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Disciplines

Engineering | Science and Technology Studies

Publication Details

Yang, J., Sun, S. S., Ning, D., Li, Z., Deng, L., Christie, M. D., Du, H., Zhang, S. W. & Li, W. H. (2019). Development and evaluation of a highly adaptive MRF-based absorber with a large effective frequency range. *Smart Materials And Structures*, 28 (10), 105003-1-105003-10.

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Development and evaluation of a highly adaptive MRF-based absorber with a large effective frequency range

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Abstract

This work developed an MRF-based absorber aiming at using the controllable storage modulus (stiffness) of MRF in its pre-yield region to achieve a wide natural frequency variation range. In the characterization test, its damping and stiffness properties in response to different amplitude and magnetic fields were explored. A sweep frequency test was then conducted to obtain the frequency shift property of the MRF absorber and the testing results demonstrate that its relative change rate of natural frequency is high up to 1986%. A mathematical model was built to predict the resonance shifting performance of the MRF absorber under different currents. Then an evaluation system with a primary mass was built to evaluate the absorber's capability on reducing vibrations. The experimental results under single-frequency excitation show that the MRF absorber has high efficiency in reducing the acceleration of the primary system under the STFT based control. The absorption experiment under the sweeping frequency excitations was also conducted and the results indicate that the absorption point of the absorber can shift its positions as the applied current changes, and that the primary acceleration remains the minimum under the STFT based control. All of the experimental results indicate that the proposed MRF absorber is competent as a vibration reduction device.

Keywords: magnetorheological fluid, variable storage modulus, absorber, vibration reduction.

1. Introduction

Dynamic vibration absorber (DVA) has been a very popular device for suppressing vibrations. It can be dated back to the early 20th century when Frahm invented the first absorber as the mass spring system [1]. The absorber generally consists of an oscillator and a stiffness component and it is usually attached to a primary system to suppress vibrations through energy transfer under the condition that the natural frequency of the absorber matches the excitation frequency. However, the vibration can be very complicated and its frequency spectrum changes with time. In such cases, the traditional DVA is not effective because of its passivity.

In order to make the DVAs more adaptive, the addition of an active force actuator to the passive DVA was proved to be able to improve the effectiveness considerably [2], which makes the emerging of active DVAs. However, the active DVA requires high power consumption, high cost and maintenance, which makes it less reliable and less acceptable than the passive DVA, though it provides better vibration suppression performance. For the sake of overcoming these drawbacks but still keeping the benefits of the active DVA, the concept of semi-active DVA has been introduced. Ever since then, many methods have been developed to realize the tunable natural frequency for the semi-active DVA [3, 4]. For example, shape memory alloys, piezo stacks, and other smart element were used to control the semi-active DVA's natural frequency [5-8].

Magnetorheological (MR) materials, mainly consisting of MR fluids and MR elastomers, are typical smart controllable materials whose stiffness and damping can be controlled by an external magnetic field and are ideal to develop semi-active DVAs. Specifically, MR elastomer (MRE) is a smart material whose mechanical properties can be controlled upon the action of a magnetic field [9, 10]. By incorporating the MRE into an absorber, it would be capable of changing its natural frequency. In the past decades, MRE-based absorbers have attracted considerable attention and resulted in a large number of research publications [11-14].

MR fluid (MRF) is widely used for implementation of smart devices. MRF materials have two working regions; one is the pre-yield region, and the other is the post-yield region. The storage modulus of MRF in the pre-yield region increases rapidly in response to the increasing magnetic field, after entering the post-yield region, the storage modulus tends to level off and become stable. The field-dependent shear stress in the post-yield region has been widely used for engineering applications, such as MR dampers for structural vibration control [15], vehicle suspension development [16], and seat suspension implementation [17-20], while its property of stiffness controllability in the pre-yield region is seldom explored or utilized for engineering applications. In fact, the large stiffness controllability range in the pre-yield working region enables MRF to be an ideal material to work as a controllable-stiffness component for adaptive absorbers implementations. Additionally, according to the literatures, the frequency variation percentage of the MRF absorber is potentially much wider than MRE absorbers because that the storage modulus variation percentage of MRF in pre-yield state is much larger than MRE.

