

Analysis of Suspension System Vibration Using Finite Element Model

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ABSTRACT

A technique enabling the level of vibration in a suspension to be predicted has been developed to improve road noise performance. Individual parts were defined and modeled to more accurately reproduce their shape and structure, and a method of assembling the component models was then formulated to enable middle-frequency vibrations to be predicted. In addition, to allow the prediction of vibration in an operating suspension, a method of experimentally identifying the force transmitted to the suspension by the road surface was developed. Measured values and predicted values were compared to verify the accuracy of the prediction technique. In addition, the technique was employed to study the reduction of vibration in an actual suspension, confirming its effectiveness in predicting the effects of structural change.

1. Introduction

With the recent pressure to shorten automotive development periods, there is a growing demand for technologies that will enable the prediction of the performance of completed vehicles at the design stage, before prototypes are built. Some examples of the application of such systems for the analysis of collision performance and of the dynamic characteristics of automobile bodies in frequencies typical for noise and vibration have been presented^{(1), (2)}. However, there have been few actual examples⁽³⁾ of applications of vibration analysis using finite element (FE) models of complete suspension systems to reduce road noise.

In spite of the fact that tire acoustic cavity noise and resonance of the complete suspension system often couple and cause issues in the middle-frequency band (200 – 300 Hz), it is still common to conduct vibration analyses of suspension components in isolation from each other⁽⁴⁾; one almost never sees examples of analyses of complete suspension systems.

In this research, a technique using FE models of the suspension components was created and used to predict

forces from the suspension on the automobile body in the middle-frequency range. And then a technology was developed for modeling the complete suspension system and employed as a method for reducing vibration.

2. Outline of Technologies for Predicting Suspension Vibrations

2.1. Approach to Modeling

The most important consideration for efficient creation of a suspension system model is how to define the components and their interconnections in the target frequency range. For example, it has been shown possible to make analyses in the low-frequency “drumming” range using comparatively small FE models (Fig. 1). However, faithful reproduction of mass and of the distribution of stiffness is essential to the accuracy of models for analysis at middle-range frequencies. This research has resulted in the development of a method for predicting middle-frequency vibration in a suspension with a large elastic FE model which reproduces the 3-dimensional shapes of components of the complete suspension system (Fig. 2).

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2.2. Part Modeling Technology

In modeling the components, top priority was given to faithful reproduction of the shape, mass and stiffness of the originals. The models of each part were defined as shown in Table 1.

Table 1 Elements list of FE model

Component	Parts	FE model	Component	Parts	FE model
Damper	Rod	CBAR, CSHELL, CONM, CELAS, CDAMP	Arm	Arm Collar Bracket	CBAR, CSHELL, CTETRA, CONM
	Outer pipe		Spring	Coil spring	CBEND, CELAS
	Inner pipe		Bush	Rubber	CONM, CELAS
	Rod guide		Ball joint	Stud	CBAR, CTETRA, CONM, CELAS
Drum brake	Valve	CSHELL, CTETRA, CONM, CELAS	Stabilizer	Rink ball	CBAR, CSHELL, CTETRA, CONM, CELAS
	Oil		Subframe	Holder bush	CSHELL, CTETRA, CONM
	Gas				
	Contact area				
Hub & bearing & spindle	Hub Bearing Spindle	CBAR, CSHELL, CTETRA, CELAS			
Wheel	Rim Disk	CSHELL, CTETRA			

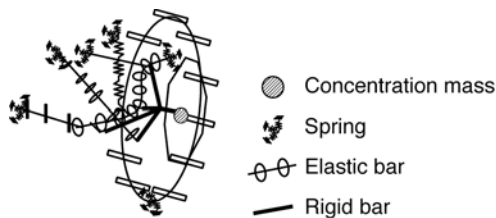


Fig. 1 Low frequency FE model



Fig. 2 Mid frequency FE model

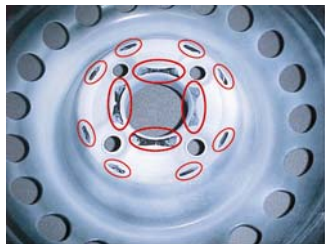


Fig. 3 Contact area of wheel disk and hub

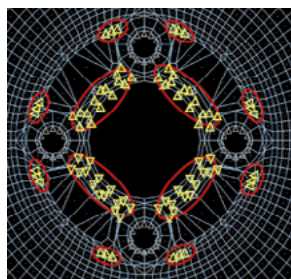


Fig. 4 Connection area of wheel disk and hub in model

2.3. Part Assembly Technology

For bushings and joints at the connections between components, the dynamic spring characteristics, were determined experimentally, and then modeled as 6-degree of freedom (DOF) scalar springs. For the other connections where direct characteristics were difficult to determine, or contact area locations, the joint characteristics were determined by inserting provisional spring elements to the model and updating it until the natural frequencies and mode shapes obtained from the calculation matched those of the experimental modal analysis.

