

Verification of Modal Arrangement for Reduction of Body Response

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ABSTRACT

This report describes a verification of cabin noise reduction for a medium class sedan at engine speeds lower than 2 000 rpm. Mode frequencies and mode shapes are defined as a modal arrangement, and a target arrangement is established to achieve significant body response reduction. In order to realize the target arrangement, a FE model of a semi-white body structure is modified, and an actual vehicle is modified as well. As a result, overall body response is reduced, and as desired a relationship between phases in the upper and lower parts of the cabin is improved. Thus, the modal arrangement is proven to be effective to reduce body noise and vibration.

1. Introduction

System characteristics related to cabin noise of a vehicle are diverse, including engine vibration, engine mount characteristics, sub-frame characteristics, drive train and suspension transfer characteristics, body characteristics and cabin acoustics. This report examines cabin noise associated with the body frame of a medium class sedan with an in-line 4 cylinder engine (displacement 2.41 – 5 AT) for engine speeds under 2 000 rpm, focusing on body characteristics. The modes that make the largest contribution to body vibration are defined as S-class modes and have been selected for further study. The mode frequency and mode shape of each of the modes is defined as its modal arrangement. A target for a set of modal arrangements that greatly reduce body vibration has been defined. In order to realize the target arrangement, the structure was analyzed using a finite element (FE) model. After making modifications to a semi-white body, it was possible to meet the target of reduction in body vibration and the target for the phase relationship of the upper and lower parts of the cabin. The arrangement example that is being described here shows that it is feasible to reduce body vibration.

2. Experimental Modal Analysis

In order to obtain modes for the vehicle under running condition, operational experimental modal analysis (operational EMA) was performed on a vehicle. In order to better grasp the body modes, modal analysis was performed on a semi-white body, which is a white body with both the windshield and the rear window installed.

2.1. Operational Experimental Modal Analysis

A vehicle was placed on a chassis dynamometer and as can be seen in Fig. 1, there were two excitation points, the front portion of the sub-frame and the knuckle of the right front wheel. Vibration data was taken at 213 different locations, including on the doors, the suspension, the sub frame, and on the body.

The vertical axis of Fig. 2 is vehicle speed and the horizontal axis is frequency. The green dotted lines are wheel harmonics and the black solid line is the first order harmonic from the engine with the vehicle running in 5th gear. The intersections of the red solid line and blue solid line are 90 km/h and 100 km/h respectively, with the green dotted

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lines and black solid line showing where the frequencies meet the first order harmonic of the engine and the wheel harmonics.

In this manner, two speeds where the wheel harmonics and the first engine harmonic do not overlap were arbitrarily chosen within a frequency range for measurement. While holding speed steady, an excitation test using two exciters and frequency response function measurement at 213 locations was performed.

The frequency response functions for the two speeds are



Fig. 1 Excitation points in operational EMA

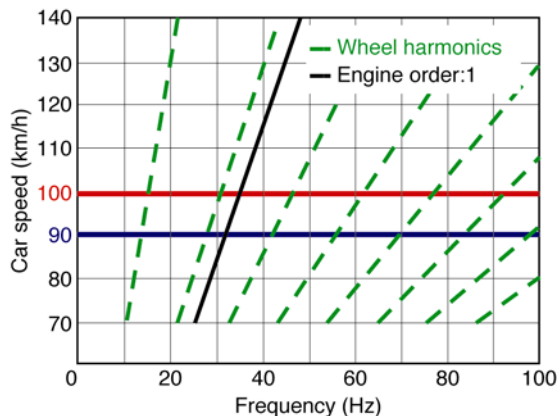


Fig. 2 Condition of operational EMA

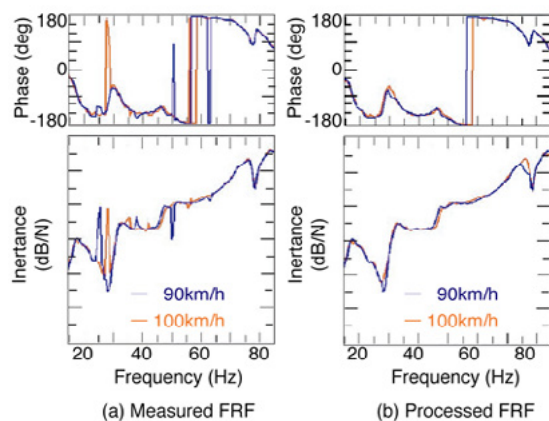


Fig. 3 Post processing of operational EMA

Table 1 S class mode frequency obtained from operational experimental modal analysis

Mode number	Frequency
S1	31.5Hz
S2	45.7Hz
S3	51.7Hz
S4	77.7Hz

shown in Fig. 3. In the frequency response function measured, a noise peak can be seen for a frequency that overlaps one of the wheel harmonics; however, a frequency response function without peaks was obtained through processing of the data.

