

Prediction Method for Vibration Transmission of Hydraulic Engine Mount

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ABSTRACT

A vibration transfer function predictive technique for hydraulic engine mounts was devised in order to reduce medium- and high-frequency band noise (e.g. engine noise, gear noise, motor noise) while the vehicle is in motion. The procedure was to model a hydraulic mount, beginning by predicting the dynamic spring characteristics of highly deformed rubber and then performing coupled analysis of the structure and the fluid sealed therein. Then, the vehicle model was created and a mode synthesis method was used to predict the vibration transfer function from engine to body. As a result, it was possible to predict the transfer function with a precision of less than 4% frequency error and 3 dB response level error at the resonance peak. It also became possible to cut analysis time from the typical 104 hours approximately to only about 35 hours by innovating the modeling technique and calculating conditions. This technique enables advance studies of hydraulic mount characteristics on a simulation model even during the initial stages of engine development, before a prototype vehicle is available.

1. Introduction

It has become imperative in recent years to decide the specifications of vehicles during the initial stages of engine development, before a prototype vehicle is available, in order to cut development costs and time. In the field of noise and vibration, auto manufacturers have to evaluate the vehicle functions and examine the specifications in the initial stage of engine development, without a prototype vehicle.

Noise and vibration while the vehicle is in motion includes engine noise, gear noise, and motor noise in the medium- and high-frequency bands of approximately 200 Hz to 1000 Hz. Reducing this noise is an important issue⁽¹⁾.⁽²⁾ There are two channels by which power train vibration is transferred as noise to the passengers' ears: airborne propagation and solid-borne propagation. The engine mount is often the main channel for solid-borne propagation. For that reason, reducing medium- and high-frequency band noise requires advanced vibration transfer characteristics of the engine mount⁽²⁾.

In this study, a technique was devised to predict the vibration transfer function using a simulation model of vehicles in which the power train and body are coupled

to each other with a hydraulic mount. This paper gives an overview of the technique and gives the results of checking the precision of its predictions.

2. Issues and Aims

Hydraulic engine mounts are often used as main mounts. Currently, however, although there is a vibration transfer function predictive technique for the low-frequency band of up to about 100 Hz⁽³⁾, there have been no reports of the same for the medium- and high-frequency bands. There are three issues involved.

- (1) Engine mount rubber often sustains the power train load and is deformed as a result, so the dynamic spring characteristics in this state have to be predicted
- (2) No fluid and structure coupling technique has been devised for hydraulic mount modeling
- (3) Hydraulic mounts have many components, which means a large number of model elements are needed, so analysis takes a great deal of time

In this paper, the fluid and structure coupling and the long analysis time were taken up as particularly unique issues of hydraulic mounts. The aim of this paper is to devise techniques to resolve the three issues above.

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3. Devising a Hydraulic Mount Dynamic Spring Characteristics Prediction Technique

As Fig. 1 shows, hydraulic mount systems consist of three components, broadly speaking: an engine bracket, a mount, and a holder. This is similar to mount systems with only vibration insulation rubber (referred to below as conventional mounts). Hydraulic mounts are typically used to enhance riding comfort dealing with low-frequency vibration as compared to conventional mount systems. Damping of the viscosity of the fluid inside as it passes through the orifice leads to the damping of vibration in the low-frequency range.

In order to predict the vibration transfer function of a vehicle using a hydraulic mount, it is necessary to predict the dynamic spring characteristics of the hydraulic mount as a unit.

3.1. Predicting Dynamic Spring Characteristics of Vibration Insulation Rubber

As Fig. 1 shows, a hydraulic mount has more components than a conventional mount. Vibratory force input from the engine bracket passes through the vibration insulation rubber and propagates to the holder. Additionally, when the mount is installed on a vehicle, it sustains load from the power train, which causes the vibration insulation rubber to be deformed. In light of these facts, a technique that predicts dynamic spring characteristics of the vibration insulation rubber as a unit was developed, taking account of its deformation under power train load.

