

Research for Vibration Analysis with Full-Vehicle Model

Toshiya YOKOI*

Kengo OKAMURA*

Jiro NINOMIYA*

Masaki NAGAHISA*

ABSTRACT

Techniques of full vehicle analysis directed at all suspension and body-related parts were combined with test-based engineering and simulation techniques in a two-sided approach intended to develop a technology for reduction of floor and steering vibration. The test-based engineering clarified the mechanism of vibration initiation by means of full vehicle modal analysis, operational deflection shape analysis, and transfer path analysis implemented by excitation of the suspension. Modifications were then made to an actual car based on the results obtained from analysis. The simulation techniques involved modeling of all the subject parts, and the creation of a finite element model (160,000 nodes) that matched the actual car with a high degree of accuracy. Sensitivity analysis was then applied to analyze the weak points, and calculations were made to implement modifications in the finite element model. Both the test-based engineering and the simulation techniques were confirmed to reduce vibration levels in the actual car, thus substantiating the validity of these analytical techniques.

1. Introduction

In new model development, there continues to be requirements to make the body lighter, as a measure for environmental protection. In addition, for collision safety performance, a variety of collision test methods continues to be established, with many requirements for body construction and body rigidity.

However, there are many times where the floor and steering vibration during driving does not match the measures being taken to reduce vehicle weight and to enhance collision safety. The quality required for floor and steering vibration is higher than it was before, and in order to balance this quality with other performance items, it has

become necessary to have effective vibration solutions that fully make clear the causes of the vibration. However, causes of vibration lie not simply in the tires and suspension; and because many parts, including the body and steering structures, are involved, it is often difficult to understand the vibration generating mechanisms. Highly accurate analysis technology is therefore required.

This research project has dealt with full vehicle analysis technology that looks at all parts contributing to floor and steering vibration. The methods of approach were test-based engineering using an actual car, and simulation technology using the finite element method. A reduction in vibration was realized by implementing countermeasures to the basic causes of vibration after analyzing the generating mechanisms. This paper reports on the details of the project.

* Tochigi R&D Center

2. Vehicle Analysis

2.1. Target Area Setting

After selecting, in an actual running test, the road surface and speed conditions where it is easy to feel the difference in vibration levels against a comparison vehicle, evaluation point vibrations levels were measured for the steering wheel and floor evaluation points. The selected conditions were a slightly rough road, at a vehicle speed of 45 km/h. The measurement results for the test vehicle and the comparison vehicle are shown in Fig. 1.

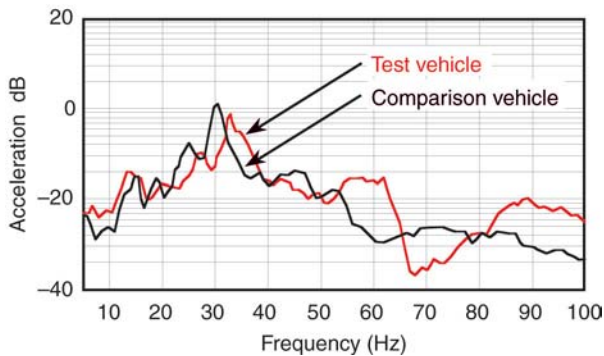


Fig. 1 Steering wheel vertical acceleration

In this way, measuring 3-direction translational acceleration of two evaluation points, the frequency range where the test vehicle is not as good as the comparison vehicle is brought out. Then, using operational deflection shapes and principal component analysis, as explained below, the ranges are categorized according to those having the same causes, from the vehicle's dynamic behavior and input force contributions. An outline explanation of the analysis method is given below.

2.2. Operational Deflection Shapes

This is a method for analyzing vehicle behavior in an actual running test. In this project, actual running test measurements were made of 3-axis acceleration at a total of 341 points on the suspension and body, and deflection shapes were specified in the relevant frequency ranges from animation observation.

2.3. Principal Component Analysis

Conventionally, front and rear wheels are excited separately, and input force contribution is determined against evaluation point response. However, this time the

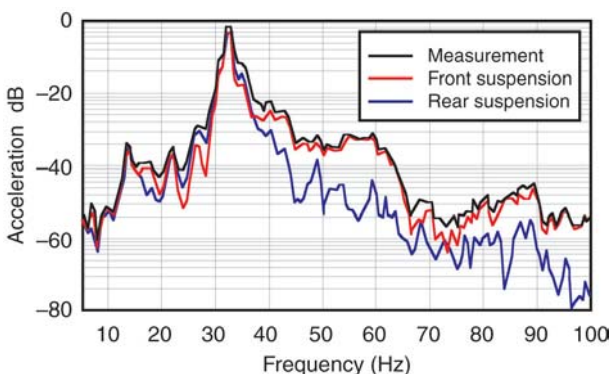


Fig. 2 Contribution of input force for steering wheel vertical acceleration

contribution of each input force was specified for each evaluation point vibration direction, applying multivariate analysis and using operational deflection shapes of actual running test data. As a result, it is possible to understand, by frequency range, the input contribution against evaluation point vibration in an actual running test. Figure 2 gives an example of principal component analysis calculation results vs. steering vertical vibration.

