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PROCESS OF DESIGNING BODY STRUCTURES FOR THE REDUCTION OF REAR SEAT NOISE IN PASSENGER CAR

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ABSTRACT–This study analyzes the interior noise that is generated during acceleration of a passenger car in terms of car body structure and panel contribution. According to the transfer method, interior noise is classified into structure-borne noise and air-borne noise. Structure-borne noise is generated when the engine's vibration energy, an excitation source, is transferred to the car body through the engine mount and the driving system and the panel of the car body vibrates. When structure-borne noise resonates in the acoustic cavity of the car interior, acute booming noise is generated. This study describes plans for improving the car body structure and the panel form through a cause analysis of frequency ranges where the sound pressure level of the rear seat relative to the front seat is high. To this end, an analysis of the correlation between body attachment stiffness and acoustic sensitivity as well as a panel sensitive component analysis were conducted through a structural sound field coupled analysis. Through this study, via research on improving the car body structure in terms of reducing rear seat noise, stable performance improvement and light weight design before the proto-car stage can be realized. Reduction of the development period and test car stage is also anticipated.

KEY WORDS : Structure-borne noise, Structural-acoustic coupling analysis, Rear seat noise, Booming noise

1. INTRODUCTION

The latest trend in the automotive industry has demanded the development of car bodies featuring high-stiffness and securement of intersystem performance for low-vibration and low-noise vehicles.

However, this demand is in conflict with lightweight design for higher fuel efficiency. As such, researchers are focusing on an optimized design to fulfill both developmental goals (Kim *et al.*, 2005).

Booming noise is that oppresses the driver's and passengers' ears. It is generated each time the specific travel space or revolution per minute (RPM) range is reached during a vehicle's acceleration/deceleration or travel at constant speed (Sim and Chung, 2002).

The vehicle studied in this research was a 1,800 cc passenger car with a 4-cylinder engine. The interior noise generated in the wide-open throttle valve (WOT) of the vehicle during acceleration was mainly by 2nd-order components with respect to the engine RPM; as for frequency components, it was less than 200 Hz for a 4-cylinder engine (Osawa and Iwama, 1986).

The main cause of booming noise, the excitation force of the engine, is transferred to the car body through the engine mount or the suspension and then causes the panel including the dashboard, floor, roof, and package tray to vibrate. Acute booming noise is generated in the car interior car when the acoustic cavity resonates (Kim and Choi, 2003).

The purpose of this study to analyze the cause of the generation of a high sound pressure level in the rear seat relative to the front seat of the vehicle during acceleration.

To this end, the correlation between body attachment stiffness and acoustic sensitivity is analyzed using a structural sound field coupled analysis.

Panel sensitive components were determined through the operational deflection shape (ODS) of sensitive frequency (Hendrick *et al.*, 1997). This study has presented the direction for improving the car body structure relative to rear seat noise through a preliminary analysis using the mother car at the initial design stage.

2. STRUCTURAL-ACOUSTIC TRANSFER FUNCTION

When there are N transfer paths of vibration energy for a vehicle, the total structure-borne noise $p(\omega)$ at specific locations can be thus expressed as the sum of the structure-borne noise generated by the vibration energy intro-

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duced through each of the transfer paths:

$$p(\omega)=\sum_{i=1}^{N} p_i(\omega) \quad (1)$$

Where, i signifies each transfer path. In addition, $p_i(\omega)$, which is structure-borne noise generated in specific locations in the car interior by vibration that has been transferred through transfer path i and is a complex number, can thus be expressed as a multiple of the excitation force f_i, which is transferred through transfer paths, and the structural-acoustic transfer function: (Nefske *et al.*, 1984).

$$p_i(\omega)=H_{pi}(w)\cdot f_i \quad (2)$$

Where, $H_{pi}(\omega)$ is the structural-acoustic transfer function of specific locations in terms of the transfer path i. This is the frequency response function (FRF) of sound pressure P_i at specific locations in terms of the excitation force F_i, which has acted on the transfer path i:

$$H_{pi}(\omega)=\frac{P_i}{F_i} \quad (3)$$

Such a transfer function can generally be measured in an anechoic room using an impact hammer. Total structure-borne noise at a specific location therefore can be expressed thus:

$$p(\omega)=\sum_{i=1}^{N} H_{pi}(w)\cdot f_i \quad (4)$$

This equation expresses the vibro-acoustical reciprocity of the noise transfer function (NTF). Vibro-acoustical reciprocity is useful parameter when it is impossible to use mechanical excitation sources such as impact hammering and excitation or to cause excitation in the direction of the tangent line of the plate. It can also be utilized when one wishes to obtain at the same time the structural-acoustic transfer functions for all transfer paths. As for vehicles, because the car body panel is low in local stiffness, it is impossible properly to exert an excitation force with an impact hammer. Consequently, it is necessary to measure the NTF by using the principle of incompatibility (Maruyama *et al.*, 1991).

