

Study of Method to Reduce Low-frequency Noise by Controlling Panel Vibration Phase

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ABSTRACT

A method of reducing the sound pressure of low frequency noises such as booming and drumming noise was devised by focusing on the phase relation between multiple panel vibrations. Low frequency noise is affected by the acoustic modes, which are determined by the shape and the size of the cabin. Accordingly, an advantageous panel phase relation was clarified for the sound pressure inside the cabin in the low frequency region, and a method of realizing this phase relation by means of the distribution of the panel resonance frequencies was invented.

Also, technology for performing BEM analysis in the low frequency region was established. The sound pressure in the cabin was predicted according to the boundary conditions obtained from the measurement results, and the contribution of each panel was clarified. From these results, the multiple panel resonant frequencies of a completed vehicle were adjusted, and the necessary phase relation for canceling the sound pressure was realized, thus the sound pressure was reduced.

1. Introduction

Due to heightened demands for environmental performance, elements that increase engine and suspension vibration have proliferated in recent years, such as the use of direct injection engines and diesel engines, and expansion of the AT lock-up range to enhance fuel economy. In addition, demands for weight reduction are also increasing, and in many cases, conflict with vibration and noise performance. As a result, development to reduce low-frequency vibration and noise is becoming more challenging.

The acoustic 1st mode frequency inside a station wagon or minivan is determined by the shape and size of the cabin. However, this frequency is close to the resonance frequency of the front suspension, so low-frequency noise, such as booming and drumming (low-frequency road noise) easily become an issue, and many man-hours are spent developing countermeasures for these phenomena. Recent research^{(1), (2)} has investigated the structural-acoustic coupling system in this area, but most countermeasures consist of reinforcing the body framework or adding mass dampers or dynamic dampers, etc. There are some cases where countermeasures focus on panels, but most of these implement countermeasures

for only a single panel⁽³⁾, and few cases take into account the relation between multiple panels.

This paper focuses on station wagons and minivans, and describes a method for reducing the sound pressure inside the cabin by appropriately distributing the resonance frequencies of the windshield, roof, tailgate and other multiple panels that comprise the cabin. This method is applied for the frequency range in which the acoustic 1st mode is dominant. To validate this method, case studies using the boundary element method (BEM)⁽⁴⁾ for low-frequency booming noise and drumming are presented.

The objective of this method is to quickly determine specifications in the initial development stages by setting an appropriate resonance frequency for each panel.

2. Method for Reducing Noise by Controlling Panel Phase

2.1. Mechanism of Noise Generation

The mechanism of low-frequency noise generation is as follows: the vibration sources are the engine and the suspension, and that vibration is conveyed to the body through the engine mounts and suspension bushings, etc. This vibration then induces vibration in the windshield,

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roof, tailgate and other panels that comprise the cabin, which results in noise produced inside the cabin. Accordingly, the acoustic mode present in the cabin affects the sound pressure. However, the acoustic mode frequency is determined by the shape and size of the cabin, and the acoustic 1st mode frequency in station wagons and minivans is approximately 40 to 50 Hz. This mode produces a sound distribution with anti-nodes (high pressure fluctuation) in the front and rear of the cabin and a node in the center as shown in Fig. 1.

2.2. Issues with Conventional Methods

Most conventional methods for reducing low-frequency noise have focused on the panel with the highest contribution, and structural changes that modify the characteristics of panel response to reduce the vibration level. In such cases, there is often a limit to how much the vibration level can be reduced by changing the structure, so efforts then have also shifted to reducing the panel vibration with the next highest contribution. However, this method of development takes a lot of time to determine countermeasure specifications. Therefore, to quickly determine the specifications within the limited development period, a new method was necessary that focuses on the characteristics of multiple panels and does not rely simply on reducing the vibration level.

2.3. Method for Reducing Noise Focusing on the Phase of Multiple Panels

This study took into account the characteristics of the acoustic 1st mode, focused on the phases of two or more panels, and proposed a method for realizing a phase relation that reduces sound pressure by distributing the resonance frequencies.

