

Development of Simulation Model for Active Controlled Engine Mount

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ABSTRACT

This paper presents the development of an active control engine mount (ACM) model required to study passive characteristics and control inputs of an ACM, which absorbs increased 1.5 order vibrations caused by three-cylinder operation in a V6 engine. By integrating a dynamic model of the moving parts of an actuator into a conventional hydraulic engine mount model, resonance due to two fluid orifices are expressed. As a result, the ACM model capable of reproducing passive, active and unit characteristics with high accuracy is established. In addition, this actuator force value is obtained with this model to cancel the sub-frame transfer load generated by engine vibrations.

1. Introduction

To improve fuel economy, the engine employed in the 2003 model INSPIRE has been provided with cylinder-on-demand type enabling combustion to be stopped in one bank of cylinders when the vehicle is cruising or decelerating. However, while it has improved fuel economy, it has also increased the 1.5 order component of engine vibrations over approximately 10 dB. Front and rear active control engine mounts (ACM)⁽¹⁾ have been employed in the new INSPIRE to absorb the increased vibrations. These ACM apply feed-forward control by detecting variations in the engine crank angle and determining the level of control input on that basis. Because the basic specifications of the feed-forward control system, such as gain and control waveform, were designed using a model that expressed physical phenomena in a simplified form, it was essential to achieve accuracy in the model. At present, the heuristic method is most commonly employed for this purpose, but this method posed issues in terms of reproducing the high-order components of the ACM driving frequency as found in an actual vehicle. A review of the literature concerning ACM shows that almost no research

has been conducted on ACM models^{(2) - (5)}. The aim of the research discussed in this paper was to construct an ACM model capable of reproducing the high-order components of the driving frequencies by integrating a dynamic model of a structure with two fluid orifices, and to determine the actuator force required to minimize transfer load by using the ACM model.

2. Constructing a Calculation Model

Figure 1 shows a cutaway view of the ACM. To increase the ride comfort performance, an electromagnetic solenoid actuator has been added to the conventional hydraulic mount, which employs fluid orifices. It has been possible to locate the actuator in the lower section of the ACM by positioning the main fluid chamber in the center and the secondary fluid chamber in the upper section. The actuator controls the displacement of the engine mounting points by means of the fluid in the main fluid chamber. It operates to maintain the displacement of the mounting points in phase as engine vibrations in order to reduce the subframe transfer load.

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Figure 2 shows the dynamic model proposed in this research. The former ACM model was a simple model in which an actuator force component was added to a hydraulic mount model⁽⁶⁾. The newly developed model employs the former hydraulic mount model, in which the engine was expressed as a simple mass with the main rubber and a fluid orifice positioned below, as a base. New terms for the mass of the moving parts of the actuator, dynamic spring constant, damping coefficient and force have been added to the lower section of the former model. In addition, the mass for the moving sections of the actuator includes the fluid mass passing through the filter orifice located in the upper section.

Equations (1a), (1b), (1c), (1d) and (2) are derived from this model. (1a) is the dynamic equation for the inertial mass of the engine, (1b) is the dynamic equation for the fluid mass passing through the orifice established to improve the ride comfort performance, (1c) is the dynamic equation for the inertial mass of the actuator, and (1d) shows the constrained conditions derived from the law of conservation of mass of a fluid under fixed pressure conditions in the main liquid cabin. Equation (2) is a coefficient substituted to simplify the equations. Subframe transfer load is expressed by Eq. (3).