However, the application of MRF working in pre-yield state on vibration absorption has rarely been investigated in existing literature except the research in [4]. On the basis of this motivation, this paper presents an advanced adaptive MRF absorber with larger frequency shift range. The structure and mechanism of the MRF absorber is detailed in Section 2. Characterizing the absorber is conducted in Section 3. Section 4 introduces the modelling of the absorber and Section 5 evaluates the absorption effectiveness of the absorber by experiments. This paper concludes in Section 6.

2. Design and working mechanism

Figure 1 shows the sectional view and the prototype photograph of the MRF-absorber. It mainly contains three parts: the base, the container, and the electromagnet. The base is square and has four pillars on its four corners. The container which is fixed to the base is round and it is used to store MRF. There is a round iron plate attached to the center of the container. It is an essential part in terms of forming the magnetic circuit working on MRF. The electromagnet serves as the oscillatory mass and it is hung over the iron plate by the four hanging ropes. Its resistance is $25\ \Omega$ and inductance is $35.45\ \text{mH}$. A gap between the iron plate and the electromagnet must be reserved so that MRF can be filled in. To avoid any collision between the electromagnet and the pillars during movement, the four hanging ropes must find their best positions to guarantee that the oscillatory mass is hung above the center of the container. As the bottom surface of the electromagnetic has an annulus notch, the effective area in contact with MRF is the superposition of the internal round area A_1 and the external annular area A_2 . B_1 and B_2 indicate the magnetic field in areas of A_1 and A_2 , respectively. As these two areas are in parallel connection, the overall lateral stiffness of MRF strength is the sum of the stiffness from the two areas. All the dimensions of the absorber are listed in Table 1: h indicates the thickness of MRF, l is the length of the hanging rope, m is the oscillatory mass, r_1 and r_2 are the inner radius and outer radius of the solenoid, respectively, and r_3 is the radius of the oscillatory mass.

Figure 1 Sectional view and photograph of the MRF absorber

Table 1 Dimensions of the absorber.

Parameters	h	l	m	r_1	r_2	r_3
Values	3.5mm	9cm	0.6kg	13.5mm	21mm	25.5mm

This absorber is especially designed to use the variable stiffness characteristic of MRF in the pre-yield region. When the absorber is subject to a horizontal vibration, the oscillatory mass will be driven to oscillate in the same direction, thereby, the MRF will operate in shear mode. Then the oscillatory mass will facilitate its motion under the operation of the pulling force from the ropes, gravity, magnetomotive force, and damping force from MRF. When the electromagnetic coil is energized by a DC power, a magnetic circuit closed by the oscillatory mass, MRFs, and iron plate will be formed. The magnetic field density is adjustable according to the levels of the applied current. Supposing that the MRFs operate in the pre-yield region, its storage modulus is field-dependent, and then the variable stiffness of the absorber can be manifested. As a result, the resonance frequency of the MRF absorber can be controlled.

3. Characterization of the MRF absorber

3.1 The mechanical characterization of the MRF absorber

An experimental platform was set up to test the performance of the MRF absorber. As shown in Figure 2, the MRF absorber was fixed to a horizontal shaking table which was excited by a shaker (Vibration Test Systems, AURORA, Model No.: VG 100-8). A DC power supplier (GW INSTEK GPC-3030D) was used to adjust the currents of the coil. A force sensor (CL-YD-302) was fixed with one of its ends connected to a fixed T-slot bar and the other to the top surface of the oscillatory mass. The fixed T-slot bar, the force sensor and the absorber are in a line which is in parallel with the moving direction of the shaker table. When the system is vibrating, the force sensor and the absorber remain motionless. A laser sensor (MICRO-EPSILON Company: ILD 1700-50) was installed to collect the displacement signal of the shaking table. The collected force signal and displacement signal were then transferred to the computer through the DAQ board (National Instruments Corporation: NI PCI-6221) to be processed.

