A Study on the Body Attachment Stiffness for the Road Noise

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The ride and noise characteristics of a vehicle are significantly affected by the vibration transferred to the body through the chassis mounting points in the engine and suspension. It is known that body attachment stiffness is an important factor of idle noise and road noise for NVH performance improvement. The body attachment stiffness serves as a route design aimed at isolating the vibration generated inside the car due to the exciting force of the engine or road. The test result of the body attachment stiffness is shown in the FRF curve data ; the stiffness level and sensitive frequency band are recorded by the data distribution. The stiffness data is used for analyzing the parts that fail to meet the target stiffness at a pertinent frequency band. The analysis shows that the target frequency band is between 200 and 500 Hz. As a result of the comparison in a mounted suspension, the analysis data is comparable to the test data. From these results, there is a general agreement between the predicted and measured responses. This procedure makes it possible to find the weak points before a proto car is produced, and to suggest proper design guidelines in order to improve the stiffness of the body structure.

Key Words : Body Attachment Stiffness, Bush Dynamic Stiffness, Vibration Isolation, NVH (Noise, Vibration and Harshness)

1. Introduction

The development of a high stiffness body is required to improve the performance of crashworthiness, NVH, and durability. However, a high stiffness body demands a heavy weight and a high cost due to the increase of panel thickness and the use of reinforcing members. In order to achieve the goal of reducing the fuel consumption of a vehicle, design optimization is needed to satisfy both development objectives of a high stiffness body and a fuel-efficient car. (Ki-Chang and Chan-Mook, 2004)

Specifications for the vibration performance of a global body structure are established through two steps. The first step is a frequency domain "mode map" constructed to minimize the interaction of a vehicle's subsystems. Secondly, computer simulation models are used for guidelines in the initial stage of design. The frequency domain allocation of a vehicle in the level resonant models is a process called "mode mapping". This process considers the input force from the road and the engine to define frequency domain allocation of resonances expected in the system. (Tony and Brian, 2000)

A benchmarking activity was undertaken to comprehend the competitive vehicle's body structure and performance. There are a number of noise and vibration sources that affect the vehi-

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cle's body. Noise and vibration occur because the power delivered through bumpy roads, the engine, and the suspension results in the resonance effect in a broad frequency band.

Vibration arises at low frequency bands of less than 50 Hz through the steering and the floor, and is determined by factors of frame stiffness, including bending, torsion, and lateral vibration mode. Noise pertaining to the body attachment stiffness occurs at high frequency bands of 100 to 600 Hz ; its main factors are the stiffness of the engine, suspension mounting points, and sensitive panel. (Ki-Chang and Suk-Ju, 2000)

The ride and noise characteristics of a vehicle are affected significantly by the vibration transferred to the body through the chassis mounting points in the engine and the suspension. It is known that body attachment stiffness is an important factor for noise reduction. According to the sensitivity of the NVH performance, the body attachment stiffness has the power to influence idle noise and road noise. (LMS and HMC Technical Report, 2001) A high stiffness helps to improve the flexibility of the bushing rate tuning. A development goal of body attachment stiffness is generally suggested to be more than 5~10 multiples of the bush dynamic stiffness. (LMS and HMC Technical Report, 2001)

This paper describes a design optimization process that increases the body attachment stiffness at the vehicle's body mounting points through which the sources of noise, including engine, suspension, and road surfaces, are delivered to the vehicle's body.

As for the body attachment stiffness, a theoretical analysis was conducted on the factors of member section types. As a result of the comparison in a mounted suspension, the analysis data is comparable to the test data if the exact test conditions are reflected with the mass effect, including the accelerometer, rigid block, bolt and weld nut. From these results, there is general agreement between the measured and predicted responses. This enabled us to find the points where the body attachment stiffness is weak and to suggest design modifications to improve these weak parts. These pre-design procedures reduced the number of test vehicles needed and made it possible to reflect the results of designing for road noise reduction.

