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SIMULATION OF ENGINE VIBRATION ON NONLINEAR HYDRAULIC ENGINE MOUNTS THROUGH FULL-VEHICLE MODEL

A.R.Ohadi [1], Gh.Maghsoodi [2] and V.Fakhari [1]

[1]Mechanical Engineering Department, Amirkabir University of Technology, Tehran, Iran
[2]Automotive Engineering department, Iran University of Science and Technology, Tehran, Iran
a_r_ohadi@aut.ac.ir

Abstract

This paper represents the simulation of vibrational behaviour of engine on nonlinear hydraulic mounts. Inertia track and decoupler are included in the model of hydraulic engine mounts and nonlinear factors such as inertia and decoupler resistances in the turbulent region are considered. In order to investigate the responses of the engine under engine and road excitations, engine is located on the 7 DOF vehicle model through its three hydraulic mounts. Primarily, the nonlinear properties of hydraulic mount are identified and a lumped parameter mathematical model of the mount is developed. After that, a thirteen degree of freedom full vehicle model including six degree of freedom for translations and rotations of engine, three degree of freedom in bounce, pitch and roll for sprung mass and one degree of freedom in vertical displacement for each four unsprung masses are developed. The 6 degree of freedom four cylinders V-shaped engine under inertia, balancing masses and mounting reaction forces and torques is considered. In deriving the governing equations of motion of sprung mass, the force exerted by the suspension system on the sprung mass and also the reaction forces from engine mounts are considered. Simulation results can be used to predict the dynamic responses of the system under engine and road excitations in the time and frequency domains. By solving the time domain nonlinear equations of motion of full vehicle model, translational and rotational motions of sprung mass and engine's body are obtained for different engine speeds and base excitations. Transmitted base forces are also determined for each loading condition. Using FFT, the response of the system in the frequency domain is studied. In addition, in order to investigate the efficiency of hydraulic mounts, the obtained results are compared to the corresponding results of engine on rubber mounts. The results show that considerable improvement in vibration and isolation of the engine and the body can be achieved in low frequency region. However, for high frequency region, behaviour of the engine on the hydraulic mounts is the same as on the rubber mounts.

1. INTRODUCTION

The engine-body system of automotives is subjected to undesirable vibrational input from the engine and road excitations. The engine excitation is typically in the range of 10 to 200 Hz, while the road excitation is typically below 30 Hz. One approach to engine vibration isolation is to carefully design the mount. This isolation approach requires the mount to exhibit the conflict characteristics of large stiffness and large damping for the low-frequency excitations and low stiffness and low damping for the high-frequency excitations. The dependence of the response of Hydraulic Engine Mount (HEM) on the frequency and amplitude of excitation permit the conflicting damping and stiffness characteristics mentioned above to be met.

The first paper about hydraulic engine mount was presented by Bernuchon [1] in 1984. In this paper the physical properties of a hydraulic mount including inertia track are investigated. Various linear and nonlinear HEM models have been presented up to now. Singh et al. [2] developed the linear lumped parameter mathematical models for both free and fixed type decoupler. By dividing the excitations into “large-amplitude low-frequency” and “small-amplitude high-frequency” categories, the decoupler nonlinearity is avoided and the two linear models (that represent the behaviour of the total system) are considered. An extensive nonlinear analysis of the hydraulic mount has been done by Kim and Singh [3]. A nonlinear lumped parameter model of HEM with inertia track is formulated and compared with experimental data over time and frequency domains from 1 to 50 Hz. Also, the nonlinear lumped parameter model is updated to include nonlinear switching behaviour of the decoupler [4-5] and experimental verification is presented.

