DESIGN AND DEVELOPMENT OF ACTIVE AND SEMI-ACTIVE ENGINE MOUNTS

By

Hossein Mansour
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© Hossein Mansour
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APPROVAL

Name: Hossein Mansour
Degree: Master of Applied Science
Title of Thesis: Design and Development of Active and Semi-Active Engine Mounts

Examinining Committee:
Chair:

Dr. Farid Golnaraghi
Senior Supervisor
Professor and Associate Director Engineering Science (Surrey); Burnaby Mountain Chair

Dr. Siamak Arzanpour
Supervisor
Assistant Professor of Engineering Science

Dr. Ash Parameswaran
Supervisor
Professor and Director of the Institute of Micromachine and Microfabrication Research

Dr. Mehrdad Moallem
Internal Examiner
Associate Professor of Engineering Science

Date Defended/Approved: August 20, 2010
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ABSTRACT

Vibration isolation in the engine compartment is a challenging design problem for all transportation means particularly in the automotive industry to attain better ride quality, improved road handling, and longer engine/parts life. Given the emergence of new vehicles with more stringent performance characteristics, engine vibration isolation has become a more demanding issue.

This thesis focuses on the modelling, development, and experimental analysis of two active and semi-active engine mounts designed specifically to address the isolation problem of Variable Displacement Engines (VDE). It has been shown, however, that the designed mounts are flexible enough to fulfil the isolation requirements of other engine types as well. Both proposed mounts are made by adding retrofittable parts to the conventionally available hydraulic engine mounts. The promising performance of the fabricated mounts, in addition to their minimal cost, fail safety, and low energy consumption, makes them appealing solutions for the auto industry.

Keywords: Variable displacement engine; Vibration isolation; Hydraulic engine mounts; Active; Semi-active; Driving condition
To my father, mother

and beloved wife
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NOMENCLATURE

\( A_p = \) equivalent piston area of the rubber \([\text{mm}^2]\)
\( \omega_{dr} = \) forcing input frequency of oscillation \([\text{rad/s}]\)
\( \omega_n = \sqrt{\frac{K}{m}} \) Natural frequency \([\text{rad/s}]\)
\( A_a = \) effective area of the actuator \([\text{mm}^2]\)
\( B_r = \) rubber linear damping \([\text{N-s/mm}]\)
\( C_{1,2,m} = \) compliance of the upper chamber, lower chamber, and MR chamber \([\text{mm}^5/N]\)
\( F_T = \) transmitted force \([\text{N}]\)
\( F_a = \) Force of the Engine mount generated by the active part \([\text{N}]\)
\( F_{mag} = \) the magnetic force of the coil imposed to the plunger \([\text{N}]\)
\( F_p = \) Force of the Engine mount generated by the passive part \([\text{N}]\)
\( F_{react} = \) The reaction force of the plunger to the chassis \([\text{N}]\)
\( I_{i,d,m} = \) fluid inertia of the inertia track, decoupler, and MR chamber \([\text{N-mm}^5]\)
\( K_{dyn} = \) Dynamic stiffness \([\text{N/mm}]\)
\( K_r = \) rubber linear stiffness \([\text{N/mm}]\)
\( P_{1,2} = \) pressure of the upper and lower chambers respectively
\( Q_{i,d,m} = \) flow from the inertia track, decoupler, and MR chamber \([\text{mm}^3/s]\)
\( R_{i,d,m} = \) fluid resistance of the inertia track, decoupler and MR chamber \([\text{N-s/mm}^5]\)
\( Y_a = \) actuator displacement
\( f_e = \) harmonic force excitation amplitude per unit mass \([\text{N/kg}]\)
\( x_r = \) relative Engine and chassis displacement \([\text{mm}]\)
\( a = \frac{\ddot{X}}{Y} \) Acceleration with respect to base excitation amplitude \([1/s]\)
\( r = \frac{\omega}{\omega_n} \) excitation frequency ratio
\( \zeta = \frac{c}{2\sqrt{km}} \) Linear damping ratio
\( \Lambda = \frac{\left| Y/f_e \right|}{f_e} \) Induced displacement with respect to a harmonic force excitation \([\text{mm.kg/N}]\)
\( \Phi = \) harmonic force excitation force transmissibility
\( \omega = \) frequency of oscillation \([\text{rad/s}]\)
\( F(t) = \) engine dynamic loading \([\text{N}]\)
\( FT = \) transmitted force per unit mass \([\text{N/kg}]\)
\( G = \) transfer function between mount displacement and passive force
\( H = \) transfer function between actuator displacement and active force
\( I = \) current to the coil \([\text{A}]\)
\( MR = \) magnetorheological
\( U = \) transfer function between overall force and mount displacement
\( V = \) Transfer function between the imposed current and mount displacement
\( X = \) Engine harmonic displacement amplitude \([\text{mm}]\)
\( Y = \) base harmonic displacement amplitude \([\text{mm}]\)
\( Z = \) control function of the actuator
\( c = \) linear damping coefficient [N-s/mm]
\( e = \) eccentricity of unbalance mass [mm]
\( k = \) linear stiffness coefficient [N/mm]
\( s = \) Laplace transformation variable
\( t = \) time [s]
\( x = \) Engine displacement [mm]
\( y = \) chassis or base displacement [mm]
\( \alpha = \) hydraulic dynamic term
\( \beta = \) dynamic flexibility of the plunger [mm/N]
\( \gamma = \) proportional coefficient of the solenoid actuator [N/A]
Vibration isolation in the engine compartment is a challenging design problem for all transportation systems, particularly in the automotive industry to attain better ride quality, improved road handling, and longer engine and parts life. The sources of excitation on the engine can be divided into two categories: 1) base excitation imposed by road pattern and acceleration of the vehicle; and 2) force excitation on the engine. The excitation force of the engine is generated from the unbalance force of rotary parts, such as crankshaft; the inertia of reciprocating parts, such as pistons; and the firing impulses inside the cylinders. A proper and effective isolation system is one that can control 1) the force/acceleration transmission to the frame; 2) induced acceleration on the engine; and 3) the relative engine-body displacement.

The handling of different types of input to fulfil these requirements is a conflicting task, mainly because of the difference in the frequency/amplitude pattern of engine vibration in its course of action. The amplitude of engine vibration is usually more than 0.3 mm at low frequencies (1-50 Hz) and less than 0.3 mm at high frequencies (50-200 Hz) (Golnaraghi and Jazar, 2001). An engine isolator is generally intended to have high damping and high stiffness at low frequency excitation, and low damping and low stiffness at high frequency excitation. Satisfying these requirements requires that the characteristics of the engine mount be a function of excitation frequency as well as amplitude (Brach and Haddow, 1993).
Passive elastomeric mounts (Lord, 1930) have been traditionally used as engine isolators; they are simple in design, compact, cost effective, and durable. The major drawback of these mounts is their low flexibility in adapting isolation performance in the frequency domain. Figure 1-1 presents a physical representation of elastomeric mounts, where the stiffness and damping are modelled by a spring and damper (Swanson, 1993). For such a system, the dynamic stiffness criteria – the ratio of the transmitted force to the relative displacement – increases linearly with the excitation frequency, which means there are only two degrees of freedom to set the dynamic response over frequency.

![Figure 1-1: The lumped model of elastomeric engine mount (a) and its dynamic stiffness (b) (Swanson, 1993)](image)

Hydraulic engine mounts are more advanced versions of passive elastomeric engine mounts (Brach and Haddow, 1993). The hydraulic type of mount uses fluid resistance as an additional damping element to provide more design parameters without compromising durability. It also benefits from a dual dynamic performance mechanism to make its isolation characteristics adaptable with the amplitude and frequency of the excitation. Hydraulic engine mounts are widely used in the automotive industry as an effective and low-cost solution. Their performance improves the noise level by 5 dB (Bemuchon, 1984; Corcoran and Ticks, 1984), and reduces the shock level by one-third.
while maintaining the idle shake constant (Kadomatsu, 1989). However, this type of mount is still passive, and is not truly flexible for adaption to different loading and road conditions.

In contrast to passive mounts, active and semi-active mounts are proposed that are adaptable to various loading conditions. Semi-active mounts change one or more dynamic properties of the system, mainly damping, in order to dissipate vibration energy efficiently. In this sense, semi-active isolators add controllability while maintaining the simplicity and cost efficiency of passive mounts. Semi-active solutions can generate only resistive force, which is considered their limitation in dealing with complex excitations. The other drawback of semi-active systems is their low flexibility toward providing the desired isolation properties over a wide range of frequencies.

Active mounts on the other hand can generate large actuating forces to counter the vibration disturbances. A typical active mount consists of actuators, vibration sensors, and controlling unit (Yu et al., 2001). Different types of active mounts are reported in the literature, including electromagnetic (Muller et al., 1994), hydraulic/pneumatic (Hodgson 1991), and piezoelectric (Shibayama et al., 1995; Ushijima 1993). All these systems have demonstrated promising results in improving the sophisticated vibration characteristics of an engine. However, they are generally hard to implement in real applications, and, in some cases, the reliability of the actuator remains a concern.

1.1 Dynamics of Engine Isolation

To gain an understanding of the dynamics of the vibrating engine on the chassis, consider a 1DoF model as shown in Figure 1-2. This model demonstrates the bounce (up-
down) movement of the engine. There are two types of excitation on the engine: unbalance force of the rotary and reciprocating parts, and base excitation imposed on the chassis.

![Diagram of engine vibration on the chassis](image)

**Figure 1-2: 1DoF representation of engine vibration on the chassis**

The Equation of motion for such a system is

$$ m \ddot{x} + c(\dot{x} - \dot{y}) + k(x - y) = m_e \omega_{dr}^2 \cos(\omega_{dr} t) $$

which can be rearranged as

$$ m \ddot{x} + cx + kx = m_e \omega_{dr}^2 \cos(\omega_{dr} t) + cy + ky $$

In equation (1-2), $m$ is the effective mass of the engine, $c$ is the damping coefficient, $k$ is the stiffness, $m_e$ and $e$ are the mass and eccentricity of the unbalance mass respectively, and $\omega_{dr}$ is the rotational frequency. The absolute displacement of the engine and base are represented by $x$ and $y$, and the relative displacement is denoted $x_r$. It should be noted that the effective stiffness and damping are more than the stiffness and
damping of one engine mount because it represents the effect of several isolators – regularly three or four installed on the engine. The force transferred through the mount can be expressed in two ways. The first is the unbalance force, which has been partially consumed by the accelerating mass of the engine, and the second is the force applied from the chassis by its relative velocity and displacement with respect to the engine. Both of these analogies are shown in equation (1-3).

\[ F_T(t) = m\ddot{x} - m_e\omega_{dr}^2 \cos(\omega_{dr} t) = c(\dot{y} - \dot{x}) + k(y - x) \quad (1-3) \]

The practical case for passive isolators is to find the optimal linear stiffness \( k \) and damping \( c \) that constrain the absolute and relative displacement of the mass \( m \), (i.e., \( x_r \) or \( x \)), while minimizing the transferred force \( F_T(t) \) criteria due to either a base excitation or an input force \( F(t) \). According to the principle of superposition, these two criteria will be studied separately.

