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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.

1. 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 is proportional to the length of flow channel and inversely proportional to the equivalent diameter of flow channel, thus increasing the thickness of the MPP and decreasing the pore diameter of the MPP are beneficial to reach higher noise-reduction levels via increasing the flow loss. This feature 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 Refs. [10–13,16] is completely different from that used in Refs. [14,15] owing to the different values of pores per inch (PPI) of the porous materials. OCMFs with a high PPI, such as PPI $\approx$ 40, 60 and 80 used in Refs. [14,15], absorb the acoustic energy via viscous and thermal dissipations, whereas OCMFs with a low PPI, such as PPI $\approx$ 20 used in Refs. [10,16], control noise by suppressing

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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
ρ_0	density of air, kg/m^3
μ	dynamic viscosity of air, N s/m^2
V_1	number of outlet guide vanes
V_2	number of upstream mounting plates
V_3	number of downstream mounting plates
Z_0	specific acoustic impedance
Z_D	specific acoustic impedance of the air inside the pore
$Z(k)$	specific surface impedance of the semi-open cell metal foam
α	absorption coefficient
β	acoustic Reynolds number
ε	porosity
ΔL	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

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 Refs. [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 Figs. 6 and 16 in Ref. [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 micro-channel 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 and Mao [17] reduced the overall noise of a centrifugal fan by up to 6 dB 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 is 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, the fundamental noise characteristics of an axial-flow fan is analyzed, 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 composes 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 in all the experimental tests. 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.