The modes during vehicle operation were obtained through performing modal analysis using frequency response functions in this manner.

Figure 4 shows synthesized frequency response functions from each mode that has a large contribution to body vibration levels (S-class modes) and actual frequency response function data from the upper part of the cabin. As can be seen from this figure, the contributions of the 4 modes S1, S2, S3, and S4 are large. The frequency of each mode is shown in Table 1.

For reference, there are modes other than these S-class modes that have large contributions; however, they are not body vibration modes, they are modes of the suspension, doors, etc.

As an example, the S1 mode shape has been placed in Fig. 5. A characteristic of this mode is that along with bending of the body, the upper and lower parts of the cabin are out of phase, which is considered an unfavorable mode shape for changing the volume of the cabin.

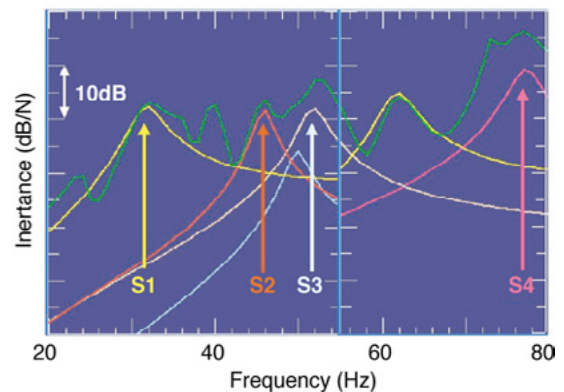


Fig. 4 Contribution of S class modes to measurement frequency response function

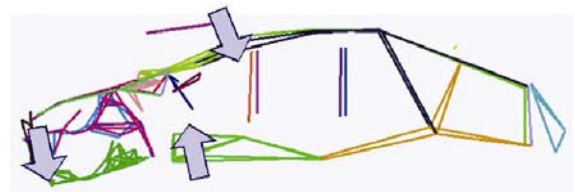


Fig. 5 Mode shape of S1

Similar to the S1 mode, the S2 mode shape is a mode that exhibits out of phase vibration of the upper and lower parts of the cabin. In order to reduce interior sound pressure, changes in the mode frequency and/or improving the mode shape such that the vibration of the upper and lower parts of the cabin change from being out of phase to in phase are considered.

2.2. Experimental Modal Analysis

In order to more clearly grasp body modes, modal analysis of a semi-white body was performed. The semi-white body was placed on an air mount. Exciters were set in 2 places, one at the front right connection point of the front sub frame and the other at the left rear connection point. Frequency response functions were measured at 131 locations in 3 axes. Modes were extracted using modal analysis.

Figure 6 shows synthesized frequency response functions from each of the modes that have a contribution to semi-white body vibration levels (W-class modes) and actual frequency response function data from the upper part of the cabin in the 40 – 60 Hz region. This diagram shows that the contributions of the 5 modes W1, W2, W3, W4 and W5 are large.

The W-class modes have been listed in Table 2. From their mode shape characteristics, it was judged that there is a high correlation between S1 and W1, S2 and W3, and S3 and W4.

As an example, the W1 mode shape is shown in Fig. 7. Similar to S1, the upper and lower parts of the cabin are out of phase and it is clear that this is an unfavorable mode shape for changing the volume of the cabin. It was confirmed that the W3 characteristics matched those of W1.

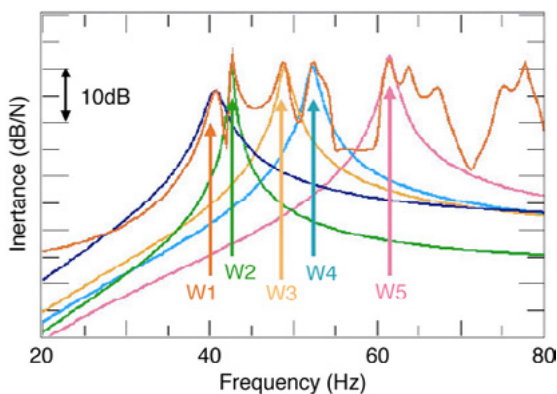


Fig. 6 Contribution of W class modes to measurement frequency response function

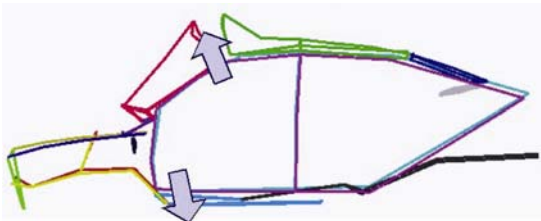


Fig. 7 Mode shape of W1

Table 2 W class mode frequency obtained from experimental modal analysis

Mode number	Mode number	Frequency
S1	W1	40.8Hz
	W2	42.6Hz
S2	W3	48.9Hz
S3	W4	52.8Hz
	W5	60.8Hz

