High-performance noise proof cover using acoustic tube

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Abstract
To prevent noise generated by devices such as compressors, generators, or motors, a noise-proof cover is usually installed around them. It is also installed to protect ultra precise devices from ambient noise. Because of space considerations, the noise radiating device or ultra precise devices to be protected from outer noise are generally situated at the center of the cover. However, this often lowers the performance of the noise-proof cover at some frequencies because of the occurrence of its inner cover acoustic modes. To solve these problems from the standpoint of the cover’s dimensions and the occurrence frequency of the inner acoustic mode in the cover, which protect devices within the cover from outer ambient noise, we used a simplified one-dimensional Transfer Matrix Method (TMM) to investigate the most effective arrangement of acoustic tubes on the inside of the cover to restrain the acoustic mode. The results show that the most effective arrangement depends on the width of the target frequency range. If the target frequency range is plus-minus 10 to 17 % around the peak caused by the acoustic mode, for example, the most effective arrangement is one in which tubes 1/4 as long as the longest length of the cover edge are set at both ends and at the center along the longest direction of the cover. Finally, the effect of the proposed structure was investigated by numerical acoustic calculations using the Boundary Element Method (BEM) and validated by experimental measurements. The BEM results correspond well to those of the experiment; the experimental results show that the sound pressure level is reduced about 6 dB over all of the frequency range around the original peak frequency plus or minus 50%.

Key words: Sound absorbing, Noise proof cover, Acoustic tube, Transfer matrix method, Boundary element method

1. Introduction

A noise-proof cover is usually installed to suppress noise radiated from devices such as compressors, generators, or motors, and it is installed to protect ultra precise devices from ambient noise, also. This cover is expected to have sufficient noise insulation performance and no openings. However, the cover usually needs several holes or gaps for air ventilation, heat exhaust, and adjustments to floor level, and so on. (In general, the opening is set as near to the floor level as possible for convenience of duct layout.) Moreover, the shielding around most of the cover creates inner acoustic modes that degrade the noise reduction performance of the cover itself.

To prevent the occurrence of these acoustic modes, sound absorbing structures are usually installed on the inner surface of the cover. While poro-elastic materials (PEM) such as fabric or foam are easy to install and absorb sound well at high frequencies, they absorb sound poorly at the low frequencies which inner acoustic modes occur in. Therefore, they must be very thick to absorb sound sufficiently at low frequencies. On the other hand, resonator-type sound absorbing structures perform better in the limited lower narrow frequency range with controllable parameters and small thickness.

Resonator-type sound absorbing structures are widely used to solve various engineering noise reduction problems. Okubo and Fujiwara have investigated applications for the top structure of highway noise barriers (so-called 'edge modified noise barriers') (Fujiwara, and Okubo,1998, 1999). Sakamoto et al. have suggested applying an array of narrow tubes to composite thin louver boards (Sakamoto, et al., 2013). Shirahata and Iwase have proposed a
Helmholtz resonator with the neck built into the back air intake to reduce the thickness of the absorbing structure (Shirahata and Iwase, 2011). An acoustic tube is also a kind of Helmholtz resonator that can reduce the noise at target frequencies. Onishi et al. have researched a sound-absorbing mechanism on the open end of duct cavity resonance (Onishi, et al., 2013). These resonator-type sound absorbing structures have higher performance with smaller thickness for lower target frequencies than PEMs. Therefore, they are suitable for setting inside cover to prevent the inner cover acoustic modes occurring.

Additionally, looking from a different angle, occurrence of inner acoustic modes in the noise-proof cover mentioned before limits the phenomenon to a specific predictable scope for frequency and mode shape. Thus, considering that the noise source to be prevented, or the ultra precise device to be protected may be generally set at the center of the cover, the first inhibited inner acoustic mode is assumed to be 2nd order in the longest cover direction. Therefore, a sound absorbing structure that perform higher in a target specific frequency range can be effectively applied.

