

Identification of Mechanism of Transmission of Flow-induced Noise through Glass for Vehicle Interior Noise Reduction

Kei AMBO*

Kazuya MORISHITA*

Takashi YOSHINO*

Minoru TERAMURA*

ABSTRACT

Seeking to identify the target for control to reduce vehicle interior noise, the research discussed in this paper determined the contribution ratio of convectional pressure and acoustic sound to high-frequency flow-induced noise (1 kHz-5 kHz) in the interior of a box modeling a vehicle interior, based on a model of cabin sound transmission from the pillar modeling the flow around the A-pillar. This study showed that the contribution ratio of acoustic sound to the overall value in the high-frequency range at the microphone in the box was approximately 93%, indicating that in the high-frequency range, acoustic sound is dominant in comparison to convectional pressure.

A comparison of the sound pressure distribution on the external surface of the front side glass of an actual vehicle, as estimated from surface pressure fluctuation and far-field sound, with the sound pressure distribution on the internal surface, as estimated from the vehicle interior sound level, showed that the difference between the sound pressure on the external and internal surfaces is almost exactly equal to the glass transmission loss. This result demonstrated that acoustic sound generated outside the vehicle is also dominant in sound transmission through the side glass in an actual vehicle, indicating that the reduction of acoustic sound generated outside the vehicle would be an effective means of increasing the comfort of the vehicle cabin in relation to high-frequency flow-induced noise.

1. Introduction

The achievement of increased fuel efficiency, looking towards the realization of a low-carbon society, and the achievement of increased driving stability and comfort as vehicles become faster are necessary elements in the further development of the automobile. The reduction of vehicle interior noise is one aspect of achieving increased automotive comfort. Sound sources for vehicle interior noise include noise from the engine, exhaust, and drive power transmission mechanism; road noise from the tires and the vibration of the suspension; noise from heating, ventilation and air conditioning (HVAC) systems; and flow-induced noise produced by air flow separation around each part of the vehicle. The electrification of powerplants is proceeding in order to realize increased fuel efficiency, and engine system noise is tending to decline as a consequence. In the FIT EV, for example, the ratio of flow-induced noise transmitted to the interior has increased with increased vehicle speed.

One of the technological issues in attempting to reduce flow-induced noise is determining how the noise is

generated and how it comes to be heard inside the vehicle; it is necessary to identify the target for control in order to reduce vehicle interior noise. Among flow-induced noise, high-frequency flow-induced noise (1 kHz-5 kHz) generated around the A-pillar⁽¹⁾, including the door mirrors, is considered particularly important. This flow-induced noise is thought to be transmitted through the glass to the interior of the vehicle, but the mechanism of its generation and transmission has not been fully clarified.

The excitation force on the glass when noise is transmitted into the vehicle interior is thought to be made up of convectional pressure acting on the glass and acoustic sound generated by the fluctuations of vortices acting on the glass. In order to identify the control target for reducing vehicle interior noise, it is important to determine the contribution ratio of this convectional pressure and acoustic sound. Using wind tunnel tests of a vehicle interior noise measurement model fitted with a glass plate, Iida et al. showed that acoustic sound possess the property of passing readily to the interior of the vehicle⁽²⁾. However, the sound source was random noise, and this research did not identify the contribution ratio of convectional pressure and acoustic

* Fundamental Technology Research Center

sound originating in aerodynamic phenomena to vehicle interior noise.

The present research therefore sought to identify the control target for reducing vehicle interior noise. First, using a model of cabin sound transmission from the pillar modeling the flow around A-pillar that modeled the flow around the pillar (Fig. 1), the research identified the contribution ratio of convectional pressure and acoustic sound to vehicle interior noise when high-frequency flow-induced noise is transmitted to the cabin. Following this, it conducted a comparison of vehicle interior and vehicle exterior noise in an actual vehicle.

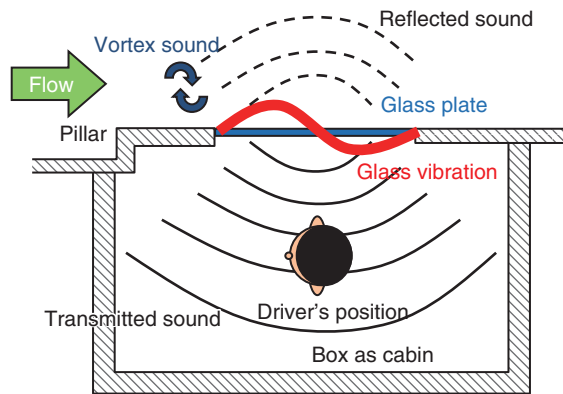


Fig. 1 Model of cabin sound transmission from pillar modeling flow around A-pillar