3. Change in Modal Arrangement

A modal arrangement is defined to be the combination of the mode frequency and mode shape. Experimental changes to the modal arrangement for reduction in the vibration of the upper part of the cabin and for improvement in the phases of the upper and lower parts of the cabin were tried.

3.1. Target for Changes to the Modal Arrangement

Targets were set based on the results of the previous paragraph.

For the W1 mode, in order to reduce underbody vibration the stiffness line was lowered and a target of a 20% increase in mode frequency, 49 Hz, was set.

As can be seen in Fig. 8, for the W2 mode, improvements have been made to the mode shape to change the phases of the upper and lower parts of the cabin from an out of phase to an in-phase relationship. In order to couple it with the W1 mode, it was targeted at 49 Hz.

The W3 mode, similar to the W1 mode, has a mode shape where the upper and lower parts of the cabin are out of phase. Through a synthesis of the frequency response function from modal parameters, it was confirmed that a frequency difference of 9 Hz or greater was needed, therefore the target frequency for W3 was set at 58 Hz.

As the W4 mode has a similar mode shape as the W2 mode, it was given the same percentage increase in frequency, 14%, for a calculation of 60 Hz.

For W5, it was confirmed that a frequency difference of 4 Hz or greater than W4 was needed through a synthesis of the frequency response function from modal parameters, therefore the target frequency for W5 was set at 64 Hz.

The target frequency for each of the W-class modes is shown in Table 3.

3.2. Frequency Response Function Synthesized from Target Arrangements

Using the results of the experimental modal analysis, the modal arrangement was changed into the target arrangement. Then a synthesis of the frequency response functions was

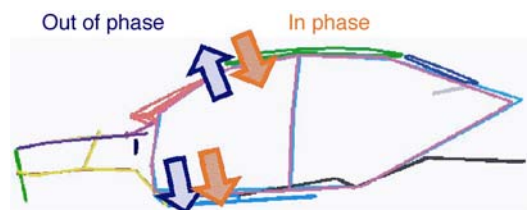


Fig. 8 Concept of mode shape change

Table 3 Target mode frequency determined by refining modal arrangement

Mode number	Base frequency	Target frequency
W1	40.8Hz	49Hz
W2	42.6Hz	49Hz
W3	48.9Hz	58Hz
W4	52.8Hz	60Hz
W5	60.8Hz	64Hz

performed. Figure 9 shows the frequency response function for the front area of the side frame, which is the front body, and the frequency response function for the upper part of the cabin, which has a large contributor to cabin noise. The vibration peak level of the front area of the side frame was not changed. It is estimated that the lowering of the stiffness line would reduce body vibration for lower engine speeds. Through coupling of the different phase modes in the upper part of the cabin it was possible to reduce the vibration level by 6 dB.

As can be seen in Fig. 10, it was also possible to confirm that the phase relationship of the upper and lower parts of the cabin was changed from out of phase to in-phase for the range of 40 – 50 Hz.

From these results it was verified that it is possible to achieve targeted vibration reduction levels through making changes to the modal arrangements in accordance with the above targets.