For this study, the authors focus on the relationship between the occurrences of acoustic modes inside the cover, their frequency, and the structure to best prevent them. To prevent inner acoustic modes at lower frequency ranges in a noise-proof cover with small openings, acoustic tubes were installed and their most effective arrangement on the inner space of the cover was investigated. The results of this investigation are summarized in this paper.

2. Comprehension of relationship between noise reduction performance and inner acoustic modes

To comprehend the acoustic characteristics of a cover with small openings, an acrylic box was made and its noise reduction performance was measured. Assuming noise proof cover for ultra precise devices to be protected from ambient noise, the sound pressure level at the center of cover and sound pressure distribution in the cover were measured when the source was set at the out of cover.

2.1 Problem establishment and reciprocity theorem

Fig. 1(a) shows the problem discussed in this paper; the sound pressure in the cover is measured when the source was set at the out of cover (Outer source problem). Using the reciprocity theorem, the force point and measurement point are mutually interchangeable (Fig. 1(b): Inner source problem). FRF $p_A/Q_B$ which means the sound pressure measured at point A when point B is excited at unit volume acceleration, equals $p_B/Q_A$ which means the sound pressure measured at point B when point A is excited at unit volume acceleration. Limits of application for this reciprocity theorem were investigated by Pierce (Pierce, 1981) and re-summarized in Japanese by Suzuki. According to Suzuki, this theorem holds in the condition in which all boundaries are locally reactive (Suzuki, 2002), and this condition is complied with in this study. Thus, under the assumption that the reciprocity theorem is complied with, each problems shown in Fig. 1(a) : outer source problem and Fig. 1(b) : inner source problem could be deemed to be mutually replaced.

In this study, only outer source problem shown in Fig. 1(a) is discussed.
2.2 Measurement for comprehension of inner acoustic modes

Fig. 2 shows a photo of the experimental measurement conditions for comprehension of inner acoustic modes. As shown in this figure, a one face open box 500 mm deep, 700 mm wide, and 800 mm high was constructed using 10-mm-thick acrylic board. Additionally, to discover the general affect of a small hole or gap set on the cover, a 30-mm thick acrylic board lid was placed 10 mm above the open face of the box. The nozzle end of a volume acceleration point source was set 1 m from the center of the largest face of the box along the outer normal direction to radiate random white noise. Nine small microphones were arrayed to a rod set along the box’s longest direction to measure the inner cover sound pressure. The measurement systems calculate frequency response functions (FRFs) of each point sound pressure to the source volume acceleration. Additionally, the sound pressure level distribution of the yellow area, at the center of the width, could be also comprehended by measuring it several times by moving the microphone array rod in the directions of the double-headed blue arrow in this figure.
2.3 Results of measurement

Fig. 3 shows FRFs of sound pressure at five points to the source volume acceleration to identify the frequency of the inner cover acoustic modes.

Fig. 3 (b) shows that the FRFs of the five points shown in Fig. 3(a) have characteristic peaks at 245, 362, 445, 573, and 653 Hz. These are the effects of the inner cover acoustic modes. Fig. 4 shows sound pressure level distributions at peak frequencies on FRFs in Fig. 3. From these figures, the occurrence of inner cover acoustic modes can be comprehended, for example the 1st mode on vertical direction at 245 Hz, the 1st on horizontal direction at 362 Hz, and the 2nd on vertical direction at 445 Hz.

"Vertical direction" means the longest cover direction, so occurrence of the 2nd acoustic mode in the longest length direction means that sound waves as long as the same as the longest cover length come and go in the cover along this direction.