### 1.1.1 Base Excitation

Base excitation, which occurs mostly because of road bumps, passes through the chassis and reaches the engine. In the absence of the unbalance excitation, equation (1-1) can be nondimensionalized as

\[ \ddot{x}_r + 2\zeta \omega_n \dot{x}_r + \omega_n^2 x_r = -\ddot{y} \quad (1-4) \]

where the parameters are

\[ \zeta = \frac{c}{2\sqrt{km}} \quad \omega_n = \sqrt{\frac{k}{m}} \quad x_r(t) = x(t) - y(t) \quad (1-5) \]

and the transferred force becomes
\[ F_T(t) = m\ddot{x} = c(\dot{y} - \dot{x}) + k(y - x) \]  \hspace{1cm} (1-6)

As observed, \( F_T(t) \) is directly proportional to the absolute acceleration of the engine mass. From equation (1-6), we see that the transfer function of the system, relating the absolute acceleration of the engine and relative displacement of the engine and chassis to the base excitation \( y \), is

\[ a = \frac{\ddot{x}}{\ddot{y}} = \frac{\omega^2 \sqrt{1+ (2\zeta r)^2}}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}} \]  \hspace{1cm} (1-7)

and

\[ \lambda = \frac{\dot{x} - \dot{y}}{\ddot{y}} = \frac{r^2}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}} \]  \hspace{1cm} (1-8)

where \( r \) is the ratio of the driving frequency \( \omega \) to the natural frequency \( \omega_n \). The absolute acceleration and the relative displacement curves are illustrated in Figure 1-3.

The frequency of base excitations induced by bumps is usually less than the system’s natural frequency. In this high-amplitude excitation, priority goes to the controlling of the relative engine-chassis displacement. According to equation (1-5), increasing the mount stiffness causes higher natural frequencies and reduces the relative displacement. Furthermore, according to Figure 1-3b, increasing the damping produces the same effect.

On the other hand, the frequency content of the road roughness and shocks exceeds the natural frequency of the system – that is, after the crossing point of the graphs in Figure 1-3c on \( \sqrt{2}\omega_n \). In this condition, increasing the damping and stiffness of the isolator produces a negative effect on the force transmissibility and may cause catastrophic accelerations on the engine or fatigue around the mount installation points.
Figure 1-3: The frequency response functions of the base excitation of a 1 DoF system, (a) motion transmissibility, (b) relative motion transmissibility, (c) acceleration transmissibility (Arzanpour, 2006)

1.1.2 Force Excitation

In this case, the base excitation is eliminated, and we look only into the unbalance force on the engine. The non-dimentionalized equation of motion for this condition is

\[ \ddot{x}_r + 2\zeta \omega_n \dot{x}_r + \omega_n^2 x_r = f_e(t) \]  \hspace{1cm} (1-9)

where \( f_e(t) \) is the nondimensionalized unbalance force and can be calculated from
\[ f_e(t) = \frac{F(t)}{m} = \frac{m_e}{m} \omega_{dr}^2 \cos(\omega_{dr} t) \]  

(1-10)

The non-dimentionalized force transmission will be in the form

\[ FT(t) = \frac{F_T(t)}{m} = 2\zeta \omega_n \dot{x} + \omega_n^2 x \]  

(1-11)

Similar to base excitation, the transfer functions can be calculated, relating the transferred force \( F_T \) and the displacement of the mass \( x \) to the input magnitude, respectively, as

\[ \Phi = \left| \frac{FT}{f_e} \right| = \left| \frac{F_T}{F} \right| = \frac{\sqrt{1+(2\zeta r)^2}}{\sqrt{(1-r^2)^2+(2\zeta r)^2}} \]  

(1-12)

and

\[ \Lambda = \left| \frac{x}{f_e} \right| = \frac{1}{\omega_n \sqrt{(1-r^2)^2+(2\zeta r)^2}} \]  

(1-13)

where \( r = \frac{\omega_{dr}}{\omega_n} \) is the frequency ratio. Generally, engine displacement is not of practical interest in higher frequencies; instead, the transmitted unbalance force is the main issue. This force causes NVH problems for the vehicle’s body and the passengers. By comparing equation (1-12) with equation (1-8), we find that the force transmissibility of the force excitation is the scaled version of the motion transmissibility in the base excitation case. As the frequency of the unbalance excitation exceeds the natural frequency of the system, decreasing the stiffness and damping of the isolator reduced the amount of force transmissibility to the chassis and body.

We can conclude from the above analysis that an ideal engine mount should be stiff enough to restrict the relative engine-chassis displacement and maintain alignment between engine and powertrain in lower frequencies, but it has to be soft in high frequencies to prevent the two-way force transmission between engine and body.
Nevertheless, an engine mount should be capable of restricting the motion of the engine against gravity forces and the forces induced by the planar accelerations of the vehicle. Such flexibility is compatible with the nature of hydraulic engine mounts.

1.2 Hydraulic Engine Mounts

A hydraulic engine mount consists of two fluid-filled chambers connected through a decoupler and inertia track, as shown in Figure 1-4. Typically, the fluid within the mount is a mixture of ethylene glycol and water. The upper or high-pressure chamber is bounded on top by a main rubber structure and on the bottom by a steel plate that houses the inertia track and decoupler. The plate is fixed to the base of the mount through a rigid structure that surrounds the lower or compliance chamber. The compliance chamber consists of a compliant rubber bellow that expands and contracts as fluid passes though the inertia track or the decoupler. Most hydraulic mounts have a bell-shaped plate bolted to the top of the pumping chamber, which prevents the decoupler plate from resonating within its cage.

![Figure 1-4: Hydraulic engine mount cross section (a) and its schematic view (b)](image)

The rubber element of the high-pressure chamber has to carry the static load of the engine in addition to providing a portion of the stiffness and damping required to
suppress the engine vibration. The reciprocal movement of the engine forces the liquid inside the pumping chamber through the decoupler and inertia track into the compliance chamber. The ability of the compliance chamber rubber to bulge relieves the pressure in the pumping chamber.

The decoupler plate has a finite travel distance within its cage; therefore, it limits the volume of the fluid that can pass freely between the upper and lower chambers. Once the plate bottoms out on the cage, the fluid stops passing through the decoupler, and it now has to pass through the inertia track. The resistance and mass of fluid within the inertia track increase the overall stiffness and damping of the mount under this condition. On the other hand, during small amplitude excitations, the fluid passes freely through the decoupler, giving the mount low damping and stiffness characteristics.

1.2.1 Mathematical Modelling

To have a visual intuition from the physics of a hydraulic engine mount its lumped model has been shown in Figure 1-5. As shown, the transmitted force through the mount is provided by two parallel elements. The first element is the stiffness and damping of the rubber part and the second is the resistive force of the hydraulic part. This is shown mathematically in equation (1-14)

\[ F_T = K_r X_r + B_r \dot{X}_r + A_p P_1 \]  

where \( F_T \) is the transmitted force, \( X_r \) is the relative engine and chassis displacement, \( K_r \) and \( B_r \) are the stiffness and damping of the rubber, \( A_p \) is the effective area of the pumping chamber, and \( P_1 \) is the pressure inside the pumping chamber. The term \( P_1 \) is
unknown in equation (1-14), but it can be found by considering the momentum and the fluid continuity equations,

\[ C_1 \dot{P}_1 + C_2 \dot{P}_2 = A_p \ddot{X}_r \quad (1-15) \]
\[ C_2 \dot{P}_2 = Q_t + Q_d \quad (1-16) \]
\[ P_1 - P_2 = I_t \dot{Q}_t + R_t Q_t \quad (1-17) \]
\[ P_1 - P_2 = I_d \dot{Q}_d + R_d Q_d \quad (1-18) \]

where \( C_1 \) and \( C_2 \) are the compliances of the pumping and compliance chambers, \( Q \) is the flow rate, \( I \) and \( R \) are the inertia and fluid-channel resistance, \( P_2 \) is the pressure in the compliance chamber, and the subscripts \( i \) and \( d \) stand for the inertia track and decoupler respectively.

![Figure 1-5: Lumped model of a hydraulic engine mount with both inertia track and decoupler](image)

Although equations (1-15) to (1-18) well represent the hydraulic part of the engine mount, an insight into the application range of different components can lead to realistic simplifications. At low frequency excitation, where the amplitude of flow fluctuation is high, decoupler can be assumed ineffective because of its limited travel; at
high frequency excitation, however, since the fluid has not enough time to accelerate and pass through the track, it is realistic to assume the inertia track is inactive. In this sense, either equation (1-17) or equation (1-18) can be eliminated, and the dynamic response can be described by two linear sets of equations depending on the frequency range of excitation. A discussion of the high frequency response of the mount is beyond the scope of this thesis, and only low frequency-high amplitude case will be studied. Therefore, the lumped model will be modified as shown in Figure 1-6, and the governing equations will be modified as in equations (1-19) to (1-21).

![Simplified lumped model of a hydraulic engine mount, without decoupler](image)

Figure 1-6: Simplified lumped model of a hydraulic engine mount, without decoupler

\[
C_1 \ddot{P}_1 + C_2 \dot{P}_2 = A_p \dot{X}_r 
\]

(1-19)

\[
C_2 \dot{P}_2 = Q_i 
\]

(1-20)

\[
P_1 - P_2 = I_i \dot{Q}_i + R_i Q_i 
\]

(1-21)

The pressure inside the pumping chamber can be obtained from equations (1-19) to (1-21) and substituted in equation (1-14) to find the transmitted force,
Finally, the transmitted force can be divided by the imposed displacement to derive the dynamic stiffness equation, as in (1-23). Dynamic stiffness response is a well-known quantity to identify the dynamic performance of an isolator.

\[
F_T(s) = (K_r + B_r s + A_p^2 \frac{I_i C_2 s^2 + R_i C_2 s + 1}{I_i C_1 C_2 s^2 + R_i C_1 s + C_1 + C_2}) X(s)
\]  
(1-22)

\[
K_{dyn}(s) = \frac{F_T(s)}{X(s)} = K_r + B_r s + \frac{A_p^2}{C_1} \frac{I_i s^2 + R_i s + 1/C_2}{I_i s^2 + R_i s + 1/C_1 + 1/C_2}
\]  
(1-23)

1.3 Experimental Verification and Parameter Identification

A hydraulic engine mount is used here to validate the mathematical model and identify its unknown parameters. This mount will be used later as a platform to develop active and semi-active engine mounts. In this regard, an experimental analysis has been conducted, as illustrated in Figure 1-7.

A magnetic shaker LDS V722 is utilized to produce a sweep sine excitation, while a Sensotec Model 41, 1000-lb precision pancake load cell measures the transmitted force, and an LVDT measures the displacement. In addition, a feedback controller is used to maintain constant amplitude of excitation over the whole frequency range.

