

Construction of an Acoustic FE Model for the Purpose of Engine Noise Reduction

Kazufumi KAWABATA* Tatsuhide SAKAI* Nobuaki HASHIMOTO*

ABSTRACT

A study was conducted in the construction of an acoustic finite element model with the velocity of vibration of the trimmed body as a boundary condition. An experimental model using velocity as the boundary condition was created by measuring the vibrational acceleration of each panel of a trimmed body by excitation measurement. The trim panel impedance was also set as a boundary condition for the acoustic FE model. It was decided that the setting values for the acoustic FE model, which provides a simple representation of multilayered trim panels, should be made approximately equivalent to actual measurement values. This hybridization of a structural experimental model and an acoustic FE model made it possible to represent the acoustic characteristics of the trimmed body up to 350Hz. It also enabled analysis of methods for efficiently minimizing the vibration level of vehicle body panel parts that contribute significantly to engine noise and for reducing the vibro-acoustic transfer function from the engine mount to passenger ear locations.

1. Introduction

To further reduce development time and number of prototypes and to achieve cost reductions in automotive development, it is essential to clarify performance requirements at the beginning of development and to predict performance during the drawing stage. With respect to the body frame in particular, high level styling, collision safety, comfort, reduction of noise and vibration and weight savings must all be achieved simultaneously. Large-scale finite element (FE) models are used to investigate body structure in development to reduce noise and vibration originating from powerplant input. However, predicting effects on noise in the cabin after modifications are made to the structure is difficult with the present technology. Furthermore, because cabin noise is also greatly affected by cabin shape, the reduction of this noise requires greater control of body vibration and acoustic characteristics rather

than a simple reduction of body vibration. Therefore, high-accuracy cabin acoustic calculation models are necessary. Past research in the development of such models includes the following examples: a method for efficiently reducing acoustic sensitivity using a hybrid model composed of experimental modal body and cabin acoustic FE models ⁽¹⁾; and, for booming noises in heavy-duty truck cabs, an acoustic FE model using the results of an experimental modal analysis of cab structure as boundary conditions ^(2, 3). In addition, trimmed body FE models have been used as structural models, instead of experimental models, to predict cabin noise up to 200Hz ⁽⁴⁾. However, these models are not sufficiently accurate for analysis up to the frequencies of interest for cabin noise caused by engine vibration input.

The research described in this paper concerns creation of a hybrid acoustic calculation model capable of determining the vibro-acoustic transfer functions from the engine mounts to passenger ear positions at frequencies up

* Tochigi R&D Center

to 350Hz. This hybrid model uses an acoustic FE model and an experimental model formulated from trimmed body vibration tests. This technique was used as a developmental tool for the reduction of cabin noise.

2. Method for Creating the Experimental and Acoustic FE Models

2.1 Experimental Model

Body panel vibration, obtained from vibration measurements in an actual vehicle, was used as the velocity boundary conditions necessary for cabin noise calculations in the acoustic FE model. The rear engine mount, which has a high level of contribution to engine noise, was subjected to burst random vibrations in a trimmed body without an engine, transmission or suspension, and the vibrational acceleration of each panel was measured. Using this data, an experimental model was formulated (Fig. 1). In determining the number of measurement points, it was necessary to consider the upper frequency limits for acoustic calculations. To increase the accuracy of acoustic calculation results it is necessary to increase measurement points in the experiment and the number of elements in the FE model. However, this in turn increases the manpower required for the experiment and the time required for calculation. Therefore, the antinodal and nodal points of the sound waves were represented by establishing six points per wave length. The frequency $f_{\max}$ up to which the FE model is capable of analysis was found using the following equation.

$$f_{\max} \leq c / (6 \cdot l_{\max}) \quad (1)$$

where c is the speed of sound, and $l_{\max}$ is the maximum distance between nodal points in the FE model. Because the upper frequency limit in this project was 350Hz, the pitch for measurement points was established at 100mm using Equation (1). Vibrational acceleration was measured at approximately 1100 points on the entire body. Sound pressure at the passenger ear position for each seat was measured at the same time as the vibrational acceleration, and used as verification for the acoustic calculation results described below.