On the assumption that the ultra precise devices to be protected from noise is located near the center of the cover, this result leads to the primary acoustic mode that should be prevented being the 2nd along the direction of the longest cover length, whose frequency \( f \) is calculated as \( f = c / l \) [Hz] using the longest cover length \( l \) and sound speed \( c \). Because the longest cover length is 800 mm in this study, the frequency of the 2nd acoustic mode in the direction along the longest cover length is calculated to be 425 Hz theoretically, which corresponds to the experimental result (445 Hz) shown in Fig. 4 within 5 % accuracy.
3. Planning for effective arrangement of acoustic tubes

It is well-known that the acoustic tubes, one of the resonator-type sound absorbing structure, cancels the surrounding acoustic pressure field at the specific frequencies determined by the tube length. However, the effective arrangement of tubes have not discussed well from a stand point of the occurrence of the inner cover acoustic modes. This chapter summarizes the results of the planning for effective arrangement of acoustic tubes inside cover, which could be considered a counter plan for the occurrence of the 2nd acoustic mode in the direction along the longest cover length mentioned in the previous chapter.

3.1 Acoustic tube

A side branch resonator is a widely used noise reduction structure for low-frequencies in a duct. It is shaped like a one-side-open tube which is branched from the target duct to maintain silence. It is 1/4 the length of the sound wave determined by the target frequency. Outward sound waves induced by coming and going in the tube cancel out the surrounding sound wave at the target frequency.

The effect of the side branch resonator comes not only when it is connected with the target duct but also when the extracted one-side-open tube is by itself in the acoustic field. It is categorized as a resonator-type noise absorbing structure. This sound absorbing structure, extracted from duct and shaped as a one-side-open tube, is called an 'acoustic tube' in this paper.

The surrounding noise reduction mechanism of the acoustic tube is explained again in Fig. 5. The acoustic tube is set in the surrounding noise field as shown in this figure. The surrounding vibrating pressure field excites the air in the tube. The vibration of the air in the tube induces standing waves at the frequency defined by its wave length being four times the tube length. During these processes, a part of the energy of the ambient noise is spent to forming and growing of the standing wave in the tube. As a result of the energy consumption, the energy of the ambient noise is reduced.

This acoustic tube could be considered as a counter plan for degraded noise reduction performance (at 445Hz shown in Fig. 3(b)) because of the occurrence of the 2nd acoustic mode in the direction along the longest cover length. Its effective arrangement on the inside of the cover is investigated in next section.
3.2 Effective arrangement of acoustic tubes on inner side of cover

The noise reduction performance at 445 Hz was degraded by the 2nd acoustic mode in the direction along the longest cover length as mentioned above. The occurrence of the acoustic mode means that the standing wave forming the anti-node of the sound pressure at both ends and the center on the mode direction because of the coming and going waves that are as long as the longest cover length. Because the wave length forming this acoustic mode is as long as the longest cover length, the optimum length of an acoustic tube to prevent the mode is equal to 1/4 of the longest cover length, and then the acoustic tube can be arranged in just four units in the cover along the direction of the longest cover length, as shown in Fig. 6. Considering the sound reducing mechanism of acoustic tubes due to the radiation of the inverse phase wave from the open end, the most effective arrangement is assumed to be having open ends of tubes set at both ends and at the center of the cover, which is the anti-node of the sound pressure. The arrangement pattern shown in Fig. 6 resulting from this consideration is verified by optimization using the one-dimensional calculation in the next section and validated by the three-dimensional numerical calculation and experimental measurement in the next chapter.

3.3 Optimization of tube arrangement patterns using simple one-dimensional model

This section mentions the most effective tube arrangement patterns optimized by the simple one-dimensional calculation model using the transfer matrix method (TMM) for the simple ideal cover structure, and discusses whether the arrangement plan shown in Fig. 6 is the best or not.