3. BASELINE TEST

To measure the interior noise level, the vehicle was accelerated to 1,000~6,000 RPM during WOT. As in Figure 1, the locations of the measurement were set along the driver's ear in the front seat and along the passenger's ear in the center of the rear seat for the test. A 1/4 inch array microphone was used and a tachometer (SE-1520) was used to measure the engine RPM. CADA-X was used as the FFT analyzer.

For the baseline test, the following were conducted:

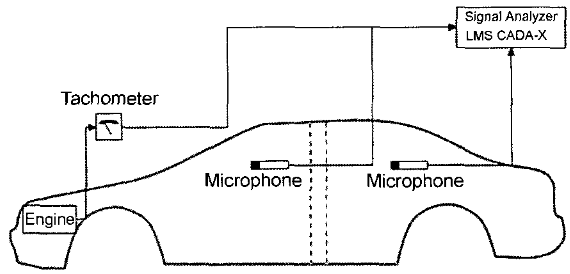


Figure 1. Test setup for measuring interior noise.

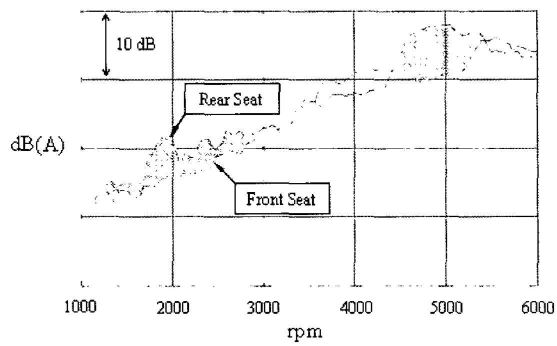


Figure 2. Overall level of interior noise.

interior noise measurement (WOT, coast down) during acceleration/deceleration; spectrum analysis; and order tracking analysis (Jeong *et al.*, 2003).

As in Figure 2, when the overall level of the front and rear seats is compared, the sound pressure level of the rear seat was greater than that of the front seat by a maximum of 5 dB(A) in the 1,000~3,000 RPM and over 4,300 RPM ranges (Jin *et al.*, 2003).

According to a spectrum analysis of the RPM of the vehicle during acceleration, as shown in Figure 3, the engine's 2nd-order components were greater than were other components.

From an analysis of the overall (OA) level in the car interior and the contribution of each of the order com-

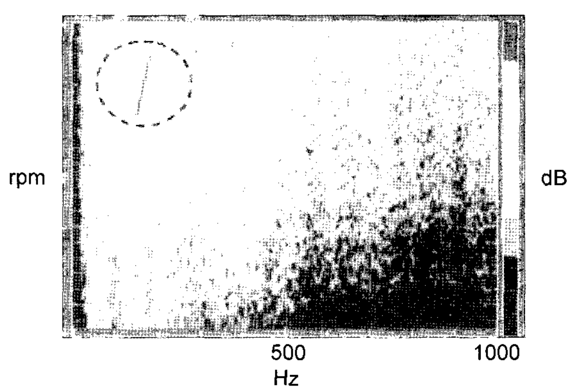


Figure 3. Color map of the rear seat.

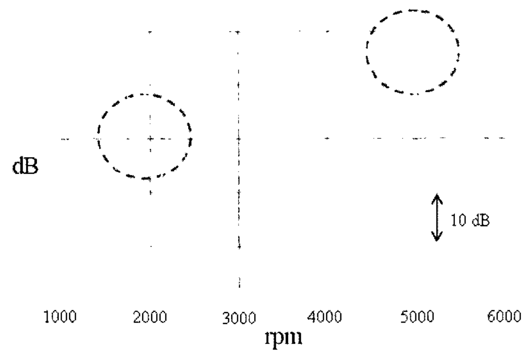


Figure 4. OA level and 2nd order of the rear seat.

ponents, as shown in Figure 4, 2nd-order components were shown to affect total noise considerably. Regarding 2nd-order components that determine the OA level, the sound pressure level was considerable in the below 3,000 RPM and over 4,000 RPM ranges, where rear seat noise was problematic during acceleration. Consequently, it is necessary to improve the noise level of the rear seat at 60~70 Hz, 150 Hz, 165 Hz, and 185 Hz.