2.4. Full Vehicle Modal Analysis

The analysis method used in target area setting was introduced above, and other methods are introduced below, used in order to more fully understand the details of this phenomenon. The first is experimental modal analysis. In this project, using an actual car in a laboratory test, the unsprung mass (lower arm) had random burst excitation applied, for the front and rear wheels separately. Through this excitation of the input force points when driving, it was possible to discover the excitation mode in an actual running test. In addition, by simultaneously carrying out constant vertical excitation of the body of 2.5 Hz, it was possible to eliminate the high frequency friction component. Using this method, 8-100 Hz frequency vibrations can be analyzed.

2.5. Transfer Path Analysis

Unsprung mass is connected to the body through multiple suspension arms, etc., and if contributions for each transfer path are known, it is also possible to take countermeasures using suspension bushes. The second analysis method is one used to determine the contributions of each path, and for this, two types of measurements are necessary. The first is measurements of suspension attachment point response when exciting the evaluation point (body transfer function measurements). The second is measurements to determine bush transfer force (input force measurements) by calculating suspension bush displacement during an actual running test from relative acceleration and by multiplying it with bush dynamic elasticity modulus. By carrying this out in all three directions for each suspension attachment point, it is possible to calculate evaluation point response as the total of input force $\times$ body transfer function. Highly rigid bushes have small displacement, and with this method it is difficult to maintain calculation accuracy, but an example of transfer path analysis calculation results against steering fore-aft vibration is given in Fig. 3.

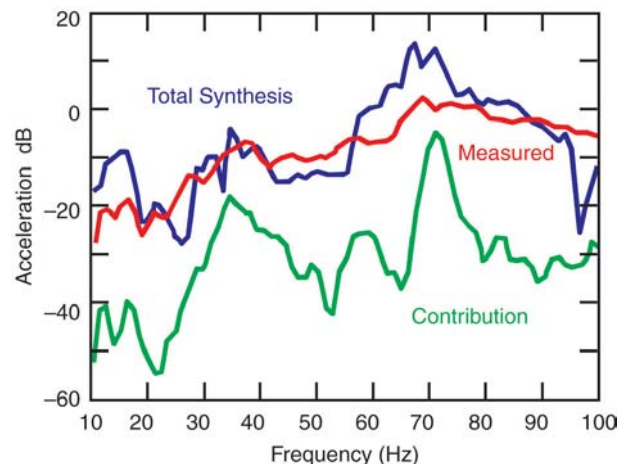


Fig. 3 Result of transfer path analysis for steering fore-aft vibration

In the chart, the red line represents steering fore-aft acceleration measurement values that are the evaluation points, and the blue line represents the calculated values. They are similar in inclination. A transfer path example, among the calculated values, is shown with the green line. It is possible to confirm that around 35 Hz and 70 Hz peaks, the contribution of the transfer path represented by the green line is large.

2.6. Weakness Analysis Results

An outline introduction of analysis methods was made above. Now two examples will be given, including the results of the analysis using both of these methods, and the various major causes in the areas where the test vehicle is worse than the comparison vehicle.

The first example is the area where vertical direction is worse for both the floor and steering wheel, around 26 Hz. Using full vehicle modal analysis in this area, (1) 25.43 Hz rear suspension vertical mode, (2) 27.89 Hz body 2nd bending mode, and (3) 32.07 Hz rear suspension fore-aft mode were discovered. From the mode shape, it is assumed that those contributing greatly to steering wheel vertical vibration are (2) and (3). The mode shapes are shown in Figs. 4–6.

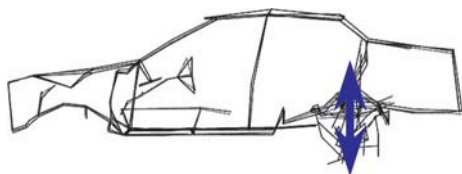


Fig. 4 Rear suspension vertical modal shape (Experimental modal analysis)



Fig. 5 Body 2nd bending modal shape (Experimental modal analysis)

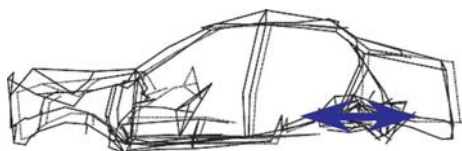


Fig. 6 Rear suspension fore-aft modal shape (Experimental modal analysis)

Checking these with operational deflection shape results, body 2nd bending was discovered at 26.86 Hz, and it was learned that rear suspension movement at that time was closer to vertical than fore-aft. (Fig. 7)

Using only experimental modal analysis, the focus is on suspension fore-aft vibration, but in this case the countermeasure must be applied for vertical vibration. It was learned that it is very important to fully and accurately understand the phenomena, using a number of methods.



Fig. 7 Body 2nd bending shape (Operational deflection shapes)

The second example concerns the steering wheel lateral vibration at 75–100 Hz. From the results of the experimental modal analysis, it was confirmed that, at 92.18 Hz, front sub-frame turning and fore-aft motion stimulate steering shaft lateral bend, and the steering wheel lateral vibration becomes larger (Fig. 8). This is a curve fitting of a high frequency range that is normally thought to be difficult, but by taking measurements fully confirming linearity, reciprocity, etc., and by repeatedly checking curve fitting, it is possible to have highly accurate curve fitting.

