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A slim subwavelength absorber based on coupled microslits

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ABSTRACT

Designing a perfect low frequency acoustic absorber with subwavelength thickness is always a challenge. Various designs including micro-perforated panels, acoustic metamaterials and metasurfaces have attracted significant interests in recent years. The present paper reports a slim subwavelength absorber based on a regular array of coupled microslits to achieve low frequency absorption. The absorption mechanism by the coupled microslits is investigated using acoustic impedance together with the reflection coefficient in the complex frequency plane. The geometry of the coupled microslits is optimized using a differential evolution algorithm to enhance the absorption in the frequency band from 300 Hz to 800 Hz under different condition. The ratio of the absorber thickness to wavelength at the lowest absorption peak is less than 3.4%, which confirms its operation in deep subwavelength regime. Finally, wide band absorption using coupled microslits is demonstrated by the experiment and simulation.

1. Introduction

Conventional acoustic absorbers for airborne sound, such as various porous, fibrous materials and wedges, usually comprise structures with a thickness comparable to working wavelength, which results in thick layers if low frequency range is to be targeted [1]. Micro-perforated panel (MPP) can be a good candidate as a subwavelength absorber, however, a backing cavity is needed and the ratio of the whole absorber thickness to working wavelength is usually larger than 5% [2–5].

Over the past decade there has been a significant amount of research efforts devoted to achieving sound absorption by thin layers using acoustic metamaterials/metasurfaces (MetaXs) with subwavelength resonant structure. MetaXs are man-made macroscopic composites designed to have properties not found in nature [6,7]. Generally, to achieve total sound absorption by tailored MetaXs, two basic problems need to be solved: 1) Matching the material impedance to background medium while preventing back-reflected wave, 2) reducing the geometric dimension of the structure while increasing the density of interaction at low frequency and providing sufficient energy losses to guarantee dissipation. Various membrane resonators [8–10], Helmholtz resonators [11–16] or Fabry-Perot (FP) resonant channels [17–21] especially with coiled up cavity [14,15,17–23], and foams with various resonant scatterers, i.e., metaporous materials [24–27] are used to realize the total absorption at low frequency range. The former two can gain deep subwavelength acoustic absorption, however, the bandwidth is narrow. The metaporous material can form a wide band absorption, however, the ratio of the overall thickness to the lowest resonant frequency wavelength is relative large, for example, about 10% in Fig. 5 of Ref. [27].

Motivated by the two coupled slits, i.e., the two dimensional (2D) Helmholtz resonators [11], the present paper aims at designing a very thin absorber using coupled microslits to acquire low frequency sound absorption. The structural parameters of the coupled microslits are optimized by a differential evolution (DE) algorithm [28,29] to enhance the absorption in the frequency range from 300 Hz to 800 Hz. The acoustic coupling between two microslits and a wide band absorption mechanism are investigated using the impedance and the reflection coefficient in the complex frequency plane [30–32]. Finally, both the experiment and simulation are used to verify the low frequency and broadband absorption by the coupled microslits.

2. Theoretical method and simulation

2.1. Theoretical method

One basic unit of the absorber with periodic infinite uniform slits is shown in Fig. 1. The slits are arranged along the x direction with period$a$, i.e., lattice constant along the x direction. The whole structure thickness is H. The structure extends infinitely along the z-axis. The width and depth of the slit (panel thickness) are $d$and h

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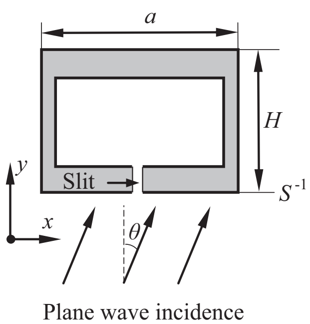

Fig. 1. Structure of one unit cell of the absorber with periodic uniform slits.

respectively, the cavity behind the slit has width w and height/length l. A plane longitudinal harmonic wave is incident from the air half space below with incident angle θ .

A harmonic incident plane wave of angular frequency ω with time dependence $e^{-j\omega t}$ can be described by the following equation

$$p_{in}(x,y) = p_0 e^{j(k_x x + k_y y)}, \tag{1}$$

where $k_x = k\sin\theta$, $k_y = k\cos\theta$ are the wave numbers in x and y directions respectively, the wave number $k$is given by $k = \omega/c_0$, c_0 is the sound velocity in air. j is the imaginary unit. p_0 denotes the amplitude of the incident acoustic pressure. The equation of air motion in a slit is [33,34]

$$j\omega\rho_0 u - \eta\frac{\partial^2 u}{\partial x^2} = -\frac{\partial p}{\partial y}, \tag{2}$$

with u being the particle velocity along y axis, ρ_0 the air density, η the dynamic viscosity. The air is assumed incompressible. For a slit depth much less than wavelength, $\partial p/\partial y$ can be substituted by $\Delta p/h$, Δp is the sound pressure drop between the ends of the slit. The above equation can be solved for u. The ratio of Δp to the average value of u over the cross sectional area of the slit gives the specific acoustic impedance