The accuracy of the model in the surface contact locations was ensured by, first, precise measurements of dimensions and locations of the static contact areas in an actual part, as shown in the example of a wheel disk tightening in Fig. 3. The connection areas of the model were then more accurately reproduced to match the characteristics of the contact areas in the actual part, as shown in Fig. 4. Finally, spring elements were placed between the areas to connect them.

2.4. Technologies for Identifying Exciting Forces

It is essential to identify the excitation forces from the road surface on the suspension during operating conditions, in order to obtain the best possible accuracy in predictions of suspension behavior.

Examples of applying road surface displacement and the exciting forces on the tire contact patch have been described^{(5), (6)}, but since either more accurate models of tires or complicated analytical procedures must be employed, the methods described above cannot be considered practical for application in automobile development. That is why assigning the experimentally determined exciting forces to the driving points on the wheel disk, a method which requires no models of the tire, was used in this study. In the method, the suspension vibration in actual operating conditions during driving on the road surface generating road noise was measured and the frequency response function at the same reference point was used when the wheel was excited. Thus, the exciting forces on the suspension exerted by the road surface were identified using the inverse matrix solution by the least-squares approximation shown in Eq. (1).

$$\{F_j\} = [H_{ij}]^{-1} \{x_i\} \quad (1)$$

F_j : Identified exciting force vector at selected driving point

H_{ij} : Matrix defining frequency response function from driving point to reference point

x_i : Response vector to vibration at reference point during operation

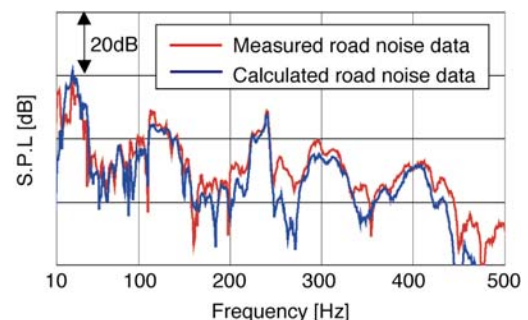


Fig. 5 Precision verification of force identification

Each driving point was assumed to be subject to 3-axis translational forces and was offset so that two moment components could also be expressed, except in the direction of tire rotation.

Figure 5 shows the results obtained when the composite sound pressure levels (SPL) calculated according to the identified exciting forces were compared with the measured cabin SPL due to road noise. It is clear that the actual SPL was well reproduced up to around 450 Hz.

2.5. Correlation with an Actual Suspension System

Figure 6 presents a comparison of the calculated and measured frequency response functions (FRF) of the suspension when the wheels are excited, and Fig. 7, the modal assurance criteria (MAC) values, which are indices of correlation between the mode shapes. The FRF were nearly identical up to about 300 Hz, and the MAC values also matched well. The present prediction technique can clearly be applied for analysis of the frequency band of tire acoustic cavity noise (200 – 300 Hz).

3. Example of Application in Prediction and Analysis of Suspension Vibrations

This section describes examples in which this technique was employed to investigate ways to upgrade performance of the rear suspension in mass-produced automobiles.

3.1. Results of Analysis of Modes of Interest

Analysis of vibratory frequency response characteristics of a suspension in the tire acoustic cavity noise band was

carried out using a FE model; as visible in Figs. 8 and 9, the main mode, coupled with the tire acoustic cavity resonance, was the mode in which the wheel disk was bending and simultaneously the hub and knuckle was bending 180° out of phase (camber mode).

3.2. Improvement of Performance in Individual Parts of Suspension

Next, the individual parts making up the suspension were investigated for ways to improve performance under excitation at the camber mode natural frequency. Focus was placed on the wheel disk, the component that absorbs the most strain energy in the camber mode. The goal was non-coupling of this component with tire acoustic cavity noise. A stiffer version of the disk was designed and vibratory frequency response characteristics of suspension were calculated (Fig. 10).