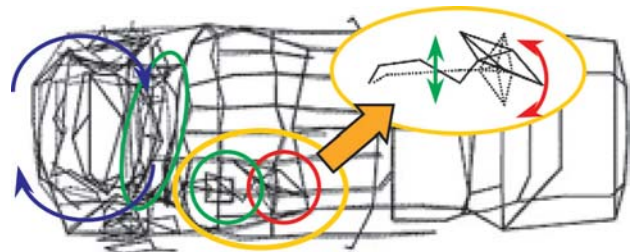


Fig. 8 Front sub frame yaw rotation modal shape (Experimental modal analysis)

3. The Effect of Vehicle Modification

3.1. Modal Sensitivity and Modal Design

Using the methods above, the major causes of each weak area were discovered, but in order to specify effective countermeasure locations and methods, it is necessary to identify a highly sensitive area.

In this point in the project, attempts were made to specify areas where countermeasures were required, using modal sensitivity analysis. An example of sensitivity analysis is given in Fig. 9.

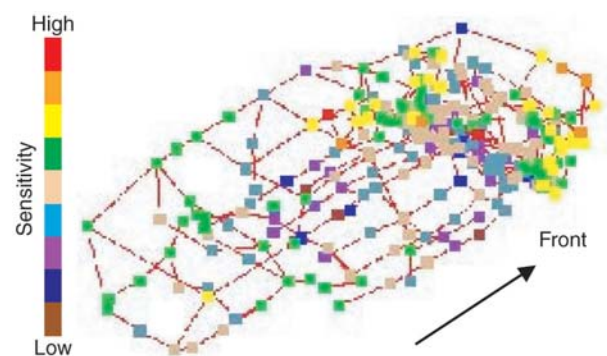


Fig. 9 Modal sensitivity

Modal sensitivity analysis is a method of calculating how much modal natural frequency and amplitude levels will change when the mass and rigidity of the part concerned changes. It is effective in identifying the areas for highly effective countermeasures.

In addition, because it is possible to add an imaginary stiffener regardless of the actual construction, it is also possible to study new constructions.

Next, use was made of a modal-modified calculation method for simulating the effects of changes to mass and rigidity. With modal-modified calculations, mass and rigidity changes are made to the modal model, and the modified model created. From natural frequency analysis of this model, it is possible to compare mode natural frequency value and amplitude levels before and after the mass and rigidity changes. Simulations of the level of effect of the countermeasures can be made, as shown in Fig. 10.

Furthermore, through synthesis of the transfer function after mass and rigidity changes, it is also possible to check characteristic changes in frequency areas far from the target mode. An example of synthesis is shown in Fig. 11.

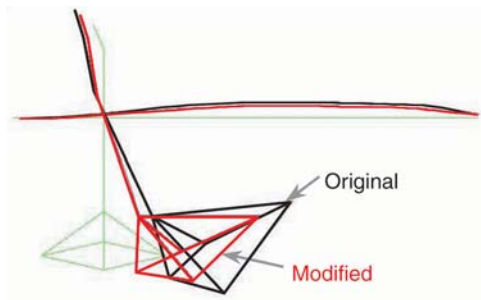


Fig. 10 Modal sensitivity
(Steering wheel lateral acceleration)

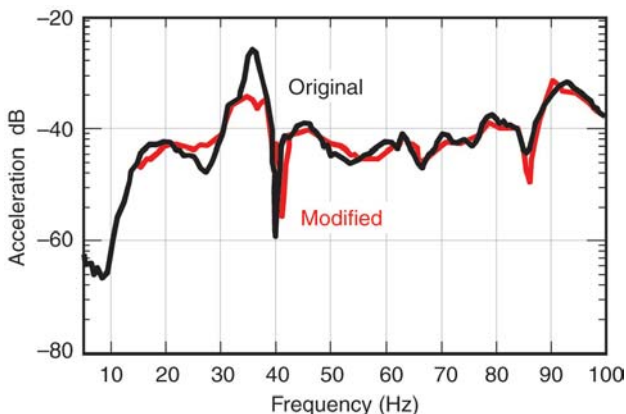


Fig. 11 FRF's synthesis

3.2. Implemented Modifications

Through the aforementioned modal sensitivity analysis and modal modification, effective modification proposals to reduce vibration came to light.

Therefore, an actual car was modified, and a vibration-countermeasure car built. Table 1 gives examples of the modifications made.

Table 1 Example of modification

Target modes	Details of modification
Local modes steering wheel, shaft hanger	· Increase stiffness for steering shaft bending · Reinforce steering hanger right connection to A-pillar
Local modes of rear sub-frame	· Front part of rear sub-frame reinforce + additional beam to body · Rear suspension lower arm connecting point reinforce
2nd body bending mode	· Increase stiffness of rear side frame · Front body (lateral and vertical) reinforce

3.3. Confirmation of Effect of Modifications

Using actual running test measurements of vibrations from the countermeasure car a study was made of the level of achievement against the target level.

The effectiveness of these countermeasures is shown in Fig. 12. There is a reduction in steering vibration level in many frequency bands, and it was confirmed that a dramatic reduction was achieved at 25–40 Hz.

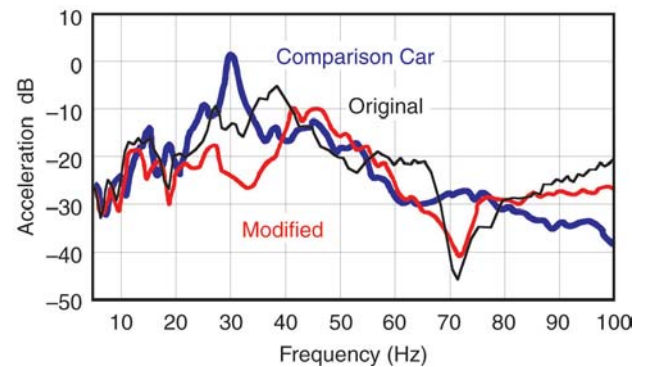


Fig. 12 Result of modification
(Steering wheel vertical acceleration)

4. Simulation Model Creation

For the analysis methods mentioned so far, a test vehicle is necessary. In making the new model development cycle shorter, however, there are cases where it is necessary to simulate vibration levels without a test vehicle.