2.2 Acoustic FE Model

The acoustic FE model used is shown in Fig. 2. Body panels and surface shapes of interior fittings were extracted from a large-scale structural FE model, and a solid model for acoustic calculation was formulated (approximately 74,000 elements). In addition to trim components such as seats and an instrument panel, acoustic speakers were modeled in the cabin to accurately represent measurement conditions in the actual vehicle. Modeling of interior fittings is important for higher-accuracy cabin sound pressure calculations. Representation of these was possible by establishing impedance or admittance as boundary

conditions for the acoustic FE model. Admittance was established for the areas shown in Fig. 3. Admittance values were obtained with the following equation, using measurements of vibrational acceleration and surface sound pressure levels of trim components obtained in acoustic vibration experiments in an actual vehicle.

$$A = 1/Z = v/p \quad (2)$$

where A is admittance, Z is impedance, v is trim vibrational velocity, and p is trim surface sound pressure level. A simple model of interior fittings was formulated on the basis of measurement results for each panel. This was accomplished by using acoustic calculation software to determine boundary conditions with almost the same impedance characteristics as the measurement results (Fig. 4). When applying velocity boundary conditions to the nodal points on the FE model using panel vibration data from the experimental model, linear interpolation was used for nodes with spatial coordinates between the vibration

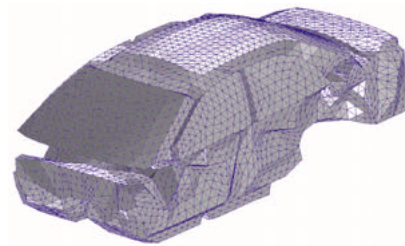


Fig. 2 Acoustic FE model

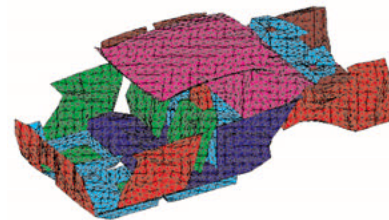


Fig. 3 Established admittance areas

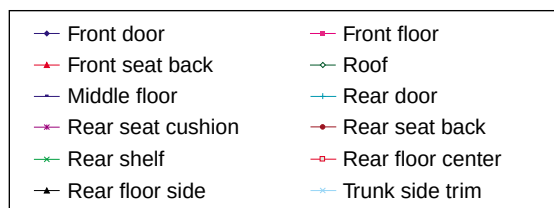
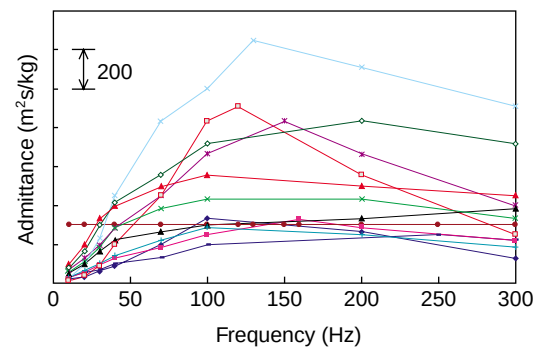


Fig. 4 Admittance values defined as boundary conditions

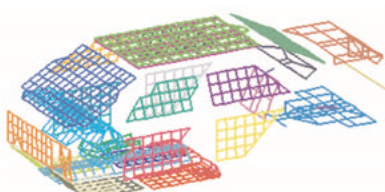


Fig. 1 Structural experimental model

data points. Finally, evaluation points for cabin sound pressure level were established at passenger ear positions in each seat corresponding to the actual vehicle measurements.

3. Verification of Calculation Accuracy of the Hybrid Acoustic Model

3.1 Verification of Velocity Boundary Conditions

3.1.1 Comparison of application of body panel vibration and trim vibration to velocity boundary conditions

Following the formulation of the experimental model, the relationship between body panel and trim vibration characteristics was analyzed. Figs. 5 and 6 compare vibration responses of trim and body panels in the floor and roof. Significant differences in vibration characteristics between trim and body panels were observed (Figs. 5 and 6). Looking at the vehicle body as a whole, there is a general tendency for trim vibration levels to become high at 300Hz or less, and for the same effect to occur in body panels in a higher frequency range.

These experimental model vibration data were then input into the acoustic FE model as a boundary condition, and the acoustic transfer function from the rear engine mount to the front seat passenger's ear positions was calculated. Individual acoustic measurements were taken for instances of panel application and for trim application in those areas where both the panel and trim data were measured, and the

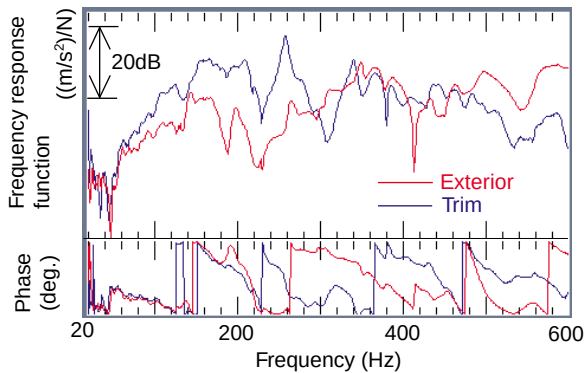


Fig. 5 Comparison of frequency response function (FRF) for the front floor upon rear engine mount excitation