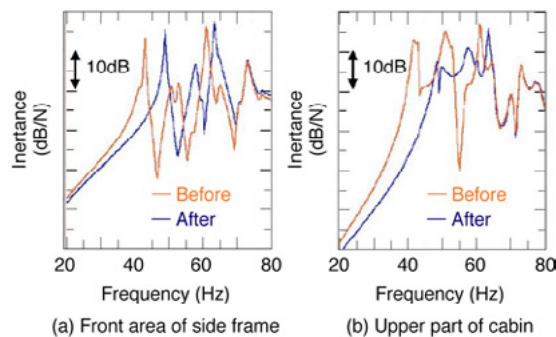


Fig. 9 Estimation of response levels by refining modal arrangement

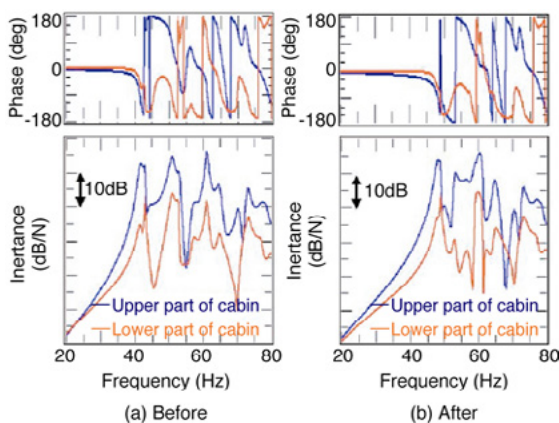


Fig. 10 Estimation of phases in upper and lower parts of cabin by refining modal arrangement

4. Changes to Structure with the FE Model

In order to implement the target modal arrangement established in the previous paragraph, a sensitivity analysis was performed for target modes using an FE model of a semi-white body. An estimation of performance was completed after structural changes were made based on these results.

4.1. Sensitivity Analysis and Structural Changes

For targeted modes, stiffness sensitivity, mass sensitivity, and sheet metal thickness sensitivity analyses were performed. This identified the places where changes to the structure are needed.

Figure 11 shows the results of the stiffness sensitivity analysis of W1 mode. This diagram shows that the mode frequency can be increased through increasing the stiffness of the yellow and pink areas. It has been found that increasing the stiffness around the window and the upper and lower weld areas of the center pillar is effective at changing this mode's modal arrangement.

Figure 12 shows the results of the mass sensitivity analysis of mode W2. This diagram shows that the mode frequency can be increased through decreasing the mass of the brown and blue areas. For this mode, reduction in the weight of the front area of the side frame and the front bulkhead was found to be effective in changing the modal arrangement.

For all of the W-class modes, structural changes were made in a FE model while monitoring the strain energy distribution and stiffness and mass sensitivity analyses. Structural changes were made at the joints of each pillar, of the side sill, of the roof side rail, of the roof rail, etc., for an approximate total of 20 places.

4.2. Performance Achievement Estimation

A refined FE model investigated previously was used to estimate performance.



Fig. 11 Stiffness sensitivity analysis



Fig. 12 Mass sensitivity analysis

Table 4 Estimated mode frequencies with refined FE mode

Mode number	Target frequency	Estimated with refined FE model
W1	49Hz	52Hz
W2	49Hz	49Hz
W3	58Hz	57Hz
W4	60Hz	59Hz
W5	64Hz	64Hz

Table 4 shows the mode frequencies for the performance estimation from a refined FE model and targeted modal arrangements. Concerning the modal arrangement, the W1 mode is 3 Hz higher than the target and the W3 mode is 2 Hz lower. Figure 13 shows the synthesized frequency response function for the front area of the side frame and the upper part of the cabin with the base and refined FE model. It also shows a 6 dB reduction was achieved for the front area of the side frame and a 12 dB reduction was achieved for the upper part of the cabin.

The phase relationship of the upper and lower parts of the cabin is shown in Fig. 14. Prior to the changes they were out of phase in the 40 – 52 Hz region. After the change they now have an in-phase relationship in this region.

Through the structural changes discussed previously, it is expected that the modal arrangement will be improved and a reduction in vibration level will be achieved when the changes are also made to an actual vehicle.

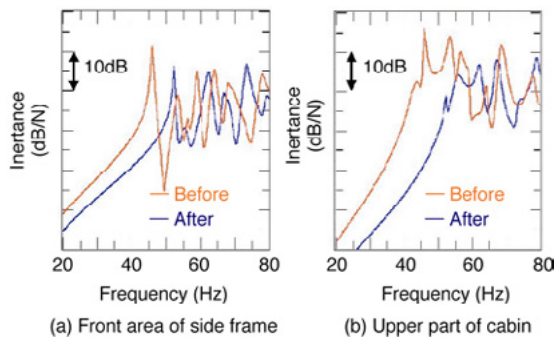


Fig. 13 Estimation of response levels by refining FE model

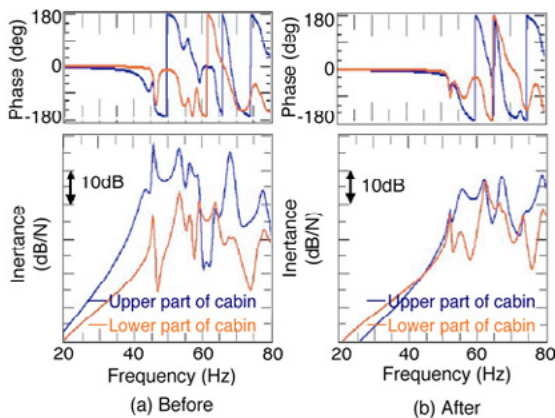


Fig. 14 Estimation of phases in upper and lower parts of cabin by refining FE model