3.1.1. Predicting deformation of vibration insulation rubber as a result of power train load

Vibration insulation rubber that sustains power train load often undergoes great deformation, measuring about 5 mm. The stress and strain of rubber that deformed significantly are in a non-linear relationship to each other. In order

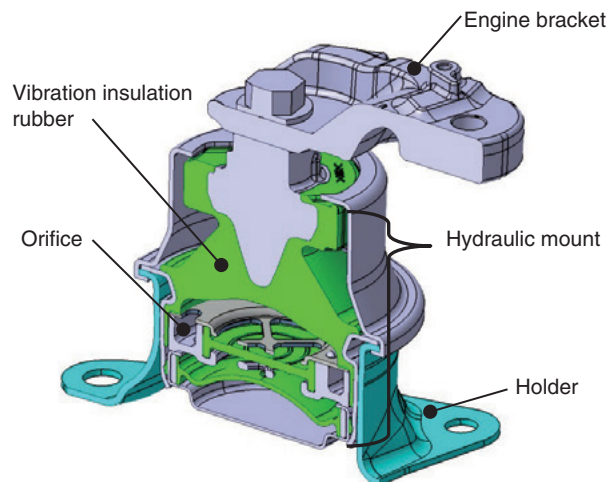


Fig. 1 Hydraulic mount systems

to reproduce this under the Finite Element Method, the Mooney-Rivlin model⁽⁴⁾ of strain energy density function W shown in Eq. (1) was used.

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) + C_{11}(I_1 - 3)(I_2 - 3) + C_{20}(I_1 - 3)^2 + C_{30}(I_1 - 3)^3 \quad (1)$$

$$I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2$$

$$I_2 = \lambda_1^2 \lambda_2^2 + \lambda_2^2 \lambda_3^2 + \lambda_3^2 \lambda_1^2$$

$$I_3 = \lambda_1^2 \lambda_2^2 \lambda_3^2 = 1$$

$$C_{ij} : \text{deformation tensor} \quad (i=0,1,2,3; j=0,1)$$

$$\lambda_i : \text{stretch ratio} \quad (i=1,2,3)$$

The coefficient C_{ij} in this model is derived from the results of biaxial tensile testing performed for each material and each hardness of the vibration insulation rubber.

Figure 2 shows the state of vibration insulation rubber deformation based on whether there is load, as found by simulation. It also shows results of static spring characteristics analysis of vibration insulation rubber. The analysis model is made of solid elements and yields results after 30 minutes of calculation. Predictive precision was -2.6%. This compares to the regulated target of within $\pm 5\%$ under power train load, depending on the model. This indicates that by databasing the coefficient C_{ij} and using the above model technique, it is possible to predict deformation form and static spring characteristics during the initial stages of engine development.

3.1.2. Predicting dynamic spring characteristics taking account of deformation under power train load

When vibration insulation rubber is subjected to the input of a resonance frequency, the dynamic spring constant rises. This is referred to as surging, and such surging has to be predicted in engine mount vibration transfer.

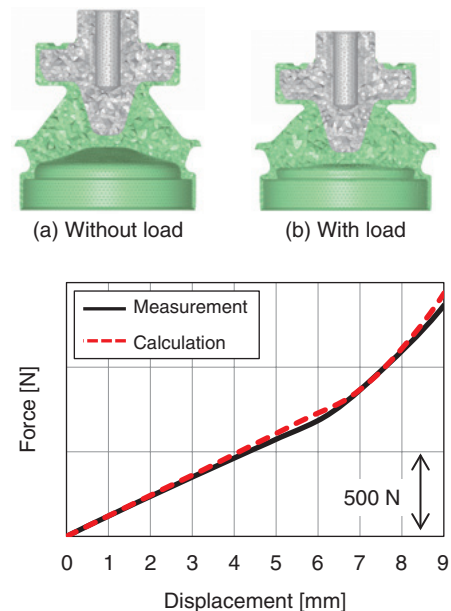


Fig. 2 Static spring analysis accuracy

The vibration insulation rubber's Young's modulus and structural damping coefficient vary according to the material and its hardness. Because the stress and strain of rubber have a non-linear relationship to each other, it is challenging to determine the Young's modulus to apply to it. The Young's modulus must be determined with the amount of strain as based on actual phenomena, or the surging frequency will deviate from actual phenomena. Thus, the following two methods were considered.

- i. Applying a Young's modulus based on strain distribution
- ii. Applying a uniform Young's modulus based on rubber form, hardness, and material

Vibration insulation rubber that is deformed as a result of sustaining power train load shows the distribution of strain, as illustrated in Fig. 3.

