

Reduction of Noise and Vibration in Compact Car Engine Mount System

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ABSTRACT

A new engine mount system has been developed to enhance noise and vibration performance and driving comfort, and thus increase competitiveness, in the next generation compact car. The basic system configuration is a principal axis of inertia type mounting system with torque rods that display excellent vibration isolation performance. In addition, the following were employed as guidelines for system characteristics in order to maximize performance. (1) Rod transmission characteristic that increases vibration isolation, to reduce booming noise and engine noise. (2) Rod mounting direction that reduces body shaking when the accelerator pedal is employed. (3) Control of engine rigid body vibration modes to increase driving comfort. (4) Dynamic stiffness of mounting points to reduce engine noise. The achievement of these characteristics has enabled the development of an engine mount system for a highly competitive compact car platform.

1. Introduction

A recent trend in automotive development is that vehicles in the same rank category have begun sharing platforms and developing derivative models such as sedans, station wagons and SUVs by interchanging the upper body and other parts. Sharing a common platform makes it possible to shorten the development period and reduce new costs, so development of platforms with high basic performance is demanded. On the other hand, booming noise while driving and engine noise are issues associated with principal axis of inertia type engine mount system commonly used in vehicles equipped with conventional in-line 4-cylinder engines. In addition, the noise and vibration produced by the large driving force and torque fluctuation vibration applied to the front and rear stoppers when starting are also issues for vehicles equipped with automatic transmissions and continuously variable transmissions (CVT).

This study aimed to construct a new engine mount system with

greatly enhanced noise and vibration performance and ride comfort as part of the development of a new public car platform with increased dynamic performance.

2. Overview

2.1. System Overview

Figure 1 shows the system configuration. The power plant is supported by the two mounting points located on the principal axis of inertia from the right side of the cylinder block towards the upper left part of the transmission, a torque rod that links the upper part of the cylinder block side mount with the top of the damper base, and a torque rod that links the bottom of the oil pan with the subframe. The structure is such that the power plant weight is borne by the two mounting points, and the driving reaction force is received by the two torque rods.

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2.2. Advantages of the Torque Rod System

The conventional principal axis of inertia type engine mount system has a structure which locates the stoppers that receive the driving reaction force in the vertical direction and supported by the front and rear beams of the subframe. The newly developed engine mount system with torque rods features a structure that receives the driving reaction force with the torque rods in the longitudinal direction as shown in Fig. 2. This structure has the following two advantages with respect to noise and vibration.

- (1) The torque rods do not simply receive the driving reaction force and control engine roll. The torque rods are elastically supported on both ends by rubber, and are comprised of rods made from steel, aluminum, plastics and other materials, so the vibration characteristics of the torque rods themselves provide significant vibration isolation effect.
- (2) The noise transfer function of the subframe, which is the input point for the vibration transmitted by the torque rods to the chassis, is generally lower in the longitudinal direction than the vertical direction. This means that an input format which is advantageous for idling vibration as well as noise and vibration when starting or otherwise operating at low engine speeds can be adopted by receiving the torque fluctuation vibration which acts in the direction of the crank axial rotation in the longitudinal direction⁽¹⁾.

The newly developed system makes maximum use of the above basic characteristics and applies the following technical elements to ensure high competitiveness. The torque rods are positioned horizontally for the reason in (2) above, but the mounting direction has been optimized to control the power plant behavior caused by the driving reaction force. Ride comfort is enhanced by employing

hydraulic engine side mounts and controlling the engine vibration in accordance with the spring characteristics of the transmission mounts. Dynamic stiffness targets that achieve the noise transfer function targets for the engine vibration transfer were set for each mounting point, and structural investigations were performed.

