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ABSTRACT

We propose a hybrid acoustic metamaterial as a super absorber for a relatively broadband low-frequency sound based on a simple construction with deep-subwavelength thickness (5 cm). The hybrid metamaterial absorber is carefully designed and constructed based on a micro-perforated panel (MPP) and coiled-up Fabry–Pérot channels. It is demonstrated analytically, numerically, and experimentally that over 99% of acoustic absorption could be achieved at a resonance frequency (<500 Hz) with the working wavelength about 30 times larger than its total thickness. It is revealed that the superior absorption is mainly caused by the friction losses of acoustic wave energy in the MPP. The frequency of the absorption peak could be tuned by adjusting the geometry parameters of the MPP and the channel folding numbers. The relative absorption bandwidth could also be tuned flexibly (up to 82%) with a fixed deep-subwavelength thickness (5 cm). The absorber has wide potential applications in noise control engineering due to its deep-subwavelength thickness, relatively broad bandwidth, and easy fabrication.

The attenuation of low-frequency sound (<500 Hz) by absorbers with deep-subwavelength thickness (≤5 cm) is of great interest in noise control engineering. However, it has been a challenging task because of the strong penetrability of low-frequency sound and weak intrinsic dissipation of common materials. Conventional sound absorbing materials (SAMs), such as porous materials, have been demonstrated to be effective for high-frequency sound absorption (>1000 Hz), but have shortcomings at low frequencies if limited thickness is constrained. In recent years, the concept of acoustic metamaterials (AMMs) provides a new idea for the design of low-frequency sound absorbers. A host of subwavelength sound-absorbing materials or devices have been developed based on resonant structures such as decorated membrane resonators (DMRs), well-constructed Helmholtz resonators (HRs) and Fabry–Pérot (FP) channels. The conventional micro-perforated panel (MPP) with a backing cavity is also a good candidate for low-frequency sound absorbers. However, large thickness (>5 cm) of the backing cavity is usually required for low-frequency applications, especially for relatively broadband frequencies below 500 Hz.

An effective approach for reducing the dimensions of the sound absorber is introducing the concept of space coiling or labyrinthine structures. In 2014, Cai et al. proposed an ultrathin sound absorbing panel (<λ/50) by bending and coiling up quarter-wavelength sound damping tubes into 2D coplanar ones. In 2016, Li and Assouar developed hybrid Helmholtz resonators (HRs) with coiled-up channels, which can generate deep subwavelength absorption with a thickness of λ/223 of the working wavelength. In 2017, Chen et al. coiled up two axially coupled tubes in series into a layer perpendicular to incident waves for reducing the sample thickness. The bandwidths of absorption peaks of these acoustic metamaterials, however, are usually narrow. By directly coiling up dead-end FP channels, Labyrinthine AMMs (LAMMs) are considered as an emerging class of low-frequency sound absorbers. A relatively large absorption bandwidth has been achieved by carefully designing many parallel channels stuffed with additional absorbent materials. Very recently, several other subwavelength hybrid acoustic absorbers have also been proposed, and they generally utilize coiled-up channels with a HR absorber to achieve low-frequency resonance.

In this letter, we carefully design and demonstrate a type of hybrid acoustic metamaterial absorber constructed based on MPPs (with perforation diameter < 1 mm) and coiled-up FP channels. Built
upon the classical micro-perforated panel absorber (MPPA), the idea of introducing coiled-up FP channels without adding to the total thickness greatly enhances the low-frequency absorption performance, and simultaneously, exhibits a relatively broad absorption bandwidth without employing additional absorbent materials.

As shown in Figs. 1(a)–1(c), the hybrid acoustic metamaterial consists of MPPs and coiled-up FP channels. The incident acoustic wave enters the channels perpendicular to the MPPs along the Z-axis. To examine the absorption characteristics, firstly, an approximate analytical two-dimensional (2D) model of one unit cell is developed. The propagation path of the acoustic wave inside the coiled-up channel is sketched by the blue dashed arrows shown in Fig. 1(d). The periodic unit cell has a dimension of length \( L \), width \( W \) and height \( H \). The two internal partition panels have the same thickness \( t_1 \). In theoretical analysis, the coiled-up FP channels can be simplified as several straight coaxially connected sub-channels in series. The effective propagation length inside the \( i \)th sub-channel is denoted \( L_{\text{eff},i} \) (\( i = 1, 2, 3 \ldots \)).

