Development of a duct-type ventilation system with high sound insulation

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ABSTRACT

In Japan, the Building Standards Law was revised in 2003 and installation of ventilation system working for 24 hours is obliged to prevent sick building syndrome. Under such a situation, it is required to develop ventilation equipment with high sound insulation performance especially for residential buildings in noisy areas. In this study, therefore, a duct-type ventilation device with high sound insulation equipped to glass sliding windows has been developed, in which two types of acoustical components, resonators with slit-shaped apertures for lower frequencies and an array of small fins for higher frequencies, are combined. In the process of designing the system, wave-based numerical investigation was performed using plane wave theory and the FDTD method. The results of the numerical studies were examined by performing full-scale model experiment using sound intensity measurement technique.

1 INTRODUCTION

When building elements such as ventilation devices are set in the exterior wall, sound insulation performance of the wall deteriorates. In Japan, the Building Standards Law was revised in 2003 and installation of ventilation device working for 24 hours was obliged to prevent sick building syndrome, therefore it is necessary to improve insulation performance of such a ventilation device. The authors propose a duct-type natural ventilation system with passive noise control components and design procedure of the noise control components attached to the duct-type ventilation device is described in this paper. In the duct-type ventilation device,
two kinds of components – series of resonators with slit-shaped apertures for noises in low frequencies and an array of small fins for those in high frequencies – were applied in order to reduce environmental noise which has components in a wide frequency range. Shapes of the acoustical components were investigated using plane wave theory and the FDTD method. In addition as a case study, A-weighted sound pressure level in a room with the designed duct-type ventilation device was calculated under assumption that the room was exposed by heavy road traffic noise with a model spectrum provided in ISO 717-1 to examine the validity of the duct-type ventilation device.

2 DESIGN OF THE ACOUSTICAL COMPONENTS

Figure 1 shows the duct-type ventilation device under investigation. To keep space to attach some acoustical components for noise control, length of the duct was made to be 1,800 mm long. As shown in Figure 2, resonators with slit-shaped apertures (see Figure 2 (A) and (B)) for lower frequencies and an array of small fins (see Figure 2 (C)) for higher frequency were applied as the acoustical components. Design frequencies of sound insulation were distributed as shown in Table 1. For noise control in low frequencies from 160 Hz to 800 Hz 7 resonators were used and for higher frequency range from 1k Hz to 2k Hz array of fins were used.

![Figure 1: Duct-type ventilation device in a room.](image1)

![Figure 2: Resonators with slit types A, B and shape of the array of fins.](image2)

<table>
<thead>
<tr>
<th>Frequency</th>
<th>160 Hz</th>
<th>226 Hz</th>
<th>320 Hz</th>
<th>400 Hz</th>
<th>500 Hz</th>
<th>630 Hz</th>
<th>800 Hz</th>
<th>1k - 2k Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acoustical components</td>
<td>Array of resonators</td>
<td>Array of fins</td>
<td></td>
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</table>

Table 1: Design frequencies of sound insulation by an array of resonators and fins
2.1 An Array of Resonators

Resonant frequency of a resonator, \( f_r \), is expressed as Eq. (1) [1].

\[
f_r = c \sqrt{\frac{c_o}{V}} \frac{1}{2\pi}, \quad c_o = \frac{S_o}{(L + \Delta L)}
\]  

(1)

where \( V \) is a volume of the resonator, \( c \) is speed of sound, \( S_o \) is an area of opening of the slit, \( L \) is a length of a neck of the slit and \( \Delta L \) is end correction of the aperture. In Eq. (1), end correction, \( \Delta L \), can not be analytically obtained for slit type resonator. Therefore, estimation method of the \( \Delta L \) was firstly investigated through numerical analysis using 2-dimensional FDTD method.

2-dimensional sound field which was composed by a duct with non-reflection terminations at both ends and a resonator as shown in Figure 3 was assumed for investigating the \( \Delta L \). An impulse response between sound source and receiving points shown in Figure 3 was firstly calculated by 2-dimensional FDTD method and the resonant frequency of the slit resonator, \( f_r \), was obtained through FFT of the calculated impulse response. The end correction, \( \Delta L \), was calculated by substituting the acquired \( f_r \) into Eq. (1). The calculation was made for 11 (for the slit type A) and 5 (for the slit type B) cases of different combinations of dimensions of the resonator, \( a \) and \( b \). The results were arranged in relationship between \( a/c \) (width of the slit normalized by width of the main duct space, \( c \)) and \( \Delta L/a \) (end correction normalized by the width of the slit, \( a \)) as shown in Figure 4. In the Figure, regression curves were also shown for respective slit types. In the design procedure described hereafter, the end correction of the slit resonator was calculated based on the equations of the regression curves.

