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The Simulation of Magnetorheological Elastomers Adaptive Tuned Dynamic Vibration Absorber for Automobile Engine Vibration Control

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Abstract:

The aim of this article is to investigate the use of a Dynamic Vibration Absorber to control vibration of engine by using simulation. Traditional means of vibration control have involved the use of passive and more recently, active methods. This study is different in that it involves an adaptive component in the design of vibration absorber using magnetorheological elastomers (MREs) as the adaptive spring.

MREs are kind of novel smart material whose shear modulus can be controlled by applied magnetic field. In this paper, the vibration mode of a simple model of automobile engine is simulated by Finite Element Method (FEM) analysis. Based on the analysis, the MREs Adaptive Tuned Dynamic Vibration Absorber (ATDVA) is presented to reduce the vibration of the engine. Simulation result indicate that the control frequency of ATDVA can be changed by modifing the shear modulus of MREs and the vibraion reduction efficiency of ATDVA are also evaluated by FEM analysis.

Keywords: magnetorheological elastomers, field-dependent shear modulus, Dynamic vibration absorber, Finite Element Method

1. Introduction

Most of mechanical, civil and construction systems are suffered from undesirable vibrations, which may damage the systems or even make the systems fail. In addition, vehicle manufacturers are continuously searching for ways in which vehicle refinement can be improved and its cost reduced. This involves the study of a wide range of noise sources from engine structure-borne noise to airborne type noise. Consumers now expect high levels of comfort and refinement from all passenger cars. In response, car manufacturers are placing vehicle refinement at the heart of their product development strategies. It is, therefore, of great importance to develop vibration control devices to reduce or suppress system vibrations [1-2].

Among all vibration control devices, research on Tuned Dynamic Vibration Absorbers (TDVAs) is, perhaps, the richest in terms of total number of investigations and the time devoted to these investigations. Since their invention in 1900s [1], TDVAs have effectively suppressed vibrations of machines and structures. However, the effectiveness of conventional TDVA is always limited due to the narrow frequency ranges. In many practical applications, off-tuning of a TDVA occurs because of varying usage patterns and loading conditions. To overcome these shortcomings, Adaptive Tuned Dynamic Vibration Absorbers (ATDVAs) are extensively studied. The ATDVA is similar to conventional TDVA but with adaptive elements that can be used to change the tuned condition.

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Generally, ATDVA designs adopt two major groups: one is to use variable geometries and the other is to use smart or intelligent materials [3]. From the important view of system reliability and maintainability of ATDVA designs, Sun et al. [4] suggested the use of intelligent materials as alternatives. Davis et al. [5] suggested the use of a piezoelectric ceramic elements as part of the device stiffness, and reported the use capacitive shunts to vary the device natural frequency from 290 Hz to 350 Hz. Flatau et al. [6] used the magnetostrictive material Terfenol-D to develop an ATDVA that achieved a natural frequency variation from 1375 Hz to 2010 Hz. Williams et al.[3] used the thermal actuation of shape memory alloys (SMA) as tuning materials to develop an ATDVA. Testing showed that the SMA ATVA natural frequency could be varied by approximately 15% from 38.5 Hz to 46.5 Hz. Though many attempts have been made, there are still some shortcomings, such as low responses, complex structures, prune to working conditions, etc.

This paper proposes to design a new ATDVA working with a new intelligent material: MR elastomers. With the magnetic field increases, the shear modulus of MR elastomers increases steadily. Removal of the magnetic field, MR elastomers immediately reverse to their initial status. Based on these unique characteristics that their mechanical properties can be magnetically controlled, MR elastomers have found a lot of applications, such as dampers, engine mounts, and shock absorbers [8]. The use of MREs to develop ATDVAs are expected to have many advantages: very fast response (less than a few milli seconds), simple structure, easy implementation, good maintenance, high stability, and effective control. Ginder et al. [8] did a pioneer work that utilized MR elastomers as variable-spring-rate elements to develop an ATDVA. Their results indicated that a natural frequency range from 580 Hz to 710 Hz at the magnetic field 0.56 Tesla. Gong et al. [9] used MR elastomers to develop an ATDVA and reported the natural frequency varying 50% from its centre frequency.