Geisberger et al. [6] develop a complete non-linear model of a hydraulic engine mount and evaluate the model using a unique experimental apparatus. It is shown that the developed model provides the appropriate system response over the full range of loading conditions. Vibration behaviour of a six degree of freedom four cylinders V-shaped automotive engine supported by three hydraulic engine mounts has been studied by Ohadi and Maghsoodi [7]. In order to investigate the efficiency of the hydraulic mounts, the obtained results were compared to the corresponding results of the rubber mounts. However, to the best knowledge of the authors, vibration behaviour of engine on full-vehicle model with rubber and hydraulic mounts are not investigated in any literature.

In the present paper, a thirteen degree of freedom full-vehicle model including engine, sprung mass and unsprung masses is developed. The dynamic responses of the system under engine and road excitations are determined in the frequency domain. In the following sections, after presenting the governing equations of the hydraulic engine mount, the equation of motions of the full-vehicle model including engine is developed. Inertia track and decoupler are included in the model of hydraulic engine mounts and nonlinear factors such as inertia and decoupler resistances in the turbulent region are considered. Before the conclusion section, vibration behaviour of engine on hydraulic engine mounts and rubber mounts under engine and road excitations are investigated through full vehicle-model.

2. HEM SYSTEM AND THE GOVERNING EQUATIONS

The cross section of a HEM is shown in Figure 1. It includes the main rubber, flexible rubber diaphragm and two fluid-filled chambers connected through a decoupler and inertia track. Inertia track is a long, narrow channel to provide the fluid damping whereas decoupler has a wider orifice with a free floating disk. During “high-frequency small-amplitude” excitations, the fluid passes freely through the decoupler into lower chamber. It makes low damping in HEM performance. However, during “low-frequency large-amplitude” excitations, large amount of fluid is flown through the inertia track and high damping will be obtained. Therefore, the damping of the hydraulic mount is dependent on the amplitude and frequency of the excitation. This feature makes superiority of HEM to the conventional rubber mount.

A lumped parameter model of HEM is illustrated in Figure 2. Parameters k_r, C_r, C_1 and

C_2 denote the structural stiffness, damping, upper and lower chamber compliances, respectively. Inertia and resistance corresponding to the inertia track and decoupler are represented by I_i, R_i, I_d and R_d, respectively. Also, the parameters and states A_d, A_p, P_1, P_2, Q_i, Q_d and X_e denote the decoupler area, effective pumping area, upper chamber pressure, lower chamber pressure, flow rate through inertia track, flow rate through decoupler and displacement excitation, respectively.

The nonlinear governing equations are obtained by bond graph method [5, 6]. Regarding to this method, the continuity equations for chambers are

$$C_1 \dot{P} = A_p \dot{X}_e - Q_i - Q_d \tag{1}$$

$$C_2 P_2 = Q_i + Q_d \tag{2}$$

The momentum equation for fluid columns in inertia track and decoupler are

$$P_1 - P_2 = I_i \dot{Q}_i + (R_i + R_i' |Q_i|) Q_i \tag{3}$$

$$P_1 - P_2 = I_d \dot{Q}_d + (R_d + R_d' |Q_d| + R_{add}) Q_d \tag{4}$$

Where R_{add} is a relatively large resistance in order to consider the effect of flow-stopping through the decoupler in low-frequency large-amplitude excitations. This nonlinear term is introduced by [6]

$$R_{add} = R_0 \, e^{(\frac{X_d}{X_0}) \arctan(\frac{Q_d}{Q_0})} \tag{5}$$

In the above equation, the parameters R_0, X_0 and Q_0 are constants and X_d denotes the decoupler position obtained by

$$X_d = \frac{\int Q_d \, dt}{A_d} \tag{6}$$

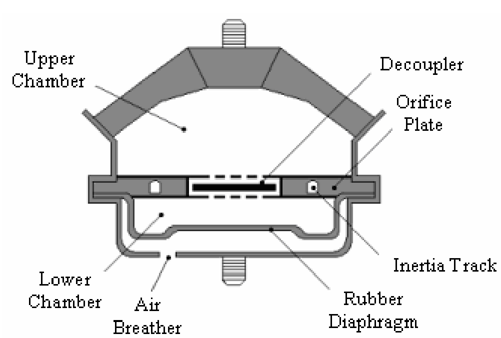

Figure 1. Cross-section of hydraulic engine mount [6].