A least square curve fitting code in MATLAB is developed to find the unknown parameters of equation (1-23). The identified parameters are shown in Table 1-1. As shown in Figure 1-8, the experimental and numerical results agree well.
Figure 1-7: The experimental setup (a) and its schematic view (b)

Table 1-1: Extracted parameter values for hydraulic engine mount

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_p$</td>
<td>3650</td>
<td>$mm^2$</td>
</tr>
<tr>
<td>$B_r$</td>
<td>0.215</td>
<td>$N-S/mm$</td>
</tr>
<tr>
<td>$K_r$</td>
<td>170</td>
<td>$N/mm$</td>
</tr>
<tr>
<td>$C_1$</td>
<td>2.05e5</td>
<td>$mm^5/N$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>2.03e7</td>
<td>$mm^5/N$</td>
</tr>
<tr>
<td>$I_i$</td>
<td>3.4e-10</td>
<td>$N-S^2/mm^5$</td>
</tr>
<tr>
<td>$R_i$</td>
<td>2.9e-8</td>
<td>$N-S/mm^5$</td>
</tr>
</tbody>
</table>
1.3.1 A Discussion on Engine Mount Equation

Since the aim of this study is to design active and semi-active engine mounts by modifying a conventional hydraulic mount, an essential step is to evaluate different parameters of the mount to find the most suitable parameter to control the overall behaviour of the mount. The hydraulic term in equation (1-23) consists of a second order transfer function in both the numerator and the denominator. The numerator has the smaller natural frequency \((C_2 > C_1)\). Therefore, the amplitude of the dynamic stiffness will grow at the low frequencies, and finally, at frequencies close to the denominator notch frequency, it becomes a flat line, a trend shown clearly in Figure 1-8. The transition between the soft region and the hard region occurs at \(\sqrt{1/I_1C_2}\) and \(\sqrt{(C_2 + C_1)/I_1C_1C_2}\). Since \(C_2\) is bigger than \(C_1\), the second frequency can be estimated by \(\sqrt{1/I_1C_1}\). The approximate level of the stiffness for a hydraulic mount for the soft region, which occurs at low frequencies (less than the first notch frequency), is
\[ K_{\text{ave}}(\text{low freq}) = K_r + A_p^2/(C_1 + C_2) \quad (1-24) \]

and for the hard region (frequencies higher than the second notch), it is

\[ K_{\text{ave}}(\text{high freq}) = K_r + B_r s + A_p^2/C_1 \quad (1-25) \]

\( K_r \) is usually set offline as the property of the bulk rubber to carry the static load of the engine, and \( B_r s \) is a small term that does not make a significant contribution to the dynamic stiffness. \( A_p \) is also a part of the engine mount’s geometry and cannot be changed significantly in the design; therefore, \( C_1 \) can be chosen as the most effective parameter to control the overall behaviour of the mount. Changing \( C_1 \) has no effect on the first notch frequency of the regular mount. The second notch, which is proportional to the square root of \( C_1 \), moves a little. The stiffness of the soft region will be affected by the \( C_1 \); however, it is predicted that the most change will occur with the amplitude at frequencies higher than the second notch. As equation (1-25) suggests, the stiffness at those frequencies has a reverse relation to the compliance of the pumping chamber. Appendix 2 presents a more detailed discussion on the effect of different parameters of the lumped model on the dynamic stiffness of the mount.

Although it is not difficult to identify the key effect of the pumping chamber compliance on the dynamic stiffness response, it is difficult to change this parameter. In fact, the compliance of the pumping chamber is a predetermined value that has been set by the selection of the bulk rubber material and its shape. The future chapters will be concerned mainly with modifying the compliance/pressure inside the pumping chamber by adding active or semi-active parts.
1.4 Literature Review

1.4.1 Modeling of Engine mounts

Hydraulic engine mounts were first presented by (Bernuchon, 1984). The main concept behind the design of hydraulic mounts was to utilize the resistive force of the fluid as an extra element to set the dynamic properties of the mount. The early versions of hydraulic mounts employed only an orifice to connect pumping and compliance chambers. However, in later versions this orifice was converted to inertia track to increase the inertial force of the fluid part. The decoupler was added later to provide an amplitude dependent characteristic to the mount. Bell has also been installed to prevent resonance of the decoupler and to minimize fluid turbulence inside the pumping chamber. Different types of hydraulic mounts with bell and decoupler are reported in literature, but they are conceptually identical, the main difference being in structural details.

(Singh et al., 1992) made the first general model of hydraulic engine mounts. Using extensive literature information, they showed that all previously developed models of hydraulic mounts are special cases of their own model. The nonlinearities of the mount have been addressed by studying the high frequency-low amplitude and low frequency-high amplitude cases separately and excluding either inertia track or decoupler equations. They also reported problems in predicting the high frequency response of the mount by the linear model.

Brach and Haddow made a more comprehensive study on the qualitative behaviour of the engine mounts (Brach and Haddow, 1993). They made a literature review on the engine vibration sources in different frequency ranges and proposed the
desired characteristics of engine mounts based on the information. In a similar study, (Flower, 1985) offered a deep insight into the physics of hydraulic mounts.

The pioneers of nonlinear analysis of hydraulic mounts are Kim and Singh. In (Kim and Singh, 1993) they investigated the nonlinear properties of a hydraulic mount without decoupler and found that the inertia track produces a quadratic fluid damping. They also found that the compressibility of the pumping chamber is due mostly to the flexibility of the rubber – not the compressibility of the fluid. They extended this study by examining the dynamic characteristics of a hydraulic mount within a simplified vehicle model (Kim and Singh, 1995). Their main contribution was to update the model of the decoupler and include its switching properties: They identified the pressure differential within decoupler as a function of the pressure differential and volume of fluid passing through the decoupler.

(Colgate et al., 1995) advanced further the nonlinear modeling of the mount, describing successfully for the first time the high frequency behavior of the mount. Using two linear models for coupled and uncoupled decoupler conditions, they used a piecewise linear technique to show the amplitude dependency of frequency response. The piecewise linear model switches between integrating large and small amplitude equations depending on the position and velocity of the decoupler. They also studied the steady-state response of the mount to a combined excitation, including two sinusoids of different amplitudes and frequencies. This experiment showed that the high frequency behavior of the mount is significantly affected when combined with a low frequency high amplitude excitation.

Using experimental methods introduced by Kim and Singh, (Geisberger et al., 2002) developed an excellent experimental apparatus to identify the nonlinear parameters
of a hydraulic engine mount. Based on the results of this experiment, a fully nonlinear model of the mount has been developed, which accurately predicts the dynamic response of the mount over a wide range of frequencies and amplitudes. This model is the simplest and most practical model to simulate the dynamics of hydraulic engine mounts.

More detailed studies have been performed to identify the nonlinearities of hydraulic engine mounts. (Golnaraghi and Jazar, 2001) developed a simple model of a hydraulic mount with only a decoupler, and experimentally validated the model over a wide range of frequencies. They used the perturbation method to solve the nonlinear equation of the decoupler. (Jazar and Golnaraghi, 2002) have found that the nonlinearity of the decoupler can cause instabilities in higher frequencies. Using analytical and numerical approaches, (Narimani et al., 2004) also showed the occurrence of frequency islands in the frequency response of the piecewise linear isolator.

A reliable evaluation of engine mounts should assess their performance when installed on an engine body system. (Ohadi and Maghsoodi, 2007) simulated the time domain vibrations of a V-shaped engine on three inclined engine mounts based on the nonlinear equation of Geisberger. The six degree of freedom model of an engine is adapted from the work of (Lewitzke and Lee, 2001), and combustion (Paul and Huston, 1980) and unbalance (Snyman et al., 1995) forces are consequently added to the model. However, the model of the vehicle was not included in their study, and the base excitation case has not been studied. According to their results, hydraulic engine mounts have a superior performance over passive mounts in low frequencies, while they do not have a considerable effect at higher frequencies. Using a similar approach, simple models of hydraulic mounts are combined with the modal analysis results of the vehicle to find
the contribution of engine mounts in vehicle vibration (Ushijima and Dan, 1986; Ishihama et al., 1994).

1.4.2 Active Engine Mounts

The trend in the auto industry is to decrease the weight of the vehicles, while the engines are getting more powerful. Based on this trend, the vibration transmissibility to the vehicle’s body is increasing, causing NVH problems. The problem is becoming more serious in the case of newly developed engines such as VDEs, with their complicated vibration pattern. The passive engine mounts are not flexible enough to address the isolation requirements of modern vehicles where active engine mounts are considered the next generation of engine mounts (Miller and Ahmadian, 1992; Swanson, 1993)

Active isolators benefit from one or more actuators, making them capable of defining the dynamic properties at each frequency. The actuators can generate force to counteract the disturbance noise and cancel it; as a result, active systems need a continuous external source of power. The common components of an active mount are a passive mount, force generating actuators, vibration sensors, and a controller. Different types of actuator are reported in the literature for engine isolation, including electromagnetic (Muller et al., 1994), servo-hydraulic (Hodgson 1991), and piezoelectric (Shibayama et al., 1995; Ushijima 1993; Hartono et al., 1994). Among these, piezoelectric actuators are the most suitable for higher frequencies because they have a very fast response, but they can not manage high amplitude displacements. Cost, size, energy consumption, and reliability are common challenges for active systems since they use moving parts, sensors, and an external source of power.
Most of the active engine mounts reported in the literature used a voice coil actuator to control the pressure inside the pumping chamber (Yushiharo et al., 1999; Nakaji et al., 1999; Hills et al., 2005). This method is more efficient for lower frequencies, however, while for higher frequencies it is more practical to impose the force directly to the chassis.

Control of the active mounts is also a challenging area because they are prone to be unstable. Active engine mounts are controlled by either open loop or closed loop strategies, with closed loop demonstrated as having a better performance. (Hills et al., 2005) compared two adaptive algorithms to control their active mount. They compared a filtered-x least-mean-square (FXLMS) adaptive filter and an error-driven minimal controller synthesis (Er-MCSI) adaptive controller, and found both of them showed the same performance in reducing chassis vibration by 50% to 90% under normal driving conditions. (Olsson, 2006) used a closed loop strategy to control an active engine mount, using a gain scheduling technique to deal with the material and large angular engine displacement nonlinearities. (Bohn et al., 2004) presented a practical study to implement the gain scheduling technique to make the isolation performance adaptable with the excitation frequency. Their results showed a considerable reduction in noise and vibration of the passenger compartment. (Karimi and Lohmann, 2007) proposed the use of Haar wavelet-based robust optimal control to control the bounce and pitch motions of the engine, and by numerical results, showed their method had advantages. (Nguyen et al., 2009) developed a hybrid active mount featuring piezostacks and rubber element. A robust sliding mode controller is formulated by considering parameter uncertainties to
reduce the engine vibrations due to base excitation. The controller is then experimentally realized, and its performance has been evaluated in both time and frequency domains.

1.4.3 Semi-Active Engine Mounts

Although active mounts are fully flexible solutions to the isolation problem of engines, they have certain drawbacks including high energy consumption, complex configuration, instability issues, high cost, and maintenance difficulties. Semi-active mounts are median solutions to the isolation problems by being both simple and controllable. They effectively dissipate vibration energy by adjusting the mount’s parameters such as damping. Moreover, semi-active isolators need a smaller control effort than their active counterparts.

