1. Introduction

Porous materials with continuous pores, such as glass wool, are useful as sound-absorbing materials. Studies on the sound absorption coefficient of wheat straw (McGinnes, et al., 2005) and tatami (Tsuchida and Kobayashi, 2005) have examined plant-based sound-absorbing structures. Another study of sound-absorbing materials for low frequencies used a Helmholtz sound absorber to demonstrate sound absorption utilizing recycled paper (Hirata, et al., 2003). Moreover, we apply plant structures with minute clearances as sound-absorbing materials (Sakamoto, et al., 2011).

This study aims to establish the sound absorption coefficient in sound-absorbing structures formed by bundles of clearances between two surfaces. In previous studies (Sakamoto, et al., 2014) (Sakamoto, et al., 2013), the authors performed and reported the results of experiments and theoretical analysis involving sound-absorbing structures having clearances between two parallel planes, folding screen clearances, and clearances that narrow in the depth direction from the incident face.

Theoretical analysis of boundary layer viscosity was conducted with reference to Tijdeman (Tijdeman, 1975) for cylinders, and to Beltman (Beltman, et al., 1998) and Stinson (Stinson, et al., 1992) for the clearances between parallel planes to calculate the propagation constant. In a previous study (Sakamoto, et al., 2014), the authors converted the
analytical method for a cylindrical coordinate system in Tijdeman (Tijdeman, 1975) to a Cartesian coordinate system and then found the characteristic acoustic impedance and propagation constant related to clearances between two planes.

In this paper, we deal with clearances between two surfaces formed by a thin material that has been folded like a sensu, or Japanese folding fan. For theoretical analysis, we divided a transfer matrix into the incident and cross direction of a sound wave. We then calculated the sound absorption coefficient by giving the characteristic acoustic impedance and propagation constant to each transfer matrix. In addition, we compared the calculation results to the measurement results (Sakamoto, et al., 2014) to obtain the sound absorption coefficient of a material that corresponded to the subject of the analysis.

By folding thin material, it is simple to produce a sound-absorbing structure in the shape of a sensu (folding fan). Materials such as old paper can be reused as sound-absorbing structures, without further treatment and requiring no energy, thus contributing to carbon fixation.

2. Measurement apparatus

We produced test samples using the clearance between two surfaces in a folding fan shape, then found and compared the sound absorption coefficient of each test sample through theoretical analysis and experimentation. Table 1 shows the types and compositions of the test samples and Table 2 shows the thin-board clearances, number of boards, and aperture ratios. For the sake of comparison, we also show information for Type C as reported in previous research (Sakamoto, et al., 2014) (Sakamoto, et al., 2013). The normal incident absorption coefficient was calculated on the basis of the standard test method ISO 10534-2 (Sakamoto, et al., 2013).

The fourth row of Table 1 shows a schematic diagram of the incident face, while the lowest row shows the test sample from the side. For the outer frame of the test sample we used a 0.5-mm-thick stainless steel box, and to construct the clearances we used high-quality paper (0.08 mm thick) as in our previous report (Sakamoto, et al., 2013). To construct the test sample, we sandwiched gauges created to match the clearance shape between the thin boards and adhered both sides of the thin boards to the frame material. We then bound both sides of the thin boards by removing the gauge.

In the test sample used in our previous report (Sakamoto, et al., 2014) (parallel clearance, folding screen-shaped clearance, with a clearance that narrows in the depth direction (Type C in Table 1)), because the thin boards in it are plane, we could use stainless plate for the material. In the folding fan-type test sample used in this paper (Type D, Type Db), however, the thin boards are bent in three dimensional surface, and we cannot fold stainless plate as shown in Figs. 1(a) and (b). Thus, we again used the high-quality paper used in our previous report (Sakamoto, et al., 2013) for the folding fan-type test sample. As shown in Fig. 1(c), this resulted in the irregular sizes of each clearance. In our previous report (Sakamoto, et al., 2014), we conducted a simulation to show the increase in the sound absorption coefficient due to clearance irregularity.