2. Determination of Glass Transmission Characteristics in Wind Tunnel Tests

Tests on the model of cabin sound transmission from the A-pillar were conducted in a closed loop low-noise wind tunnel of the specifications shown in Table 1 and Fig. 2(a). The model used a step, modeling a pillar, which connected the lower surface of the wind tunnel nozzle and the ground plate [Fig. 2(b)]. A glass plate was positioned on the downstream side of the step. As shown in Fig. 2(c), this glass plate had the same surface area as the front side windows in the FIT HYBRID. A box modeling a vehicle interior was positioned underneath the glass plate. To help eliminate the effect of acoustic resonance in the interior of the box, the entire box, other than the glass plate forming its roof, was covered in acoustic material to create an anechoic chamber. Measurements were conducted under the conditions shown in Table 2. Multi-field microphones, able to produce accurate measurement results in both reverberant and anechoic chambers, were used to measure far-field sound at a position of 1 m above the step as well as sound within the box (Fig. 3).

As Fig. 4 shows, differences in the thickness of the glass produce differences in the frequencies and sound pressure inside the box, indicating a different mechanism of transmission through the glass. In addition, while it was

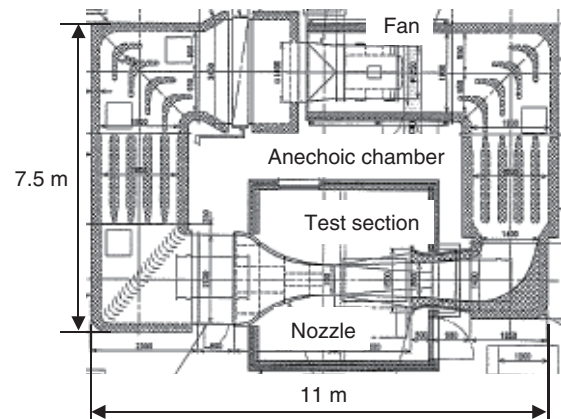
Table 1 Wind tunnel specifications

Maximum velocity	50 m/s
Background noise	54 dBA (30 m/s)
Nozzle exit size	$0.7 \times 0.7 \text{ m}^2$
Anechoic chamber size	$4 \times 4 \times 3 \text{ m}^3$

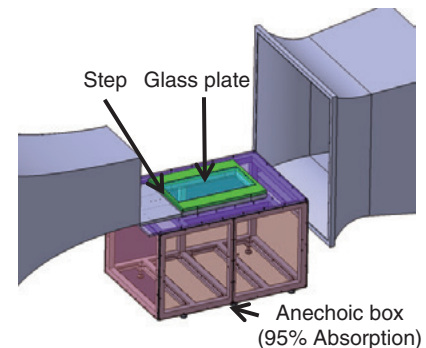
Table 2 Measurement conditions

Step height, H [mm]	30
Nozzle exit velocity, V [m/s]	33
Glass thickness [mm]	3, 4, 5
Microphone sampling frequency [Hz]	32000

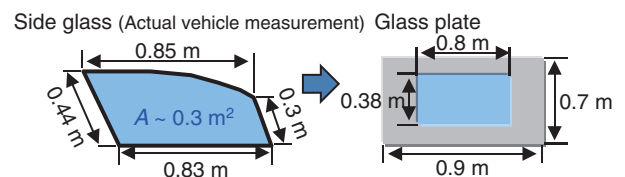
predicted that the vibration of the glass would change with its thickness, the fact that no change was observed in far-field sound for different thicknesses of glass indicates that the influence of vibration of the glass on far-field sound and the flow field is minimal.