The crankshaft of automobile engine is subject to dynamic torsional deformation as well as dynamic axial displacement, both causing vibration. Vibration control is of great importance because excessive vibration levels may damage the engine and create an annoying human environment. In this paper, the purpose of the new ADTVA is to suppress the vibration of an engine when the external stimulant forces with variant frequencies are added on this engine. Hence, before developing the new ATDVA, it is proposed to examine the validation of this ATDVA. FEM can be used over the complete frequency range of interest. It provides a valuable tool for the rapid assessment of the noise implications of design changes, thereby reducing the number of options requiring detailed analysis. Following this, on the basis of the principle of the MREs and the vibration suppression, the finite element method is applied to build an engine model according to the properties of the engine 1.3L VVT2. The transmissibility of the engine model corresponding to stimulant forces with variant frequencies is calculated and recorded by the FEM too. Based on the simulation results, MATLAB is used to analyze and conclude the relationship between the properties of the MREs and the effect of the vibration suppression.

This paper is structured as four sections. In Section 1, the intelligent material MR Elastomer is introduced, and its' changeable range of the shear modulus is discussed. In Section 2, the principle of the TDVA is explained. Section 3 is to demonstrate the Finite Element Modeling and the simulation results. Finally, the conclusion is given.

2. The MR Elastomer in the ATDVA

MR elastomers are suspensions of magnetized particles dispersed in a polymer medium, such as rubbers [7]. When individual particles are exposed to an applied magnetic field, magnetic dipole moments pointing along the magnetic field are induced in the particles. Pairs of particles then form head-to-tail chains. After the matrix is cured, the particles are locked into place and the chains are firmly embedded in the matrix[11]. For particle concentrations of interests, the shear modulus G of the elastomer filled with dispersed rigid particles can be calculated with acceptable accuracy by following function [10],

$$G = G_0\ (1 + 2.5\phi + 14.1\phi^2) \tag{1}$$

Here G_0 is the shear modulus of the unfilled elastomer, while ϕ is volume fraction of particles.

According to [8], the field induced modulus of MREs is in direct proportion to the magnetic intensity controlled by the electrical current intensity in the coil enlaced on the absorber. By assuming MRE is as same as the one in [8], corresponding to the elastomer with variant density of rigid parcels, the changeable range of the shear modulus ΔG can be represented by

$$\frac{\Delta G}{G(0)}(\max) = \frac{0.1911\ MPa}{G_0} \qquad \text{when } \phi = 0.266. \tag{2}$$

Here typically, G_0 is close to 1.1 MPa. Therefore, $\Delta G/G(0)(\max)$ could be roughly equal to 50%. Following this, the changeable range of the shear modulus of the MRE is located between 1.1MPa and 1.5MPa.

3. ATDVA

Suppose a mass-spring system or a primary system is excited by a harmonic driving force $P\sin\omega t$. When the driving frequency equals to the natural frequency of the system, the response is infinite. The TDVA is essentially a secondary mass attached to the primary system via a spring and a damper. When the natural frequency of the TDVA is tuned such that it coincides with the frequency of unwanted vibration in the primary system, the vibration of the primary system is greatly reduced. Thus, the energy of the primary system is apparently "absorbed" by the TDVA (Fig.1).

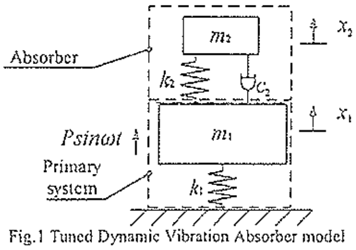

Fig.1 Tuned Dynamic Vibration Absorber model

The vibration reduction efficiency can be preparatory predicted by using a simplest model based on classical vibration theory. According to [12], a series of parameters are defined and shown in table 1.