In general, the absorption coefficient for an acoustic absorbing system with a rigidly backed panel can be calculated based on the impedance theory as

\[
\alpha = 1 - \left| \left( Z_r/Z_0 - 1 \right)/\left( Z_r/Z_0 + 1 \right) \right|^2, \tag{1}
\]

in which \( Z_r \) refers to the surface impedance of the acoustic absorber. \( Z_0 = \rho_0 c_0 \) denotes the characteristic impedance of air with \( \rho_0 \) and \( c_0 \) referring to the mass density and the sound speed of air, respectively. The surface impedance of the proposed absorber can be calculated in a general form as:

\[
Z_i = Z_{iM} + \xi \cdot Z_{iC1}, \tag{2}
\]

where \( Z_M \) and \( Z_{C1} \) (see supplementary material for more detailed derivation) denote the surface impedance of the MPP and the first channel at the entrance, respectively. The quantity \( \xi = S_i/W \) is defined as the area modification factor with \( S_i = W \times L \) and \( S_i = W_0 \times L_0 \) referring to the cross-sectional areas of the unit cell and channels, respectively.

For numerical verification purpose, FEM acoustic models of hybrid absorbers are also built using COMSOL Multiphysics 5.3. The viscous-thermal losses are taken into account using the embedded Acoustic-Thermoacoustic interaction module. All the walls of the acoustic absorber are assumed as hard boundaries due to huge impedance mismatch between air and solid panels. Two metamaterial absorbers made of the same space coiled-up channels with two different MPPs are considered first. The geometric parameters of the absorber are given in the caption of Fig. 2. The absorption coefficients obtained based on the theoretical calculation (circles) and FEM models (lines) are compared. A high absorption peak is achieved at 350 Hz and 460 Hz with a total thickness of the absorber (5 cm) within a deep subwavelength scale. To further explore the low-frequency absorption mechanism, we plot the central cross-sectional intensity maps of the acoustic pressure and the velocity of acoustic particles with arrows at the resonance frequency (460 Hz) in Fig. 2(b). At the resonance frequency, a much larger velocity at the interface of the MPP and at the entrance of the first channel is observed. As a result, the incidence wave energy is mainly dissipated due to the larger friction between the acoustic wave and small perforations. Therefore, the low-frequency absorption mechanism of the acoustic absorber is the conversion of acoustic energy into heat energy at the resonance frequency. A series of hybrid absorbers are further constructed with the total thickness unchanged at 5 cm, but with different channel folding numbers \( n = 3, 4, 5 \). All other parameters are given in the caption of Fig. 3. To reveal the underlying physics of the total absorption of the presented absorbers, we investigate the reflection coefficient \( r \) in the complex plane using a graphic method. Generally, a reflection coefficient processes a complex conjugate zero and a pole in the lossless case. If the energy leakage is equal to the loss at the resonant frequency, zero will be located exactly on the real frequency axis. Figures 3(a)–3(c) illustrate the distributions of \( |\log_{10}|r|^2 \) in the complex frequency plane for folding numbers \( n = 3, 4 \) and 5, respectively. As is shown, all zeros fall nearly at the real frequency axis; thus, nearly perfect absorption is obtained at the resonance frequency. And, it can be observed that the absorption could be tuned by adjusting folding numbers \( n \). Moreover, the absorption bandwidth can also be illuminated.
by the leakage rate between the zero and the pole.\(^\text{14}\) Figs. 3(a)–3(c) show that a broader bandwidth and a larger leakage rate are obtained with decreasing folding number \(n\). Figures 3(d)–3(f) depict the absorption coefficient for different cases; in each case, the theoretical solution (circles) agrees well with the FEM results (lines). Near perfect absorption peaks occur at 330, 270, and 225 Hz [corresponding to zero points in Figs. 3(a)–3(c)] with relative bandwidths of 59.1%, 53.7% and 50.6% for folding numbers \(n = 3, 4\) and 5, respectively. Here, the relative bandwidth refers to the ratio of the full width at half maximum of the absorption coefficient to the resonance frequency. The absorption peak clearly shifts to a lower frequency with increasing folding number \(n\). The thickness is only 1/30 of the working wavelength at the resonance frequency (225 Hz).

