Control of Engine Roll and Bounce Vibrations Using Hydraulic Mounts*

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A method for designing an engine mounting system using hydraulic insulators has been developed with the aim of controlling low-frequency engine vibrations in the roll and bounce modes. This method includes a design optimization theory for each hydraulic insulator and a procedure for determining the equivalent mass for various modes at the mount locations, and is applicable to integrated system optimization. Computer simulation results show that resonance peaks can be effectively reduced with a hydraulic engine mount system designed on the basis of this method.

Key Words: Vibration Control, Hydraulic Engine Mount, Optimum Design, Vibration Mode, Equivalent Mass

1. Introduction

Improving the ride comfort of motor vehicles has long been a focus of research, and a great deal of work has been done to date toward the reduction of vehicle vibrations. Research efforts in this area have become even more vigorous in recent years, especially with regard to semi-active and full active suspensions. These suspension systems reduce road surface input and attenuate very-low-frequency vibrations to achieve improvements in both handling and ride comfort.

Moreover, new types of engine mounts, including hydraulic insulators, are being developed to reduce

vehicle shaking caused by vibration input at around 10 Hz from the suspension and to isolate the transmission of engine vibrations to the body during idling. However, since design optimization methods have yet to be established for hydraulic mounts, they are generally designed and manufactured through a process of trial and error. As a result, the outstanding characteristics inherent in hydraulic mounts are not always obtained.

In a previous work⁽¹⁾, we presented a design optimization method and a set of equations derived for optimizing the design of hydraulic mounts. The method was developed on the basis of a theoretical analysis conducted with a one-mass system model consisting of an equivalent engine mass and a hydraulic mount. The equations include formulations for determining the optimum mass ratio and optimum damping ratio. If the spring ratio of the mount, i.e., the ratio of the support spring to the expansion spring of the fluid chamber, is known, the optimum geometry can be determined. The validity of these equations for use in designing hydraulic mounts has been verified both theoretically and experimentally. Low-frequency engine vibrations, however, involve a vibration system with two degrees of freedom, namely, roll and

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bounce.

Taking the nature of this problem into account, we have devised a design optimization method for a mounting system that uses two hydraulic mounts to control engine vibrations in the roll and bounce modes. The two vibration modes involved were first analyzed and an equivalent mass was determined for each mount location. Then, using the above-mentioned design optimization equations of the one-mass system model, each mount was designed. The mount designs were then analyzed using the transfer matrix method, and the results showed that vibrations in the roll and bounce directions were effectively reduced. This confirmed the validity of the method used for designing actual engine mounts. In addition, useful insights were gained concerning the application of hydraulic mounts for the purpose of controlling vibrations in the roll and bounce modes.

2. Engine Vibration Characteristics

2.1 Two-degree-of-freedom model for vibration control

Vehicle ride comfort is greatly influenced by two resonances, referred to as roll and bounce vibrations, that typically occur at frequencies of around 10 Hz. The roll mode occurs around the principal axis of inertia, as shown in Fig. 1. A two-degree-of-freedom model of this engine vibration system is shown schematically in Fig. 2. As seen in the figure, two rubber mounts are used to support the engine in the vertical direction. In this model, M and J represent the engine mass and the moment of inertia around the engine's center of gravity, K_{E1} and K_{E2} are the spring constants of mount 1 and mount 2, and L_1 and L_2 are the distances from the center of gravity to each mount respectively. It is assumed that the force generated by the engine is an external force that acts on the center of gravity.

Table 1 shows the values that were calculated for these mount parameters and the engine used in this

Fig. 1 Engine's center of gravity and mount locations

study. In the following discussion, these values will be applied to the engine vibration system in Fig. 2, which represents the vibration characteristics to be controlled.

2.2 Vibration characteristics to be controlled

The equation of motion for the engine vibration system in Fig. 2 can be given as

$$\tilde{M}\ddot{X} + \tilde{C}\dot{X} + \tilde{K}X = f, \quad (1)$$

where

$$\tilde{M} = \begin{bmatrix} M & 0 \\ 0 & J \end{bmatrix}$$

$$\tilde{C} = \begin{bmatrix} C_{E1} + C_{E2} & -C_{E1}L_1 + C_{E2}L_2 \\ -C_{E1}L_1 + C_{E2}L_2 & C_{E1}L_1^2 + C_{E2}L_2^2 \end{bmatrix}$$

$$\tilde{K} = \begin{bmatrix} K_{E1} + K_{E2} & -K_{E1}L_1 + K_{E2}L_2 \\ -K_{E1}L_1 + K_{E2}L_2 & K_{E1}L_1^2 + K_{E2}L_2^2 \end{bmatrix}$$

$$X = \{x, \theta\}^T, f = \{F, 0\}^T.$$

Using the values in Table 1, we can solve Eq. (1) as an eigenvalue problem and calculate the eigenvalues and eigenvectors. The eigenfrequencies and vibration mode shape can then be found and represented schematically, as shown in Fig. 3. As a result, the vibration characteristics that are to be controlled can be summed up as follows.