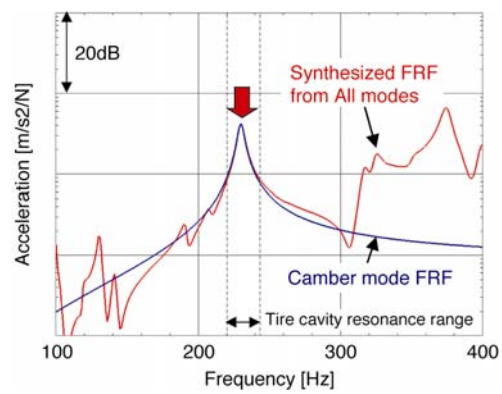


Fig. 8 Mode contribution of target point FRF

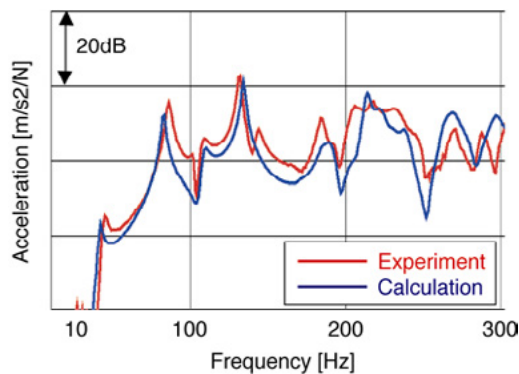


Fig. 6 Precision verification of FRF

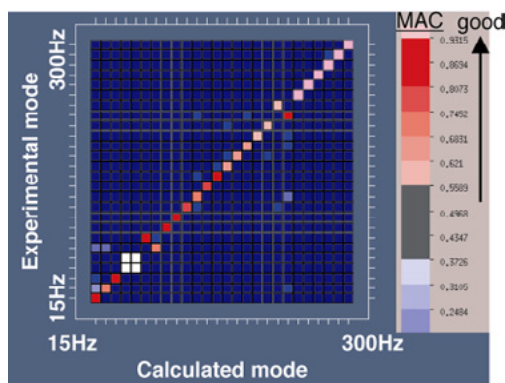


Fig. 7 Precision verification of MAC

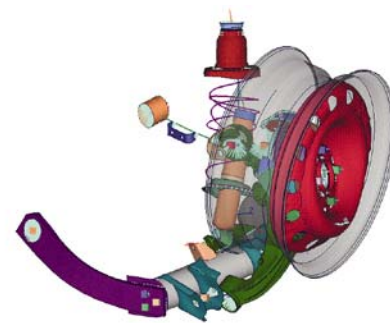


Fig. 9 Mode shape of Camber mode

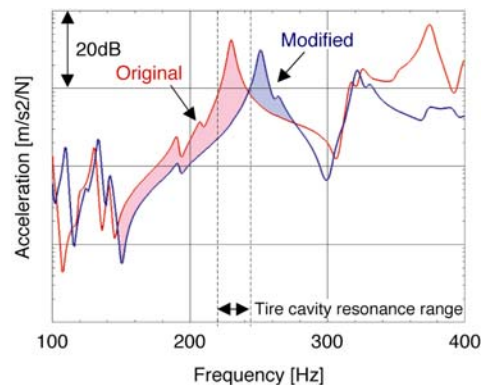


Fig. 10 Improvement of camber mode by increased wheel disk stiffness

Vibration level was substantially reduced in the tire acoustic cavity noise band, but since examination of the entire medium-frequency band showed that the peak in the camber mode response spectrum remained nearby, simply shifted to a slightly higher frequency, it was clear that this approach had not completely succeeded.

3.3. Improvement of Performance in Complete Suspension Systems

An analysis of the contributions of stiffness and mass of every component of the suspension system was carried out with the objectives of reducing mode amplitude and re-adjusting the camber mode peak to lower levels. The analysis showed that the highest contributions were by the knuckle stiffness and the mass of the trailing arm end (Fig. 11).

One constraint was to avoid increases of the weight of the overall suspension system. As a countermeasure, a structural change was tried, with a move of the shock absorber location

from the middle of the trailing arm to the end, to allow modification of the knuckle geometry and shape and to increase the mass of the trailing arm end (Fig. 12).

The resulting changes in the suspension FRF were estimated. The conversion to the mode shape in which mass efficiency was improved and the increase in modal mass effected a large reduction in the amplitude of the camber mode. Dramatic reductions in vibration acting on the suspension are expected from this result (Fig. 13).