The main advantage of semi-active mounts over active ones is that semi-active isolators have no actuating force, so the power consumption and risk of instability is considerably lower for semi-active isolators. Several designs are offered for semi-active engine mounts. (Muler et al., 1996) have proposed a design for a semi-active mount, which opens different inertia tracks with partial vacuum to control the damping of a passive mount. (Graf et al., 1987) presented a semi-active mount that tunes the pressure difference between the upper and lower chambers using hydraulics. It has been shown that the effective compliance of the pumping chamber can be changed in this way. (Kim and Singh, 1995) used a manifold vacuum and a controlled solenoid to control an air pocket inside the pumping chamber. They showed that their design could set the overall damping of the mount on two different values. (Foumani et al., 2002) have used shaped memory alloy wires with on-off controller to tune the compliance of the pumping chamber. In the next study (Foumani et al., 2003) they used a rotary disk mounted on the
inertia track plate to change simultaneously the length of the inertia track and the effective area of the decoupler. However, most of the recent semi-active mounts are designed based on Electrorheological (ER) and Magnetorehological (MR) fluids.

Electro-rheological (ER) and magneto-rheological (MR) fluids are two types of smart materials that have been extensively used in semi-active vibration isolators. ER fluids are a mixture of semi-conducting particles in a dielectric carrier liquid. The shear stress of ER fluid is enhanced when it is exposed to a high electric field in the range of 8 kV/mm (Peel et al., 1996). Although ER fluid has a long history in the literature (Ushijima et al., 1988; Petek et al., 1988; Duclos 1988), it attracts little attention in real applications, the reason being high voltage requirement, low shear strength, and limited operating temperature (Choi et al., 2003).

MR fluids, on the other hand, are more reliable and provide more flexible control capabilities in design than ER fluids. A typical MR fluid is composed of magnetically polarizable particles suspended in a fluid media. When an external magnetic field is applied, the metal particles align themselves with respect to the field and form chains, considerably increasing the viscosity of the mixture. Properties of MR fluids are defined by factors such as type of the metal particles, properties of the viscous fluid, particle volume density, and strength of the magnetic field. Several US patents from LORD Corporation (Carlson et al., 1995) and Delphi Technologies, Inc. (Baudendistel et al., 2003; Hopkins et al., 2006) describe the design of MR fluid hydraulic mounts. Paulstra Corporation has also described the design and use of an MR suspension bushing, which has a performance comparable to that of a multi-link suspension (Piquet et al., 2007). Other than those described by (Arzanpour and Golnaraghi, 2008a), most of the designed
MR mounts are made by replacing the common mixture of water and ethylene glycol with MR fluid. This design increases the risk of early failure and leakage because hydrocarbon-based MR fluids react with natural rubber. Although other ethylene glycol or silicone oil-based MR fluids have been invented which are compatible with natural rubber (Barber and Carlson, 2009), filling the whole mount with MR fluid is not economically feasible.

Magnetorheological elastomer is another type of smart material that has potential for use in semi-active isolators. Magnetorheological elastomers are composite materials of an elastic element with embedded magnetic particles. The magnetic particles are suspended in the elastomer and can be aligned if a magnetic field is applied while the elastomer is cured (Boczkowska et al., 2007). Since MR elastomers do not have a fluid part they are more reliable and easier for mass production. (Shen et al., 2004) proposed a new method of MR elastomer fabrication, by fabricating polyurethane and rubber-based MR elastomers. They could reach a 28% increase in modulus by applying a magnetic field. In similar studies (Zhou, 2004) and (Gong et al., 2005) could achieve 55% and 66% increase in modulus, respectively.

1.5 Variable Displacement Engines Isolation

The emerging demand for more adaptable engine isolation systems arises from the development of new types of engine with more sophisticated performances. Among these are hybrid engines and variable displacement engines (VDEs). To date, hybrid engines are more widely used in the auto industry than variable displacement engines. However, VDEs are now becoming more popular because hybrid engines are fuel efficient in only heavy traffic while they do not affect the fuel consumption in highways.
Moreover, hybrid engines are more expensive and less reliable because of adding different electrical components to the power train.

VDEs on the other hand can save up to 25% in light load conditions (Jackson and Jones, 1976). The main concept used to design VDEs is that a vehicle in steady speed requires a small amount of power (less than 30 HP) compared to the maximum power of modern vehicles, which exceeds 300 HP (Ashely, 2004). Based on this fact, VDE technology deactivates half of the engine cylinders in light load conditions and brings them back in case more power is needed.

VDEs have been commercialized by car manufacturers since 2004 with different names such as Variable Cylinder Management by Honda, Active Fuel Management by General Motors, Multi-Displacement System by Chrysler, and Active Cylinder Control by Daimler. Varying vibration pattern induced by VDEs in the absence of a proper isolating system is still a challenge for the widespread use of this technology.

![Figure 1-9: A comparison of the forces generated by regular engines (a) and variable displacement engines (b)](image)

The main vibration problem of VDEs stems from the fact that the overall generated torque of the engine should remain constant when switching between different cylinder activation modes. As a result, in the half cylinder mode, the combustion pressure doubles, and consequently increases the excitation force on the engine (Matsuoka et al.)
2004). Figure 1-9 shows the excitation force of VDE in normal and half-cylinder modes. To deal with this changing pattern and maintain the isolation performance, a VDE mount should be half as stiff in the deactivation mode while being able to switch back to its original condition in normal operating mode. Switching between two isolation characteristics is beyond the capability of passive engine mounts. To address this requirement, active and semi-active mounts are the next frontiers.

1.6 Thesis Overview

This thesis focuses on the design and development of active and semi-active engine mounts specifically designed to address the isolation requirement of VDEs. In chapter 1, the need for low-cost versatile engine mounts is established and the isolation requirements of variable displacement engines are identified. The problem of engine isolation is first investigated, and the challenges to the design of a proper engine mount are described. Later, the conflicting requirements of engine isolation are shown by a simple 1DoF model. Hydraulic engine mounts are introduced and studied as the best conventionally available solutions to the isolation problem of the engines, and their dynamic stiffness equation is derived analytically. The chapter also presents a review of the available literature for engine isolation.

Chapter 2 describes the design of an active engine mount, which has been designed on the platform of a hydraulic engine mount. This active mount benefits from a solenoid actuator that distinguishes it from previously developed active mounts. The actuator has a reciprocal movement, which controls the pressure inside the pumping chamber. The dynamic stiffness of the developed active mount is derived, and shown to be controllable with the current imposed on the magnetic coil. The chapter also performs
the proof of concept experiments and identifies lumped parameters of the analytical model based on the experimental results.

Chapter 3 is concerned mostly with the control of the developed active engine mount for different practical cases. In the first part, the active mount is controlled to address the isolation requirements of variable displacement engines. In this regard, the notch frequency, amplitude, and switching frequency of the mount are shown to be highly flexible for the frequency range of dominant excitations on the engine. The requirements of regular engines for different driving conditions are also investigated. The desired characteristics of the mount to deal with different driving conditions are extracted from an industrial datasheet, and the mount is controlled to address them separately.

In chapter 4, the design of a semi-active engine mount is described, which is also designed to be used for VDE application. This semi-active mount is designed by retrofitting an auxiliary MR chamber inside a hydraulic engine mount. The mathematical model of the mount is first modified to include the effect of the MR chamber and the dynamic stiffness is numerically simulated. The numerical predictions are then validated with the experimental results to prove the capabilities of our design for VDE application.

Finally, chapter 5 highlights the key findings of this study and presents recommendations for future works.
1.7 Contributions Made in This Thesis

This thesis has contributed to the advancement of knowledge in many areas. The key contributions herein have resulted in the following Journal and Conference papers:


2: DESIGN AND SIMULATION OF THE ACTIVE ENGINE MOUNT

In the first chapter, the isolation requirements of engine-body system have been described for both regular engines and VDEs. Moreover, hydraulic engine mounts are introduced as the best conventionally available devices to address these requirements while their performance still needs improvement to effectively suppress engine vibration. In this regard, the compliance/pressure of pumping chamber has been selected as the most suitable parameter to alter the performance of the passive mount. In this chapter, design and development of an active mount will be described. A solenoid-based actuator is replaced for the decoupler and is controlled to gain the desired dynamic properties. This mount is primarily designed to address the isolation requirements of VDEs, while later on it will be shown that it is flexible enough to be used for the isolation of regular engines as well. Design, development, and proof of concept experiments are described in this chapter, and more practical cases are postponed to the next chapter.

2.1 Design of the Active Mount

As it is described in chapter one, pressure inside the pumping chamber is the most suitable parameter to control the dynamic behaviour of a hydraulic engine mount. To alter this parameter, an electromechanical actuator is designed and retrofitted inside the inertia track plate of a passive hydraulic mount. Decoupler is eliminated from the original design, although its performance is still available by letting the plunger move freely (it has a restricted motion) or by a control signal that derives the actuator which acts similar
to a reciprocating pump. The plunger of the actuator is faced to the pumping chamber from the top and to the atmosphere from the bottom. The plunger is sealed with a thin rubber diaphragm and a compression spring keeps it in its proper position. Figure 2-1 shows different parts of the electromechanical actuator designed for the active mount.

![Electromechanical actuator for the active mount](image)

The actuator used for this active mount is composed of a plunger and a solenoid coil (for more details see appendix 3). Most of the previously developed active engine mounts were using voice coil to generate actuating force and the application of solenoid-based actuators is rather new in the area (Arzanpour and Golnaraghi, 2008). A solenoid-based actuator is more compatible with the harsh environment of the active mount. It is considerably cheaper, more durable, and it has less design/fabrication complexity compared to voice coils.

The main problem with magnetic actuators is their limited displacement functionalities. Since the magnetic force has an inverse relation to the distance of the plunger and the coil, in most cases, the effective operational range of the plunger does not
exceed few centimetres. However, this problem does not affect the performance of our device, as the amplitude of engine mount’s vibration is less than 1 cm, which produces approximately the same displacement on the plunger. The amplitude is even smaller (0.1–0.3 mm) in higher frequencies where the magnetic actuator is in its most effective range.

The actuator used in our design does not benefit from a permanent magnet; as a result, the magnetic coil can only attract the plunger. This problem is addressed by using a compression spring underneath the plunger and giving a DC current to the actuator. The DC offset generates a static displacement for the plunger against the compression spring; therefore, if the imposed current goes below the level of the DC offset the plunger moves farther from the coil compared to its static equilibrium. This will enable us to produce a reciprocal movement on the plunger.

The process of pressure modification starts by activating the electromechanical actuator coil. Then, the magnetic field moves the plunger depending on the imposed current. The motion of the plunger changes the effective volume of the pumping chamber; consequently, the pressure of the pumping chamber will be affected and the dynamic stiffness will change according to the volume change. The other side of the plunger is open to the atmosphere; therefore, $C_2$ is not directly affected by the movement of the plunger.

2.2 Mathematical Modelling and Numerical Simulation

The lumped model of the proposed active mount is illustrated in Figure 2-2. This model is the same as the lumped model of a passive hydraulic mount except for a
reciprocating part faced to the pumping chamber. This reciprocating part represents the actuator.

