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Planar acoustic notch filter for low frequency sound wave suppression

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ABSTRACT

In this study, a subwavelength acoustic notch filter with a tapered neck is proposed. This type of structure is capable of suppressing tonal noise at relatively low frequencies, but with a thickness of 1/16 of the targeting sound wavelength. Unlike the traditional acoustic Helmholtz resonator, which is bulky for low frequency noise control, the proposed structure can be formed as a planar acoustic notch filter, occupying a small volume and making the system compact. It is found that when the proposed units are mounted on a waveguide, the achieved noise reduction performance has a close relationship to the separation distance of the resonators. Near total reflection can occur when the propagation phase lag comes up to 90 degrees if acoustic damping is neglected. The damping influence has also been discussed, indicating damping should be reduced for this application. In the experimental study, since the tapered form can reduce damping, it is demonstrated that such sample can have better sound amplification performance compared to the uniform sample. The tapered samples are then used to suppress tonal noise emitted by a loudspeaker. At the optimal configuration, measured results show sound pressure can be suppressed by 84 percent when the units start to work. The proposed planar acoustic notch filter can be used when ventilation and noise suppression are simultaneously required in a narrow space.

Introduction

Noise suppression and insulation have always been a great concern for modern industry and transportation. Large noise levels bring about an uncomfortable and harmful influence on human beings, exciting the structure and shortening the service life [1]. In general, the abatement of relatively low frequency acoustic noise (below 1 kHz) faces the toughest solving approach: since the acoustic wavelength in this band is long, this type of noise makes it difficult to be absorbed or isolated.

Various approaches have been attempted to solve this task. Traditionally, the passive control method, such as porous material, can be used to efficiently absorb broadband sound energy when its thickness is up to a quarter wavelength of the interesting sound wave. However, the porous material has to be very thick for the low frequency noise and its performance deteriorates sharply in a dusty environment. For narrow band noise reduction, Helmholtz resonator [2,3], or quarter wavelength resonator can be an alternative approach [4,5]. However, to suppress low frequency noise, these apparatuses occupy a large space. For instance, to suppress the 500 Hz noise, the length of the quarter wavelength resonator can be up to 17.2 cm and needs to become bulkier to lower frequency noise suppression. This hinders the installation of this type of device for narrow spaces.

Alternatively, active noise control or active structural acoustic control can be adopted to suppress low frequency noise [6–8]. However, in the active control system, a sensor, actuator, real-time controller, signal conditioning, and power amplifier are all required in the system – increasing the overall cost of the system. Meanwhile, the control law may suffer from instability problems, and the power supply may not be satisfied or allowed in many severe applications [9].

Recently, the development of acoustic metamaterials have brought new insights to the low frequency noise control [10]. For instance, the breakthrough membrane type acoustic metamaterial, using tensioned thin film and proof mass, can realize excellent noise insulation performance in the low frequency range and break the mass law [11]. In further step, the meta cells with different configurations can be assembled into very thin and lightweight plate type acoustic metamaterial structures [12–14], realizing noise insulation for broadband application.

For the duct noise control application, the membrane and plate acoustic metamaterials block air ventilation. This problem draws the attention of many researchers and generates some interesting results. For instance, it is found the coupling of Helmholtz resonators' [15,16], as well as the coupling of novel hybrid membrane resonators [17], can be substantially increased with proper structure design, realizing

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enhanced noise reduction performance.

Meanwhile, the space coiling concept indicates that a compact acoustic device can be fabricated to suppress low frequency noises [18–20]. However, increasing the coiling length can increase the viscous loss of the air passage, reducing the quality factor of the reactive acoustic device.

Inspired by the above findings, a compact and passive noise control apparatus is demanded to suppress low frequency noise without simultaneously hindering airflow. In this study, a subwavelength acoustic notch filter is proposed, which is based on Helmholtz resonator theory but differs in that it occupies a small space.

The detailed structural descriptions of the acoustic notch filter unit are given in Section "Proposed acoustic notch filter unit". In Section "Numerical analysis", numerical simulation is carried out to predict its acoustic behavior, and two identical resonators with different configurations are adopted to realize excellent waveguide noise suppression. With proper design, their coupling can be substantially increased, which is favorable for boosting noise suppression performance. In Section "Experimental study", the proposed samples are fabricated and the sound suppression performance is evaluated via experimental study. The test results show the proposed device can simultaneously realize satisfied noise suppression and retain air ventilation functions.

Proposed acoustic notch filter unit

Theory of Helmholtz resonator

The Helmholtz resonator belongs to a reactive acoustic component, which can generate sound reflection to the incident sound wave, producing a sound dip effect at specific frequencies.

