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# A Computer Simulation Method for Dynamic Analysis of Hydraulic Engine Mount System

Sang Beom Lee*, Dong Woon Choi**, Hong Jae Yim*

*Graduate School of Automotive Engineering, Kookmin University
**Inno Simulation

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**Abstract**

In this paper, a computer simulation method is presented for the dynamic analysis of a hydraulic engine mount system. The hydraulic engine mount system controls the damping characteristics using the viscosity of fluid flow. The complex stiffnesses of the main rubber for the hydraulic engine mount system are computed using a finite element analysis. The equations of motion considering the parameters of the hydraulic engine mount system are derived. To investigate the effects of the hydraulic engine mount system, the computer simulation running over a typical rough road is carried out using a vehicle dynamic model. These results are compared with those of the conventional rubber mount system.

***Keywords:*** *Complex stiffness, Damping coefficient, Decoupler, Dynamic analysis, Hydraulic engine mount, Inertia track, Vibration analysis*

## I. Introduction

One of the important problems encountered in the automotive design is the reduction of engine vibrations and ultimately the dynamic forces transmitted from the engine to the vehicle body structure. In general, rubber mounts or hydraulic mounts have been used to provide static support for the engine and to isolate the motion of the engine from the vehicle body structure. In general, to isolate a vehicle body structure from a vibrating member, low stiffness is required for the mount. However, in order to limit motion of the engine and to increase the life of the mount, high stiffness is required for the mount. Low stiffness of the mount is required when small input amplitudes are transmitted through the mount, and high stiffness and high damping is required when large input amplitudes are transmitted through the mount[1,2].

Corresponding author: Sang Beom Lee (sblee@kmu.kookmin.ac.kr)
Kookmin University, Seoul 136-702, Korea

Since most conventional rubber mounts have constant stiffness and damping coefficient without having interrelation with frequency and amplitude of the mount, it is difficult to meet the two conflicting characteristics described above. These contrasting objectives have motivated the development and application of hydraulic engine mount systems. The hydraulic engine mount systems can control the damping characteristics by using the fluid viscosity. Thus, for the hydraulic mount systems, it is possible to get better isolation effects than the conventional rubber mount systems.

In this paper, a computational method is presented for the dynamic analysis of the hydraulic engine mount system. The complex stiffnesses of the main rubber for the hydraulic engine mount system are computed byfinite element analysis for visco-elastic materials and hydrostatic elements. A numerical analysis method is presented to solve nonlinear equations of the hydraulic engine mount system, which consists of the engine mass, the fluid in the inertia track, and the vertical inertia force of the reciprocating mass in the engine. Also, dynamic properties of the hydraulic engine mount system are analyzed in the frequency domain. Effects of the hydraulic engine mount system running over a rough road profile are investigated using a vehicle dynamic model and the results are compared with those of the conventional rubber mount system.

## II. Hydraulic Engine Mount System

Hydraulic engine mount systems are complicated devices consisting of several components. Generally, a hydraulic engine mount system consists of a main rubber, an upper chamber, a lower chamber, a decoupler, an inertia track, and bellows. Figure 1 shows the configuration of a typical hydraulic engine mount system.

The main rubber is usually of conical shape and used to provide static support for the engine weight. The main rubber also forms the upper chamber of the hydraulic engine mount. A hydraulic pressure is generated due to the dynamic deflection of the main rubber. The bellows is a thin rubber membrane that forms the lower chamber of the hydraulic engine mount. The low stiffness of the bellows allows the lower chamber to act as an accumulator for the fluid transferred from the upper chamber. The decoupler is a flexible membrane and serves to minimize the amplitude of the fluid in the inertia track under small displacements. The inertia track is a long pipe joining the two chambers through the plate. The inertia track forms the major damping element of the hydraulic engine mount system. The length and cross sectional area dimensions of the inertia track confine a volume of fluid[3].