Table.1 Parameters for building the mathematical model

$\omega_{n1} = \sqrt{\frac{k_1}{m_1}}$	Natural frequency of structure	$\omega_{n2} = \sqrt{\frac{k_2}{m_2}}$	Natural frequency of TDVA
$\varsigma_1 = \frac{c_1}{2\omega_{n1} m_1}$	Damping ratio of structure	$\varsigma_2 = \frac{c_2}{2\omega_{n2} m_2}$	Damping ratio of TDVA
$\mu = \frac{m_2}{m_1}$	Mass ratio	$r = \frac{\omega}{\omega_{n1}}$	Force frequency ratio
$g = \frac{\omega_{n2}}{\omega_{n1}}$	Natural frequency ratio	δ_{st}	Static deflection

Assume the stimulant force as $F = k_1 \delta_{st} e^{st}$, the transmissibility equation for the force-exited system can be expressed as following:

$$\frac{X_1}{\delta_{st}} = \frac{(g^2 - r^2 + 2\varsigma_2 grj)}{(-r^2 + 2\varsigma_1 rj + 2\varsigma_2 \mu grj + 1 + \mu g^2)(-r^2 + 2\varsigma_2 rgj + g^2) - \mu(2\varsigma_2 grj + g^2)^2} \tag{3}$$

As expressed in this equation, without changing the mass of the structure and the TDVA, the only method to change the displacement of the system is to adjust the spring and damper of the TDVA.

In this example, by assuming the engine as the primary system, and the TDVA replaced by the ATDVA, the vibration suppression can be achieved successfully by adjusting k_2 and c_2. Because the values of the k_2 is related to the values of the shear modulus of the MREs, changing the shear modulus of the elastomer could be an efficient method to adapt the requirements of the vibration suppression. As discussed in Section 2, since the shear modulus of the MREs can be adjusted from 1.1MPa to 1.5MPa by changing the flux, it is proposed that using MREs can directly develop a relationship between the flux and the natural frequency of the system. In the following section, based on the parameters of the MR Elastomer, a FEM is applied to examine the validation of the new ATDVA.

4. Finite Element Modeling and Analyzing

In order to examine the validation of the MRE semiactive vibration absorber, the engine 1.3L VVT2 is introduced in this paper. By respectively setting the passive tuned vibration adapter and the MRE semiactive vibration absorber on this engine in ANSYS simulation environment, the structural vibrations of the engine system are tracked and analyzed so as to demonstrate the effects of the proposed MRE vibration absorber.

4.3 Engine 1.3L VVT2 and Finite Element Modeling

Based on the properties of engine 1.3L VVT2, this simulation model is simplified as a block with overall dimension of 607×634×605 and mass of 138kg. Furthermore, because treating the source of the vibration is the most effective and often the most economical solution to vibration problems, it is proposed to isolate the engine from the car base by the engine mounts with rubber part. Normally, there are several pairs of engine mounts to fix the engine on the car base. Therefore, in the simulation model, the block representing the engine is attached by four rubber pads which are fixed in the simulation environment. Another function of the engine mounts is to isolate the car from the engine vibration in working status. The vibration transmissibility depends on disturbance frequency. Because a passive system is only effective for disturbance with frequencies much higher than its natural frequency, the first compulsory work is to set the natural frequency of the system much less than the rotation speed of the engine which is from 63Hz to 100Hz so as to eliminate the influence from the engine. In this example, by predefining the properties of the four rubber pads, the natural frequency of the system is located around 43Hz.

Table.2 properties of 1.3L VVT2

Model	Unit	Value
Number of Cylinders		4
Number of Valves		16
Displacement	cm3	1297.5
Rated Power/Speed	kW/rpm	65/6000
Max. Torque/Speed	Nm/rpm	118/3800
Fuel Quality	RON	93
Emission	Euro IV	
Overall Dimension(L×W×H)	mm	607×634×605
Net Weight	kg	138

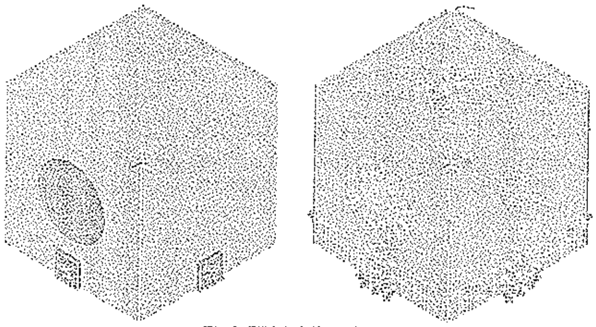

Fig.2 FE Modeling of the engine

As the basis for numerical parametric studies for further evaluation of the dynamic performance of the proposed new ATDVA, the baseline model is built as shown in Fig.2. In order to reduce the calculations without influencing the accuracy of the simulation, the element used to build this model is solid brick with 20 nodes; the meshing coefficient is set as '9' which is rough but acceptable.