3.4. Results of Verification in Actual Assembly

A prototype was constructed to verify the validity of the prediction for improved performance. A knuckle with modified geometry was installed and a weight was attached to the trailing arm end corresponding to the mass of the shock absorber. Figure 14 shows the predicted results, and Fig. 15, the measured results. These confirm the effectiveness of this method. It was also found that the new suspension system offers reduced road noise in the cabin (Fig. 16).

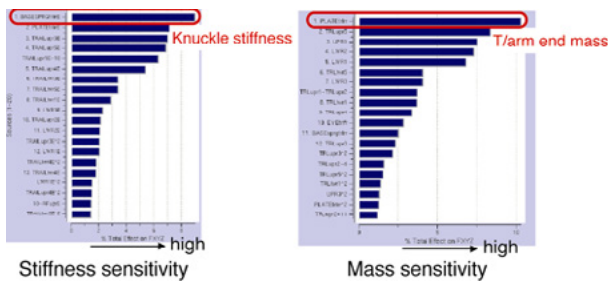


Fig. 11 Sensitivity of suspension components to changes in stiffness and mass



Fig. 12 Case of structure change for improvement of road noise

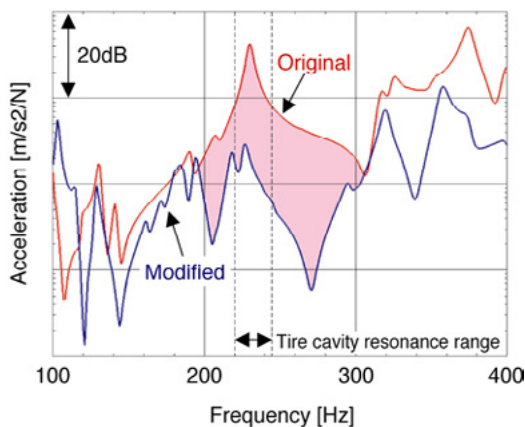


Fig. 13 Improvement of camber mode by optimization of stiffness and modal mass

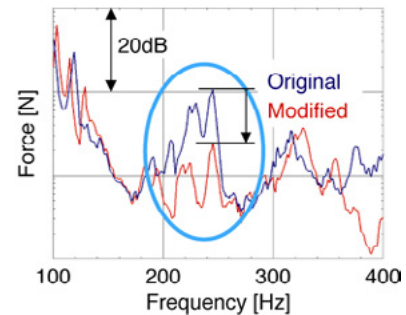


Fig. 14 Estimate of improvement using FE model

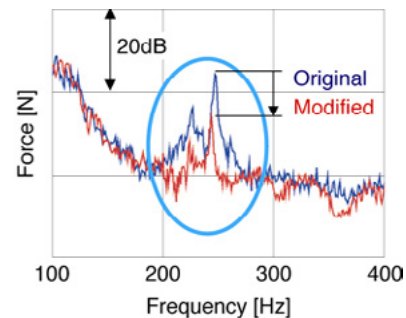


Fig. 15 Verification of improvement by experiment

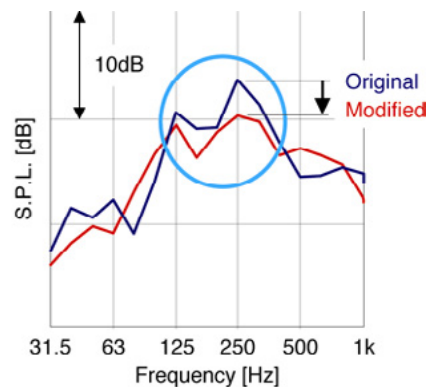


Fig. 16 Improvement of road noise by optimization of modal mass

4. Observations on Modal Masses

It has been shown that vibration forces which contribute greatly to road noise can be reduced at its driving points by modifying the mass distribution to increase the modal mass of a selected mode. Next, some observations of the differences between suspension types in vibratory frequency response characteristics were made, with reference to modal mass.

4.1. Suspension Types and Modal Masses

The sums of suspension vibratory forces in different suspension types were analyzed for comparison, as shown in Fig. 17. It is clear in the figure that the torsion-beam type offered much lower absolute spectral line values in the middle-range frequencies. The frequency response of the torsion-beam suspension was superior to the multi-link suspension.

