Design and Analysis of a Compact Sound Absorber Made of Multiple Parallel Helmholtz Resonators

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1 Introduction

The noise reduction at multiple frequencies simultaneously remains a challenge in many engineering applications such as aerospace, industry, and ground transportation. Yang et al. [1] proposed an acoustic metamaterial made of multiple parallel hexagonal Helmholtz resonators (HR) and demonstrated its sound absorption performance at low frequency. Guo et al. [2] investigated a checkerboard sound absorber made of inhomogeneous HR with extended necks and showed dual absorption peaks. Laly et al. [3] presented a design of acoustic metamaterial consisting of parallel Helmholtz resonators that are periodically embedded within a porous material and illustrated multiple transmission loss (TL) peaks.

In this paper, a compact sound absorber made of twelve parallel Helmholtz resonators is proposed for noise reduction at multiple frequencies. The global cylindrical cavity of the material is partitioned into twelve sub-cavities and one extended neck is connected to each sub-cavity. The sound absorption coefficient and the TL obtained using finite element method present twelve resonant peaks at different frequencies where the surface impedance matches with the air impedance.

2 Finite element design and analysis of the proposed sound absorbing material

Figure 1(a) shows a compact sound absorber made of multiple parallel Helmholtz resonators. The global cylindrical cavity of the material is partitioned into twelve sub-cavities where four are located in the center and one extended neck is connected to each sub-cavity. The necks are illustrated in colors and the walls are considered rigid.

Each sub-cavity with the associated neck represents a Helmholtz resonator, so the material design in Fig. 1 (a) consists of 12 parallel Helmholtz resonators. A normal incidence plane wave with pressure amplitude of 1 Pa is applied on the inlet plane with plane wave radiation condition in Figs. 1 (b) and (c) and the surface average acoustic pressures at planes M_1 and M_2 positions are determined numerically. The two-microphone transfer function method is used to evaluate the sound absorption coefficient and the surface impedance of the proposed metamaterial. The distance between the planes M_1 and M_2 in Fig. 1 is set to 30 mm and the distance from the plane M_2 to the surface of the metamaterial is 150 mm.

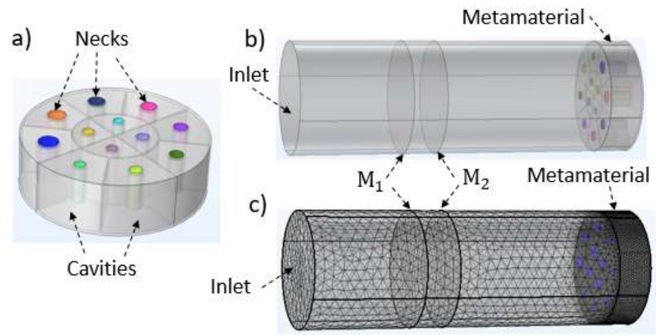

Figure 1: Numerical models (a) metamaterial based on twelve parallel resonators (b) geometry (c) mesh.

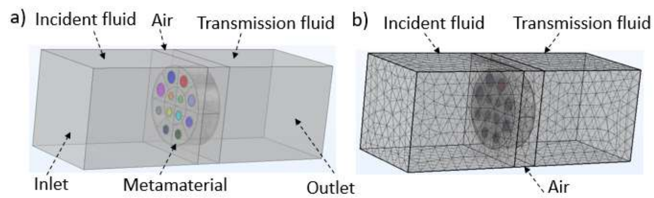

Figure 2: Periodic unit cell for TL evaluation (a) geometry of the PUC (b) mesh of the PUC.

The sound absorber in Fig. 1 (a) is periodically arranged in air and the periodic unit cell (PUC) used to determine numerically the TL is presented in Fig. 2. The geometry and the mesh of the PUC shown in Fig. 2(a) and (b) consist of an incident and transmission fluids that are connected to the metamaterial.

All the domains in Figs. 1 and 2 are modelled using the pressure acoustics module of Comsol Multiphysics while the air within each neck is characterized using the thermo-viscous acoustic module to account for the viscous and thermal dissipation effects. In Fig. 2, the lateral dimension of the PUC are 120 mm x 120 mm and the length of the incident and transmission fluids is set to 150 mm. The thickness of the metamaterial in Figs. 1 and 2 is set to 30 mm with outer diameter of 100 mm and the inner cavity diameter is 46 mm. In Fig. 2, a plane wave radiation condition is applied at the inlet and outlet planes and the periodic boundary condition is applied on each pair of parallel plans. The transmission loss is obtained by the following relation [4]

$$TL = 10\log_{10}(W_{\text{in}} \, / \, W_{\text{out}}) \qquad (1)$$

where W_{in} and W_{out} are respectively the incoming power at the inlet plane and the outgoing power at the outlet plane.