In Method i, a Young's modulus is applied based on the strain distribution derived in Section 3.1.1. In Method ii, a Young's modulus and structural damping coefficient are applied uniformly to the vibration insulation rubber model. Points A and B in Fig. 3 are the points where the greatest and least respective amounts of strain as derived in Section 3.1.1. were found. Figure 4 shows the relationship between stress and strain. The slope is the Young's modulus. The difference in slope of the same rubber hardness at Points A and B is small for the engine mount, which is the subject of this paper, and the relationship between stress and strain shown in Fig. 4 could be deemed as linear.

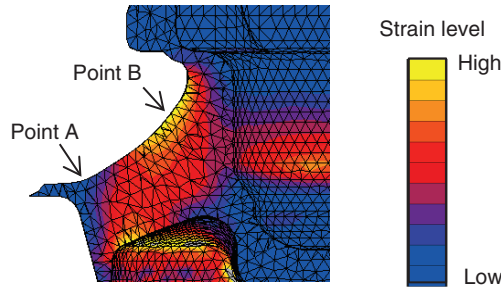


Fig. 3 Distribution of strain

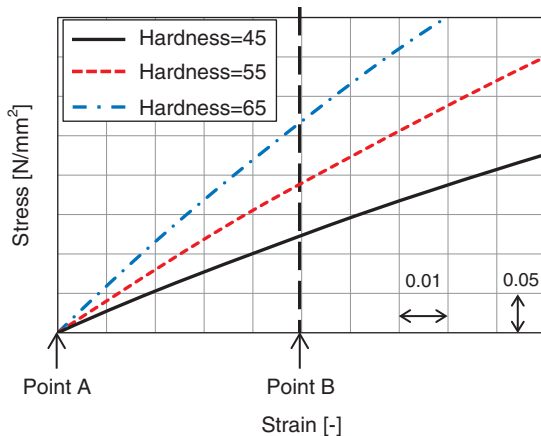


Fig. 4 Stress-strain characteristics

On the other hand, the difference in slope at hardness levels 45 and 55 was 54.5% at Point A and 55.7% at Point B, which is not negligible.

Therefore, it was assumed that equivalent calculation accuracy was obtained by creating a database for each hardness level and applying the same Young's modulus to all elements. A database was made not just for each rubber hardness but also for each rubber form and material, and structural damping coefficients were also included in a database. This method (Method ii) requires creating new databases, but it has an advantage in that it makes calculation time only about 10 minutes, even as it provides equivalent calculating precision to Method i (the precision difference between these two methods is within 2.8%). Based on the above consideration, it was decided to use Method ii.

Following are details on the method for databasing the Young's modulus and structural damping coefficient of Method ii. The method of measuring dynamic spring characteristics is as shown in Fig. 5. The vibration insulation rubber is affixed to a shaker and given the same initial strain that occurs in a vehicle when a power train is installed. After that, forced vibration is applied to the engine bracket at a fixed acceleration and the frequency is increased at fixed intervals. In the simulation model, rigid elements were used to reproduce a similar affixing as when the vibration insulation rubber was measured and a similar input as when the measurement was applied from the engine

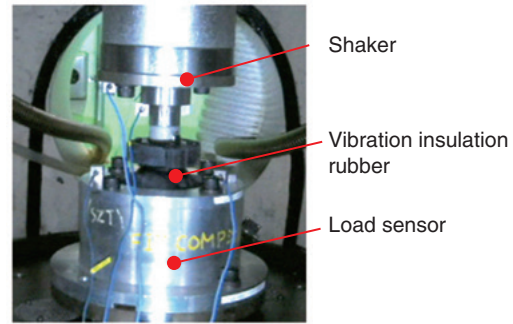


Fig. 5 Test setting

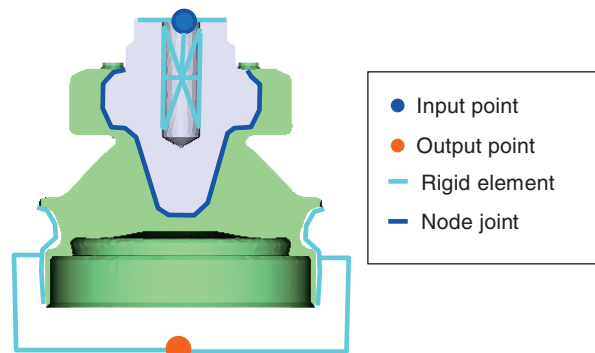


Fig. 6 FE model

bracket (Fig. 6). Proceeding in the above manner, it was possible to find a Young's modulus and structural damping coefficient satisfying the target precision during surging, namely within $\pm 3\%$ for frequency and $\pm 10\%$ for dynamic spring level, as shown in Fig. 7.