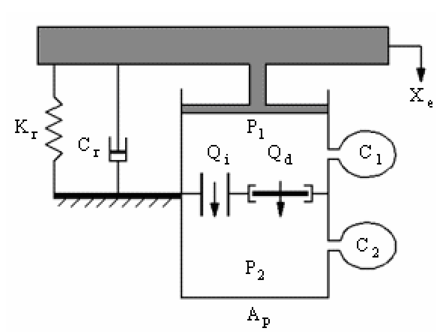

Figure 2. Lumped parameter model [6].

The nonlinear transmitted force equation is defined as below:

$$F_T = K_r X_e + C_r \dot{X}_e + (A_p - A_{d-fnc})(P_1 - P_2) + A_p P_2 + A_d (R_d + R'_d |Q_d|) Q_d \tag{7}$$

Where A_{d-fnc} is a continuous function of the decoupler position and pressure differential and introduced by [6]

$$A_{d-fnc} = \frac{A_d}{\pi}\left(\arctan\left(\frac{X_{d-\max} - \left(\frac{2}{\pi}\right) X_d \arctan\left(\frac{P_1 - P_2}{P_0}\right)}{X_1}\right)\right) + \frac{A_d}{\pi}\left(\frac{\pi}{2}\right) \tag{8}$$

The constants X_1 and P_0 are defined to normalize the function.

3. MOTION EQUATIONS OF THE FULL-VEHICLE MODEL

Consider a 13 degree of freedom full-vehicle model as shown in Figure 3. The vehicle is front wheel drive type and the engine is supported by three inclined mounts. A thirteen degree of freedom full-vehicle model including six degree of freedom for translations and rotations of engine, three degree of freedom in bounce, pitch and roll for sprung mass and one degree of freedom in vertical displacement for each four unsprung masses under engine and road excitations are developed. In order to derive motion equations of full-vehicle model, two coordinate systems are used, the engine coordinate system located at centre of mass of the engine and the sprung mass coordinate system with its origin at centre of mass of the sprung mass (Figure 4). In both coordinate systems, the z–axis is parallel to the crank shaft and the x-axis is in the vertical direction.

3.1 Motion Equations of the Engine

The engine is assumed to be a rigid body of mass M_e connected to a chassis by three HEMs as shown in Figure 3. Suppose that the engine coordinate axes are coincident with the principal axes of inertia. Therefore, general translational and rotational equations of engine motion in its coordinate system become:

$$\sum F_x = M_e \ddot{x}_e \;\; ; \;\; \sum F_y = M_e \ddot{y}_e \;\; ; \;\; \sum F_z = M_e \ddot{z}_e \tag{9}$$

$$\sum T_x = I_{ex} \ddot{\theta}_{ex} - \left(I_{ey} - I_{ez}\right) \ddot{\theta}_{ey} \ddot{\theta}_{ez} \;\; ; \;\; \sum T_y = I_{ey} \ddot{\theta}_{ey} - \left(I_{ez} - I_{ex}\right) \ddot{\theta}_{ez} \ddot{\theta}_{ex}$$

$$\sum T_z = I_{ez} \ddot{\theta}_{ez} - \left(I_{ex} - I_{ey}\right) \ddot{\theta}_{ex} \ddot{\theta}_{ey} \tag{10}$$

Where I_{ex}, I_{ey}, I_{ez} and θ_{ex}, θ_{ey}, θ_{ez} represent the moments of inertia and rotational angles about axes of engine coordinate system, respectively. Also, $\sum F_x$, $\sum F_y$, $\sum F_z$ and $\sum T_x$, $\sum T_y$, $\sum T_z$ are the resultant forces and moments acting on engine block in the x, y and z directions, respectively.