Fig. 1  Paper portion of folding fan, element of test sample and incident face of completed test sample
Table 1  Schematic drawing and dimension of paper test samples

<table>
<thead>
<tr>
<th>Type</th>
<th>Type D Folding fan</th>
<th>Type D&lt;sub&gt;b&lt;/sub&gt; With back air space</th>
<th>Type D Length l = 100 mm</th>
<th>Type C Wedge</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average clearance d [mm]</td>
<td>0.12, 0.22, 0.52, 1.04</td>
<td>0.22</td>
<td>0.22, 0.52</td>
<td>0.22, 0.52, 1.04</td>
</tr>
<tr>
<td>Back air space [mm]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Incidence face 25.7×25.7mm Surrounded by 0.5 mm casing</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Side view</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Length l = 50 mm (Except Type D l = 100 mm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aluminum block</td>
<td>Back air space</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2  Typical specifications of paper test samples

<table>
<thead>
<tr>
<th>Type</th>
<th>Average clearance on incident face [mm]</th>
<th>Number of thin sheets</th>
<th>Number of clearance</th>
<th>Aperture ratio of incident face</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type D</td>
<td>0.12</td>
<td>123</td>
<td>122</td>
<td>0.556</td>
</tr>
<tr>
<td></td>
<td>0.22</td>
<td>83</td>
<td>82</td>
<td>0.675</td>
</tr>
<tr>
<td></td>
<td>0.52</td>
<td>42</td>
<td>41</td>
<td>0.798</td>
</tr>
<tr>
<td></td>
<td>1.04</td>
<td>23</td>
<td>22</td>
<td>0.855</td>
</tr>
<tr>
<td>Type D&lt;sub&gt;b&lt;/sub&gt;</td>
<td>0.22</td>
<td>83</td>
<td>82</td>
<td>0.556</td>
</tr>
<tr>
<td>Type C</td>
<td>0.22</td>
<td>83</td>
<td>82</td>
<td>0.556</td>
</tr>
<tr>
<td></td>
<td>0.52</td>
<td>42</td>
<td>41</td>
<td>0.675</td>
</tr>
<tr>
<td></td>
<td>1.04</td>
<td>23</td>
<td>22</td>
<td>0.855</td>
</tr>
</tbody>
</table>
3. Theoretical analysis
3.1 Calculating the sound absorption coefficient from the transfer matrix

In this paper, to consider sound wave attenuation in the clearances between two surfaces, we converted the Tijdeman method (Tijdeman, 1975) from a cylindrical coordinate system to a Cartesian coordinate system (Sakamoto, et al., 2014). For the clearances between two surfaces, a model of a sound wave entering a clearance of thickness $H$ in the positive $x$ direction from a space of thickness $H_0$ in the $z$ direction is shown in Fig. 2. Here, $H_0$ is the unit thickness for each individual clearance including the non-opening portion. In other words, it is clearance thickness $H$ divided by the aperture ratio.

![Fig. 2 Model of a narrow clearance between two surfaces](image)

When the thickness $H$ of the clearance between two surfaces in Fig. 2 is sufficiently small with respect to the length of the $y$ direction of the clearance, sound wave propagation is determined by the $x$ and $z$ directional components of the clearance, and in the $y$ direction propagation is seen to be uniform. Thus, the cross sectional area of the clearance on the $y$–$z$ plane can be written as the product of clearance thickness $H$ and the unit length of the $y$ direction.

Using $H$, the cross sectional area of the $y$–$z$ plane in the clearance, length $l$, characteristic impedance $Z_c$, and propagation constant $\gamma$, the transfer matrix $T$ and the four-terminal constants $A–D$ of the unit sound element can be expressed by Eq. (1) (Suyama and Hirata, 1979).

$$T = \begin{bmatrix} A & B \\ C & D \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l) & \frac{Z_c}{H} \sinh(\gamma l) \\ \frac{Z_c}{H} \sinh(\gamma l) & \cosh(\gamma l) \end{bmatrix}$$ (1)

For the entrance and terminal of the clearance, given the respective sound pressures $p_1$, $p_2$ and the $x$ direction component of the particle velocities $u_1$, $u_2$, then the transfer matrix can be expressed as follows in Eq. (2), considering that the end of the clearance is rigid and that $u_2 = 0$ holds for the particle velocity.