For example, consider the case where sound pressure is produced from only two panels: the front windshield and the tailgate. At low frequencies that are relatively unaffected by the acoustic 1st mode, such as the idle noise range (20 to 25 Hz) for second-order vibration of a 4-cylinder engine, panel phase relations that increase the volumetric variation in the cabin space also increase the sound pressure. On the other hand, in the range where the acoustic 1st mode is dominant, phase relations that increase the volumetric variation exhibit a sound distribution with opposite phase at the front and back of the cabin due to the acoustic mode characteristics as

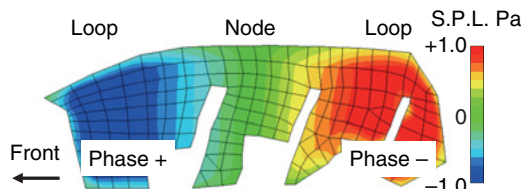


Fig. 1 Sound pressure distribution of 1st acoustic mode

shown in Fig. 2. This means that the sound produced from the windshield is cancelled by the sound produced from the tailgate, so the sound pressure is reduced. In other words, it is beneficial to have a structure where the front and rear panels vibrate out of phase with each other in the frequency range of the 1st acoustic mode of the cabin interior.

Investigations were performed for validation using this method of sound pressure reduction by controlling the panel phases.

2.4. Controlling the Panel Phase according to the Resonance Frequency

The fundamental characteristic of a basic single degree of freedom model is that the phase inverts at the resonance frequency. Therefore, it was thought that this feature could be used to realize a panel phase relation that reduces sound pressure by optimally distributing the resonance frequencies of the panels with high contributions to the sound pressure inside the cabin.

Vibration tests were conducted using completed vehicles, and the vibration phase characteristics were compared for the center of the windshield and front pillar section. The results confirmed that the phase inverts at the 1st mode resonance frequency of the windshield as shown in Fig. 3. These results indicate that the phase characteristics of each panel in a completed vehicle can be aligned by changing the respective resonance frequencies.

2.5. Panel Phase Combinations

A method of achieving a phase relation that cancels the acoustic 1st mode was investigated from the viewpoint of the resonance frequencies of two panels. By setting the resonance frequency of the windshield lower and the resonance frequency of the tailgate higher than the acoustic mode, the phases of the two panels in the acoustic mode range become opposite to each other and provide destructive interference with the cabin interior mode, and the sound pressure can be reduced. Figure 4

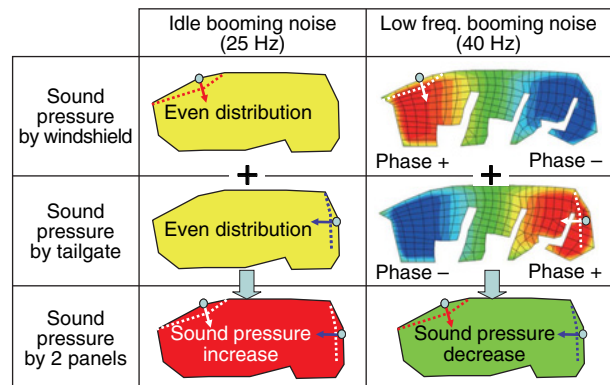


Fig. 2 Mechanism of noise generation

shows these resonance frequencies and phases.

It was found that noise is increased or cancelled depending on the phase relation of the panels, which is determined by the location of their respective resonance frequencies. Completed vehicles have high sound pressure levels at frequencies near the acoustic 1st mode, so this study set out to control the panel phase and reduce the sound pressure inside the cabin around the frequency of the 1st acoustic mode of the completed vehicles.

The panel phase relation is determined by whether the resonance frequency of each panel is set higher or lower than the acoustic 1st mode frequency of 43 Hz. Four panels had a high degree of contribution to the sound pressure inside the cabin for the vehicle used in these tests. In descending order of highest contribution, these panels were the windshield, the roof, the rear floor, and