Three kinds of forces and moments are applied to the engine block: inertia forces and moments generated by the moving links of the slider-crank mechanism, forces and moments due to rotating balancing masses, reaction forces and moments applied to the engine by three HEMs. These forces and moments have been derived in previous works [7, 8].

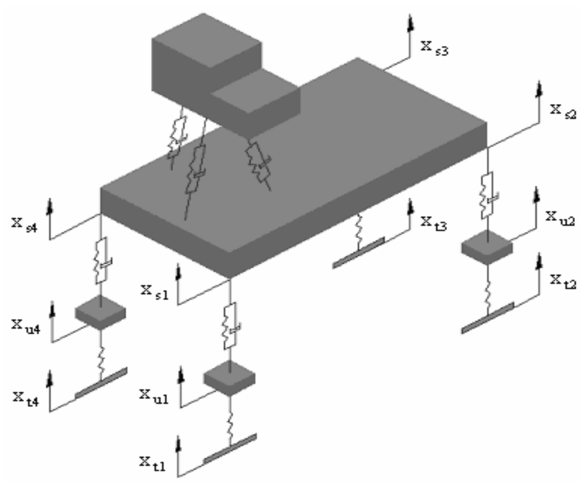

Figure 3. Full-vehicle model.

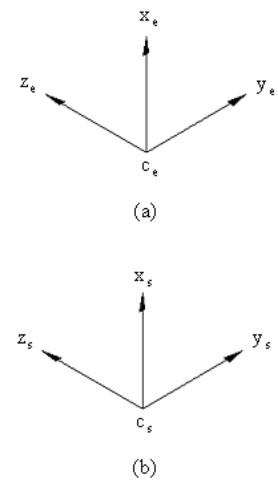

Figure 4. the coordinate systems of (a) engine (b) sprung mass.

3.2 Sprung Mass and Unsprung Mass Motion Equations

The motion equations of sprung mass in its coordinate system can be obtained by

$$M_s \ddot{x}_s = F_{mx-s} + F_{sx} \;;\; I_{sy} \ddot{\theta}_{sy} = T_{my-s} + T_{sy} \;;\; I_{sz} \ddot{\theta}_{sz} = T_{mz-s} + T_{sz} \tag{11}$$

Where M_s, I_{sy} and I_{sz} denote the mass of sprung mass, moment of inertia of sprung mass about y and z axes. Also, F_{mx-s} , T_{my-s}, T_{mz-s} and F_{sx}, T_{sy}, T_{sz} represent forces and moments exerted to the sprung mass due to the HEMs and the suspension system, respectively.

The quantities associated with each unsprung mass are represented by index i ($i=1,2,3,4$). Therefore, the motion equations of each unsprung mass are derived as follow

$$M_{ui} \ddot{x}_{ui} = F_{sx} - k_{ti} \left(x_{ui} - x_{ti} \right) \tag{12}$$

Where M_{ui}, k_{ti}, x_{ui} and x_{ti} denote the unsprung mass, tire stiffness, unsprung mass displacement and the road excitation, respectively.

4. SIMULATION RESULTS

Vibration behaviour of engine on HEMs and rubber mounts under engine and road excitations are investigated through full vehicle-model. In this regard, the parameters associated with the engine and the HEMs are considered the same as references [8] and [6], respectively. The parameters corresponding to the sprung and unsprung masses, tire and suspension system are selected the same as [9]. It is assumed that hydraulic and rubber mounts have the same elastic stiffness and damping values. The amplitude of road excitation and the speed of engine are assumed to be 5 mm and 3000 rpm, respectively. The nonlinear equations ((9)-(12)) and internal dynamic equations of hydraulic mounts ((1)-(8)) are solved by MATLAB SIMULINK and therefore time response of engine under road and engine excitations are obtained. Also, Using FFT method, the frequency response of engine under road excitations is studied.