3.3.1 Simple one-dimensional calculation model using TMM

A simple one-dimensional calculation model using TMM is introduced to optimize the arrangement patterns of acoustic tubes. Fig. 7 shows the approach to modifying the simple one-dimensional model. The internal air is separated and modeled to several tube shaped elements shown in this figure. As shown in this figure, neglecting all bends of tubes, the internal air including inner tube has modeled and divided to several simple tube shaped one-dimensional acoustic elements. On the right figure in Fig. 7, an each point-end-line indicates an each element. Fig.8 shows parameters and physical quantities of the each element as shown as a point-end-line in Fig. 7. Each tube
shaped one-dimensional acoustic element $i$ has two parameters (cross section $s_i$ and length $l_i$) and four physical quantities (pressure $p_1, p_2$ and inside direction air particle velocity $u_1, u_2$ at each end) as shown in Fig. 8.

These physical quantities comply with the following equation.

$$
\begin{pmatrix}
  p_2 \\
  u_2
\end{pmatrix} = \begin{pmatrix}
  \cosh(\gamma l_i) & \rho_i c_i \sinh(\gamma l_i) \\
  -\frac{1}{\rho_i c_i} \sinh(\gamma l_i) & -\cosh(\gamma l_i)
\end{pmatrix} \begin{pmatrix}
  p_1 \\
  u_1
\end{pmatrix}.
$$

(1)

where $c_i$ is sound speed, $\rho_i$ is density, $\gamma = \alpha + 2\pi f/c_i$ is complex propagation constant of medium in the tube, $\alpha$ is the wave attenuation coefficient per unit propagating length, and $f$ is the calculation frequency. According to Harris (Harris, 1963), the wave attenuation coefficient $\alpha$ depends on the frequency, air temperature and humidity, it is about 0.012 [1/m] at 2kHz, about 0.02 [1/m] at 4kHz in the condition of the temperature 20°C and humidity 5%. Simplifying the discussion, the wave attenuation coefficient was set as $\alpha = 0.01$ [1/m] constant in this study (Investigation of dependency to the wave attenuation coefficient is the future research). The above equation could be transformed into a formula similar to the one-dimensional finite element model (1D-FEM).

$$
\mathbf{B}_i \mathbf{p} = \mathbf{u}
$$

$$
\mathbf{B}_i = \frac{1}{\rho_i c_i \sinh(\gamma l_i)} \begin{pmatrix}
  -\cosh(\gamma l_i) & 1 \\
  1 & -\cosh(\gamma l_i)
\end{pmatrix} \mathbf{p} = \begin{pmatrix}
  p_1 \\
  p_2
\end{pmatrix}, \quad \mathbf{u} = \begin{pmatrix}
  u_1 \\
  u_2
\end{pmatrix}.
$$

(2)

where $\mathbf{B}_i$ is the admittance matrix, $\mathbf{p}_i$ is the pressure vector, and $\mathbf{u}_i$ is the inside direction velocity vector of element $i$. 

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Fig. 7  Modification to the simple one-dimensional model

![Modification to the simple one-dimensional model](image1)

Fig. 8  One-dimensional acoustic element

![One-dimensional acoustic element](image2)
The two connected elements shown in Fig. 9 comply with the following equation.

\[
\begin{align*}
p_2 &= p_{2'} = p_2 \\
s_1u_2 + s_2u_{2'} &= 0
\end{align*}
\]  
(3)

![Compatibility condition diagram](image)

Fig. 9  Compatibility condition

Therefore,

\[
\begin{pmatrix}
b_{111} & b_{112} & 0 \\
b_{121} & b_{122} + b_{211} & b_{212} \\
0 & b_{221} & b_{222}
\end{pmatrix}
\begin{pmatrix}
p_1 \\
p_2 \\
p_3
\end{pmatrix}
= 
\begin{pmatrix}
u_1 \\
u_2 \\
u_3
\end{pmatrix}.
\]  
(4)

where \(b_{ij}\) is the \(i^{th}\) row \(j^{th}\) column component of admittance matrix of element \(i\) : \(\mathbf{B}_i\).