$$\begin{bmatrix} p_1 \\ Su_1 \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} p_2 \\ Su_2 \end{bmatrix} = \begin{bmatrix} Ap_2 \\ Cp_2 \end{bmatrix}$$ (2)

Here, giving the upstream sound pressure and $x$ direction component of the particle velocity in Fig. 2 as $p_0$, $u_0$, and using of $p_0 = p_1$, $S_0 u_0 = S u_1$, we can express specific acoustic impedance $Z_0$ with the following Eq. (3). Here, we see that the specific acoustic impedance $Z_0$ becomes larger due to the reciprocal of the aperture ratio of the clearance $H_0/H$.
In general, the relationship between the specific acoustic impedance on the incident face \( Z_0 \) and reflectance \( R \) can be expressed in Eq. (4) below.

\[
\frac{Z_{\text{inc}}}{Z_{\text{air}}} = \frac{1 + R}{1 - R}
\]  

(4)

The theoretical value for the sound absorption coefficient \( \alpha \) can be obtained from Eq. (5).

\[
\alpha = 1 - |R|^2
\]  

(5)

3.2 Propagation constant and characteristic impedance when considering sound wave attenuation in clearances between two surfaces

Given sound pressure \( p \) as shown in Eq. (6), the propagation constant \( \gamma \) in a clearance between two surfaces is given as in Eq. (7) (Tijdeman, 1975). Here, \( X \) and \( Y \) are arbitrary constants, and \( k \) is the wave number.

\[
p = X e^{\gamma x} + Y e^{-\gamma x}
\]  

(6)

\[
\gamma = k \sqrt{1 - \frac{\cosh \left( \frac{5}{j^2} s \right) - 1}{\left( \frac{5}{j^2} \right) \sinh \left( \frac{5}{j^2} s \right) - 1}} \frac{\kappa}{N}
\]

(7)

Next, characteristic impedance \( Z_c \) for a clearance between two surfaces with consideration for attenuation is shown.

If the \( x \) component of traveling wave particle velocity and sound pressure are given as \( u^* \), \( p^* \), then characteristic impedance \( Z_c \) is given in Eq. (8).

\[
Z_c = \frac{p^*}{u^*}
\]

(8)

\( u^* \), \( p^* \) in a clearance between two surfaces with attenuation are shown in Eqs. (9) and (10) (Tijdeman, 1975). Here, \( P_a \) is atmospheric pressure and \( t \) is time.

\[
u^* = c \frac{\gamma}{\kappa k} \left[ 1 - \frac{\sinh(\frac{5}{j^2} \eta s) - \sinh(\frac{5}{j^2}(\eta - 1) s)}{\sinh(\frac{5}{j^2} s)} \right] (-Ye^{-\gamma x}) e^{j \omega t}
\]

(9)
The obtained characteristic impedance $Z_c$ is a function of the $z$ direction of the clearance. To make $Z_c$ a function of the $x$ direction only, we integrate $Z_c$ by $z$ and average within the $y$–$z$ cross-section to find $\overline{Z_c}$ which is equal in the $y$–$z$ cross section through Eq. (11) (Sakamoto, et al., 2014).

\begin{equation}
\overline{Z_c} = \frac{1}{H} \int_0^H Z_c dz
\end{equation}

Figures 3(a)–(c) show respective schematics for one clearance corresponding to Type D with $d = 0.52$, $0.22$, and $0.12$ mm. Positions where the clearance reaches $0.52$, $0.22$, and $0.12$ mm are all shown. These clearances are the typical thicknesses used in this study (Sakamoto, et al., 2014) (Sakamoto, et al., 2013).
3.3 \( y \) Direction element decomposition

To calculate the sound absorption coefficient of Type D, we must find the specific acoustic impedance by viewing the test sample from the incident face. To do so, we express the clearance using a transfer matrix related to sound pressure and volume velocity based on a one-dimensional (x dimension) wave equation (Suyama and Hirata, 1979).