4. ANALYSIS PROCEDURES

4.1. Trimmed Body Model

For an interior noise analysis, it is necessary to construct a structural model that most closely approximates the actual vehicle. This study therefore reviewed optimal structural plans for interior noise reduction using a trimmed body structure model that excludes the power train, suspension, and exhaust system.

In addition, the structural model was constructed with a BIW (body in white), door, hood, tail gate, glass, sub frame, rear cross member, and steering system deemed appropriate to the interior noise analysis. The trimmed body model, which consists of 300,000 nodes, 320,000 elements, and 65,000 MPC, is shown in Figure 5:

In the structural model, it is important to appropriately express the structures through which the BIW and moving parts are joined. Depending on the modeling effect of the door side weather strip and the hood over slam bumper, the vibration sensitivity can vary in low frequency ranges. The contribution of such quality deviation factors requires an analytical model and a correlation through test results. This study, however, has omitted this procedure, leaving it to future studies.

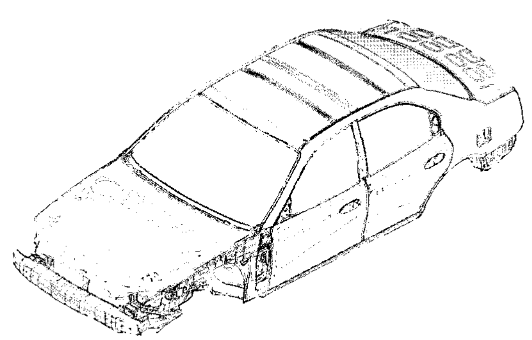


Figure 5. Finite element model of the trimmed body.

4.2. Acoustic Mode Analysis

To conduct a structural sound field coupled analysis, the reliability of the sound field model as well as that of the structural model must be secured. The sound field inside the car is surrounded by glass and interior trim and this interior structure includes the seats.

Trim excluding the seats scarcely absorbs sound in low frequency ranges, and is thus negligible. The seats, however, must be taken into consideration in modeling, because they have a sound absorption rate of 0.5 sabin in the 100 Hz range.

The sound field model consists of 2,920 nodes and 4,944 solid elements. The front and rear seats are included as a solid model within the cavity.

The sound field model analysis revealed that 8 modes

Table 1. Acoustic natural frequency and mode.

NO	Frequency (Hz)	MODE
1	80.2	(1,0,0)
2	124.5	(0,1,0)
3	136.6	(2,0,0)
4	143.0	(0,1,0)
5	151.3	(1,1,0)
6	161.9	(0,0,1)
7	185.9	(3,0,0)
8	191.4	(2,1,0)

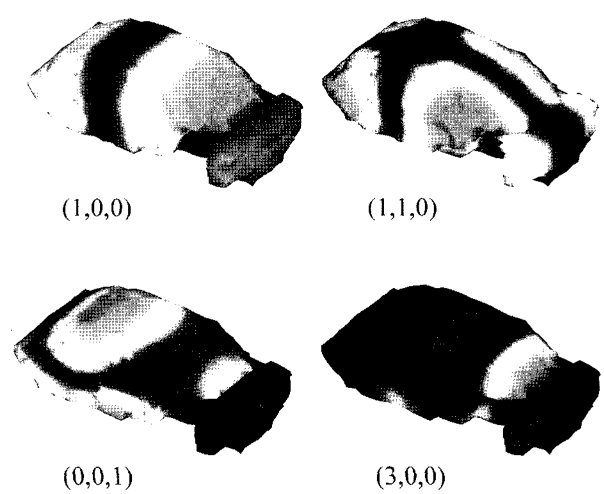


Figure 6. Acoustic modes by 3D FEM model.