Figure 2 Experimental setup for characterizing the MRF absorber

In order to characterize the force-displacement relationships of the MRF absorber, a series of test were conducted. These tests use sine waves with 5Hz frequency as excitations. There are five current levels (0A, 0.06A, 0.12A, 0.18A, and 0.24A) powering the absorber coil in each test, the corresponding magnetic flux density in the internal round area and the annular region are measured and given in Table 2. Figure 3 and Figure 4 are two examples of these force-displacement (F-A) relationships which show the influence of the magnetic field on the mechanical property, where the displacement is 0.2mm and 0.4mm, respectively. It can be seen from the two figures that as the magnetic field increases, a transition from the damping property to the stiffness property can be observed. That is because the strengthened magnetic field reinforced the chain structures between the magnetic particles in the MRF and then the mechanical property changes from the damping property to the stiffness property. The transition currents are 0.12A and 0.18A, respectively, when the amplitude is 0.2mm and 0.4mm. Specifically, when the current is higher than the transition current, the MRF absorber behaves its variable stiffness characteristic; otherwise, it behaves variable damping property. It can also be found that the critical current (i.e. the critical magnetic field strength) where the transition appears is amplitude dependent: the MRF absorber subject to smaller loading amplitude requires less power to enable the happening of damping-to-stiffness transition. In other words, it would be much easier for the absorber to show stiffness property when it is used to control small amplitude vibrations.

Table 2 Magnetic flux density vs. applied current.

I(A)	0	0.06	0.12	0.18	0.24
B_1 (mT)	0	14.91	24.85	34.63	44.41
B_2 (mT)	0	9.17	16.60	23.91	31.21

Figure 3 Force-displacement relationship of the MRF absorber (displacement=0.2mm)

Figure 4 Force-displacement relationship of the MRF absorber (displacement=0.4mm)

To verify its amplitude-dependent property, the MRF absorber was also tested under sinewaves with different amplitudes. For this test, the current was constant. Figure 5 is one example when the $B_1=34.63\text{mT}$ and $B_2=23.91\text{mT}$ (corresponding to 0.18A).

Figure 5 Force-displacement relationships under different amplitude

Likewise, the hysteresis loops in Figure 5 also show a transition, but it is from stiffness property to damping property: i.e. the absorber performs stiffness property when the loading amplitude is smaller than critical (transition) amplitude, otherwise, it performs damping property. The reason is that when the amplitude is small, the MRF in the absorber works in the pre-yield region where the storage modulus increases with the increasing shear strain, while the MRF enters into the post-yield region when the loading amplitude exceeds the critical amplitude,.

From Figure 5 it can be concluded that for each certain magnetic density, there is corresponding critical amplitude to make the transition occur. Therefore, the critical amplitude can be determined as the function of the magnetic strength. In order to find all critical amplitude under different currents (magnetic fields), more tests were conducted and the testing results are plotted in Figure 6. The correspondence relationship between the critical amplitude ($A_{critical}$) and the current forms a boundary which divides the plain into two areas. For a constant current, when the loading amplitude is above the boundary, the absorber works in the damping dominant area (post-yield area); otherwise, it works in the stiffness dominant area (pre-yield area). The damping and the stiffness both show field-dependent characteristics in their areas: the damping and the stiffness increase (Figures 3-5) when the current (magnetic strength) increases.

Figure 6 The boundary dividing the damping and stiffness dominant areas.

3.2 The frequency shift performance of the MRF absorber.

Apart from the force-displacement relationships, frequency shift property is also an important indication of the absorber performance. In this experiment, the experimental setup shown in Figure 2 is used. The force sensor is disassembled and the oscillator of the MRF absorber is free to move in this test. Two accelerometers (CA-YD-106 SINOCERA Piezotronics, Inc.) are used to measure the accelerations of the absorber and the excitation, respectively, and pass them to the computer for processing via the DAQ board. The LabVIEW program is designed to be the controller and display unit by which the harmonic excitation can be generated and the experimental results can be displayed and recorded.