The clearance thickness of Type D continually changes in the lateral (y) direction with respect to the sound wave incidence direction while continually decreasing in the direction of sound wave incidence (x). Thus, we used an approximation method using element decomposition in the y and x directions, respectively, to calculate the transfer matrix.

In this section, we find a transfer matrix that performs element decomposition for the clearance in the y direction at a decomposition location in the x direction in one clearance. In the next section, we continue by combining transfer matrices of a number decomposed in the x direction.

Regarding the \( i \)-th position in the x direction, to divide the clearance in the y direction into \( n \) elements, as shown in Fig. 4(b), we approach a wedge-shaped cross section clearance on the y–z surface as shown in Fig. 4(a). In other words, we analyze a clearance between two surfaces in which clearance thickness \( H \) changes in \( n \)-steps.

By connecting the transfer matrices (as shown in Fig. 4(b)) for \( n \) clearances between two surfaces in parallel, the result can be expressed in the equivalent circuit shown in Fig. 5. At this point, we view each element as an independent clearance, and we ignore sound pressure and particle velocity propagation between elements (Verdière, et al., 2013).
Giving the transfer matrices for “clearances connected in parallel” and “the next clearance to add” as $T_{i,n}$ and $T_{i,1to n-1}$, respectively, in Eq. (12), we can express the transfer matrix $T_{i,1to n}$ which connects these in parallel as shown in Eq. (13).

$$T_{i,n} = \begin{bmatrix} A_{i,n} & B_{i,n} \\ C_{i,n} & D_{i,n} \end{bmatrix} \quad T_{i,1to n-1} = \begin{bmatrix} A_{i,1to n-1} & B_{i,1to n-1} \\ C_{i,1to n-1} & D_{i,1to n-1} \end{bmatrix}$$

$$T_{i,1to n} = \begin{bmatrix} A_{i,1to n} & B_{i,1to n} \\ C_{i,1to n} & D_{i,1to n} \end{bmatrix} = \begin{bmatrix} A_{i,n}B_{i,1to n-1} + A_{i,1to n-1}B_{i,n} \\ B_{i,n} + B_{i,1to n-1} \\ C_{i,n} + C_{i,1to n-1} + \frac{(A_{i,1to n-1} - A_{i,n})(D_{i,n} - D_{i,1to n-1})}{B_{i,n} + B_{i,1to n-1}} \\ D_{i,n}B_{i,1to n-1} + D_{i,1to n-1}B_{i,n} + B_{i,n} + B_{i,1to n-1} \end{bmatrix}$$

3.4 x Direction element decomposition and combining x, y direction elements

In the previous section, we found transfer matrices for elements dividing one clearance in the $x$ direction. In this section, we combine those elements to find a transfer matrix $T$ for one clearance.

A wedge-shaped cross section clearance of the $x$–$z$ plane as shown in Fig. 6(a) is approached as shown in Fig. 6(b). In other words, we approach so that clearance thickness $H$ changes in the $x$ direction in $i$-steps.

![Actual and Approximated Clearance Shapes](image)

Fig. 6 Approximation of clearance shape of depth ($x$) direction division

The transfer matrix for one clearance can be expressed as an equivalent circuit as shown in Fig. 7 by connecting transfer matrices divided into $i$ in the $x$ direction in cascade.

![Equivalent Circuit](image)

Fig. 7 Equivalent circuit of depth ($x$) direction division

The number of decomposed elements is $n$ divisions in the lateral ($y$) direction with respect to the direction of sound wave incidence, and $i$ divisions in the direction of sound wave incidence ($x$). Thus, for the equivalent circuit shown in Fig. 7, $i$ transfer matrices (from $T_{1,1to n}$ to $T_{i,1to n}$) (connecting each of $n$ transfer matrices from $T_{i,1}$ to $T_{i,n}$ in parallel) can be connected in cascade to be expressed in detail with an equivalent circuit as shown in Fig. 8.