(a) Schematic image of low-noise wind tunnel facility



(b) Schematic view of WT set-up



(c) Dimensions of glass plate

Fig. 2 Experimental set-up

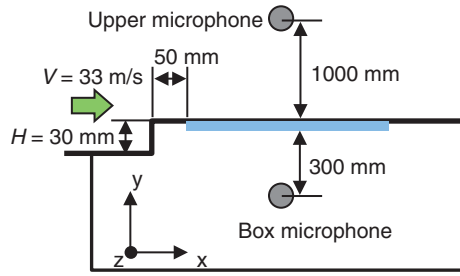


Fig. 3 Microphone positions

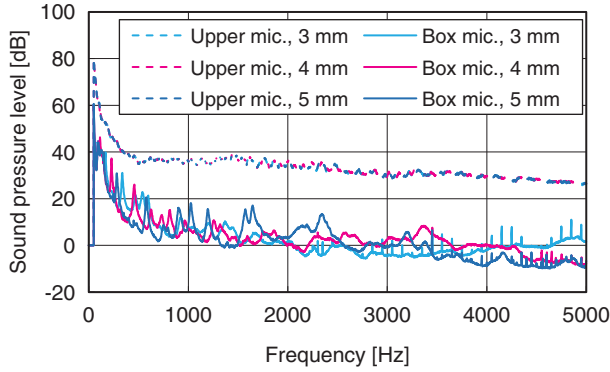


Fig. 4 Experimental results

3. Contribution ratio of Convective Pressure and Acoustic Sound

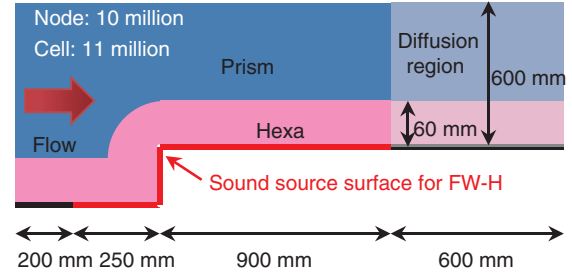
3.1. Method of Identification of Aeroacoustic Sound Source

Powell's theory of vortex sound locates the aeroacoustic sound source in the fluctuations of vortices generated in the vicinity of physical bodies⁽³⁾. Research has been conducted on the use of a statistical method (Stochastic Noise Generation and Radiation) in the identification of sound sources in order to reduce calculation cost⁽⁴⁾, but the present research employed incompressible unsteady Large-Eddy Simulation (LES) in order to study the sound source, the vortex motion of the separation flow, in greater detail. A hexa mesh, consisting mainly of hexa and prism elements, was used in the analysis (Fig. 5). Table 3 shows the CFD calculation conditions.

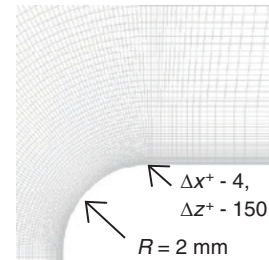
Far-field sound 1 m above the step was predicted based on pressure fluctuations in the sound source surface shown in Fig. 5(a) using the Ffowcs Williams-Hawkings (FW-H) method. Because the width in the direction of the span differed in the analysis and the test, the sound pressure in the analysis results was corrected using a correction formula proposed by Kato, et al.⁽⁵⁾ Figure 6 shows a comparison of the predicted and measured values for the frequency distribution of far-field sound, in addition to the instantaneous vortex distribution on the hexa mesh. Figure 6(a) shows that the results for far-field sound obtained using the FW-H method correspond well with the

Table 3 CFD calculation conditions

Software	Fluent ver. 13.0 SP2
Inlet velocity, V [m/s]	33
Step height, H [mm]	30
Reynolds number [-]	66000

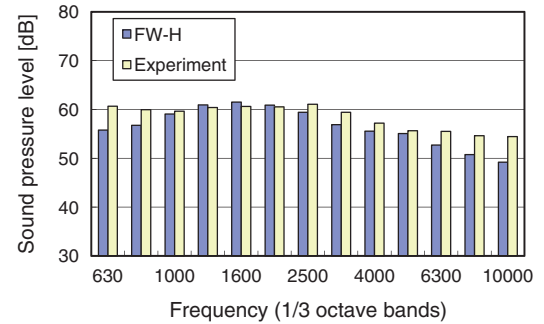


(a) Hexa/Prism mesh (Hexa mesh)

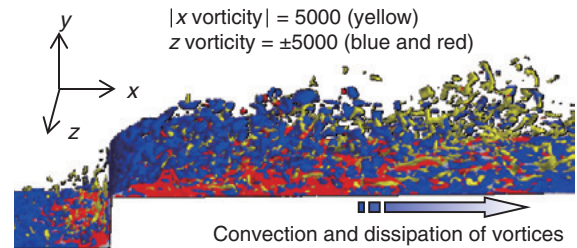


(b) Hexa mesh (enlarged view)

Fig. 5 CFD mesh



(a) Far-field sound spectra at 1 m above step



(b) Instantaneous vortex distribution

Fig. 6 CFD results

values obtained from the tests in the 1 kHz-5 kHz high-frequency flow-induced noise band. Figure 6(b) shows that directly following air-flow separation by the step, numerous horizontal vortices with Z-vorticity, (shown as blue and red) are produced, while following reattachment, numerous vertical vortices (yellow) with X-vorticity are produced, and the number of horizontal vortices declines.

3.2. Computational Aeroacoustic Analysis

Next, in order to identify the generation and propagation of acoustic sounds in the flow field using a decoupled solution, a computational aeroacoustics analysis (CAA)⁽⁷⁾, in which the Lighthill equation⁽⁶⁾, the governing equation, was discretized and solved using the finite element method, was employed. As shown in Fig. 7, an analytic mesh was generated in part of the flow field; a Lighthill tensor obtained from unsteady LES that extended the width in the direction of the span to 190 mm using hexa mesh was mapped as the sound source, and the status of generation and propagation of sound in the frequency domain was analyzed. A maximum mesh size of 5 mm was used in the CAA analysis, and the resolution for wavelengths of 5000 Hz was 10 points or more. Using infinite elements for the boundaries in the directions of the circumference and span, far-field sound was predicted using fifth-order extrapolation. Table 4 shows the calculation conditions for the CAA analysis.