Figures 8 and 9 give examples from the materials databases created for this research, illustrating databases for a heat-tolerant super-low-motion amplifier for the conical mount shown in Fig. 6. Figure 10 shows results of general usefulness testing by making several models using the values established from these databases. In all cases, the results satisfied the target precision of within $\pm 3\%$ for frequency and $\pm 10\%$ for dynamic spring level. Therefore,

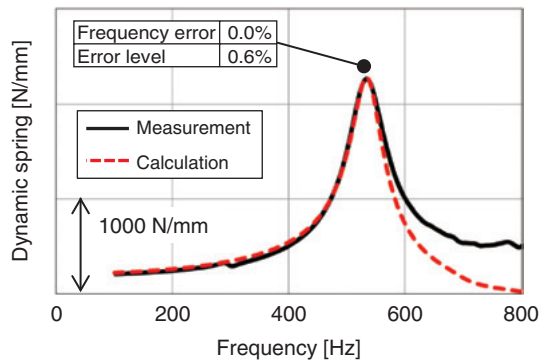


Fig. 7 Accuracy of dynamic spring analysis

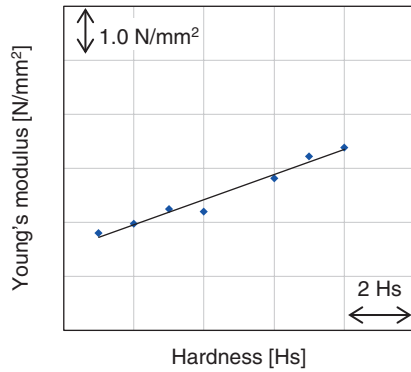


Fig. 8 Material database of Young's modulus

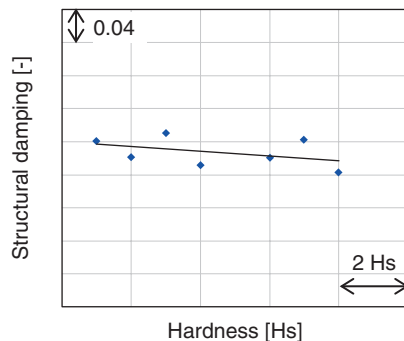


Fig. 9 Material database of structural damping

once the form, hardness, and material of vibration insulation rubber have been determined, it is now possible to estimate the Young's modulus and structural damping coefficient at the initial stages of engine development. Using the above technique, one can now predict the dynamic spring characteristics of the vibration insulation rubber as a unit, taking account of its deformation under power train load, during the initial stages of engine development.

3.2. Predicting Dynamic Spring Characteristics of Hydraulic Mount Components

Hydraulic mounts have an incompressible fluid sealed inside the mount, as illustrated in Fig. 1. To simulate this precisely, it is necessary to do a coupled analysis of fluid and structure. In the medium- and high-frequency bands, the mount input is of very slight amplitude, so the fluid flow was considered nonexistent. Taking Eq. (2) as a precondition, Eq. (3) expresses the coupled analysis as based on the fluid and structure pressure.

$$\rho \ddot{u} = -\nabla p \quad (2)$$

ρ : density
 $\ddot{u}$: acceleration
 p : pressure

$$\begin{bmatrix} M_s & 0 \\ -A^T & M_f \end{bmatrix} \begin{Bmatrix} \ddot{u}_s \\ \ddot{p} \end{Bmatrix} + \begin{bmatrix} B_s & 0 \\ 0 & B_f \end{bmatrix} \begin{Bmatrix} \dot{u}_s \\ \dot{p} \end{Bmatrix} + \begin{bmatrix} K_s & A \\ 0 & K_f \end{bmatrix} \begin{Bmatrix} u_s \\ p \end{Bmatrix} = \begin{Bmatrix} p_s \\ p_f \end{Bmatrix} \quad (3)$$

M : mass, B : damp, K : stiffness, u : displacement
 s : structure, f : fluid, A : boundary matrix, P : load

The fluid's volume elasticity modulus K_f is defined in Eq. (4).