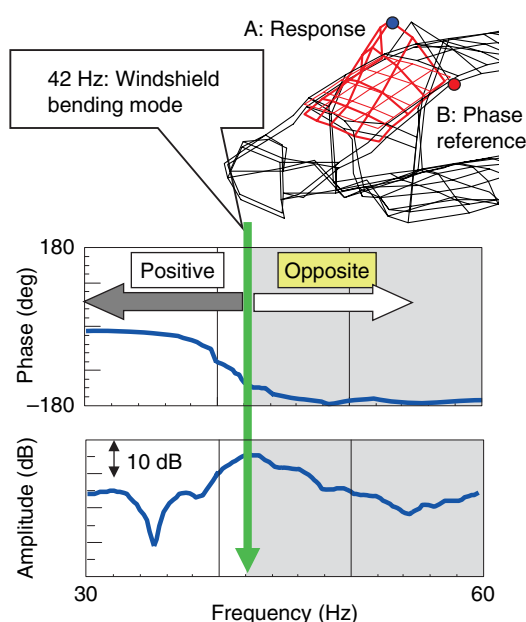


Fig. 3 Transfer function from front pillar to center of windshield

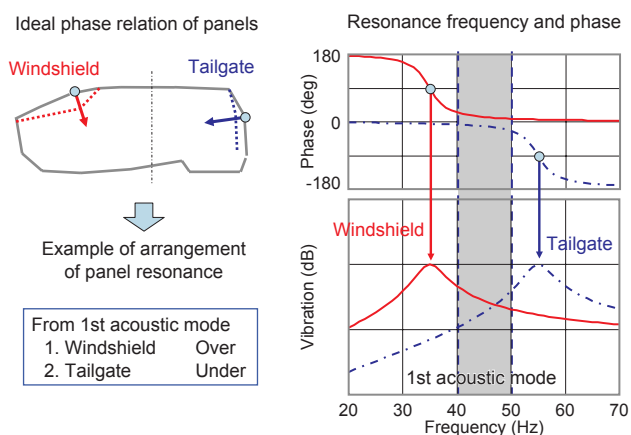


Fig. 4 Ideal arrangement of panel resonance

Table 1 Combination of panel resonance frequency

	Windshield	Roof	Rear floor	Tailgate	Phase condition
A	↑	↑	↑	↓	Even
B	↑	↑	↓	↓	Poor
C	↑	↓	↑	↓	Excellent
D	↑	↓	↓	↓	Good
E	↓	↑	↑	↓	Good
F	↓	↑	↓	↓	Excellent
G	↓	↓	↑	↓	Poor
H	↓	↓	↓	↓	Even

■ Unsuitable phase condition

↑ Resonance frequency : over 1st acoustic mode

↓ Resonance frequency : under 1st acoustic mode

the tailgate. Therefore, various combinations were investigated to determine whether the resonance frequency of each panel should be set higher or lower than 43 Hz. For structural reasons it is a challenge to set the resonance frequency of the tailgate higher than 43 Hz, so it should be set lower. As a result, the possible resonance frequency combinations are the eight combinations A to H shown in Table 1. In addition, whether a certain phase increases or cancels the noise can be determined from the test results by using the windshield as the reference. This is shown in the table, with shaded distributions indicating phase relations that increase the noise produced from the windshield. Combinations with few or no shaded areas are best, so this shows that combinations C and F are reliable.

Investigations were performed for validation using this approach of reducing sound pressure by controlling the panel phases.

3. Analysis Method Using BEM

3.1. Prediction of the Sound Pressure inside the Cabin Using BEM

The sound pressure inside the cabin is a complex summation of the noise produced from the vibration of each panel inside the cabin. To know which panel is important with respect to the generated the sound pressure, a method for clarifying the panel contribution is necessary. Therefore, an analysis method was introduced that models the cabin space using the boundary element method (BEM), and then calculates the sound pressure from the panel vibration.

First, a closed cabin space, including seats, was modeled as shown in Fig. 5.

When the boundary surfaces are modeled as rigid walls, the calculated acoustic mode frequency is higher and the damping is lower than the measurement results using a completed vehicle. This can cause a sharp peak to appear, which increases the difference between the calculated and measured the sound pressure inside the

cabin and lowers the accuracy. This is thought to be because the acoustic impedance of the boundary wall surfaces in the completed vehicle state is lower than that of the calculation model, due to the effects of the interior materials. Therefore, discrete characteristics tests were conducted on a number of interior material types, and except for the glass sections that have high acoustic impedance, the average results value was set for the boundary wall surfaces. This method reduced the difference between the calculated and measured values to within ± 3 dB as shown in Fig. 6.

3.2. Analysis of Panel Contribution

This method can be used to calculate the sound component produced by a panel from the vibration velocity of that panel. The panel contribution can then be analyzed by comparing the magnitude and phase of the sound component of each panel. A method for reducing sound pressure is examined below using the results of panel contribution analysis for two panels as an example.