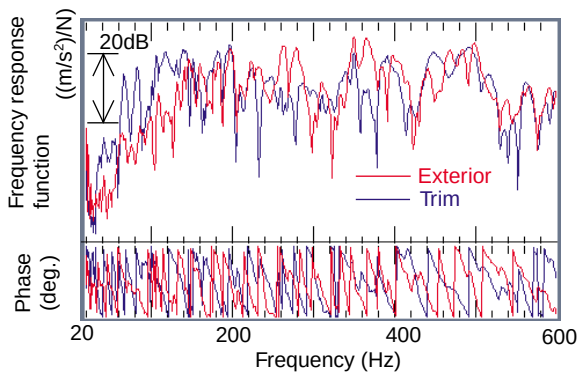


Fig. 6 Comparison of FRF for the roof upon rear engine mount excitation

effects on the acoustic transfer function to passenger ear positions was verified. Fig. 7 shows a comparison between calculated and measured vibro-acoustic transfer function values at these ear positions. It further shows that by selecting trim data as a boundary condition, the transfer function increases by 5 dB or more, particularly in the area above 200Hz, while calculated values more closely approach the actual measured values. However, despite the fact that a correlation can be observed between acoustic transfer function bottom and peak frequencies, there is a 5-10 dB difference between 120 and 240 Hz, and therefore the acoustic characteristics of actual vehicles are not accurately represented. Thus, an attempt was made to examine the disparity by calculating the effect of vibration in each panel on sound pressure levels at passenger ear positions in the acoustic FE model.

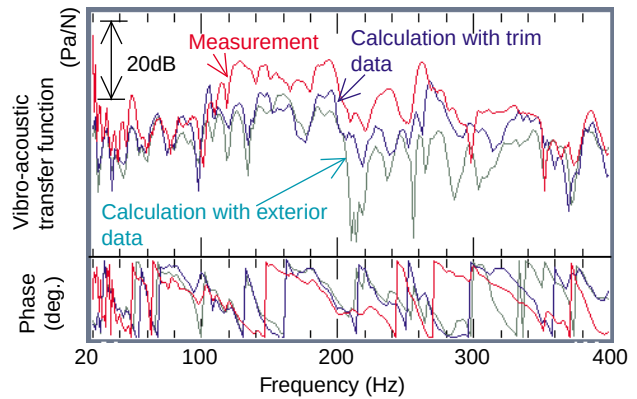


Fig. 7 Comparison of vibro-acoustic transfer function for the front/left seat upon rear engine mount excitation

3.1.2 Effect of vibration in each panel on cabin noise

A boundary element (BE) model of the same form was formulated from the FE model shown in Fig. 2, and the velocity contribution from each panel to the cabin noise was calculated, with a focus on the range between 100 and 250Hz, for which calculation accuracy was insufficient. Fig. 8 shows the acoustic transfer vector for sound pressure at passenger ear positions at 124Hz. Contribution is especially high for the dashboard lower panel, and is in reverse phase with the front window-side (instrument panel-side). Fig. 9 shows a high contribution for other frequencies at the front of the body, with the dashboard-side in reverse phase with the instrument panel-side, as was shown in Fig. 8.

Next, a method of calculating the contribution of panel vibrations to passenger ear position sound pressure in the FE model was examined. First, the boundary velocity, which up to this point had been applied to all points simultaneously, was divided for individual panels, as shown in Fig. 10, and vibro-acoustic transfer functions from the engine mount to passenger ear positions were calculated for each of the panels (Fig. 11). By recombining these individual transfer functions, vibro-acoustic transfer functions for the entire body were obtained. These functions were then compared with calculation results obtained when the boundary velocity was used for all points simultaneously (Fig.12). Fig. 11 shows a high contribution from the front of the body in the 120-200Hz range, the range for which accuracy for vibro-acoustic transfer functions at passenger