As shown in Fig. 1, a conventional Helmholtz resonator is composed by a large volume cavity and an opening at the end of an extended neck.

Through an acoustical-mechanical analogy, the acoustic cavity behaves like a mechanical spring and the air at the neck behaves like a proof mass. Accordingly, acoustic resonance can occur at specific frequencies, which is similar to a mechanical resonator.

The acoustic resonance frequency can be predicted using Eq. (1).

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{l'V}} \tag{1}$$

where c is sound speed; S represents the cross-sectional area of the neck; V is the volume of cavity; and l' is the modified neck length, which is the summarized result of the original neck length and the end correction length.

According to Eq. (1), when the cross sectional area of the neck is kept constant, increasing the length of the neck or the volume of the cavity is feasible to generate lower acoustic resonance frequency, making the acoustic resonator bulky. When the length of the neck and the volume of the cavity are kept as constant, decreasing the cross sectional area of the neck is also applicable to generate lower acoustic resonance. However, the vicious loss can be substantially increased with a narrow neck, as the vicious loss is in reverse proportion to the cubic order of the neck's radius [21].

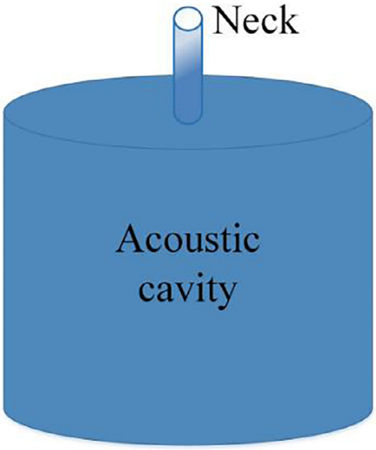

Fig. 1. Schematic diagram of Helmholtz resonator.

Description of the proposed acoustic notch filter unit

Based on the Helmholtz resonator theory, the schematic diagram of proposed acoustic notch filter unit is shown in Fig. 2.

The proposed unit is in a cube format (40 mm × 40 mm × 40 mm) and is composed of an embedded neck, an acoustic cavity, and rigid of surrounding walls. The thickness of wall is 2 mm. An open inlet is located at one side of the wall, linking the outside surrounding air and the internal acoustic cavity via the embedded neck. Here, the width of the neck at the inlet is 3 mm. The width of the neck is gradually increased from the inlet to the end. At the end of the neck, its width is increased to 5 mm.

Compared with Fig. 1, the proposed unit can be seen as a kind of Helmholtz resonator. However, unlike the conventional Helmholtz resonator, two major distinctions exist. Firstly, the acoustic neck is placed within the structure, rather than an extended configuration, making the overall structure very compact. Secondly, rather than a uniform neck configuration, a tapered form is adopted for the embedded neck.

When the tapered neck form is adopted, the geometric transition between the neck and the acoustic cavity will become smoother than the classical Helmholtz resonator – friction loss can be substantially decreased and better impedance matching can be achieved [22]. When acoustic resonance occurs, the reactance part becomes zero and only the resistance part exists. Correspondingly, decreasing the sound resistance will increase the quality value of the resonant system, and the tapered configuration can achieve better noise suppression than the classical uniform neck configuration [23]. Notably, as the neck's width is gradually increased, the tapered form acoustic resonance frequency is higher than the uniform case, which can be found from Eq. (1). Other non-linear tailored neck configurations have also been investigated [24], generating similar conclusions to the tapered neck configuration.

Owing to the fact that the neck is not uniform and an embedded neck configuration is used in this study, it would be difficult to derive a formula to calculate the exact acoustic resonance frequency. Therefore, a numerical study is carried out in Section "Numerical analysis" to calculate the model's acoustic resonance frequency and disclosing its acoustic characteristics.

Numerical analysis

Acoustic resonance prediction of single unit

Numerical simulation is carried out in the COMSOL$^{\text{TM}}$ finite element analysis environment and the detailed computation theories can be found in the relevant Refs. [25–27].

In this numerical model, pressure acoustics and frequency domain are selected to disclose the acoustic properties of the proposed acoustic notch filter unit. At the inlet, the incident sound pressure is set to 1 Pa. The calculated result shows the acoustic resonance occurs at 551 Hz and the corresponding Sound Pressure Level (SPL) distribution is shown in Fig. 3.

As shown in Fig. 3, the incident sound pressure can be significantly amplified at the acoustic resonance frequency. In contrast, when the frequency is off the resonance frequency, the sound pressure amplification gain is sharply reduced.

In this simulation, the acoustic fluid is assumed to be inviscid, which means acoustic damping is omitted. Correspondingly, the sound pressure amplification gain at the resonant frequency is over-predicted. This simulation is essential because it confirms that the proposed unit can