Figure 8 shows a comparison of predicted and measured values for the directivity of far-field sound in the high-frequency range, in addition to the sound pressure distribution close to the step. Figure 8(a) shows that the results of predictions using CAA analysis display a better correspondence with results obtained from tests than predictions based on the FW-H method. This can be considered to result from the fact that CAA analysis is able to more accurately study the generation of quadrupole sound by the fluctuations of the vortices and its process of propagation as dipole sound due to reflection and scattering at boundaries with physical body⁽⁸⁾ than the FW-H method. An area of high acoustic pressure is conspicuous at the front of the region of air-flow separation at the step in Fig. 8(b). This area of high acoustic pressure corresponds to the area in which numerous horizontal vortices with Z-vorticity were shown to be produced in the comparison shown in Fig. 6(b).

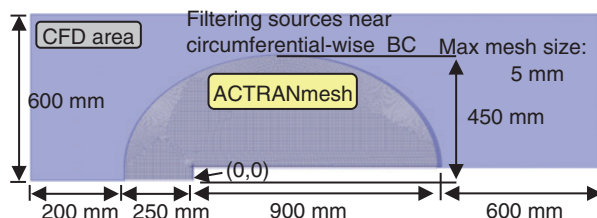


Fig. 7 CAA analysis

Table 4 CAA calculation conditions

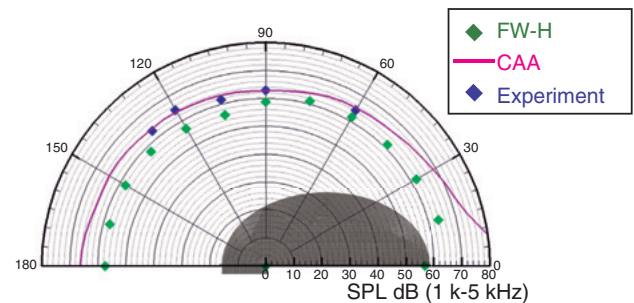
Software	ACTRAN/AA ver. 11.1
Sound source	Lighthill-Volume
Maximum frequency	$f_{\max} = 10752$ Hz
Frequency filter	$\Delta f = 14.3$ Hz

Table 5 VA analysis calculation conditions

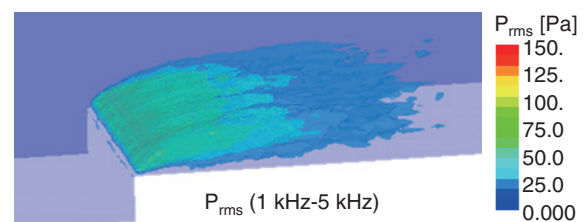
Software	ACTRAN/VA ver. 11.1
Vibration source	Acoustic and/or convectional pressure
Glass density	2500 kg/m ³
Glass Young's modulus	7.16×10^{10} (damping 3%)
Fixed point modeling as spring	Stiffness 90000 N/m
Maximum mesh size	5 mm (glass), 9.6 mm (air)
Maximum frequency	$f_{\max} = 10752$ Hz
Frequency filter	$\Delta f = 14.3$ Hz

3.3. Method of Analysis of Vibro Acoustic

Far-field sound, the sound pressure distribution within the area under analysis, and the excitation force acting on the glass were determined from the CAA analysis discussed above. Here, a vibroacoustic (VA) analysis was conducted, as shown in Fig. 9. The glass excitation force was mapped as a vibration input, and the frequency response of the glass and a CAA analysis of the interior of a box modeling a vehicle interior were coupled for analysis. The influence of the coincidence effect, in which transmission loss declines at specific frequencies, was considered in the analysis of the frequency response of the glass. Prior to this analysis, the



(a) Far-field sound direction at 1 m from step



(b) Near-field sound distribution

Fig. 8 CAA results

vibration mode of the glass and its acoustic transmission loss, as determined from the results of basic unit tests, were calibrated using the glass constraint conditions and damping rate as parameters⁽⁹⁾. Table 5 shows the calculation conditions for the VA analysis. The mesh size for the space inside the box was set as the wavelength of the frequency in the glass normal direction only, divided into six parts; infinite elements were used for boundary conditions, and the effect of rebound of interior sound was ignored.

Figure 10 compares predicted and measured values for the frequency distribution of far-field sound and box-interior sound. The excitation force employed included the convectional pressure and acoustic sound determined in the CAA analysis. While multiple samples were not used for the CAA analysis, and statistical variations therefore occur for each frequency, Fig. 10 shows that the results of prediction of sound pressure levels correspond well with measurement results. Table 6 shows a comparison of predicted and measured overall (OA) values (the total of sound pressure levels in the high-frequency range) for far-field and box-interior sound.

