Simulation of the dynamic behavior of the magneto-rheological engine mount for automotive applications

T. Feyzi*,1, M. Esfahanian2, R. Tikani3 and S. Ziaei-Rad4

[1] MSc student, Department of Mechanical Engineering, Isfahan University of Technology, Isfahan Iran.
[2, 3, 4] Assistant Professor, Department of Mechanical Engineering, Isfahan University of Technology, Isfahan Iran.

* t.feizi@me.iut.ac.ir

Abstract

Increasing current vehicle development trends for small, light, front wheel drive vehicles with low idle speeds have been forced automotive industries to use hydraulic engine mounts for further improvement in Vibration, Noise, Vibration and Harshness (NVH) performance of their vehicles. However, with the development of modern vehicle designs such as hybrid vehicles and variable engine management systems which have different operational modes, more sophisticated engine mounting systems are required to effectively response to the each operational mode. The MR engine mount is a semi-active hydraulic engine mount, containing MR fluid, that can alter its dynamic behaviour as a result of applying a magnetic field. In this paper, the modeling of a new semi-active MR engine mount has been conducted. It was shown that by increasing the magnetic field, the dynamic stiffness of the engine mount can be changed.

Keywords: Hydraulic engine Mount, MR fluid, Dynamic stiffness, Vibration isolation

1. INTRODUCTION

The vehicle engine mounting system serves three principal functions:

- Supporting the engine weight
- Isolation of the engine vibrations (caused by engine eccentricity during its operation)
- Preventing engine bounce and excessive motions (mostly caused by rough roads, idle shake, vehicle acceleration and deceleration)

To isolate the vibration caused by the engine-unbalanced disturbances (occurring in a relatively high frequency range with low amplitude), low elastic stiffness and damping is needed as the forces transmitted to the structure are proportional to the stiffness and damping of the mounts. However, to prevent engine bounce (usually occurring in the low frequency range with high amplitude and shock excitations) engine mounts should simultaneously have high stiffness and damping [1].

Ordinary engine mounts, such as elastomeric and passive hydraulic engine mounts, can easily serve the first function. Since the second and third functions are conflicting, an engine mount with amplitude and frequency dependant dynamic characteristics is needed. Elastomeric mounts cannot serve the second and third functions simultaneously, because their dynamic characteristics are almost independent with excitation frequency. Moreover, modern car design trends for lighter bodies and more power-intensive engines which adversely affect vibratory behaviour and increase the vibration and noise level, has forced the automotive industry to turn increasingly to passive hydraulic engine mounts.

A typical hydraulic mount with decoupler (Fig. 1) requires incorporation of the following features [1]:

- An elastomeric mount capable of supporting the load and acting as a piston to pump the liquid into the button chamber.
- Two separate chambers for fluid transfer.
- An orifice or inertia track to generate damping.
- A fluid medium.
- Sealing between chamber and the outside.
- Decoupler to permit low amplitude by-pass of the damping.

For the high frequency and low amplitude vibration, the fluid flowing between chambers can easily pass through the decoupler, therefore the dynamic stiffness and damping of the hydraulic mount is reduced and the mount acts like an elastomeric mount. Hence, the mount shows good isolation performance. For the large input amplitude and shock

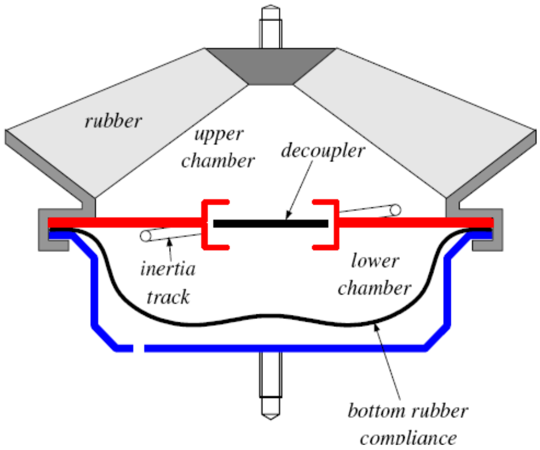

Fig. 1. Schematic of a passive hydraulic engine mount

excitations, the decoupler bottoms on its seats terminating the flow of liquid around it. Therefore, the flow goes through the inertia track which creates additional damping and increases dynamic stiffness. However, passive hydraulic engine mounts cannot resolve all the problems that arise during vehicle operation.