Considering these compatibility conditions, the total system formula could be built as follows.

\[
\mathbf{C}\mathbf{p} = \mathbf{u}.
\]  
(5)

where \(\mathbf{C}\) is the admittance matrix of the total system composed by elements of each admittance matrix of element \(i\) : \(\mathbf{B}_i\). Similarly, \(\mathbf{p}\) is the pressure vector and \(\mathbf{u}\) is the volume velocity vector of total system.

By solving Eq. (5) using adequate boundary conditions, which construct the inside direction velocity vector \(\mathbf{u}\), pressure vector \(\mathbf{p}\) is obtained. Fig. 10(a) shows the modification of the inside cover acoustic cavity without a tube to the eight-division model. Fig. 10(b) shows its calculation results for pressure frequency response at three noise evaluating points around the cover center and the summation of these three responses when the top end is forced at unit velocity. Here, ‘Normalized frequency’ in Fig. 10(b) means the ratio of calculation frequency to that of the 2nd acoustic mode: \(c/L\). ‘Normalized SPL’ means the sound pressure level (SPL) based on the equivalent pressure of source velocity: \(\rho \times u_0\). (This is derived from the theorem that sound pressure propagating one-dimensionally is proportional to the source velocity. If in a three-dimensional propagation like in the previous chapter, sound pressure is proportional to the source acceleration). The calculation results show that the pressure frequency response at the cover center has only one peak around the frequency of the 2nd acoustic mode, which describes qualitatively the experimental result shown by the black line in Fig. 3(b). On the other hand, that at other noise evaluating points around the cover center has three peaks corresponding to the 1st, the 2nd, and the 3rd acoustic mode. Thus only the 2nd mode could be appeared on the all three response, the authors focused on the reduction of the 2nd acoustic mode as the primary one to be prevented, and the evaluating point assumed to be the center of the cover.
As an example of the acoustic tube installation effect, Fig. 11 shows three models of inside covers with tubes and the calculation results for pressure frequency response at three noise evaluating points around the cover center. In this study, the base inside cover cavity is modeled as being divided by eight (the same as Fig. 10(a)), the acoustic tube length was set to quarter of the longest length of the cover (called ‘\( \lambda/4 \) tube’), the cross section of tube was set to 10% of the cover in the normal to the wave propagation direction, and the source was set at the top of the base inside cover cavity. As shown in Fig. 11, the original peak is divided into two or more smaller ones, and the level at the original peak frequency reduces greatly on each frequency response of three noise evaluating points. As shown in Fig. 11, peak reduction effect around the normalized frequency 1 plus or minus 10%, caused by the tube installation, could not so much changed even if the location of the noise evaluating point changed approximately \( L/8 \). However, the SPL reducing frequency range depends on the tube arrangement because of appearance of these divided two peaks which are generally known as a collateral effect in the technical field of the dynamic damper. This peaks caused by weak occurrence of the prevented mode at another frequency different to the original. For example, the addition of tubes expands the distance of two collateral peaks, tube arrangement of Model C shown in Fig. 11 (c) is best for wide normalized frequency range 0.85 to 1.15. However, the addition of tubes causes the another peaks around the original peaks, it leads that Model C is worth than the Model A shown in Fig. 11 (a) for normalized frequency 0.95 to 1.05.
Thus, it seems to be that the tube arrangement can be optimized for the designed target frequency range width. From the frequency response of the Model C, it is considered that the collateral peaks would be shown in the range in the frequencies around the original peak plus or minus 25%. So, the target frequency to be reduced is set as plus or minus 25% or narrower in this study.

