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Performance Analysis of Different Modified MR Engines Mounts

T. Feyzi, R. Tikani, M. Esfahanian*, S. Ziaei Rad

Department of Mechanical Engineering, Isfahan University of Technology, Isfahan 84156-83111, Iran

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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 and harshness (NVH) performance of the 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 each operational mode. Magnetorheological (MR) engine mount is a semi-active hydraulic engine mount, containing MR fluid, which can alter its dynamic behavior as a result of applying magnetic field. In this paper, design concept of two MR mounts is presented and their dynamic behavior is simulated. It is shown that the simulation methods used in this paper for simulating the dynamic behaviors of the MR mounts are effective with which the dynamic characteristic analysis and design optimization of MR mounts can be performed before its prototype development. Because of increasing demands for semi-active MR mounts in automotive industries, this can ensure their low cost and high quality for development.

Keywords: Hydraulic engine; 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]:

* Corresponding author. Tel.: +98 311 391 3110; Fax: +98 311 391 2718.
E-mail address: mesf1964@cc.iut.ac.ir (M. Esfahanian).

- 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 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.

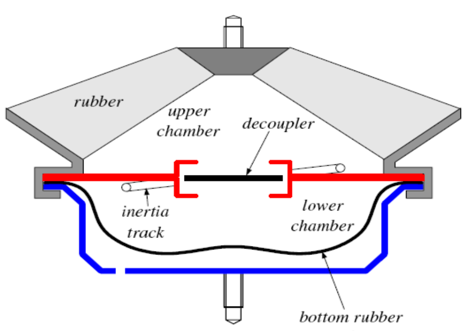


Fig. 1
Schematic of a passive hydraulic engine mount [2].

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 (second design) is illustrated in Fig. 2. In the first design, inertia track is not included and the mount has just an MR valve. 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.

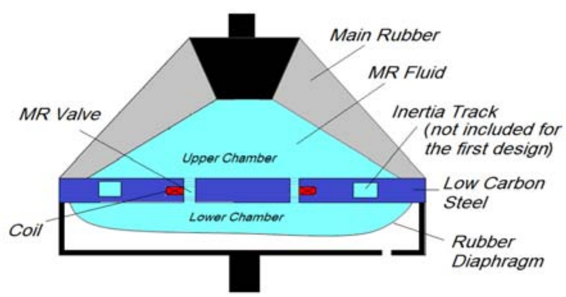


Fig. 2
Schematic of the MR engine mount.

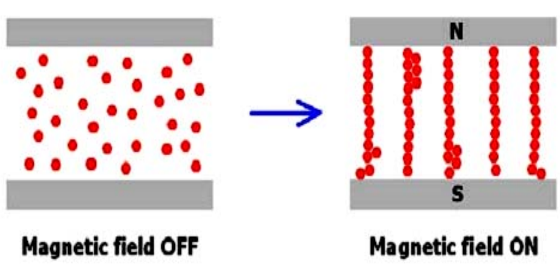


Fig. 3
MR fluid with and without magnetic field [4].

The first design of MR mount is very similar to a passive hydraulic engine mount with an inertia track. However, the inertia track is omitted and an MR valve is replaced. The inertia track has a fixed resistance against the fluid flow, but the resistance of a MR valve can be changed through applying a magnetic field as a result of the change in the viscosity of the MR fluid. When there is no magnetic field applied, the mount acts like a passive hydraulic engine mount which has an inertia track having the same inertia and resistance properties as the MR valve. When the magnetic field is applied on the MR fluid contained in the MR valve, the resistance of the MR valve against fluid flow is increased resulting in a higher damping for the mount. As a result, dynamic stiffness of the mount increases at lower frequencies. With further increase in magnetic field, the MR valve is essentially closed and there will be no more flow between the top and bottom chambers of the MR mount. At this point, the dynamic characteristic of the mount resembles that of a conventional rubber mount.