Therefore, the next step was to create a full vehicle finite element model for simulation technology. In order to obtain a correlation with experimental modal test data, the objective was to create a model with which it was possible to calculate frequency response function (referred to as FRF hereafter) between the unsprung mass (lower arm) and floor or steering wheel. Multiplying this by the road surface input force determined from an actual running test, it was possible to make a simulation calculation of actual car floor and steering vibrations.

4.1. Flow of Model Creation

The subject parts were suspension, engine, exhaust, trimmed body, and steering system (gear box, hanger, column, wheel). The trimmed body mentioned here is the actual car, excluding suspension, engine and exhaust, and steering system (wheel, column, and gear box).

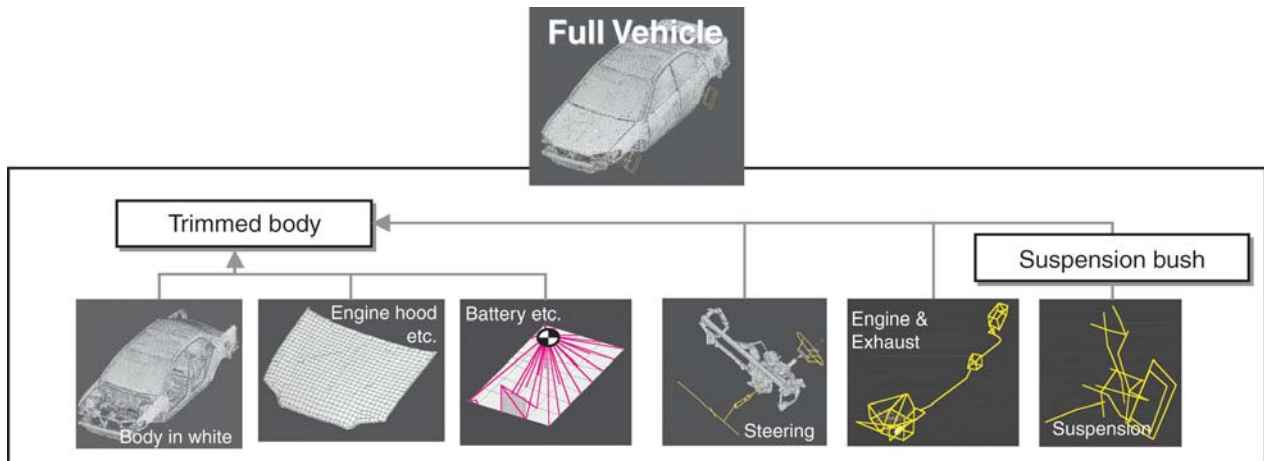


Fig. 13 Flow of full vehicle modeling

First, a finite element model was created for each component, and according to the results of component jig tests, a correlation study was carried out. The modeling method varied for each part, considering the part's level of effect on vibration. Next, the parts were connected, creating a full vehicle finite element model.

The method was to have 3 or 4 people working on model creation simultaneously, and to study the causes of problems in the finite element model so it could be carried out at an early stage. The flow of model creation is shown in Fig. 13.

4.2. Trimmed Body

The trimmed body finite element model used was the one created for acoustic analysis. This trimmed body model was created from the model of the body-in-white, and among the trimmed parts, parts that affect accuracy, such as the engine hood, are modeled in detail. Parts such as the battery which are considered to have little effect on accuracy are modeled as concentrate masses. The sub-frame is also modeled in detail. The final model is compared with the body-in-white and trimmed body test models, making a correlation study. Suspension attaching point local rigidity is very important here, so a correlation study is carried out between attaching point inertance and the index.

4.3. Engine and Exhaust

Because the engine has no resonance points at 200 Hz or less according to the excitation results, it is shown as a concentrated mass, given mass and inertia moment (6 direction) applied.

As for the exhaust, modeling is necessary because of the large number of modes at 100 Hz and less. However, the exhaust pipe is expressed as a BAR element, while the catalyst unit and muffler are a concentrated mass. From the results of a correlation study with experimental modal analysis results, it was learned that it is necessary to have detailed modeling of the flexible joint, and it was expressed as a SHELL element. The model created is shown in Fig. 14.

4.4. Suspension

For the suspension, component experimental modal analysis tests were carried out, and the modal parameters obtained were converted into a direct matrix input force

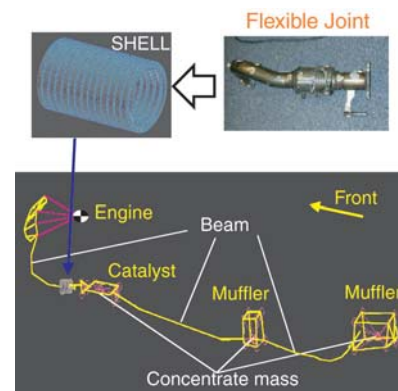


Fig. 14 Engine and Exhaust model

format (DMIG format), thereby creating the finite element model. The advantage of this method is that the suspension, which is strongly non-linear making finite element modeling difficult, can be expressed as a very small model of 74 nodes. Furthermore, data obtained from the experimental modal analysis is input directly, making it highly accurate. On the other hand, the disadvantage is that once the model is created, it is not possible to have a design change. In order to carry out component testing, at this time, the suspension was attached to a sub-frame firmly fixed to a concrete jig, and excitation testing carried out. The created model is connected to the sub-frame through a suspension bush. The dynamic elasticity modulus measured on the component was used as a bush characteristic. The model created is shown in Fig. 15.