2. Experimental Procedures

The characteristics of road noise were obtained through a baseline test. The coupling effects between the body structure and the acoustic cavity of a compartment were obtained by means of modal tests for structure and acoustic cavities. The local stiffness of joint areas between the chassis system and the body was determined by measuring the input point inertance. Noise sensitivities of the body joints to operational force were obtained a test for the measurement of the noise transfer functions. (Kang-Ho and Seung-Jin, 2003)

2.1 Road noise

As shown in Fig. 1, three typical parts are considered in the design process. The first part is the exciting source, the second is the transfer path, and the final part is the response. In passenger vehicles, the power train and road loads are major input sources. Tires, suspension, body structure, and cavities are included in the transfer path. The improvement of the response is the final goal of the NVH analysis.

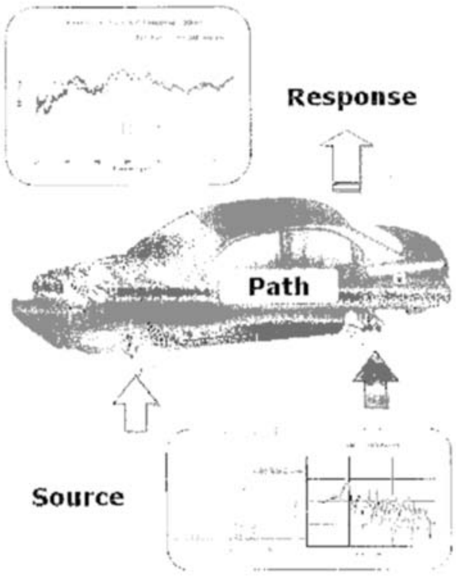


Fig. 1 The typical process of road noise analysis

The road noise load is developed from a road surface profile that is measured either from a test track road surface or from a chassis dynamometer used for evaluating road noise. The profile is converted from height vs. distance to a frequency-based input spectrum based on the vehicle speed used when evaluating road noise. The surface data measured from the chassis roll is shown in Fig. 2.(Ji-Un and Walter, 2003)

Having the road surface data in height vs. distance format permits the conversion to a spectrum of height vs. spatial frequencies. The process of converting the road surface data from height vs. distance is done with a Fast Fourier Transform (FFT).

The approach used to develop the road noise load considers the Power Spectral Density (PSD) and performs a power function regression analysis in a range of useful wave numbers (from 0.001 cycles/mm to 0.010 cycles/mm). An expanded PSD with a regression analysis is shown in Fig. 3. In this graph, the blue curve is the Fast Fourier Transformed curve, and the red curve is the regression of the blue curve.

This spectral representation of road noise is independent of vehicle speed. Therefore, once the road surface has been measured, it can be used for other vehicle simulations and speeds.

MSC.Nastran was used as a solver because it is easy to use the enforced displacement in the frequency domain. Generally, the measured responses remain between the upper and lower bounds of the predicted responses. The averaged road noise response is compared to the measured and predicted responses measured at driver's ear and rear seat position, as shown in Fig. 4. From these results, general agreement, between the measured and predicted responses can be seen.(Ji Un and Walter, 2003)

2.2 Body attachment stiffness

The automotive body structure must have sufficient stiffness at the power train and suspen-