Figure 3 shows the normal incidence sound absorption coefficient of the metamaterial presented in Fig. 1 (a) where the length of each neck is set to 20 mm. The corresponding

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normalized surface impedance is presented in Fig. 4. The radii of the four necks in the center are 3 mm, 3.5 mm, 2.8 mm and 3.2 mm while the radii of the other eight necks are 2 mm, 2.5 mm, 2.75 mm, 3 mm, 3.5 mm, 1.75 mm, 1.5 mm and 2.25 mm. The mesh consists of 201354 domain elements with 40658 boundary elements and the number of degrees of freedom is 562157.

In Fig. 3, the sound absorption coefficient presents twelve resonant frequencies that are 202 Hz, 226 Hz, 264 Hz, 286 Hz, 320 Hz, 360 Hz, 390 Hz, 464 Hz, 522 Hz, 564 Hz, 606 Hz and 664 Hz where the absorption peak values are respectively 0.45, 0.67, 0.9, 0.96, 0.96, 0.94, 0.93, 0.76, 0.96, 0.9, 0.87 and 0.78. At these resonant frequencies, the normalized acoustic resistance and reactance in Fig. 4 are close to one and zero respectively. Note that the sound absorption in Fig. 3 drops at frequencies where the resistance in Fig. 4 is higher.

Figure 5 shows the TL of the metamaterial where the radii of the four necks in the center are set to 5.5 mm, 5 mm, 4 mm and 4.5 mm with a length of 20 mm. The radii of the other necks are 4 mm, 4.5 mm, 5 mm, 5.5 mm, 6 mm, 6.5 mm, 7 mm, 7.5 mm with a length that is varied from 20 mm to 25 mm with a step of 1 mm. In Fig. 5, one observes twelve resonant frequencies that are 528 Hz, 582 Hz, 636 Hz, 686 Hz, 714 Hz, 736 Hz, 786 Hz, 808 Hz, 850 Hz, 900 Hz, 914 Hz and 998 Hz where the TL resonant peak values are 8 dB, 11 dB, 12.7 dB, 16 dB, 13 dB, 14 dB, 16.7 dB, 13.5 dB, 17 dB, 18 dB, 18.6 dB and 16.8 dB respectively.

The sound absorption coefficient and the TL of the proposed metamaterial show twelve resonant peaks in Figs. 3 and 5. The resonant frequencies can be tuned to specific frequencies by adjusting the parameters of the necks and the volumes of the sub-cavities.

3 Conclusion

A compact sound absorbing material made of twelve parallel Helmholtz resonators was proposed and its acoustic performance was studied using finite element method. The sound absorption coefficient and the transmission loss showed twelve resonant peaks at different frequencies where the surface impedance was close to the air impedance. The proposed sound absorber can help in different engineering applications for noise reduction at twelve different frequencies simultaneously.

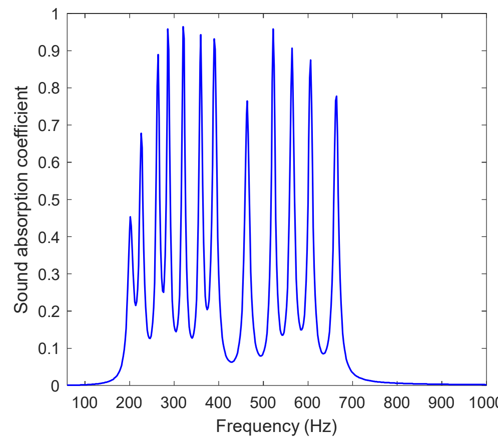

Figure 3: Sound absorption coefficient of the metamaterial.

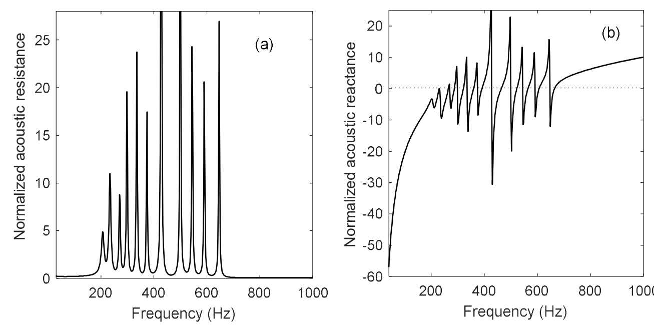

Figure 4: Normalized surface impedance of the metamaterial (a) acoustic resistance (b) acoustic reactance.

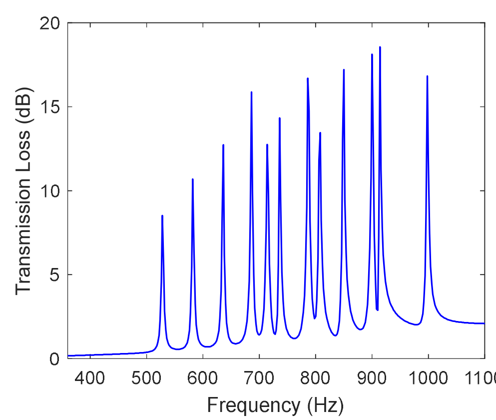

Figure 5: Transmission loss of the metamaterial.

Acknowledgments

The authors would like to acknowledge the National Research Council Integrated Aerial Mobility Program for the financial support.

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