperature situation. The influence tendency of the natural frequencies of MRE-based SDVA is discussed. The effect of temperature is more obvious than the control current.

Keywords: magnetorheological elastomers (MREs), intelligent materials, semi-active dynamic vibration absorber (SDVA), marine gearbox vibration

1. Introduction

The vibration of the marine gearbox is one of the main sources of ship vibrations, which not only affects crews’ living environment, but also greatly affects the safety and reliability of the ship. The working conditions of a gearbox may directly affect the dynamic behaviours of the whole machine, and therefore the vibration of the gearbox will affect the normal operation and lifetime of the machine. As the marine gearbox is working with a variable speed, the excitation frequency is of wideband range. Several methods may be used to reduce the vibration of the marine gearbox, such as passive vibration controller, semi-active vibration controller, to name a few. The installation of a dynamic vibration absorber (DVA) is found to be an effective and feasible method. The type of DVA can be categorized into classical passive, active and semi-active DVA. Generally, a passive
DVA consists of a single mass-damper-spring system. Once designed, the physical parameters of the passive DVA are fixed. Since a classical passive absorber operates at a single excitation frequency, resonances in the system appear just above and/or below the excitation frequency, making the effective bandwidth of the absorber very narrow and becoming inefficient as the excitation frequency shifts [1]. To overcome these disadvantages and extend the effective bandwidth of passive absorbers, active and semi-active absorbers are developed and introduced. The active dynamic vibration absorber (ADVA) is achieved using external actuator forces to offset the excitation forces and suppress the vibration [2]. The active DVA can achieve an excellent vibration attenuation effect. However, there are still some inevitable disadvantages for an active DVA, such as the high energy consumption, complicated mechanical structure. The main advantage of the active dynamic absorber is that it enables fast response to disturbances. Some disadvantages of the active dynamic absorber are the potential use for large actuator forces, which may require high power inputs. This may lead to complexity of the system and the possible instability particularly when the excitation frequency moves away from the natural frequency of the system [3]. Semi-active DVA (SDVA) controls the structural vibration by changing its dynamic parameters, such as the stiffness and damping. The advantages of semi-active control lies in the fact that it requires less energy, which lowers the costs, and its complexity is reduced in comparison with active systems, while being nearly as effective. There exist two major categories of variable stiffness elements, namely, mechanically variable spring and intelligent materials. Walsh and Lamancusa reduced transient vibrations using a mechanically variable spring but found disadvantage of long response time of the mechanical structure and difficulty in meeting the design requirements of the system, which rapidly changes the vibration characteristics [4]. Intelligent materials also can be used as variable stiffness elements such as piezoelectric ceramics and magneto-rheological elastomers (MREs). Piezoelectric elements have relatively high stiffness that can lead to relatively large absorber masses. Moreover, piezoelectric materials should not be put into tension, as they are fragile and have displacement amplitude limitations. To make the device more robust and reduce the mass of absorber, magneto-rheological elastomers (MREs) have been used as variable stiffness elements [1,5].

MREs are composites whose highly elastic polymer matrices are filled with magnetic particles. 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 storage modulus can be controlled by the external magnetic field rapidly, continuously, and reversibly [6]. Such properties make MREs promising in many applications, such as SDVAs, stiffness tunable mounts and suspensions, and variable impedance surfaces. Watson applied a patent using MREs for a suspension bushing [7]. Ginder developed an adaptive tunable vibration absorber using MREs [8]. Hoang developed a torsional adaptive tunable vibration absorber using an MRE for vibration reduction of a powertrain test rig; the experimental results show that the ATVA can work in a frequency range from 10.75 to 16.5 Hz (53% relative change), and the numerical simulations show the steady-state vibration of powertrain can be significantly reduced [9,10]. Kim proposed a real time control system for the MRE based TVA to suppress the cryogenic cooler vibrations [11]. Kallio proposed a tunable spring element utilizing an MR elastomer [12]. The device works in squeeze mode and allows compressive loading to be applied on it. The magnetic field adjusts its compressive modulus and thus presents its tunable property. ATVAs can also be designed with a combined mode, such as the combining shear mode and squeeze mode. Ni designed an MR elastomer AVTA in combined shear and squeeze modes; In this design, the piezoelectric actuator installed on a movable arm was used to apply controllable compressive loading to the MR elastomer samples [13]. Numerical simulations indicate that such a design realizes more effective suppression capabilities than it does in shear mode only.