3. Basic Characteristics of the Torque Rod System

3.1. Reduction of Engine Booming Noise by Double Vibration Isolation Effects

A double vibration isolation structure establishes resonance systems in series within the vibration transfer path and provides significant vibration isolation effects after those resonance frequencies. Rubber floating of the subframe reduces the road and engine noise based on this principle. In the torque rod system, the rod weight and the spring setting of the rubber bushes on both ends make it possible to obtain these effects from frequencies of 100 Hz or less. Therefore, isolation of vibration can be expected to reduce booming noise and engine noise during driving. In vehicles with a conventional principal axis of inertia type mount system, booming noise caused by the 2nd order inertial force of a 4-cylinder engine is significantly affected by the side and rear mount vertical input as shown in Fig. 3. In contrast to this, the torque rod system provides significant isolation effects by attenuating the engine vibration at engine speeds higher than the rod resonance frequency as shown in Fig. 4. However, the vibration isolation effects decrease at the rod resonance frequency, so booming noise while driving is vastly

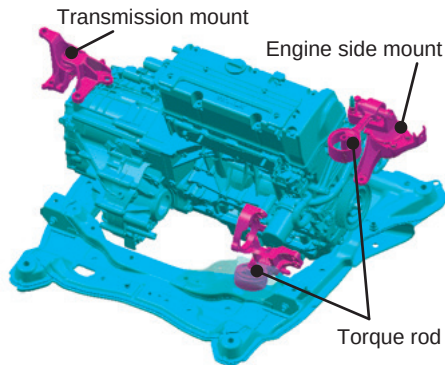


Fig. 1 Torque rod mount system configuration

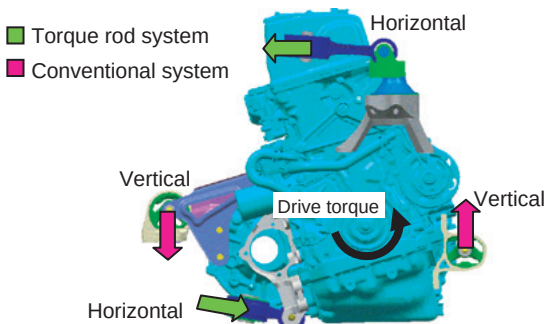


Fig. 2 Direction of principal input force

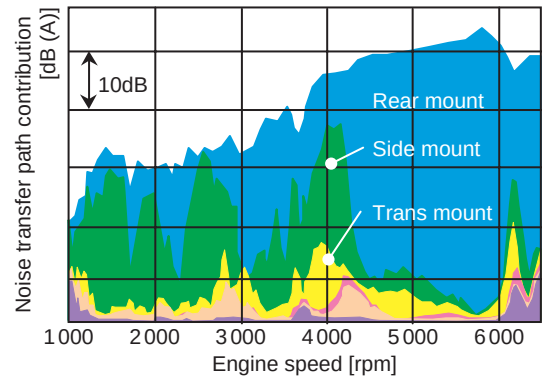


Fig. 3 2nd order booming noise transfer path contribution in conventional system

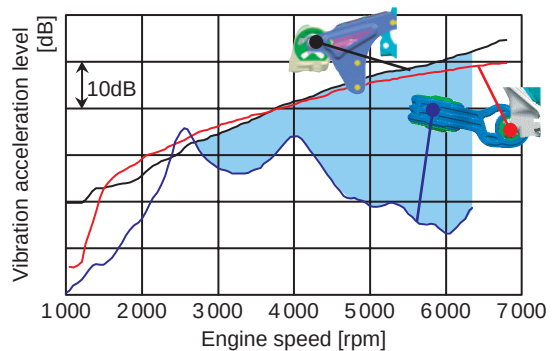


Fig. 4 Torque rod double floating effect in 2nd order vibration

reduced by setting the resonance frequency in a low sensitivity frequency band in view of the chassis sensitivity and other factors. Figure 5 shows the effects confirmed using a test vehicle.

