

Experimental Investigation and Semi-Active Control Design of A Magnetorheological Engine Mount

Seyed Salman Hosseini¹ and Javad Marzbanrad^{2,*}

¹School of Automotive Engineering, Iran University of Science and Technology, Tehran, Iran
²Faculty at School of Automotive Engineering, Iran University of Science and Technology, Tehran, Iran
*Corresponding Author: Javad Marzbanrad. Email: marzban@iust.ac.ir.

Abstract: In this paper; the dynamic characteristics of a semi-active magnetorheological fluid (MRF) engine mount are studied. To do so, the performance of the MRF engine mount is experimentally examined in higher frequencies ($50\sim170$ Hz) and the various amplitudes ($0.01 \sim 0.2$ mm). In such an examination, an MRF engine mount along with its magnetically biased is fabricated and successfully measured. In addition, the natural frequencies of the system are obtained by standard hammer modal test. For modelling the behavior of the system, a mass-spring-damper model with tuned PID coefficients based on Pessen integral of absolute error method is used. The parameters of such a model including mass, damping ratio, and stiffness are identified with the help of experimental modal tests and the recursive least square method (RLS). It is shown that using PID controller leads to reducing the vibration transmissibility in the resonance frequency (=93.45 Hz) with respect to the typical passive engine mount by a factor of 58%. The average of the vibration transmissibility decreasing is also 43% within frequency bandwidth ($50\sim170$ Hz).

Keywords: Semi-active engine mount; PID controller; experimental data; least square method; magnetorheological fluid

1 Introduction

Reducing noise, vibration and harshness (NVH) of passenger cars is one of the most important challenges. Engine vibrations are one of the NVH sources of vehicles. Since recent designs go toward cars with high speed along with the large power-intensive engine, therefore, NVH problem must be taken into account more than ever before. For solving this problem, the use of smart materials is common in new car designs. The magnetorheological fluid is a well-known type of these materials. It has widespread use in new car designs. Several parts of a car including suspensions, brakes and engine mounts use a magnetorheological (MR) fluid [1-3]. These parts with MR can significantly reduce the NVH. One of the important parts in reducing NVH is the engine mounts. The typical hydraulic engine mounts cannot effectively isolate and reduce transferred engine to body vibrations at high engine revolutions per minute (RPM) [4]. Using Semi-Active Controlled Engine Mount (SACEM) is an acceptable method to reduce NVH effect [5]. Such a system has been used for isolating the chassis from the high-frequency engine excitation. By high-frequency, we mean idle speed of 600 RPM for a maximum one of 6000 RPM corresponded to sinusoidal excitation at 20~200 Hz [6]. Implementation of SACEM needs a material, which its viscosity can be changed. In [7], it has been used as a magnetorheological (MR) fluid

controlled by varying the applied magnetic field. Increasing the intensity of the applied magnetic field leads to enhance the viscosity of MR and hence the critical frequency of SACEM dynamically varies [8, 9]. To control the magnetic field, a magnetic coil is required in which the electrical current is varied [10-13]. The maximum value of coil current is less than 3 A [14, 15]. To further understanding the performance of semi-active engine mounts, the various models have been proposed in the literature [16, 17].

One of the well-known models is mass spring damper. This model has been used in the design of a prototype engine mount in [17-19] and showed that such a model can be accurately described dynamic characteristics of the engine mount. For the online identification of these model parameters, one can use the Recursive Least Squares (RLS) method [20]. Based on the obtained model, the transmitted force from the engine to the body may be attenuated using the PID controller due to its simple implementation [21]. However, it is desired to design an appropriate PID controller for the semi-active engine mounts. Therefore, it is needed for a systematic approach to the design of such a structure.

In this paper, it has been firstly fabricated an MR engine mount with an embedded coil based on a common engine mounts. Such a structure is then tested and its vibration amplitude and frequency are measured. After that, the obtained measurements are used in the identifying of the parameters of the Mass-Spring-Damper model. Next, using the achieved model, a PID controller is designed. Finally, it is shown that the implemented MR engine-mount along with the designed PID controller has vibrations with low-level amplitude.

Firstly, the test process and data acquisition device are presented. Thus, semi-active mount design and experimental controlling setup are described. In addition, proposes the system modelling and PID controller design has been proposed.