In the design procedure of an array of the slit resonators, optimal combination of design parameters of dimensions of resonators (\( a \) and \( b \)) and slit type (A and B) were determined through trial and error using the 1-dimensional plane wave theory [2]. As results of the procedure, two types of array of resonators shown in Figure 5 were obtained; Type A for design frequency range from 160 to 800 Hz and Type B for that from 226 to 800 Hz. The frequency characteristics of their insertion loss calculated by the 1-dimensional plane wave theory are shown in Figure 6 in which high insertion loss is realized in the respective design frequency range.

![Sound field for calculating impulse response to estimate resonant frequency, \( f_r \).](image)

(C and \( d \) are constant (\( c = 50 \text{ mm}, \ d = 48 \text{ mm} \).)

![End correction of slit types A and B. \( \Delta L/a \) was obtained when \( b \) is 70, 100, 200 mm.](image)
2.2 An Array of Fins

An array of fins to reduce sound propagation in high frequency range (from 1k to 2k Hz) were set in a downstream area with 276 mm long from series of slit resonators which is 1474 mm long in total. Design of an array of fins was made by the 2-dimensional FDTD method.

As shown in Figure 7, two fins were assumed to be set in allowed area to make three sub spaces in which the sound was reflected in upstream direction. Therefore three widths of the sub areas, \(a\), \(b\) and \(c\) were designed parameters in this procedure. The combinations of the values of \(a\), \(b\) and \(c\) were changed as shown in Table 2 and the best combination were selected based on calculation results by the FDTD analysis. Frequency characteristics of the insertion loss of the array of the fins were obtained as follows. Firstly, the impulse response between source and receiving points for sound field with array of fins (see Figure 7 (A)) was calculated by the FDTD analysis. At the same time, the impulse response for sound field without fins (see Figure 7 (B)) was also calculated. The calculated impulse responses were converted in frequency transfer functions through FFT and the insertion loss was obtained as level difference of the magnitudes of the frequency components between the two responses. The calculation result of the insertion loss is shown in Figure 8. In case of Type D (bold line), high insertion loss is obtained in the widest frequency range among them. The Type D was adopted in this study.

![Designed arrays of resonators.](image)

![Insertion loss by the arrays of resonators.](image)

**Table 2: Combinations of the intervals between the fins.**

<table>
<thead>
<tr>
<th></th>
<th>(a)</th>
<th>(b)</th>
<th>(c)</th>
</tr>
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<tbody>
<tr>
<td>Type C</td>
<td>70</td>
<td>90</td>
<td>110</td>
</tr>
<tr>
<td>Type D</td>
<td>80</td>
<td>90</td>
<td>100</td>
</tr>
<tr>
<td>Type E</td>
<td>90</td>
<td>90</td>
<td>90</td>
</tr>
</tbody>
</table>

Unit: mm
3 TRANSMISSION LOSS OF THE VENTILATION DEVICE

From the design procedure described above, two types of the duct-type ventilation device with high sound insulation performance as shown in Figure 9 were obtained. In order to confirm their sound insulation performance, two laboratory experiments for measuring sound transmission loss were performed.

Measurement of transmission loss was performed in rooms of a combination of a reverberation room and anechoic room shown in Figure 10. Broad-band noise was radiated from a loudspeaker in the reverberation room, and sound power level $L_W$ radiated from the ventilation device into the anechoic room was measured using scanning intensity method. Averaged sound pressure level $L_p$ in the reverberation room was also measured and element normalized level difference $D_{n,e}$ was calculated by Eq. (2).

$$D_{n,e} = L_p - 6 - \left( L_{in} + 10 \log_{10} \left( \frac{S_M}{A_0} \right) \right)$$  (2)

where $L_{in}$ is averaged sound intensity level on the measurement surface in the anechoic room, $S_M$ is total area of the measurement surface in the anechoic room, and $A_0$ is the reference area. In this paper $A_0$ equals to 1. When surface pressure intensity indicator $F_{pi}$ was over 10 dB, measured data was excluded. Ventilation device with acoustical components used in this experiment is Type F and G (see Figure 9). A full-scale model of the ventilation device was
made with aluminum of 2 mm thickness. Result is shown in Figure 11. In case of Type F, deterioration of transmission loss is observed in wide frequency except 1.25k and 1.6k Hz band. On the contrary, in Type G, effects of the arrays of resonators and fins were simultaneously realized and $D_{n,e}$ was much improved in wide frequency range.