3.2. Isolation of Torque Fluctuation Input

A large driving force is produced when a vehicle equipped with automatic transmission or CVT starts, and booming noise and vibration may occur due to an increase in torque fluctuation vibration and decrease of the mount transfer characteristics. In the conventional principal axis of inertia mount system, the vertical input component from the rear mount is dominant among the effects of the input from the engine mounts with respect to this phenomenon as shown in Fig. 6(a).

This is because the noise transfer function of the subframe is dependent on the following factors which are more severe in the vertical direction, as shown in Fig. 9.

- (1) The body shear bending mode shape, shown in Fig. 7 has a large effect on the sound pressure, and is easily excited by the subframe vertical input.
- (2) The subframe section shape is generally thin due to layout restrictions as shown in Fig. 8, which is disadvantageous in terms of vertical stiffness.

Therefore, in the principal axis of inertia type engine mount system with torque rods, engine vibration caused by driving reaction force and torque fluctuation can be input in the longitudinal direction, which is extremely advantageous with respect to these noise and vibration phenomena. Figure 6(b) shows the distribution of the sound pressure contribution for the principal axis of inertia

type engine mount system with torque rods, which shows that the sound pressure due to the main contributing factor of the torque rod longitudinal input is reduced, and that the overall sound pressure is also greatly reduced. Figure 10 shows the 2nd order booming noise reduction effects in a test vehicle.