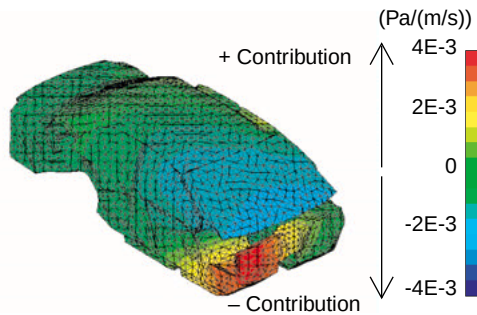


Fig. 8 Acoustic transfer vector at 124Hz

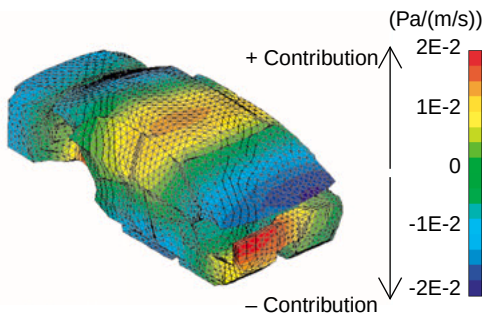


Fig. 9 Acoustic transfer vector at 182Hz

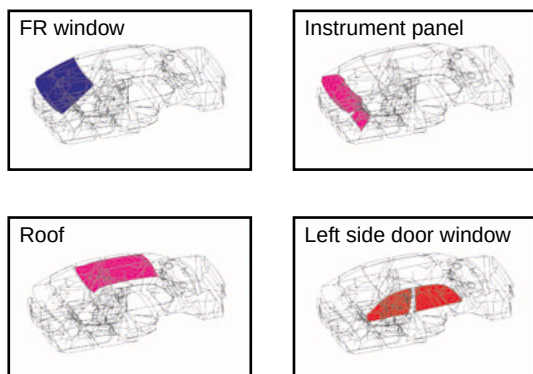


Fig. 10 Examples of set divided panels

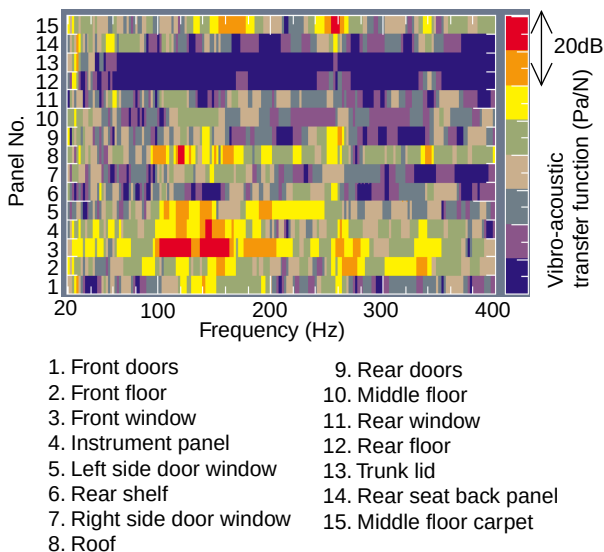


Fig. 11 Panel contribution to interior engine noise level

ear level was insufficient in calculations. This is the same tendency as demonstrated by the acoustic transfer vectors in the BE model. Fig. 12 shows that when vibro-acoustic transfer functions are calculated from the engine mount to passenger ear positions, combining individual acoustic calculation results for each panel produces better results than performing a single acoustic calculation using a single velocity boundary condition for all points.

3.1.3 Panel vibration mode phase effect

Next, the vibro-acoustic transfer function at passenger ear positions was considered, with a focus on the 120 and 190Hz peaks for which there was insufficient correlation between measured and calculated values. In particular, the boundary velocity data was re-verified for the front of the body, which has a high level of contribution to cabin noise. Front floor vibration data was calculated separately for the trim and panels, and the effect on cabin noise was verified (Fig. 13). Results demonstrate that, as shown in Fig. 5, between 100 and 200Hz the trim vibration level exceeds that of the panels, but looking at the vibro-acoustic transfer function to passenger ear positions, there is a high contribution from panels between 100 and 150Hz. Therefore, the vibration modes in this frequency band were focused on using the experimental model, and as Fig. 14

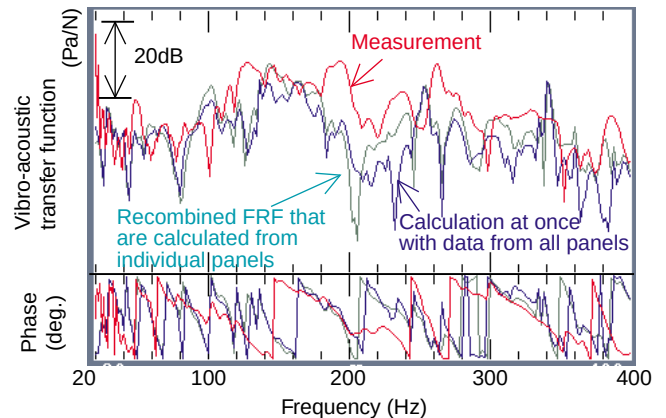


Fig. 12 Comparison of vibro-acoustic transfer function for the front/left seat upon rear engine mount excitation