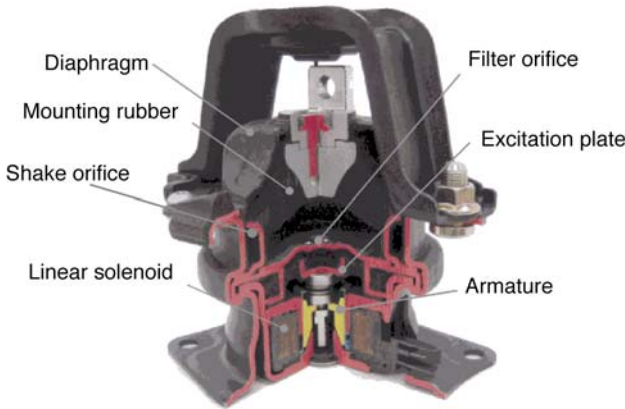


Fig. 1 Cutaway view of ACM

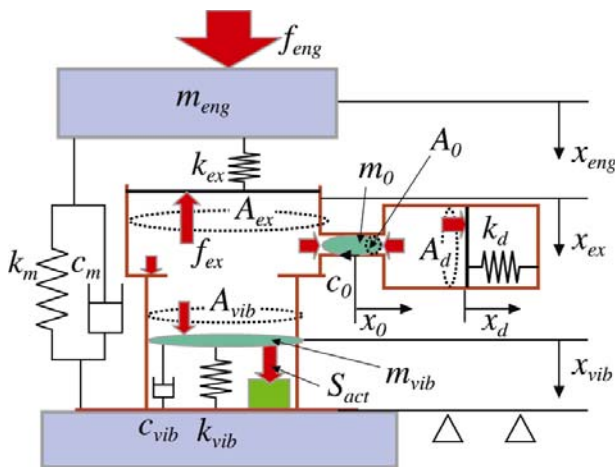


Fig. 2 ACM model

$$\begin{cases} \ddot{x}_{eng} + 2\xi_{eng}\omega_{eng}\dot{x}_{eng} + \omega_{eng}^2 x_{eng} = \frac{1}{m_{eng}} \{f_{eng} - f_{ex}\} & (1a) \\ \ddot{x}_0 + 2\xi_0\omega_0\dot{x}_0 + \omega_0^2 x_0 = \frac{1}{m_0} \left(\frac{A_0}{A_{ex}}\right) f_{ex} & (1b) \\ \ddot{x}_{vib} + 2\xi_{vib}\omega_{vib}\dot{x}_{vib} + \omega_{vib}^2 x_{vib} = \frac{1}{m_{vib}} \left\{s_{act} + \left(\frac{A_{vib}}{A_{ex}}\right) f_{ex}\right\} & (1c) \\ f_{ex} = k_{ex} \left\{ -\left(\frac{A_{ex}}{A_{vib}}\right) x_0 - \left(\frac{A_{vib}}{A_{ex}}\right) x_{vib} + x_{eng} \right\} & (1d) \end{cases}$$

However,

$$\begin{cases} \omega_{eng} = \sqrt{\frac{k_{ex}}{m_{eng}}}, \quad \xi_{eng} = \frac{c_m}{2\sqrt{m_{eng}k_{ex}}} \\ \omega_0 = \left(\frac{A_0}{A_{ex}}\right) \sqrt{\frac{k_{ex}}{m_0}}, \quad \xi_0 = \frac{c_0}{2\left(\frac{A_0}{A_{ex}}\right)\sqrt{m_0k_{ex}}} \\ \omega_{vib} = \sqrt{\frac{k_{vib}}{m_{vib}}}, \quad \xi_{vib} = \frac{c_{vib}}{2\sqrt{m_{vib}k_{vib}}} \end{cases} \quad (2)$$

$$LOAD = f_{ex} + k_m x_{eng} + c_m \dot{x}_{eng} \quad (3)$$

- x_{eng} : Displacement of engine mass (m)
- x_0 : Displacement of fluid mass passing shake orifice (m)
- x_{vib} : Displacement of moving parts of actuator (m)
- m_{eng} : Engine mass (kg)
- m_0 : Fluid mass passing shake orifice (kg)
- m_{vib} : Mass of moving parts of actuator (kg)
- k_m : Main spring constant (N/m)
- k_d : Spring constant of diaphragm (N/m)
- k_{vib} : Spring constant of excitation plate (N/m)
- k_{ex} : Extended spring constant of main rubber (N/m)
- A_{ex} : Effective area of main rubber (m²)
- A_0 : Cross-sectional area of shake orifice (m²)
- A_{vib} : Effective area of excitation plate (m²)
- A_d : Effective area of diaphragm (m²)
- $LOAD$: Subframe transfer load (N)

3. Verification of Calculation Model

A study was conducted to verify the accuracy of the model described in Section 2 above. The accuracy of passive characteristics was verified in the dynamic spring constant, and the accuracy of active characteristics was verified in the displacement of moving parts during operation of the actuator.

3.1. Passive Characteristics

Equations (1a) – (1d) were transposed as shown in Eq. (4) to express frequency response. This expression was solved for X_{eng} , X_0 and X_{vib} , and Eq. (3) was used to generate Eqs. (5) and (6). These equations express the dynamic spring constant.

$$\begin{aligned} x_{eng} &= X_{eng} e^{j\omega t}, \quad x_0 = X_0 e^{j\omega t}, \quad x_{vib} = X_{vib} e^{j\omega t} \\ f_{eng} &= s_{act} = 0 \end{aligned} \quad (4)$$

$$\begin{aligned} |K_s| &= abs \left[k_{ex} \left\{ -\left(\frac{A_{ex}}{A_{vib}}\right) \left(\frac{AH_{12} - BH_{12}}{H_{11}H_{22} - H_{12}H_{21}} \right) \right. \right. \\ &\quad \left. \left. - \left(\frac{A_{vib}}{A_{ex}}\right) \left(\frac{BH_{11} - AH_{11}}{H_{11}H_{22} - H_{12}H_{21}} \right) \right\} + 1 + k_m + j\omega c_m \right] \end{aligned} \quad (5)$$