Ushijima and Takano [3] reported that conventional hydraulic mount displays excellent characteristics when subjected to simple sinusoidal inputs. But, this type of mount is not practical for superimposed inputs because of the significant nonlinearity of the decoupler. Moreover, a hydraulic mount is yet a passive element that is efficient only within a limited range of operation. Emerging automotive technologies, i.e., electric hybrid, hydraulic hybrid and variable cylinder management, which have various operational modes, require the mounts that can alter their characteristics during each mode of operation [4].

To solve some problematic aspects of the passive hydraulic mounts and to improve their dynamic performance further, adaptive vibration control techniques have been applied to engine mounting designs. Adaptive mounts can be active or semi-active. Semi-active mounts are more common because of their simple design and lower cost. Semi-active mounts can alter their dynamic characteristics (stiffness and damping) for effectively responding to the different input excitations during operation.

Semi-active control can change the dynamic response of the system through controlling system parameters. The controlled parameters for a semi-active engine mount system can be stiffness and damping. Vahdati and Ahmadian [5] proposed a new design concept for semi-active hydraulic mounts through mathematical modelling and simulation. Their study focused on changing the mount dynamic characteristics by affecting the bottom chamber's volumetric stiffness. Foumani et al. [6] embedded shape memory alloy wires inside the rubber spring of the mount to make its compliance variable. They could alter the dynamic characteristics of the mount by switching between maximum and minimum values of the upper chamber compliance.

Currently, semi-active mounts rely on changes in geometry of the flow paths or on changes in properties of the working fluid. As changes in geometry of the flow paths in real time require rather complex actuation mechanisms to be incorporated in the mount, it is more desirable to be able to change the mount response through changes in the working fluid characteristics. Consequently, semi-active fluid mounts have been proposed to use electro-rheological (ER) or magneto-rheological (MR) fluids as the working fluid. These kinds of fluids can change their rheology upon the application of an electric/magnetic field. It has been proven that MR fluids develop higher yield stress compared with ER fluids. Therefore, MR fluids are more suitable for applications that require high levels of energy dissipation.

MR fluids often work in one of the three modes: flow (or valve) mode, shear mode and squeeze mode. The most popular type is the flow mode since it is somewhat simpler to design devices making use of the fluid working in that mode. The flow mode happens when the fluid flows between two fixed parallel boundaries that are perpendicular to the direction of the applied field [7].

Hong et al. [8] introduced an ER mount capable to support a static load of 70 kg, with the fluid working in flow mode. Baudendistel et al. [9] patented a fluid mount with annular fluid flow paths running between a reservoir and a pumping chamber. Carlson and Jolly [7] introduced the MR fluid and its applications to vibration isolation. They also compared MR fluids to ER fluids. Stelzer et al. [10] proposed a compact MR isolator to mount the air conditioning compressor in a vehicle based on the flow mode. Choi et al. [11] presented a mixed-mode MR engine mount that has the MR fluid operate in flow and shear modes simultaneously.

This study focuses on the modelling and simulation of two designs for MR engine mounts. The main idea of the designs was first introduced by Barber and Carlson [12]. However, they neither proposed any mathematical model for their design nor they did any simulation on them. The first design contained a simple MR valve while in the second; the mount utilizes both an MR valve and an inertia track. In this paper, first, the performance concept of two mounts is explained. Then, mechanical models and the governing equations of the mounts are constructed. Finally, the simulation of the mathematical equations is carried out and the effectiveness of the mounts is validated theoretically.

2. DESIGN

Schematic design of the MR mount with MR valve and inertia track is illustrated in Fig. 2. The operational concept of these MR mounts is similar to a passive hydraulic mount. It is composed of two rubber components (the rubber spring and rubber bellow) and two fluid chambers like a passive hydraulic engine mount. The chambers are filled with MR fluid which contains micron-sized iron particles.