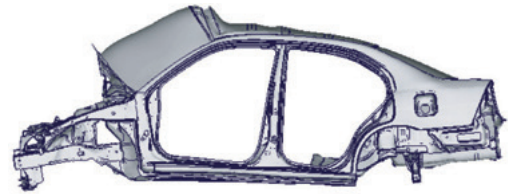


Fig. 7 Body bending mode



Fig. 8 Subframe section

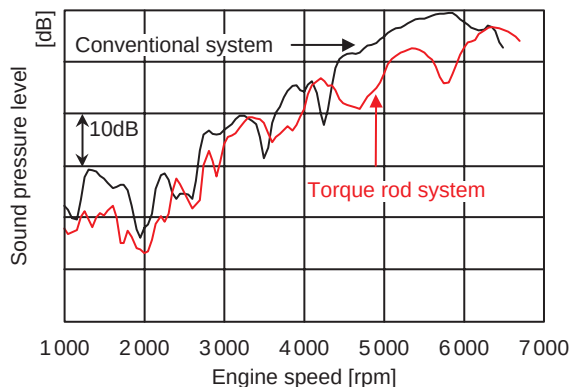


Fig. 5 Enhancement of 2nd order booming noise level

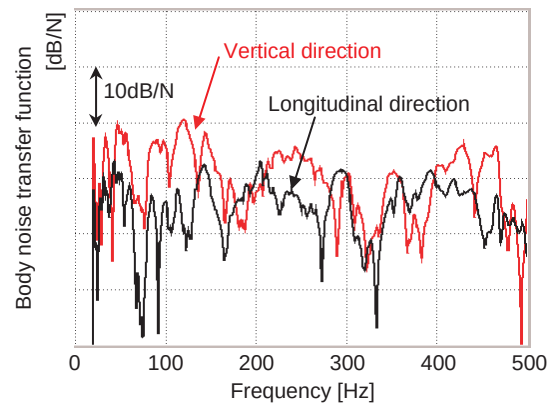


Fig. 9 Body noise transfer function of subframe

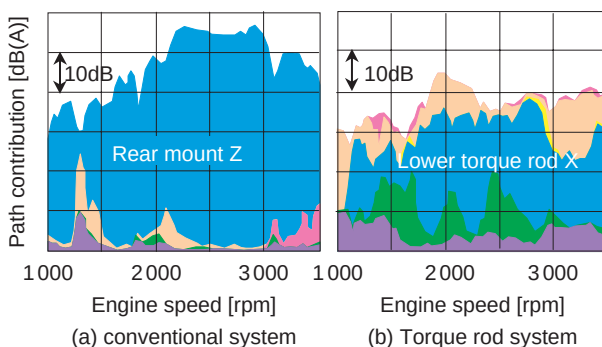


Fig. 6 2nd order booming noise transfer path contribution

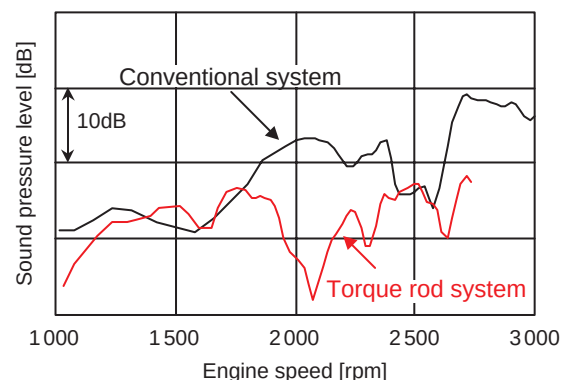


Fig.10 Enhancement of 2nd order booming noise

4. Torque Rod High Frequency Vibration Transfer Characteristics

Like booming noise while driving, engine noise performance can also be reduced up to a frequency of 1 kHz or more which is affected by the mount transfer characteristics by controlling the vibration transfer characteristics of the torque rods, based on the chassis sensitivity and other factors. This study defined the double vibration isolation effects of the torque rods by the transmissibility of the discrete torque rods as shown in Fig. 11.

$$\text{Transmissibility} = F_1/F_0 = X_1 \times K_1/F_0 = \int A_1 \times K_1/F_0$$

The torque rod transmissibility was quantitatively predicted using a discrete CAE model that takes into account the rod and the bushes on both ends, for which the rod shape and bush spring rates were determined. Figure 12 shows the calculation results for the transfer function. The vibration isolation effects generally tend to increase as the rod weight increases and the resonance frequency is set at low frequencies. However, vibration isolation performance can be satisfied by setting the peak of the torque rod transfer characteristics in the frequency band with low chassis sensitivity in the same manner as that of the booming noise region, described in the previous section.

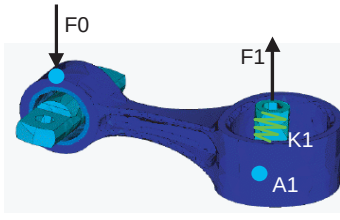


Fig. 11 Transmissibility simulation model of torque rod

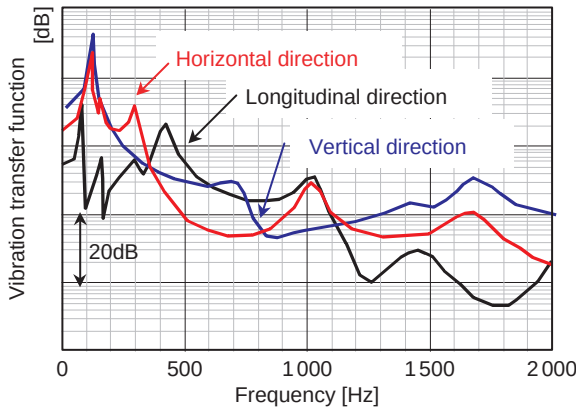


Fig. 12 Vibration transfer function of torque rod

5. Torque Rod Mounting Direction

The required rotational stiffness of the engine mount system is set to ensure that there are no part interference issues within the engine room and that the shock and body shake phenomena of manual transmission vehicles are at an acceptable level. In the torque rod system, the mounting direction is also important in addition to the bush spring rate. The torque rods have a small spring rate in the bush rotational direction, and rod displacement can only be restrained in the axial direction. Therefore, roll stiffness can be

increased by reducing the offset angle between the displacement direction of the rod parts that are connected to the power plant and the rod axial direction. On the other hand, the translational motion of the power plant may have an adverse effect on body shaking, but this can be enhanced by shortening the distance between the power plant center of rotation and center of gravity⁽²⁾. The roll center of the power plant when accelerating or decelerating can be controlled according to the torque rod mounting direction as shown in Fig. 13. Therefore, this study determined the torque rod mounting direction as shown in Fig. 14 to minimize the translational motion of the center of gravity. This reduced the body shaking by approximately 30%.