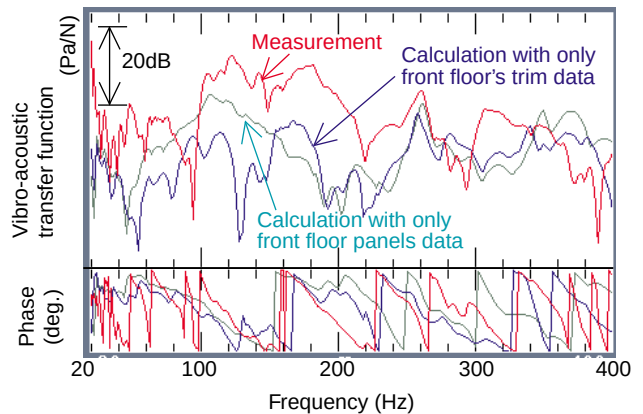


Fig. 13 Contribution of front floor vibration to vibro-acoustic transfer function for the front/right seat upon rear engine mount excitation

shows, it is clear that the front door trim did not exhibit a low order membrane vibration mode similar to the floor panels. Also, phase was unstable at each measurement point for the trim. It is believed that poor adhesion between trim materials and panels, and measurement accuracy of trim cover vibration are the cause for this difference in mode shapes. Other areas, including the middle floor, roof, trunk, etc., were also analyzed, but no major discrepancies were observed similar to the front floor situation.

Panel data for the front floor alone were then used in recalculating the vibro-acoustic transfer function to passenger ear positions. Results are shown in Fig. 15. The method applied to calculate these transfer functions was to individually apply velocity boundary conditions to each panel, and then to recombine the individually calculated transfer functions. Using front floor panel data enabled correlation between calculated and measured values to be improved between 100 and 160Hz and 220 and 240Hz. By this means it was possible to construct a structural-acoustic hybrid model representing the acoustic characteristics of an actual vehicle up to 350Hz, with the exception of the 180Hz peak.

3.2 Verification of Accuracy of the Acoustic FE Model

3.2.1 The acoustic transfer vector for cabin noise

The lack of correlation between measurements and calculations in the peak at 180Hz shown in Fig. 15 is considered from the perspectives of acoustic modes and acoustic transfer vectors calculated in the acoustic FE model. Fig. 16 shows calculation results for acoustic transfer vectors at 182Hz. It can be seen that the contributions of the dashboard lower panel and the front border of the front window, in particular, are both high (almost the same level), and are in reverse phase. Fig. 17 shows the acoustic mode at 183Hz calculated in the acoustic

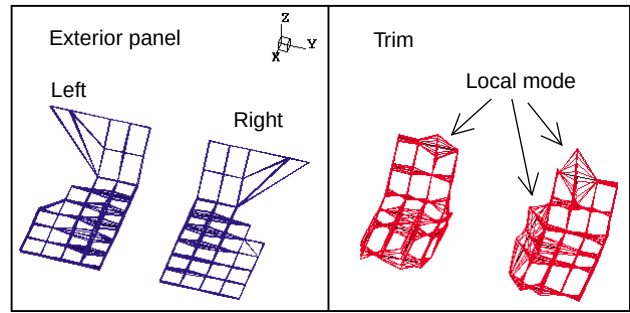


Fig.14 Running mode of front floor at 142Hz

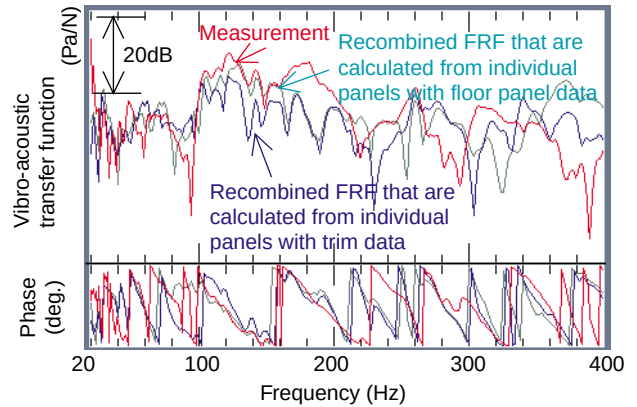


Fig. 15 Effect of using the floor panel data on vibro-acoustic transfer function for the front/right seat upon rear engine mount excitation