As illustrated in Fig. 3, in the presence of magnetic field, the iron particles form chain clusters that develop yield strength and change MR fluid from a free-flowing liquid to a semi-solid. MR fluid flows back and forth between two chambers through two separate paths, inertia track and MR valve (inertia track does not exist in the first design). Inertia track, just like in a passive hydraulic mount, is a long narrow track with high inertia and resistance against fluid flow which provides a large stiffness and damping at low frequency and large amplitude excitations. MR valve is an annular flow gap which has more cross sectional area, less length and therefore less inertia and resistance against fluid flow than inertia track. Magnetic field is applied on the MR fluid contained in the MR valve by means of a coil embedded around the MR valve.

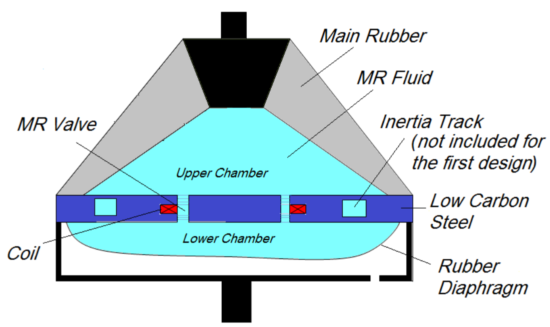

Fig. 2. Schematic of the MR engine mount

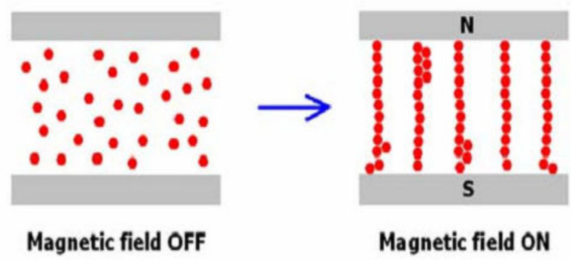

Fig. 3. MR fluid with and without magnetic field

The design of the MR mount is so that it can provide more stiffness and damping to prevent engine bounce from shock excitations and minimize excessive engine motion at low frequency and large amplitude vibrations. Furthermore, it can provide less stiffness and damping to obtain a low transmissibility and hence isolate engine vibrations from vehicle frame at high frequency and low amplitude vibrations of the engine.

At high frequency and low amplitude excitations, magnetic field is not applied (off-state) and MR fluid has the lowest viscosity and can easily pass through MR valve and inertia track. In this situation, most part of the fluid flows through the MR valve rather than inertia track because of its lower inertia and resistance. Therefore, dynamic stiffness and damping of the MR mount is decreased and hence isolation performance of the mount is enhanced.

At low frequency and large amplitude or shock excitations, magnetic field is applied on the MR fluid by means of the coil and heightens the resistance of the fluid in the MR valve. Therefore, the MR valve resistance against the fluid flow is intensified and hence the part of fluid passing through the MR valve is decreased. The higher the magnetic field strength, the higher the MR valve resistance, the lower the amount of fluid passing through the MR valve and adversely the higher the part of fluid passing through the inertia track. At a critical magnetic flux, the MR valve is essentially closed and almost all of the fluid flowing between chambers passes through the inertia track. At this point the MR mount shows the behaviour of a classic hydraulic engine mount with an inertia track and due to high inertia and resistance of the inertia track, dynamic stiffness and damping of the MR mount is increased over a narrow frequency range.

3. MODELLING

The mechanical model of the mount, as seen in Fig. 4, is used to derive the governing equations for both the MR fluid mounts.