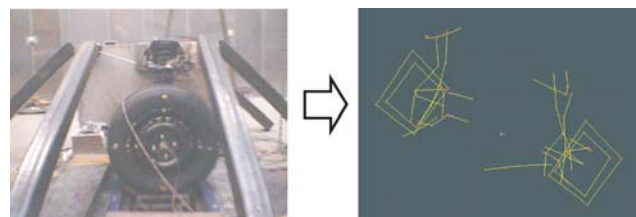


Fig. 15 Testing of suspension and FE-model

4.5. Steering System

In order to reproduce steering vibration accurately, detailed modeling of the hanger and column was carried out, as SHELL elements. However, the gear box and wheel were simply modeled as BAR elements. A correlation study was carried out between them and the results of the excitation test. The model created is shown in Fig. 16.

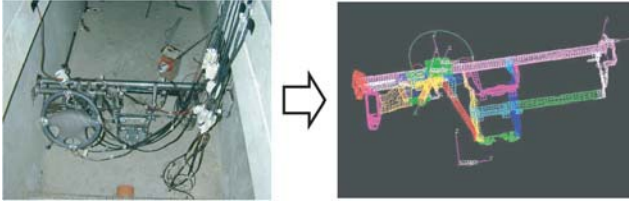


Fig. 16 Steering system

4.6. Full Vehicle Modeling

By linking together all the sub-models, a full vehicle model was created, as shown in Fig. 17.

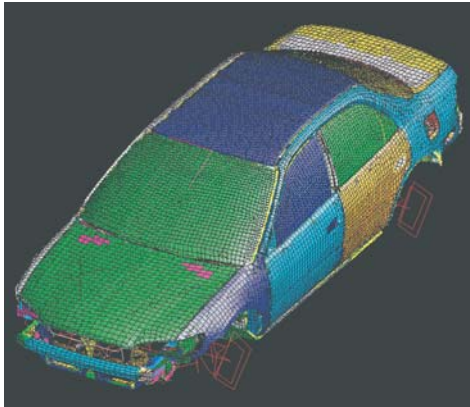


Fig. 17 Full vehicle FE-model

The floor and steering responses (3 directional) against suspension input force using the prepared model were compared with actual car measurement results, and a correlation study made. Here, accuracy was improved mainly by changing the rigidity of connections between sub-models.

The accuracy of the final finished model appears in the FRF between unsprung mass and steering or floor, as shown in Figs. 18 and 19.

In the 10–90 Hz frequency band, the results of a comparison with actual car measurement data showed that the accuracy of the finite element model created in this project was sufficient to be used in simulating vibration levels.

5. Study of Construction using a Modification Simulation

5.1. Outline

Using the full vehicle model for vibration analysis as explained in Section 4, studies were made of proposals for modifications to reduce vibration to the target levels, and the effectiveness of the proposals were reviewed. First, areas that

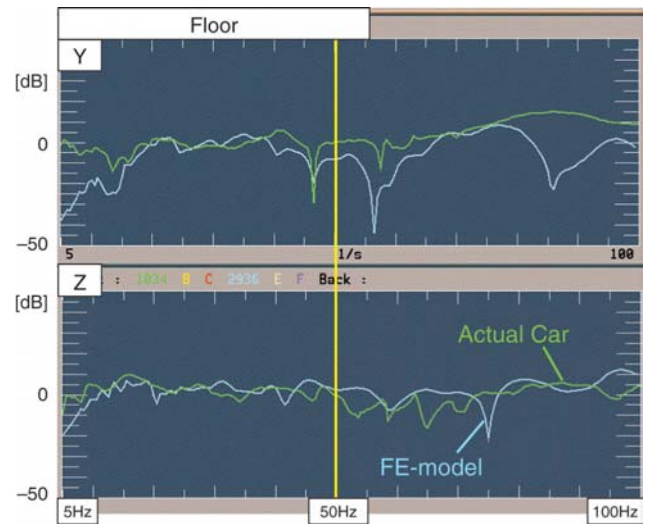


Fig. 18 Final synthesis with full vehicle model

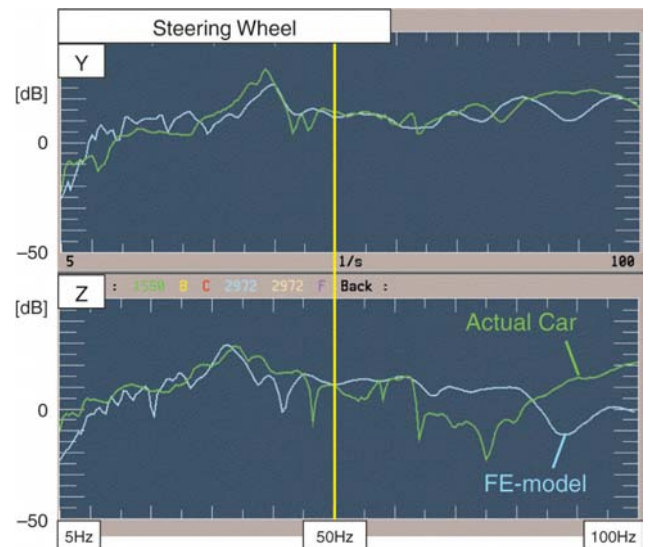


Fig. 19 Final synthesis with full vehicle model

should be modified were identified using sensitivity analysis. Next, simple modifications were made, and the benefits simulated.