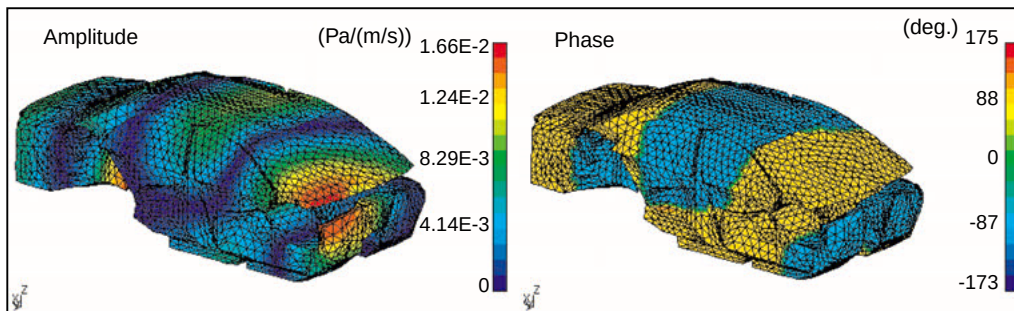


Fig. 16 Acoustic transfer vector to front/right seat at 182Hz

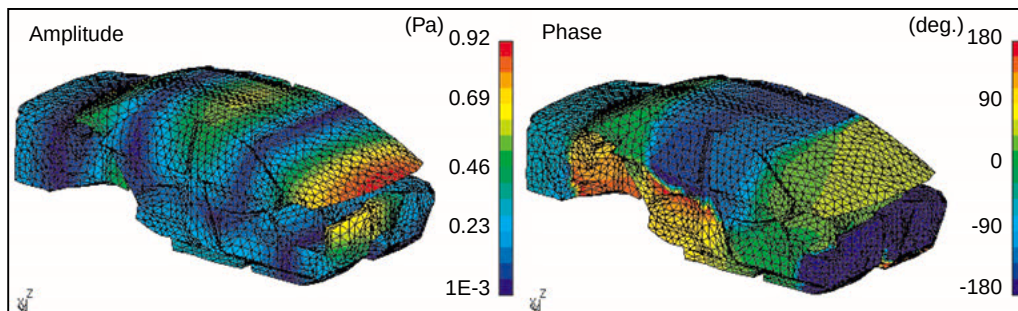


Fig. 17 Calculated acoustic mode (183Hz)

FE model. Figs. 16 and 17 show the same tendencies in amplitude level patterns. This is thought to be because acoustic modes exert a strong influence on acoustic transfer vectors.

3.2.2 Comparison of measured acoustic-acoustic transfer functions with calculated values

This stage was to verify whether acoustic characteristics calculated in the acoustic FE model accurately represent the acoustic characteristics of a real vehicle. Comparison and verification were conducted using the acoustic-acoustic transfer function H from passenger ear position sound pressure level to cabin boundary surface sound pressure level. As shown in Fig. 18, acoustic speakers were placed at passenger ear positions in a real vehicle, and the cabin boundary surface sound pressure level was measured. Using these measurements, H is derived with the following equation

$$H = P/\dot{Q} \quad (3)$$

where P is cabin boundary surface sound pressure and Q is volume acceleration produced by acoustic speakers. Sound pressure measurement points were kept to one or two per panel, with 35 in total in the cabin. The boundary velocity v as determined by equation (4) should be established for the speaker section in the acoustic FE model.

$$\begin{aligned} P/\dot{Q} &= P/i\omega Sv \\ v &= \dot{Q}/i\omega S \end{aligned} \quad (4)$$

where ω is angular frequency and S is the area of the speaker opening.

Fig. 19 shows measurement results for acoustic-acoustic transfer function H . It can be seen that the level is high at the front of the body for the 160-200Hz range in which correlation was poor between calculated and measured values for passenger ear positions, as shown in Fig. 15. This corresponds to the areas with a high contribution to

