Tonal and broadband noise control of an axial-flow fan with metal foams: design and experimental validation

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Abstract

This paper presents an application of metal foams for controlling aerodynamic noise of an axial-flow fan. A configuration of a semi-open cell metal foam combined with a backing cavity is employed to attenuate the tonal component while an open-cell metal foam is used to absorb the broadband component. An acoustic impedance model is employed to determine the optimal geometrical parameters of the metal foam and the cavity. Experimental results confirmed that the tonal noise is greatly reduced by the semi-open cell metal foam and the open cell metal foam shows a potential of absorbing the broadband noise.

Keywords

Axial fan; Aeroacoustics; Noise control; Metal foam

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Nomenclature

\( a \) = diameter of the inner surface of the casing, m

\( b \) = diameter of the hub, m

\( d \) = pore size, mm

\( H \) = height of the cavity, mm

\( k \) = layers number of micro-perforated plates

\( L \) = thickness of the semi-open cell metal foam, mm

\( l \) = the total length of the semi-open cell metal foam, mm

\( M \) = rotational Mach number

\( M_0 \) = acoustic resistance

\( N \) = number of rotating blades of the axial-flow fan

\( n \) = rotating speed, rpm

\( R_0 \) = acoustic reactance

\( t \) = thickness of the micro-perforated plate, mm

\( \rho_0 \) = density of air, kg/m\(^3\)

\( \mu \) = dynamic viscosity of air, N·s/m\(^2\)

\( \nu \) = ratio between the frequency of sound and that of source pulsation

\( Z_0 \) = specific acoustic impedance

\( Z_1 \) = number of outlet guide vanes

\( Z_2 \) = number of upstream mounting plates
\(Z_3\) = number of downstream mounting plates

\(Z_D\) = specific acoustic impedance of the air inside the pore

\(Z(k)\) = specific surface impedance of the semi-open cell metal foam

\(\alpha\) = absorption coefficient

\(\beta\) = acoustic Reynolds number

\(\varepsilon\) = porosity

\(\Delta L\) = noise reduction level, dB

**Abbreviations**

BPF = blade passing frequency

OCMF = open cell metal foam

OASPL = overall A-weighted sound pressure level

PPI = pores per inch

SCMF = semi-open cell metal foam

SPL = sound pressure level

TPR = total pressure rise

UMP = upstream mounting plate

VFR = volume flow rate
I. Introduction

Propagation of acoustic waves in the air is a specific unsteady flow phenomenon and it can usually be treated as inviscid, as the acoustic energy loss is negligible in this situation. However, if sound propagates in micro-channels, viscous and thermal dissipations become important and will greatly affect the flow feature, generating significant acoustic energy loss along the flow direction, which is usually the acoustic propagation direction. Owing to these features, porous materials, such as glass fiber and mineral wool, are usually used as sound-absorption materials. However, these conventional sound-absorption materials do not have satisfactory moisture- and fire-proof performances, limiting their applications for practical engineering problems.

In 1970s, Maa [1] developed a micro-perforated plate (MPP) absorber which showed a good sound-absorption performance and a wide application range if the MPP is made by metals or other moisture- and fire-proof materials. There have been many examples of successful applications of MPPs in various engineering applications, examples can be found in reviews [2, 3]. Applications of the MPP in controlling aerodynamic noise of fans and engines have also been reported, see for examples [4-8].

From the viewpoint of fluid dynamics, flow loss accumulates along the flow channel length while decreases with increasing channel equivalent diameter due to reduced flow velocity, thus increasing the thickness of the MPP and decreasing the pore diameter are beneficial to damping flow energy thus achieving higher noise reduction. This was also shown by the acoustic impedance model of Maa [1], which characterized the relationship between the acoustic impedance and the structural parameters of the MPP. However, it is usually technically difficult or expensive to produce the MPP with high thicknesses and small pore sizes.
Since 1990s, a new type of porous material, metal foam, has shown appealing performances of sound-absorption and of moisture- and fire-proof characteristics. According to the connectivity of its cellular structures, the cell of the metal foam can be classified into the following three categories: open cell, semi-open cell and closed cell. The closed cell metal foam is not a good sound-absorption structure because air cannot go through the neighboring cells [9]. The features of irregular, sinuous and connective micro-channels make the open cell metal foam (OCMF) an efficient sound-absorption material. OCMFs have been applied to control aerodynamic noise of airfoil trailing edges [10, 11] and high-speed train pantographs [12, 13]. Recently, OCMFs have also been used to reduce turbofan [14, 15] and centrifugal fan noise [16, 17].