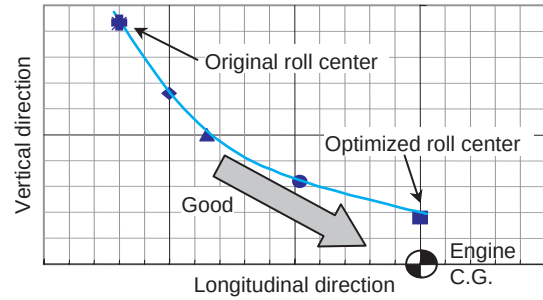


Fig.13 Optimization of engine roll center axis

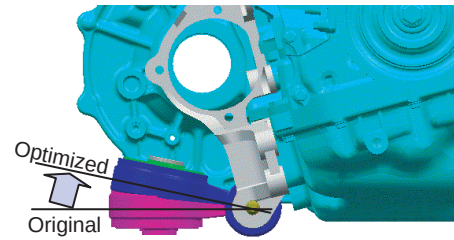


Fig.14 Optimization of torque rod angle

6. Enhancement of Ride Comfort

It is important to control the power plant rigid body vibration modes in order to reduce low frequency (5 to 20 Hz) chassis vibration due to the road surface input⁽³⁾. The newly developed system focused on the bounce mode and pitch mode (longitudinal axial rotation) of the power plant vibration, and performed mode control. The horizontal positions of the power plant center of gravity and the elastic center of the mount can be shifted as shown in Fig. 15, and bounce and pitch vibration can be excited with respect to the tire input by increasing the vertical spring ratio of the engine side mount and transmission mount. This produces a phase difference between the vertical vibration on the right and left sides of the power plant as shown in Fig. 16, which reduces chassis vibration by causing mutual interference between the inputs to the chassis. The results of bench vibration tests are shown in Fig. 17, and confirmed that vibration decreased by approximately 3 dB compared to the conventional model.

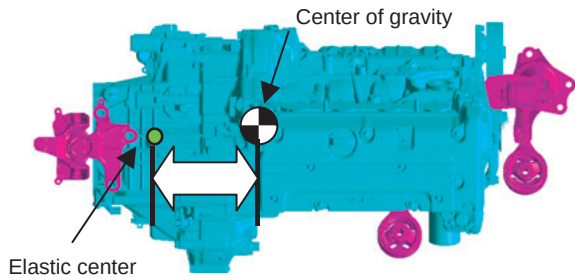


Fig. 15 Horizontal distance between engine center of gravity and elastic center of mount system