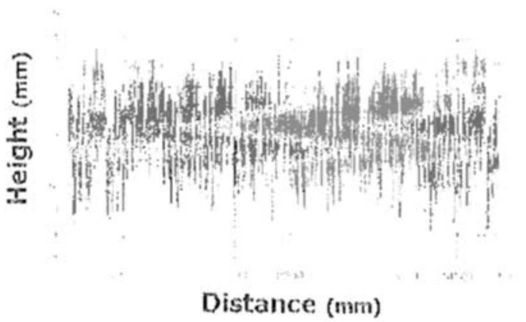


Fig. 2 Measured road surface profile

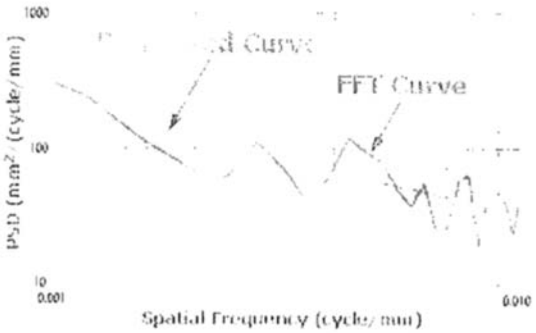
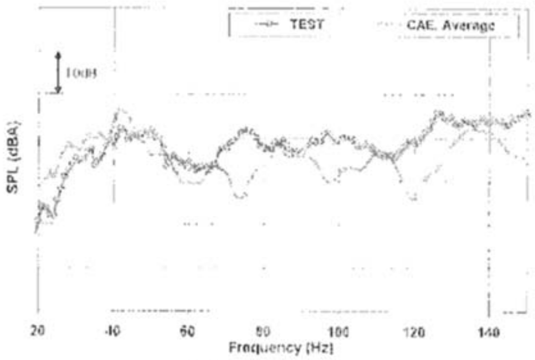
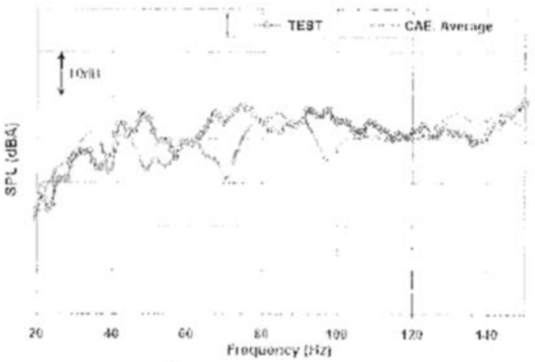


Fig. 3 Road surface PSD



(a) Driver's ear



(b) Rear seat center

Fig. 4 Averaged road noise response

sion attachment points. The body attachment stiffness takes advantage of the isolation provided by bushings. High stiffness helps to improve the flexibility of the bushing rate tuning between the road noise and handling performance. (Ki-Chang and Suk-Ju, 2000)

The body attachment stiffness serves as a route design aimed at isolating the vibration generated inside the vehicle due to the exciting force of the engine or suspension as shown in Fig. 5.

The body attachment stiffness is one of the key NVH targets for most vehicle manufacturers. Generally, a design guide for target stiffness is 5~10 times greater than the bush dynamic stiffness.

According to the sensitivity of the NVH performance, body attachment stiffness is a main factor of idle noise and road noise as shown in Fig. 6, and it has the influence power equal to 30% of idle noise comparison and 40% of road noise comparison. (LMS and HMC Technical Report, 2001)

The object of the test for measuring inertance is to estimate the stiffness of the body joints to which the chassis components are attached. The stiffness of the structure can be estimated by means of its inertance, which is defined as the frequency response functions between the accelerations of structures as responses and the forces as inputs. Particularly, the local stiffness of a structure can be estimated by the input point inertance that makes use of the spectra of acceleration and force measured at identical positions.

The test of the body attachment stiffness was conducted on the major mounting points of a vehicle with the four corners of its body fixed with air ride, as shown in Fig. 7. Excitation was applied in XYZ directions near the points where the stiffness was to be measured. Responses were acquired at either the flat areas near the excitation points or at the center of the bush inner pipe.

The test results of the body attachment stiffness are in the FRF curve data, as shown in Fig. 8. The stiffness level and sensitive frequency band are recorded in the data distribution. The stiffness data is used for the analysis of parts that fail to meet the target stiffness at a pertinent frequency band. The target frequency band is between 200 and 500 Hz.