![Diagram of the lumped model of the active engine mount]

**Figure 2-2: The lumped model of the active engine mount**

Since most of the hydraulic mount components are maintained, the active mount’s governing equations are similar to the passive one other than the continuity equation. In this regard, the system of equation for the active mount will be in the form of,

\[
\begin{align*}
C_1 \ddot{P}_1 + C_2 \dot{P}_2 &= A_p \dot{X}_r + A_a \dot{Y}_a \\
C_2 \dot{P}_2 &= Q_l \\
P_1 - P_2 &= I_l \dot{Q}_l + R_l Q_l
\end{align*}
\] (2-1)

(2-2)

(2-3)

Where \(A_a\) is the effective pumping area, and \(Y_a\) is the displacement of the plunger. Solving equations (2-1) to (2-3) for \(P_1\) will result in

\[
P_1(s) = \frac{A_p X_r(s) + A_a Y_a(s)}{C_1} \frac{I_l s^2 + R_l s + 1/C_2}{I_l s^2 + R_l s + 1/C_1 + 1/C_2}
\] (2-4)

Although the above equation contains some properties of the hydraulic mount, the plunger displacement \(Y_a(s)\) can be set to adjust the dynamics of the isolator. \(Y_a(s)\) can be
controlled as a function of the relative displacement of engine and frame \( X_r(s) \) according to equation (2-5).

\[
Z(s) = Y_a(s)/X_r(s) \tag{2-5}
\]

The dynamic stiffness equation of the active engine mount can then be obtained by combining equations (2-4), (2-5), and (1-14) leads to

\[
K_{dyn}(s) = K_r + B_r s + \frac{A_p^2 + A_a A_p Z(s)}{C_1} \frac{I_1 s^2 + R_1 s + 1/C_2}{I_1 s^2 + R_1 s + 1/C_1 + 1/C_2} \tag{2-6}
\]

To have a better insight from the advantage of retrofitting the actuator in a hydraulic mount, the passive and active terms are separated from each other as shown in Figure 2-3.

![Block diagram of the force generated by the active mount](image)

**Figure 2-3:** Block diagram of the force generated by the active mount

The control parameter is \( X_r(s) \), which is the relative displacement of the engine and the frame. \( F \) is the force, and the subscripts \( a, p, \) and \( T \) stand for active, passive, and transmitted, respectively. The transfer function \( Z(s) \) is defined as the ratio of engine mount and plunger displacements; \( H(s) \) is the plunger displacement and active force, and \( G(s) \) is the mount displacement and passive force. \( H(s) \) and \( G(s) \) are taken from equation (2-6) as, .
The key point here is that most of the mount’s force is provided by the passive part (i.e., \( G(s) \)), and the active part only tunes it. Since the active part generates a small portion of the overall force, the actuator can be chosen significantly smaller than the completely active mounts. This will reduce the weight, size, cost, and power consumption of the actuator. In addition, using a typical passive mount to generate the major part of the resistive force guarantees the same performance of the passive isolation systems in case of failure in the actuator. The nonlinearities of the main rubber part also bound the amplitude of the engine vibrations if instability develops in the controller.

2.3 Numerical-Experimental Verification

An experimental setup is built to investigate the performance of the active mount. Figure 2-4 shows different components of the testbed. The active engine mount is supported by a frame on top and an LDS V722 shaker, which excites the mount from the bottom. The setup is mounted on a heavy concrete base to minimize the effect of the base vibration on the measurements. Dytran accelerometer, Model 3145AG LIVM, is also used as a feedback to the shaker controller. The displacement of the shaker with respect to the fixed frame (which means the elongation of the engine mount) is measured by LVDT. A Sensotec Model 41, 1000 lb precision pancake load cell is also used for sensing the transmitted force. The first natural frequency of the frame is about 20 Hz that as it will be shown in the experimental results produces a small peak around this frequency.
According to equation (2-5) position-control of the plunger needs frequency, amplitude, and phase of the relative engine and chassis displacement. Since the standard dynamic stiffness response experiments are performed at fixed displacement amplitude, the output of the displacement sensor is used as the frequency reference. Based on the excitation frequency, the controller generates the right command signal, which is then amplified and fed to the actuator.

A set of preliminary experiments are performed to validate the mathematical model of the active hydraulic mount. Moreover, these experiments are used to evaluate
the parameters of the isolator. The special cases include fixed, free, and proportional plunger movement. Fixed plunger condition is generated by giving the actuator a constant current, enough to hold it stationary against the changes of the pressure inside the mount. The amount of the current was selected to press the spring of the plunger to its solid length. As a result, the active mount shows a dynamic stiffness response similar to a hydraulic mount. For the second case, the current to the actuator is set to zero, so the plunger was free to move based on the pressure change inside the mount. The resultant dynamic stiffness is soft compared to the passive mount. In fact, the resistive force in this case is mostly generated by the rubber part; therefore, the dynamic stiffness is close to a straight line. Finally, a current proportional to the shaker displacement is used to excite the mount. The dynamic stiffness has the same shape as a hydraulic mount, but it acts a little softer at low frequencies and harder at high frequencies. The amplitude and phase results for these three cases are depicted in Figure 2-5.

![Figure 2-5: Active mount dynamic stiffness response when the actuator is fixed (dotted line), free (solid line), and when Z(s) = I (dashed line)](image)

To simulate the dynamic stiffness of active mount based on equation (2-6) we started with the fixed actuator case, which behaves similar to the passive mount. Later on,
the area of the plunger \( (A_d) \) is substituted to the equation (2-6) and \( Z(s) \) is tuned to minimize the error between numerical and experimental results of the free and proportionally controlled actuators. Figure 2-6, indicates relatively good agreement between experimental and numerical results in both magnitude and phase. The updated parameters for the active mount are shown in Table 2-1.

![Figure 2-6](image)

**Figure 2-6:** Comparison of magnitude and phase of numerical (dashed) and experimental (solid) results for (a) free, (b) fixed, and (c) controlled actuator
Table 2-1: identified parameter values for active engine mount

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_p$</td>
<td>3650</td>
<td>$\text{mm}^2$</td>
</tr>
<tr>
<td>$A_s$</td>
<td>1788</td>
<td>$\text{mm}^2$</td>
</tr>
<tr>
<td>$B_r$</td>
<td>0.215</td>
<td>N-$\text{S/mm}$</td>
</tr>
<tr>
<td>$K_r$</td>
<td>170</td>
<td>N/$\text{mm}$</td>
</tr>
<tr>
<td>$C_1$</td>
<td>2.05e5</td>
<td>$\text{mm}^3/N$</td>
</tr>
<tr>
<td>$C_2$</td>
<td>2.03e7</td>
<td>$\text{mm}^3/N$</td>
</tr>
<tr>
<td>$I_i$</td>
<td>3.4e-10</td>
<td>N-$\text{S}^2$/mm$^5$</td>
</tr>
<tr>
<td>$R_i$</td>
<td>2.9e-8</td>
<td>N-$\text{S}/\text{mm}^5$</td>
</tr>
</tbody>
</table>

A small difference is observed between the numerical and experimental results of the proportionally controlled actuator. The reason is the assumption that the proportional control of the current will result in a proportional displacement with the same phase on the plunger. However, this assumption is not always valid especially in higher frequencies. A more detailed model of the active mount is presented in Appendix 1, which includes the dynamics of the plunger. It should be noted that the position control model is still precise if a closed loop control is used with a feedback from the position of the plunger (not from the relative displacement of the engine and chassis).

2.4 Summary

Design and development of an active mount is described in details. The active mount is made by retrofitting an actuator inside a conventional passive hydraulic mount. The actuator is aimed to control the fluid pressure inside the pumping chamber with no direct effect on the pressure of the compliance chamber. Being retrofittable to a passive mount provides the new mount with several advantages over previously developed active mounts. These advantages include more stability; cost efficiency; fail-safety, and low power consumption. The agreement between numerical and experimental results validates the mathematical model and illustrates the capability of the designed active mount for real application.
3: ACTIVE ENGINE MOUNT EXPERIMENTAL ANALYSIS

In this chapter, more sophisticated control functions are implemented in the controller to demonstrate the abilities of the newly designed active mount in generating different dynamic characteristics. First, the vibration isolation criteria for the VDE application are investigated; these include the notch frequency location, the amplitude of the dynamic stiffness, and the switching frequency between soft and hard regions. Then, several driving conditions are investigated and proper control strategies are utilized to demonstrate the mount’s capability to fulfil the isolation requirements for each condition. The driving conditions include no load, accessory load, cruising at high and moderate speeds on a rough road, cruising at low to moderate speeds on a smooth road, and accelerating the vehicle.

3.1 Capabilities of the Active Mount for VDE Application

As described in section 1.5, Variable Displacement Engines (VDEs) switch between full cylinder and half cylinder modes, dramatically changing the frequency and amplitude of the generated excitation force. To address this varying pattern and to maintain the isolation performance, an engine mount should be able to adjust its isolation properties. Based on the information provided by our industrial collaborator, CooperStandard Automotive Inc. (CSA), the main parameters that need tuning are the notch frequency location, the amplitude of the dynamic stiffness, and the switching frequency between soft and hard regions. The ability of our designed mount to manage these needs is evaluated in this section.
3.1.1 Notch Frequency Location

The notch frequency location is a representation of dynamical systems, and changing its location shows the flexibility of the mount to tune its parameters. For hydraulic mounts particularly, in notch frequencies the fluid resonates within the inertia track, causing maximum energy dissipation by the hydraulic part. For this active mount, the notch frequency location is relocated by adjusting \( Z(s) \) in the dynamic stiffness equation.

![Figure 3-1: Active mount dynamic stiffness response for setting the notch frequency for passive mount (dotted) and active mount (dashed and solid)](image)

The experimental results are depicted in Figure 3-1. The results show that the dynamic stiffness of the regular hydraulic mount (dotted line) is comparable to that obtained by the active mount. The active mount is capable of changing the notch frequency from its original values of 12 Hz and 16 Hz to any desired value between 5 Hz to 20 Hz. This range can even be expanded by adjusting the phase and amplitude of the applied current to the controller.
3.1.2 Dynamic Stiffness Amplitude

The other important capability of an ideal mount is its flexibility to adjust the level of the stiffness. Similar to the notch frequency relocation, $Z(s)$ can be set to tune the level of the stiffness. The experiments are conducted for the frequency range of 10-20 Hz.

![Graph showing dynamic stiffness response for setting the level of stiffness](image)

**Figure 3-2: Active mount dynamic stiffness response for setting the level of stiffness, passive mount (red dotted), hard active mount (black solid), and soft active mount (brown dashed)**

As depicted in Figure 3-2, the experimental results show that the maximum value of the dynamic stiffness can be set to 400 N/mm as opposed to 250 N/mm for the hydraulic mount in the hard region. On the other hand, the stiffness of the active mount can be reduced to 2 N/mm in soft region. The numbers show a huge controllability in the dynamic stiffness response.

3.1.3 Setting the Switching Frequency

Another experiment is planned to demonstrate the ability of the active mount to alter its dynamic stiffness in low frequencies and then set it back to normal. The experiments are performed for frequencies between 20 Hz and 40 Hz. The result
compares active hydraulic mount responses obtained by different control strategies with the regular hydraulic mount responses (red dashed line). As shown, with appropriate control the stiffness level of the mount can be maintained at a certain level and then set back to normal over a wide range of frequencies (in Figure 3-3 from 27 Hz to 36 Hz).