A series of harmonic excitations with sweeping frequency (f_e) were generated and used to excite the absorber to obtain the transmissibility of MRF absorber in the pre-yield region. Figure 7 shows the transmissibility responses under different magnetic fields. The magnetic strength is adjusted via changing the applied current from 0A to 0.5A with a step of 0.05A. In such cases, the stiffness and the damping are very complicated parameters as they are field-dependent. Figure 7 shows that the resonance frequency shifts when the magnetic field changes. This means that the MRF operates in the pre-yield region and the absorber performs variable stiffness characteristic. Table 3 lists the natural frequencies in correspondence to different current levels (magnetic fields). It changes from 2.14Hz at 0A to 42.5Hz at 0.5A, producing a relative change up to 1986%. This number is much higher than that of the reported MR absorber in existing literatures. This result indicates that this MRF absorber is very competent as the vibration-reduction device.

Figure 7 Frequency shift performance of the MRF absorber

Table 3 Current and the corresponding natural frequency

Current(A)	0	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5
F(Hz)	2.14	3.46	4.6	7	14.2	17.35	21.87	25.9	34.5	41.36	42.5

4. Modelling of the MRF absorber.

This section aims to obtain a relationship between the applied current and the natural frequency of the MRF absorber. To achieve this, an empirical model describing the storage modulus of MRF as the function of the magnetic flux density was first used. Then the expressions of the natural frequency of the MRF absorber were determined by representing this system with a single degree of freedom. Before working out the relationship between the natural frequency and the applied current, the magnetic flux density as the function of the applied current was also determined.

4.1 An empirical model for MRF

MRF used in this work is from Guohao Sensing Technology Research Institute (model: GH-MRF-250, density 2.55g/cm³). Its shear properties were measured using a rheometer (Physica MCR 301, the Anton Paar Company, Germany) in oscillatory mode. Figure 8 shows the storage modulus (G) in response to the sweeping magnetic field (B) when the shear strain is 0.7%. It can be seen that the measured storage modulus in the pre-yield region increases gradually under a small magnetic density while rapidly at moderate magnetic flux density, and then it tends to be saturated. Figure 9 presents the relationship between the storage modulus and the shear strain; it shows that the storage modulus changes rapidly with the variation of shear strain. It means that the storage modulus is very sensitive to the change of the shear strain. Figure 10 provides the yield stress characteristics and the viscosity information. The viscosity (η) in Figure 10 was measured without the present of magnetic field and the shear rate ($\dot{\gamma}$) increased from 0.1S⁻¹ to 1000S⁻¹. It is seen that the viscosity decreases sharply when the shear rate is very small and then levels off when the shear rate keeps increasing.

An empirical model for MRF is used here to describe the relationship between the storage modulus and the applied magnetic flux density [21]:

$$G = G_0 + (G_\infty - G_0)(1 - e^{-a_1 B^{a_2}}) \quad (1)$$

where G_0 , G_∞ , a_1 , and a_2 are empirical constants, and G is the storage modulus of MRF under the magnetic field. G_0 is the storage modulus when the magnetic field is zero, and G_∞ is the saturated storage modulus. These two parameters can be read from the experimental data. a_1 and a_2 can be determined by fitting the empirical model to the experimentally responses. They are calculated by minimizing the estimation sum squared error between the estimated and measured data. The values for the above parameters are listed in Table 4. The estimated performance of the fluids is compared with the experimental data in Figure 8. It can be seen that the fitting results show excellent resemblance between the measured and the predicted data.

Table 4 Empirical constants for the storage modulus.

Parameters	G_0	G_∞	a_1	a_2
Values	$2.51 \times 10^3 \text{ Pa}$	$3.5 \times 10^6 \text{ Pa}$	6.8339	1.8248

Figure 8 Comparison between the measured and the estimated data.

Figure 9 Storage modulus as the function of the shear strain.

(a)

(b)

Figure 10 (a) The yield stress characteristic; (b) The viscosity information.

4.2 Modelling the MRF absorber

Figure 11 Force analysis of the oscillatory mass.