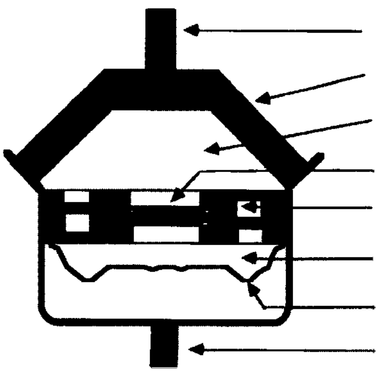


Figure 1. Typical geometry of a hydraulic engine mount.

The hydraulic engine mount system is worked by the fluid flow between the upper chamber and lower chamber as well as the stiffnesses of the main rubber element. The main rubber has the three directional stiffnesses: the vertical stiffness, the bulge stiffness, and the horizontal stiffness. To obtain the damping and the stiffness, which is changeable according to the input amplitude, the hydraulic engine mount system has two-flow path, i.e. the decoupler and the inertia track. For the higher frequencies with smaller input amplitude, the fluid flows mainly through the decoupler, and the transmissibility is improved. For the higher input amplitude like an engine bounce, the fluid flows into the decoupler and the inertia track. According to the flow rate increase, the larger damping of the hydraulic engine mount system can be obtained.

## III. Fe Analysis of Main Rubber

The main rubber of the hydraulic engine mount system consists of three complex stiffnesses. Equation (1) shows the complex vertical stiffness of the main rubber without the fluid in the mount. Equation (2) shows the complex bulge stiffness of the main rubber and results from the hydraulic pressure generated in the upper chamber due to the pumping of the fluid through the inertia track. Equation (3) shows the complex horizontal stiffness of the main rubber.

$$K_m^* = K_m(1 + \eta_m i) \tag{1}$$

$$K_b^* = K_b(1 + \eta_b i) \tag{2}$$

$$K_h^* = K_h(1 + \eta_h i) \tag{3}$$

where $\eta_m$, $\eta_b$, and $\eta_h$ are the loss factors.

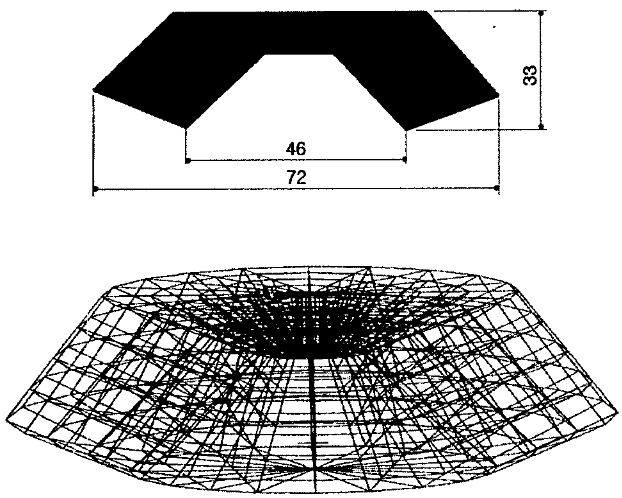


Figure 2. FE model of the main rubber.

Table 1. Physical properties of the rubber.

| Property | Value |
|---|---|
| Shear modulus | 9.8E5 (N/m²) |
| Poisson's ratio | 0.5 |
| Density | 920 (kg/m³) |
| Loss factor | 0.08 |

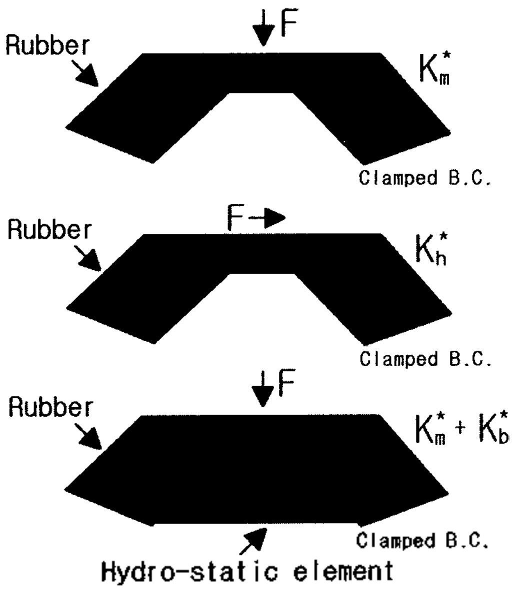


Figure 3. Constraint and load conditions of the FE model.