$$K_f = C^2 \rho \quad (4)$$

C : speed of sound

The first thing to do was to judge whether coupled analysis of the fluid and structure is possible. In order to start from a simple structure, an analysis model and hydraulic engine mount trial equipment from which the membrane had been removed was prepared. The membrane

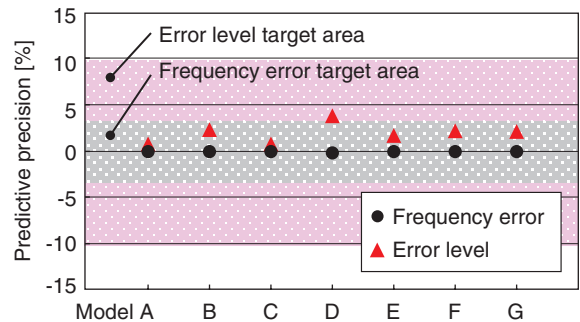


Fig. 10 General-purpose inspection of predictive technique

was made of rubber and separated the fluid chamber inside the mount into two parts, above and below, as shown in Fig. 11.

The measuring technique was similar to the measurement of dynamic spring of the vibration insulating rubber as a unit in Section 3.1.2.: trial equipment was put into the dynamic spring tester, forced vibration was applied at a fixed acceleration, and the frequency was increased at fixed intervals. In the analysis model, the same internal structure with the membrane removed as in the trial equipment was also prepared in the analysis model. In the model, similar affixing and input as in the actual experiment were given. The vibration insulation rubber material values used were values showing correlation between analysis and experiment as in Figs. 8 and 9. Additionally, the fluid model was defined with solid elements. In order to perform more precise coupling, the boundary nodes of the fluid model elements and solid model elements were set to have the same coordinates (Fig. 12). The material values of fluid elements were defined as density ρ , damping B_f , and speed of sound C .

Results of analysis as compared to measurement are shown in Fig. 13, along with dynamic spring characteristics of vibration insulation rubber as a unit, as measured in Section 3.1.2. The results showed that the technique was able to simulate the hole resonance close to 200 Hz as well as the dissipation of the surge close to 500 Hz. Hole resonance is a phenomenon such that the vibration insulation rubber is dynamically vibrated, and the enclosed fluid, which is under pressure from the vibration insulation rubber, resonates as a result of open holes in Device A shown in Fig. 11. The results showed that this hole

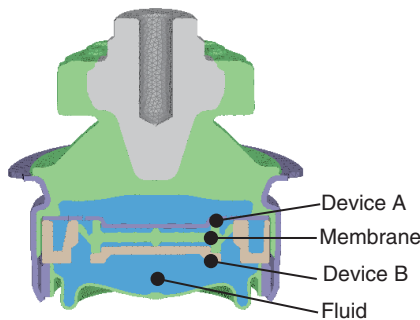


Fig. 11 Hydraulic mount

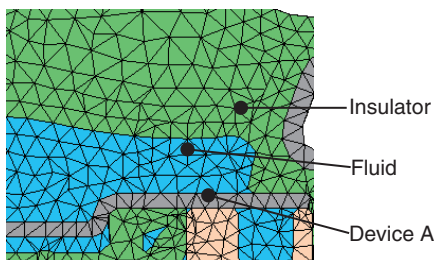


Fig. 12 Boundary condition of liquid model

resonance satisfies the target precision (frequency error within $\pm 10\%$, dynamic spring level error within $\pm 20\%$), so it may be concluded that it is possible to use pressure to perform coupled analysis of fluid and structure.

Next, the predictive precision with actual products containing membranes was tested. First, to confirm the impact of the membrane on hydraulic mount dynamic spring characteristics, three products were prepared with different membrane hardnesses. Then dynamic spring characteristics were measured in the same manner as for the vibration insulation rubber as a unit and for trial equipment with the membrane removed, as described above. Measurement results are given in Fig. 14. With the previously mentioned specimens with the membrane removed, after the hole resonance at 200 Hz, the dynamic