Table 6 shows that the sound pressure for the microphone positioned in the box in the CAA analysis is around 2 dB lower than the measured value, but because the same tendency can be observed for far-field sound, the main factor producing this result is thought to be the fact that the numerical analysis exclusively evaluates sound ideally transmitted from an aeroacoustic sound source, while the

Table 6 Overall sound pressure level (1 kHz-5 kHz) at upper and box microphone

Microphone location	CAA [dB]	Experiment [dB]
Upper (far-field) mic	65.1	67.1
Box mic	36.6	38.9
Δ SPL (Transmission loss)	28.5	28.2

test results incorporate the effect of environmental noise. In addition, there is a good correspondence between values predicted using CAA analysis and measured values for the difference between far-field and box-interior sound (Δ SPL), which expresses glass transmission loss. This is considered to demonstrate that calibration based on the results of the basic unit tests of the glass plate made it possible to conduct an accurate VA analysis.

Verification of the method for identification of the aeroacoustic sound source and the CAA and VA analysis methods, as described above, demonstrated the realization of a level of analytic accuracy making it possible to achieve a good quantitative correlation with test results for high-frequency flow-induced noise, creating the potential for identification of the contribution ratio of convectional pressure and acoustic sound to noise in the vehicle interior using these technologies.

3.4. Determination of Contribution Ratio of Convectional Pressure and Acoustic Sound

The contribution ratio of convectional pressure and acoustic sound to sound pressure inside the box was determined by conducting a VA analysis of the type discussed in Section 3.3 while varying the excitation force acting on the glass.

Because the excitation force acting on the glass in Section 3.1 (Fig. 11) was determined using incompressible unsteady LES, it represents the excitation force due exclusively to convectional pressure. The excitation force acting on the glass in Section 3.2 (Fig. 12) is thought to incorporate acoustic excitation force produced by convectional pressure and acoustic sound. This is because a wave equation (the Lighthill equation) derived from the Navier-Stokes equation, the governing equation of the flow,

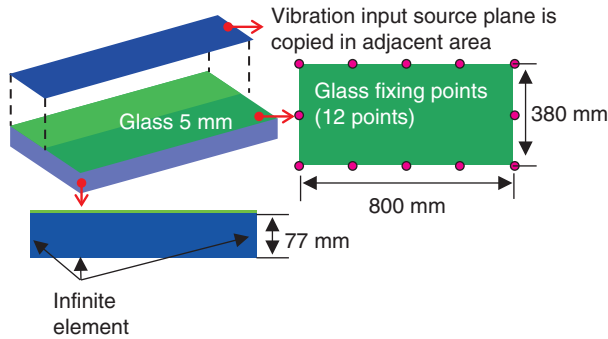


Fig. 9 VA analysis

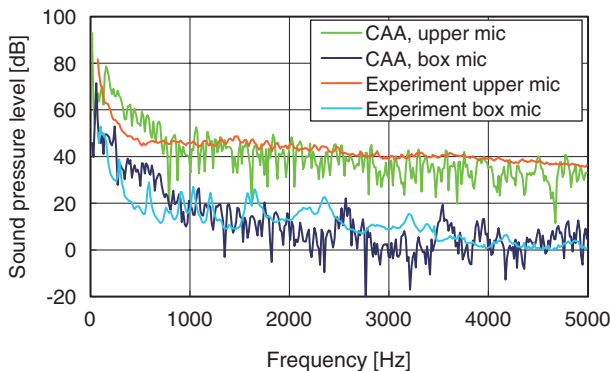
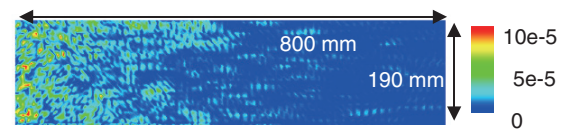


Fig. 10 Sound spectra at upper and box microphone positions (Glass 5 mm)



(a) 1 kHz point load (N)



(b) 4 kHz point load (N)

Fig. 11 Convectional pressure load (Glass 5 mm)

and a continuity equation, was employed as the governing equation. The pressure fluctuations in Section 3.1 and 3.2 were employed as the excitation force input for Section 3.3, and the box-interior sound in response to differences in the pressure fluctuations of the glass surface was determined.

In the case of box-interior sound when convective pressure is used as the excitation force, the difference between calculated and measured values tends to become greater as the frequency increases (Fig. 13).