2 System Modelling Approach

In order to model the engine mount, the mass-spring-damper model with a base excitation was used as shown in Figure 1. The schematic diagram of this model is as follows.



Figure 1: Mass-spring- damper system with base excitation

The model of this system is as follows:

$$m\ddot{x} + c\dot{x} + kx = c\dot{y} + ky \tag{1}$$

where k is the final stiffness, c is the equivalent damping coefficient, and m is the mass of the engine mount. The damping consists of two parts, including Intrinsic damping of magnetorheological fluid (MRF) and damping due to the magnetic field. The total damping is the sum of these parts:

$$c = c_{MR} + c_{field} \tag{2}$$

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2.1 Parameters Estimation

In order to estimate the system parameters, the difference of squares identity method has been used. To this end, discretization was performed using the finite-difference method (FDM) as stated in Eqs. (3-a) and (3-b):

$$\ddot{x} = \frac{x(k-2) - 2x(k-1) + x(k)}{T^2}$$
(3-a)

$$\dot{x} = \frac{x(k-1) - x(k)}{T}$$
(3-b)

Therefore, by substituting the Eqs. (3-a) and (3-b), the Eq. (1) can be written as follows:

$$m\left(\frac{x(k-2) - 2x(k-1) + x(k)}{T^2}\right) + c\left(\frac{x(k) - x(k-1)}{T}\right) + kx(k) = c\left(\frac{y(k) - y(k-1)}{T}\right) + ky(k)$$
(4)

by considering the system model in the $y = \sum_{i=1}^{\varphi} \theta$ form, the recursive least square (LQR) method can be used to identify the system parameters. The regression form of Eq. (4) is as follows:

$$\varphi_1 x(k-2) + \varphi_2 x(k-1) + \varphi_3 x(k) + \varphi_4 y(k-1) = y(k)$$
(5)

with a simple mathematical manipulation, the values of ϕ_i , i = 1, 2, 3, 4 are given as follows:

$$\varphi_1 = \frac{m}{T^2 \left(k + \frac{c}{T}\right)} \tag{6-a}$$

$$\varphi_2 = \frac{\frac{-2m}{T^2} + \frac{c}{T}}{k + \frac{c}{T}} \tag{6-b}$$

$$\varphi_3 = \frac{\frac{m}{T^2} - \frac{c}{T} + k}{k + \frac{c}{T}} \tag{6-c}$$

$$\varphi_4 = -\frac{c}{T\left(k + \frac{c}{T}\right)} \tag{6-d}$$

The sampling rate of the sensor is 20,000 data per second; therefore, T is considered as sampling time. By simplifying the parameters, their relation can be expressed as:

$$\frac{\varphi_4}{\varphi_1} = \frac{cT}{m} \tag{7}$$

The RLS equations are as follows:

$$k(n) = \frac{\lambda^{-1} P(n-1) u(n)}{1 + \lambda^{-1} u^T(n) P(n-1) u(n)}$$
(8-a)

$$\varepsilon(n) = d(n) - w(n-1)u(n) \tag{8-b}$$

$$w(n) = w(n-1) + k(n)\varepsilon(n)$$
(8-c)

$$P(n) = \lambda^{-1} P(n-1) - \lambda^{-1} k(n) u^{T}(n) P(n-1)$$
(8-d)

In Eq. (8), the input is the displacement of the vibrator and the output is the displacement of the top section of the engine mount.

3 Materials and Methods

The experimental setup consists of a vibrator, the fixture system, sensors, and an electronic control unit. In the following, these parts are described.

3.1 Test Setup

The vibrator is the source of vibrations in several frequencies. This system is capable to produce frequencies up to 200 Hz. The top side of this vibrator is fixed and is used for applying the preload. The applied preload used for performing this test is about one-third of the weight of the engine, i.e., about 700 N. The shaker has been installed in the middle section of this setup. This setup is shown in Fig. 2. In order to record the displacement of the engine mount during testing, an accelerometer sensor (KS74C100 with less than 2% accuracy) has been mounted on the top side of the engine mount.