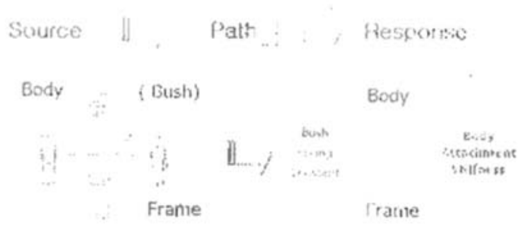


Fig. 5 Schematic diagram of bush mounting parts

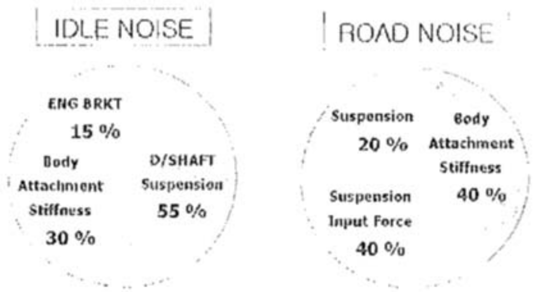
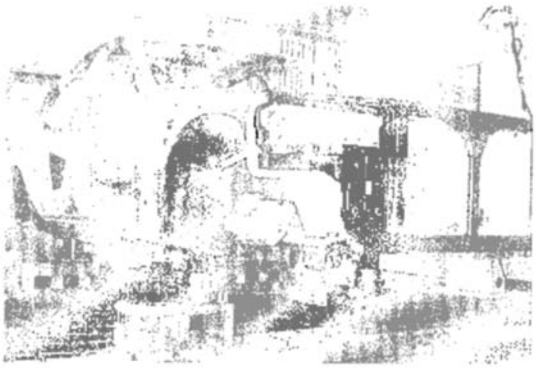


Fig. 6 The sensitivity of NVH performance



(a) Test boundary condition

(b) The point of exciting and response

Fig. 7 Test Condition of body attachment stiffness

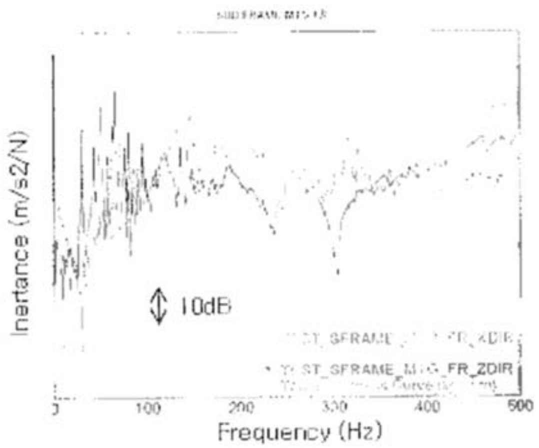


Fig. 8 The test data in mounting position of sub frame

In Fig. 8, the horizontal axis is a frequency band and the vertical axis is the level of acceleration. As a result of the mounting position of the subframe, the X-direction (before and after) and the Z-direction (upper and lower) curves are satisfied with the target stiffness curve. But, the Y-direction (left and right) curve, compared with the target stiffness, is necessary for the stiffness enlargement to improve bush isolation for road noise performance.

3. Analysis Procedures

Fig. 9, shows the body design process of the rear suspension attachment points for road noise performance. After deciding on the suspension type, we obtained the static force in order to find main load direction from the chassis group. After obtaining the result data, the result was in an up-and-down direction from the rear strut and rear cross member attachment parts, while the trailing arm was in a front-to-rear direction. The analysis condition was that the excitation points were at the center of the bush inner pipe and the bottom of the bolt ; the response point was the center of the bush inner pipe.

Fig. 10, shows the procedure of the body attachment stiffness analysis. A detailed mesh and a half trimmed FE model were used to reduce the CPU time in the high frequency range. When