In this work, the purpose of modeling this absorber is to obtain the natural frequency as a function of the applied current. As the movement of the oscillatory is small, the absorber motion satisfies the presume that the oscillatory mass remains on the same plane during movement. Figure 11 shows the force analysis for the oscillatory mass when the offset angel is θ . It is assumed that the forces working on the oscillatory mass are evenly distributed on its four points, and then one of them was taken out as an example. The oscillatory mass is under the operation of the pulling force from the ropes, the gravity, the force from MRF, and the magnetomotive force. The force analysis of each force working on the oscillator is detailed as follow.

The expression for MRF stiffness is:

$$k_{\text{MRF}} = \frac{GA}{h} \quad (2)$$

where k_{MRF} is the MRF stiffness in the effective area of A . As mentioned previously, A is the effective area of MRF in contact with the magnetic field and it consists two parts, i.e. an internal round area (A_1) and an external annular region (A_2). The magnetic flux density working on these two regions are different, thereby, the stiffness of MRF produced in these two areas should be calculated separately (k_1 and k_2). As the stiffnesses from the two regions are in parallel connection and both contribute to the overall stiffness, the overall equivalent stiffness should also be a superposition. Therefore:

$$k_1 = \frac{G_1 A_1}{h}, k_2 = \frac{G_2 A_2}{h}, \text{ and } k_{\text{MRF}} = k_1 + k_2 \quad (3)$$

G_1 and G_2 are the MRF storage modulus in correspondence to the working areas of A_1 and A_2 .

For the magnetic attractive force, F_m , between the iron plate and the oscillatory mass in vertical direction, it can be obtained according to [22]. The integration of the exerted force between the electromagnet and the iron plate should be:

$$F_m = \int [\mu_e \cdot (H \cdot n) \cdot H - \frac{\mu_e}{2} H^2 \cdot n] dA \quad (4)$$

with n the normal vector to the surface A, H is the magnetic field and, μ_e is the permeability of MRF. As the direction of the magnetic field is the same with the normal vector, and it is assumed that there are no other magnetic fields outside the particular working area, equation (4) can then be simplified as:

$$F_m = \frac{1}{2\mu_0} (\int B_1^2 dA_1 + \int B_2^2 dA_2) = \frac{1}{2\mu_0} (B_1^2 A_1 + B_2^2 A_2) \quad (5)$$

whose direction is the same with gravity.

By considering the force analysis of the oscillatory mass, as shown in Figure 11, the dynamic equation of the whole absorber system in horizontal direction can be expressed as:

$$m\ddot{x} + c\dot{x} + k_{\text{MRF}}x + 4 \times \frac{1}{4} (mg + F_m) \tan \theta = 0 \quad (6)$$

where m is the oscillatory mass, x is the lateral displacement of the oscillatory mass, c and k_{MRF} are the damping and stiffness coefficients of MRF, respectively. θ is the offset angel during movement and it is so small that $\tan \theta \approx \sin \theta = \frac{x}{l}$, then equation (6) can be derived as:

$$m\ddot{x} + c\dot{x} + \left(k_{\text{MRF}} + \frac{mg + F_m}{l}\right)x = 0, \quad (7)$$

Then the natural frequency of the absorber can be obtained as:

$$f = \frac{1}{2\pi} \sqrt{\frac{G_1 A_1}{hm} + \frac{G_2 A_2}{hm} + \frac{g}{l} + \frac{F_m}{ml}} \quad (8)$$

where $A_1 = \pi r_1^2$

$$A_2 = \pi r_3^2 - \pi r_2^2$$

$$G_1 = [G_0 + (G_\infty - G_0)(1 - e^{-a_1 B_1^{a_2}})]$$

$$G_2 = [G_0 + (G_\infty - G_0)(1 - e^{-a_1 B_2^{a_2}})]$$

Define that:

$$B_1 = c_1 I + d_1, \quad B_2 = c_2 I + d_2 \quad (9)$$

where c_1, c_2, d_1 and d_2 are the constants to be determined.

