Principles of Sound Absorbers

Xiaojun Qiu

Abstract This chapter introduces the principles of sound absorbers and the factors that affect acoustic absorption. The basic wave phenomena related to sound propagation such as acoustic reflection, absorption and scattering are introduced first and then the corresponding parameters such as transmission loss, absorption coefficient, scattering coefficient and flow resistance are explained. Sound absorbers can be made from porous materials or resonant structures, and the main mechanisms for sound absorption are acoustic impedance matching on the absorbers' boundary and acoustic energy dissipation inside the absorbers. Porous absorbers are materials where sound propagation occurs in a network of interconnected pores so that viscous and thermal effects cause acoustic energy to be dissipated. Resonant absorbers have two common forms: membrane/panel absorbers and Helmholtz absorbers. The main difference between porous absorbers and resonant absorbers is that the former is effective for broadband from mid to high frequency while the latter is usually only effective in a narrow tunable low frequency band. The applications of sound absorbers are focused on the discussion of reverberation time control and sound pressure level control in rooms. Finally, specific discussions are given to acoustic textiles and its recent developments. The contents of this chapter are focused on the fundamental aspects of sound absorption and absorbers to serve as an introduction to students and designers with basic knowledge in mathematics and physics.

Keywords Sound reflection · Sound absorption · Porous absorbers · Resonant absorbers · Reverberation time · Sound pressure level

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1 Principles of Sound Absorbers

This chapter introduces the principles of sound absorbers and the factors that affect acoustic absorption. The basic wave phenomena related to sound propagation such as acoustic reflection, absorption and scattering are introduced first and then the corresponding parameters such as transmission loss, absorption coefficient, scattering coefficient and flow resistance are explained. The mechanisms of porous absorbers and resonance absorbers are discussed next as well as designing factors that affect acoustic absorption. The applications of sound absorbers are focused on the discussion of reverberation time control and sound pressure level control in rooms. Finally, specific discussions are given to acoustic textiles and it recent developments. The contents are focused more on the fundamental aspects of sound absorption and absorbers to serve as an introduction to students and designers with basic knowledge in mathematics and physics.

The approaches that are commonly used for sound control can be classified into the following three categories:

- control the sound radiation and generation from sound sources
- control the propagation and transmission of the generated sound
- control the sound pressure received by listeners.

The applications of the above approaches depend on the specific problems to be solved under specific situations. For example, if measures can be taken around the sound sources, then the mechanisms of the sound generation or their radiation pattern can be modified to reduce the sound power generated by the sound sources. This is the preferred method and should be applied first whenever it is feasible. If the sound is already generated and radiated into spaces, different measures can be implemented depending on the properties of environments. In free field, sound barriers can be used to reflect the sound or diffract the sound so that the sound received at some locations is reduced. In enclosures, in addition to sound barriers, sound absorption materials and structures can be applied in the space or on the boundaries to absorb sound and to reduce the reflection of the sound energy. With the sound absorption treatments in enclosures, the reverberant sound pressure as well as the reverberation time can be reduced. If the sound pressure is still too high after the treatments on sound sources and propagation paths, personal protection measures can be taken, which can be enclosures and ear mufflers.

Two main challenges in the applications of the above-mentioned passive sound control approaches are: low frequency sound control and ventilation requirements. Active noise control is a method of reducing existing noise by the introduction of controllable secondary sources to affect generation, radiation, transmission and reception of the original primary noise source. It can provide better solutions to low frequency noise problems than the current passive noise control methods when there are weight and volume constraints. It also provides an alternative noise control solution to applications where current passive noise control methods cannot be applied. The fundamental theories and methods of active noise control have become well established over the last 30 years; however, successful industry and civil applications of the technology are still limited in some specific cases such as headsets and earplugs, propeller aircrafts and cars [1].

1.1 Sound Propagation, Reflection and Absorption

Sound is a longitudinal mechanical wave, where the displacement of the medium at each point is normal to the local wavefront surface when the disturbance is travelling in a medium [2]. Sound speed is different in different medium, and that in air under normal atmospheric pressure at 20 °C is about 343 m/s. There is an energy loss when sound propagates in the medium, for example, in textiles; however, the attenuation of sound caused by air can usually be neglected in the low frequency range. For example, the sound pressure level attenuation is about 1 dB for a sound wave of 250 Hz travelling 1000 m; however, for a sound wave at 4000 Hz, the attenuation can be from 24 to 67 dB depending on the temperature and humidity of the air [3]. More about the measures for describing the sound propagation loss in textiles will be introduced in Sect. 1.1.1.