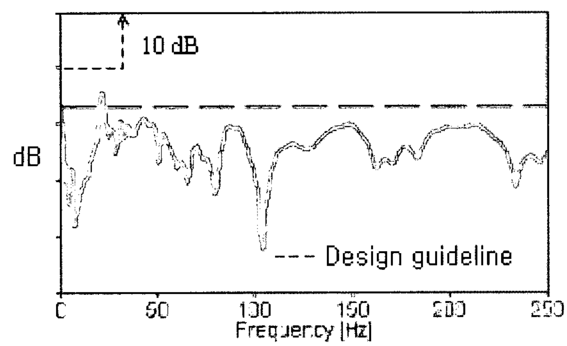
cause problems to the structural mode and coupling in car interior noise in the below 200 Hz range, as listed in Table 1. The sound field mode around the 60~70 Hz, 150 Hz, 165 Hz, and 185 Hz ranges, which are sensitive frequency ranges relative to rear seat noise, were presented in Figure 6.

The sound field model is classified into the car interior and the trunk and connected to the MPC at the rear of the package tray. The seats imitate the seat back and have a density that is ten times standard air density. They effectively move the resonance mode along the length of the sound field of the car interior.

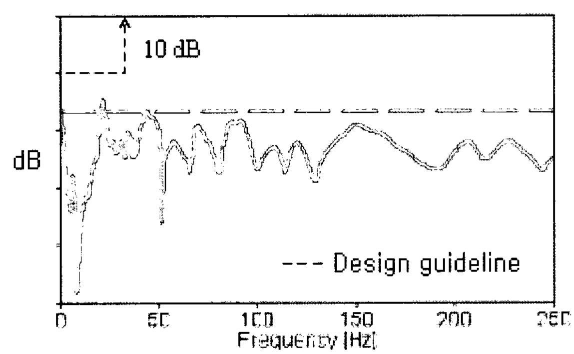
4.3. Structural-Acoustic Coupling Analysis

To predict structure-borne interior noise in low frequency ranges, body attachment stiffness and acoustic sensitivity arising from the coupling of the structural model and the sound field model must be analyzed. Acoustic sensitivity along the driver's ear and passenger's ear in the rear seat according to the excitation of the input point of the power train and the suspension attachment parts is used in the panel contribution analysis of acceleration noise and road noise (Kim and Lee, 1999).

For the coupling of the sound field model and the structure, the ACMODL command is used in NASTARN.

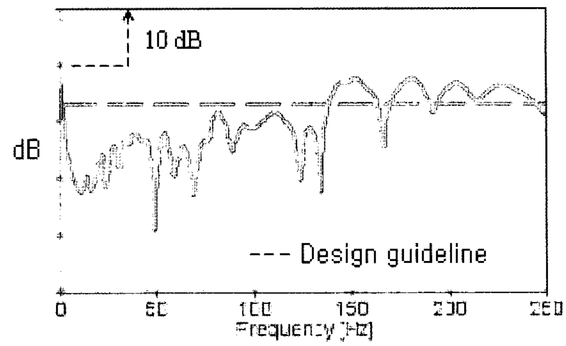


(a) Driver ear

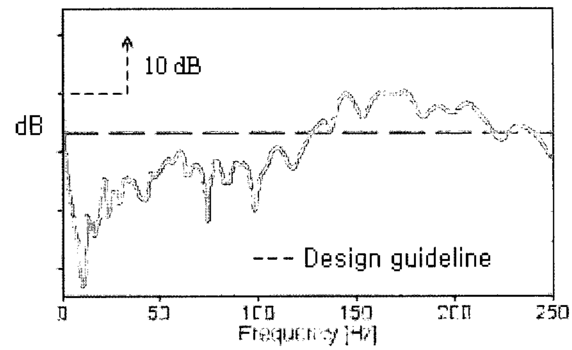


(b) Rear seat

Figure 7. Acoustic sensitivity by exciting engine mounting point.



(a) Driver ear



(b) Rear seat

Figure 8. Acoustic sensitivity by exciting the rear suspension mounting point.

In addition, the BIW method, a new coupled method, is applied.

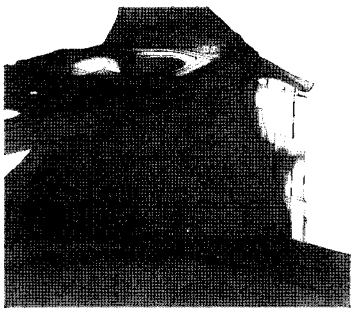
Figure 7 shows the acoustic sensitivity level in the driver's seat and the rear seat according to the vertical excitation of the engine attachment parts. In frequency ranges of 250 Hz or less, acoustic sensitivity relative to the design goal was stable.