Determination of the difference between box-interior sound when the acoustic excitation force due to convective pressure and acoustic sound is used as the glass excitation force and when convective pressure is used as the glass excitation force made it possible to define the contribution ratio of acoustic sound. Figure 14 shows the frequency distribution for the contribution ratio of acoustic sound and the contribution ratio of acoustic sound at the OA value in the high-frequency range for the microphone in the box. In Fig. 14(a), while convective pressure displays a contribution rate of around 10% between 1.0 kHz and 1.6 kHz, in the higher-frequency 2.5 kHz-5.0 kHz range, its contribution ratio is only several percent, and the contribution ratio of acoustic sound becomes dominant. In Fig. 14(b), the contribution ratio of acoustic sound at the OA value is approximately 93%, indicating that acoustic sound is dominant in the high-frequency range.

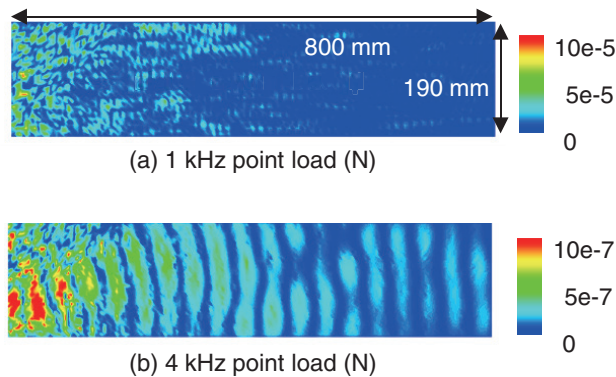


Fig. 12 Acoustic and convective pressure load (Glass 5 mm)

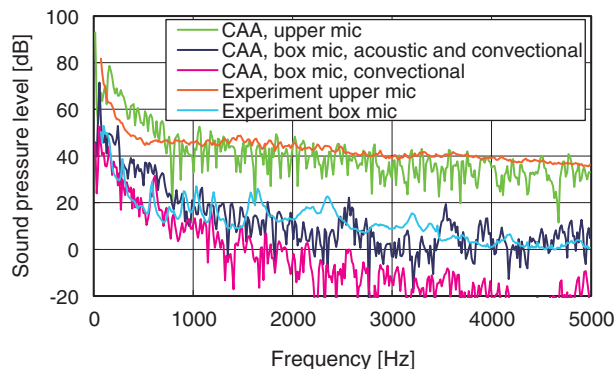
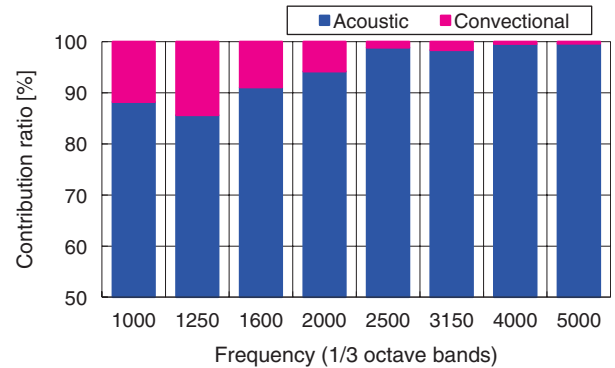
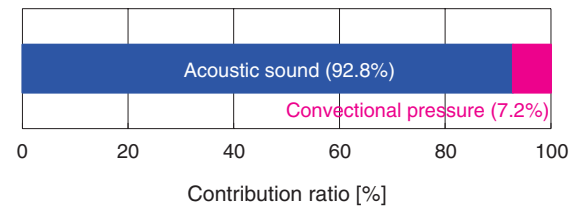


Fig. 13 Sound spectra at upper and box microphone positions

Figure 15 shows a sound propagation diagram⁽¹⁰⁾ using pressure contours at 4 kHz, including the glass and the step. This figure also demonstrates that acoustic sound generated close to the step is propagated through the air, and transmitted into the interior of the box with no phase difference.



(a) Contribution ratio for 1/3 octave bands



(b) Overall sound pressure level (1 kHz-5 kHz)

Fig. 14 Contribution ratio of acoustic sound

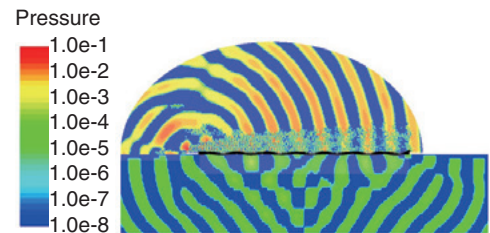


Fig. 15 Sound propagation at 4 kHz (Sound pressure [Pa])

4. Simultaneous Measurement of Interior and Exterior Acoustic Sound in an Actual Vehicle

Nouzawa, et al. studied the effect of acoustic noise in each frequency band on the vehicle interior⁽¹¹⁾, but research has not focused on sound generated by flow-induced noise passing through the side glass as a means of increasing interior comfort with respect to high-frequency flow-induced noise. The present research therefore attempted to clarify the relationship between the acoustic pressure at the glass surface, estimated from the far-field external noise, the pressure fluctuation on the glass surface, and the noise transmitted to the interior of the cabin.

