SOME DISCUSSIONS OF MR ENGINE MOUNT ON VIBRATION ATTENUATION AND FORCE TRANSMISSION

Shyh-Chin Huang

Mechanical Engineering, Ming Chi University of Technology, No.84, Gongzhuan Rd., Taishan Dist., New Taipei City 24301, Taiwan

e-mail: schuang@mail.mcut.edu.tw

In this paper, vibration attenuation and force transmission of an MR engine mount were discussed. Two different models, the conventional 1-DOF and the 2-DOF containing vehicle chassis dynamics, were compared in theoretical studies. The engine’s displacement response and transmitted force to the vehicle body were derived and used as performance indicators. Theoretical results showed that the displacement response would be overestimated and the transmitted force underestimated in the simplest 1-DOF model. The differences could be up to 40% (displacement) and 20% (force) depending on engine mount’s damping ratio. An MR damper operated in flow mode was then designed and manufactured. Experiments were arranged to verify if MR performed as expected. Data for transmissibility were measured in the experiments and the results showed satisfactory agreement with theoretical calculations. This investigation enhanced the capability of MR damper implemented into engine mount for vibration and force reduction at low speeds. It also showed to the audience that in vehicle dynamics that the chassis dynamics might need to be included in designing the engine mount.

1. Introduction

Engine mounts are implemented to reduce engine’s shaking motions due to imbalances. For effective isolation it demands high damping in low frequency, and low damping in high frequency excitation [1]. Conventional passive mount of fixed damping constant could not meet the requirement therefore there have been developed semi-active/active engine mounts to meet the goal. Active mounts play the best but require the largest energy inputs. Semi-active hence provides the most cost-efficient, versatile selections to the auto engineers. To maximize the variability of mount damping and stiffness, hydraulic types of engine mount with adjustable parameters are usually adopted as semi-active components [2]. Among the techniques of altering engine mount’s parameters, electro-rheological fluid (ERF) and magneto-rheological fluid (MRF) incorporated into hydraulic engine mount are two of the very promising applications [3-5]. ER and MR fluids possess the property of adjustable fluid viscosity such that the damping constant can be altered to meet the engine operation. The feasibility of the application is further enhanced by the short response time, wide control bandwidth and compact size.

ERF and MRF technologies were discovered and developed about at the same years of 1940s. Rabinov discovered the MRF effect at the US National Bureau of Standards [6,7]. Wislow worked on the ERF technology. Though both technologies were discovered in the 1940s, more research
works have been carried out on ERF than on MRF. ERF effect depends on an electrostatic field and MRF effect depends on a magnetic field. These two different technologies require about the same power level. ERF yet requires for much larger voltage, up to thousand volts and mini-amperes, MRF normally needs about 2 to 24 V and some amperes. Lately, researchers [8] concluded that MRF products have between 20 and 50 times higher control effect than the equivalent ERF products and MRF technology today appeared better stability with regard to contaminants. Based on all these MRF technology advantages, a very high level of interest to introduce MRF products has been created during the recent years. In addition to the applications to engine mounts, there were those in suspension systems [9-11]. In MRF/ERF applications, there are three different operational modes: shear mode (used in brakes and clutches), valve/flow mode (used in dampers) and squeeze mode [8]. Hong et al [12] experimentally compared a squeeze mode and a flow mode ERF mount and concluded that the squeeze mode could stand for larger static load and performed better at low levels of excitation. Ciocanel et al [13] further designed an MRF mount that operated at a mixed mode, where the squeeze mode operated and the flow mode took hand once a maximum allowable displacement was reached. Their simulations showed that increased magnetic strength reduced transmissibility and increased system’s resonance frequency. Their results also yielded the conclusion that combining both modes in operation resulted in an increased effectiveness of the mount.