Fig. 4 Mechanical model of the MR mount

Under displacement excitation, x(t), the continuity equations for the fluid flowing between chambers through the inertia track and the MR valve are given by:

$$C_1\dot{P}_1 = A_p\dot{x} - Q_1 - Q \tag{1}$$

$$C_2\dot{P}_2 = Q_1 + Q_2 \tag{2}$$

where C_1 and C_2 are the volumetric compliances of the upper and lower chambers, P_1 and P_2 are the pressures in the upper and lower chambers, Q_1 and Q_2 are the fluid volume fluxes through the inertia track and MR valve, respectively and A_p is the effective piston area of the rubber spring. The pressure drop due to the flow of the MR fluid through the inertia track and MR valve of the mount are evaluated from the linear momentum equation as:

$$P_1 - P_2 = I_1\dot{Q}_1 + R_1 Q_1 \tag{3}$$

$$P_1 - P_2 = I_2\dot{Q}_2 + R_2 Q_2 + \Delta P_{MR} \tag{4}$$

where I_1 and I_2 are the fluid inertias in the inertia track and MR valve, R_1 and R_2 are the resistances of the inertia track and MR valve against fluid flow, respectively and ΔP_{MR} is the pressure drop due to the yield stress of the MR fluid. According to Srinivasan et al. [12], the pressure difference induced by the MR effect can be expressed as:

$$\Delta P_{MR} = C\frac{L}{h}\tau_y(H)sign(\dot{x}_i) \tag{5}$$

where C is a constant value in the range of 2 to 3 depending on the steady state flow conditions. In this study, it is assumed that C=2, which corresponds to low flow conditions. L is the length inside the MR valve where the magnetic field is effective, as illustrated in Fig. 2, h is the distance between the magnetic poles, which is equal to the height of the MR valve channel, $\tau_y(H)$ is the MR fluid yield stress that is magnetic field strength (H) dependant. MRF-132LD is used in this study [14], the yield stress for this type of fluid is assumed to be a function of the magnetic field strength as follows [15]:

$$\tau(H) = 1.93H^{1.73} \tag{6}$$

The transmitted force to the base of the mount is obtained from [16]:

$$F_T(t) = K_r x + B_r\dot{x} + A_p P_1 \tag{7}$$

where K_r and B_r are dynamic stiffness and damping properties of the rubber spring, respectively. The complex stiffness of the mount at an excitation frequency of ω_0 is expressed as [16]:

$$K(s) = \frac{L(F_T(t))}{L(X(t))}\Big|_{\omega=\omega_0} = K_s + jK_l \tag{8}$$

where L represents Laplace transform and $s=j\omega$. K_s is the storage stiffness and K_l is the loss stiffness. Dynamic stiffness K_d and loss angle ϕ are defined as:

$$K_d = \sqrt{K^2{}_s + K_l{}^2}\,, \qquad \phi = \arctan(\frac{K_l}{K_s}) \tag{9}$$

It is noteworthy that the time domain analysis is important especially in the presence of the large engine motions. But in this paper and also many other similar works [2, 4-6], the dynamics of the model has been discussed by using dynamic stiffness parameter, that is the complex stiffness of the mount. Therefore, by calculating the stiffness and damping of the system, one may simulate the dynamic behaviour of the structure in the time domain.

4. SIMULATION RESULTS

A MATLAB program, with the parameters listed in Table 1 for the MR mounts, was used to simulate

Table 1. parameters used for the simulation

parameter	value	parameter	value
C_1 (mm^5/N)	3.8×10^4	I_2 (Ns^2/mm^5)	6.83×10^{-11}
C_2 (mm^5/N)	3.3×10^6	R_1 (Ns/mm^5)	1.81×10^{-7}
A_p (mm^2)	2500	R_2 (Ns/mm^5)	1.92×10^{-8}
B_r (Ns/mm)	0.1	K_r (N/mm)	220
h (mm)	3	L (mm)	10
I_1 (Ns^2/mm^5)	7.93×10^{-9}		

dynamic properties of the mount by using equations (1-9).

The variation of dynamic properties (dynamic stiffness and phase angle) of the MR mount with the magnetic field strength is illustrated in Fig. 5. When no magnetic field is applied (off-state), MR fluid can easily pass through the MR valve and inertia track without any significant resistance against its flow. Therefore, the dynamic stiffness is low over a wide frequency range and the phase angle is increased gradually (Fig. 5 graph 1). At the off-state, the MR mount shows a good isolation performance due to its low dynamic stiffness and phase angle (damping) and can be used to isolate engine vibrations at high frequency and low amplitude excitations.