It is worth mentioning that the noise control mechanism of the OCMF used in references [10-13, 16] is completely different from that used in references [14, 15] owing to the different values of pores per inch (PPI) of the porous materials. OCMFs with a high PPI, such as PPI≈40, 60 and 80 used in references [14, 15], absorb the acoustic energy via viscous and thermal dissipations, whereas OCMFs with a low PPI, such as PPI≈20 used in references [10, 16], control noise by suppressing the wall pressure fluctuation, which is the acoustic dipole source. This feature is consistent with the conclusions of Maa [1, 18-20], where significant viscous and thermal dissipations occur only in micro-channels in the order of sub-millimeter. The acoustic model of Maa [1] also indicated the sound-absorption performance of micro-channels is highly dependent on the frequency of sound. Especially, the sound-absorption coefficient is usually small at low frequencies because the absolute value of the negative acoustic reactance (imaginary part of the complex acoustic impedance) increases with the decrease of the sound frequency and causes an unsatisfactory sound-absorption performance at low frequencies. This conclusion has been validated by the experimental results given in references [14, 15], where the sound-absorption
coefficient sharply reduces with the decrease of the sound frequency and noise level is hardly reduced at low frequencies (see Fig. 6 and Fig. 16 in reference [14], respectively).

Maa developed an efficient technique to improve the sound-absorption performance at low frequencies, where a cavity is constructed at the back of the MPP to generate the Helmholtz resonance. This combination of MPP with backing cavity produces a peak sound-absorption at the cavity resonant frequency, which is dependent on the cross-sectional area of the microchannel and the volume of the backing cavity. Owing to the preceding features, a good sound-absorption performance at low frequencies can be achieved by adjusting the volume of the backing cavity. Inspired by the above technique, it is natural to combine the metal foam with a cavity to control the low-frequency noise.

Moreover, the acoustic model of Maa suggested that an improved sound-absorption performance could be achieved at a wider frequency range if multi-layer MPPs with backing cavities are assembled successively. But multi-layer MPP absorbers have not been widely used because of difficulties in its structural design and manufacture [21]. In 1990s, a new type of metal foam with semi-open cell was developed which has a similar structural feature to the multi-layer MPP absorber. Experimental and theoretical studies of Lu et al. [22] showed that the acoustic model of Maa for multi-layer MPPs was also valid for determining the structural parameters of the semi-open cell metal foam (SCMF). With this acoustic impedance model, Meng [23] optimized the structural parameters by adopting the genetic algorithm and indicated that the distribution of graded geometrical parameters affects significantly the acoustic absorbing performance of SCMFs. Recently, Xu et al. [17] reduced the overall noise of a centrifugal fan by up to 6dB using this SCMF, and showed its potential in controlling aerodynamic noise of
turbomachines. In this paper, metal foams are applied to control aerodynamic noise generated from an axial-flow fan.

The remainder of this paper is organized as follows. Section 2 describes the main geometrical and operating parameters of the axial-flow fan and analyzes the noise generation and radiation features. In Section 3, a hybrid acoustic liner made by SCMF and OCMF are employed to control the tonal and broadband noise, respectively, and the method of Lu [22] is used to determine the structural parameters of the SCMF. In Section 4, an experimental test is carried out to analyze the effect of metal foams on the aerodynamic performance and sound pressure level (SPL) of the axial-flow fan. Conclusions are given in Section 5.

2. Noise generation and radiation features of axial-flow fan

Since the surface impedance and the absorption coefficient of an acoustic liner are functions of the frequency, detailed information of the sound spectrum are beneficial to provide an input for designing efficient acoustic liners. Numerical simulations and experimental tests can be employed to obtain the sound spectrum, however manufacturers prefer simpler methods to design liners at low cost and quick turnaround. Thus, a more efficient and simpler method, with suitable assumptions in computational modelling and reasonable accuracy, is welcomed by manufacturers to guide the design of acoustic liners. In this section, an axial-flow fan is described and an acoustic model is employed to analyze the characteristics of the acoustic power spectrum of the investigated axial-flow fan, which will be utilized in the next Section to design the acoustic liners for tonal and broadband noise treatment.