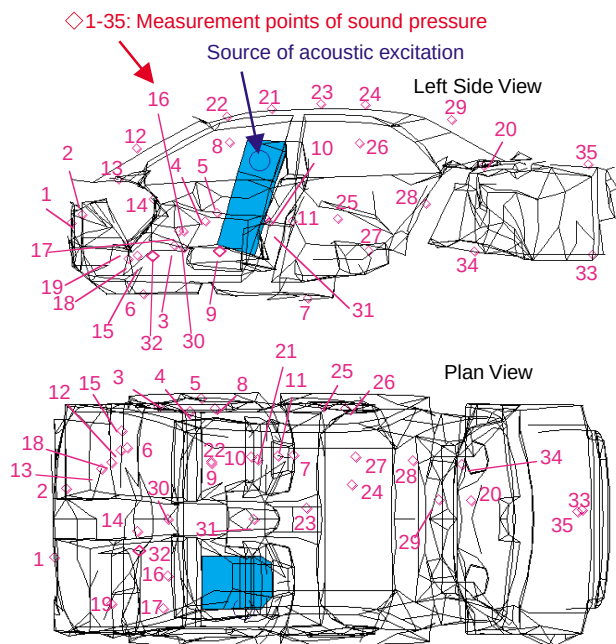


Fig. 18 Source of acoustic excitation and measurement points of sound pressure

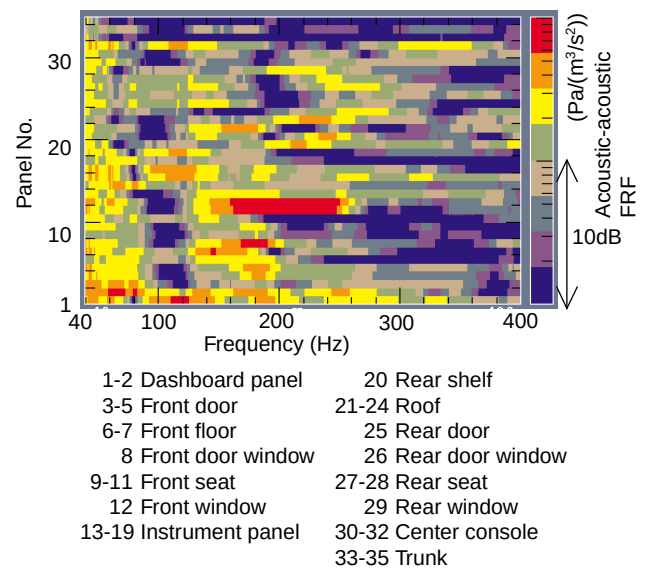


Fig. 19 Experimental acoustic-acoustic FRF

sound pressure shown in Fig. 16. Comparisons between calculated and measured values for acoustic-acoustic transfer function H with a focus on the front of the body are shown in Figs. 20 and 21. These figures show that correlation between calculated and measured values is poor, and in particular the calculation model is incapable of representing the 180Hz peak at the dashboard panel. These results demonstrate that there are some inaccuracies in the

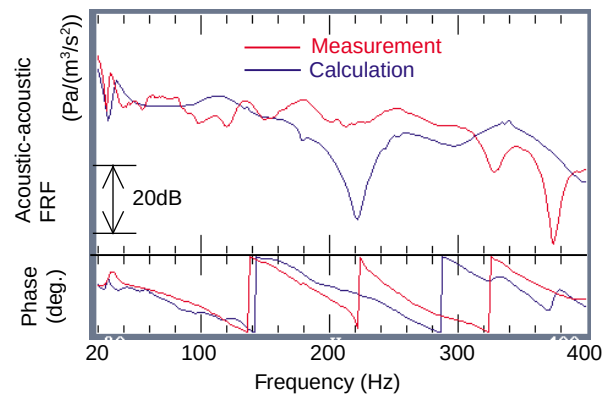


Fig. 20 Comparison of acoustic-acoustic FRF for the dashboard lower panel

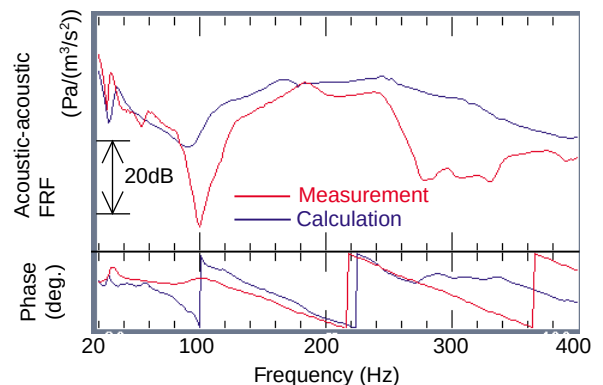


Fig. 21 Comparison of acoustic-acoustic FRF for the front window

acoustic FE model for the front of the body, specifically, the area between the front window, instrument panel and dashboard panel. Because the steering column, hanger, pedals, air-conditioning and heating units and ducts functioning as ventilation channels are fitted in this area, it is difficult to adequately represent the form of the cabin in the FE model. However, unlike the BE model, complex fluid material characteristics can be established by section in the FE model. Therefore, it was examined whether this modeling feature could be used to establish fluid material characteristics different from air characteristics and so achieve a representation of the acoustic characteristics within the dashboard of an actual vehicle.

3.2.3 Improvements in acoustic model calculation accuracy by changing volume densities

Volume densities different to air were established in the areas shown in Fig. 22, in an attempt to improve the accuracy of the acoustic FE model. Fig. 23 shows a comparison of acoustic-acoustic transfer function H in the area around the lower part of the instrument panel before and after the volume density changes. It can be seen that changes to the volume density reduced differences in level between calculated and measured results, particularly in the area above 250 Hz, and increased the accuracy of the acoustic FE model. However, no improvement in H between 160 and 200 Hz was observed. This range presented difficulties for the calculation of the vibro-acoustic transfer function from engine mounts to passenger ear positions. It is clear that an improvement in accuracy cannot be expected from a change in volume density in only one area.