From reviews of the existing engine mount models, the author found most of them used as simple as the 1-DOF model, which included just engine mass and mount’s stiffness and damping. The suspension combined with the chassis usually obtained the natural frequency as low as a few Hertz and for such a case the coupling effect between chassis-suspension and engine-mount should be of some significance. This effect may need to be considered in engine mount design. This paper hence intends to derive how an MR damper reduces the vibration response and force transmission in 1-DOF and 2-DOF models and further looks into the differences between models. To verify the reduction effect, a flow mode MR is designed and manufactured. The MR damper’s performance is then calculated and compared with the experimental measurement.

2. Dynamic Model

Figure 1 shows two different models for an engine-mount system, where (a) is the widely used 1-DOF model and (b) is the more realistic 2-DOF model. The difference between them is the chassis dynamics being included in 2-DOF model. The MR damper plays as a semi-active component by which the damping force is adjusted via changing the applied magnetic field. MR resistive force, denoted $f_{MR}$, can be derived from several different approaches. The equation of motion (EOM) for 1-DOF model can be written as:

$$m_1\ddot{x} + k_1x = f_e - f_{MR},$$

where $m_1$ and $k_1$ represent the engine mass and the mount stiffness. $f_e$ denotes the engine unbalanced excitation and, without loss of generality, can be assumed to be

$$f_e = me0^2e^{i\omega t} = F_ee^{i\omega t}$$

The MR damping force can be further expressed as

$$f_{MR} = c_p\dot{x} + f_{yield}\frac{\dot{x}}{|\dot{x}|},$$

in which there are two parts: the passive damping force $c_p\dot{x}$ and the magnetic field dependent force $f_{yield}$. Details of these two terms depending on the fluid properties and MR damper geometry
will be illustrated in section 3. Equation (1) is hence rewritten as

\[ m_1 \ddot{x} + c_1 \dot{x} + k_1 x = f_e \]  

(4)

where

\[ c_1 = c_p + \frac{f_{\text{field}}}{|\dot{x}|} \]  

(5)

Note that \( c_1 \) is varying with the magnetic field in general and equals the passive constant if the coil current is shut off, i.e., \( c_1 = c_p \). The engine displacement response and force transmissibility with regard to Eq. (4) can be easily derived from any vibration text book as

\[
X^* = \begin{bmatrix}
X \\
X_{st}
\end{bmatrix} = \frac{1}{\sqrt{(1-\gamma^2)^2 + (2\zeta_1 \gamma)^2}}
\]

(6)

\[
TR_1 = \frac{F_T}{F_e} = \frac{\sqrt{1+(2\zeta_1 \gamma)^2}}{\sqrt{(1-\gamma^2)^2 + (2\zeta_1 \gamma)^2}}
\]

(7)

where \( X_{st} = F_e / k_1, \omega_1 = \sqrt{k_1 / m_1}, \gamma = \omega_1 / \omega_1, \zeta_1 = c_1 / 2 \sqrt{m_1 k_1} \) stand for static displacement, engine-mount natural frequency, excitation to mount frequency ratio and MR damping ratio (varying).

Since engine is of variable speed the transmitted force \( F_T \) through engine mount to the chassis is more suitably expressed as

\[
F_T = F_e \cdot TR_1 = m \omega_1^2 \cdot \frac{\omega_1^2}{\omega_1^2} \cdot TR_1 = m \omega_1^2 \cdot \gamma^2 \cdot TR_1 = m \omega_1^2 \cdot \gamma^2 \cdot \frac{1+(2\zeta_1 \gamma)^2}{\sqrt{(1-\gamma^2)^2 + (2\zeta_1 \gamma)^2}}
\]

(8)

It is seen the transmitted force is proportional to \( \gamma^2 \cdot TR_1 \), called the force transmissibility. Similarly, the EOM’s for 2-DOF model, Figure 1(b), are derived as well

\[
\begin{align*}
    m_1 \ddot{x}_1 + c_1 (x_1 - x_2) + k_1 (x_1 - x_2) &= f_e \\
    m_2 \ddot{x}_2 + c_2 (x_2 - x_1) + k_2 (x_2 - x_1) + k_1 (x_1 - x_2) &= 0
\end{align*}
\]

(9)