When sound encounters different medium or space, discontinuity in the propagation, reflection happens, where the incident wave arriving at the boundary interact with it to produce waves travelling away from the boundary [2]. The reflected wave or scattered waves follow certain rules. For example, for the specular reflection in which a plane incident wave is reflected by a uniform plane boundary, the normal wavenumber component of the incident field is reversed on reflection, and the wavenumber component parallel to the boundary is unaltered, so the angle of reflection is equal to the angle of incidence.

There is usually an energy loss when sound is reflected from a boundary, and the changes in amplitude and phase taken place during the reflection can be represented by the complex reflection factor (R) or sound pressure reflection coefficient as

$$R = \frac{p_{\mathrm{r}}}{p_{\mathrm{i}}} \tag{1.1.1}$$

where p_{r} and p_{i} are complex amplitudes of the reflected and incident waves, respectively. The sound pressure reflection coefficient is a property of the boundary, whose magnitude and phase values depend on the frequency as well as the direction of the incident wave. More about the acoustic absorption and scattering on the boundary will be introduced in Sects. 1.1.2 and 1.1.3. Because the propagation and absorption properties of a textile depend on its flow resistance, the flow resistance will be introduced in Sect. 1.1.4.

1.1.1 Propagation

For the sound wave propagating in porous textiles, the porous gas-filled medium is often treated as an equivalent uniform medium for analysis purpose; so a propagation factor $e^{j\omega t-\gamma x}$ is used to describe the dependence of the propagating wave on time t and the propagation coordinate x [3]. Here, $\omega = 2\pi f$ is the angular frequency and f is the wave frequency. The propagation constant γ is also called the propagation coefficient, which is a complex number and can be described by,

$$\gamma = \alpha + jk \tag{1.1.2}$$

where the propagation wavenumber $k = \omega/c$ is also called the phase coefficient, α is called the attenuation coefficient, and c is the speed of sound in the medium. The attenuation coefficient α describes the reduction in amplitude of a progressive wave with the distance in the propagation direction by $e^{-\alpha x}$. This coefficient is actually the amplitude attenuation coefficient instead of the energy attenuation coefficient.

When a sound wave is incident upon a partition or a textile layer, some of it will be reflected back to the incidence medium and some will be transmitted through the partition or the textile layer. The fraction of incident energy that is transmitted is called the transmission coefficient (τ). If there is no other energy dissipation or reflections, the relationship between the transmission coefficient and the attenuation coefficient is

$$\tau = 1 - \alpha^2 \tag{1.1.3}$$

The transmission loss, TL is defined as the logarithm of the reciprocal of the transmission coefficient as

$$TL = -10\log_{10}\tau \tag{1.1.4}$$

In general, the transmission loss through a porous layer depends upon the angle of incidence and is a function of material density, thickness, flow resistivity and frequency. In the low frequency range, the transmission loss of common porous layers is usually less than 20 dB, but can be greater than 20 dB in high frequency range. The transmission loss usually increases with the material density, thickness, flow resistivity and frequency.

1.1.2 Absorption

There is usually an energy loss when sound propagates in a medium. Although the energy loss can be caused by some kinds of energy dissipation such as absorption, the term "attenuation coefficient" is used in Sect. 1.1.1 to describe the reduction in amplitude of a progressive wave with the distance in the propagation direction in the medium. When the sound wave encounters different medium or space discontinuity in the propagation, reflection happens. The energy loss associated with the reflection from a boundary is described by the sound pressure reflection coefficient, which is the change in amplitude and phase of the propagating wave taken place during the reflection. It is a property of the boundary, and the value depends on the frequency as well as the direction of the incident wave.