![Figure 3-3: Active mount dynamic stiffness response for setting frequency of stiffness switching; the red dashed line is for the passive hydraulic mount](image)

### 3.2 Proposed Control Strategy for the Active Mount

Most of the studies on active engine mounts focus on the “force transmission to the frame” as the sole control objective (Yu et al., 2001; Hagino et al., 1986), while ignoring two other equally important criteria – acceleration on the engine, and relative engine-body displacement. Such an approach may cause catastrophic acceleration on the engine or misalignment between engine and body in response to the actuating force. The approach followed in our controller is based on selecting an appropriate control function, \( Z(s) \), from a predefined library of functions that corresponds to the current operating condition of the vehicle (such as “high speed cruising on a smooth road”). These control functions are designed to produce the required isolation properties demanded by the manufacturer. A sample of such a datasheet is shown in Figure 3-4.
The vehicle operation conditions can be determined by a combination of information on engine RPM, throttle condition, vehicle speed, and/or acceleration on the chassis. The needed quantities are usually available in modern vehicles for other applications such as automatic transmission or Electronic Stability Control (ESP).
proposed control scheme is shown in Figure 3-5. To the best of our knowledge, the control scheme used here is unique in its kind.

![Diagram of the proposed control architecture to control the active mount](image)

**Figure 3-5: The proposed control architecture to control the active mount**

The control strategy used here is a closed loop control scheme, which gets feedback from the relative displacement of the engine and chassis. Although the desired value of the relative displacement is set to zero, the control functions are not designed to cancel it. Instead, the feedback from displacement is intended to identify the amplitude and phase of the engine displacement, while the control functions are designed to satisfy all three acceleration and displacement requirements simultaneously. In other words, the experimentally extracted dynamic stiffness of the active mount for each control function and its ability to satisfy the requirements for the associated driving condition are the criteria for evaluating the suitability of that control function. In this sense, the steady state error of the mount displacement is not necessarily zero for each function.

To clarify the benefit of the proposed control strategy, we should take into account the information available from the ideal isolation properties. These properties are usually qualitative and perception-based, and in each excitation condition, the priority goes to a different criterion based on personal preference and driving condition. An
important point: In the lack of a reliable objective function, a right trend is the only reliable known information about the desired isolation characteristics. Defining a strict set point and forcing the engine isolation system to stay around that point is waste of power while there is still no guaranty that the performance is satisfactory.

3.3 Capabilities of the Active Mount in Different Driving Conditions

This section considers different loading conditions on the engine mount, including no load, accessory load, cruising at high and moderate speed on a rough road, cruising at low to moderate speed on a smooth road, and accelerating the vehicle. Most of the time a vehicle works on one of these conditions. Each condition produces a dominant excitation at a limited frequency range, which requires that a specific isolation property be suppressed properly. In evaluating the engine mount for each driving condition, lower and higher ranges of frequencies are considered separately, because their desirable trends are not necessarily the same.

The low frequency region is designed to decrease the negative effect of the engine vibration on vehicle dynamics. As all the dominant natural frequencies of the suspension, driveline, and engine-body system are less than 30 HZ, there is a risk that different modes with close natural frequencies couple to each other and cause catastrophic vibrations. A soft engine mount in this condition increases the response time of the vehicle to a step steering and significantly degrades the road handling (similar to a tire with a tall sidewall). The isolation requirements to address this issue will be referred as “vehicle dynamics requirements.”
As opposed to the previous case, a high frequency excitation can not significantly influence the road handling, and its negative effect is limited to the passenger’s inconvenience. The isolation properties to address this drawback will be referred as “ride comfort requirements.” Because of safety concerns, the vehicle dynamics requirements have priority over ride comfort requirements in low frequencies unless the vehicle is stationary.

Figure 3-4 shows the desired engine mount characteristics for each specific driving condition to address vehicle dynamics and ride comfort requirements. The illustrated datasheet has been provided by General Motors to Cooper Standard Automotive to design the hydraulic engine mount used as the test case in our study.

In this section, the real-imaginary format is used to illustrate the complex dynamic stiffness. Although the amplitude-phase format is more common in industry, the real-imaginary format is preferred as it has a direct relation to the stiffness and damping of the isolator. In addition, this format is consistent with the requirements mentioned in the datasheet of Figure 3-4. To convert these two formats to each other, equations (3-1) and (3-2) can be utilized

\[
\text{real} = \text{mag} \cdot \cos(\varphi) \quad \text{imag} = \text{mag} \cdot \sin(\varphi) \quad (3-1)
\]

\[
\text{mag} = \sqrt{\text{real}^2 + \text{imag}^2} \quad \varphi = \arctan\left(\frac{\text{imag}}{\text{real}}\right) \quad (3-2)
\]

where "real" is the real part, "imag" is the imaginary part, "mag" is the magnitude, and "\(\varphi\)" is the phase.
3.3.1 No Load Condition

For the stationary condition that applies in neutral or park with no accessory load (such as A/C, headlights, rear defog), the engine speed is between 650 and 800 RPM. The main concern in this condition is how to deal with the rigid body mode of the engine, and the dominant excitation is the first engine RPM that is in the range of 5-20 HZ. For this case, the mount is required to have low stiffness and high damping. Low stiffness enables rigid body rotational mode of the engine and reduces the amount of force transfer to the body due to the crankshaft’s torque. However, random torque input of the crankshaft falls in the same frequency range; therefore, a high damping is required to suppress the rotational mode of the engine.

Figure 3-6: Active mount dynamic stiffness response, set for stationary in neutral or park with no accessory load (solid line), compared with the passive mount response (dotted line)

The experimental results for this case are depicted in Figure 3-6. Compared with the hydraulic mount, the average stiffness of the active mount is reduced by half, while the damping is doubled. The vehicle is not moving, so the vehicle dynamics requirements need not to be considered.
3.3.2 Accessory Load Condition

The accessory load condition occurs for a stationary vehicle in drive, but with accessory load (A/C, headlights and rear defog) at idle (no throttle input). Accessory load decreases the engine RPM to 600-750 RPM; therefore, the engine will not work as smoothly as the no-load case. Since the excitation force of the engine deviates from a perfect sinusoidal wave, the first engine RPM (i.e., 600-700 RPM) is no longer the dominant excitation frequency. Harmonics (i.e., the whole multiples) of engine RPM will emerge in the excitation force. Given the datasheet provided in Figure 3-4, the major excitation will be the second harmonic of the engine RPM, which falls in the range of 20-40 Hz. A proper engine mount for this case should have low stiffness and low damping to enable the rigid body rotational mode of the engine and prevent the crankshaft’s torque from being transferred to the body. The experimental results of the active mount and the passive mount are compared in Figure 3-7.

![Figure 3-7: Active mount dynamic stiffness response, set for stationary vehicle with accessory load (solid line), compared with the passive mount response (dotted line)](image)

The result shows that the active mount is softer and has lower damping than the passive mount. The average level of stiffness is 170 N/mm compared with 250 N/mm for
the passive mount; for the damping, it is reduced from 70 N/mm to 40 N/mm. As in the no-load case, vehicle dynamics requirements are not considered because the vehicle is stationary.

3.3.3 Cruising At Low to Moderate Speed on A Smooth Road

For cruising at low speed on a smooth road, both vibration and vehicle dynamics requirements need moderate stiffness and damping. Conventional passive mounts are able to fulfil these requirements. The dominant excitation frequency for this situation is 6-40 Hz, in which the decoupler of a conventional mount is closed. The same characteristics can be produced in an active mount by applying enough current to the actuator and holding the plunger stationary against changing pressure in the pumping chamber, as depicted in Figure 2-6b.

3.3.4 Cruising At High Speed or Moderate Speed on A Rough Road

Noise and Vibration Requirements

For cruising at steady speed at 45-80 mph, the engine normally works at 1000-3000 rpm. The excitation falls mainly in the frequency range of 35-400 Hz. For this case, the engine mount should have low stiffness and low damping to stop the induced force of the engine from passing through the engine mounts. However, this range is mostly outside the scope of active vibration suppression, especially with a mechanical actuator. To evaluate the performance, an experimental analysis is conducted on the lower frequency range (35-60 HZ). High frequency requirements, above 60 HZ with very low amplitude, can be considered in the design of the main rubber. According to the
experimental results depicted in Figure 3-8, the active mount has a lower stiffness and
damping than the passive mount.

![Active mount dynamic stiffness response](image)

**Figure 3-8:** Active mount dynamic stiffness response, set for cruising at steady speed in high gear
(solid line), compared with the passive mount response (dotted line)

**Vehicle Dynamics Requirements**

For cruising at high speed (above 45 mph) on a smooth road, and cruising at low
to moderate speed (20-55 mph) on a rough road, it is required that the mount have high
stiffness and damping in the frequency range of 6-20 Hz. The high stiffness requirement
is intended to optimize the apparent mass relative to suspension dynamics. In addition,
high damping is required to optimize power-train response to suspension dynamics. The
experimental result for the frequency range of 6-20 Hz is compared with the result for the
hydraulic engine mount in Figure 3-9. The results show extreme increase in both stiffness
and damping level. The average level of the stiffness in the frequency range of 5-20 Hz is
three times that of the passive one. In addition, the magnitude of the damping is
approximately twice the damping of the passive mount. Both stiffness and damping
follow the same trend as the passive ones do, with no dramatic change observed in the
pattern.
Figure 3-9: Active mount dynamic stiffness response, set for cruising at high speed or moderate speed on a rough road (solid line), compared with the passive mount response (dotted line)

3.3.5 Accelerating the Vehicle

This condition usually occurs when the driver accelerates the vehicle to roll into the highway speed. In this case, the engine vibration excitations cover a wide range of 35-600 Hz. The upper bound of this excitation frequency, 60-600 Hz, will be handled by the low-pass characteristics of the main rubber, and the active actuator should address only the frequency range of 35-60 Hz. An ideal mount should have moderate stiffness and low damping in this condition. The low damping is needed to enable high frequency transmissibility of the mount, and the moderate stiffness is needed to hold the engine against the reaction of the engine torque. As shown in Figure 3-10, this requirement can be achieved by our active mount. Although the damping of the active mount is about 30% higher than the damping of the passive mount in the frequencies below 40 Hz, it decreases to one-third of the passive mount’s damping at 60 Hz.
Figure 3-10: Active mount dynamic stiffness response, set for accelerating the vehicle (solid line), compared with the passive mount response (dotted line)

3.4 Summary

The implementation of several control functions is described to show the capabilities of the developed mount in real applications. First, three criteria are selected to show the flexibility of the mount to manage VDE isolation requirements, and later, those for a regular engine at different driving conditions are investigated. The three criteria for VDE application are the notch frequency location, the amplitude of the dynamic stiffness, and the switching frequency between soft and hard regions; the conditions investigated for a regular engine are no load, accessory load, and cruising in different speed ranges and road conditions. The results of the experiments proved high degrees of flexibility and adaptability for the developed engine mount to satisfy the requirements for both VDEs and regular engines applications.
4: SEMI-ACTIVE ENGINE MOUNT

In this chapter, the design and implementation of a semi-active engine mount for VDE application are described and its performance is investigated by experiments. The vibration isolation requirements of VDE were detailed in chapter 1, and the capabilities of our designed active mount will be demonstrated experimentally in chapter 3. The proposed semi-active mount discussed in this chapter is designed as an alternative solution for vibration isolation of VDEs to reduce cost and complexity. Our semi-active mount is designed on the platform of the conventional hydraulic engine mounts. This modular design eliminates the need to design from scratch, considerably reducing manufacturing cost and time. A novel auxiliary magneto-rheological fluid chamber is developed, and retrofitted inside the pumping chamber. Since the MR chamber controls the stiffness and damping of the mount, it falls in the category of semi-active systems. First, the design of the MR auxiliary chamber is described. To determine the model of the new engine mount, the dynamic equation of the hydraulic mount is modified to include the effect of the retrofitted auxiliary chamber. Finally, the semi-active mount is numerically simulated, and the results are verified experimentally.