This study aims to investigate the application of MREs in semi-active DVAs for reducing vibration of marine gearbox. A new shear-mode semi-active dynamic vibration absorber (SDVA) is designed based on the stiffness variable characteristics of magneto-rheological elastomers. A compact and utilizable shear-mode vibration absorber is developed. Semi-active vibration absorber applied
for the marine equipment is developed and its natural frequency-shift performance is tested. Experiments on the natural frequencies are carried for SDVA under different magnetic fields and temperatures. The influence tendency of the natural frequencies of MRE-based SDVA is discussed. It is of significance for the engineering design and application of the vibration control of marine equipments.

2. Design principle of SDVA

The dynamic model of a semi-active DVA is shown in Figure 1. The main system is described by a single-degree-of-freedom model, and the DVA is attached as a spring-damper-mass system with variable stiffness. The equations of motion for the system are expressed as:

\[
\begin{align*}
\left\{ \begin{array}{l}
m_1 \ddot{x}_1 + c(\dot{x}_1 - \dot{x}_2) + k_1 x_1 + k_2 (x_1 - x_2) = P \sin \omega t \\
m_2 \ddot{x}_2 + c(\dot{x}_2 - \dot{x}_1) + k_2 (x_2 - x_1) = 0
\end{array} \right.
\end{align*}
\]  

Where \( m_2, c \) and \( k_2 \) are the mass, damping coefficient and stiffness of the SDVA respectively; \( m_1 \) and \( k_1 \) are the mass and stiffness of the primary system respectively. \( x_2 \) and \( x_1 \) are the absolute displacement of the DVA and the primary system respectively; \( P \sin \omega t \) is the harmonic excitation force acting on the primary system. The solution to equation (1) for the displacement amplitude of the primary system is given as:

\[
\frac{X_1}{\delta_{st}} = \frac{1}{\sqrt{\left\{ \left( \frac{1-f^2}{f^2} \right)^2 + (2\zeta f)^2 \right\}}} \quad \left[ \left( \frac{g^2}{g^2} - [1+\mu]g^2 +1 \right)^2 + (2\zeta f)^2 \right]
\]  

(2)

While \( \delta_{st} = \frac{P}{k_1}, \omega_a = \sqrt{\frac{k_1}{m_1}}, \omega_s = \sqrt{\frac{k_2}{m_2}}, \mu = \frac{m_2}{m_1}, f = \frac{\omega}{\omega_a}, g = \frac{\omega}{\omega_s}, \zeta = \frac{c}{2m_s \omega_s}. \) It can be seen from equation (2) that the response of isolated primary system is minimum \((\frac{X}{\delta_{st}} = 0)\) when \( f \) is taken to be 1. In other words, when the natural frequency of dynamic absorber is equal to excitation frequency of external force \((\omega_a = \omega)\), the vibration absorption capability of the absorber is maximum. The semi-active dynamic vibration absorber (SDVA) introduced in this paper is based on this dynamic principle, which can vary its natural frequency as the excitation frequency is...
changed. In doing so, the natural frequency of dynamic absorber traces the excitation frequency of external force on time.

3. Design and experimental analyses of SDVA

3.1 Structure of SDVA

The structure of the MRE-based SDVA is shown in Figure 2 and Figure 3. The SDVA consists of three main components: the oscillator or dynamic mass, smart spring elements with MREs, and the base attached to the isolated system directly. The smart spring elements (MREs) connect the dynamic mass and the base. The dynamic mass and the base are made of low-carbon steel. The current coils are strapped on the base to emit the magnetic field acting on MREs and some grooves are slotted in the dynamic mass to install the current coils to increase the magnetic field intensity. The mass of the oscillator is 7.7 kg and the current coils of the SDVA are 230 rounds. Furthermore, the outer and inner diameter of MREs is 50 mm and 40 mm respectively.