From the above, the transfer matrix $T$ for each clearance in Type D can be expressed in Eq. (14).
By substituting transfer matrix $T$ into Eq. (1) and using Eqs. (3) and (5), we can find the specific acoustic impedance and sound absorption coefficient of Type D.

\[
T = \begin{bmatrix}
A & B \\
C & D
\end{bmatrix} = \begin{bmatrix}
A_{1,1} & A_{1,2} & \cdots & A_{1,n} \\
A_{2,1} & A_{2,2} & \cdots & A_{2,n} \\
\vdots & \vdots & \ddots & \vdots \\
A_{n,1} & A_{n,2} & \cdots & A_{n,n}
\end{bmatrix}
\begin{bmatrix}
B_{1,1} & B_{1,2} & \cdots & B_{1,n} \\
B_{2,1} & B_{2,2} & \cdots & B_{2,n} \\
\vdots & \vdots & \ddots & \vdots \\
B_{n,1} & B_{n,2} & \cdots & B_{n,n}
\end{bmatrix}
\begin{bmatrix}
C_{1,1} & C_{1,2} & \cdots & C_{1,n} \\
C_{2,1} & C_{2,2} & \cdots & C_{2,n} \\
\vdots & \vdots & \ddots & \vdots \\
C_{n,1} & C_{n,2} & \cdots & C_{n,n}
\end{bmatrix}
\begin{bmatrix}
D_{1,1} & D_{1,2} & \cdots & D_{1,n} \\
D_{2,1} & D_{2,2} & \cdots & D_{2,n} \\
\vdots & \vdots & \ddots & \vdots \\
D_{n,1} & D_{n,2} & \cdots & D_{n,n}
\end{bmatrix}
\]

(14)

Fig. 8   Equivalent circuit for each clearance of Type D
3.5 Transfer matrix of Type D_b

For Type D_b, we divide the portion that is identical to Type D and the back air space to consider separately. After the sound wave propagates through each clearance, it enters a common back air space to all clearances. Thus, the transfer matrix for Type D_b can be found through the following procedure. Figure 9 shows the transfer matrix $T_b$ of Type D_b expressed as an equivalent circuit.

First, as with Type D, we find the transfer matrix $T$ of one clearance using Eq. (14).

Next, we connect transfer matrices $T$ for the number of clearances shown in Table 1 in parallel as $T_{all}$. We can also use Eq. (13) above for connecting the transfer matrices in parallel.

Then, we consider the transfer matrix of the back air space. The back air space is a large, single wedge-shaped clearance. Thus, the transfer matrix can be found using the analytic method provided in Section 3.4. In other words, the transfer matrix is decomposed in the sound wave incidence direction. Thus, the transfer matrix of the back air space $T_{BA}$ is obtained through Eq. (14).

Finally, we join the transfer matrices linked in parallel $T_{all}$ and the transfer matrix of the back air space $T_{BA}$ in a cascade connection. This lets us express the transfer matrix $T_b$ for Type D_b as follows in Eq. (15).

\[
T_b = \begin{bmatrix} A_b & B_b \\ C_b & D_b \end{bmatrix} = \begin{bmatrix} A_{all} & B_{all} \\ C_{all} & D_{all} \end{bmatrix} \begin{bmatrix} A_{BA} & B_{BA} \\ C_{BA} & D_{BA} \end{bmatrix}
\] (15)

By substituting transfer matrix $T_b$ into Eq. (1) and using Eqs. (3) and (5), we can find the specific acoustic impedance and sound absorption coefficient of Type D_b.

---

**Fig. 9** Equivalent circuit of Type D_b with back air space
4. Comparing the calculated and measured results

4.1 Type D

Figures 10–13 show a comparison of the calculated and measured results for the sound absorption coefficient for a test sample of length \( l = 50 \) mm for each clearance. Similarly, Figs. 14–16 show results for each clearance for a test sample of length \( l = 100 \) mm. Also, Figs. 10–13 show results for Type C (Sakamoto, et al., 2014) (Sakamoto, et al., 2013) for comparison. In the calculations for this section, the divided width in the \( x \) and \( y \) directions were set to 0.5 mm, a value for which the calculated values sufficiently converge.