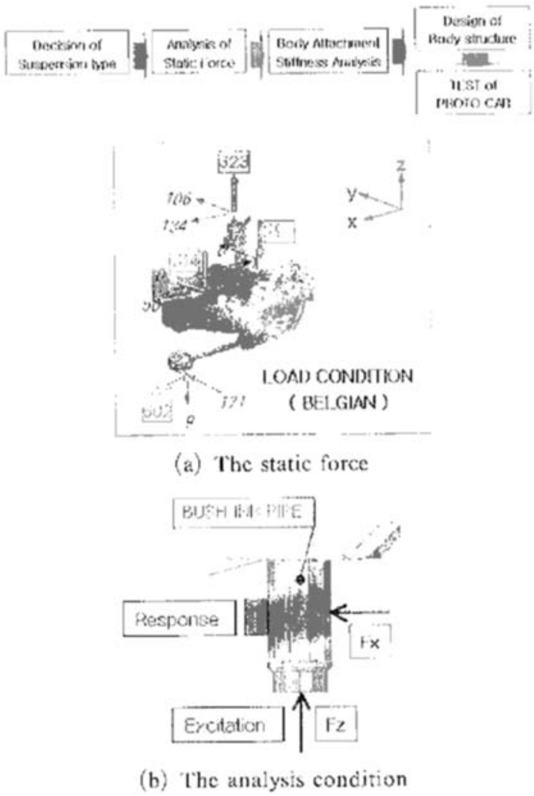


Fig. 9 The design process of the body attachment stiffness

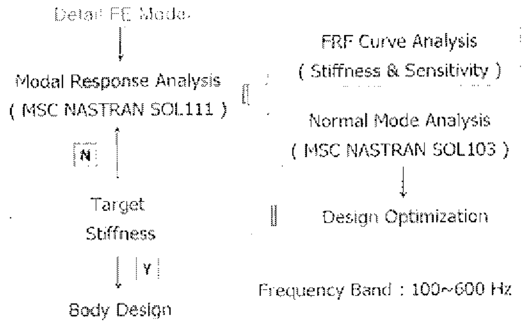


Fig. 10 The procedures of body attachment stiffness analysis

constructing the half model, the trim line must be set after considering the points of analysis ; and the shape and welding status of mounting points should be correctly reflected in the analysis model.

The modal response analysis was used in the analysis of the body attachment stiffness. The stiffness level and sensitive frequency band are recorded in the data distribution. The normal mode analysis was used to analyze parts that failed to meet the target stiffness at a pertinent frequency band. After applying the improvement methods, this paper analyzes the mounting points to see if they meet the target stiffness.

MSC.Nastran SOL 111 calculates the modal values by using modal stiffness and modal mass matrix and it is also known as a modal response analysis. This analysis type is used to locate the parts whose vibration levels are relatively high and to estimate the body mounting stiffness. If the parts fail to meet target stiffness level at production, improvement methods can be found through proper oscillation analysis at the frequencies that are suspected to be problematic.

3.1 Comparison between the full and half model

In this paper, the half model was constructed by considering the boundary conditions and the analysis time. After obtaining the analysis results, both the full and half models showed remarkable similarity in the 100 to 800 Hz frequency band, as shown in Fig. 11. During the analysis, the half model was more effective in terms of the time taken and hard disk space needed.

In the body attachment stiffness analyses conducted on the outer mounting point's Z-direction, excitation was applied in various directions near the points whose stiffness was to be measured. The responses were acquired at the flat areas near the excitation points.

To verify the analysis results according to the boundary conditions, the half model's trim line was constrained to six degrees of freedom to establish a fixed model while setting up a free model with PARAM, AUTOSPC, and YES modes. The results of the analyses in both cases are proved that the analyses are the same. However, if the AUTOSPC mode is not taken into account in the free model, the analyses results differ from each other. AUTOSPC automatically constrains the degree of freedom with a very low stiffness level. If the AUTOSPC takes, it automatically constrains a singularity when there's a singularity problem in the stiffness matrix.