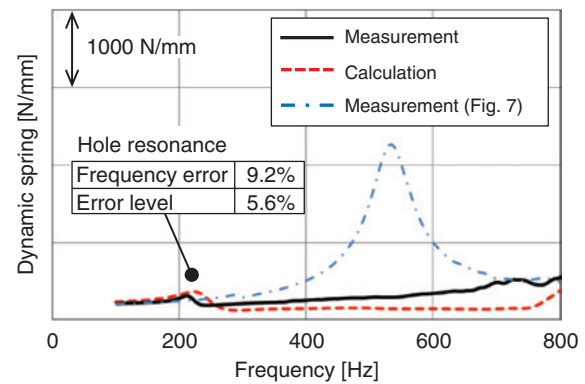


Fig. 13 Hydraulic mount without membrane model

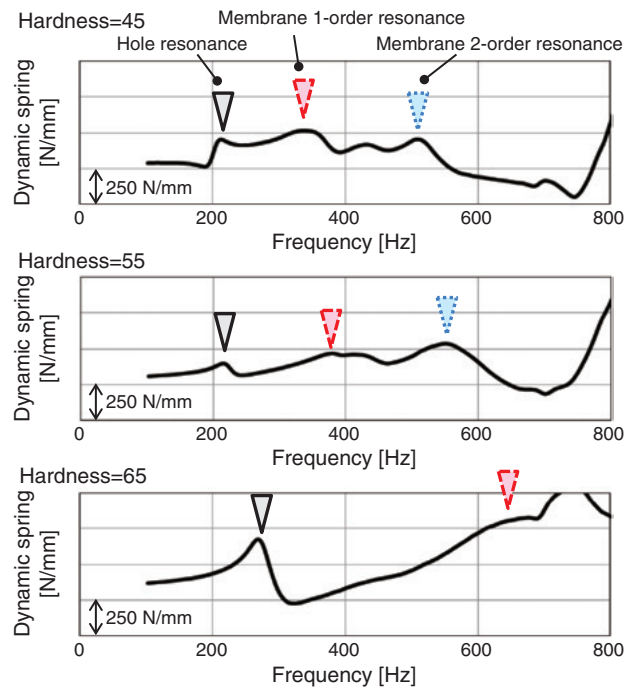


Fig. 14 Dynamic spring of hydraulic mount affected membrane hardness

spring was flat until almost 800 Hz. However, this graph shows that having a membrane causes resonance to occur between 300 Hz and 500 Hz, and as the hardness increases, so does the resonance frequency. The above indicates that the membrane affects hydraulic mount dynamic spring characteristics, so it is necessary for the analysis model to be able to reproduce membrane resonance as well.

As for the boundary conditions of the membrane and surrounding structures, these are not rigidly joined to each other in actual products, but rather are held together with a crimped structure. Therefore, the technique of joining them with a spring element was considered. For the spring constant, a value identified from results of several types of model was used.

Figure 16 and Table 1 show results of modeling with a spring element. In only about two hours of calculation time, the model was able to satisfy measuring target precision for both hole and membrane resonance (frequency error within $\pm 10\%$, dynamic spring level error within $\pm 20\%$). Based on the above, one can now predict the dynamic spring characteristics of hydraulic mount components, taking account of deformation under power train load, starting from the initial stages of engine development.

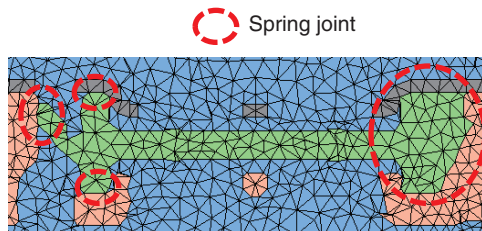


Fig. 15 Membrane mesh model

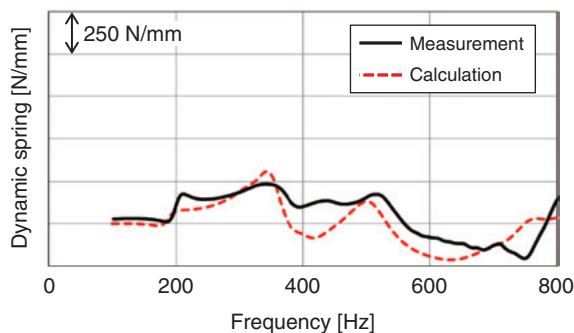


Fig. 16 Dynamic spring analysis result of hydraulic mount systems

Table 1 Accuracy of dynamic spring analysis

	Hole	Membrane 1-order	Membrane 2-order
Frequency error	-5.0%	0.9%	-3.1%
Error level	-19.7%	15.2%	-9.8%

4. Results of Application to Vibration Transfer Function for Vehicle

The mount vibration transfer function was predicted with the mount on a vehicle, as shown in Fig. 17. The new technique joined the existing technique⁽²⁾ — a computational model for metal fixtures like the bracket and holder — with power train and body models. Also, the vibration insulation rubber sustained the load of the power train and thus underwent great deformation. In attempting to reduce the technique's calculating costs, the power train model and body model were simplified to cover only those components affecting the engine mount transfer function. Also, modal reduction was performed at the mount bolting point and a vehicle model was developed by joining the components by component mode synthesis⁽⁵⁾.