When the windshield component and the roof

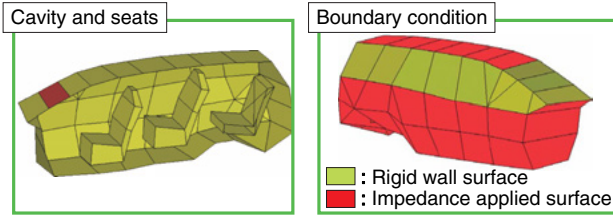


Fig. 5 BEM cavity model

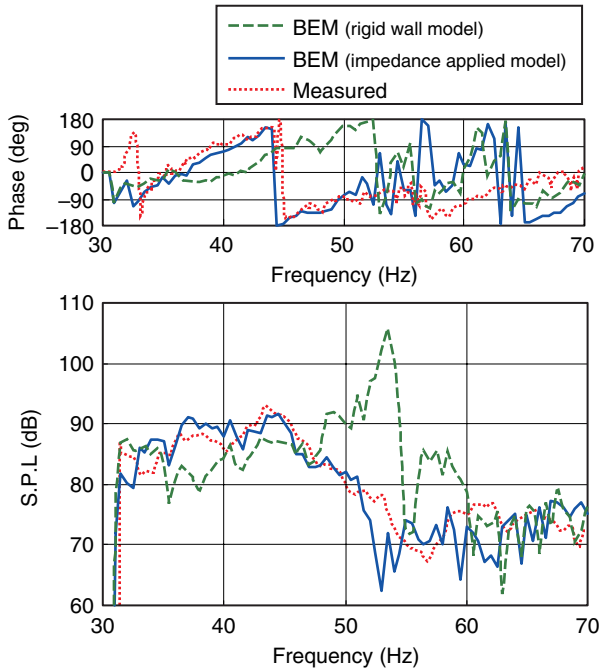


Fig. 6 Calculated S.P.L from actual car vibration

component make large contributions to the sound pressure inside the cabin, the sound pressure can be reduced by establishing a phase relation where these two sound components cancel each other. Figure 7 shows an example with the phase of the windshield component delayed by 90° . This makes the phase difference with the roof component approximately 180° , which reduces the total vector. In other words, it is thought that if this condition can be realized, the sound pressure inside the cabin can be reduced.

3.3. Application to Drumming (Low-frequency Road Noise)

3.3.1. Issues related to understanding the panel phase

The engine excitation force is the input source for the phenomenon of low-frequency booming noise, so the phase relation of multiple panels can be easily understood by using the engine pulse as the measurement reference. On the other hand, when applying this method to drumming (low-frequency road noise), the vibration input from the road surface is conveyed to the body through four tires. This means there are multiple uncorrelated inputs, so the signal processing should be enhanced in some way to understand the phase relations.

Therefore, the coherent component method was used for phenomena with multiple inputs. This method extracts a reference with a high degree of contribution to understand the phase relations as described below.

3.3.2. Coherent component method

The drumming input path is through the vibration of the suspension, so the phase relation of each panel was understood by breaking down the operational sound pressure and panel vibration into components that have a high correlation with the vibration of each wheel suspension.

Consider the case where k number of independent vibration inputs are conveyed from the suspension. Here, the measured the sound pressure inside the cabin P is expressed as the total of k components as shown in Eq. (1).

$$P = P_1 + P_2 + P_3 + \dots + P_k \quad (1)$$

In this manner, the phase relation of each panel is

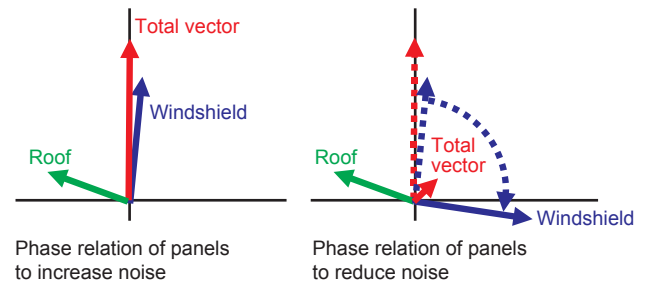


Fig. 7 Panel contribution analysis

investigated by breaking down the sound pressure into the components produced by each input. In this case, the sound component P_j attributable to the input j is obtained by multiplying the input j by the square root of the cabin sound coherence γ_j as shown in Eq. (2).

$$|P_j| = |P| \cdot \sqrt{\gamma_j} \quad (2)$$

The phase characteristics of P_j are obtained by the cross power spectrum of the input signal and the sound pressure inside the cabin.