Through these studies, it was possible to carry out modifications that effectively reduce vibration levels.

5.2. Specification of Areas for Modification

In order to make effective modification, the following three methods were used.

(1) Finding the Dominant Modes

This is a method that discovers the dominant modes that greatly affect the evaluation index, and can be used to determine the areas that must be qualitatively strengthened. Figure 20 shows the steering wheel lateral bending mode shape, which greatly affects one evaluation index.

(2) Strain Energy

This method is used in order to determine the areas that should be quantitatively modified. The strain energy density of all the dominant modes selected in (1) is totaled, and modification areas specified. Figure 21 shows distribution of strain energy density totals for each group. Here, strain

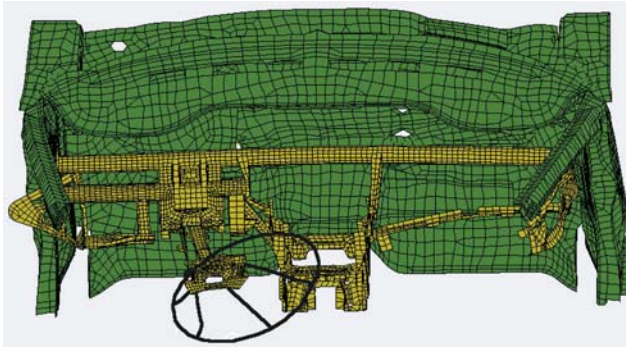


Fig. 20 Steering wheel lateral bending mode

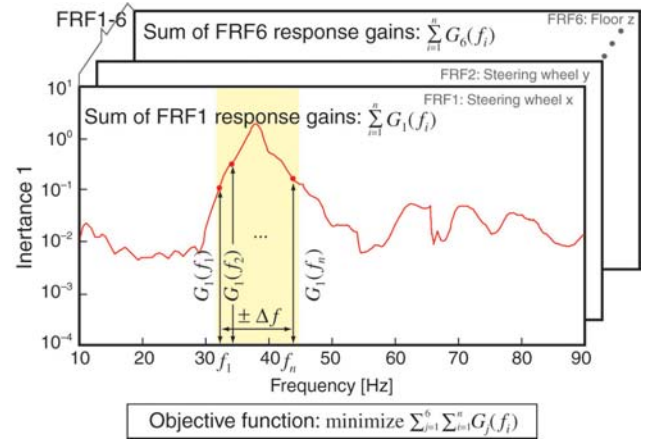


Fig. 22 Objective function of sensitivity analysis

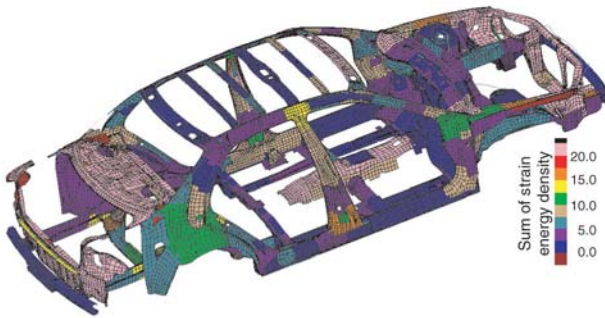


Fig. 21 Strain energy density of body shells

energy density was calculated with each part comprising the body-in-white as one group. With this method, groups, that is parts, that would give a high modification benefit were specified, according to the magnitude of the strain.

(3) Frequency Response Function (FRF) Sensitivity

What follows is an explanation of the method using FRF sensitivity as an actual evaluation function. In sensitivity calculations, the normal finite element method is used. Table 2 gives the objective function, constraint conditions, and design variance for sensitivity analysis. In objective function, the FRF level of a total of the 6 directions — floor 3 directions and steering 3 directions — according to suspension input force is used. As shown in Fig. 22, at each FRF, the focus frequency $\pm \Delta f$ [Hz] response gain sum is determined, and minimizing the sum of FRF for floor and steering three directions is the objective function.

Table 2 Calculation conditions of sensitivity analysis

Objective function	minimize	$\sum_{i=1}^n \sum_{j=1}^6 G_j(f_i)$
Constraint condition	subject to	$W_{opt} \leq W_{init}$
Design variance	shell thickness	

$G_j(f_i)$: FRFj Response gain at f_i ($i = 1, 2, \dots, n$; $j = 1, 2, \dots, 6$)

n : Number of evaluation frequencies

W_{opt} : Optimized weight

W_{init} : Initial weight

The FRF sensitivity obtained through these calculations is shown in Fig. 23. Here, as well as in (2), sensitivity was calculated with each part comprising the body-in-white as one group. With this method, sensitivity concerning direct FRF is calculated, and areas that can be very effectively modified are effectively specified, even more than when using strain energy.