In this situation, the part of the fluid passing

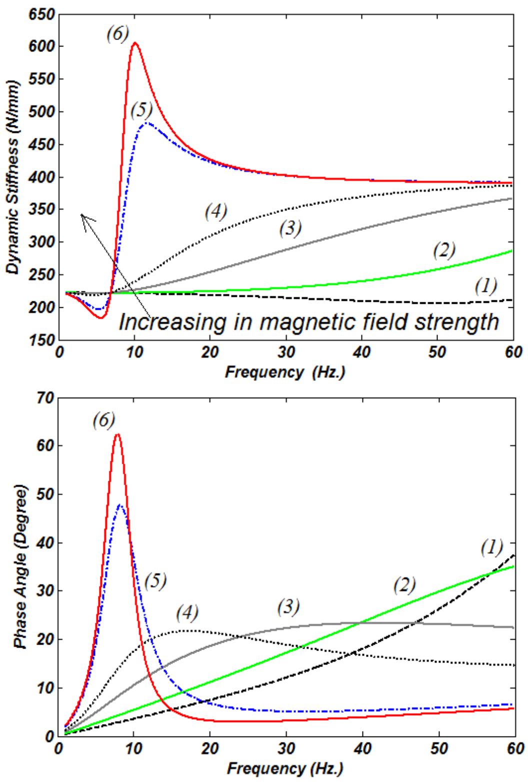

Fig. 5. Variation of the dynamic characteristics of the second design of MR mount with applied magnetic field strength (a) dynamic stiffness (b) phase angle

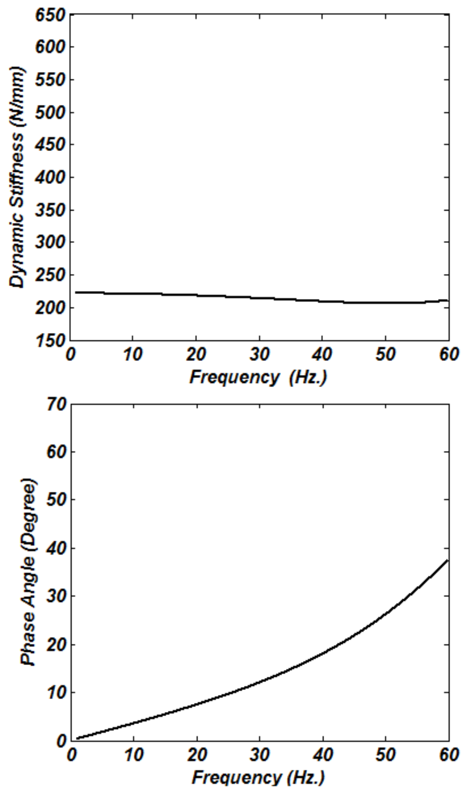

Fig. 6. Dynamic behaviour of a passive hydraulic engine mount with an orifice the same as the MR valve (a) Dynamic stiffness (b) phase angle

through the MR valve is more significant than inertia track and the fluid flow through the inertia track can be neglected. As illustrated in Fig. 6, this assumption is true. It is apparent from Fig. 6 that the dynamic stiffness and phase angle of a passive hydraulic engine mount with an orifice having the same inertia and resistance properties (I and R) as the MR valve, and the MR mount at off-state (Fig. 5 graph 1) are unique.

When the magnetic field is applied the resistance of the MR fluid contained in the MR valve is increased.

Due to increased resistance of the MR valve as a result of applying magnetic field, the dynamic stiffness and phase angle of the mount is increased compared with off-state (Fig. 5 graph 2). With more increase in magnetic field, MR valve resistance increases more and therefore, as it is obvious from graphs 3 and 4 in Fig. 5, the MR mount shows higher dynamic stiffness and phase angle over a wide frequency range. When the magnetic field strength exceeds a specific value, the part of the fluid flowing through the inertia track becomes more significant than MR valve due to enormous resistance of the MR valve as a result of extremely high amount of applied magnetic field. So, as it is apparent from graphs 5 and 6 in Fig. 5, the initial dynamic stiffness of the mount decreases to a minimum at a relatively low frequency (the low-stiffness notch) and the damping reaches a maximum due to resonant oscillation of the fluid mass within the inertia track. At higher frequencies the mount had high stiffness and low damping much like a simple elastomeric mount due to blocking of the inertia track by the fluid at high frequencies. At a critical magnetic field strength, MR valve is completely blocked and the MR fluid passes entirely through the inertia track (graph 6 in Fig. 5). In this condition the dynamic stiffness at frequencies below the idle speed is increased 2.5 times more than the static stiffness. At this situation, the MR mount acts as a passive hydraulic engine mount with an inertia track having the same inertia and resistance properties as the inertia track properties of the MR mount, as illustrated in Fig. 7.