Assuming there are m number of panels and focusing on the vibration of panel n , the component V_{nj} that correlates to input j is obtained from the coherence γ_j between the panel n vibration and input j as shown in Eq. (3).

$$|V_{nj}| = |V_n| \cdot \sqrt{\gamma_{nj}} \quad (3)$$

The phase characteristics of V_{nj} are obtained by the cross power spectrum of the panel n vibration. P_j is calculated as shown in Eq. (4) by applying the obtained vibration input V_{nj} that correlates to j to the BEM model, and at the same time the contribution of each panel can also be calculated.

$$P_j = \sum_m \{ [H_j] \cdot V_{nj} \} \quad (4)$$

H_j : Acoustic vibration transfer function of panel j

4. Results of Verification in Completed Vehicles

4.1. Low-frequency Booming Noise Reduction Effects

4.1.1. Sound pressure reduction effects of controlling the panel phase

Structural changes to the vehicle body were investigated using CAE with the goal of realizing combinations C and F in Table 1. The results showed that a significant increase in weight would be needed to raise the resonance frequency of the windshield for combination C. Weight restrictions are a major factor in the development of the subject vehicles, so combination F was determined to be more realistic, and this combination was verified using a completed vehicle. In combination F, the roof and rear floor sound components have phase relations that cancel the windshield sound component, and the tailgate sound component phase increases it. Therefore, overall noise reduction effects can be anticipated by achieving this phase relation.

Here, the test vehicle used for verification of low-frequency booming sound is labeled “a.” The resonance frequencies of each panel in vehicle “a” are 40 Hz for the windshield and 43 Hz for the roof, both of which are close to the acoustic 1st mode frequency. Structural investigations were performed using CAE to set the resonance frequencies of the windshield and the floor lower, and the resonance frequency of the roof higher than the acoustic 1st mode frequency, and these structural changes were then made on a test vehicle to realize

combination F. The acoustic 1st mode is dominant over a range centered on the mode frequency of 43 Hz, so the resonance frequencies were set at 33 Hz for the windshield, 60 Hz for the roof, and 36 Hz for the floor. This distributed the panel resonance frequencies sufficiently away from the acoustic mode frequency, and realized combination F in Table 1 as shown in Fig. 8.

After making these structural changes in a completed vehicle, the vibration phases of the windshield and roof were checked. The results confirmed that the phase relations cancel each other in the range where the acoustic 1st mode is dominant as shown in Fig. 9 (upper level). This resulted in significant low-frequency booming noise reduction effects of -10 dB or more around 43 Hz (engine speed: 1300 rpm), compared to before the structural changes.

The results of panel contribution analysis confirmed that the relation between the windshield sound component and roof sound component changed from characteristics that increased noise to those that cancel noise after the structural changes. (Fig. 10)

4.1.2. Comparison of results with the conventional method

The results were compared with the conventional method to compare the effects of phase control. The conventional method sets out to reduce the panel vibration level. Therefore, structural changes were made to the same vehicle “a” to realize combination A in Table 1, which has phase relations that increase the acoustic mode. Therefore, noise reduction effects of countermeasures that only reduce the panel vibration level were confirmed.

In this case, the resonance characteristics of the roof were kept the same as those of combination F, but the windshield resonance frequency was changed to 60 Hz, and the rear floor resonance frequency to 45 Hz (Fig. 11). The characteristic values were increased by increasing the rigidity of the panel support structures, and the structure was changed for the vibration level to be equal to that of combination F. Figure 9 (lower level) compares the low-frequency booming noise reduction effects with the windshield and roof phases. Unlike combination F, the