(1) The first-order eigenfrequency is 6.72 Hz. This represents a roll vibration mode having a node at

Table 1 Rubber mount specifications

Mass and Inertia	$M=80\text{ kg}, J=6.7\text{ kgm}^2$
Stiffness	$KE1=KE2= 18 \times 10^4 \text{ N/m}$
Damping	$CE1=CE2= 1.8 \times 10^4 \text{ Ns/m}$
Distance	$L1=0.1\text{ m}, L2=0.3\text{ m}$

Fig. 2 Two-degree-of-freedom model of engine vibration system

Fig. 3 Vibration modes

a point 168 mm from the center of gravity in the direction of mount 2.

(2) The second-order eigenfrequency is 11.58 Hz. This is a bounce vibration mode having a node at a distance 486 mm from the center of gravity in the direction of mount 1.

To obtain the best vibration damping effect from an insulator, it should be placed at the antinode of the vibration mode. Therefore, based on the foregoing analytical results, mount 1 should serve to suppress the roll mode and mount 2 the bounce mode.

3. Analysis by Transfer Matrix Method

This section presents an analysis performed by means of the transfer matrix method, which is a useful tool for equivalent mass identification and for computing the frequency response when hydraulic mounts are installed.

3.1 Transfer matrix representation

The engine vibration system model in Fig. 2 was first represented in the form of the transfer matrix shown in Fig. 4 for the purpose of performing a frequency response analysis and equivalent mass identification. Here, B_1 and B_5 are the transfer matrices for mount 1 and mount 2; B_2 and B_4 are rigid massless levers representing the distances L_1 and L_2 from each mount to the engine's center of gravity respectively; and B_3 is a transfer matrix of the engine at its center of gravity. These transfer matrices and state vectors can be expressed as

$$B_i = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ K_{Ei} + jC_{Ei} & 0 & 0 & 1 & f \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}, (i=1, 5) \quad (2)$$

$$B_i = \begin{bmatrix} 1 & -L_s & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & -L_s & 0 \\ 0 & 0 & 0 & 1 & f \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \begin{matrix} (s=1, 2) \\ (i=2, 4) \end{matrix} \quad (3)$$

Fig. 4 Vibration model used in transfer matrix analysis (with rubber mount support)

$$B_3 = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & -J\omega^2 & 1 & 0 & 0 \\ -M\omega^2 & 0 & 0 & 1 & f \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \quad (4)$$

where f in each matrix is the external force term. When there is no external force acting on the model, $f=0$.

The state vector at the i -th point is defined as

$$Z_i = \{Y, \phi, T_b, F, 1\}^T, \quad (5)$$

where Y , ϕ , T_b and F are the lateral deflection, slope, bending moment and shear force, respectively.

Using these transfer matrices, the frequency response at the center of gravity and at the locations of mount 1 and mount 2 was analyzed under the assumption that external force was acting on the center of gravity. The results obtained are shown in Fig. 5. The frequency responses show good agreement with the calculated results in Fig. 3.

3.2 Equivalent mass identification

Figure 6 shows the design optimization equations for the hydraulic mounts based on a one-mass system model of the engine's vertical motion. If we know the ratio between the main spring of the mount and the expansion spring, $N=K_m/K_e$, it is possible to determine the optimum mass ratio, μ_{opt} , and optimum damping ratio, ζ_{opt} . Furthermore, if the engine mass,

Fig. 5 Vibration frequency response at mount locations

Fig. 6 Equivalent dynamic model of hydraulic mount

M , is already known, the optimum mount design can be obtained. However, in the engine vibration model in Fig. 2, the equivalent masses at the engine mount locations are still undetermined, making it impossible to use the design optimization equations.

To solve this problem, we applied an equivalent mass identification method using multiple degrees of freedom. With this method⁽²⁾, an added mass was attached at the location of each mount. The value of the added mass was varied several times within a range of 1 to 10 kg in order to minimize the effect on the vibration modes and assure good calculation accuracy. Using the equation noted below, the equivalent mass was found for each mode in relation to the given values. The equivalent masses thus found were then subjected to a linear regression analysis to determine the equivalent mass values corresponding to a condition without any added masses. In the equation below, M_{ji} is the equivalent mass at the j -th point of the i -th mode and can be calculated with this expression if we know the value of the added mass, Δm_{ji} , the eigenfrequency, Ω_i , of the i -th mode of the original vibration system and the eigenfrequency, ω_{ji} , of the i -th mode of the vibration system incorporating the added masses.