Table 2. Complex stiffness of the main rubber.

| Type | Stiffness (N/mm) | Loss factor |
|---|---|---|
| Vertical complex stiffness | 393 | 0.077 |
| Horizontal complex stiffness | 116 | 0.050 |
| Bulge complex stiffness | 110 | 0.077 |

In order to obtain the complex stiffnesses numerically, a finite element model of the main rubber was modeled using ABAQUS. Figure 2 and Table 1 shows the finite element model and the physical properties of the main rubber, respectively.

The vertical stiffness and the horizontal stiffness respectively can be determined by using the displacement and the phase difference, which are generated when the dynamic load of the sinusoidal wave is acted to the main rubber in each direction. And to determine the bulge stiffness, the fluid is assumed as incompressible fluid and modeled using the hydrostatic elements. Figure 3 shows the load conditions and the boundary conditions to obtain the three stiffnesses.

Table 2 shows the complex stiffnesses and the loss factors of the main rubber computed using finite element analysis.

## IV. Dynamic Analysis of Hydraulic Engine Mount System

The typical model of hydraulic engine mount system is shown in Figure 4. And the parameter values used in this study are listed in Table 3[4].

As shown in Figure 4, the main rubber can be described as vertical complex stiffness and bulge complex stiffness, and the inertia track can be described as mass and equivalent damping. The stiffness of the bellows of the lower chamber has much lower stiffness than the main rubber.

In Figure 4, $A_c$ and $x_c$ are the area and the displacement of the upper chamber, respectively. $A_t$ and $x_t$ are the area and the displacement of the inertia track; and $M_{en}$ is the engine mass; and $F_{en}$ is the vertical inertia force due to the engine excitation. $m$ and $c$ are the mass and the damping coefficient of the fluid in the inertia track.

The flow rate through the inertia track can be written as Equation (4) by using the equation of continuity. And, the equation of equilibrium in the inertia track can be written as Equation (5).

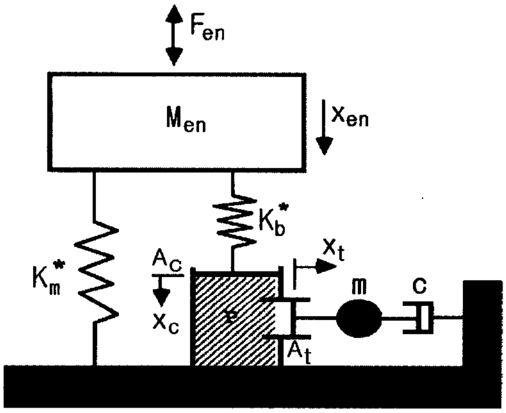


Figure 4. Mathematical model of the hydraulic engine mount.

Table 3. Parameter values of the hydraulic engine mount.

| Parameter | Value |
|---|---|
| Area of equivalent upper chamber, $A_c$ | 1250 (mm²) |
| Area of inertia track, $A_t$ | 9 (mm²) |
| Length of inertia track, $L_t$ | 125 (mm) |
| Hydraulic diameter of inertia track, $D_h$ | 3 (mm) |
| Minor loss, $K$ | 1.5 |
| Density of fluid, $\rho$ | 1040 (kg/m³) |
| Kinematic viscosity, $\nu$ | 1.8E-6 (m²/s) |
| Vertical stiffness of main rubber, $K_r$ | 393 (N/mm) |
| Bulge stiffness of main rubber, $K_b$ | 110 (N/mm) |
| Loss factor of main rubber, $\eta$ | 0.077 |
| Equivalent piston mass, $M_p$ | 1.67 (kg) |
| Length of crank, $R$ | 0.0445 (m) |
| Length of connecting rod, $L$ | 0.155 (m) |
| Mass of engine, $M_{en}$ | 75 (kg) |

$$A_c\dot{x}_c = A_t\dot{x}_t = A_t u \qquad (4)$$

$$\rho L_t A_t \ddot{x}_t = pA_t - p_0 A_t - \left(f\frac{L_t}{D_h} + K\right)\frac{\rho}{2}|u|uA_t \qquad (5)$$

where $u$ is a flow velocity, and $f$ is a friction coefficient of the pipe.