We further plot the imaginary parts of relative impedance \(Z_s/Z_0\) of the acoustic absorbers in Figs. 3(d)–3(f) for different folding numbers \(n\). Total absorption could be realized when the impedances match well, meaning that the imaginary parts of relative impedance are equal to zero, while the real parts become unity. Thus, the sound energy could be totally dissipated by the resistance. The imaginary curve crosses zero at 330, 270 and 225 Hz for folding numbers \(n = 3, 4\) and 5, indicating the existing resonant state [corresponding to absorption peaks in Figs. 3(d)–3(f)]. The real parts of relative impedance are equal to 0.94, 1.01, and 1.27 for folding numbers \(n = 3, 4\), and 5, respectively. The real part of relative impedance is not strictly equal to one. As a result, the absorption coefficient is little less than 100%.

In order to elaborate study the absorbing performance of the hybrid absorber, we further investigate the trajectory of zeros of the reflection coefficient by changing the geometry parameters. The associated geometric parameters are given in the caption of Fig. 4. A near perfect absorption peak (99.96%) is obtained at 380 Hz for a particular case with a relative absorption bandwidth of 73.7%, as shown in Fig. 4(a). The theoretical solution (line) agrees well with the FEM results (circle). In regard with the concern on the absorption bandwidth of the proposed design, it is worth comparing the present absorber [i.e., the case illustrated in Fig. 4(a)] with one typical existing absorber reported recently by Liu et al.,\(^\text{30}\) which is designed with a comparable thickness \([H = 53 \text{ mm}, \text{see Fig. 2(g) in Ref. 30}\] using a single labyrinthine channel, and it works efficiently at a somewhat higher frequency (426 Hz). Note that the maximum absorption coefficient of the reference absorber is only about 65%, and the relative bandwidth of high absorption \((\alpha > 0.5)\) is only about 20%, being much narrower than our present absorber (73.7%). The performance of the reference

![FIG. 3.](image1.png) (a)–(c) Representation of the reflection coefficient in the complex frequency plane for folding numbers \(n = 3, 4\) and 5, respectively. Other geometric parameters are: \(d = 0.9\ \text{mm}, b_0 = 1\ \text{mm}, \rho = 0.018, L_1 = L_2 = L_3 = 12\ \text{mm}, H = 50\ \text{mm}\) and \(H_1 = H_2 = 12\ \text{mm}\). (d)–(f) Sound absorption coefficients of the absorber for various folding numbers \(n\). The solid line and circle show the theoretical prediction and the FEM results, respectively. The imaginary of relative impedance is also given. The arrows denote the absorption peak frequencies.

![FIG. 4.](image2.png) (a) Sound absorption coefficient of a particular case of absorber. The solid line and circle show the theoretical prediction and the FEM results, respectively. The imaginary and real parts of relative impedance are also presented. The associated geometric parameters are: \(d = 0.5 \text{ mm}, b_0 = 0.64\ \text{mm}, \rho = 0.018, L_1 = L_2 = L_3 = 12\ \text{mm}, H = 50\ \text{mm}\) and \(H_1 = H_2 = 12\ \text{mm}\); (b) \(\log_{10}|\rho|^2\) in the complex frequency planes for the optimized sample. Each line shows the trajectory of its zero by changing a single geometry parameter.
The color map in Fig. 4(b) corresponds to the above-mentioned particular case in which the critical coupling condition is fulfilled, i.e., the impedances match well such that the imaginary part of relative impedance is equal to zero, while the real part reaches unity, as observed in Fig. 4(a). The zero of the reflection coefficient is exactly located on the real frequency axis in Fig. 4(b). We also present in Fig. 4(b) that the trajectory of this zero can be seen that the zero point in the color map (resonance frequency) could also be easily adjusted by changing the geometric parameters (including the thickness \( t_0 \), the diameter \( d \), the permeation point \( p \) of the MPP, the width of the second channel \( L_2 \), and the height of the coiled-up channel \( H \)). There is a clear tendency that the zero point is shifted to a lower frequency when the thickness \( t_0 \) or height \( H \) of the coiled-up channel is increased. Also, note that near-perfect absorption can be obtained at the resonance frequency over a wide range of heights \( H \). The trajectory of permeation rate \( p \) shows that the resonance frequency increased with increasing permeation. In the case of diameter \( d \), a small diameter suggests that the acoustic resistance is overly increased, while a large diameter implies that the intrinsic losses are not enough, thus critical coupling in both cases cannot be fulfilled. In the case of \( L_2 \), if the width of the second channel decreases, the total permeation rate of the surface MPP will increase, and thus the trajectory of the zero is twisted.