However,

$$\begin{cases} H_{11} = -\omega^2 + 2j\omega\xi_0\omega_{n0} + \omega_{n0}^2 + \frac{k_v}{m_v}\left(\frac{A_v}{A_n}\right)^2 \\ H_{12} = \frac{k_v}{m_v}\left(\frac{A_v}{A_n}\right)\left(\frac{A_v}{A_n}\right) \\ H_{21} = \frac{k_v}{m_{vb}}\left(\frac{A_{vb}}{A_n}\right)\left(\frac{A_v}{A_n}\right) \\ H_{22} = -\omega^2 + 2j\omega\xi_{vib}\omega_{vib} + \omega_{vib}^2 + \frac{k_v}{m_{vb}}\left(\frac{k_v}{m_{vb}}\right)^2 \\ A = \frac{k_v}{m_v}\left(\frac{A_v}{A_n}\right) \\ B = \frac{k_v}{m_{vb}}\left(\frac{A_{vb}}{A_n}\right) \end{cases} \quad (6)$$

Figure 3 shows a comparison of the dynamic spring constant calculated using Eq. (5), and values measured in tests. Employing parameters based on design values, a good correlation between calculated values and measured values was obtained. The effect of resonance of the fluid orifices appears as peaks in the region of several Hz, and the effects of resonance in the moving parts of the actuator and the fluid which passes through the filter orifice appear as peaks between 100 and 150 Hz.

3.2. Active Characteristics

A time series simulation was conducted based on Eq. (1), and the results were compared with values measured in tests. Actuator force was found from a polynomial approximation of the non-linearity of the relationship between force characteristic and current as shown in Fig. 4 and the measured current shown in Fig. 5. Figure 6 shows the calculated force.

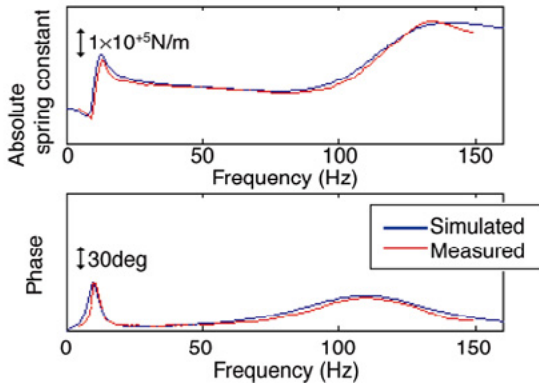


Fig. 3 Passive characteristic

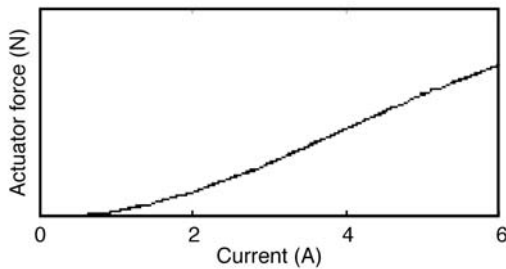


Fig. 4 Force characteristic of solenoid actuator

Figure 7 shows time series data for displacement of the moving parts of the actuator, and Fig. 8 shows a comparison of simulation results and measurements of the frequency characteristics of the displacement. There is a disparity of 0.4% in 1st-order components of driving frequencies and 9.4% in 2nd-order components between the simulation results and the measured values, enabling the model to be judged as possessing a sufficient degree of accuracy. The disparity in 3rd-order components is larger at 119%, and this is considered to be a result of the simplification of the friction loss characteristic of the moving parts of the actuator and the actuator force characteristics in the polynomial approximation.