$$M_{ji} = \Delta m_{ji} \omega_{ji}^2 / (\Omega_i^2 - \omega_{ji}^2)$$

The specific procedure used here was to express the rigid lever with a mass attached only at the location of the mount (Fig. 7) as transfer matrix B_2 and to calculate the change in the eigenfrequency. The equivalent masses (expressed in round figures) found with this method are noted below.

$$M_1 = 120 \text{ kg} \quad M_2 = 40 \text{ kg}$$

$$B_2 = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & -L_1 & 0 \\ -\Delta m_1 \omega^2 & 0 & 0 & 1 & f \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}$$

3.3 Transfer matrix of hydraulic mount

Using the symbols and coordinates of the hydraulic mount in Fig. 6, the transfer matrix between points 1 and 2 in Fig. 4 can be expressed as

$$\begin{bmatrix} Y \\ \phi \\ T \\ F \\ 1 \end{bmatrix}_1 = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ Z & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} Y \\ \phi \\ T \\ F \\ 1 \end{bmatrix}_2, \quad (6)$$

where

$$Z = \frac{[K_m K_e - r^2 m \omega^2 (K_m + K_e)] + j r^2 (K_m + K_e) C \omega}{(K_e - r^2 m \omega^2) + j r^2 C \omega}$$

$$r = A_u / A_o. \quad (7)$$

Since the damping effect of the rubber damper used with the mount support spring is smaller than that of the fluid, it has been omitted from the equation.

This transfer matrix was used in performing the frequency response calculations presented in the following section.

4. Hydraulic Engine Mount Design Taking into Account Engine Vibration Modes

Figure 8 shows a vibration simulation model incorporating two hydraulic mounts for the purpose of controlling engine vibrations in the roll and bounce modes. Because the equivalent mass at the location of each mount was found in the previous section, the following equations can be used to design the mounts.

$$\left(\frac{A_u}{A_o} \right)^2 m_i = M_i \frac{N_i (N_i + 2)}{2(N_i + 1)^2}, \quad (i=1, 2) \quad (8)$$

$$\left(\frac{A_u}{A_o} \right)^2 C_i = \frac{2\sqrt{M_i K_i} 3N_i^2 (N_i + 2)}{16(N_i + 1)^3}, \quad (i=1, 2) \quad (9)$$

Here, A_u and A_o are the sectional areas of the fluid chamber and the orifice of the mount, respectively. Setting $N_1 = N_2 = 6.6/18$ and $(A_u/A_o)_1 = (A_u/A_o)_2 = 21.0/0.42$, the mass and damping coefficient of the fluid flowing through the orifice of each mount can be found as

$$m_1 = 1.09 \times 10^{-2} \text{ kg} \quad C_1 = 0.337 \text{ Ns/m}$$

$$m_2 = 0.437 \times 10^{-2} \text{ kg} \quad C_2 = 0.21 \text{ Ns/m}$$

These values can be obtained through a suitable selection of the viscosity and coefficient of kinematic viscosity of the fluid used and the length of the orifice passage.

5. Validation of Mount Design by Simulation

Having determined the design values of the hydraulic mounts, the next step is to verify the effectiveness of the mounts in controlling engine vibrations. This is done by using the transfer matrix

Fig. 7 Model with added mass for equivalent mass identification in each mode

Fig. 8 Simulation model of engine supported by two hydraulic mounts

method and the simulation model in Fig. 8 to conduct a numerical analysis. Typical results are shown in Fig. 9.

In Fig. 5, the resonance peaks in each mode were seen to be approximately eight times higher than the baseline. The results in Fig. 9 indicate that both peaks were reduced to a level where they were only about 2.5 times higher than the baseline values. The simulation model used in performing these calculations had equivalent models of the hydraulic mounts in place of the springs shown in the model in Fig. 4.

It was noted earlier that the equivalent engine mass in each resonance mode plays an important role in the mount design. Simulations were conducted to verify the importance of their role. Two engine mounts were designed with incorrect equivalent masses of $M_1=80$ kg and $M_2=80$ kg, instead of using the true values of $M_1=120$ kg and $M_2=40$ kg, i.e., the conditions in Table 1. The vibration damping effect with the mounts was then calculated. Typical results are shown in Fig. 10. The results in Fig. 9 clearly show more effective vibration damping, thereby substantiating the advantages of the design method presented here.

Fig. 9 Damping effect obtained with two hydraulic mounts tuned to separate modes

Fig. 10 Influence of equivalent mass estimation error

6. Design Approach for Combined Resonance Modes of Hydraulic Mounts

Using the simulation technique explained in the previous section, several practical and systematic design approaches that utilized the characteristics of hydraulic mounts to control resonances in both the roll and bounce modes were studied.