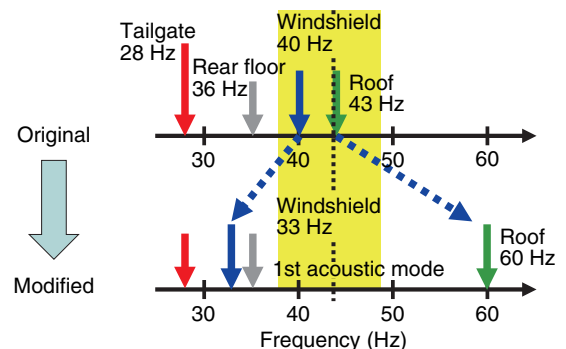


Fig. 8 Modification F of panel resonance

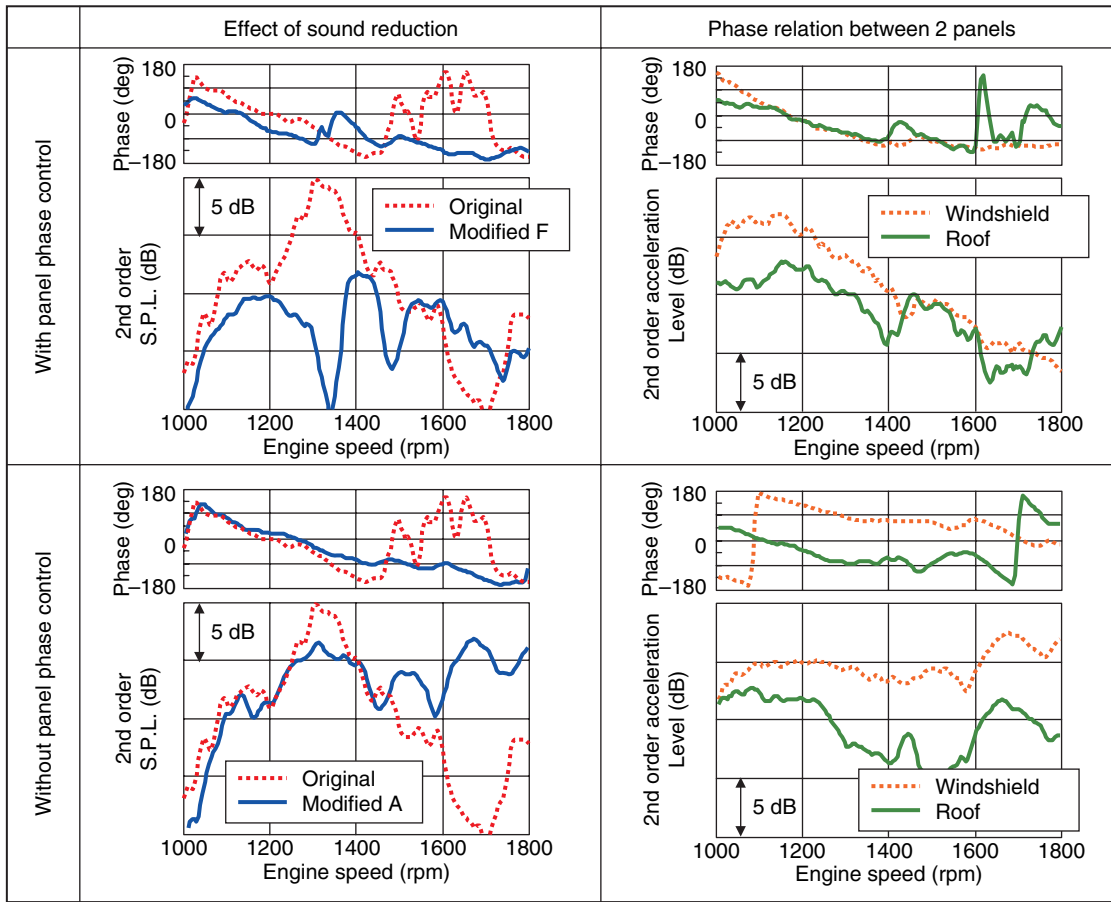


Fig. 9 Result of low frequency booming noise and panel phase relation

windshield and roof are out of phase. This is a phase relation that increases the acoustic response at the 1st mode, so while noise reduction effects of -4 dB were obtained, the effects were less than those for combination F. Both the windshield and roof vibration were reduced to approximately the same vibration level as those for combination F, therefore to obtain additional noise reduction, the vibration level should be reduced even more. However, it is a challenge to realize such a structure. This verified that combination F, which uses phase control, provides superior noise reduction effects as compared to the conventional method of combination A.

Thus, low-frequency noise issues can be quickly addressed by investigating this type of resonance distribution from the initial development stages.

4.2. Drumming Reduction Effects

4.2.1. Coherent component extraction results

Here, the test vehicle used to verify drumming is labeled “b.” The input contributions were identified for vehicle “b.” First, as the input components for the suspension vibration, the coherent components corresponding to the vibration of the four wheels in three directions were extracted using Eq. (1). In the subject