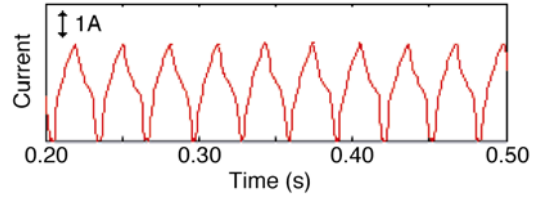


Fig. 5 Measured current

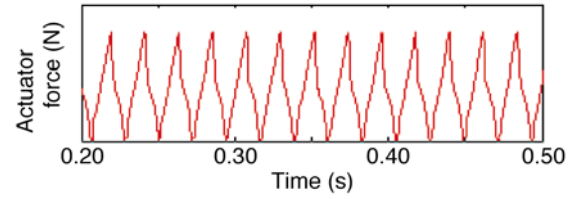


Fig. 6 Calculated force

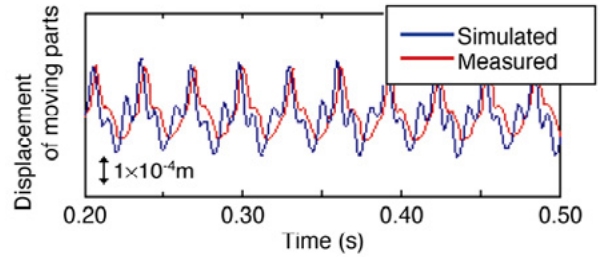


Fig. 7 Displacement of moving parts

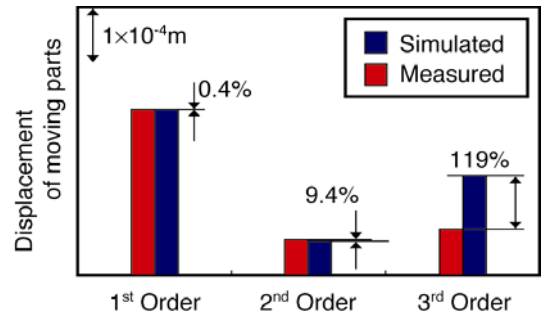


Fig. 8 Frequency characteristic of moving parts

4. Required Force Characteristic of ACM

Using this model, it is possible to determine the required actuator force to cancel engine vibrations. Substitution of Eq. (7) into Eqs. (1a) – (1d) enables the actuator force (S_{act}) at which subframe transfer load is at its minimum ($LOAD \rightarrow 0$) to be found. This generates Eqs. (8) and (9).

$$x_{eng} = X_{eng}e^{j\omega t}, x_0 = X_0e^{j\omega t}, x_{vib} = X_{vib}e^{j\omega t} \quad (7)$$

$$f_{eng} = 0, s_{act} = S_{act}e^{j\omega t}, LOAD \rightarrow 0$$

$$S_{act} = \{RFA + RFB (RFC + RFD)\}X_{eng} \quad (8)$$

$$= RF(\omega)X_{eng}$$

However,

$$\left\{ \begin{array}{l} RFA = \left(\frac{A_a}{A_m}\right)(k_m + j\omega c_m) \\ RFB = -m_{vib}(\omega^2 - 2j\omega\xi_{vib}\omega_{vib} - \omega_{vib}^2) \\ RFC = \left(\frac{A_a}{A_{ab}}\right)\frac{k_a + k_s + j\omega c_a}{k_a} \\ RFD = -\frac{1}{m_0}\frac{k_a + j\omega c_a}{\omega^2 - 2j\omega\xi_0\omega_0 - \omega_0^2}\left(\frac{A_a}{A_m}\right)^2 \end{array} \right\} \quad (9)$$

Figure 9 shows the required force function calculated using Eqs. (8) and (9). The actuator force required to minimize subframe transfer load against any given engine vibrations can be determined, enabling study of actuator specifications and control current to the ACM.

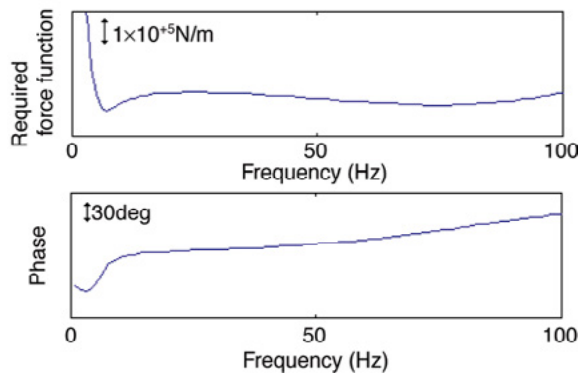


Fig. 9 Required force function

5. Conclusion

The research described in this paper constructed a detailed ACM model. The following results were achieved:

- (1) The addition of a dynamic model of the moving parts of the actuator to a conventional hydraulic mount model and the expression of resonance in two fluid orifices enabled the construction of a model in which the passive characteristics (dynamic spring constant) were expressed with a high degree of accuracy.
- (2) The model expresses the high-order components of driving frequency, which normally represent an issue in achieving consistency with the characteristics of actual vehicles, with a high degree of accuracy.
- (3) The ACM model enables the actuator force required to cancel subframe transfer load to be determined.

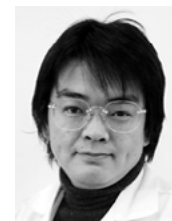
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