3.3.2 Optimization of tube arrangement patterns

Constructible arrangements of $\lambda/4$ tubes are $2^9=512$ total patterns for a model of base inside cover cavity divided by eight. Pressure frequency responses at the cover center for all 512 arrangements are calculated by the method mentioned in the previous sub section. Fig. 12 shows the best arrangement patterns and their pressure frequency responses at the cover center for some target reduction frequency ranges when the tube length $l$ is set as $l=L/4$. Target reduction frequency ranges are set to around the original peak frequency plus or minus 0% (the original peak frequency), 5, 10, 20, and 25%. Red dashed line shows the pressure frequency response at the cover center when the tube length $l$ is 10% shorten (set as $l=0.9L/4$). Table 1 shows the pressure reduction effects of optimization of both cases on the tube length ($l=L/4$ and $l=0.9L/4$). As shown in Fig. 12, to reduce pressure at the original peak frequency, $\lambda/4$ tubes should be set at both the bottom end and the area between top end and center (Fig. 12(a)). On the other hand, to reduce pressure over a wide frequency range, tubes are set at only the area around the top without the bottom
end (Fig. 12(e)). For the target frequency range set to plus or minus 5% to 20%, tubes should be around both ends and around the center like that shown in Fig. 6.

As shown in Fig. 12, when the tube length is 10% shorter, the most reducing frequency is shifted 10% higher. Therefore, the target frequency range should be set as more than the shorten ratio of the acoustic tube length.

Optimized model for some target frequency ranges and their pressure frequency response

Fig. 13 shows the relationship between the target frequency ranges and the best optimized tube arrangement. In this figure, a black square means the tube location and a white square means the location without tube. Fig. 13(a) is the result of the case of the tube length $l=L/4$, and Fig. 13(b) shows the that of the case of the tube length is shorter 10% ($l=0.9L/4$). For example tube location for the target frequency range 0% and tube length $l=L/4$ is same as the arrangement shown in Fig. 12(a). Although, resolutions of the tube location and the target frequencies are rough, transitions of the optimized arrangement are comprehended. As shown in Fig. 13(a), the best optimized arrangement of tube length $l=L/4$ is sensitively depends on the target frequency range less than 9%. The reason can be inferred that the local small peaks near the original affect greatly to the result. For target frequency range 9 to 21%, tube arrangements like shown in Fig. 6 are the best stably. However, for the frequency range more than 22%, the optimized arrangement changes suddenly to the arrangement shown in Fig. 12(e). This reason can be explained as follows: The distance of the first divided peaks reach the limit of the expansion, for example shown in Fig. 12(d) at normalized frequency 0.78 and 1.23, target frequency range exceeds the limit distance of the first divided two peaks.
Then, the optimized arrangement becomes the one that has wide target frequency shown in Fig. 12 (e). Thus, these results explain that the proposed tube arrangement plan shown in Fig. 6 is realistic in a standpoint of the space layout, efficient optimized pattern for pressure reduction around the original peak frequency plus or minus 9 % to 21 %.

However in case of the real structure, backlash clearance would be needed to take the waves into the tubes. So, the tube length should be shortened for the backlash clearance. Fig. 13 (b) means the optimized arrangement when the backlash clearance is set as 10% of the tube length. As mentioned before, when the tube length is shortened, the most reduced frequency is shifted to higher frequency, the target frequency range should be set as more than the shorten ratio of the acoustic tube length. Therefore, as shown in Fig. 13(a) and (b), tube arrangements like shown in Fig. 6 are best for the target frequency range 10 to 17% in the both cases of the tube length tube length \( l=L/4 \) and tube length \( l=0.9L/4 \).

![Table 1](image)

**Table 1** Pressure reduction effect of optimization.

<table>
<thead>
<tr>
<th>Target Frequency range</th>
<th>Original Over-all pressure in the target range [(dB)]</th>
<th>Original Over-all pressure in the target range [(dB)]</th>
<th>Optimization Over-all pressure in the target range [(dB)]</th>
<th>Optimization Over-all pressure in the target range [(dB)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>±0%</td>
<td>40.0</td>
<td>-157.6</td>
<td>197.6</td>
<td>-26.6</td>
</tr>
<tr>
<td>±5%</td>
<td>40.3</td>
<td>-6.9</td>
<td>47.2</td>
<td>-0.5</td>
</tr>
<tr>
<td>±10%</td>
<td>40.3</td>
<td>2.6</td>
<td>37.7</td>
<td>6.6</td>
</tr>
<tr>
<td>±20%</td>
<td>40.3</td>
<td>15.6</td>
<td>24.7</td>
<td>27.3</td>
</tr>
<tr>
<td>±25%</td>
<td>40.3</td>
<td>28.5</td>
<td>11.8</td>
<td>27.6</td>
</tr>
</tbody>
</table>