3.2 Theoretical analysis of shift-frequency property

The shear storage modulus $G$ of MREs consists of two terms, namely the zero-field shear storage modulus $G_0$ and the magneto-induced shear storage modulus $\Delta G$:

$$G = G_0 + \Delta G \quad (3)$$

The natural frequency of SDVA can be expressed as:

$$\hat{f} = \hat{f}_0 + \Delta \hat{f} \quad (4)$$

where $\hat{f}_0$ is the initial natural frequency and $\Delta \hat{f}$ is the magneto-induced frequency:

$$\hat{f}_0 = \frac{1}{2\pi} \sqrt{\frac{G_0 A}{m_s h}} - \frac{1}{2\pi} \sqrt{\frac{G_0 \pi (R_i + R_s) l_i}{\rho (R_i - R_s) \pi (R_i^2 - R_s^2) l_s}} = \frac{1}{2\pi} \sqrt{\frac{G_0 (R_i + R_s)}{\rho (R_i - R_s) (R_i^2 - R_s^2)}} \quad (5)$$

$$\Delta \hat{f} = \frac{1}{2\pi} \sqrt{\frac{G_0 (R_i + R_s)}{\rho (R_i - R_s) (R_i^2 - R_s^2)}} \left( \left| 1 + \frac{\Delta G}{G_0} \right| - 1 \right) \quad (6)$$

where $R_i$ and $R_s$ are the outer and inner radius of dynamic mass respectively, $R_i$ is the inner radius of MREs, and $l_s$ is the length of SDVA as shown in Figure 2.

From equation (5), the initial natural frequency of the SDVA can be designed to match the primary system. Equation (6) reveals that the frequency-shift capacity is not only related to the MR effect but also related to the initial shear storage 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 magnetic dipoles model considering the interaction between all particles.
at the same chain is employed [14]. Figure 4 shows two adjacent particles (dipoles) in a chain along with the direction of applied magnetic field. The interaction energy of the two dipoles of equal strength $|m|$ and direction is:

$$E_{12} = \frac{1}{4\pi\mu_0\mu_i} \left[ \frac{\vec{m}_1 \cdot \vec{m}_2 - 3(\vec{m}_1 \cdot \vec{r})(\vec{m}_2 \cdot \vec{r})}{r^3} \right] = \frac{|m|^2 (1 - 3\cos^2 \theta)}{4\pi\mu_0\mu_i r^5} = \frac{|m|^2 (1 - 3r_0^2 + x^2)}{4\pi\mu_0\mu_i (r_0^2 + x^2)^{3/2}}$$  \hspace{1cm} (7)

where $\mu_i$ is the relative permeability of MREs, $\vec{m}$ is the dipole moment of each particle, and where some basic trigonometric identities have been used in the second equality. By defining the scalar shear strain of the particle chain as $\varepsilon = x/r_0$, the interaction energy can be written as:

$$E_{12} = \frac{|m|^2 (\varepsilon^2 - 2)}{4\pi\mu_0\mu_i r_0^2 (\varepsilon^2 + 1)^{5/2}}$$  \hspace{1cm} (8)

Now it is assumed that the particles are aligned in long chains and that there is only magnetic interaction between adjacent particles within the chain, i.e., there are no multi-pole interactions. For a dipole $i$, the dipole moment $\vec{m}_i$ is determined by:

$$\vec{m}_i = \frac{1}{6\pi d^3} \mu_0 \mu_i \chi H_i$$  \hspace{1cm} (9)

where $d$ is the particle diameter, $\chi$ is the susceptibility of iron particles, and $H_i$ is the magnetic field strength at this point, $\mu_0 = 4\pi \times 10^{-7}$ H$ \cdot$ m$^{-1}$, $\mu_i$ is the permeability of MREs. The field experienced by a dipolar particle $i$ is the resultant of an applied external field $H_0$, and ignores the local field induced by the particles around the particle $i$.

The total energy density (energy per unit volume) associated with the one dimensional shear strain can be calculated by multiplying the particle-to-particle energy, given by equation (8), by the total number of particles and dividing by the total volume:

$$U = \frac{n \cdot E_{12}}{V_e} = \frac{n \cdot V_i \cdot E_{12}}{V_e \cdot V_i} = \frac{\phi \cdot E_{12}}{V_i} = \frac{3\phi(\varepsilon^2 - 2)|m|^2}{2\pi^2 \mu_0 \mu_i d^3 r_0^2 (\varepsilon^2 + 1)^{5/2}}$$  \hspace{1cm} (10)

where $V_e$ is the total volume of MREs, $V_i$ is the volume of each particle, $n$ is the total number of particles, and $\phi$ is the volume fraction of particles in the composite. The stress induced by the application of a magnetic field can be computed by taking the derivative of inter-particle energy density with respect to the scalar shear strain:

$$\sigma = \frac{\partial U}{\partial \varepsilon} = \frac{9\phi(4 - \varepsilon^2)|m|^2}{2\pi^2 \mu_0 \mu_i d^3 r_0^2 (\varepsilon^2 + 1)^{3/2}}$$  \hspace{1cm} (11)