Figure 8 expresses the acoustic sensitivity level in the driver's seat and the rear seat according to the sideways excitation of the rear suspension attachment parts. In the frequency ranges of 150~200 Hz, the acoustic sensitivity was unsatisfactory relative to the design goal.

According to an analysis of the panel sensitive components through the ODS of sensitive frequency, as shown in Figure 9, the contribution of the components surrounding the woofer speaker on the package tray center panel and the quarter inner rear panel was considerable.

This study has analyzed the effect of the excitation source relative to rear seat noise and confirmed changes in acoustic sensitivity according to the excitation of the rear suspension attachment parts relative to the engine attachment parts. According to an acoustic sensitivity frequency range analysis, sensitive frequency relative to rear seat noise was similarly in the 150~200 Hz range. A panel sensitivity analysis revealed the need

(a) Package tray center panel



(b) Quarter inner rear panel

Figure 9. Panel contribution parts by exciting rear suspension.

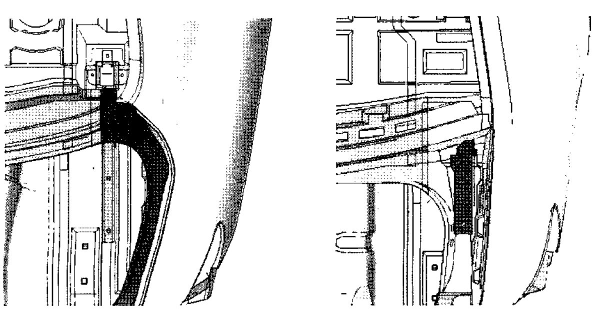
to increase the stiffness of the rear opening near the package tray and of the rear wheelhouse inner joint.

4.4. Design Modification

This study has presented the direction for improving the car body structure relative to rear seat noise through a preliminary analysis using the mother car at the initial design stage. According to a subsequent proto-car analysis, it was possible to secure stable performance.

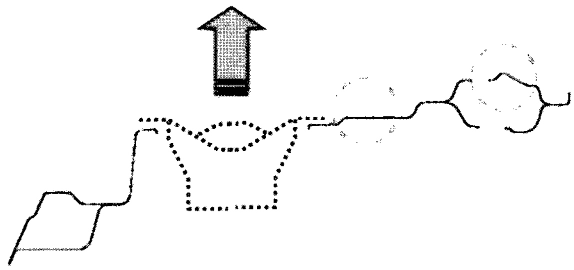
To increase rear opening stiffness according to the exertion of load on the rear suspension attachment parts, as in Figure 10, design- and layout-related design guidelines were presented at the initial design stage.

In terms of design, the trunk lid opening line of the

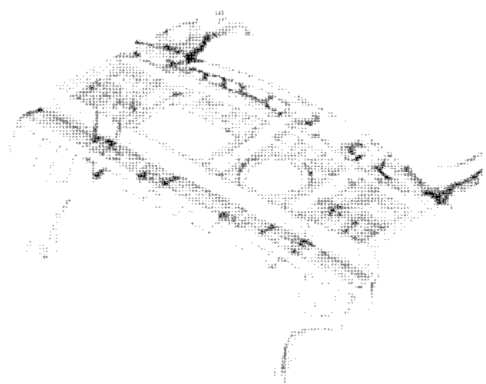


(a) Goose neck hinge (b) 4 link hinge

Figure 10. Design modification of trunk lid hinge.



(a) Package tray section of base model



(b) Curved surface and center reinforcement

Figure 11. Design modification of package tray center panel.

side outer panel was changed from a corner round to a straight line, thus increasing local stiffness. In terms of layout, it is possible to secure rear opening stiffness through the securement of the package tray side cross-section by changing the trunk lid hinges from a goose-neck type to a 4-bar link type.

As shown in Figure 11(a), the package tray center panel considerably affects abnormal noise generation by panel vibration when the woofer speaker is excited. This is a component where the panel contribution relative to rear seat noise is considerable.

As for the direction for improvement, as shown in Figure 11(b), a double curvature surface form was designed on the panel around the speaker and the air purifier to increase local stiffness and frontal and rear reinforcement were applied to the package tray center, resulting in a box combination with the rear window frame.

When local stiffness is weak in the sideways structural members that support the bottom of the rear glass of the rear window frame rear seat noise can be problematic due to changes in the sound field caused by rear glass vibration.

