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Letter

Design of multi-layered porous fibrous metals for optimal sound absorption in the low frequency range

Wenjiong Chen [a], Shutian Liu [a,*], Liyong Tong [b], Sheng Li [a]

[a] State Key Laboratory of Structural Analysis for Industrial Equipment, Dalian University of Technology, Dalian 116023, China

[b] School of Aerospace, Mechanical and Mechatronic Engineering, The University of Sydney, NSW 2006, Australia

HIGHLIGHTS

- A method for enhancing sound absorption coefficient of fibrous metals is presented.
- A fibrous layout with given porosity of multi-layered fibrous metals is suggested.
- Appropriate surface porosity balances dissipation and reflection of acoustic energy.

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ABSTRACT

We present a design method for calculating and optimizing sound absorption coefficient of multi-layered porous fibrous metals (PFM) in the low frequency range. PFM is simplified as an equivalent idealized sheet with all metallic fibers aligned in one direction and distributed in periodic hexagonal patterns. We use a phenomenological model in the literature to investigate the effects of pore geometrical parameters (fiber diameter and gap) on sound absorption performance. The sound absorption coefficient of multi-layered PFMs is calculated using impedance translation theorem. To demonstrate the validity of the present model, we compare the predicted results with the experimental data. With the average sound absorption (low frequency range) as the objective function and the fiber gaps as the design variables, an optimization method for multi-layered fibrous metals is proposed. A new fibrous layout with given porosity of multi-layered fibrous metals is suggested to achieve optimal low frequency sound absorption. The sound absorption coefficient of the optimal multi-layered fibrous metal is higher than the single-layered fibrous metal, and a significant effect of the fibrous material on sound absorption is found due to the surface porosity of the multi-layered fibrous.

As a new type of sound-absorbing materials, porous fibrous metal (PFM), e.g. stainless steel and FeCrAl, has been found to be effective in reducing noise. Compared with the conventional nonmetallic porous fibrous material, PFM becomes attractive due to its mechanical properties, e.g. large surface area, low density, excellent permeability and higher mechanical strength. Hence under extreme circumstances (such as acoustical liner of turbofan engine inlet) PFM may be suitable for applications [1,2]. Therefore recently the study of the sound absorption performance of PFM is of considerable interest.

PFM often has high porosity, as metal fibers cross over each other and there exist a lot of small air passages in the material. Due to these specific pore geometries, energy dissipation in the forms of damping and thermal loss leads to superior sound absorption performance of the PFM. Similar to most sound porous materials, uniform (homogeneous) PFM has excellent sound absorption properties in middle–high frequency range, however, in the low frequency range, the capability of sound absorption is relatively poor. The most common way to improve the performance in low frequencies is to increase the thickness of material [3]. However, this approach may limit the development of PFM to control noise in microelectronics and precision instruments. An alternative way is to optimize the pore geometric parameters to influence the sound propagation in the fibrous material. It has also been found that the gradient porosity can improve PFM's sound absorption performance [1,3–5]. An analysis and design model is in need to be established for designing optimal pore geometric parameter distribution.

* Corresponding author. Tel.: +86 411 84706149; fax: +86 411 84706149.
E-mail address: stliu@dlut.edu.cn (S. Liu).

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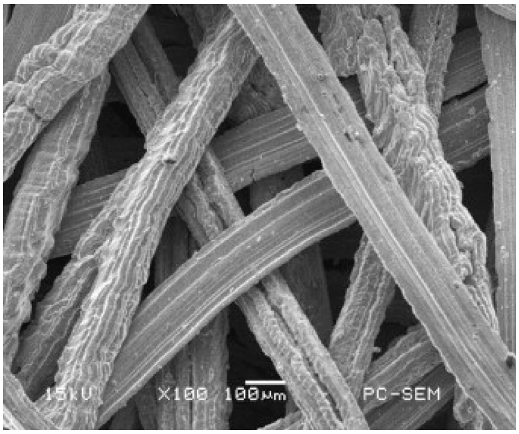

Fig. 1. A typical structural configuration of PFM [12].

Many researchers have studied the acoustic model for predicting the sound absorption properties of uniform porous materials. These models generally fall into two types: empirical and phenomenological models. Bazley and Delany [6] have proposed one of the most representative empirical model. This model can be used to study the single parameter's influence on sound absorption properties, such as state air-flow resistivity, and may not be directly used for studying microscopic structure of porous material. Apart from the empirical model, a number of phenomenological models have also been established, for example, the Biot model [7], Johnson–Champoux–Allard (JCA) model [8,9], Wilson model [10], Lafarge model [11], etc. Accuracy is improved by these models through introducing extra macroscopic acoustic parameters, which are usually obtained through solving the viscous boundary value problems of the pore structure. Relationship of material's sound absorption properties and the pore structure can be established using the JCA model, since it is accurate in both high and low frequencies. Therefore, the JCA model is employed to determine the sound absorption characteristics of PFM in this study.

In this paper, an optimization method of designing multi-layered PFMs is also proposed for maximizing low frequency sound absorption. We use the JCA model to investigate the effects of the pore structural parameters (fiber diameter D and gap w) on sound absorption coefficient. The impedance translation theorem is also introduced to calculate the sound absorption coefficient of multi-layered PFMs. To validate the present analytical model and design method, we compare the obtained results of computation to experimental data available in literature. With the average sound absorption in low frequencies as the objective function and the fiber gaps as the design variables, an optimization method of multi-layered PFMs is established. By using the proposed optimization method, we identified an optimal fiber distribution pattern for optimal sound absorption performance in the low frequency range.