6.1 Relationship between roll-bounce vibration mode coupling strength and vibration reduction

The design value of the main spring of the mounts was varied in relation to the mount support state in Fig. 5 and the vibration reduction that was obtained when the eigenfrequencies were separated to 5.0 Hz and 13.5 Hz was calculated. The results obtained are shown in Fig. 11. Conversely, the two resonance modes were made to overlap so that the eigenfrequencies coincided at 10.5 Hz. The frequency response results for that state without hydraulic damping are shown in Fig. 12, and the optimized damping effect is shown in Fig. 13.

When the eigenfrequencies were separated, the resonance peaks were smaller and vibrations were reduced over the entire frequency range examined

Fig. 11 Damping effect obtained when resonance frequencies are separated

Fig. 12 Response for overlapping resonance frequencies (with rubber mount)

here. The reasons for this difference in damping effect can be understood as follows. When the vibration modes overlap, in addition to the concentration of energy at one peak, vibration occurs in the simple bounce mode. This results in a situation where the engine is most susceptible to excitation caused by vertical force input. A second reason is that a stronger damping effect cannot be produced by positioning the mount at the antinode to increase the fluid velocity. These results suggest that the eigenfrequencies should be separated when two hydraulic mounts are applied to reduce engine vibrations in the roll and bounce modes.

6.2 Relationship between vibration modes and vibration reduction obtained with only one hydraulic mount

Hydraulic mounts are generally more expensive than conventional rubber mounts. It would be ideal if the desired damping effect could be achieved using only one hydraulic mount. To examine this possibility, simulations were conducted to investigate the design concept for a mounting system with only one hydraulic mount. This investigation focused on the connection between the mount location and the vibra-

tion modes.

Figure 14 shows the simulation results obtained when a hydraulic mount was used only at the mount 1 location with the aim of controlling vibration modes on the high-frequency side. While vibrations were effectively reduced at frequencies above 10 Hz, peaks were still evident between 6–7 Hz, albeit at lower levels. Figure 15 shows the results obtained when a hydraulic mount was used on the opposite side at the mount 2 location. In the latter case, the peaks between 10–11 Hz were not appreciably reduced. The difference in the results between Figs. 14 and 15 is attributed to the flow characteristics of the fluid. In the case of Fig. 14, the fluid flowed through the orifice, producing a damping effect on the high-frequency side. Such an effect was not produced in the case of Fig. 15 because the fluid did not flow at frequencies higher than its resonance frequency. As a result, the damping effect was greatly reduced.

The results in Fig. 16 indicate that a relatively good damping effect can be obtained even with only one mount when the engine vibration modes overlap.

Fig. 13 Response for overlapping resonance frequencies (with 2 hydraulic mounts)

Fig. 15 Response obtained with one hydraulic mount tuned to the mode on the low-frequency side

Fig. 14 Response obtained with one hydraulic mount tuned to the mode on the high-frequency side

Fig. 16 Response obtained for overlapping resonance frequencies (with one hydraulic mount)

7. Conclusions

Equations for designing hydraulic engine mounts were derived in relation to a one-mass system. Computer simulation results confirmed the validity of the equations for designing two-degree-of-freedom engine mounting systems that effectively control vibrations. The following insights have been gained concerning the use of hydraulic mounts to reduce engine vibrations in the roll and bounce modes.

(1) When two hydraulic mounts are applied to reduce vibrations, the eigenfrequencies of the modes to be controlled should be separated as much as possible.

(2) When one hydraulic mount is used, a relatively good damping effect can be obtained by making the roll and bounce modes overlap.

(3) When one hydraulic mount is used and the eigenfrequencies of the roll and bounce modes are separated, a good damping effect can be obtained by tuning the mount to the mode on the high frequency side.

Appendix

When a hydraulic mount is used in the model shown in Fig. 6 between points 1 and 2 in Fig. 4, its stiffness, Z , force, F , and displacement, Y , can be expressed by the relationship

$$F = Z \cdot Y. \quad (\text{A} \cdot 1)$$

Z represents a parallel combination of the stiffness, Z_1 of the main spring in Fig. 6 and the stiffness, Z_2 of the other portions. Hence,

$$Z = Z_1 + Z_2 \quad (\text{A} \cdot 2)$$

$$Z_1 = K_m. \quad (\text{A} \cdot 3)$$

Z_2 represents a serial combination of the expansion spring of the fluid chamber wall, K_e and the lever portion, Z_3 .

Hence,

$$1/Z_2 = 1/K_e + 1/Z_3. \quad (\text{A} \cdot 4)$$

Z_3 represents a parallel combination of the fluid mass in the orifice, m , and damper C . Since it is attached to the end of the lever (ratio of r),

$$Z_3 = r^2(-m\omega^2 + j\omega C). \quad (\text{A} \cdot 5)$$

Substituting this equation into Eq. (A·2) yields Eq. 6 in the text.

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