Then a gauss meter was used to measure the magnetic flux density produced in between the iron plate and the oscillatory mass in response to different current levels and the data were used to fit out the constants in equation 9, which are listed in Table 5.

Table 5. Values of Constant parameters

Parameters	c_1	c_2	d_1	d_2
Values	163	121.77	5.29	1.99

By substituting equation (9) to equation (8), the relationship between the natural frequency (f) and the current (I) can be determined. Figure 12 shows the comparison between the measured data and the simulated data. Overall, the model can predict the resonance frequency of the absorber well although there is discrepancy between the two curves under some currents. The reason of the discrepancy is that the vibration amplitude might change under different frequency and current status, and the modulus is very sensitive to the vibration amplitude (please refer to Figure 9). As the natural frequency is a function of the storage modulus, it will also change when the vibration amplitude changes. In this paper, the data for modelling MRF was obtained from the experimental results when the shear strain is constant 0.7% (Figure 8), so the model for the absorber was built based on the assumption that the shear strain remains 0.7%. However, the natural frequency information of the absorber was collected under varying currents (magnetic fields), which means that the excitation strain cannot remain always at 0.7%. Therefore, there is difference between the estimated and the measured natural frequency. Although the discrepancy exists, this model still can provide meaningful guideline to design the MRF absorber.

Figure 12 Natural frequency as the function of the applied current

5. Absorption evaluation

In order to evaluate the effectiveness of the absorber on reducing vibrations, a series of experimental tests were performed. This experiment divides into two parts. The first tests used the excitation signals with single frequency; the second part measured the absorption transmissibility under sweeping frequency. Figure 13 shows the control diagram. The absorber is attached to a spring-mass system, serving as the primary system, which is subject to the horizontal excitation during the whole test. For all the tests in this section, passive control and semi-active control were considered. Passive control means the absorption system operates in an open loop without any feedback or real-time control, while the semi-active control adopts a control algorithm to form a closed loop. The control algorithm used is the Short Time Fourier Transform (STFT) which aims to transfer the responses in time domain into the frequency domain signals and work out the dominant frequency of the vibration excitation. As Figure 13 shows, an accelerometer (SINOCERA PIEZOTRONICS: CL-YD-302) was used to capture the acceleration of the primary system. The measured acceleration was then sent to the STFT controller (National Instruments Corporation: NI PCI-6221). The control algorithm programmed by LabVIEW can identify the dominant frequency and calculates the desired current according to the relationship between the current and the absorber resonance. The desired current signal generated by the controller will be sent to a power amplifier (SINOCERA YE5871) to enlarge its power. Then the controlling current from the power amplifier will be sent to the absorber which will tune its resonance frequency to match the excitation frequency. The working principle of the STFT control algorithm is explained by the following equations.

Figure 13 The schematic diagram for absorption evaluation.

For the first step, define the time segment as $x_\tau(t)$ and it can be calculated by multiplying the signal $x(t)$ by a window function $h(t)$:

$$x_\tau(t) = x(t)h(t - \tau) \quad (10)$$

where τ is the fixed time, and t is the running time. The hamming window is used as the window function. After that, the Fourier transform for the modified signal is calculated as:

$$X_\tau(\omega) = \frac{1}{\sqrt{2\pi}} \int x(t)h(t - \tau)e^{-j\omega t} dt \quad (11)$$

The energy density of the windowed signal at fixed time τ can be calculated by:

$$P(\tau, \omega) = |X_\tau(\omega)|^2$$

$$= \left| \frac{1}{\sqrt{2\pi}} \int x(t)h(t - \tau)e^{-j\omega t} dt \right|^2 \quad (12)$$

which can provide the time–frequency distribution. Then the dominant excitation frequency at time τ is given by:

$$\langle \omega \rangle_\tau = \frac{1}{|x(\tau)|^2} \int \omega |X_\tau(\omega)|^2 d\omega \quad (13)$$

Upon obtaining the above dominant frequency of the excitation, the natural frequency f of the absorber should be equal to the dominant frequency of the excitation so as to attenuate the vibration of primary system. Then the desired current can be determined using the following relationship which is obtained by fitting the experimental data shown in Table 3:

$$I = 2.55 \times 10^{-3} + 1.65 \times 10^{-2}f - 1.28 \times 10^{-4}f^2 \quad (14)$$

The absorber will then adjust its resonance frequency to match the excitation frequency after receiving the desired current signal to obtain the optimal performance.