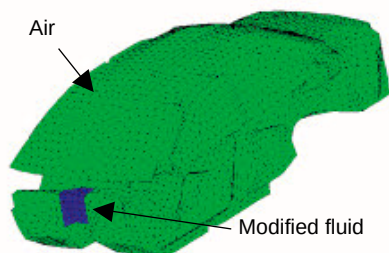


Fig. 22 Area of modified fluid properties on the FE model

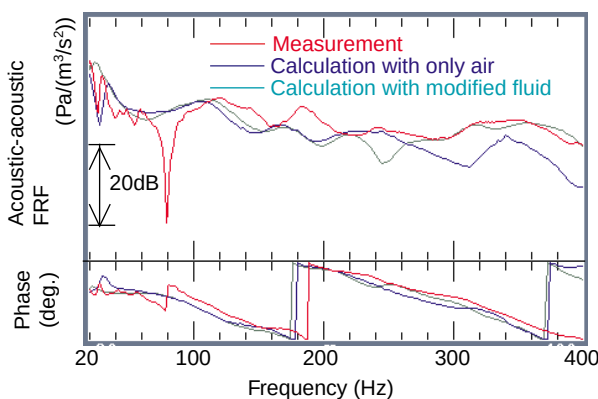


Fig. 23 Effect of fluid properties modification on acoustic-acoustic FRF under the instrument panel

4. Conclusion

An experimental model formulated from vibrational acceleration measurements in an actual vehicle and an acoustic FE model were used in the construction of a hybrid acoustic calculation model capable of analyzing vibro-acoustic transfer functions from engine mounts to passenger ear positions up to 350 Hz. This model can be used as a developmental tool to reduce cabin noise due to engine vibration input. As a result, technological difficulties and methods necessary to improve the accuracy of the model have been clarified.

- (1) As a result of examining the use of experimental data as velocity boundary conditions in the acoustic FE model, it is clear that trim vibration data should be used rather than panel vibration data.
- (2) To prevent boundary velocity interpolation errors in the acoustic FE model, a method was examined in which boundary velocity was divided into individual panels and the vibro-acoustic transfer functions obtained for each by acoustic calculations were recombined. Correlation between calculated and measured values was improved in comparison with the method in which a single boundary velocity is used for all points simultaneously and transfer functions are calculated once. This method enabled the construction of a hybrid model capable of representing the acoustic characteristics of an actual vehicle up to 350 Hz, with the exception of the 180 Hz peak.
- (3) To improve representation of the front of the body, which contributes greatly to cabin noise, a method for establishing a volume density different to air was attempted for a portion of the FE model. As a result, there was an improvement in the discrepancies between calculated and measured values of acoustic-acoustic transfer functions from passenger ear positions to the cabin boundary surface. It was demonstrated that the accuracy of the acoustic FE model could be improved.

However, to achieve further improvements in model accuracy, it will be necessary to reduce discrepancies in level between calculated and measured values for the acoustic-acoustic transfer functions throughout the cabin. To achieve this, sound pressure measurement points for actual vehicle acoustic excitation should not be limited to representative points on each panel. Rather, a more detailed experimental acoustic modal model must be formulated, and volume density settings optimized across the entire acoustic FE model through correlation between experimental modes and those calculated in the acoustic FE model.

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References

- (1) Kimura, Keisuke: Technical Development Using the Structural and Acoustic Hybrid Model for Reducing Vibro-acoustic Transfer Function, Society of Automotive Engineers of Japan, Inc., 1998 Fall Conference, Technical Lecture Preprint Collection, No.78-98, p.1-4(1998)
- (2) Kurokawa, Isamu: Reduction of Booming Noise in Heavy Duty Truck Cab – Part 1 Contribution Analysis of Floor Panel Vibration, Society of Automotive Engineers of Japan, Inc., 1998 Fall Conference, Technical Lecture Preprint Collection, No.78-98, p.5-8 (1998)

- (3) Kurokawa, Isamu: Reduction of Booming Noise in Heavy-duty Truck Cab – Part 2 Reduction of Booming Noise by Floor Structure Modifications, Society of Automotive Engineers of Japan, Inc., 1998 Fall Conference, Technical Lecture Preprint Collection, No.78-98, p.9-12 (1998)
- (4) Sung, Shung H., et al.:Development and Experimental Evaluation of a Vehicle Structural Trimmed-Body Model, Proceeding of the 1999 Noise & Vibration Conference, P-342, Vol.2, p.1135-1140 (1999)

■ Authors ■



Kazufumi KAWABATA



Tatsuhide SAKAI



Nobuaki HASHIMOTO