Figure 2: The test setup

3.2 Control Unit and Sensors

The biased coil and MRF are embedded into the engine mount and the control unit is attached to the coil. The coil is embedded in the inner part of a hydraulic engine mounts and fill with MR fluid. The fluid used for experiments is MRF-132 DG made by Lord Company. Fig. 3, shows the fabricated engine mount and its control unit.

The accelerometer is KS74C100 and the data acquisition module is M302 designed by Manfred Weber Metra Mess Company. The properties of the sensor and acquisition module are given in Tabs. 1 and 2 respectively.

Figure 4 shows the acceleration recorded with this sensor in the 100-millisecond duration. The velocity is the integral of these data, which a sample of these data is shown in Fig. 5. By integrating again, the displacement. Figure 6 shows the displacement of the engine mount. Frequency analysis based on Fast Fourier Transform (FFT) was used to reveal the spectral properties of data. Figure 7 shows the power spectral of engine mount displacement.



Figure 3: Designed engine mount and control box

 Table 1: Characteristics of accelerometer

| Sensitivity | 100 mV/g |
|--------------------------------|---------------|
| Measuring range | \pm 60 g |
| Linear frequency range (±3 dB) | 0.13-22000 Hz |
| Weight | 32 g |

 Table 2: Characteristics of the data acquisition module

| Input | 2 IEPE; 1 digital trigger |
|-----------------|---------------------------|
| Frequency range | 0.3 to 2000 Hz (-3 dB) |
| Accuracy | < 2% |



Figure 4: Acceleration of the engine mount



Figure 5: Velocity of the engine mount



Figure 6: Displacement of the engine mount



Figure 7: Power spectral of the displacement

3.3 Test Procedure

The engine mount was excited in seven frequencies with eight controlling conditions. The identification of mount properties needs rich wave periodic displacement excitation [22]. A sinusoidal excitation signal $u(t) = \sin(\omega_0 t)$ is used. This signal is a persistent exciting signal of the second-order and contains different frequencies and amplitudes. The excitation signals are 50, 70, 90, 110, 130, 150 and 170 Hz. The domain of excitation decreases with the increasing of the frequencies. Table 3 shows the frequencies and domains of the excitation signals and the dynamic stiffness of the engine mount.

Figure 8 shows the base excitation vibration in the produced by test setup. The vibration domain at 50 Hz is about 180 μ m. By increasing the frequency, the domain of excitation reduced. At 170 Hz the domain of excitation reached 10 μ m.

| Excitation frequency (Hz) | Excitation domain (mm) | Dynamic stiffness (KN/m) |
|---------------------------|------------------------|--------------------------|
| 50 | 0.18 | 419.1 |
| 70 | 0.16 | 421.2 |
| 90 | 0.14 | 440.9 |
| 110 | 0.1 | 437.4 |
| 130 | 0.06 | 443.3 |
| 150 | 0.03 | 490.1 |
| 170 | 0.01 | 483.4 |

Table 3: Dynamic characteristics of excitation



Figure 8: The diagram of the base excitation signal of the engine mount



Figure 9: Damping of the system in different currents

The system parameters are estimated with the Recursive Least Square (RLS) method. RLS is a common method for the identification of structures subjected to forces and multiple disturbances [23]. Figure 9 shows the system damping ratio which estimated with RLS method in different currents.

3.4 Determination of Natural Frequency

The natural frequency of the system is identified with a standard hammer impact test. This method has several advantages including easy excitation to high frequencies, the good definition of force in the striking direction without transverse forces [24]. The hammer test setup consists of an impulse hammer, accelerometer, signal acquisition device, and fixtures. Figure 10 shows the schematic view of the hammer test setup.



Figure 10: Schematic of the hammer test setup



Figure 11: Logarithmic decrement of vibration in the impulse excitation

In the impulse test, the damping ratio is estimated from logarithmic decrement. The logarithmic decrements of the domain can be shown as a push curvature in Fig. 11. The first maxima and second maxima are shown in Fig. 11. By considering two maxima, the decay ratio is:

$$\frac{x_n}{x_0} = e^{-2\pi n} \frac{\xi}{\sqrt{1 - \xi^2}}$$
(9)

Where ξ is the damping ratio:

$$\xi = \frac{c}{2m\omega_n} \tag{10}$$

where *m* and ξ are the intrinsic constant properties of the. The logarithmic decrement Δ for the small value of ξ is:

$$\Delta = \ln \frac{x_1}{x_2} = \frac{2\pi\xi}{\sqrt{1-\xi^2}}$$
(11)