4.1 Design and Installation of the Auxiliary MR Chamber

As explained in section 1.5, an ideal engine isolator for VDE application should maintain the same performance as a regular engine mount in normal operations and be capable of becoming soft when the engine switches to half-cylinder mode. Similar to our
designed active mount, here the compliance of the pumping chamber, $C_1$, is selected as the most effective parameter to control the overall behaviour of the mount.

Although $C_1$ is tied to the selection of the mount material and geometry, it can be regulated indirectly by introducing an auxiliary controllable chamber inside the pumping chamber. This chamber is a compressible element that can expand or contract in response to the pressure change inside the chamber. Figure 4-1 shows a schematic of the auxiliary MR chamber.

![Figure 4-1: The schematic view of the MR chamber (a) inside the mount and (b) standalone](image)

The MR chamber is cylindrical, and an iron core with electrical winding is placed in the middle of the cylinder. Both the iron core and the cylinder are integrated as a unit by an aluminium cap at the top. The chamber is sealed by two thin rubber diaphragms at top and bottom. The space between the wire winding and cylinder is filled with MR fluid. The advantage of this design is, only a small volume of MR is needed to fill the auxiliary MR chamber. The air, which is trapped in the top section, creates a spring effect to push back the MR fluid.

The core and the body of the auxiliary MR chamber are both made of soft iron, providing good magnetic field conduction. Application of current to the winding induces
a magnetic field inside the core that can be easily controlled to adjust the viscosity of MR fluid. Because of the high permeability of soft iron and the short gap between the core and body, most of the magnetic field is concentrated in the gap. The field leaves the core in radial direction – that is, aligning the iron particles perpendicular to the MR fluid flow for the maximum flow resistance. Figure 4-2 shows the direction and concentration of the induced magnetic field.

The auxiliary MR chamber is bolted to the metal seat of the hydraulic mount, as shown in Figure 4-3. The bell of the mount is removed to provide more space for the MR chamber. The length of the MR chamber is selected to provide enough clearance between the chamber and the inertia track plate; therefore, the working amplitude of the vibration for the designed semi-active mount is the same as the passive one. When the pressure
inside the pumping chamber increases, the rubber diaphragm in the bottom of the MR chamber bends inside, and pushes the MR fluid up. The air at the top is then compressed. As a result, the pressure inside the pumping chamber drops, and the dynamic stiffness decreases.

![Figure 4-3: Fabricated Semi-Active engine mount (a) and its schematic cut view (b)](image)

4.2 Mathematical Modelling and Numerical Simulation

Adding the new auxiliary chamber inside the hydraulic mount introduces an adjustable element to the passive hydraulic engine mount. When the auxiliary MR chamber responds to the pressure change inside the pumping chamber, both the damping (MR fluid flow) and the spring (enclosed air) elements exist. Since both elements have the same displacement and since their forces are added, they can be considered parallel. Here, the decoupler is maintained in the design. This arrangement guarantees the same isolation performance of passive mounts in higher frequencies.
A lumped model of the semi-active system is depicted in Figure 4-4. In this model, $C_m$ represents the flexibility of the entrapped air. The inertia parameter of $I_m$ is considered to model the MR fluid mass, and $R_m$ represents the damping of the MR fluid flow. Although the resistive force of the MR fluid is a hysteretic force, which is generally velocity and displacement dependent, a viscous damping force is assumed. The linearity assumption is shown to be realistic for the first-order approximation of the response in small amplitude excitations (Shen et. al., 2006; Spencer et. al, 1997; Ikhouane and Rodellar, 2007).

![Lumped model of the designed semi-active engine mount with low frequency excitation assumption](image)

For this model continuity and momentum equations will be in the form of
\[ Q_m + C_1 \dot{P}_1 + C_2 \dot{P}_2 = A_p \dot{X} \]  \hspace{2cm} (4-1)

\[ C_2 \dot{P}_2 = Q_l \]  \hspace{2cm} (4-2)

\[ P_1 - P_2 = I_q \dot{Q}_l + R_l Q_l \]  \hspace{2cm} (4-3)

\[ \dot{P}_1 = I_m \dot{Q}_m + R_m \dot{Q}_m + \frac{1}{C_m} Q_m \]  \hspace{2cm} (4-4)

where, \( Q_m \) is the flow rate of the MR fluid. Substituting equations (4-1) to (4-4) in equation (1-14) will result in

\[ P_1 = \frac{A_p \dot{X}}{C_2/(I_l(C_2 s^2 + R_l C_2 s + 1) + C_1) + [C_m/(I_m C_m s^2 + R_m C_m s + 1)]} \]  \hspace{2cm} (4-5)

And finally the dynamic stiffness of the semi-active mount is,

\[ K_{dyn}(s) = K_r + B_r s + \frac{A_p^2}{C_2/(I_l C_2 s^2 + R_l C_2 s + 1) + C_1} + [C_m/(I_m C_m s^2 + R_m C_m s + 1)] \]  \hspace{2cm} (4-6)

By comparing equation (4-6) with equation (1-23) we see that the two mounts have the same performance when the MR chamber is fully activated and the fluid flow resistance is maximum. The dynamic stiffness of the semi-active mount, based on the extracted passive mount parameters and the auxiliary chamber properties, is simulated and depicted in Figure 4-5.
As can be seen, when we change the viscosity of the MR fluid ($R_m$) the magnitude of the dynamic stiffness shifts, while the phase shows a crossing behaviour. This demonstrates the ability of our semi-active mount to change its stiffness and damping either in the same direction, meaning less than the crossing frequency, or in the opposite direction, meaning more than the crossing frequency. This gives us the opportunity to design the MR chamber so that it can change the stiffness and damping of the mount in the desired directions, which are not necessarily the same for them (as discussed in section 3.3).

It is common for semi-active isolators to use closed loop control schemes; however, an on-off controller serves well for the purpose of this application, given that only the upper and lower boundaries of dynamic stiffness – either hard or soft – are required by VDEs. This controlling technique also reduces the complexity and cost for real application implementation. In the fully on mode, the auxiliary chamber is at the maximum magnetic field, which closes the MR passage completely, and as a result, the auxiliary chamber is inactive and the mount is hard. On the other hand, in the fully off
mode no magnetic field is applied to MR fluid, and therefore, it passes relatively freely through the gap, so the mount becomes soft.

4.3 Experimental Analysis

The experiments on the semi-active mount are performed for the frequency range of 5 Hz to 60 Hz, covering the VDE major excitation range. The amplitude of the excitation is selected 0.1 mm pk-pk. This selection will not theoretically affect the result, because the dynamic stiffness is the ratio of the transmitted force to the amplitude of the motion. The experiment is performed at maximum, medium, and no current. At the maximum current, the high magnetic field blocks the MR fluid passage and the diaphragm cannot move; as a result, the semi-active engine mount behaves similar to a conventional hydraulic mount. But in the no-current case there is no magnetic field, so the MR fluid moves at its minimum shear stress and the mount poses the minimum dynamic stiffness.

![Figure 4-6: Experimentally extracted dynamic stiffness of the semi-active mount in fully on (solid line) and fully off (dotted line) and partially on (dashed line) conditions](image)

60
By comparing the maximum and no-current cases depicted in Figure 4-6, it is found that the first notch frequency remains at 6 Hz for both cases. For low frequencies, the semi-active mount is 20 N/mm softer than the conventional hydraulic mount; for high frequencies, the second notch frequency moves slightly to the left – as was predicted by our model in section 1.3.1. The semi-active mount is much softer than the conventional hydraulic bushing for high frequencies. In the frequency range of 25-55 Hz, the semi-active mount is 50 N/mm softer than the conventional hydraulic mount. In fact, the cases discussed here are just two extremes that show the borders of change in dynamic characteristics of the designed semi-active mount. The response can also be maintained between those boundaries by adjusting the electrical current, as depicted in Figure 4-6.

Inertia, compliance, and resistance of the auxiliary chamber are the main design parameters to set the behaviour of the response as a function of excitation frequency. $C_m$ is influenced mostly by the size of air enclosure and the flexibility of the rubber diaphragm. The inertia of the MR chamber is also defined by the mass of the MR fluid and the diaphragms, which can be adjusted by changing the diameter and the length of the chamber, as well as by the thickness of the diaphragm, and by the gap size. In addition, the gap size defines the limits of damping for the mount. Another parameter that affects damping characteristics is the fluid compound: A high density of iron particles in the MR fluid results in more shear stress. These parameters can be used to design the optimum auxiliary MR chamber for each special application.
4.4 Summary

This chapter discusses the design and development of a new semi-active engine mount designed to regulate the complicated vibration pattern of the newly developed variable displacement engines (VDE). As was shown in chapter 1, the compliance of the pumping chamber is the most efficient parameter for controlling the overall dynamic stiffness of the mount. Since the compliance of the pumping chamber is constrained by some limitations, an auxiliary MR chamber is designed, to retrofit inside the pumping chamber. The new compliance chamber is a controllable pressure regulator, which can alter the dynamic performance of the mount. Conventional hydraulic passive mounts are selected as the platform for the semi-active mount design. The experimental results show that the semi-active mount can change considerably the dynamic stiffness response. The minimum cost, the least-design modifications from the hydraulic mount, and the low power requirements are the important features of this semi-active mount.
5: CONCLUSION AND FUTURE WORKS

5.1 Conclusions

The design and development of two active and semi-active engine mounts are
described in this thesis. In chapter 1, the isolation requirements of engines and
specifically variable displacement engines were described in detail. The mathematical
model of the hydraulic engine mounts was derived, and a sensitivity analysis was made
on different lumped parameters. The compliance-pressure of the pumping chamber has
been selected as the most suitable parameter to control the overall behaviour of the
mount. This conclusion has been used in the design of our proposed active and semi-
active engine mounts.

The design and development of an active mount were described in chapter 2. The active
mount is made by retrofitting an actuator inside a conventional hydraulic engine mount.
The actuator is intended to control the fluid pressure inside the pumping chamber, with
no direct effect on the pressure of the compliance chamber. The fact that the new mount
is retrofittable to a passive mount and avails of a special control strategy provides the
new mount with several advantages over previously developed active mounts, including
improved stability, cost efficiency, fail-safety, and low power consumption. The validity
of the mathematical model was evaluated by special control functions, including free
actuator, fixed actuator, and proportionally controlled actuator. Although the numerical
and experimental results show acceptable agreement with each other, an improved model
is proposed in Appendix 1 to include the dynamics of the actuator. This model can
potentially predict more accurately the dynamic properties of the mount. The findings in chapter 3 showed that the designed active mount can successfully relocate the dynamic stiffness notch frequencies, demonstrating the flexibility of the system and its remarkable adaptability for special cases. The mount can become as soft as 2 N/mm, compared with 170 N/mm of the regular hydraulic mount; and as hard as 400 N/mm, compared with 250 N/mm for a hydraulic mount. Given this performance, the proposed active mount appears to be an ideal solution to the isolation requirements of the VDEs, where different stiffness and damping characteristics are needed. The chapter also proposed a special control scheme to identify the driving condition from engine RPM, throttle condition, velocity, and acceleration on the chassis. The controller then selects a proper control function from a predefined library of the functions to remedy the isolation requirements of the associated driving condition – guaranteeing a satisfactory performance in fulfilling all criteria for engine isolation. The driving conditions investigated in this thesis are in accordance with a statement of requirement provided by General Motors for Cooper Standard Automotive, our industrial collaborator. Our designed mount successfully meets the isolation requirements for all proposed driving conditions.