The equation of equilibrium according to the bulge stiffness can be written as Equation (6).

$$K_b^*(x_c - x_{en}) + pA_c = 0 \qquad (6)$$

By considering Equations (5) and (6), the equation of motion can be written as Equation (7).

$$M_e\ddot{x}_c + C_e\dot{x}_c - K_b^* x_{en} + K_b^* x_c = 0 \qquad (7)$$

where $C_e$ and $M_e$ is the equivalent damping coefficient and the equivalent mass of the inertia track respectively, and can be written as follows.

$$C_e = \frac{4}{3\pi}\left(f\frac{L_t}{D_h} + K\right)\rho A_t\left(\frac{A_c}{A_t}\right)^3 X_c\omega$$

$$M_e = \rho L_t\frac{A_c^2}{A_t}$$

The equilibrium equation of the hydraulic engine mount system including the vertical inertia force due to the engine excitation can be written as Equation (8).

$$M_{en}\ddot{x}_{en} + (K_m^* + K_b^*)x_{en} - K_b^* x_c = F_{en} \qquad (8)$$

By combining Equations (7) and (8), the equation of hydraulic engine mount system can be written as Equation (9).

$$\begin{bmatrix} M_{en} & 0 \\ 0 & M_e \end{bmatrix}\begin{Bmatrix} \ddot{x}_{en} \\ \ddot{x}_c \end{Bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & C_e \end{bmatrix}\begin{Bmatrix} \dot{x}_{en} \\ \dot{x}_c \end{Bmatrix} + \begin{bmatrix} K_m^* + K_b^* & -K_b^* \\ -K_b^* & K_b^* \end{bmatrix}\begin{Bmatrix} x_{en} \\ x_c \end{Bmatrix} = \begin{Bmatrix} F_{en} \\ 0 \end{Bmatrix} \qquad (9)$$

$$F_{en} = 4M_p\omega^2\frac{R^2}{L}\cos 2\omega t$$

where $M_p$ is the reciprocating mass per cylinder of the engine, $\omega$ is the angular velocity of the crankshaft, $R$ is the radius of the crank, and $L$ is the length of the connecting rod[5].

The dynamic responses such as the displacement, the transmitted force, etc. of the hydraulic engine mount system can be simulated by solving the Equation (9). To solve the Equation (9), the Newton-Raphson method is used.

Figures (5) and (6) show the comparisons of the engine displacements and the transmitted forces for the hydraulic mount and the rubber mount, respectively. As shown in these figures, for the hydraulic engine mount, the displacement of the engine and the transmitted force are decreased near the 300 rpm.

Figures (7) and (8) show the variation of the dynamic stiffness and the dynamic loss factor due to the variation of the input displacements for the hydraulic engine mount system.

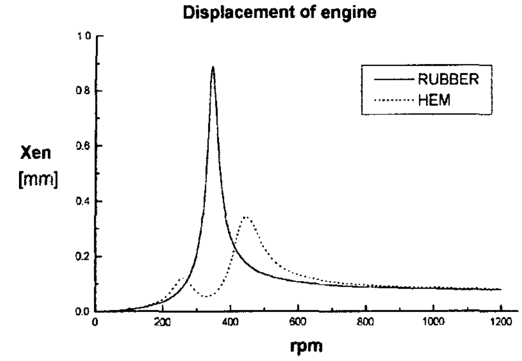


Figure 5. Displacement of the engine.

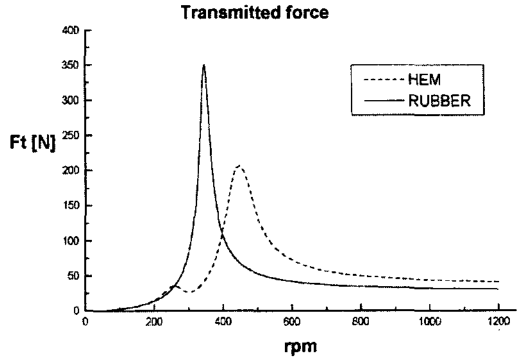


Figure 6. Transmitted force.