2.1 Main parameters of the Axial-flow fan

Fig. 1 shows a sketch of the single-stage axial-flow fan studied in this paper. The stage of this axial-flow fan comprises of 26 rotating blades (RBs) and 17 outlet guide vanes (OGVs). Eight
upstream mounting plates (UMPs) and three downstream mounting plates (DMPs) are fixed to support the casing. Some characteristic parameters of the fan are listed in Table 1. Note that the primary aim of this investigation is to control the noise at the outlet of the fan, thus acoustic liners are chosen to be fixed downstream of the fan.

Fig. 1 Schematic of acoustic liner installed in the axial-flow fan

Table 1 characteristic parameters of the axial fan

<table>
<thead>
<tr>
<th>No</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Nominal rotating speed $n$ (rpm)</td>
<td>2800</td>
</tr>
<tr>
<td>2</td>
<td>Diameter of the inner surface of the casing $a$ (m)</td>
<td>0.7</td>
</tr>
<tr>
<td>3</td>
<td>Diameter of the hub $b$ (m)</td>
<td>0.39</td>
</tr>
<tr>
<td>4</td>
<td>Number of rotating blades $N$</td>
<td>26</td>
</tr>
<tr>
<td>5</td>
<td>Number of outlet guide vanes $Z_1$</td>
<td>17</td>
</tr>
<tr>
<td>6</td>
<td>Number of upstream mounting plates $Z_2$</td>
<td>8</td>
</tr>
<tr>
<td>7</td>
<td>Number of downstream mounting plates $Z_3$</td>
<td>3</td>
</tr>
<tr>
<td>8</td>
<td>Blade passing frequency $f_{BPF} = nN/60$ (Hz)</td>
<td>1213.3</td>
</tr>
</tbody>
</table>

2.2 Analysis on Noise Generation of Axial-flow fan

Researches on axial-flow fan aerodynamics and aeroacoustics revealed the following phenomena as potential reasons of aerodynamic noise for axial-flow fan.
(1) Interactions of RBs with UMPs and OGVs. Although these interactions radiate both tonal and broadband noise [24-26], the tonal components are usually dominating. There are two mechanics responsible for these periodic interactions [24]. The first is the potential interaction, which causes pressure fluctuations on both the upstream and downstream blades surfaces. Some investigators have suggested that the potential interaction dominates only when the separation of the adjacent cascade rows is less than 50% of the blade chord [27, 28]. For the axial-flow fan studied in this work, the chords of the UMPs and OGVs are very long, so we believe that the potential interaction is a possible reason generating pressure fluctuations on blade surfaces. The second is the wake interaction, which only causes the pressure fluctuation on the downstream blades owing to the impingent of the wake flow from upstream blades. Based on the above analysis, Table 2 lists the dominant frequencies of the wall pressure fluctuation due to the periodic potential and wake interactions on different fan components. The dominant frequency of the wall pressure fluctuation on both the UMPs and OGVs is equal to the BPF. However, the wall pressure fluctuation on the RBs is dominated by two different frequencies. At the leading edge of the RBs, the dominant frequency is $Z_2n/60 = 373.3\text{Hz}$ due to the potential and wake interactions between the UMPs and the RBs, whereas the dominant frequency at the RB trailing edge is $Z_1n/60 = 793.3\text{Hz}$ due to the potential interaction between the RBs and the OGVs. For noise radiated from the stationary UMPs and OGVs, the sound frequency is equal to the frequency of the wall pressure fluctuation, but tonal noise radiated from the RBs is equal to the BPF and its higher harmonics even the wall pressure fluctuation is dominated by two different frequencies [29].

(2) Incoming turbulence, vortex shedding from trailing edge (TE) [30], tip clearance (TC) flow [31], etc. These mechanisms also generate both tonal and broadband noise. Note that the
frequencies of the tonal noise radiated from the periodic potential and wake interactions depends on the rotating speed, the frequency of tonal noise due to these mechanisms (i.e. TE noise and TC noise) is also related to the mass flow rate (local flow velocity) \([30, 31]\). It has been found that TE and TC noise dominate only when there is a strong vortical flow, and we usually can alleviate this phenomenon by reducing the TE thickness and the radial gap between the blades tip and the casing. Therefore, in this paper, we don’t consider the tonal noise due to TE and TC flow. Experimental results in Section 4 also indicate that the primary mechanism of tonal noise generation for the present fan is periodic interaction because the discrete frequencies of tonal noise is related to the rotating speed rather than the mass flow rate.