In order to expand the relative absorption bandwidth, we further integrated two parallel unit cells with different parameters as one sample resonator as shown in Fig. 5(a). Unit 1 and unit 2 have the same cross-section of coiled-up channels (folding number \( n = 3 \)), but with different top MPPs. The permeation rate \( p \), the diameter \( d \), the thickness \( t_0 \) of MPP resonator unit 1 are 0.048, 0.4 mm and 0.64 mm, respectively, while the corresponding parameters of MPP resonator unit 2 are 0.018, 0.9 mm and 0.64 mm, respectively. The other parameters of both samples are the same and are listed in the caption of Fig. 5. For verification purpose, we conducted experiments using the standard two-microphone method \(^{10} \) on a B&K 4206A impedance tube of square cross-section. In our experimental setup, we first integrated ten parallel unit cells as one sample resonator instead of one unit cell. The given parameters of ten unit cells in one sample are exactly the same, which generates a symmetric resonant mode. Two resonator units with different geometric parameters were fabricated using three-dimensional printing.

The measured and predicted results are shown in Fig. 5(b). The absorption peak of unit 1 and unit 2 are achieved at 350 Hz and 460 Hz with a total thickness of the absorber (5 cm) within a deep subwavelength scale. The slight but acceptable deviation between prediction and experiment can be observed, which may have resulted from the unavoidable machining error in preparing the MPPs having very small geometric features like small diameters \( d \) and small thicknesses \( t_0 \). Figure 5(c) illustrates the distributions of \( \log_{10}(r^2) \) in the complex frequency plane of the integrated two parallel units 1 and 2 (integrated sample), in which two zeros and poles are obviously observed. Their corresponding absorption peaks can be found in Fig. 5(d). The red lines in Fig. 5(d) show the experimental absorption spectrum of the hybrid absorber. There is some difference between the test and theoretical results of the integrated sample. This may be caused by the fact that the two resonance frequency peaks of the integrated units are designed relatively close. Thus, the two resonant units may exhibit some coupling and merging effects due to the influence of uncertainties in the fabricated sample. To reveal the underlying coupled mechanism, the absorption of each separate units can be referred in Fig. 5(b). Note that the maximum absorption coefficients of two unit cells are 95% and 96% at 355 Hz and 470 Hz, respectively, meaning that no perfect absorption is obtained in each separate unit cell in Fig. 5(b). The relative bandwidths of absorption of each unit cells are 54.9% and 64.9%. After integrating these two units, nearly total absorption can be obtained due to the influence of the coupling effects. The amplitudes of the maximum absorption coefficient exceeded over 99% in both simulated and experimental results. Moreover, the relative bandwidth of absorption is widened up to 82.2% as shown in Fig. 5(d).

In summary, we have proposed a type of hybrid acoustic metamaterial absorber based on microperforated panels and coiled-up Fabry–Pérot channels, which can efficiently absorb the incident acoustic wave energy at very low frequencies (<500 Hz) with a broad relative bandwidth of absorption. The high-efficiency and tunable absorption characteristics of the proposed absorber are examined analytically, and also verified by numerical simulations and experiments. We reveal that the absorption is mainly caused by the friction losses of acoustic waves in microperforated panels. The phenomenon is also explained by graphically analyzing the reflection coefficient in the complex plane. The relative absorption bandwidth is further broadened up to 82.2% by integrating two parallel absorber unit cells with different parameters. Due to the deep subwavelength thickness, relatively broad bandwidth, and easy fabrication, the proposed hybrid absorbers have wide potential applications in noise control engineering.
See supplementary material for the derivation of explicit formulations for determining $Z_M$ and $Z_L$ in Eq. (2).