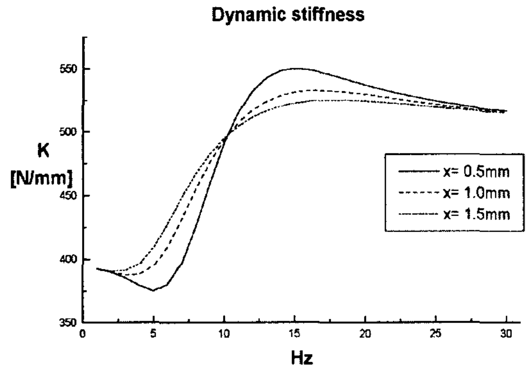


Figure 7. Dynamic stiffness of the hydraulic engine mount.

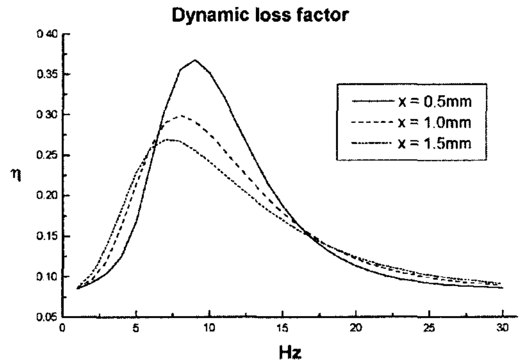


Figure 8. Dynamic loss factor of the hydraulic engine mount.

In these figures, we can see that the damping significantly affects the dynamic stiffness and the dynamic loss factor between 5 and 15 Hz.

## V. Computer Simulation of Vehicle

In general, the engine mount is excited by the engine vibration and the input amplitude of the rough road.

In this study, a multi-body dynamic simulation model of full vehicle system is developed for the dynamic analysis of the vehicle body and the engine using DADS, which is a general-purpose multi-body dynamic analysis code. The vehicle dynamic model is composed of 32 rigid bodies that are constrained by kinematics joints. This vehicle dynamic model, which has the hydraulic mount and the rubber mount respectively, run over a typical rough road profile of 5 cm at the constant speed of 18 mph.

In this study, in the inertia track of the hydraulic mount, the damping coefficient is set to 0.24 Ns/m by considering linear damping[6].

Equation (11) shows the differential equation for the forces transmitted to the engine and the vehicle body due to the displacements of the engine and the vehicle body. For the rubber mount, the forces transmitted to the engine and the vehicle body is computed by multiplying the vertical stiffness to the relative displacements of the engine and the vehicle body.

$$\begin{Bmatrix} \dot{x}_t \\ \ddot{x}_t \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\left(\dfrac{A_t}{A_c}\right)^2 \dfrac{K_b}{m} & -\dfrac{c}{m} \end{bmatrix} \begin{Bmatrix} x_t \\ \dot{x}_t \end{Bmatrix} + \begin{bmatrix} 0 & 0 \\ \dfrac{A_t}{A_c}\dfrac{K_b}{m} & -\dfrac{A_t}{A_c}\dfrac{K_b}{m} \end{bmatrix} \begin{Bmatrix} x_{en} \\ x_{body} \end{Bmatrix} \tag{10}$$

$$\begin{Bmatrix} f_{en} \\ f_{body} \end{Bmatrix} = \begin{bmatrix} \dfrac{A_t}{A_c}K_b & 0 \\ -\dfrac{A_t}{A_c}K_b & 0 \end{bmatrix} \begin{Bmatrix} x_t \\ \dot{x}_t \end{Bmatrix} + \begin{bmatrix} -(K_m+K_b) & K_m+K_b \\ K_m+K_b & -(K_m+K_b) \end{bmatrix} \begin{Bmatrix} x_{en} \\ x_{body} \end{Bmatrix} \tag{11}$$

Table 4 shows the specification of the engine used in

Table 4. Specification of the engine.