If equation (9) is substituted into equation (11) and the derivative of the stress is further taken with respect to the scalar shear strain, the magneto-induced shear storage modulus can be written as:

$$\Delta G = \frac{\partial \sigma}{\partial \varepsilon} = \frac{\mu_0 \mu_i (4\varepsilon^4 - 27\varepsilon^2 + 4) \chi^2 H_0^2}{8h_j^2 (\varepsilon^2 + 1)^{3/2}}$$  \hspace{1cm} (12)
where \( h_0 = \frac{r_0}{d} \) is the indication of the gap between two adjacent particles in a chain. When the shear strain is small, equation (12) can be simplified as:

\[
\Delta G \approx \frac{\phi \mu_r \chi^2 H_0^2}{2h_0^2}
\]  

(13)

Substituting equation (13) into equation (6), then equation (6) can be rewritten as:

\[
\Delta f = \frac{1}{2\pi} \sqrt{\frac{G_0 (R_r + R_i)}{\rho (R_r - R_i) (R_r^2 - R_i^2)}} \left[ 1 + \frac{\phi \mu_r \chi^2 H_0^2}{2h_0^2 G_0} - 1 \right]
\]  

(14)

It can be seen from equation (13) that the magneto-induced shear storage modulus rises quadratically with the applied external field \( H_0 \). In addition, the change of shear storage modulus of MREs is greatly affected by the ratio of mean distance between two adjacent particles to the mean radius of the particles. As can be observed from equation (14), it should have a relatively high particle volume fraction of MREs, increase the applied external field, reduce the gap between two adjacent particles in a chain, and have the high-saturated magnetic flux density of iron particles to obtain a large shift of natural frequency of SDVA.

### 3.3 Experimental analysis of the frequency-shift property of SDVA

To investigate the frequency-shift property of SDVA, the experimental setup is shown in Figure 5–6 and the experimental procedure is as below. The developed MRE-based SDVA is placed at the centre of the vibration test rig. An exciter via the power amplifier generates the sweep-frequency sine excitation that is applied to the vibration test rig and the vibration test rig produces the horizontal direction movement. Two accelerometers are placed on the oscillator and the base to measure their responses, respectively. The measured signals are sent to the dynamic signal analysis system. The transfer function can be achieved by using FFT analysis and the natural frequency of SDVA is subsequently obtained. The control current applied to the MRE-based SDVA is supplied from the adjustable direct current power source. In addition, the temperature of MREs in SDVA is measured using by the thermometer. The whole system can measure the natural frequencies of SDVA under different control currents and temperatures. The results are shown in Table 1 and Figure 7.
Figure 6: The frequency-shift property test of dynamic absorber.

Table 1: The natural frequencies of dynamic absorber with different test situations

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Figure 7: The natural frequencies of dynamic absorber with different control currents under different temperatures.

It can be seen that the natural frequencies of SDVA range from 65 Hz to 139 Hz is coincide with the normal working frequencies of marine equipments. The natural frequency of SDVA has a decreasing trend with an increase in the temperature. However, it has an increasing tendency when the control current is increased at the same temperature situation. The effect of temperature is more obvious. As the temperature factor is difficult to control in the test process, big challenges exist in practical applications that how to make the dynamic vibration absorber work as stable as possible in the most test situations listed above.
4. Conclusions

In this work, a new design of a shear-mode SDVA is developed based on the stiffness variable characteristics of magneto-rheological elastomers. A compact and utilizable shear-mode vibration absorber is developed. Semi-active vibration absorber for the marine equipment is built up and its natural frequency-shift performance is tested. The natural frequencies of SDVA under different magnetic fields and temperatures are evaluated by experiments. Experimental results show that the natural frequency of SDVA has a decreasing tendency as the temperature is increasing. However, it has an increasing trend with an increase in the control current at the same temperature situation. The effect of temperature is more obvious than the control current. It is of significance for the engineering design and application of the vibration control of marine equipments.

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