The second design of 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. It can also 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 viscosity of the fluid in the MR valve. So, 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 the entire part of the fluid flowing between chambers passes through the inertia track. At this point, the MR mount shows the behavior 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. 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

$$C_1\dot{P}_1 = A_p\dot{x} - Q_1 - Q_2 \tag{1}$$

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

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:

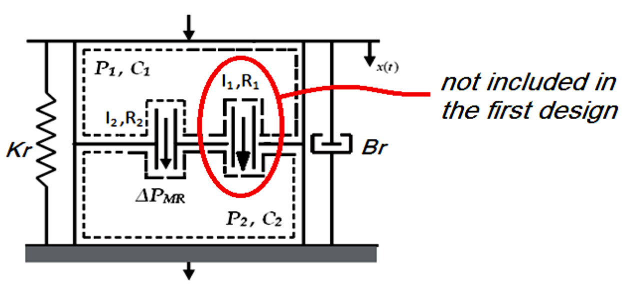


Fig. 4
Mechanical model of the MR mount.

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

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

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. [13], 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]

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

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

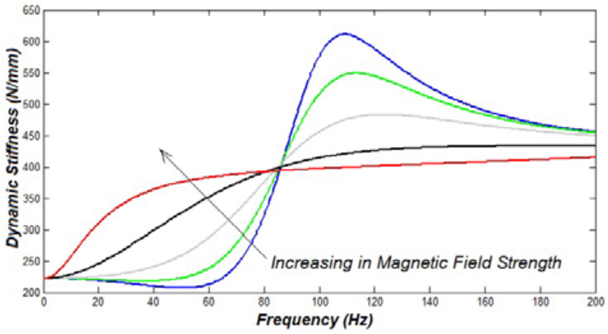
$$K_d = \sqrt{K^2{}_s + K_l^2}, \qquad \varphi = \arctan\left(\frac{K_l}{K_s}\right) \tag{9}$$

4 SIMULATION RESULTS

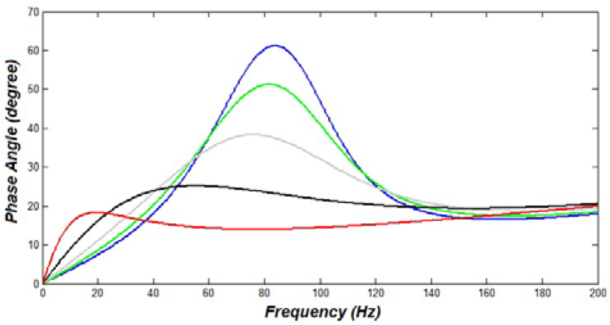
A MATLAB program, with the parameters listed in Table 1 for both the MR mounts, was used to simulate dynamic properties of the mount by using Eqs. (1-9). The effect of the magnetic field strength on the dynamic properties (dynamic stiffness and phase angle) of the first design of the MR mount is illustrated in Fig. 5.

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}		



(a)



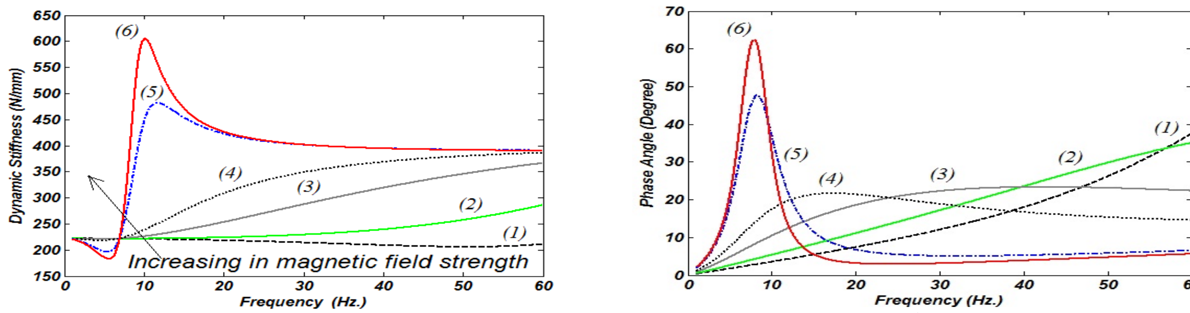
(b)

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

When the magnetic field increases it affects the MR mount dynamics greatly, as a result of an increase in the flow resistance. The reduced flow rate results in a higher damping for the mount. Therefore, the notch depth rises while the peak drops. When the magnetic field increases more, the dynamic stiffness of the mount becomes very similar to that of a conventional rubber mount, i.e. an almost constant dynamic stiffness. This is because of the high value of the applied magnetic field which limits the fluid flow through the MR valve. Consequently, the fluid flow between the top and bottom chambers of the MR mount is stopped. At this point, the stiffness of the MR mount is approximately equal to the sum of the elastomeric and the volumetric stiffness of the rubber spring. This behavior represents an opportunity for continuously tuning the low-frequency stiffness of such mounts.