Here,

P_s : Convictional pressure and acoustic sound

P_f : Acoustic sound

P_i : Sound pressure on the surface of the glass in the vehicle interior

Table 7 shows an overview of the measurement equipment. Values measured using surface microphones attached to the external surface of the glass (Fig. 16) were used for P_s ; for P_f , the acoustic pressure distribution at the side of the vehicle calculated using the beamforming method [Fig. 17(b)] from noise measurements taken by a planar microphone array positioned at a distance [Fig. 17(a)] was used. In order to evaluate sound transmitted to the vehicle interior, noise measurements were conducted using a spherical microphone array [Fig. 18(a)], and the distribution of P_i was calculated using the spherical beamforming method [Fig. 18(b)]. As Fig. 16(a) shows, the surface microphones were attached to the glass discretely. Because of this, for each physical quantity (P_s , P_f , and P_i), the values measured at the corresponding surface microphone measurement points (#1-#16) for two plates of side glass (plate thickness: 3.5 mm) fixed in place using different methods were made representative values, and their averages were used.

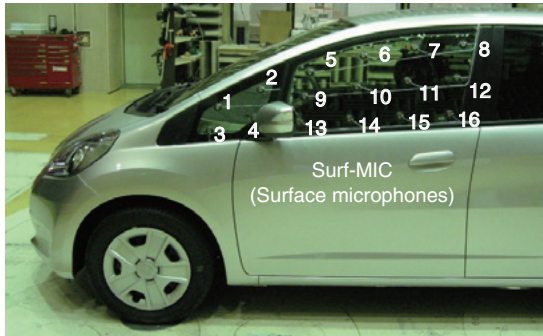
The relationship between the acoustic pressure at the surface of the side glass, estimated from the far-field external noise, the pressure fluctuation on the glass surface, and the noise transmitted to the interior of the cabin was determined using the following method: The values produced by subtracting the dB figures for P_i from the dB

Table 7 Experimental equipment

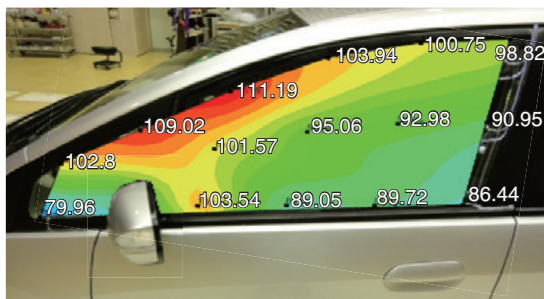
Manufacturer	Brüel & Kjær Sound & Vibration Measurement A/S
16-ch surface microphones (Surf-MIC)	4948 and 4949
42-ch half-wheel array microphones (BF)	Type 8608
50-ch spherical array microphones (SBF)	Type 8606

figures for P_s and P_f , respectively, and the transmission loss figures for the glass plates used in Chapter 2 (plate thickness: 3 mm and 4 mm) were compared (Fig. 19). The results for the transmission loss of the glass plates and the difference between P_f and P_i ($P_f - P_i$) are similar. By contrast, the results for the difference between P_s and P_i ($P_s - P_i$), which are equal to convictional pressure, are higher than the figures for glass transmission loss. Based on this result, it can be conjectured that there is almost no transmission of convictional pressure to the vehicle interior as sound pressure fluctuations.

Both numerical analysis and vehicle tests showed that with regard to the transmission of noise through automotive window glass, it is the acoustic excitation force produced by acoustic sound, P_f , which is transmitted to become sound in the cabin due to the glass transmission loss.



(a) Image of set-up



(b) Contour plot of sound pressure distribution (dB)