On the purpose of observing the transmissibility of the engine model, a series of stimulant forces are added on this engine. By considering the working status of the engine, the forces from the engine rotation provide the vibration in the vertical (Z) and horizontal (X and Y) directions. In this simulation process, the frequency domain of the stimulant forces provided by the internal forces from the engine rotation are both set from 20Hz to 100Hz, while the phase difference between the two directions (X and Z) of the forces from the engine rotation is set as $\pi/2$ to simulate the rotary eccentric force.

By setting the frequency independent damping of the system as 0.05, the transmissibility of the engine block in the three orthogonal directions can be illustrated in Fig.3. In this figure, it is obvious that the most prominent vibration is in the vertical (Z) direction, while the others' amplitudes of the vibrations are much smaller. Hence, the focus of this FEM should be put on suppressing the vibration in the vertical direction. Because the amplitude of the vibration is directly related to the stimulant force, the transmissibility of the system could be described as $Amplitude/Force$ (m/N).

4.2 Tuned Vibration Absorber

In order to suppress the vibration in a continuous frequency domain, an ATDVA model is added on the engine model as shown in Fig.4. The materials of the ATDVA are referred to the requirements of the design. Most part of the ATDVA is meshed roughly, while the MR Elastomer is meshed in detail so as to guarantee the accuracy of the simulation.

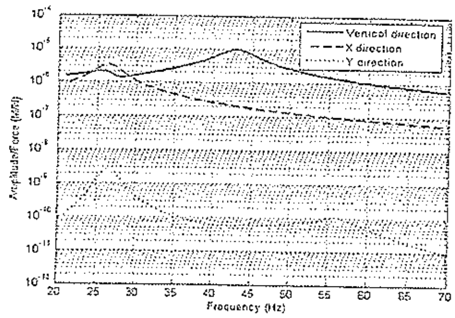

Fig.3 Transmissibility response of the engine

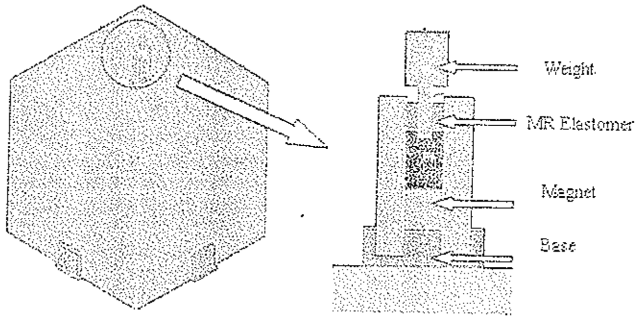

Fig.4 FE Modeling of the engine equipped with ATDVA

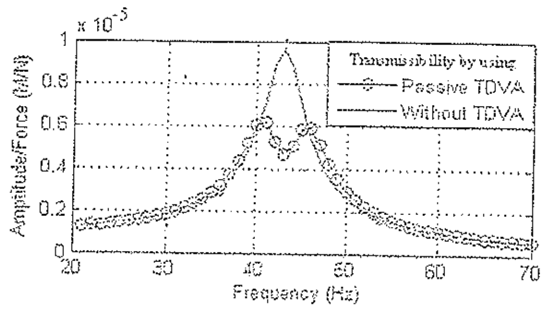

Fig.5 Effect of TDVA

Based on the discussion in Section 2 and 3, in this paper, the way to reduce the disturbance from the car base is to change the shear modulus of the MR Elastomer so as to keep the natural frequency of the ATDVA as same as the frequency of the stimulant forces. In this example, the mass of the ATDVA is set as 1% of the engine, while the shear modulus is set around 1.3MPa. The simulation results are illustrated in Fig.5.