To increase rear opening stiffness, this study has

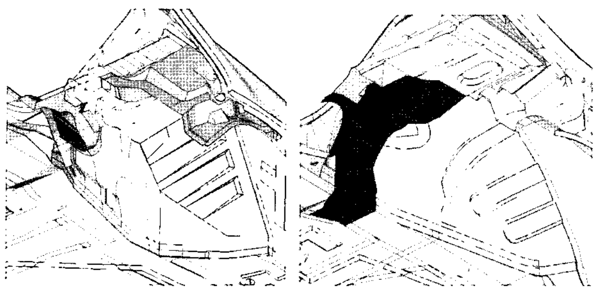


Figure 12. Design modification of the rear wheelhouse and the rear floor.

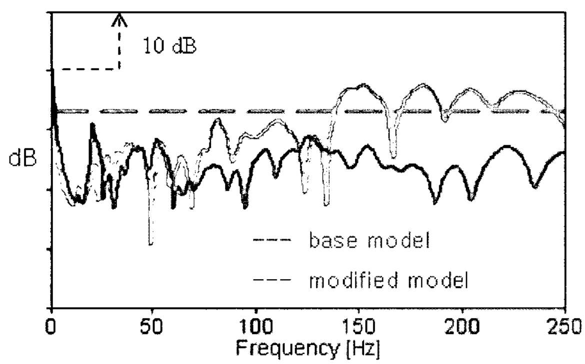
presented directions for improving the car body structure, as shown in Figures 10 and 11, from which rear seat noise is expected to be alleviated.

According to a trimmed body acoustic sensitivity analysis, the rear suspension attachment parts showed sensitivity in the 150~200 Hz range when a load was exerted sideways. A road test road noise evaluation confirmed a correlation where rear seat noise became problematic in similar ranges.

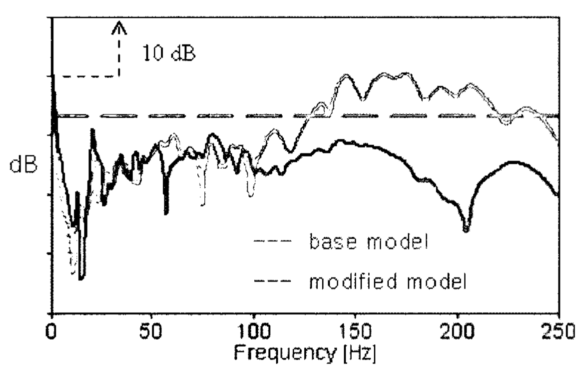
As in Figure 12, this study created a connection with the package tray side member by vertically applying vertical structural members to the rear wheelhouse inner and the rear floor joint.

Additional structural members are called package tray pass through members. When a load is exerted on the rear suspension relative to the previous structure, contribution to transmission force diffusion and panel vibration sensitivity reduction is expected to be considerable.

This study has presented directions for improving the car body structure at the rear opening and the rear wheelhouse joint through a preliminary analysis of the mother car at the initial design stage. According to an improvement effect analysis of the proto-car development vehicle, as shown in Figure 13, when the rear suspension was excited sideways, the acoustic sensitivity of the front and rear seats was reduced, thus meeting the design goal. According to a subsequent road test evaluation, a reduction in road noise reduction was confirmed around the 150 Hz range. Handling performance and product durability are thereby expected to be positively affected also.



(a) Driver ear



(b) Rear seat

Figure 13. Acoustic sensitivity by exciting the rear suspension mounting point.

5. CONCLUSIONS

To analyze the cause of high sound pressure levels of the rear seat relative to the front seat generated during the acceleration of vehicles, this study has implemented a design improvement process using a mother car at the initial design stage. The following conclusions have been obtained:

According to a baseline test of the test vehicle, 2nd-order components of the engine during acceleration exhibited problematic rear seat noise in the below 3,000 RPM and over 4,000 RPM ranges. Consequently, the rear seat requires noise improvement at 60~70 Hz, 150 Hz, 165 Hz, and 185 Hz.

According to an analysis of body attachment stiffness and acoustic sensitivity through trimmed body structural sound field coupling, the engine attachment parts, when excited, showed little contribution relative to rear seat noise. Meanwhile, the rear suspension attachment parts, showed acoustic sensitivity in a problematic frequency range when excited sideways.

This study has presented directions for improvement by identifying panel sensitive components through the ODS of sensitive frequency. Stable performance improvement before the proto-car stage has been realized. A consequent reduction in the test car stage and development period is anticipated.

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