Fig. 16 Surface microphones



(a) Image of set-up

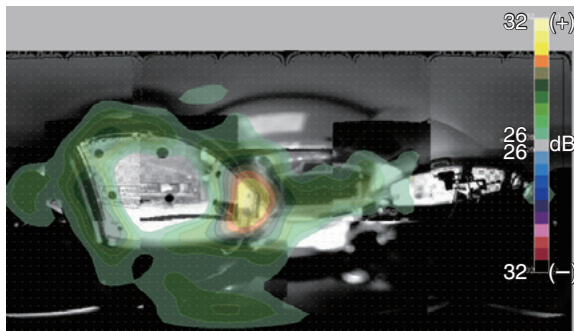


(b) 1/3-octave sound intensity map

Fig. 17 Beamforming



(a) Image of set-up



(b) Noise distribution within vehicle cabin

Fig. 18 Spherical beamforming

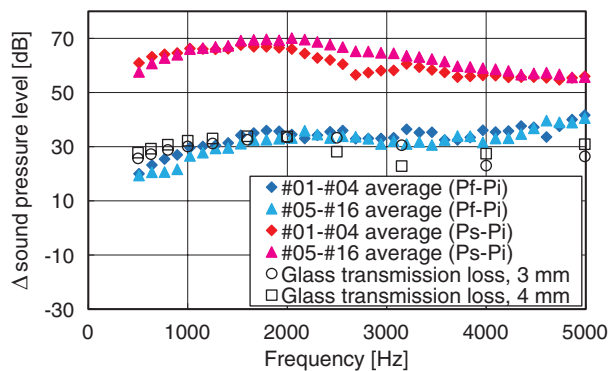


Fig. 19 Differences in sound spectra

5. Conclusion

Seeking to identify the target for control to reduce vehicle interior noise, the research discussed in this paper determined the contribution ratio of convectional pressure and acoustic sound to high-frequency flow-induced noise (1 kHz-5 kHz) in the interior of a box modeling a vehicle interior, based on a model of cabin sound transmission from the pillar modeling the flow around the A-pillar. This study showed that the contribution ratio of acoustic sound to the

OA value in the high-frequency range at the microphone in the box was approximately 93%, indicating that in the high-frequency range, acoustic sound is dominant in comparison to convectional pressure.

A comparison of surface pressure fluctuations on the front side glass of an actual vehicle, the sound pressure distribution on the external surface of the front side glass, as estimated from far-field sound, and the sound pressure distribution on the internal surface of the front side glass, as estimated from the vehicle interior sound level, showed that the difference between the sound pressure on the external and internal surfaces is almost exactly equal to the glass transmission loss.

This result indicated that efforts to reduce acoustic sound generated by fluctuations in aerodynamic flow would be an effective means of reducing vehicle interior noise.

References

- (1) Sumitani, K., Maeda, K., Ichinose, K.: Jidousha to Ryutairikigaku: Shyataimawarinagare to Kuurikitokusei, Nagare, Vol. 23, p. 445-454 (2004) (in Japanese)
- (2) Iida, A.: Kuriki ni Kiinsuru Syanaishouon no Yosoku, Jidosyagijyutukai Sinpojiumu Tekisuto, Vol. 20124281, No. 18-11, p. 1-6 (2012) (in Japanese)
- (3) Powell, A.: Theory of vortex sound, Journal of the Acoustical society of America, 36, p. 177-195 (1964)
- (4) Kamatsuchi, T.: Linearized Euler Equation Niyoru Yokugatawari no Nagare to Oto no Renseikaiseki, Nagare, Vol. 23, p. 285-294 (2004) (in Japanese)
- (5) Kato, C., Takano, Y., Iida, A., Fujita, H., Ikegawa, M.: LES niyuru Ryutaion no Yosoku (Dai-ippou, Nijigen-enchu Karano Housyaon), TRANSACTIONS OF THE JAPAN SOCIETY OF MECHANICAL ENGINEERS Series B, Vol. 60, No.569, p. 126-132 (1994) (in Japanese)
- (6) Lighthill, M. J.: On sound generated aerodynamically. I. general theory, Proceedings of the Royal society of London, Series A, vol. 211, p. 564-587 (1952)
- (7) Caro, S., Ploumhans, P., Gallez, X., Morgenthaler, V., Mathey, F., Ciapa, G.: Identification of the Appropriate Parameters for Accurate CAA, 11th AIAA/CEAS Aeroacoustics Conference, vol. 4, p. 2407-2423 (2005)
- (8) Mochiduki, O., Maruta, Y.: Nagare no Oto ha Dokokarakurunoka, Ryutaionkougakunyumon-Yutakana Otokankyo wo Motomete-, Asakura Publishing Co., Ltd., p. 74-97 (1996) (in Japanese)
- (9) Anbo, K., Yoshino, T., Morishita, K., Teramura, M.: Transmission mechanism of flow-induced noise from the forward facing step through the glass plate, Actran Users Conference (2012)
- (10) Ambo, K., Yoshino, T., Teramura, M.: Zenpostepu kara Hasseisuru Kazakirion no garasutoukamekanizumu, Nihon kikaigakkai ronbunshu Bhen, Vol. 79, No. 803 (2013) (in Japanese)

- (11) Nouzawa, T., Li, Y., Kasaki N., Nakamura T.:
Jidosyaniokeru kurikisouon no ongenkouzounitsuite,
TRANSACTIONS OF THE JAPAN SOCIETY OF
MECHANICAL ENGINEERS Series B, Vol. 75, No.
758, p. 101-107 (2009) (in Japanese)

■ Author ■



Kei AMBO



Kazuya MORISHITA



Takashi YOSHINO



Minoru TERAMURA