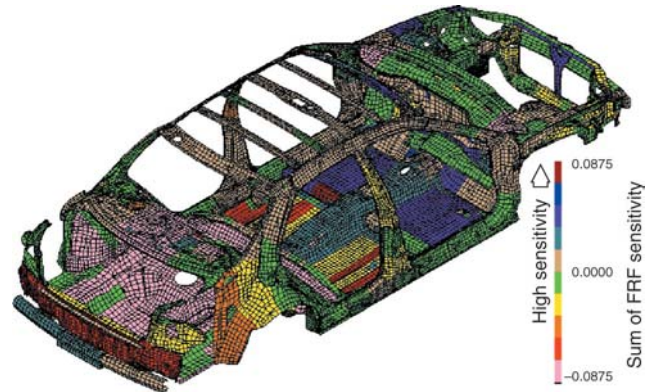


Fig. 23 FRF sensitivities of body shells

5.3. Results of Simulating the Effect of Modification Proposals

The results of simulated effects from the modifications carried out using the methods described in the previous paragraphs are given below. Figure 24 shows examples of body modifications and Fig. 25 shows examples of steering system modifications, with their effects on the evaluation index. From these diagrams, it can be seen that it is possible to reduce vibration by about 10–30% with each modification. Figures 26 and 27 shows the floor 2-direction and steering wheel 2-direction actual running test vibration level simulation results when these modifications are carried out. From these results, by making these modifications, it is possible to simulate a reduction in the 70–90 Hz range and 35–40 Hz steering vibration level.

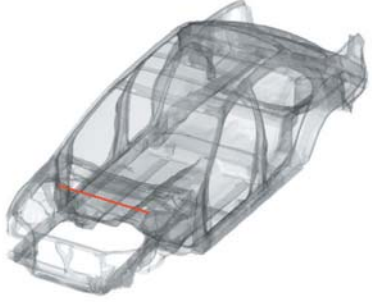
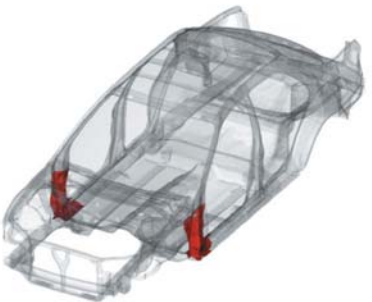
Countermeasure		Effect
Lateral stiffness between front damper base		Floor y, z: -20%
A-pillar lower and side sill connection		Floor Y, Z: -10% Steering wheel x, y, z: -10%
Thickness x 2		

Fig. 24 Examples of body refinement for floor and steering wheel response

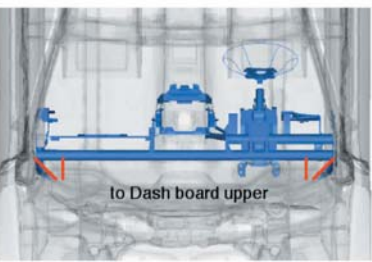
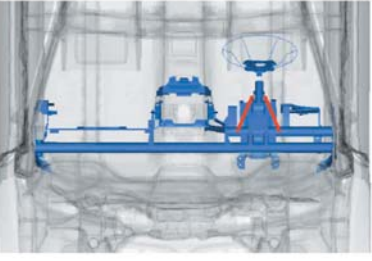
Countermeasure		Effect
Connection from steering hanger to body		Floor y, z: -25% Steering wheel x, z: -30%
Connection from steering outer shaft to hanger		Steering wheel y: -25%