Based on the above qualitative analysis, we can conclude that tonal noise is dominant for this axial-flow fan, and the UMP-RB and RB-OGV interactions are the main reasons generating pressure fluctuations on the blade surfaces. The corresponding dominant frequencies for toner noise given in Table 2 will be used in the design of the acoustic liners.

**Table 2 Dominant frequencies of wall pressure fluctuation due to periodic interaction**

<table>
<thead>
<tr>
<th>Components</th>
<th>Dominant frequency</th>
<th>Mechanics</th>
</tr>
</thead>
<tbody>
<tr>
<td>UMPs</td>
<td>( f_{BPF} = Nn/60 = 1213.3 \text{Hz} )</td>
<td>Potential interaction between UMPs and RBs</td>
</tr>
<tr>
<td>RBs</td>
<td>( Z_2n/60 = 373.3 \text{Hz} )</td>
<td>Potential and wake interactions between UMPs and RBs</td>
</tr>
<tr>
<td>RBs</td>
<td>( Z_1n/60 = 793.3 \text{Hz} )</td>
<td>Potential interaction between RBs and OGVs</td>
</tr>
<tr>
<td>OGVs</td>
<td>( f_{BPF} = Nn/60 = 1213.3 \text{Hz} )</td>
<td>Potential and wake interactions between RBs and OGVs</td>
</tr>
</tbody>
</table>
3. Acoustic Liner Design

3.1 Semi-open and Open Cell Metal Foams

As shown in Fig. 1, a hybrid acoustic liner is designed to control the axial-flow fan noise, where an SCMF combined with a backing cavity is used to reduce the tonal component at the BPF and an OCMF is used to absorb the broadband component. The main geometrical parameters of these metal foams are listed in Table 3.

Table 3 Geometrical parameters of metal foams

<table>
<thead>
<tr>
<th>Metal foams</th>
<th>Porosity (%)</th>
<th>Pore Size (mm)/PPI</th>
<th>Pore Opening (mm)</th>
<th>Thickness (mm)</th>
<th>Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Semi-open cell</td>
<td>65%</td>
<td>~4.0/6</td>
<td>~0.7-0.8</td>
<td>20</td>
<td>0.2+0.3(Two sections)</td>
</tr>
<tr>
<td>Open cell</td>
<td>96%</td>
<td>~0.6/40</td>
<td>-</td>
<td>10</td>
<td>0.3</td>
</tr>
</tbody>
</table>

3.2 Surface Impedance Model of Metal Foams

The SCMF is flush mounted on the casing downstream of the RBs. Note that the SCMF composes of two sections, as shown in Fig. 1, to fit long enough SCMF without altering the current fan geometry. The specific impedance \( Z_0 \) is used to characterize its acoustic absorbing performance, it is a complex number defined by

\[
Z_0 = (R_0 + iM_0)/\rho_0 c_0
\]

(1)

where \( R_0 \) and \( M_0 \) are the acoustic resistance and the acoustic reactance, respectively. For the porous material, these two parameters are functions of the acoustic Reynolds number \( \beta = 0.5d\sqrt{\omega \rho_0/\mu} \), where \( d \) is the equivalent diameter of the porous material (pore opening of the SCMF), \( \rho_0 \) is the density of air and \( \mu \) is the dynamic viscosity of air. As Reynolds number is the ratio between the inertial force and the viscous force, in order to achieve a good acoustic absorption performance via viscous and thermal dissipations, the acoustic Reynolds number...
should be small enough. The corresponding thickness of the acoustic boundary layer is about half of the equivalent diameter of the porous material (pore opening of the SCMF). It is suggested by Ingard [32] that the equivalent diameter of the porous material should be in the order of sub-millimeter to achieve a good acoustic absorption performance in air.