Table 1 represents the natural frequencies of the engine motion on full-vehicle model with hydraulic mounts. Figures 5 to 10 show the comparisons of the engine displacements in the translational and rotational directions and the transmitted forces to the body versus frequency of road excitation for the hydraulic and the rubber mounts. The peak points in these figures are related to the natural frequencies of engine. However, because of damping, these frequencies are not exactly coincident with the values presented in table 1. These figures indicate that, in the low frequency region, the magnitudes of displacements of the engine with hydraulic mounts are less than the corresponding values of the engine with the rubber mounts. Therefore, considerable improvement in vibration and isolation of the engine and the body can be achieved in low frequency region.

Table 1.Natural frequencies of engine on full-vehicle model with hydraulic mounts.

	f_1 (rad/s)	f_2 (rad/s)	f_3 (rad/s)	f_4 (rad/s)	f_5 (rad/s)	f_6 (rad/s)
Hydraulic Mounts	30	31	41	48	68	78

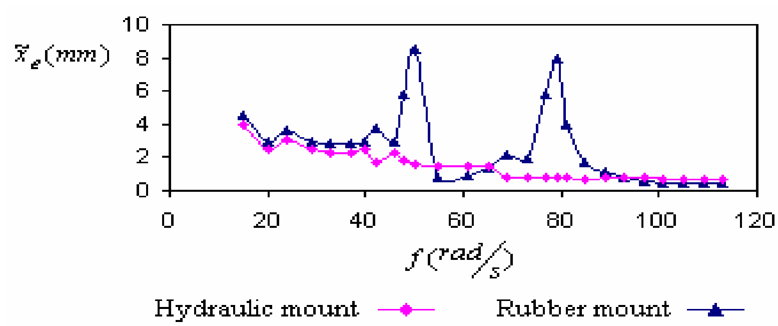

Figure 5. Variation of steady state amplitude of engine displacement in “x” direction versus frequency of road excitation.

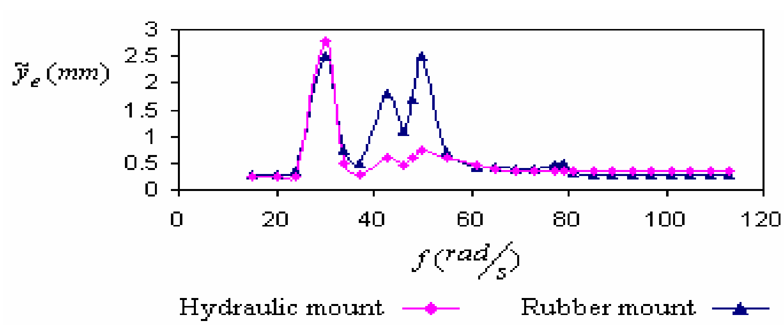

Figure 6. Variation of steady state amplitude of engine displacement in “y” direction versus frequency of road excitation.

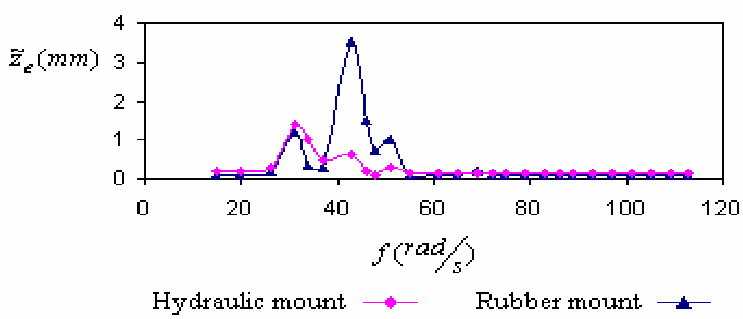

Figure 7. Variation of steady state amplitude of engine displacement in "z" direction versus frequency of road excitation.