As shown in this figure, there are two peaks around the Resonance frequency, and this passive TVA is only capable of reducing the amplitudes between the two peaks. In order to extend the capability of the TVA, it has been proposed that an Adaptive Tuned Dynamic Vibration Absorber (ATDVA) employing MREs must be designed and manufactured. The basic magnetic field is generated by a permanent magnet and the accessional magnetic intensity is controlled by the electrical current intensity in the coil. The induced magnetic field is imposed in the direction of particles' chains in MREs and it works at shear modulus. Hence, this absorber is capable of instantaneously changing its stiffness, thus it can switch between resonance frequencies, increasing its effective bandwidth as compared to classical TDVA for vibration control. Because a specific MR Elastomer with changeable shear modulus from 1.1MPa to 1.5MPa is applied in this simulation, the validation of the new ATDVA must be examined based on this restriction. Therefore, this simulation could be considered as achieving a function to study the correlation between the shear modulus of the MREs and the vibration amplitude. Following this, by increasing the values of the shear modulus of the MREs, the amplitudes of the system stimulated by a series of multi-frequency forces are derived as shown in Fig.6. By adjusting the values of the MREs' modulus according to disturbing force frequency, the effect of the new ATDVA is highlighted by comparing to the passive TVA in Fig.7.

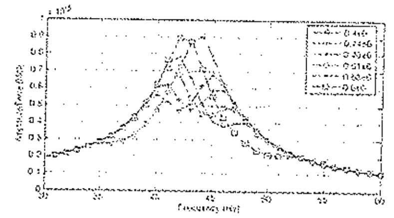

Fig.6 Vibration amplitudes corresponding to variant shear modulus of MREs

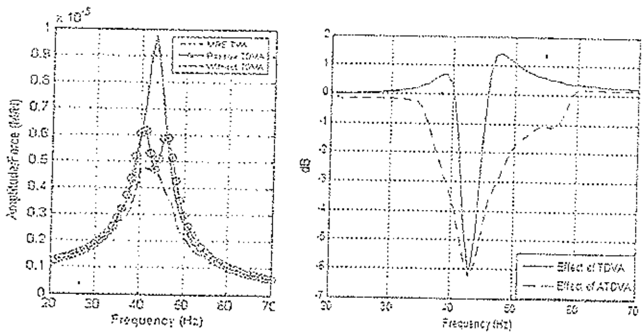

Fig.7 The transmissibility of the system with passive and Semiactive TDVA

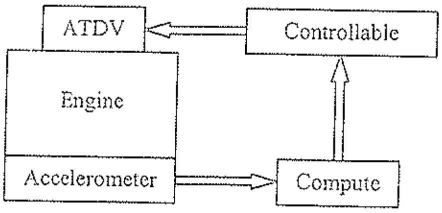

Fig.8 ATDVA System

As demonstrated in above figure, it is obvious that the new ATDVA performs much better than the passive TDVA whose shear modulus cannot be adjusted. In the frequency domain of interest which is from 35Hz to 50Hz, the vibration of the system cannot only be suppressed in the natural frequency of the system, but also can be suppressed in other frequencies. Therefore, it is valid to design and manufacture this new ATDVA and equip it on the engine for vibration suppression.

Based on the validation provided by simulation, the principle of the new ATDVA can be illustrated as in Fig.8. Detecting the vibration by the accelerometers mounted on the engine, a system will analyse the vibration information and provide control command to the controllable power so as to change the current intensity in coils which adjust the magnetic field intensity around the MREs. As discussed in Section 2 and 3, the ATDVA will vary the natural frequency of the system in variant vibration situations, so that the new system will not only achieve the vibration isolation at different frequencies but also fulfil the overall performance requirements.

5. Conclusion

In context of this paper, a new ATDVA employing MR Elastomer has been introduced to the engine vibration problem. By analyzing the principle of the MRE ATDVA, FEM is applied to evaluate the validation of this new vibration absorber. From the simulation results, there are several conclusions can be made as following.

Changing the magnetic intensity controlled by the electrical current intensity in the coil, the shear modulus of the MREs can shift in a reasonable range. By evaluating the analysis results of the simulation, it can be concluded that the elastic coefficient of the absorber can be continuously changed to shift the natural frequency of the ATDVA.

By tracking the frequency of the stimulant forces, the new ADTVA is capable of suppressing the vibration in a wider frequency domain. The FEM also provide a reliable evidence that the transmissibility of the whole system can be influenced greatly by shifting the shear modulus of the MRE in the practical range.

Finally, investigating the relationship between the shear modulus and the transmissibility of the engine system, the FEM provides a reliable trend that, in order to suppress the vibration of the engine, the higher the frequency of the stimulant force is, the larger the shear modulus of MREs is required. Thereupon, the electrical current intensity in the coil should be increased continuously so as to suppress the vibration of the system when the frequency of the stimulant force is increasing.

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