

New Vibration Control Methodology in Engine Mount System for Low-Fuel Consumption Engines

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ABSTRACT

With growing demands for better fuel economy and reduced carbon emissions there is a need for smaller and more fuel efficient engines. At the same time, to improve passenger comfort there are also demands placed on improved vehicle quietness [1]. A Homogeneous Charge Compression Ignition (HCCI) system or a higher compression ratio system can be used to obtain better fuel economy but the enhanced combustion rate causes an increase in engine vibration in the medium to high frequency range [2, 3]. To ensure vehicle quietness, this issue of structure-borne noise that is transmitted from the engine mounts to the body must be addressed.

In this paper a simple anti-vibration active mount system is introduced that can significantly reduce structure-borne noise at medium to high frequencies. This is achieved by adding mass to the insulator which leads to resonance at lower frequencies, in order to obtain double anti-vibration performance. This works to significantly reduce the transmission of approximately 300Hz to 1000Hz noise to the body. To suppress the increased vibration due to resonance of added mass, a control force which is proportional to the vibration velocity is applied. Additionally, an equation is developed for reducing force transmitted in directions other than direction of principal force. The resulting active double anti-vibration engine mount system can be used with fuel efficient engines to ensure improved vehicle quietness.

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INTRODUCTION

There are increasingly greater demands today for more fuel-efficient engines that are designed to achieve higher fuel economy and reduce carbon dioxide emissions. At the same time, there are also strong demands for improved vehicle quietness in order to enhance passenger comfort.

The adoption of a higher compression ratio or a Homogeneous Charge Compression Ignition (HCCI) system for reducing fuel consumption produces a much faster combustion rate that increases engine vibration at medium to high frequencies. As a result, while the engine consumes less fuel, reducing the structure-borne noise transmitted from the engine mounts to the body is an issue that must be addressed to achieve a quiet vehicle.

That requires an engine mounting system capable of satisfying the requirement for quietness. However, the conventional technique of adding resonance to an engine mount to obtain a double anti-vibration effect has a drawback in that the resonance increases structure-borne noise. Active control mounts can markedly reduce low-frequency structure-borne noise, but because of their control and structural

complexity, they cannot realistically control noise up to the medium and high frequency range [4,5,6].

This paper presents a simple anti-vibration active mount that can markedly reduce structure-borne noise at medium to high frequencies. The noise reduction effect obtained with this mount is described on the basis of an investigation of the physics principle.

Specifically, a mass is added to the insulator for setting resonance at lower frequencies in order to get double anti-vibration performance. This works to significantly reduce the transmission of approximately 300Hz to 1000Hz noise to the body. Moreover, the increased vibration due to the added-mass resonance is suppressed by applying a control force proportional to the vibration velocity. Additionally, a relational equation is introduced to cancel the added-mass resonance in directions other than the direction in which the large force input from the engine. As a result, it was found that this mount makes it possible for a fuel-efficient engine to provide both improved fuel economy and quietness.

ENGINE MOUNT ISSUES

This section describes various issues concerning conventional engine mounts that isolate force inputs from the engine and also a double anti-vibration engine mount that is effective in isolating vibrations at medium to high frequencies.

Engine Mount Structure

Figure 1 shows the conceptual structure of a conventional engine mount. An engine mount consists of a bracket on the engine side, a bracket on the body side and a rubber insulator in between connecting the two brackets. This structure provides the connection between the engine and the vehicle frame. In general, the large force input from the engine acts in the direction (referred to here as the direction of principal force) that bears the engine torque and engine mass. For that reason, the attachment points of the insulator to the engine and to the vehicle body are generally aligned approximately with the axis in which the principal force acts as shown in the figure. On the other hand, because of the space needed for installing the insulator, the points where engine forces are input and where forces are input to the body differ in the direction orthogonal to the principal force axis as illustrated in the figure.