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REFERENCES

Supplementary material to:

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As described in the manuscript, the surface impedance of the proposed hybrid absorber is given in a general form as:

\[ Z_s = Z_M + \xi \cdot Z^L_{C1}, \quad (S1) \]

where \( Z_M \) and \( Z^L_{C1} \) denotes surface impedance of micro-performed panel and the first channel at the entrance, respectively. The quantity \( \xi = S_o / S_1 \) is defined as the area modification factor with \( S_o = W \times L \) and \( S_1 = W_o \times L_o \) referring to the cross-sectional areas of the unit cell and channels, respectively. Based on the theory of Crandall,1 the surface impedance \( Z_M \) can be obtained as:

\[ Z_M = \frac{j\omega \rho_o \tau_0}{\bar{p}} \left[ 1 - \frac{2J_1(\gamma \sqrt{-j})}{(\gamma \sqrt{-j})J_0(\gamma \sqrt{-j})} \right]^{-1} + \frac{\sqrt{2\eta}}{\bar{p}d} + \frac{j0.85\omega \rho_o d}{\bar{p}}, \quad (S2) \]

in which \((\tau_0, d, \bar{p})\) denote the thickness of panel, the diameter and the total porosity of perforation of top panel, respectively. \( \bar{p} = p * S_p / S_{tot} \), \( S_p \) is the area of perforated parts of top panel and \( S_{tot} \) is total area of the top panel. \( \omega \) is the angular frequency. \( \eta \) refers to the dynamic viscosity coefficient of air. The quantity \( \gamma = d \sqrt{\rho_o \omega / 4\eta} \) denotes \( \sqrt{2}/2 \) times the ratio of the perforation diameter to the thickness of the viscous boundary layer. \( J_1 \) and \( J_0 \) is the Bessel functions of the first kind of the first

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and zeroth order. For each channel, the surface impedance \( Z_i^L \) at the top entrance can be obtained:

\[
Z_i^L = Z_i^{c} + jZ_i^{c} \cot \left( k_i l_i \right) \quad \left( \text{if}, \quad i = 1, 2, 3\ldots N-1 \right),
\]

\[
Z_N^L = -j \frac{\rho_i^{eq}}{C_i^{eq}} \cot \left( \omega \sqrt{\rho_i^{eq} C_i^{eq}} l_i \right) \quad \left( \text{if}, \quad i = N \right),
\]

where \( Z_i^{c} = Z_i^{L} / \zeta_i \) and \( \zeta_i = S_{i+1} / S_i \), with \( S_i \) referring to the cross-sectional area of channel \( C_i \). The last channel \( (i = N) \) backed with a rigid wall can be considered as a FP channel. The quantities \( \rho_i^{eq} \) and \( C_i^{eq} \) in Eq.\((S3)\) are defined as complex density and compressibility functions,\(^2\) respectively, which can be obtained by:

\[
\rho_i^{eq} = \rho_0 \left\{ \frac{v L_i^2 W_0^2}{4i\omega} \sum_{k=0}^{\infty} \sum_{n=0}^{\infty} \frac{\alpha_k^2 \beta_n^2 \left( \beta_n^2 + \frac{i\omega}{v} \right)^{\alpha_k^2 \beta_n^2 + \frac{i\omega}{v}}}{\alpha_k^2 \beta_n^2 + \frac{i\omega}{v}} \right\}^{-1},
\]

\[
C_i^{eq} = \frac{1}{P_0} \left\{ 1 - \frac{4i\omega(\gamma - 1)}{v' L_i^2 W_0^2} \sum_{k=0}^{\infty} \sum_{n=0}^{\infty} \frac{\alpha_k^2 \beta_n^2 \left( \beta_n^2 + \frac{i\omega \gamma}{v'} \right)^{\alpha_k^2 \beta_n^2 + \frac{i\omega \gamma}{v'}}}{\alpha_k^2 \beta_n^2 + \frac{i\omega \gamma}{v'}} \right\}^{-1},
\]

in which, the quantities \( v = \mu / \rho_0 \) and its derivatives \( v' = \kappa / \rho_0 C_v \) can be calculated with \( \mu \), \( \kappa \), and \( C_v \) referring to the air viscosity, thermal conductivity, and specific heat at constant volume, respectively. The quantities \( \alpha_k = (k + 1/2) \pi / W_0 \) and \( \beta_n = (k + 1/2) \pi / L_i \) are constants, \( P_0 \) and \( \gamma \) refer to the air pressure and the ratio of specific heat, respectively.

**References**