3.2 The influence of weight factors

To reflect the exact test conditions in the analyses, the weight of the test elements, including an

(a) Full model (b) Half model

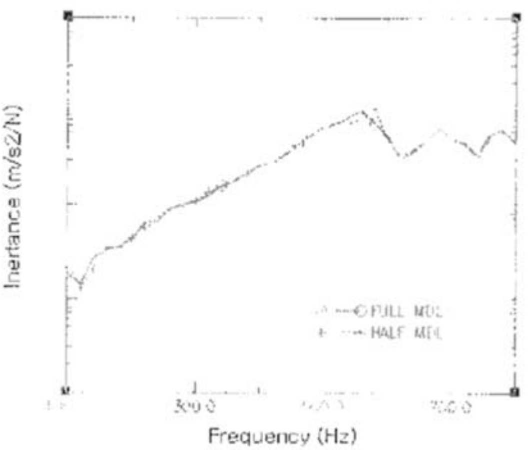
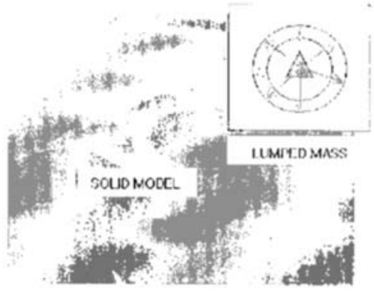


Fig. 11 The comparison between full and half model

NO	MASS FACTORS	MASS (g)
1	RIGID BLOCK	120.5
2	BOLT	32.6
3	WELD NUT	12.1
4	ACCELEROMETER	30.0

(a) Weight factors of test condition



(b) The modeling methods of body attachment parts

Fig. 12 The influence of test weight factors accelerometer, rigid block, bolts, and weld nut, was concentrated in the mounting hole. In this paper, two modeling guides are suggested to reflect the real shape and mass effect of a vehicle as shown in Fig. 12.

As a result of the comparison, we found that the solid model reflected both the weight and the moment of inertia more accurately when compared to the test data when the weight of mass elements was concentrated.

3.3 The influence of section shape

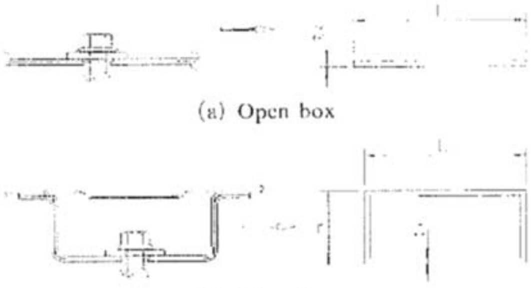
If a body mounting point's section is box-shaped, it is more effective in absorbing loads put onto the point through the body mounting section, and it minimizes the thickness of the panels compared with an open-shaped section. Fig. 13, compares the influence of open and closed sections on the body mounting point in a simplified manner. (Ki-Chang and Suk-Ju, 2000)

In order to secure the same stiffness as a closed section type needs to satisfy the following formula.

$$\frac{1}{12}LH^3-\frac{1}{12}(L-2t_1)(H-2t_1)^3=\frac{1}{12}Lt_2^3$$
$$t_2=\sqrt[3]{H^3-\frac{1}{L}(L-2t_1)(H-2t_1)^3} \tag{1}$$

Table 1 A requirement thickness (t_2) in open section compared with a height (H) in closed section

	Case 1	Case 2	Case 3	Case 4
H (mm)	10	20	30	40
t_2 (mm)	9.8	17.0	24.0	30.2



(a) Open box

(b) Closed box

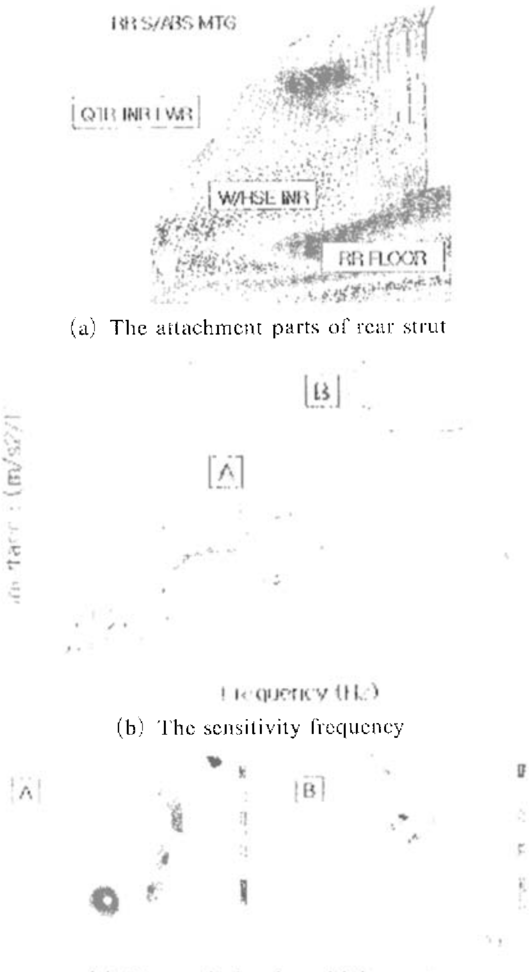
Fig. 13 The shape of open and closed section box