Table 5 Verified mode frequencies with modified semi-white body

Mode number	Estimated with FE model	Verified with modified semi-white body
W1	52Hz	51.9Hz
W2	49Hz	49.2Hz
W3	57Hz	56.3Hz
W4	59Hz	59.1Hz
W5	64Hz	63.5Hz

5. Verification of a Modified Semi-white Body

The changes that were made in the FE model were used to make modifications to a semi-white body. Verification of the changes to the modal arrangement, developed by these modifications, was performed through experimental modal analysis, using the conditions described in section 2.2. The information in Table 5 confirms that the mode frequencies estimated by the FE model match the results of the experimental modal analysis.

The frequency response functions of the upper part of the cabin and the front area of the side frame for the base and modified condition of the semi-white body are shown in Fig. 15. Through moving the stiffness line down the vibration level of the front area of the side frame was lowered 6 dB. The upper part of the cabin's vibration level was lowered 7 dB at the peak frequency under 60 Hz and was lowered 10 dB at other frequencies.

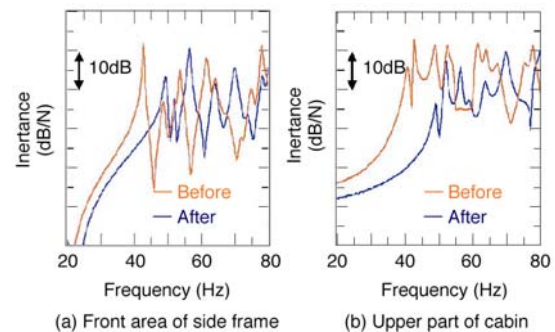


Fig. 15 Verification of response levels by modifying semi-white body

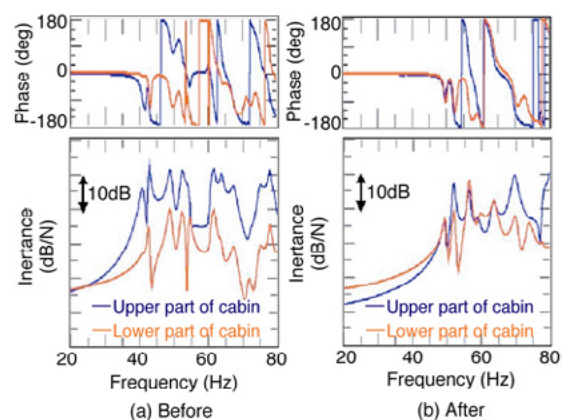


Fig. 16 Verification of phases in upper and lower parts of cabin by modifying semi-white body

The phase relationship for the upper and lower parts of the cabin is shown in Fig. 16. Prior to changes made to the semi-white body, the upper and lower parts of the cabin were out of phase over the broad region of 42 – 60 Hz. After making the targeted changes to the mode shape, the upper and lower parts of the cabin became in phase for the ranges below 52 Hz and above 60 Hz.

6. Conclusion

This report focused on the reduction of cabin noise for engine speeds under 2 000 rpm for a medium class sedan with an in-line 4 cylinder engine. This was done through investigation of the modal arrangement of a semi-white body. The following results were achieved.

- (1) The modal arrangement was changed with a focus on the phases of the upper and lower parts of the cabin. Through analysis it was estimated that by coupling the different modes of specific target areas, the body vibration level could be reduced.
- (2) Structural changes that could achieve the targeted mode shapes and frequencies were first investigated through analysis and then verified through modifications to a semi-white body.

This verifies that the examination of the modal arrangement is effective at reducing the level of body vibration.

The changes described in this report will be applied to the trimmed body and vehicle. This will be done in order to further verify that the modal arrangements used in this report are effective at reducing cabin noise levels.

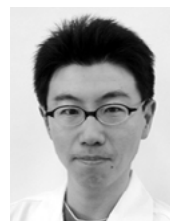
This method has been exhibited to be a useful tool to reduce the overall vehicle weight while improving noise and vibration performance, crash performance, and driving stability.

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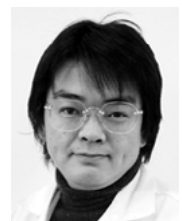
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