4 TRANSMISSION LOSS OF A GLASS SLIDING WINDOW WITH THE VENTILATION DEVICE

When a ventilation device is additionally set to a glass sliding window, sound insulation performance deteriorates compared to the situation with only the glass sliding window. In this section, composite transmission loss of a wall system composed of glass sliding window and duct-type ventilation device was measured in order to confirm the effect of the proposed the duct-type ventilation device with noise control components.

4.1 Measurement of Transmission Loss in Reverberation Room Method

Composite transmission loss of a window system which is composed by a glass sliding window and a duct-type ventilation device was measured by the reverberation room method. The
glass sliding window used in this experiment is a double sliding window (1980 mm *1980 mm) with two sheets of glass of 6 mm thickness. As a duct-type ventilation device, Type F and Type H were used and the noise control effect by the acoustical components set in Type H was examined. Between two reverberation rooms, glass sliding window and duct-type ventilation device were set together as shown in Figure 12. Averaged sound pressure level in each reverberation room and reverberation time of the receiving room were measured. From these results, transmission loss was calculated by Eq. (3).

\[
TL = (\bar{L}_1 - \bar{L}_2) + 10 \log_{10} \left( \frac{S}{A_2} \right), A_2 = \frac{55.3 \cdot V_2}{T_2}
\]

(3)

where \( \bar{L}_1 \) and \( \bar{L}_2 \) are averaged sound pressure levels in the sound source and the receiving rooms, \( S \) is the total area of the glass sliding window and the ventilation device, \( c \) is speed of sound, \( V_2 \) is the volume of the receiving room and \( T_2 \) is the reverberation time of the receiving room.

Measurement result is shown in Figure 13. In the Figure, sound transmission loss of glass sliding window (without ventilation device) is also shown as a reference. As is seen in the Figure, sound insulation performance reduces when a ventilation device is additionally attached to a window. In case of Type F (without noise control component) set with the glass sliding window, \( TL \) deteriorates in wide frequency range except 1.25k and 1.6k Hz band. (In 1.25k and 1.6k Hz bands, Type F originally had good performance as shown in Figure 11.) When noise control components were added in the ventilation device (Window with Type H), \( TL \) is much improved compared to the case without noise control component (Window with Type F).

5 CASE STUDY ON SOUND PRESSURE LEVEL IN A ROOM

A case study was performed assuming that the duct-type ventilation device and glass sliding window are set in the exterior wall and the exterior wall is exposed by a road traffic noise. The
spectrum of the exterior noise was assumed referring to sound level spectrum No. 2 shown in ISO 717-1 (A-weighted sound pressure level along arterial roads with heavy traffic volume) [3]. Over-all level of the exterior noise was assumed 75 dB to simulate noise by heavy traffic volume. A-weighted sound pressure level in a room, $SPL_i$, was calculated by Eq. (4).

$$SPL_i = SPL_o - TL + 10 \log_{10}(\frac{S}{A}) + 6$$

(4)

where $TL$ is the transmission loss of the exterior wall and measured value of transmission loss in the section 4 was applied. $S$ is the total area of the ventilation device and the window and $A$ is the equivalent sound absorption area. The equivalent sound absorption area was assumed as shown in Table 3. Frequency characteristics of the calculated sound pressure level are shown in Figure 14. $SPL_i$ of Type F increases about 10 dB in 630 Hz band and 5 dB in over all compared with the case of only a glass sliding window. On the other hand in the case with Type H set on a glass sliding window, the increase of the interior noise is much reduced and the increase of A-weighted sound pressure level (O.A.) was restrained in only 1 dB.

Table 3: Equivalent sound absorption area in a room with flooring.

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>125 Hz</th>
<th>250 Hz</th>
<th>500 Hz</th>
<th>1k Hz</th>
<th>2k Hz</th>
<th>4k Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.0</td>
<td>5.5</td>
<td>5.0</td>
<td>5.0</td>
<td>5.0</td>
<td>5.5</td>
<td></td>
</tr>
</tbody>
</table>

Unit: m²

Figure 14: Estimation of the $SPL$ in a room with ventilation device and glass sliding window.

6 CONCLUSIONS

To improve sound insulation performance of a duct-type ventilation device, the authors proposed a duct-type ventilation device to which a combination of series of slit resonators and an array of fins were applied. In this report, the design procedure was described and its
effectiveness was confirmed by laboratory experiments. As a result of the experiments, it was found that the proposed duct-type ventilation device is effective for noise reduction against typical road traffic noise.

7 REFERENCES