Fig. 2 Schematic of the SCMF: (a) idealized cross-section (b) electroacoustic analogy

Maa [18] developed an acoustic model for an MPP with a backing cavity, whose specific acoustic resistance and reactance are expressed as follows

\[
R_0 = \begin{cases} 
\frac{32\mu}{d^2} \left[ t + \frac{8d}{3\pi} \right] & \beta < 1 \\
\frac{32r\mu}{d^2} \left[ \sqrt{1 + \beta^2/32} + \sqrt{\beta d/4t} \right] & 1 < \beta < 10 \\
\frac{8\mu\beta}{\sqrt{2}d^2} \left[ t + \frac{8d}{3\pi} \right] & \beta > 1 
\end{cases}
\]

(2)

\[
M_0 = \begin{cases} 
4\omega\rho_0 (t + \frac{8d}{3\pi})/3 & \beta < 1 \\
\omega\rho_0 \left[ 1 + 1/\sqrt{9 + \beta^2/2 + 0.85d/t} \right] & 1 < \beta < 10 \\
\frac{8\beta\mu(t + \frac{8d}{3\pi})}{\sqrt{2}d^2} + \omega\rho_0 (t + \frac{8d}{3\pi}) & \beta > 10 
\end{cases}
\]

(3)

where \( t \) is the thickness of the perforated layer (i.e., the length of the micro channel). With this acoustic model, Lu et al. [22] showed that the SCMF can be regarded as multi-layer micro-perforated panels with backing cavities, as sketched in Fig. 2.
In Fig. 2, $L$ is the thickness of the SCMF, $D$ is the pore size of the SCMF, $Z_0$ is the specific acoustic impedance of the pore opening computed by Eqs. (1)-(3) and $Z_D$ is the specific acoustic impedance of the air inside the pore, which can be computed by $Z_D = -i \cot(0.909 \omega D/c_0)$. Unlike the MPP, the parameter $t$ in Eqs. (2) and (3) cannot be measured directly for the SCMF, and it is computed by the following approximation $t \approx (1-\varepsilon)D \frac{(1-\varepsilon)D}{3.55 - \frac{6(d/D)^2}{d/D}}$, where $\varepsilon$ is the porosity of the SCMF [22]. The number of layers for the SCMF is computed by $n = \lfloor L/0.806 D \rfloor$, where $\lfloor \cdot \rfloor$ means round-off calculation. With the above approximations, the specific acoustic impedance of the SCMF is computed by

$$Z^{(n)} = \begin{cases} 
Z_0 + \frac{1}{Z_D + \frac{1}{Z^{(n-1)}}} & n > 1 \\
\frac{1}{Z_D} + \frac{1}{Z^{(n-1)}} & n = 1 
\end{cases} \quad (4)$$

where $Z^{(n)}$ represents the specific surface impedance of the SCMF, which is equal to that of MPP with $k$ layers. Note that this formulation is suitable for the SCMF directly placed on a rigid wall. If there is a cavity with a height of $H$ between the SCMF and the rigid wall, the specific acoustic impedance is computed by

$$Z = \frac{1}{Z^{(n)} + \frac{i}{\cot(\omega H/c_0)}} \quad (5)$$

It can be found that Eq. (5) reduces to Eq. (4) if the height of the cavity is equal to zero. With the above model of the specific surface impedance, the acoustic absorption coefficient is computed as follows

$$\alpha = \frac{4 \text{Re}(Z)}{[1 + \text{Re}(Z)]^2 + [\text{Im}(Z)]^2} \quad (6)$$
which implies that the acoustic absorption coefficient achieves the maximum when the specific acoustic resistance and reactance are equal to one and zero, respectively.

Considering a configuration with SCMF on a rigid wall, the effect of the SCMF thickness on the absorption coefficient is analyzed by employing its structural parameters listed in Table 3 using the empirical formula given above. Fig. 3 shows that increasing the SCMF thickness improves its acoustic absorption performance. Moreover, the peak frequency of the absorption coefficient is greatly affected by the SCMF thickness, and the acoustic absorption performance in low frequencies can be improved by increasing the SCMF thickness. As analyzed in Section 3, the dominant frequency of tonal noise radiated from the investigated fan is the BPF (1213Hz), thus a sample of SCMF with thickness 20mm is recommended to be installed as the acoustic liner downstream of the fan. However, the absorption coefficient of this SCMF at the BPF is around 0.8 and a further increase of the absorption coefficient is beneficial to achieve greater reduction in noise level.

Fig. 3 Variation of the absorption coefficient with the thickness of the SCMF

Fig. 4 displays the effect of the height of the backing cavity on the absorption coefficient with the SCMF thickness fixed to 20mm. It is observed that the height of the backing cavity has no obvious effect on the peak frequencies of the absorption coefficient. Increasing the height of the backing cavity increases the absorption coefficient at the BPF, however, this leads to a decrease
in the bandwidth of the effective sound absorption. Additionally, the absorption coefficient at the BPF is insensitive to the height of the backing cavity when it is larger than 30mm. Thus, we select the height of the backing cavity as 20mm to balance the peak value and the bandwidth of the high absorption coefficient.