5.1 The vibration absorption performance of the MRF absorber under a constant frequency excitation

Figure 14 shows the primary acceleration (a_{primary}) responses in time domain under excitation frequencies of 10Hz, 20Hz, and 30Hz. By clicking the switch in the control program, the control mode can be switched flexibly between the passive control and the semi-active control. For the comparison purpose, the control mode is switched from passive to semi-active during each test in the following three figures. It can be seen that the primary acceleration shows an obvious reduction when the semi-active control is turned on. The maximum relative reduction among the three figures is high up to 47.8%.

(a)

(b)

(c)

Figure 14 Primary acceleration performances under single-frequency excitation

5.2 The vibration absorption performance of the MRF absorber under a sweep frequency excitation

To further confirm the effectiveness of the absorber in terms of reducing the primary acceleration, harmonic excitations with sweeping frequency from 1Hz to 60Hz were also considered. To achieve a constant excitation vibration, the amplitude of the controlling signal of the shaker was constantly set to be 1.5V. To provide a reference for comparison, the natural frequency of the primary system without the attachment of the absorber was firstly measured and it is shown by the bold black curve in Figure 15. The other curves in Figure 15 are the absorption performances of the system with the presence of the absorber under constant current (magnetic field). The absorber under such conditions can be considered as passive absorbers with different resonance frequencies. For each of the absorption curves (0A-0.5A), there is an absorption point where the amplitude of the transmissibility is reduced because the natural frequency of the absorber matches the excitation frequency.

Figure 15 The absorption performance of MRF absorber

After the evaluation of the passive absorbers (under different constant currents), the vibration absorption performance of the semi-active MRF absorber under the STFT control is evaluated and the transmissibility of the primary system is also presented in Figure 15, shown as the bold blue curve. It can be seen that this semi-active-controlled transmissibility has the minimum amplitude through almost all the frequency range and it passes through almost all the absorption points, indicating that this controlled MRF absorber is highly efficient in tracing the excitation frequencies and reducing the vibration.

6. Conclusion

An MRF-based absorber with large effective frequency bandwidth was successfully developed. The unique of this absorber lies in its utilization of MRF's operation in the pre-yield region: the storage modulus of MRF increases with the increasing magnetic field. To observe the property of the absorber, characterizing experiment was conducted. The experimental results show that the mechanical property of the absorber performs a transition from damping characteristic to stiffness characteristic in response to the increase of the external magnetic field under fixed amplitude, and a reverse transition in response to the increasing amplitude under fixed magnetic field. And then a critical relationship between the applied current and the loading amplitude was obtained. The transmissibility performance demonstrates frequency shift property with a relative change up to 1986% which is much higher than that the other traditional MR absorbers can

achieve. This verifies that this MRF-based absorber is able to perform variable stiffness successfully. An empirical model which describes the relationship between the natural frequency of the MRF absorber and the applied current was successfully built. Then a series of absorption experiments were conducted to evaluate the effectiveness of the absorber on reducing vibrations. Excitation signals with single frequency and sweeping frequency are both considered. For the purpose of comparison, passive absorber and semi-active absorber with STFT control are both conducted. The experimental results under single-frequency excitations demonstrate the high effectiveness of the MRF absorber in reducing the primary accelerations with a reduction rate of 47.8%. The testing results under sweeping frequency demonstrate that the absorption points can also shift when the applied current changes. In addition, the testing results also prove the high efficiency of the semi-active MRF absorber in absorbing vibrations through its always minimum transmissibility.

Acknowledgements

The work is supported by ARC Discovery grant (DP150102636) and Linkage grant (LP150100040), and China Postdoctoral Science Foundation Grant (No. 2019M652204).

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