Figure 1 shows a typical structural configuration of PFM [12]. To simplify the calculation, PFM is idealized as a parallel fiber array with repetitive hexagonal distribution patterns of individual fiber having circular cross-section (Fig. 2(a)). Therefore, the parameters of pore structure are only the fiber diameter D and fiber gap w. We perform numerical computations on the scale of the half unit cell (as shown in Fig. 2(b)). The porosity ϕ of the PFM can be conveniently calculated by

$$\phi = 1 - 4\sqrt{3}\pi\left(\frac{D}{2}\right)^2 \Big/ [9(w+D)^2]. \tag{1}$$

In PFM, fibers' density and stiffness are much higher than those of the air in the fibrous material. Therefore, acoustic propagation and dissipation through a rigid frame of porous media can be macroscopically described as a layer of equivalent fluid having the bulk modulus K_{eq} (thermal interaction dependent) and frequency-dependent effective density ρ_{eq} (which is mainly dependent on the viscous interaction between the fluid and frame) [13,14]. In this study, the JCA model [8,9] is employed to calculate the effective density and bulk modulus as

$$\rho_{\mathrm{eq}}(\omega) = \frac{\rho_0\alpha_\infty}{\phi}\left[1 - \mathrm{j}\frac{\sigma\phi}{\omega\rho_0\alpha_\infty}\sqrt{1 + \mathrm{j}\omega\rho_0\eta\left(\frac{2\alpha_\infty}{\sigma\phi\Lambda}\right)^2}\right], \tag{2}$$

$$\frac{1}{K_{\mathrm{eq}}(\omega)} = \frac{\phi}{\gamma P_0}\left\{\gamma - (\gamma-1)\left[1 - \mathrm{j}\frac{8\eta}{\omega\rho_0 Pr\Lambda'^2} \times \sqrt{1 + \frac{\mathrm{j}\omega\rho_0 Pr}{\eta}\left(\frac{\Lambda'}{4}\right)^2}\right]^{-1}\right\}, \tag{3}$$

where $\mathrm{j} = \sqrt{-1}\rho_0$ and α_∞ denote the fluid density and tortuosity, ϕ the open porosity, σ the static airflow resistivity, ω the angular frequency, η the air viscosity, Λ and Λ' represent the viscous and thermal characteristic lengths respectively, γ the specific heat ratio, P_0 the atmospheric pressure, and Pr is the Prandtl number. The JCA model consists of five macroscopic acoustic parameters: $\phi, \sigma, \alpha_\infty,\ \Lambda$ and Λ'.

These macroscopic acoustic parameters are determined by the velocity field and property of fluid in fibrous material, which can be defined as

$$\sigma = \frac{\eta}{\phi\langle w_x\rangle}, \tag{4}$$

$$\alpha_\infty = \langle E_x^2\rangle / \langle E_x\rangle^2, \tag{5}$$

$$\Lambda = \frac{\int_{\Omega_f}|E|^2\,\mathrm{d}V}{\int_{\partial\Omega}|E|^2\,\mathrm{d}S}, \qquad \Lambda' = 2\frac{\int_{\Omega_f}\mathrm{d}V}{\int_{\partial\Omega}\mathrm{d}S}, \tag{6}$$

where the following symbol designates a fluid-phase average

$$\langle * \rangle = \frac{1}{|\Omega_f|}\int_{\Omega_f}(*)\mathrm{d}V \tag{7}$$

and the subscript "x" denote the x component of $*$.

The viscous boundary value problems were solved using finite element analysis at the micro-structure scale under both asymptotic low ($f \to 0$) and high ($f \to \infty$) frequencies, to deduce the macroscopic parameters in the equivalent fluid JCA model [15].

Using the solution of steady state Navier–Stokes (NS) problem, we can deduce the low-frequency limit ($f \to 0$) leading parameter (i.e. the static airflow resistivity σ). The NS problem is defined as

$$\begin{aligned} &\Delta\boldsymbol{w} = \nabla\pi - \boldsymbol{e}, \quad \text{in } \Omega_f, \\ &\nabla\cdot\boldsymbol{w} = 0, \quad \text{in } \Omega_f, \\ &\boldsymbol{w} = \boldsymbol{0}, \quad \text{on } \partial\Omega. \end{aligned} \tag{8}$$

Here, Δ and ∇ represent the local 2-d del and Laplacian differential operators, $\boldsymbol{e}$ is the unit vector, and Ω_f and $\partial\Omega$ denote fluid domain and fluid surface. In addition, $\boldsymbol{w}$ and π represent a static velocity field and associated scalar pressure. As shown in Fig. 3(a), we can numerically (including via the finite element method) solve the velocity field by imposing a no-slip boundary condition at the fluid–solid interface and placing a periodic condition on π and $\boldsymbol{w}$ at the inlet/outlet surfaces. Once we get the solution for a given micro-structure, we can obtain the macroscopic static airflow resistivity by Eq. (4).

Comparing to the pore size, the boundary layer is small under extreme high frequency ($f \to \infty$), so we can ignore its viscous