(a) In case of the tube length \( l=L/4 \)

(b) In case of the tube length \( l=0.9L/4 \) (10% shorter than (a))

![Fig. 13](image)

**Fig. 13** Relationship between the target frequency ranges and the best optimized tube arrangement:

A black square means the tube location and a white square means the location without tube.
4. Validation of pressure reduction effect of proposed tube arrangement

This chapter summarizes the validation results using three-dimensional calculation and experimental measurement of the pressure reduction effect of the proposed tube arrangement for the real cover structure.

4.1 Three-dimensional calculation using boundary element method (BEM)

Pressure reduction effect of the proposed arrangement calculated by the one-dimensional TMM model mentioned in previous chapter is validated by three-dimensional calculation using the boundary element method (BEM).

4.1.1 Verification of BEM

First, the original cover model without tubes is verified by comparison with the experimental results before the calculation of the model with tubes. Fig. 14 shows the calculation model for the cover without the acoustic tubes and calculated pressure distribution results at some specific frequencies. Here, the cover dimensions are set the same as those of the experimental specimen shown in Fig. 2, the source location and sound pressure evaluating plane are defined the same as in that experiment also, and its output power is set as unit volume acceleration $1 \text{ m}^3/\text{s}^2$. Additionally, the density of the air is set as $\rho = 1.225 \text{ kg/m}^3$, the sound speed of the air is set as $c = 340 \text{ m/s}$. Comparing that calculated pressure distribution in Fig. 14 (b) with the measured one in Fig. 4, the BEM results correspond well to the measured ones with regard to occurrence frequencies and distributions of each acoustic mode. The accuracy of the occurrence frequency of acoustic mode is 3% or less.

Fig. 15 compares calculated and measured results for pressure frequency response at the cover center. As shown in this figure, the objective peak to be prevented at 445 Hz in measured results is simulated by the calculated peak at 439 Hz with an accuracy of 1%. Comparing Fig. 15 with Fig. 14(b), the calculated peak at 439Hz is caused by the 2nd acoustic mode in the direction along the longest cover length, the same as the measured one.

From above the comparison between calculated results and measured results, this BEM calculation has verified that occurrence frequencies and distributions of the 2nd acoustic mode that should be prevented.

(a) Illustration of calculation model       (b) Calculated pressure distribution result at some specific frequencies

Fig. 14  Schematic view of calculation model for cover without acoustic tube and calculated pressure distribution results
4.1.2 Calculation of pressure reduction effect using BEM

The effect of the proposed tube arrangement optimized in the previous chapter is calculated using BEM in this subsection by comparing the calculated results of the cover model with and without the tubes. Fig. 16 shows the calculation model and detailed dimensions for the effect of the cover with the acoustic tube. Here, the cross section of the tubes is defined as a square that has 50-mm-long edges and each tube is defined as 180 mm long with backlash clearance of 20 mm (10% of the ideal tube length 200 mm). Moreover, the same as in Fig. 6, tubes are set inside the cover as their open end set at both ends and the center along the longest length of the cover. The conditions of the source, location, and output power are set the same as in the previous section. The location of pressure frequency response evaluating point is set as same as previous section also.