Then, the values of t_2 according to the values of H can be stated by the following.

Here, the panel thickness of the closed section is 3.0 mm. If the height (L) of the closed box section is 30 mm, the panel thickness of the open box section is needs to be 80 mm in order to achieve the same stiffness.

3.4 The application of the design process

To accurately understand weak mounting points and in search of effective improvement methods, application cases were arranged on the rear strut mounting point of non-package tray vehicles during the initial stage of design as shown in Fig. 14.



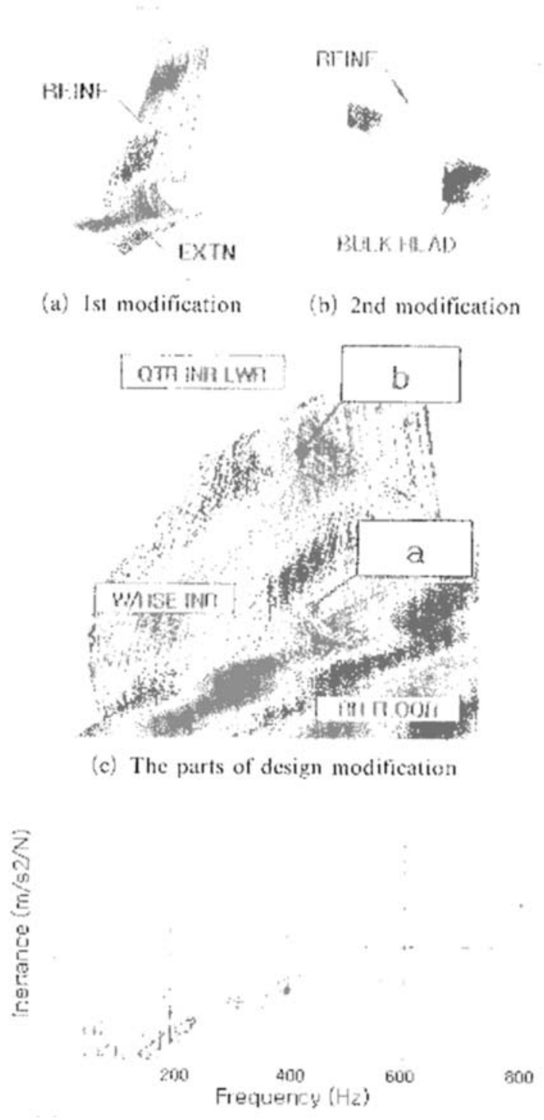
(a) The attachment parts of rear strut

(b) The sensitivity frequency

(c) The analysis of sensitivity parts

Fig. 14 The analysis data at initial stage of design

The rear suspension's strut is mounted on the rear shock absorber housing; a high level of mounting stiffness against road noise is important in this mounting point. After conducting a mounting stiffness analysis on the rear shock absorber housing mounting point, we found that the vibration level was relatively high at two frequencies; between 200 and 400 Hz, and at around 600 Hz.



(a) 1st modification (b) 2nd modification

(c) The parts of design modification

Fig. 15 The improvement according to design modification

The normal mode analysis was used to find the points where the stiffness level was relatively high at given frequency bands; the points where the distortion level was significantly high were analyzed and a revised design was developed.