\( m_2, k_2 \) and \( c_2 \), respectively, denote the chassis mass, suspension stiffness and damping constant. By a similar approach but more complicated calculations, the engine’s response \( (X_1) \), chassis’ response \( (X_2) \) and force transmissibility to the chassis are solved to be

\[
X_1^* = \frac{X_1}{X_{st}} = \frac{(1+\mu \lambda^2 - \mu \gamma^2)^2 + (2\zeta_1 \gamma + 2\mu \zeta_2 \lambda \gamma)^2}{(\mu \gamma^4 - \mu \gamma^2 \lambda^2 - \mu \gamma^4 - 4\mu \gamma^2 \zeta_2 \lambda \gamma + \mu \lambda^2)^2 + (2\mu \zeta_2 \lambda \gamma + 2\mu \zeta_2 \lambda^2 \gamma - 2\zeta_1 \gamma^3 - 2\mu \zeta_2 \lambda \gamma^3 - 2\mu \zeta_1 \gamma^3)^2}
\]

(10)

\[
X_2^* = \frac{X_2}{X_{st}} = \frac{(1+\mu \lambda^2 - \mu \gamma^2)^2}{(\mu \gamma^4 - \mu \gamma^2 \lambda^2 - \mu \gamma^4 - 4\mu \gamma^2 \zeta_2 \lambda \gamma + \mu \lambda^2)^2 + (2\mu \zeta_2 \lambda \gamma + 2\mu \zeta_2 \lambda^2 \gamma - 2\zeta_1 \gamma^3 - 2\mu \zeta_2 \lambda \gamma^3 - 2\mu \zeta_1 \gamma^3)^2}
\]

(11)

\[
\gamma^2 \cdot TR_2 = \gamma^2 \frac{-(-\gamma^2 \mu^2 + (-2\mu \zeta_1 \gamma)^2)^2}{(\mu \gamma^4 - \mu \gamma^2 \lambda^2 - \mu \gamma^4 + 4\mu \gamma^2 \zeta_2 \lambda \gamma + \mu \lambda^2)^2 + (2\mu \zeta_2 \lambda \gamma + 2\mu \zeta_2 \lambda^2 \gamma - 2\zeta_1 \gamma^3 + 2\mu \zeta_2 \lambda \gamma^3 - 2\mu \zeta_1 \gamma^3)^2}
\]

(12)

Note that the displacement responses are normalized with respect to the engine’s static displacement \( X_{st} \) of 1-DOF model so that the following comparisons are on the same base. Additional dimensionless parameters are defined as follows: \( \omega_s = \sqrt{k_1 / m_1}, \lambda = \omega_1 / \omega_s, \mu = m_1 / m_2, \zeta_2 = c_2 / 2 \sqrt{m_2 k_2} \), respectively denote the suspension natural frequency, suspension to mount frequency ratio, engine
to chassis mass ratio, and the suspension damping ratio. Compare Eq. (14) to (8), it is obvious that the transmitted force depends on not only engine mount property but also the chassis dynamics.

3. Engine MR damper

Among the available types of MR operations, we select a flow (valve) mode type and design an MR damper as shown the schematic diagram in Figure 2. The pressure drop created in this mode is the sum of the viscous (pure rheological) component $\Delta P_\eta$ and the magnetic field dependent component $\Delta P_f(H)$. The value of this pressure drop is defined using the following approximation [8]:

$$\Delta P = \Delta P_\eta + \Delta P_f(H) = \frac{12\eta QL}{bh^3} + \frac{3\tau_y(H)L}{h}$$

(13)

$$Q = A_p \times v = \frac{\pi}{4}(D^2-d^2)\times v$$

(14)

Use the corresponding geometrical parameters and plug eq. (15) into eq. (3) the MR damping constant, expressed in terms of passive and active portion, becomes

$$f_{yield} = \frac{3\tau_y(H)L\pi}{4h}(D^2-d^2)$$

(15)

$$c_p = \frac{12\eta L}{bh^3}\left(\frac{\pi}{4}(D^2-d^2)^2\right)$$

(16)