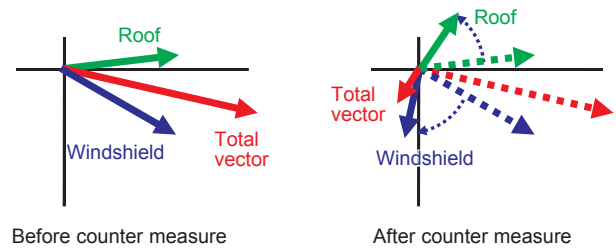


Fig. 10 Panel contribution vector at 43Hz for booming noise

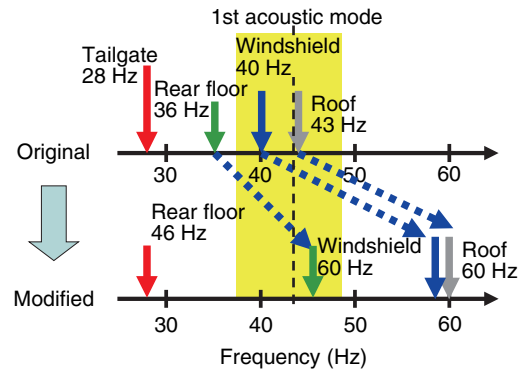


Fig. 11 Modification A of panel resonance

frequency range, the suspension vibration level is higher in the longitudinal direction than in the other directions, so the coherent components were compared for the longitudinal direction. Figure 12 shows the results.

These results show that the input component from the rear wheels has a high contribution in the 40 Hz range. Therefore, the panel vibration velocity components were calculated in the same manner from the measurement results using the longitudinal vibration of the rear wheel suspension as the reference, and applied as the boundary conditions in the BEM model.

Figure 13 compares the sound pressure component of the rear wheel suspension with the BEM model calculation results. Like the analysis results for low-frequency booming noise, an accuracy of within ± 3 dB was obtained. These results show that sufficient accuracy can also be obtained when applying the coherent component method to drumming.

The panel contributions at 43 Hz were confirmed from the BEM analysis results, and the tailgate component was found to be dominant and to have a phase relation that increases the acoustic mode and with the windshield component. Therefore, possible countermeasures would be to first reduce the amplitude of the tailgate vibration, and then to change the phases of these two panels to cancel the acoustic mode.

The panel resonance frequencies for this test vehicle "b" were confirmed to have a distribution as shown in Fig. 14. The distinguishing features are that the roof curvature is small so that the resonance frequency is high, and the tailgate resonance frequency is also high. The

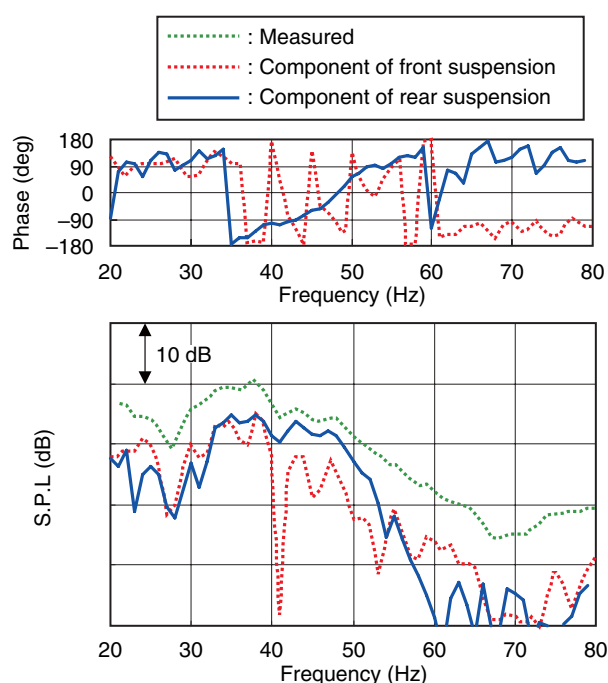


Fig. 12 Result of calculated coherent component

characteristic value of the tailgate is closer to the acoustic 1st mode frequency than that of the roof panel, so the tailgate is predicted to have a greater effect. Therefore, a combination corresponding to A in Table 1 was judged to provide significant noise reduction effects. That is to say, setting the windshield resonance frequency higher and the tailgate resonance frequency lower than those of the acoustic mode was predicted to cancel the acoustic mode.

In test vehicle "b", the windshield resonance frequency was raised by 5 Hz by countermeasures to increase the rigidity of the front roof rail, and the resonance frequency of the tailgate was lowered by 5 Hz by reducing the spring force of the tailgate stopper. As a result, the panel contributions changed as shown in Fig. 15, and the two panels changed to phases that cancelled the acoustic mode as expected. This phase relation provided reduction effects of -6 dB in the 40 Hz range as shown in Fig. 16.