In Figs. 10 and 11, in which \( l = 50 \) mm and \( d = 0.52 \) mm or higher, and in Fig. 14, in which \( l = 100 \) mm and \( d = 1.04 \) mm, experimental values are higher than the calculated values across a wide frequency range. Paper was used for the test sample, and because of the large irregularities in the flexure and clearance, there is a significant difference between the calculated and measured values. We will discuss this below.

The difference between the experimental and calculated values can be attributed to the use of approximation in the theoretical analysis and the supposition that all clearances are equal size and that there is no flexure in the wall surface. The reason is because an experimental value of Type C made by stainless steel gave good agreement with a calculated value shown in a previous report (Sakamoto, et al., 2014).

In a previous report (Sakamoto, et al., 2014), we found calculated values for the sound absorption coefficient that incorporated irregular clearances into the theoretical analysis. Our results showed that the sound absorption coefficient increased as the irregularity of the clearance increased. In addition, in the previous report we showed this same finding through experimentation. Specifically, we compared a stainless steel test sample with a paper test sample. The results of this experiment showed that because the stainless steel test sample had less clearance irregularity in comparison with the paper test sample, it also had a lower sound absorption coefficient. Thus, it gave values closer to the calculated values.

In Figs. 12 and 13, in which \( l = 50 \) mm and \( d = 0.22 \) mm or lower, and in Figs. 15 and 16, in which \( l = 100 \) mm and \( d = 0.52 \) mm or lower, the difference between the experimental values and calculated values is primarily due to a shift in the sound absorption coefficient curve. In other words, the calculated values have a sound absorption coefficient curve at a higher frequency side than the experimental values. This is thought to be because sound attenuation within the clearance was estimated to be smaller than the actual sound attenuation in the calculated values. In other words, because the actual sound velocity was lower than the calculated values, the effective thickness of sound absorption material was thicker than the apparent thickness and the sound absorption coefficient curve of the experimental values was at a lower frequency. This can be explained by the fact that the clearance thickness given during calculation matches more closely when it is slightly smaller than the clearance of the experimental values (Sakamoto, et al., 2014).

![Fig. 10](Type D Experiment, Type D Calculation, Type C Experiment, Type C Calculation)

![Fig. 11](Type D Experiment, Type D Calculation, Type C Experiment, Type C Calculation)

Fig. 10  Comparison between calculations and experiments (Type D, \( d = 1.04 \) mm, Type C, \( d=1.04 \) mm, \( l = 50 \) mm)

Fig. 11  Comparison between calculations and experiments (Type D, \( d = 0.52 \) mm, Type C, \( d=0.52 \) mm, \( l = 50 \) mm)
Here, in the most of the frequency range of Figs. 10–12, calculated values are slightly lower for Type D than Type C. Values for Type D are also slightly lower than Type C in the experimental values. Thus, the trends of the experimental and calculated values match closely. This shows that the differences between Type C and Type D are reflected in the theoretical analysis.

Fig. 12  Comparison between calculations and experiments (Type D, $d = 0.22$ mm, Type C, $d=0.22$ mm, $l = 50$ mm)

Fig. 13  Comparison between calculation and experiment (Type D, $d = 0.12$ mm, $l = 50$ mm)

Fig. 14  Comparison between calculation and experiment (Type D, $d = 1.04$ mm, $l = 100$ mm)

Fig. 15  Comparison between calculation and experiment (Type D, $d = 0.52$ mm, $l = 100$ mm)

Fig. 16  Comparison between calculation and experiment (Type D, $d = 0.22$ mm, $l = 100$ mm)
4.2 Type D<sub>b</sub>

Figure 17 shows the calculated values for the sound absorption coefficient when modifying the back air space thickness in Type D<sub>b</sub>, while Fig. 18 shows the measured values. The calculated and experimental values without a back air space are the results shown in Fig. 12 of the previous section.

Similar trends are seen in the calculated and experimental values, as discussed below.

For back air spaces 10 mm and 20 mm thick, the sound absorption peak frequency is reduced in comparison to cases without a back air space, and sound absorption characteristics for low frequencies are improved. The back air space thicknesses do not show clear decreases in the sound absorption coefficient when compared to cases with no back air space. In particular, when the back air space is 10 mm thick, performance improves over cases without a back air space.