$$Z_s = \frac{\Delta p}{\bar{u}} = j\omega\rho_0 h\left[1 - \frac{1}{K\sqrt{j}}\tanh(K\sqrt{j})\right]^{-1}, \tag{3}$$

here $K = d/2\sqrt{\omega\rho_0/\eta}$. It is now necessary to add the end corrections of the slit. The resistance due to air flow friction on the surface of the panel is $\sqrt{2\omega\rho_0\eta}/2$, the mass reactance due to the piston sound radiation at both ends of the slit is $dF(e)/2$. $F(e)$ denotes of the complete elliptic integral where an ellipse with suitable shape approximating the slit [34]. For normal incidence of sound wave on the panel, the overall relative acoustic impedance of the panel is [34]

$$Z_o = \left(j\omega\rho_0 h\left[1 - \frac{1}{K\sqrt{j}}\tanh(K\sqrt{j})\right]^{-1} + \frac{\sqrt{2\omega\rho_0\eta}}{2} + j\omega\rho_0\frac{dF(e)}{2}\right)/\sigma\rho_0 c_0, \tag{4}$$

where σ is the filling fraction of the slit embedded in panel, $\sigma = d/a$. After some simplifications the real part of the relative special impedance of the panel is

$$x_h = \frac{12\eta h}{\sigma\rho_0 c_0 d^2}\left(\sqrt{1 + \frac{K^2}{18}} + \frac{\sqrt{2}}{12}K\frac{d}{h}\right), \tag{5}$$

and the imaginary part is

$$y_h = \frac{\omega h}{\sigma c_0}\left(1 + 1/\sqrt{5^2 + 2K^2} + \frac{dF(e)}{2h}\right). \tag{6}$$

The rigid back cavity of the unit can be regarded as an addition to the special reactance normalized to the impedance of air

$$y_c = -a/w\cot(\omega l/c_0). \tag{7}$$

The total relative surface impedance of the absorber is

$$Z = Z_o + jy_c \text{ or } Z = x_h + j(y_h + y_c), \tag{8}$$

where x_h and $y_h + y_c$ refer to the resistive and reactive parts of the acoustic impedance respectively. For plane wave incidence, the sound absorption coefficient is [2]

$$\alpha = \frac{4x_h\cos\theta}{(1 + x_h\cos\theta)^2 + (y_h\cos\theta - \cot(\omega l\cos\theta/c_0))^2}. \tag{9}$$

Further, for the two coupled slits arranged periodically along the$x$direction, as showed in Fig. 2, the total relative surface impedance Z of the absorber is [35]

$$\frac{a}{Z} = \sum_{i=1}^{2}\frac{a_i}{Z_i}, \tag{10}$$

where the subscript $i = 1, 2$, a_i and Z_i denote the outside width and the relative impedance of the respective slit, $a = \sum_{i=1}^{2} a_i$.

2.2. Finite element simulation

To verify the above theoretical predication, finite element (FE) model is built with commercial software COMSOL Multiphysics 5.1 with Acoustic-Thermoacoustic interaction module. The Floquet periodic boundary condition has been applied on both sides of the unit. The effects of the viscous friction and the heat transfer are included in the linearized compressible Navier-Stokes equation, the continuity equation, and the energy equation. In the domain below the s^-surface (in Fig. 1) there is a virtual impedance tube with width a fitted to the absorber unit, in which the incident plane wave is exerted, and a Perfectly Match layer (PML) is added at the bottom of the virtual impedance. All the walls of the slit and cavity are assumed to be acoustically rigid. On s^- surface, the reflect pressure is

$$p_r = \sum_n A_n e^{i(k_{x,n}x + k_{y,n}y)}, \tag{11}$$

here $k_{x,n} = k\sin\theta + 2n\pi/a$, $k_{y,n} = \sqrt{k^2 - k_{x,n}^2}$, and

$$A_n = \frac{1}{a}\int_0^a p_r(x, y_{s^-})e^{-ik_{x,n}}dx. \tag{12}$$

The energy reflection coefficient, i.e., the ratio of the reflection energy to the incident energy [36]

$$r = \sum_{n=-N''}^{n=N'} |A_n|^2 k_{y,n}/\left(|p_0|^2 k_y\right) \tag{13}$$

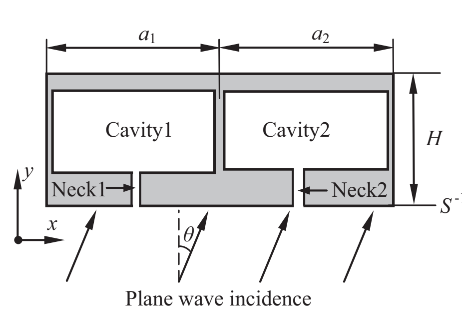

Fig. 2. Structure of one unit cell of the absorber with periodic coupled slits.