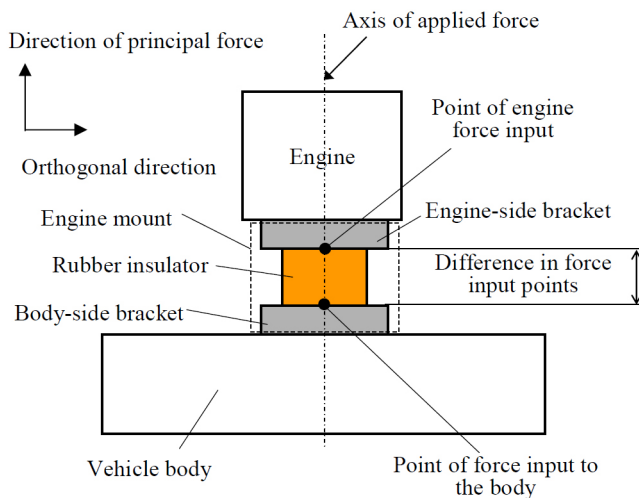


Figure 1. Elements composing engine mount

Engine Mount Issues

For fuel-efficient engines designed to reduce fuel consumption and CO₂ emissions, the reduction of structure-borne noise transmitted through the engine mounts is an issue of concern. Such noise can be effectively reduced by lowering the stiffness of the insulator at medium to high frequencies, however, that also has the attendant effect of decreasing the static stiffness of the insulator. That trade-off makes it difficult to balance vehicle handling and stability with ride comfort.

For that reason, it is necessary to have an engine mount system that allows insulator stiffness to be substantially reduced only at medium to high frequencies while still maintaining static stiffness.

Structure and Issues of a Double Anti-Vibration Engine Mount

Figure 2 shows the conceptual structure of a double anti-vibration engine mount, which has a mass added to the insulator. As a result, it generates low-frequency resonance, referred to here as added-mass resonance, consisting of the added mass and the parallel stiffness of the insulators on the excitation source side and the body side. This mount structure is designed to isolate vibration at medium to high frequencies by using the added-mass resonance while maintaining static stiffness.

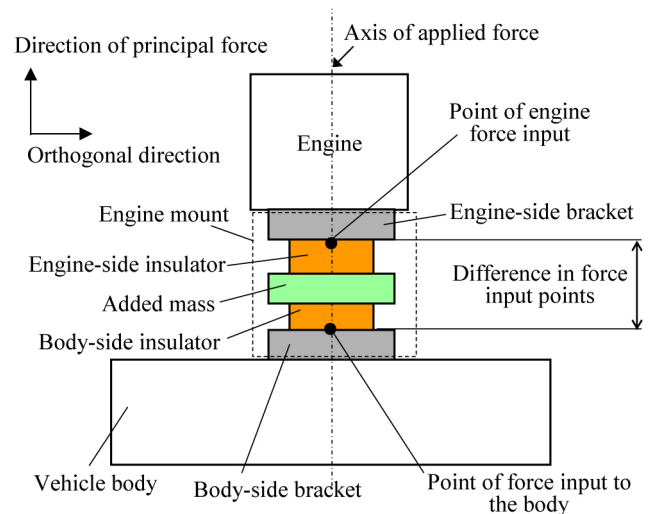


Figure 2. Elements composing double anti-vibration engine mount

However, there is concern that this added-mass resonance generated at low frequencies in the direction of principal force might substantially increase the structure-borne noise transmitted by the mounts and heard as interior noise. Specifically, with this added mass an anti-vibration effect is obtained starting from frequencies 1.4 times higher than the resonance frequency. This means that to obtain an anti-vibration effect, say, beginning from 300 Hz, the added-mass resonance must be positioned lower than 200 Hz. However, assuming a top engine speed of 6000 rpm, the force of the fundamental engine order of an inline 4-cylinder engine is applied up to 200 Hz and that of a V6 engine up to 300 Hz. Consequently, the added-mass resonance might cause a drastic increase in booming noise, which would be a problem.

Moreover, because the added mass has degrees of freedom in directions other than the direction of principal force, there is concern that the added-mass resonance in the orthogonal direction might also noticeably increase structure-borne noise. Accordingly, if the added-mass resonance can be

fundamentally controlled, it would be possible to significantly improve the anti-vibration effect at medium to high frequencies while still maintaining the desired static stiffness.

CONTROL PRINCIPLE OF ACTIVE ANTI-VIBRATION ENGINE MOUNT

This section proposes a new active vibration control principle for an engine mount constructed on the basis of the double anti-vibration mount described above. This control principle is intended to reduce structure-borne noise in a medium to high frequency range of 300 Hz to 1 kHz.