The design and development of a new semi-active engine mount was discussed in chapter 4. This engine mount was designed specifically to meet the isolation requirements of variable displacement engines (VDE). Conventional hydraulic mounts were selected as the platform to design the semi-active mount. The compliance of the pumping chamber was chosen as the best parameter to control the overall dynamic stiffness. Since the compliance of the pumping chamber is constrained to some limitations, an auxiliary MR chamber was designed and retrofitted inside the pumping
chamber. Dynamic stiffness of the designed semi-active mount was derived, and experimentally demonstrated to have the ability to control the stiffness and damping either in the same or in the opposite direction by setting the design parameters. This novel design allowed us to lower the dynamic stiffness by as much as 50 N/mm in the frequency range of VDE excitation. The mount was first tested in fully on and fully off modes to evaluate its suitability for VDE isolation application. It was also tested for intermediate current level, and it demonstrated the capability of sweeping between the two boundaries. The main advantages of this semi-active mount are minimum cost, little host engine mount modification, and low power requirement.

5.2 Recommendations:
The following recommendations are made as a result of this research:

- Include the non-linear equation of the solenoid in the model of the active mount
- Couple the model of the active mount with full vehicle model to evaluate its performance in dealing with different inputs to the engine
- Perform proper modelling and simulation to evaluate the effect of several active mounts acting on the same engine simultaneously (considering their interaction and control challenges)
- Implement a multi-objective optimal control to the active mount and, compare its results with the proposed control scheme in this study.
- Model the semi-active engine mount with finite element softwares considering fluid-structure interaction to study the effect of different design parameters of the MR chamber
APPENDICES
Appendix A

The governing equations of the active mount are derived in chapter 2 based on the position control of the plunger. However, this is not a practical case, because the imposed current to the coil generates force, not displacement. Getting feedback from the position of the plunger is also not a practical solution because it is costly and is difficult to implement. In this regards, in this appendix the dynamics of the actuator will be included in the equations, and finally, the dynamic stiffness will be calculated as a function of the implemented control function to the current.

First, consider rewriting the momentum and continuity equation of the active mount

\[ C_1 \dot{P}_1 + C_2 \dot{P}_2 = A_p \ddot{X} + A_a \dot{Y}_a \]  \hspace{1cm} (A-1)

\[ C_2 \ddot{P}_2 = Q_t \]  \hspace{1cm} (A-2)

\[ P_1 - P_2 = I_t \dot{Q}_t + R_t Q_t \]  \hspace{1cm} (A-3)

which will lead to

\[ P_1(s) = \frac{A_p X_r(s) + A_a Y_a(s)}{C_1} \frac{1}{s^2 + R_t s + 1/C_2} \]  \hspace{1cm} (A-4)

and

\[ K_{dy}m(s) = K_r + B_p s + \frac{A_p^2 A_a Z(s)}{C_1} \frac{1}{s^2 + R_t s + 1/C_2} \]  \hspace{1cm} (A-5)

The equation of motion for the plunger can be added here to relate the position of the plunger \((Y_a)\) with the magnetic force of the coil \((F_{mag})\) and pressure inside the pumping chamber \((P_1)\).
\[ M_a \ddot{Y}_a + C_a Y_a + K_a Y_a = F_{mag} - P_1 A_a \]  \hspace{1cm} (A-6)

where \( M_a \) is the mass of the plunger, and \( C_a \) and \( K_a \) represent the damping and stiffness of the actuator with respect to the body of the actuator, respectively.

Equation (A-6) can be expressed in Laplace form

\[ Y_a(s) = \frac{F_{mag}}{M_a s^2 + C_a s + K_a} - P_1(s) \frac{A_a}{M_a s^2 + C_a s + K_a} \]  \hspace{1cm} (A-7)

For easiness, we can define the dynamic flexibility of the plunger as

\[ \beta = \frac{1}{M_a s^2 + C_a s + K_a} \]  \hspace{1cm} (A-8)

and the hydraulic dynamic term as

\[ \alpha = \frac{l_1 s^2 + R_1 s + 1/C_2}{l_2 s^2 + R_1 s + 1/C_1 + 1/C_2} \]  \hspace{1cm} (A-9)

By substituting these terms, equation (A-7) becomes

\[ Y_a(s) = F_{mag} \beta - P_1(s) A_a \beta \]  \hspace{1cm} (A-10)

And by substituting \( Y_a \) into equation (A-4), the pressure of the pumping chamber becomes

\[ P_1(s) = \frac{A_p X(s) \alpha}{C_1} + \frac{A_a F_{mag} \alpha \beta}{C_1} - \frac{P_1(s) A_0^2 \alpha \beta}{C_1} = \frac{A_p X(s) \alpha + A_a F_{mag} \alpha \beta}{C_1 + A_0^2 \alpha \beta} \]  \hspace{1cm} (A-11)

Now, assume a linear relation between the imposed current to the coil and the induced force on the plunger. This assumption is valid for small amplitude vibrations of the plunger around its equilibrium position. In practice, the equilibrium position will be defined based on the DC current to the coil and the stiffness of the compression spring. Linearity assumption implies that
\[ F_{mag} = \gamma I \]  

(A-12)

As with the position control case, the relative displacement of the engine and chassis will be used as a feedback, but now the current to the coil will be controlled instead of the position of the plunger.

\[ V(s) = I(s)/X_r(s) \]  

(A-13)

And the dynamic stiffness equation becomes

\[ K_{dyn}(s) = K_r + B_r s + \frac{A^2_\alpha \alpha + A_a A_p V(s) \alpha \beta \gamma}{C_1 + A^2_\alpha \alpha \beta} \]  

(A-14)

Equation (A-14) calculates the force of the mount to the engine from the relative displacement of the engine and chassis. However, the reaction of the imposed force to the plunger should be included if the transmitted force to the chassis has to be calculated. The reaction force can be found as

\[ F_{react} = -M_a \ddot{Y}_a - P_1 A_a = C_a \ddot{Y}_a + K_a Y_a - F_{mag} \]  

(A-15)

By substituting \( Y_a \) and \( P_1 \) from equations (A-10) and (A-11), we get

\[ F_{react} = (C_a s + K_a)(F_{mag} \alpha - \frac{A_p X_r(s) \beta \alpha + A_a F_{mag}(s) \alpha \beta^2}{C_1 + A^2_\alpha \alpha \beta}) - F_{mag} \]  

(A-16)

After substituting \( F_{mag} \) from equation (A-12), the reaction force can be included in the dynamic stiffness equation as
\[ K_{dyn}(s) = K_T + B_T s + \frac{A_p^2 \alpha + A_a A_p V(s) \alpha \beta}{c_1 + A_a^2 \alpha \beta} + (C_a s + K_a)(V(s) \gamma \alpha - \frac{A_p X(s) \beta \alpha + A_a V(s) \gamma(s) \alpha \beta^2}{c_1 + A_a^2 \alpha \beta}) - V(s) \gamma \]

(A-17)

Note that equation (A-17) is valid only for calculating the transmitted force to the chassis; equation (A-14) should still be used for calculating force transmitted to the engine.
Appendix B

In this appendix, the outcome of change in different lumped parameters of passive engine mount is numerically investigated. Equation (1-23) is simulated to find the dynamic stiffness of the engine mount according to the extracted parameters in section 1.3. Each parameter is then varied from its original value by -66\%, -33\%, +33\%, and +66\%. The real and imaginary parts of the dynamic stiffness are plotted versus frequency, which represent the stiffness and damping of the hydraulic mount.

![Image](image_url)

Figure B-1: Typical change in the stiffness (top) and damping (bottom) of the engine mount by change in $C_1$, the red line represents the original values (Simulated)

In the first step, the effect of change on the compliance of the pumping chamber ($C_1$) is studied, and shown in Figure B-1. As predicted by equations (1-24) and (1-25), the average level of dynamic stiffness is reduced by increasing the compliance of the pumping chamber both at lower and higher frequencies. The effect of the compliance of the lower chamber is also investigated – there is no noticeable change in either stiffness or damping.
At the next stage, the stiffness \((K_r)\) and damping \((B_r)\) of the main rubber are varied to evaluate their effect on the overall dynamic response. As shown in Figure B-2 and Figure B-3, changes in the stiffness and damping of the rubber part directly affect the
stiffness and damping of the mount, respectively. However, these are interconnected, and their selection is constrained to bear the static load of the engine.

Finally, the effect of inertia track parameters is investigated. As shown in Figure B-4, by increasing $I_l$ both stiffness and damping deviate further from a straight line; however, the average level of stiffness and damping is not significantly affected by change in $I_l$.

![Figure B-4: Typical change in the stiffness (top) and damping (bottom) of the engine mount by change in $I_l$; the red line represents the original values (simulated)](image)

As in the previous case, the resistance of the inertia track ($B_r$) does not significantly affect the average level of the stiffness and damping. As illustrated in Figure 5-5, the most affected region is the frequencies close to the first resonance of the inertia track. By decreasing $B_r$, the resonance shows a more distinct behaviour, while by increasing it, the stiffness and damping curves follow approximately a straight line.
Figure 5-5: Typical change in the stiffness (top) and damping (bottom) of the engine mount by change in $R_i$; the red line represents the original values (simulated)
Appendix C

This appendix contains the datasheet of the actuator used for the design of the active mount. The AWG number for our actuator is 21.

**Magnetic Sensor Systems**

*Tubular Low Profile Clapper Solenoid*

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**HEAT SINK:** For proper heat dissipation, body of solenoid should be mounted on an equivalent of 5.0” x 5.0” x 1/4” aluminum plate in an unrestricted flow of air.

9901 Woodley Avenue, Van Nuys, California 91406
Telephone: (818) 785–6244 Fax: (818) 785–5713
www.solenoidcltv.com
MAGNETIC SENSOR SYSTEMS

S-16-264

MECHANICAL DIMENSIONS

TOLERANCES: (UNLESS NOTED)
O.XXX: ±0.005
O.XX: ±0.010
O.X: ±1/64
COIL RESISTANCE: ±10%
DIMENSIONS IN INCHES [mm]

SOLENOID SHOWN ENERGIZED

TYPICAL PULL FORCE VERSUS STROKE

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REFERENCE LIST


Arzanpour, S. & Golnaraghi, M.; 2008b, “Development of a bushing with an active compliance chamber for variable displacement engines”, Vehicle System Dynamics, Taylor and Francis Ltd, 46, 867-887


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