| | |
|---|---|
| [illegible] | 1.34 (kg) |
| [illegible] | 1.13 (kg) |
| [illegible] | 0.00349 (kg·m²) |
| [illegible] | 0.0445 (m) |
| [illegible] | 0.155 (m) |
| [illegible] | 0.0563 (m) |
| [illegible] | 0.045 (m) |
| [illegible] | 0.11 (m) |
| [illegible] | |
| [illegible] | 225 (kg) |
| [illegible] | 8.51 (kg·m²) |
| [illegible] | 24.18 (kg·m²) |
| [illegible] | 19.09 (kg·m²) |
| [illegible] | 0.54 (kg·m²) |
| [illegible] | 5.39 (kg·m²) |
| [illegible] | 0.27 (kg·m²) |

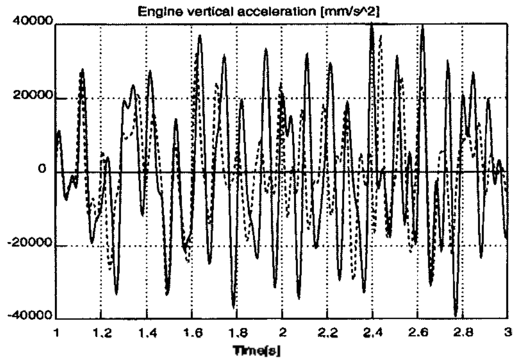


Figure 9. Vertical acceleration of the engine.

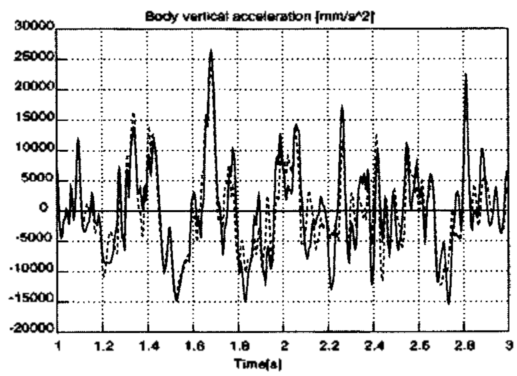


Figure 10. Vertical acceleration of the body.

this study [4]. Figures 9 and 10 show the computer simulation results obtained from the dynamic analysis. Figure 9 shows the vertical acceleration of the engine, in which two simulation results are compared. In the figure, the solid line denotes the rubber mount and the dotted line denotes the hydraulic mount. Figure 10 shows the comparison of the vertical accelerations for the vehicle body. As shown in these figures, we can see that smaller magnitudes of accelerations are obtained from the vehicle with the hydraulic mount than from the vehicle with the rubber mount. These results show that more improved isolation effects can be obtained by using the hydraulic mount.

## VI. Conclusions

In this paper, a computational method is presented for the dynamic analysis of hydraulic engine mount system. The complex stiffness of the main rubber of the hydraulic engine mount system is computed by finite element analysis for the visco-elastic materials and hydrostatic elements. The numerical analysis method is presented to solve nonlinear equations of the hydraulic engine mount system, with the engine mass, the fluid in inertia track and the vertical inertia force of reciprocating mass in the engine. Also, dynamic properties of the hydraulic engine mount system are analyzed in the frequency domain. Effects of the hydraulic engine mount system running over the rough road are investigated using the vehicle dynamic model. These results are compared with those of the rubber mount system. These results obtained from the computer simulation show that more improved isolation effects can be obtained by using the hydraulic mount.

## Acknowledgements

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## [Profile]

• Sang Beom Lee

Professor, Graduate School of Automotive Engineering, Kookmin University
The Journal of the Acoustical Society of Korea, Vol. 17, No. 1E.

• Dong Woon Choi

He received the B.S. and M.S. degree in Mechanics and Design from Kookmin University, Seoul, Korea, in 1995 and 1997, respectively. Since September 2001, he has been an intern researcher at the Inno Simulation Co., Seoul, Korea. His major research area is vehicle structural analysis.

• Hong Jae Yim

Professor, Graduate School of Automotive Engineering, Kookmin University
The Journal of the Acoustical Society of Korea, Vol. 17, No. 1E.