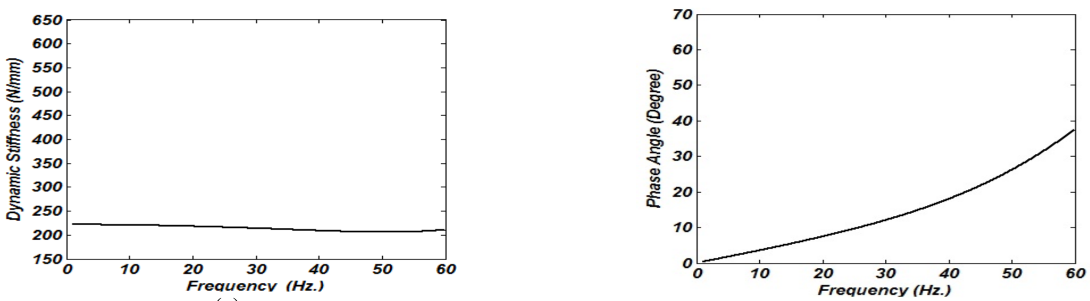
The variation of dynamic properties (dynamic stiffness and phase angle) of the MR mount with the magnetic field strength is illustrated in Fig. 6. When no magnetic field is applied (off-state), MR fluid can easily pass through the MR valve and inertia track without any significant resistance against 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 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. 7 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 (graph 1in Fig. 6) are unique. When the magnetic field is applied the resistance of the MR valve against the fluid flow 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.



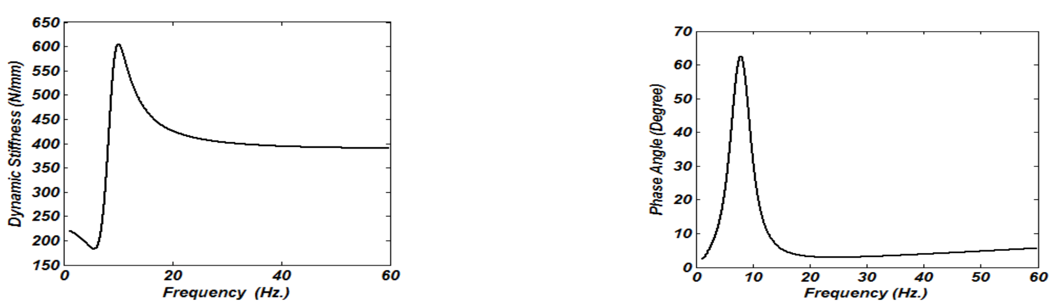
(a) (b)

Fig. 6
Variation of the dynamic characteristics of the second design of MR mount with applied magnetic field strength; (a) dynamic stiffness and (b) phase angle.



(a) (b)

Fig. 7
Dynamic behaviour of a passive hydraulic engine mount with an orifice the same as the MR valve; (a): dynamic stiffness and (b): phase angle.



(a) (b)

Fig. 8
Dynamic behaviour of a passive hydraulic engine mount with an orifice the same as the inertia track of the MR mount; (a): dynamic stiffness and (b): Phase angle.

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. 8.

5 CONCLUSIONS

In this study, the mathematical models of two designs of semi-active MR mount (first design with only an MR valve and the second one with both an MR valve and an inertia track) were derived. The resulting system of equations of motion was constructed in a MATLAB program to simulate the behavior of the mount. From the simulation results, it was proven that both the mounts could alter their dynamic stiffness and damping via applying a magnetic field and show a highly tunable response. It was shown that the first design (MR mount with only an MR valve) can change its dynamic properties from a standard (non MR) hydraulic mount with a simple orifice, which has a lower stiffness and damping at low frequencies and higher stiffness at high frequencies (off-state), to a simple elastomeric mount (on-state) with a frequency independent stiffness (higher at low frequencies and lower at high frequencies compared with off-state) and low damping. So, the low frequency dynamic stiffness of such MR mounts can be continuously tuned. Also, it was shown that the second design (MR mount with both an MR valve and inertia track) 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.

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