Acoustic absorption coefficient at a boundary, also called sound absorption factor or sound power absorption coefficient, is defined as the fraction of the incident acoustic power arriving at the boundary that is not reflected, and is therefore regraded as being absorbed by the boundary [2],

$$\alpha = 1 - \frac{E_{\mathrm{r}}}{E_{\mathrm{i}}} \tag{1.1.5}$$

where E_{r} and E_{i} are the reflected acoustic power and the incident acoustic power arriving at the boundary, respectively. The values of the acoustic absorption coefficient can range from 0 to 1.0. A value of 0 refers to the condition when there is a total reflection, whereas 1.0 indicates perfect absorption and no reflection. It is also a function of frequency and incident wave direction. The (random incidence) statistical absorption coefficient can be obtained by,

$$\alpha_{\mathrm{st}} = \frac{1}{\pi}\int\limits_{0}^{2\pi} \mathrm{d}\varphi \int\limits_{0}^{\pi/2} \alpha(\theta)\cos\theta\,\sin\theta\,\mathrm{d}\theta \tag{1.1.6}$$

where θ and φ are the angle of elevation and azimuth, respectively. The Sabine absorption coefficient is the random incidence absorption coefficient deduced from the reverberation time measurement via the Sabine equation.

Porous absorbers are materials where sound propagation occurs in a network of interconnected pores (open pore structure) in such a way that viscous and thermal effects cause acoustic energy to be dissipated. Air is a viscous fluid, and consequently acoustic energy is dissipated via friction with the pore walls. As well as viscous effects, there will be losses due to the thermal conduction from the air to the absorber material, which might be more significant at low frequency. For a porous absorber to create significant absorption, it needs to be placed somewhere where the particle velocity is high. The particle velocity close to a room boundary is usually zero, so the parts furthest from the backing surface are often most effective. The material needs to be greater than one-tenth wavelength thick to cause significant absorption, and about one-fourth wavelength to absorb most incident sound. More will be discussed in Sect. 1.2.

It is often difficult to obtain absorption at low frequencies with porous textile absorbers because the required thickness of the material is large and the treatments are often placed at room boundaries where the absorbers are inefficient due to the low particle velocity. Resonant absorbers can be an alternative solution. There are two common forms of resonant absorbers: membrane/panel absorber and Helmholtz absorber. For a membrane/panel absorber, the mass is usually a sheet of mass-loaded vinyl or plywood which vibrates; while for a Helmholtz absorber, the mass is a plug of air in the opening of the perforated sheet. The spring in both the cases is provided by air enclosed in the cavity. The resonant frequency of this type of absorbers can be tuned. More will be discussed in Sect. 1.3. Sound absorbers are usually employed for adjusting the reverberation of the room, suppressing undesired sound reflections from remote walls (echoes), and reducing the acoustical energy density and hence the sound pressure level in noisy rooms. More will be discussed in Sect. 1.5.

1.1.3 Scattering

The degree of scattering and absorption in a room are important factors related to the acoustic quality of the room. Although scattered polar responses can give much information about the scattering from a textile surface, it yields too much detail, and a single value is desired in practice to allow easy comparison of diffuser quality. There is also a need for a scattering coefficient to evaluate the amount of dispersion generated by a surface to allow accurate predictions using geometric room acoustic models [4]. The geometric models use a scattering coefficient to determine the proportion of the reflected energy that is reflected in a specular manner and the proportion that is scattered.

The scattering coefficient, s_θ for an incident wave arriving at the boundary with an angle θ is defined as the value calculated by one minus the ratio of the specularly reflected acoustic energy to the total reflected acoustic energy,

$$s_\theta = 1 - \frac{E_\mathrm{s}}{E_\mathrm{r}} \tag{1.1.7}$$

where E_s is the specularly reflected acoustic energy and E_r is the total reflected acoustic energy from the incident acoustic power arriving at the boundary. Theoretically, s_θ takes a value between 0 and 1, where 0 means a totally specularly reflecting surface and 1 means a totally scattering surface. When there is no subscript, random incidence scattering coefficient s is used, which is defined as the value calculated by one minus the ratio of the specularly reflected acoustic energy to the total acoustic energy reflected from a surface in a diffuse sound field.