The front side of the rear wheel housing inner panel was found to be sensitive part at a 200 to 400 Hz frequency band. The sensitivity of this point to vibration is more significant in coupe and 5-door vehicles with no package trays. Therefore, the sensitivity of the part at the junction point with the floor needs to be strengthened.

The upper mounting point of the rear strut was found to be a sensitive point at around 600 Hz. The stiffness of the points near this mounting point needs to be strengthened.

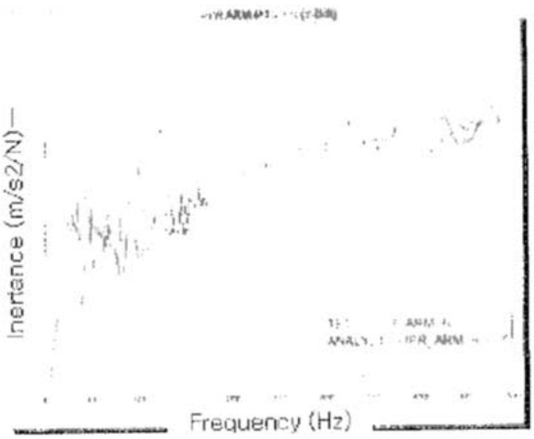
The first design modification was adding a vertical reinforcement in the junction between the rear floor and the rear wheel housing inner panel. As a result, the vibration level of the rear suspension mounting points decreased between the 200 and 400 Hz frequency band; however the peak of 600 Hz was not affected.

The second design modification was to include two bulk heads in the body attachment structure of the rear strut in order to increase the local stiffness level at around 600 Hz. As shown in Fig. 15, the vibration level at around 600 Hz decreased and the frequency around 600 Hz moved to the rear.

4. The Results of Body Attachment Stiffness

Fig. 16 shows the comparison of the test data and the analysis data of the body attachment stiffness at the front and rear suspension mounting points. After the comparison, it was found that the analysis data is similar to the test data.

This analysis procedure makes it possible to find out the weak points before a proto car is assembled, and to propose proper design guidelines in order to improve the stiffness of the body structure for improved road noise performance.



(a) The attachment part of front suspension

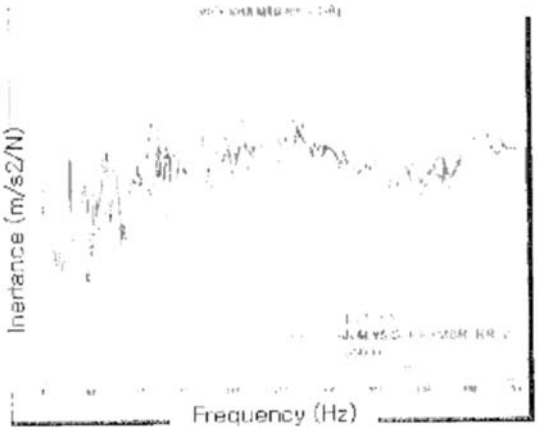


Fig. 16 The comparison of test and analysis data

5. Conclusions

In this paper, we suggested an analysis process for calculating the body attachment stiffness at the suspension mounting points of a vehicle for improved road noise performance. A vehicle with a high stiffness and light weight is a key goal in the refinement of passenger cars to meet customers' requirements of NVH performance and fuel economy.

According to the sensitivity of the NVH performance, body attachment stiffness is a main factor of road noise; and it has the power to influence up to 40% of the total road noise.

We established techniques for the analysis of mounting stiffness on the vehicle body's mounting points, and evaluated the level of accuracy of these analyses through correlating the test results and the analysis results.

Using package tray pass-through structures to increase the body attachment stiffness of the suspension mounting points is expected to minimize road noise by isolating the vibration of the bush mounted unit.

Vehicles with good NVH performance can be designed at the initial design stage and reduce the number of proto vehicles used during development. These improvements can lead to a shortening of the time needed to develop better vehicles.

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