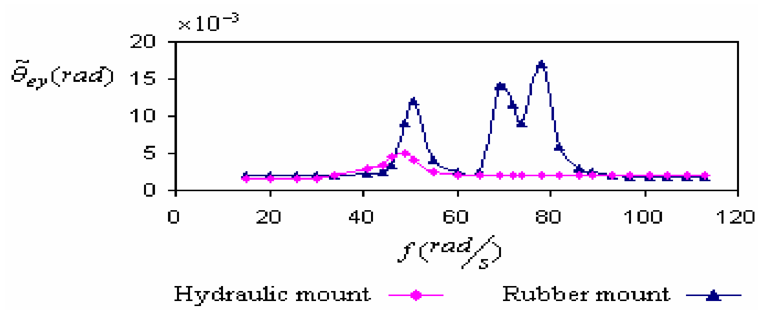

Figure 8. Variation of steady state amplitude of engine rotation about "y" axis versus frequency of road excitation.

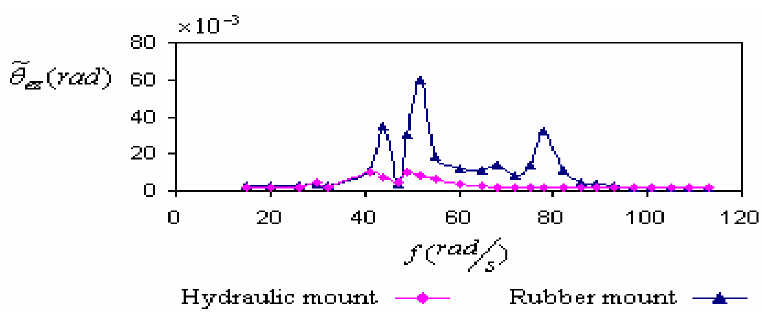

Figure 9. Variation of steady state amplitude of engine rotation about "z" axis versus frequency of road excitation.

Since hydraulic mounts can not produce enough damping for high frequency excitations, the displacements and rotations of engine on hydraulic mounts will be the same as engine on rubber mounts. On the other hand, in high frequency region, the transmitted force from hydraulic mounts to base is greater than corresponding from rubber mounts. Since engine movement in high frequency region for both mount types are identical, the forces due to stiffness and damping terms are the same, however, the term related to fluid pressure will cause increase in transmitting force.

5. CONCLUSIONS

This paper represents the simulation of vibrational behavior of engine on nonlinear hydraulic engine mounts. For this purpose, the nonlinear properties of hydraulic mounts are identified and a mathematical model of the system is developed. The governing equations of the motion are derived in time domain. For the comparison of rubber and hydraulic mounts, a thirteen degree of freedom full-vehicle model including six degree of freedom for translations and rotations of engine, three degree of freedom in bounce, pitch and roll for sprung mass and one degree of freedom in vertical displacement for each four unsprung masses are developed. The dynamic responses of the system under engine and road excitations are determined in frequency domain. The results show that more improved isolation effects can be obtained by using the hydraulic mounts in low frequency region. The assembled model would be of some practical use to engine-mount designers. In addition, the presented assembled model would be very helpful for the optimization of engine mounting systems.

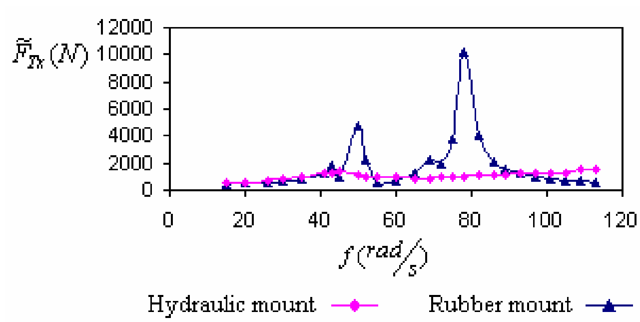

Figure 10. Variation of steady state amplitude of transmitted force to the body by engine mounts in "x" direction versus frequency of road excitation.

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