The directional diffusion coefficient $d_{\theta,\varphi}$ is a measure of the uniformity of diffusion produced by a surface for one sound source [4],

$$d_{\theta,\varphi} = \frac{\left(\sum_{i=1}^{n} 10^{L_i/10}\right)^2 - \sum_{i=1}^{n} \left(10^{L_i/10}\right)^2}{(n-1)\sum_{i=1}^{n} \left(10^{L_i/10}\right)^2} \tag{1.1.8}$$

where the subscript θ is the angle of incidence relative to the reference normal of the surface, and φ indicates the azimuth angle, n is number of receivers, L_i are a set of sound pressure levels in a polar response of the scattering sound. The directional

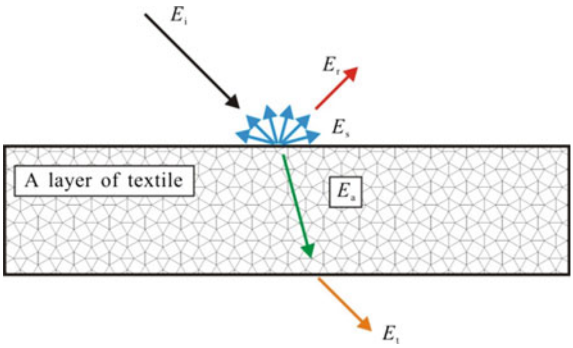

Fig. 1 Energy relationships when a propagating wave is incident upon a lay of textile

diffusion coefficient has a value between 0 and 1, corresponding to that one receiver receives non-zero scattered sound pressure or complete diffusion, respectively. The random incidence diffusion coefficient calculated from weighting the directional diffusion coefficients for the difference source positions is a measure of the uniformity of diffusion for a representative sample of sources over a complete semicircle for a single plane diffuser, or a complete hemisphere for a hemispherical diffuser.

Both the scattering and diffusion coefficients are simplified representations of the true reflection behaviour [4]. The purpose of the diffusion coefficient is to enable the design of diffusers and to allow acousticians to compare the performance of surfaces for room design and performance specifications. While the scattering coefficient is to characterize surface scattering for use in geometrical room modelling programs, the diffusion coefficient is different from, but related to, the random incidence scattering coefficient. The scattering coefficient is a rough measure of the degree of scattered sound, while the diffusion coefficient describes the directional uniformity of the scattering, i.e. the quality of the diffusing surface.

Figure 1 summarizes the relationships of different kinds of energy when a propagating wave is incident upon a layer of textile. The total input energy brought from the incident wave is E_i, which is equal to the summation of E_r, E_s, E_a and E_t, i.e., the energy reflected and scattered from the boundary, the energy dissipated inside the textile layer and the energy transmitted through the layer.

1.1.4 Flow Resistivity

Most textiles that can be used for acoustic purposes show open porosity due to many interconnected pores or voids inside. The acoustic performance of a porous textile is mainly determined by its flow resistivity, which is an intrinsic property of the textile and is a measure of how easily air can enter a porous textile material and the resistance that the air flow meets through the material [4]. Flow resistivity is also known as static flow resistivity and is used to describe some of the structural properties in an indirect manner. It may be used to establish correlations between the structure of the materials and some of the acoustical properties. It plays a critical role in the calculation of many intrinsic acoustic properties of porous textiles, such as the characteristic impedance, the propagation constant and the sound absorption coefficient. Flow resistivity has a unit of N.s.m^{-4} and is defined as the unit-thickness specific flow resistance (σ) by [2],

$$\sigma = \frac{R_\mathrm{f}}{h} \tag{1.1.9}$$

where R_f is the specific flow resistance of a uniform textile layer of thickness h and is defined as

$$R_\mathrm{f} = \frac{\Delta p\, S}{V_0} \tag{1.1.10}$$

where Δp is the pressure drop through the layer with a surface area of S under conditions of slow steady flow with a volume flow rate of V_0.