The reason for this is that, near the terminal of the test sample, the regions where the clearance is small relative to the boundary layer thickness and where the aperture ratio is small are eliminated. Thus, sound waves propagate to the terminal of the test sample and increase the effective thickness of the sample. The thickness of the velocity boundary layer is dependent on the frequency, and is approximately 50 μm (Wesley, 1958) at 1 kHz.

On the other hand, when the back air space thickness exceeds 30 mm, the sound absorption peak frequency becomes higher, and a decrease in the overall sound absorption coefficient was seen. This is thought to be due simply to the reduction in sound-absorbing material.

The trends described above match in both the experimental and calculated values. This suggests that the theoretical analysis was also valid for the Type D<sub>b</sub> sample, which had a back air space.
5. Conclusion

We conducted a study related to sound absorbing materials created by folding thin material in the shape of a Japanese folding fan.

In order to conduct a calculation of the shape of a clearance in the folding fan-shaped test sample, we performed an approximation of element decomposition in both the incidence direction of the sound wave and the lateral direction. In a theoretical analysis, we considered sound wave attenuation due to boundary layer friction in the clearance between two surfaces.

In addition, we calculated the sound absorption coefficient by applying the characteristic acoustic impedance and propagation constant to individual divided transfer matrices. We compared experimental and calculated values for the sound absorption coefficient. Trends in both matched closely. Errors in the calculated values for large clearances could be attributed mainly to irregularity in the size of the clearances. Frequency shift of the calculated values for small clearances were due to errors in estimating the sound attenuation.

Sound absorbing structures shaped like a folding fan possess sound absorption characteristics are similar to the wedge-shaped sound absorbing structures shown in previous reports. This was also shown in calculated values based on theoretical analysis.

In cases with an appropriate back air space, sound absorption peak frequencies decreased relative to cases without a back air space, and sound absorption characteristics were superior across a wide frequency range. We showed that calculated values of this effect matched the trends of experimental values, and we demonstrated the validity of the theoretical analysis.

Nomenclature

\( A - D \) : Four terminal constants
\( c \) : Speed of sound in air \([\text{m/s}]\)
\( d \) : Average clearance \([\text{mm}]\)
\( H \) : Thickness of clearance \([\text{m}]\)
\( H_0 \) : Thickness of clearance divided by aperture ratio \([\text{m}]\)
\( i \) : Number of division of clearance in \( x \) direction
\( j \) : Imaginary unit
\( k \) : Wave number \([1/\text{m}]\)
\( l \) : Length of \( x \) direction \([\text{m}]\)
\( n \) : Number of division of clearance in \( y \) direction
\( p \) : Sound pressure \([\text{Pa}]\)
\( P_s \) : Atmospheric pressure \([\text{Pa}]\)
\( R \) : Reflection coefficient
\( s \) : Ratio of thickness of clearance to boundary layer thickness \([\text{m}]\)
\( t \) : Time \([\text{s}]\)
\( T \) : Transfer matrix
\( u \) : \( x \) direction component of particle velocity\([\text{m/s}]\)
\( x \) : \( x \) co-ordinate \([\text{m}]\)
\( X \) : Arbitrary constant
\( y \) : \( y \) co-ordinate \([\text{m}]\)
\( Y \) : Arbitrary constant
\( z \) : \( z \) co-ordinate \([\text{m}]\)
\( Z_{\text{air}} \) : Characteristic impedance of air
\( Z_c \) : Characteristic impedance
\( Z_{\text{av}} \) : Averaged characteristic impedance \([\text{Ns/m}^3]\)
\( Z_0 \) : Specific acoustic impedance \([\text{Ns/m}^3]\)
\( \alpha \) : Sound absorption coefficient
\[ \gamma \text{: Propagation constant [1/m]} \]
\[ \eta = \frac{\gamma}{H} \text{: Normalized thickness of clearance} \]
\[ \kappa \text{: Ratio of specific heat} \]
\[ \sigma \text{: Square root of the Prandtl number} \]
\[ \omega \text{: Angular frequency [rad/s]} \]

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References