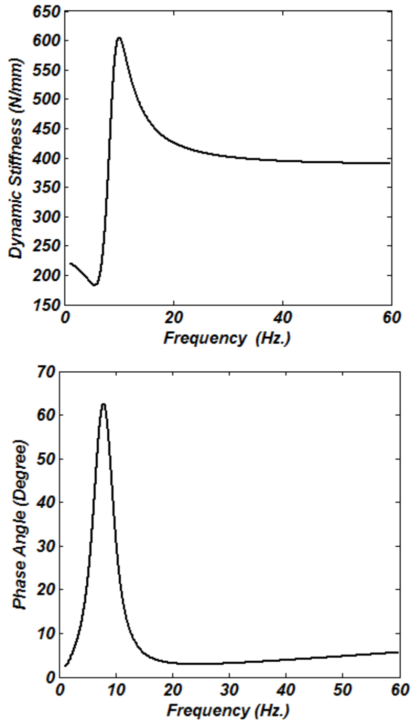

Fig. 7. Dynamic behaviour of a passive hydraulic engine mount with an orifice the same as the inertia track of the MR mount (a) Dynamic stiffness (b) Phase angle

The control task for the proposed design can be done by using a solid-state relay (SSR) which can control the close and open state of switch according to the input signal coming from the controller (Fig. 8). The accelerometer can be put on the lower part of the mount. Once it gives a high voltage which means a large motion situation occurring, switch will be closed and the current will be applied to the magnetic coil; thus dynamic stiffness of the mount increases and the

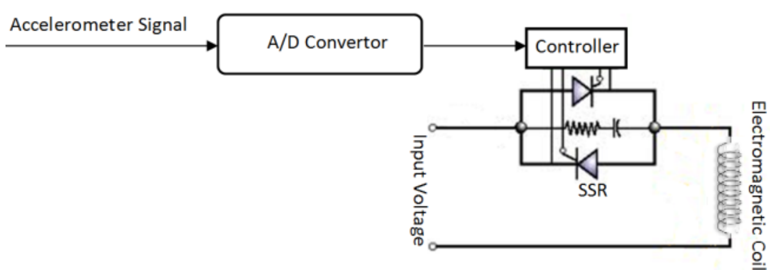

Fig. 8. A schematic diagram for control of the MR mount

engine stays in its place rigidly. Under normal conditions, when accelerometer gives low voltage signal, the switch is open and no current will be applied to the magnetic coil. Therefore, the mount acts like a passive hydraulic engine mount.

4. CONCLUSION

In this study, the mathematical model of a semi-active MR mount (with an MR valve and an inertia track) was derived. The resulting system of equations of motion was constructed in a MATLAB program to simulate the behaviour of the mount. From the simulation results, it was proven that the mount could alter its dynamic stiffness and damping via applying a magnetic field and showed a highly tuneable response. Also, it was shown that the MR mount can alter its dynamic characteristics from a low stiffness and damping mount, which is suitable for the isolation of vibrations at higher frequencies, to a high stiffness and damping mount, which is appropriate for low frequency and shock excitations to prevent engine bounce.

REFERENCES

[1] Y. Yunhe, G. N. Nagi, V. D. Rao, “A literature review of automotive vehicle engine mounting systems”, Mechanism and Machine Theory, vol. 36, pp. 123-142, 2001.

[2] J. Christopherson, G.N. Jazar, “Dynamic behaviour comparison of passive hydraulic engine mounts. Part 1: Mathematical analysis”, Journal of Sound and Vibration, vol. 290, pp. 1040–1070, 2006.