The transfer function from the bolting point between the power train and engine mount to the bolting point between the body and engine mount, as found by component mode synthesis, is shown in Fig. 18. The results (peak frequency precision within 4% and peak level precision within 3 dB

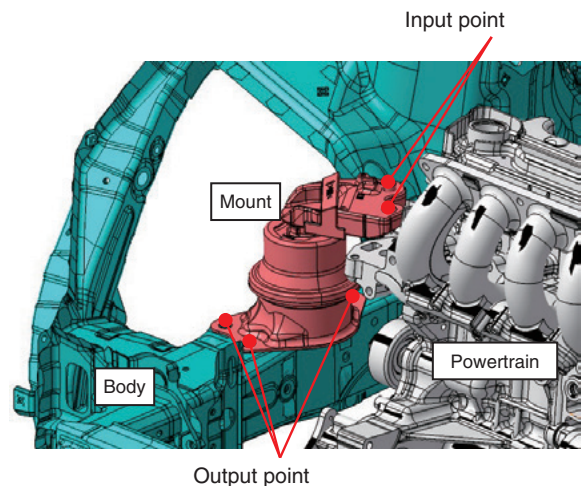


Fig. 17 Layout of mounting systems

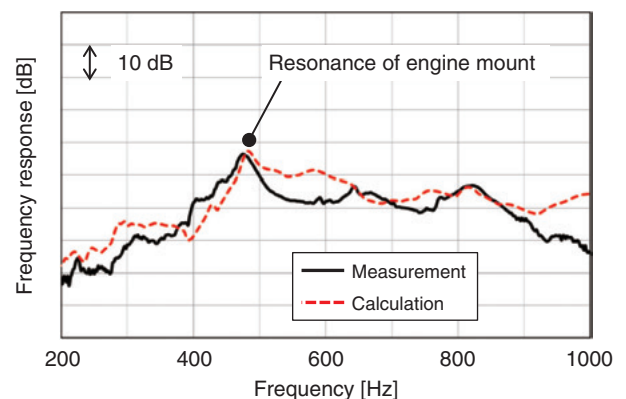


Fig. 18 Comparison between calculation and measurement for frequency response

of measured results on a vehicle) were precise enough to satisfy the targets (peak frequency precision within $\pm 5\%$ and peak level precision within ± 3 dB). Therefore, the model was able to predict the hydraulic mount vibration transfer function for engine, gear, and motor noise, etc. in the medium- and high-frequency bands of approximately 200 Hz to 1000 Hz with adequate precision.

In addition, by innovating the modeling and calculating conditions, etc., it was possible to perform analysis in just 35 hours: 24 hours to build the model and 11 hours to run the calculations. It conventionally took 56 hours to build the initial model and 48 hours to run the calculations, making a total of 104 hours. This was because of the time needed to mesh complex structures and the lengthy calculation time resulting from the increase in, for example, the coupling elements used at interfaces.

5. Conclusion

The following results were obtained in relation to a prediction technique for vibration transfer functions of hydraulic engine mounts in the medium- and high-frequency bands.

- (1) Through databasing the relationship of Young's modulus and structural damping coefficient in respect to rubber hardness, a technique was developed that predicts the dynamic spring characteristics of vibration insulation rubber while accounting for great deformation.
- (2) In addition to the technique described in (1), the dynamic spring characteristics of hydraulic engine mounts is able to be predicted by using coupled analysis of fluid sealed inside and structure.
- (3) By innovating the calculation techniques, using modal reduction of a large model, simplifying boundary conditions, and so on, the technique reduced calculating costs by about 66%.

The above made it possible to develop a technique to predict vibration transfer functions on an actual vehicle with sufficient precision for hydraulic engine mounts, which highly contributes to gear, motor, and other engine noise vibration transfer functions. This prediction technique can be applied in vehicle development, even in the initial stages of engine development, before a prototype vehicle is available.

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