![Graph showing variation of absorption coefficient with height of backing cavity](image)

Fig. 4 Variation of the absorption coefficient with the height of the backing cavity

![Photos of acoustic liner installed in the axial-flow fan](image)

Fig. 5 Photos of acoustic liner installed in the axial-flow fan: (a) front view; (b) back view

Note that a drawback of the SCMF, similar to MPP absorbers, is the difficulty to achieve a good acoustic absorption performance in a wide frequency range. The experimental results of the acoustic impedance given in references [14, 15] for samples of OCMF with PPIs from 40 and 60 have shown a good acoustic absorption performance in wide frequency range, and these samples of OCMF have been successfully applied to control the turbofan noise. Recently Xu and Mao [16,
employed OCMFs with PPIs of 20, 40, 70 and 80 to reduce a centrifugal fan noise. However, to the best knowledge of the authors, no acoustic model has been developed for the OCMF to characterize the relationship between the acoustic absorption coefficient/acoustic impedance and the structural parameters.

Detailed research on this topic is ongoing, and this paper studies experimentally the effect on the broadband noise reduction of the OCMF, whose structural parameters are listed in Table 3. As shown in Fig. 5, the OCMF is installed in the downstream flow channel of the rotating blades, where metal foams with high value PPI are applied aiming to absorb the acoustic energy via viscous and thermal dissipations in the micro-channel with the diameter in the order of sub-millimeter [9].

4. Experimental Validation and Discussion

Experimental tests were carried out to measure the aerodynamic performance and noise of the axial-flow fan. A standard test facility was manufactured at the National Industrial Fan Testing Center of China according to China Standards GB/T 1236-2000: Industrial fan performance testing using standard airways, and GB/T 2888-1991: Methods of noise measurement for fans, blowers, compressors and roots blowers. The volume flow rate (VFR), total pressure rise (TPR) and A-weighted SPL were measured for the fan operating at different VFRs. An in-house system was used to measure the aerodynamic performance, and an LMS SCADAS Mobile SCM01 system was employed to sample and analyze the acoustic signals. For the acoustic experiments, two PCB microphones were placed 1m at the side of the upstream inlet and the downstream outlet of the experimental facility, respectively.

Since the experimental facility is quite large, the tests were carried out in a test plant without acoustic treatment on the surrounding walls and ground. Before the tests, the background noise
was measured, and its A-weighted SPL spectrum is given in Fig. 6. The overall SPL of the background noise was about 70dB. Experimental results of the OASPL of the fan operating at different VFRs are displayed in Fig. 7. The original fan noise is generally above 115dB, which is over 45dB higher than the background noise, implying the effect of the background noise on the acoustic measurement results can be neglected.

![Fig. 6 A-weighted SPL of background noise](image)

![Fig. 7 Variation of OASPL with the VFR: (a) fan outlet; (b) fan inlet](image)

Experimental results indicate that the acoustic liner significantly reduces noise at the fan outlet (Fig. 7a) whereas it is not beneficial in reducing noise at the fan inlet (Fig. 7b). This result validates that the acoustic liners installed downstream of the fan only control the noise at the fan.
outlet, which is consistent with the purpose of the acoustic liner design. Because of this feature, from now on we concentrate on the acoustic measurement results at the fan outlet.

Experimental results also show that the SCMF with the backing cavity greatly reduces the overall A-weighted sound pressure level (OASPL) by about 5dB and the OCMF further reduces the OASPL by about 2dB for the fan over the whole operating range. Note that the OASPL of the axial-flow fan varies with the VFR, but the noise reduction level is almost independent of the VFR. This result implies that the liners are effective over the measured VFRs and the air flow has little effect on the surface impedance of the acoustic liner if the Mach number of the air flow is very low, which does not exceed 0.2 in this axial-flow fan. This experimental result also indicates that the acoustic model given in Section III works reasonably well and the assumption to ignore the effect of air flow on the surface impedance is acceptable.

Before analyzing the noise reduction mechanism of the acoustic liner, experimental results given in Fig. 8 show that both types of metal foams influence the aerodynamic performance of the fan, which has also been reported in references [14, 15]. This drawback widely exists for
acoustic absorbers owing to viscous dissipative effects. For the current axial-flow fan, the two acoustic treatments invoke reduction of the TPR of about 4% and 8%, respectively.