[3] T. Ushijima, K. Takano, H. Kojima, “High performance hydraulic mounts for improving vehicle noise and vibration”, SAE Paper # 880073, 1988.

[4] M. Nguyen, “A Novel Semi-Active Magnetorheological Mount for Vibration Isolation”, MSc thesis, Department of Mechanical Engineering, The University of Toledo, August 2009.

[5] N. Vahdati, M. Ahmadian, “Single Pumper Semi-active fluid mount”, Proceedings of IMECE2003, Washington, D.C., November 16-21, 2003.

[6] M. S. Foumani, A. Khajepour, M. Durali, “Application of SMA to a new adaptive hydraulic mount”, Proceedings of the 2002 SAE International Body Engineering Conference and Automotive & Transportation Technology Conference, July 9-11, 2002.

[7] J. D. Carlson, R. M. Jolly, “MR fluid, foam and elastomer devices”, Mechatronics, vol. 10, pp. 555-569, 2000.

[8] S. R. Hong, S. B. Choi, W. J. Jung, I. B. Ham, D. K. Kim, “Vibration control of an ER mount subjected to high static loads”, Journal of Sound and Vibration, vol. 242(2), pp. 740-748, 2001.

[9] T.A. Baudendistel, S.G. Tewani, J.M. Shores, M.W. Long, R.E. Longhouse, C.S. Namuduri, A.A. Alexandridis, “Hydraulic mount with magnetorheological fluid”, US Patent No. 6,622,995 B2, 2003.

[10] G.J. Stelzer, M.J. Schulz, J. Kim, R.J. Allemang, “A magnetorheological semi-active isolator to reduce noise and vibration transmissibility in automobiles”, Journal of Intelligent Material Systems and Structures, vol. 14, pp. 743-765, 2003.

[11] S.B. Choi, H.J. Song, H.H. Lee, S.C. Lim, J.H. Kim, H.J. Choi, “Vibration control of a passenger vehicle featuring magnetorheological engine mounts”, International Journal of Vehicle Design, vol. 33, no. 1-3, pp. 2-16, 2003.

[12] D. E. Barber, J. D. Carlson, "Performance Characteristics of Prototype MR Engine Mounts Containing LORD Glycol MR Fluids", Journal of Physics: Conference Series 149 (2009) 012035

[13] A.V. Srinivasan, M.D. McFarland, “Smart Structures: Analysis and Design”, Cambridge University Press, New York, 2001.

[14] Lord Corporation, Cary, NC, MRF- 132LD, 1999.

[15] Y.T. Choi, N. M. Wereley, “Comparative Analysis of the Time Response of Electrorheological and Magnetorheological Dampers Using Non-dimensional Parameters”, Journal of Intelligent Material Systems and Structures, vol. 13, pp. 443–451, 2002.

[16] R. Singh, G. Kim, P.V. Ravindra, “Linear analysis of automotive hydraulic-mechanical mounts emphasis on decoupler characteristics”, Journal of Sound and Vibration, vol. 158, pp. 219–243, 1992.

Nomenclature

A_p	effective piston area
B_r	damping property of the rubber spring
C	constant value in the range of 2 to 3
C_1	volumetric compliances of the upper chamber
C_2	volumetric compliances of the lower chamber
h	distance between magnetic poles
H	magnetic field strength
I_1	fluid inertia in the inertia track
I_2	fluid inertia in the MR valve
K_d	dynamic stiffness
K_l	loss stiffness
K_r	dynamic stiffness of the rubber spring
K_s	storage stiffness
L	length inside the MR valve where the magnetic field is effective
P_1	pressures in the upper chamber
P_2	pressures in the lower chamber
Q_1	fluid volume fluxe through the inertia track
Q_2	fluid volume fluxe through the MR valve
R_1	resistance of the inertia track against fluid flow
R_2	resistance of the MR valve against fluid flow
ΔP_{MR}	pressure drop due to the yield stress of the MR fluid
τ_y	MR fluid yield stress
ϕ	loss angle