Physical Characteristics in the Direction of Principal Force

An excitation experiment was conducted to make clear the physical model of the double anti-vibration mount having the structure shown in [Figure 2](#). The solid red line in [Figure 3](#) shows the experimental force transmission characteristic to the body in the direction of principal force when a given unit of engine displacement was applied in that direction. The solid blue line shows the calculated force transmission characteristic obtained with a one-degree-of-freedom (1-DOF) physical model, consisting of the mass (m_a) of the added mass and parallel springs representing the stiffness of the engine-side insulator (k_d) and that of the body-side insulator (k_v). The good agreement seen between the experimental and calculated values indicates that the double anti-vibration mount can be expressed by the 1-DOF physical model shown in [Figure 4](#).

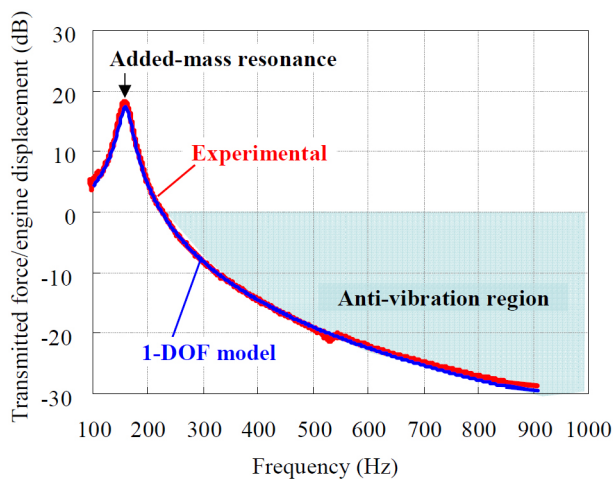


Figure 3. Force transmission characteristics of double anti-vibration mount

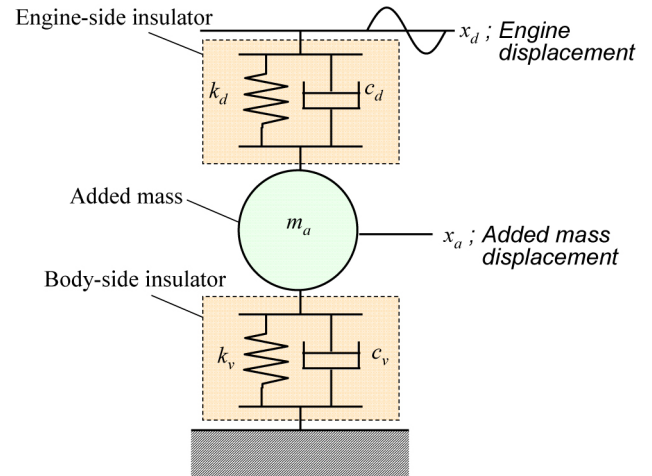


Figure 4. Physical model of double anti-vibration mount

Feasibility of Controlling Added-Mass Resonance Using Passive Technology

The issue of controlling the added-mass resonance using passive technology is described here. First, we will use the physical model of the double anti-vibration mount to illustrate the issues involved in trying to control the added-mass resonance by increasing insulator damping.

The equation of motion of the added mass can be written as shown in [Eq. \(1\)](#) below. The input to the added mass is assumed to be the displacement excitation from the engine. Because the force transmitted from the engine mount to the body (F_{trans}) is transferred only from the body-side insulator, it can be expressed as shown in [Eq. \(2\)](#). Applying the Laplace transform to [Eqs. \(1\) and \(2\)](#) and arranging the results yields [Eq. \(3\)](#). It will be noted that upper-case X_n is the variable obtained by applying the Laplace transform to x_n . Similarly, F_n is obtained in the same way from f_n . The subscript n is arbitrarily chosen.

$$m_a \ddot{x}_a + (c_d + c_v) \dot{x}_a + (k_d + k_v) x_a = c_d \dot{x}_d + k_d x_d \quad (1)$$

$$f_{trans} = c_v \dot{x}_a + k_v x_a \quad (2)$$

$$F_{trans} = \frac{(c_v S + k_v)(c_d S + k_d)}{m_a S^2 + (c_d + c_v)S + (k_d + k_v)} X_d \quad (3)$$

c_v : damping by body-side insulator

c_d : damping by engine-side insulator

x_d : engine displacement

x_a : Added mass displacement