When ξ is small it can be approximated from Eq. (12) as follows:

$$2\pi\xi = \frac{\pi c}{m\omega_n} \tag{12}$$

By using the Eq. (13) the natural frequency can be written as:

$$\omega_n = \frac{\pi c}{m\Delta} \tag{13}$$

The first maximum in the test are the 2951 μ m and the second is 1440 μ m. By using Eq. (11), the natural frequency of the system is 93.45 Hz. The transmissibility ratio is as follows:

$$Tr = \sqrt{\frac{1 + \left(2\zeta\left(\frac{\omega}{\omega_n}\right)^2\right)}{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_n}\right)\right)^2}}$$
(14)

The maximum of this function is at $\xi = 0$. by increasing the ξ the transmissibility is reduced.

4 Controller Design

PID controller is the most common controller used for industrial process control applications. To find the set point for PID controller, one may find the maximum ξ in the excitation frequency. So, for the experimental dataset, a surface has been fitted. The fitted surface is shown in Fig. 12. The equation of this surface is:

$$C = -791.6 + 26.13I + 27.03f + 4.601I^{2} + 0.533fI - 0.26f^{2} - 12.29I^{3} + 0.3672I^{2}f - 0.0008If^{2} + 0.0007f^{3}$$
(15)

where *C* is the mount damping ratio, *f* is excitation frequency and *I* is the coil amps. The current unit is Amps and the frequency is Hz. The applied current must be selected in such a way that $\frac{\delta C}{\delta I} = 0$. The derivation of the damping with respect to frequency is obtained as follows:

 $26.13 + 9.202I + 0.533f - 36.87I^2 + 0.7344If - 0.0008f^2 = 0$ ⁽¹⁶⁾



Figure 12: System damping ratio and fitted surface

by solving this equation in the excitation frequency, the desired current has been obtained:

$$I_{des} = 0.01 \left(f - \sqrt{\left(78.61f + \left(0.73f + 9.2\right)^2 - 0.12f^2 + 3853.65\right)} \right) + 0.12$$
(17)

The error is defined as the difference between the coil's current and desired current. The PID controller is:

$$e = I_{coil} - I_{des} \Rightarrow I = K_p e + K_i \int^e dt + K_d \frac{de}{dt}$$
(18)

The stability of the system is determined by the step response. Therefore, the system is simulated with the estimated parameters and step response within the test bandwidth (50~170 Hz) and current range from 0 to 3 A. Because of technical constraints, the maximum current is set to three Amps. Figure 13 shows the system step response in the frequency of 70 Hz. The critical time and ultimate gain are calculated from the step response in each frequency and current. The Pessen integral of the absolute error method was used for tuning the controller gains [25]. The gains in the other currents are interpolated. In addition, at frequencies 70 and 90 Hz, the gains manually re-tuned and the best performance and corresponding gain are chosen. The system transmissibility is shown in Fig. 14. In the current less than 1.5 A, the system is underdamped and has a resonant. In the next currents up to three Amp, the system is overdamped and no resonant frequencies occurred. The PID controller has a lower transmissibility ratio in the near resonance frequencies and the system is overdamped. The controller minimizes the transmissibility ratio.



Figure 13: Step response of system in 70 Hz



Figure 14: The transmissibility ratio in the mount with the controller

5 Conclusion

In this paper, a magnetorheological semi-active engine mount is investigated. First, a magnetorheological mount is designed. To identify, mount parameters, vibration tests in several frequencies and domains are done. In addition, to determine the system intrinsic properties an impulse test is done. The parameters are identified by the recursive least square method. A surface is fitted on these parameters by using the curve-fitting method and the desired current for each frequency is

calculated. To control the mount damping property and obtaining the minimum transmissibility, a PID controller is designed based on the passing integral of the absolute error method. In addition, technical constraints in applying the coil current is considered. By implementing the controller, the results show that this controller can reduce the vibration in the resonant area by 58% and approximately 43% in high frequency (>100 Hz) domain. By using this mount, the high-frequency vibrations in the resonant area and after that reduced about 43% and it improves ride comfort in the passenger car.

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