Fig. 25 Examples of steering refinement for floor and steering wheel response

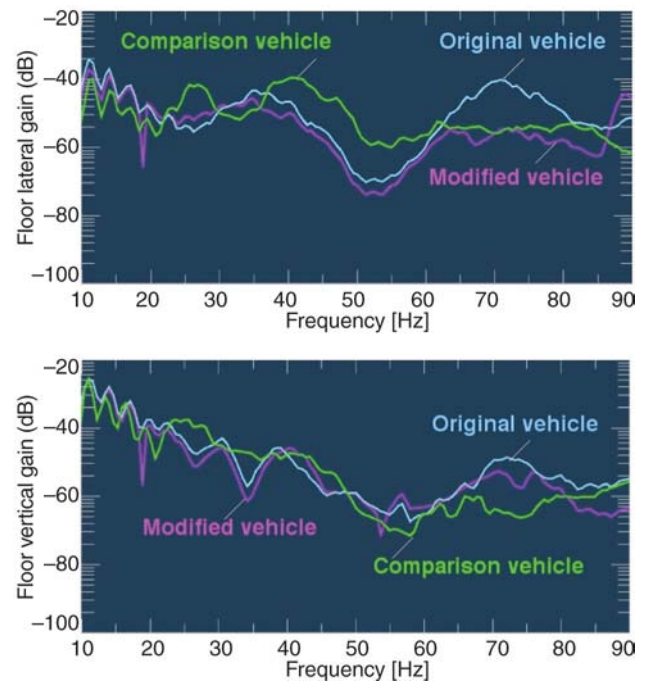


Fig. 26 Floor response gain of comparison, original and modified vehicles

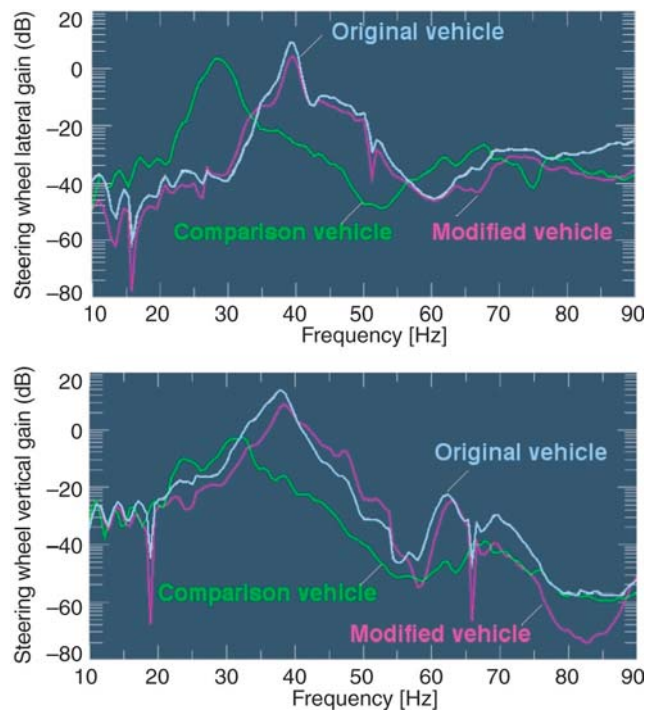


Fig. 27 Steering wheel response gain of comparison, original and modified vehicles

5.4. Results of Measuring the Effect of Modification Proposals

Based on the countermeasures obtained from the calculation results, actual car modifications were carried out, and the effects checked. The floor vertical vibration level is shown in Fig. 28, and steering wheel vertical vibration level in Fig. 29.

The improvement effect in the 70–90 Hz range and in the 35–40 Hz steering response are the same as the calculation results, and there was a reduction in vibration level.

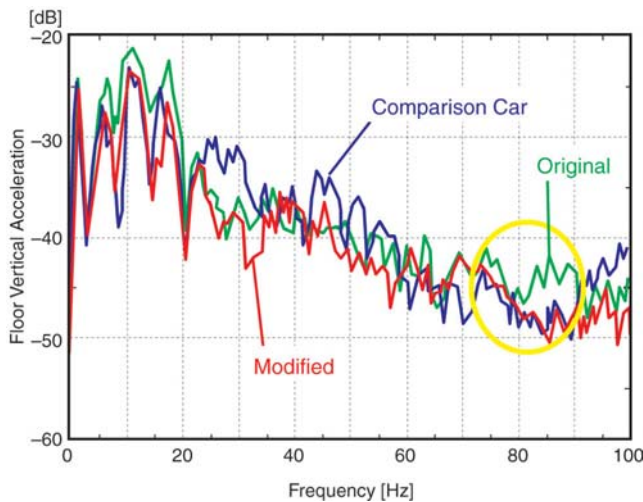


Fig. 28 Floor vertical acceleration

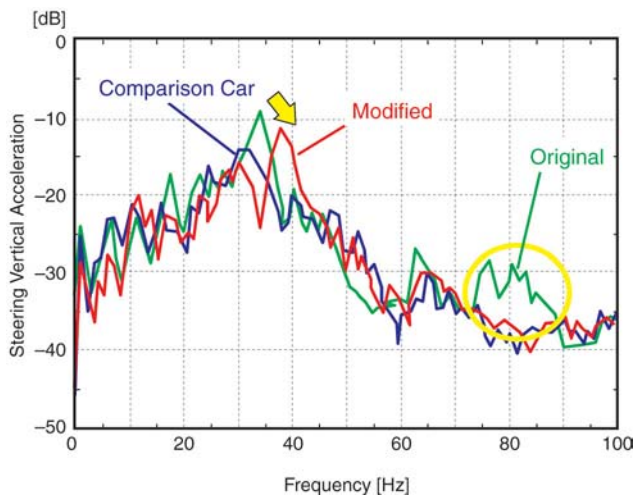


Fig. 29 Steering vertical acceleration

6. Summary

Through this research project, the following achievements were made, in the analysis of floor and steering vibration.

- (1) Full vehicle analysis was tackled, using both experimental methods and simulation technology. It was possible to simulate modifications that address the major causes of vibration.
- (2) Modifications were made according to the simulation results, and it was confirmed that the vibration reduction benefit tendency was the same as the simulations.

In addition, in this full vehicle analysis, experimental methods and simulation technology were used, and once again the advantages of each were brought to light.

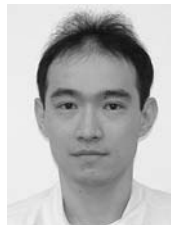
- Advantages of experimental methods
 - It is possible to accurately check vibration levels.
 - It is possible to accurately understand the actual vibration generating mechanisms.
- Advantages of simulation technology
 - Vibration level simulations and mechanism analysis can be carried out without an actual car.
 - It is possible to study constructions that are impossible with an actual car.

In reality, a full understanding of the actual phenomenon was gained through experimental methods; mechanism analysis and study of modification proposals was carried out through both experimental methods and simulation technology. Then, the effects of the modifications were checked through experimental means. This type of development — a fusion of experimental methods and simulation technologies (hybrid development) — is very advanced in terms of the amount of time needed for analysis, analysis accuracy, and effective countermeasures.

7. Acknowledgements

The authors would like to take this opportunity to offer their heartfelt gratitude to all those at LMS International who generously gave their support and cooperation in this research project.

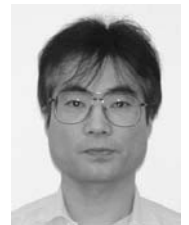
■ Authors ■



Toshiya YOKOI



Kengo OKAMURA



Jiro NINOMIYA



Masaki NAGAHISA