Fig. 16(a) compares the calculated pressure level frequency response of the cover models with and without the tubes (results of the model without tubes is same as shown in Fig. 15 previously). Fig. 17(b) shows the sound pressure distributions at the specific peak frequencies in Fig. 15 and Fig. 17(a). Fig. 17(b) explains that the frequency response peak at 439 Hz on the tubeless model has disappeared due to the installation of tubes (it is shown that the sound pressure in the tubes are higher than the other area in the cover), and frequency characteristics Fig. 17(a) have a large dip around 400 to 600 Hz. The frequency at the bottom of the dip shown in pink line in Fig. 17(a) is 472 Hz (wave length $\lambda = 720$ mm), it is correspond to real tube length $\lambda/4 = 180$ mm, shifted to the higher frequency from the original peak frequency. This dip means that tubes are behaving as acoustic dynamic dampers around this frequency range, taking acoustic energy of inside cover into their tube actively.

On the other hand, other peaks are generated at 521 and 380 Hz. The peak at 521 Hz is the collateral effect mentioned in previous chapter. According to general knowledge of dynamic dampers, this collateral effect and these frequency transitions of the prevented mode occur at two divided frequencies lower and higher than original. Fig. 17(b) explains that the prevented 2nd order acoustic mode at 439 Hz in the tubeless model should occur at 330 Hz as lower frequency and 521 Hz as higher frequency in the with-tube model. In addition to this, another peak occurs at 380 Hz in these calculation results.

To summarize this section, the original peak at 429 Hz should be reduced by installing tubes. However, some peaks caused by its collateral effect must be considered. The occurrence of these collateral effects will be confirmed by experimental measurements in the next section.

![Fig. 15 Comparison between calculated and experimental results of pressure frequency response at cover inner center without tubes](image-url)
4.2 Validation of pressure reduction effect using experimental measurement

An experimental measurement was done to validate the pressure reduction effect of proposed arrangement calculated by three-dimensional calculation using BEM mentioned in previous section. Fig. 18 (a) shows a photo of the test specimen with the tubes. Here, the test specimen is made of 10-mm-thick acrylic boards and the tube is 2 mm thick. The other dimensions are the same as the BEM model: the tube cross section is square, edges are 50 mm long, each tube is 180 mm long, and the backlash clearance is 20 mm.

Fig. 18 (b) compares measured sound pressure FRFs at the inner center with and without tubes. As shown in this figure, peak reduction effect around 450 Hz obtained by BEM calculation is reproduced in this experiment. The frequency at the bottom of the dip shown in red line is 468 Hz (wave length $\lambda = 726$ mm), it is almost the same as tube length $\lambda/4 = 180$ mm. Additionally, two peaks of collateral effects forecasted by calculation are also shown at 357 Hz and 524 Hz, but these peak levels are almost 10 dB lower than the original peak level at 445 Hz.

The 'partial over all' (energy summation in the limited frequency range) of sound pressure level of frequency range around the original peak (220 ~ 680 Hz) is 85 dB with the tubes and 91 dB without them. Thus, the sound pressure reduction effect of the proposed tube arrangement is concluded to be 6 dB when the evaluation point is set at the center of the cover.
5. Conclusion

We proposed a new high-performance noise proof cover using acoustic tubes.

We investigated the effective arrangement of acoustic tubes on the inside of the cover from the standpoint of the cover's dimensions and occurrence frequency of the inner acoustic mode by using a simplified one-dimensional Transfer Matrix Method (TMM).

The results showed that the effective arrangement depends on the width of target frequency range. If the target frequency range was set as plus or minus 10 to 17% around the peak caused by the inner cover acoustic mode, for example, the most effective arrangement resulting from TMM was having tubes 1/4 as long as the longest cover edge set at both ends and at the center along the longest direction of the cover.

Finally, the effect of the proposed structure was investigated by numerical acoustic calculations using Boundary Element Method (BEM) and validated by experimental measurements. The results of BEM closely corresponded to those of the experiment, which showed that the sound pressure level was reduced about 6dB in the frequency range around the original peak frequency plus-minus 50% when the evaluation point is set at the center of the cover.

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