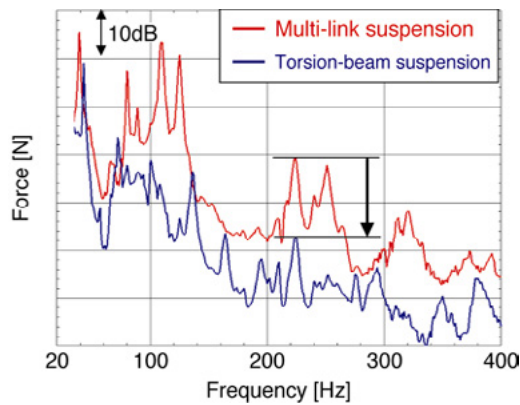


Fig. 17 Comparison of different suspension systems as calculated by FE model

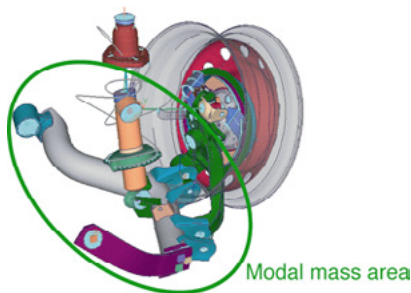


Fig. 18 Modal mass area of multi-link suspension in camber mode

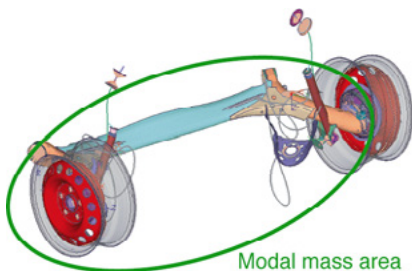


Fig. 19 Modal mass area of torsion-beam suspension in camber mode

The camber mode was the contributory mode in the region for tire acoustic cavity noise in both systems. The mode shapes in the systems were compared; Fig. 18 shows that only part of the system for one wheel in each pair vibrates in the rear multi-link type; this makes for a low modal mass, and in turn, high mode amplitude. In contrast, vibration occurs over a large portion of both wheels in the pair in the rear torsion-beam type (Fig. 19); vibratory forces are exerted on both wheel masses, so the modal mass is high, causing low mode amplitude.

4.2. Increase of Modal Mass in Multi-link Suspension

In order to increase the modal mass in the multi-link suspension, whose modal mass would otherwise be excessively low, the mass distribution in one side wheel of the suspension must be optimized. Even after this re-design, however, the absolute value of its modal mass is still nowhere near that of the torsion beam system in which both wheels can act as masses. The present method was therefore used to predict the behavior of a sub-frame floating version of the multi-link system, which incorporated the masses of both wheels and a sub-frame. The results are presented in Fig. 20. The suspension response was comparable to that of a torsion-beam system.

5. Conclusion

The following results were obtained using this technology for predicting suspension system vibration.

- (1) The development of this technology for finite element modeling of suspensions and experimental identification of suspension excitation forces has provided a method for reproducing suspension vibration comparable to the actual automobile operation.
- (2) This technology also allows analysis of complete suspension systems and prediction of improvements in performance. Its effectiveness was verified in tests of an actual assembly.
- (3) It was shown, both in predictions and tests with an actual assembly that optimizing the modal mass distribution of a suspension system is an effective method for reducing suspension vibration, and that vibratory forces can be reduced at the suspension attachment points.

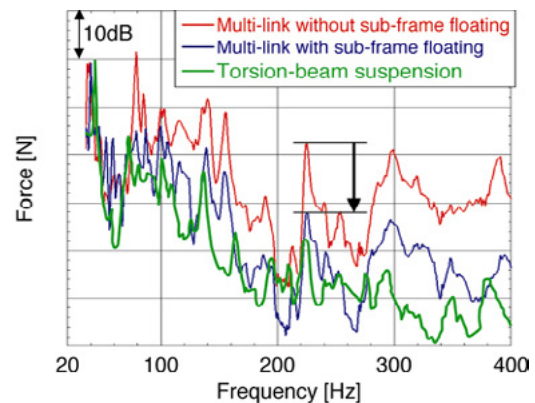


Fig. 20 Comparison between sub-frame floating and torsion-beam suspension as calculated by FE model

- (4) The rear torsion-beam suspension was shown to be superior to the rear multi-link type, from the viewpoint of the modal mass and of its effect on the frequency response function of each type of suspension.

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