Note that the above parameters are defined as: $D$ and $d$: piston’s outer and inner diameter; $\eta$: dynamic fluid viscosity with no magnetic field; $\tau_y$: field-dependent yield stress; $H$: magnetic field intensity; $L$, $b$ and $h$ are flow channel length, width and gap. The field-dependent yield stress depends on what type of MR fluid chosen. We use MRF-122EG made by LOAD Co. and the fluid’s yield stress versus magnetic field strength curve provided by LOAD is shown in Figure 3. The magnetic field intensity is of the expression

$$H = \frac{NI}{2\pi R}$$

(17)

where $N$: number of coils, $I$: current; $R$: toroid mean radius. Note that the values of MR’s parameters described above are summarized in Table 1, where all the geometric parameters are in mm unit.

4. Results and discussions

Numerical and experimental results are now illustrated. First, the differences between 1-DOF and 2-DOF model are compared. Table 2 lists all the parameters, presumably representing an ordinary vehicle, used for numerical calculations. The damping ratio $\zeta_1$ depending on the passive fluid and input current to the coil and is varying in the simulations. From Table 2, the values are calculated, $\omega_0=20$ rad/s, $\omega_2=9.6$ rad/s $\lambda=\omega_f/\omega_0=0.48$, and $\zeta_2=0.24$. Figure 4 compares the difference in (a) engine displacement response and (b) the force transmission for 1DOF and 2DOF models. Figure 4(a) shows that 1-DOF’s engine response is larger than that of 2-DOF in the vicinity of $\gamma=1$ and Figure 4(b) shows that the transmitted force is yet to the opposite. These differences arise from the entrance of chassis dynamics. Provided that 2-DOF better represents a real vehicle, then 1-DOF model overestimates the engine’s displacement response but underestimates the engine force transmitted to the chassis. The “errors” vary with the mount’s damping ratio. Table 3 illustrates the error percentages at the peaks for various mount damping ratios $\zeta_i$. The comparisons show that displacement response errors increase with decreasing damping ratio up to 40% at as low damping ratio of 0.1. The errors in force transmission yet increase with increasing damping ratio up to 20%
at a higher damping ratio of 0.75.

As the engine starts, it always runs through $\gamma = 1$ therefore there exists time duration of large response amplitude and transmitted force. The significance of MR damper is it can activate in just mini-second such that the damping constant increases suddenly and the response being suppressed efficiently. If the MR is turned on before $\gamma = \sqrt{2}$ and returned to its passive state afterwards, the engine vibration will be significantly improved. As described in Eq. (5) the MR damper contains the passive portion, which is basically fixed as the geometry and fluid type are fixed. The damping constant $c_p$ of Eq. (16) and corresponding passive damping ratio $\zeta_1$ depending on fluid type and passage design in the demonstrated case is given in Table 1. The magnetic field-dependent damping force, Eqs. (15,17), are calculated to be $f_{yield} = 1.99N$ and $f_{yield} = 3.99N$ respectively for $I = 0.5A$ and $1.0A$. From Fig. 3 it is noticed the yield strength will eventually reach saturation and our tests show that it happens as the current exceeds 1A.

The next goal of the research is to see if the MR’s performance works as we expected. To verify this, a simple experiment is set up. Since all the derived equations are in dimensionless, it is unnecessary to build a scale as a real engine mount. A 1-DOF system is scaled down as shown in Table 1. Note that the purpose of this test is to verify if MR behaves as the theoretical calculations so that we rather not setup a 2-DOF model since that is expected to generate more measurement errors due to line-up precision. Figure 5 illustrates the experimental set-up and the photo of MR resting on a shaker. The function generator outputs a sine wave of various frequencies to the power amplifier then the shaker. A Load current controller controls the input current to MR damper. The displacement output of the top mass is picked up by an Eddy current sensor. The ratios of output to input are then calculated for transmissibility. Analytical calculations of force transmission is evaluated and shown in Figure 6(a), in which three curves for passive, 0.5A and 1.0A. Experimental results are shown in 6(b). Comparisons of these two figures show surprisingly good agreement. The improvement of force transmission in dB for theory and experiment at the peaks are digested in Table 4. Experimental results are slightly less than the theoretical calculations but the improvement is still of significance.