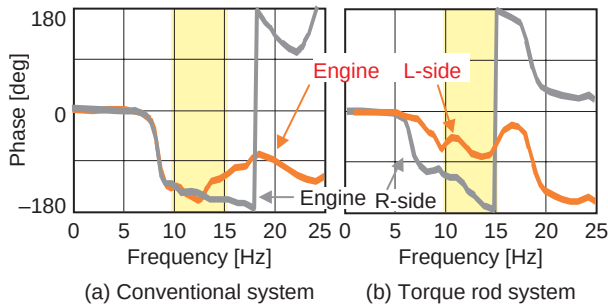


Fig. 16 Comparison of body transmissibility phase

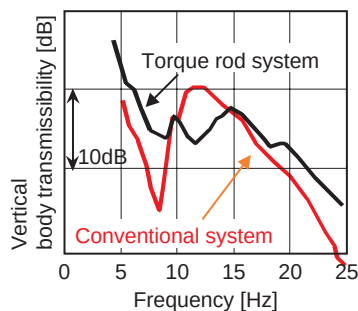


Fig. 17 Comparison of body transmissibility

7. Engine Mount Mounting Point Stiffness

In order to achieve the high frequency noise transfer function target for the engine mounting points, the correlation between the noise transfer function of the mounting points and dynamic stiffness values was first obtained through experiments. The noise transfer function and the dynamic stiffness values are roughly proportional within the same chassis, so the system target was replaced with the dynamic stiffness values and structural investigations were performed using a finite element model. The mounts fastened to the side frame employ mounting bolts with a collar structure as shown in Fig. 18(a), (b) and (c), stays were attached between the mounts and the wheel housing or the upper member as shown in Fig. 18(d) and (e) to increase lateral stiffness. In addition, the target stiffness was ensured by using aluminum die cast and steel as appropriate for the mounts and stays. Figure 19 shows the noise transfer function level achieved at the side engine mount. This shows that the acoustic sensitivity in the 100 to 300 Hz range can be greatly reduced by increasing the dynamic stiffness to the target value.

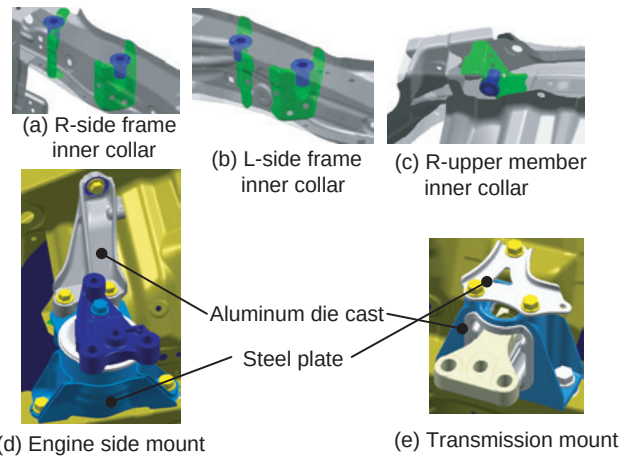


Fig. 18 Body construction in mount area

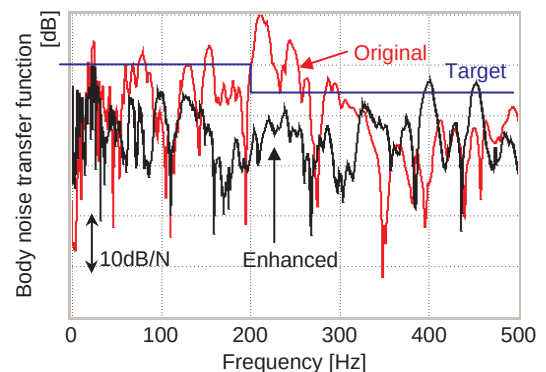


Fig. 19 Enhancement of body noise transfer function at engine side mount

8. Conclusion

The following technologies for reducing noise and vibration were applied to a principal axis of inertia type engine mount system with torque rods developed for a full model change for the next generation compact car.

- (1) Torque rod high frequency vibration isolation performance
Vibration isolation performance and weight reduction were simultaneously achieved by defining the double vibration isolation effects with the transfer characteristics of the discrete torque rods and performing optimization using a finite element model.
- (2) Torque rod mounting direction
Body shaking was reduced by ensuring rotational stiffness and by controlling the engine center of rotation.
- (3) Enhanced ride comfort
Chassis vibration was reduced by determining the vertical spring of the mounts on both sides so that the engine bounce vibration and pitch vibration are coupled.
- (4) Enhanced dynamic stiffness of mounting points
Engine noise was reduced by quantitatively setting dynamic stiffness targets that have high correlation with the noise transfer function, and by determining the structure based on these results.

Applying these technical elements in addition to the reliable basic noise and vibration characteristics of the torque rods made it possible to develop a highly competitive principal axis of inertia type engine mount system with torque rods for a public car platform.

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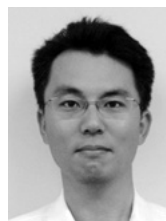
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