![Figure 9](image)

**Fig. 9** A-weighted sound pressure spectra at the best efficiency point: (a) comparison between the original fan and the SCMF; (b) comparison between SCMF and the combination of SCMF with OCMF.

A-weighted SPL spectra were measured for the fan operating at the best efficiency point, they are compared in **Fig. 9** for the two noise treatments with the original fan. The results show that tonal components dominate for the investigated axial-flow fan, and discrete frequencies of tonal noise are related to the BPF owing to the interaction between the rotating blades and the adjacent stationary components, which is consistent with existing studies [14, 28]. The measurement results revealed the following features of the fan noise control:

1. As shown in **Fig. 9(a)**, the acoustic liner made from SCMF significantly reduces the tonal noise at the BPF, showing the surface impedance model for the SCMF used in designing the acoustic treatment provides a good tool for acoustic estimation.

2. As shown in **Fig. 9(b)**, the acoustic liner made from OCMF reduces the broadband noise, but the level of noise further reduction is very limited compared with the acoustic liner made
from SCMF. An acoustic model is expected to establish the relationship between the acoustic impedance and the structural parameters of the OCMF in future research.

Moreover, a simple and fast method for predicting the noise reduction level is beneficial for acoustic designers to analyze the effect of acoustic liners in reducing the fan noise. Obviously, many factors, such as the surface impedance and the length of the acoustic liner, affect its noise reduction capability. Since the cross-section of the flow channel is circular and the SCMF is installed in the inner surface of the casing, the following approximation for estimating the noise reduction level of the duct muffler is employed [33].

\[
\Delta L = 6.14 \cdot l \cdot \sqrt{\frac{2k \text{Im}(Z)}{|Z|^2 R} - k^2 + \left(\frac{2k \text{Im}(Z)}{|Z|^2 R} - k^2\right)^2 + \left(\frac{2k \text{Re}(Z)}{|Z|^2 R}\right)^2}
\]  \(7\)

where \(k\) is the wave number, \(R\) is the radius of the inner surface of the casing. In this research, the total length of the SCMF is \(l=0.5\)m, which is combined by two sections due to the limitation of installation position.

Eq. (7) indicates that increasing the length and reducing the radius of the acoustic liner are beneficial to achieve better acoustic absorption performance, also shows that the noise reduction level reaches the maximum when the real and imaginary parts of the specific acoustic impedance are equal to one and zero, respectively. It should be noted that Eq. (7) is derived for plane waves, so the noise reduction level is assumed to be proportional to the length of the acoustic liner and is independent of the position of the acoustic liner. However, the sound wave generated from the axial fan is obviously not a plane wave, so this discrepancy is likely to generate errors in computing the noise reduction. Fig. 10 displays variation of the predicted noise reduction level with the frequency by calculating from Eq. (7). The result indicates that there are two peaks in
the noise reduction level at around 1200Hz and 3500Hz, respectively, and the first frequency is approximately equal to the BPF.

Fig. 10 Predicted noise reduction levels for SCMF

Comparing the experimental result shown in Fig. 9(a) with the predicted result shown in Fig. 10 showed that both the experimental and predicted results display a great noise reduction at the BPF, with a measured reduction of 12dB, which is lower than the predicted result of about 16dB as shown in Fig. 10. Fig. 10 also shows that a significant noise reduction at 3500Hz can be achieved but this is not found in the experimental result. The above results conclude that the formulation for analyzing the noise reduction of the duct muffler could only approximately estimates the noise reduction level and its frequency of the acoustic liner flush mounted in the casing of the axial-flow fan, further work would develop a more accurate method for noise reduction prediction by considering the influences of blade rotation on sound radiation and casing scattering on sound propagation.
5. Conclusion

This paper presents an application of metal foams to control the tonal and broadband noise of an axial-flow fan. In order to suppress the tonal component, an acoustic impedance model is used to design the structural parameters of the SCMF and the backing cavity, and the OCMF with a PPI of 40 is employed to attenuate the broadband component. Experimental results validate the efficiencies of the SCMF and OCMF to control the tonal and broadband noise, respectively. It is expected to improve acoustic models of the SCMF and OCMF for accurately describing the acoustic absorption in micro-channels in future research.

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