5. Conclusions

This paper illustrated the effects of chassis dynamics to the engine mount response and transmissibility by comparing 1-DOF and 2-DOF models. The results indicate that without chassis dynamics considered the engine response is overestimated and the transmitted force is underestimated. The differences can be up to 40% and 20% depending on engine mount damping ratio. An MR damper of flow mode operation was designed and its performance was verified by vibration tests. The results showed that experimental measurements agreed very well with the theoretical calculations. This study enhanced the applicability of MR damper on engine mount and the improvement could be up to 3dB near the resonance frequency.

<table>
<thead>
<tr>
<th>$m$(kg)</th>
<th>$k$ (N/m)</th>
<th>$D$</th>
<th>$d$</th>
<th>$N$</th>
<th>$L$</th>
<th>$h$</th>
<th>$b$</th>
<th>$R$</th>
<th>$\eta$(Pascal·s)</th>
<th>$c_p$(Ns/m)</th>
<th>$\zeta_1$</th>
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<tr>
<td>3.75</td>
<td>1.90(10^4)</td>
<td>36</td>
<td>9</td>
<td>1100</td>
<td>9</td>
<td>2</td>
<td>6</td>
<td>27.5</td>
<td>0.042</td>
<td>86.05</td>
<td>0.16</td>
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<table>
<thead>
<tr>
<th>$m_1$(kg)</th>
<th>$k_1$(N/m)</th>
<th>$m_2$</th>
<th>$k_2$</th>
<th>$c_f$(Ns/m)</th>
<th>$c_2$</th>
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<tr>
<td>100</td>
<td>4.0(10^3)</td>
<td>1330</td>
<td>1.23(10^3)</td>
<td>varying</td>
<td>6780</td>
<td>13.3</td>
<td>0.48</td>
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Table 3: Comparisons of 1-DOF and 2-DOF results at the peaks

<table>
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<tr>
<th>$\zeta$</th>
<th>$X^*$</th>
<th>$X^*_1$</th>
<th>Error (%)</th>
<th>$r^2 TR_1$</th>
<th>$r^2 TR_2$</th>
<th>Error (%)</th>
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<tr>
<td>0.1</td>
<td>4.97</td>
<td>3.55</td>
<td>40</td>
<td>5.08</td>
<td>5.52</td>
<td>-8.0</td>
</tr>
<tr>
<td>0.25</td>
<td>1.99</td>
<td>1.55</td>
<td>28.4</td>
<td>2.24</td>
<td>2.63</td>
<td>-14.8</td>
</tr>
<tr>
<td>0.5</td>
<td>0.99</td>
<td>0.81</td>
<td>22.2</td>
<td>1.41</td>
<td>1.73</td>
<td>-18.5</td>
</tr>
<tr>
<td>0.75</td>
<td>0.67</td>
<td>0.56</td>
<td>19.6</td>
<td>1.2</td>
<td>1.5</td>
<td>-20.0</td>
</tr>
<tr>
<td>1</td>
<td>0.5</td>
<td>0.43</td>
<td>16.3</td>
<td>1.12</td>
<td>1.41</td>
<td>-20.6</td>
</tr>
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Table 4: Force transmission improvement by MR

<table>
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<tr>
<td>Theoretical</td>
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<td>-1.6dB</td>
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<td>Experimental</td>
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Fig. 1: MR Engine mount model: (a) 1-ODF, (b) 2-DOF

Fig. 2: Designed flow mode MR damper

Fig. 3: Characteristic curve of MRF-122EG fluid
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Figure 4: Comparisons of 1-DOF and 2-DOF model for (a) engine displacement response and (b) force transmission.

Fig. 5: (a) Experimental set up and (b) photo of experiment

Figure 6: (a) Theoretical calculations and (b) experimental measurements of force transmission for two current values.
REFERENCES

4 Matsuoka H, Mikasa T and Nemoto H 2004 NV countermeasure technology for a cylinder-on-demand engine development of active control engine mount Proceedings of the SAE World Congress.