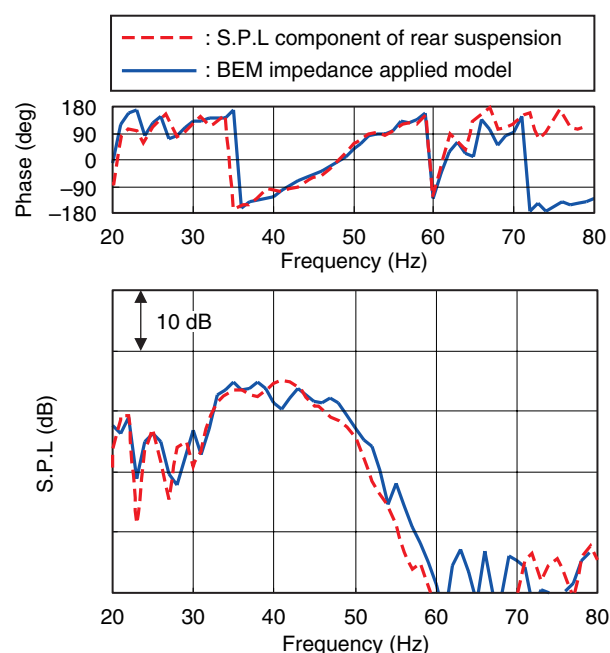


Fig. 13 Compared with S.P.L. measured and calculated

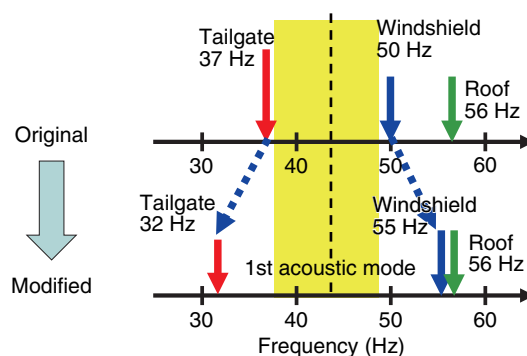


Fig. 14 Modification of panel resonance for drumming

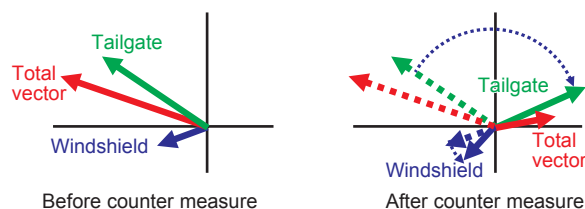


Fig. 15 Panel contribution vector at 43Hz for drumming

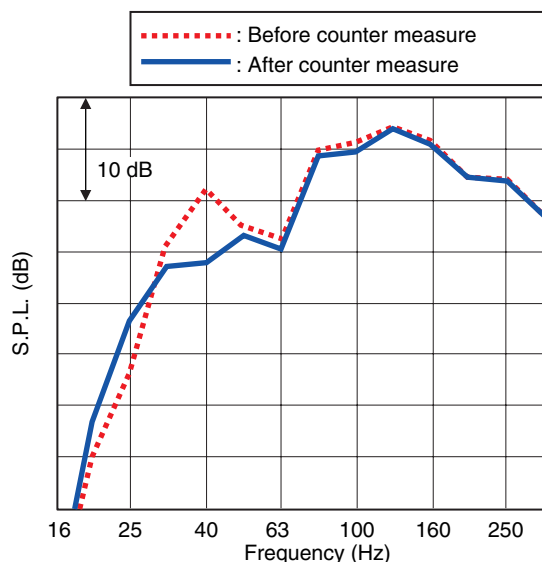


Fig. 16 Effect of panel phase control for drumming

5. Conclusion

The following points were clarified with respect to low-frequency noise, such as low-frequency booming noise and drumming, in station wagons and minivans.

- (1) It was shown that the panel phase characteristics can be controlled by the panel resonance frequency, and resulted in sound pressure reduction inside the cabin through cancellation by appropriately distributing the resonance frequencies of multiple panels.
- (2) The ability to reduce the sound pressure inside the cabin by controlling the panel phase was verified by making actual structural changes to completed vehicles; the effectiveness of these countermeasures was then confirmed.
- (3) Sound pressure was accurately calculated from the panel vibration measurement results using the boundary element method (BEM); the panel phase relations and contributions to sound pressure were then clarified.
- (4) The coherent component method was applied not only to low-frequency booming noise with a single input source, but also to the drumming phenomenon, which has multiple uncorrelated input sources. In both cases, analysis accuracy sufficient for practical application was obtained.

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