(Static) flow resistance is the ratio of the pressure drop across a porous element to the volume velocity flowing through it under conditions of steady low speed flow. The flow resistance is almost independent of the volume velocity at low speeds; however, it is frequency dependent. Dynamic specific flow resistance of a thin (compared to an acoustic wavelength) porous textile layer is the real part of the complex specific flow impedance at a specified frequency, which is defined as the complex ratio of the pressure drop across the layer to the relative face velocity through the layer. When the frequency approaches zero, the dynamic specific flow resistance varies little with frequency, so it almost equals to the static flow resistance. For fibrous porous materials such as fibreglass and rockwool, their flow resistivity can be estimated with [3]:

$$\sigma = 27.3\left(\frac{\rho_\mathrm{m}}{\rho_\mathrm{f}}\right)^{k_1}\frac{\mu}{d^2} \tag{1.1.11}$$

where ρ_m is the porous material bulk density, ρ_f is the fibre material density, μ is the dynamic gas viscosity (1.84×10^{-5} kg m^{-1} s^{-1} for air at 20 °C), d is the fibre diameter of the porous material, and the constant $k_1 = 1.53$ for fibreglass and rockwool with a diameter between 1 and 15 μm and might be a different value for other fibres and different diameters.

1.2 Porous Absorbers

Porous absorbers are materials where sound propagation occurs in a network of interconnected pores in such a way that viscous and thermal effects cause acoustic energy to be dissipated [4]. Carpets, acoustic tiles, open cell acoustic foams,

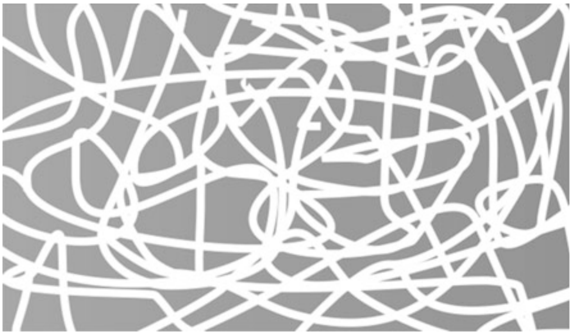

Fig. 2 Illustration of an open pore structure for sound absorption

curtains, cushions, cotton and mineral wools such as fibreglass are such materials. Air is a viscous fluid, and consequently acoustic energy is dissipated via friction with the pore walls provided that the size of pore is sufficiently small. As well as viscous effects, there are losses due to thermal conduction from the air to the absorber material, especially at low frequency. For the absorption to be effective there must be interconnected air paths through the material; so an open pore structure is necessary, as shown in Fig. 2 [5]. The viscous and thermal effects are affected by the pore diameter, network shape and layout, density and other physical properties of the material such as cell size and thickness, so the micromodels for understanding the underneath physical mechanisms of sound absorption in porous materials are quite complicated.

Fortunately, these micro effects for sound absorption of a uniform textile layer can be approximately represented by a macro quantity, i.e. the specific flow resistance. As introduced in Sect. 1.1.4, the flow resistivity of a textile layer is an intrinsic property of the textile and is a measure of how easily air can enter a porous textile material and the resistance that the air flow meets through the material. Equation (1.1.11) shows that the flow resistivity of certain fibreglass and rockwool is proportional to the dynamic gas viscosity and the ratio of the porous material bulk density to the fibre material density, but inversely proportional to the squared fibre diameter; however, it varies little with frequency.

For the sound wave propagating in porous textiles, the porous gas-filled medium is often treated as an equivalent uniform medium for analysis purpose, which can be characterized in dimensionless variables by a complex density and complex compressibility [3]. These quantities can be calculated from the flow resistivity and then be used to calculate the characteristic impedance and the complex propagation constant of the porous material. The following empirical formulas can be used to estimate the characteristic impedance, Z_m, and the complex propagation constant, k_m, of the porous material directly from the flow resistivity for fibreglass and rockwool materials with a small amount of binder and having short fibres smaller than 15 μm in diameter [6],

$$Z_m = \rho_0 c_0 \left[1 + 0.0571 \left(\frac{\rho_0 f}{\sigma} \right)^{-0.754} - j0.087 \left(\frac{\rho_0 f}{\sigma} \right)^{-0.732} \right] \quad (1.2.1)$$

$$k_m = k_0 \left[1 + 0.0978 \left(\frac{\rho_0 f}{\sigma} \right)^{-0.700} - j0.189 \left(\frac{\rho_0 f}{\sigma} \right)^{-0.595} \right] \quad (1.2.2)$$

where ρ_0 is the air density, c_0 is the speed of sound, $k_0 = 2\pi f/c_0$ is the wavenumber in air and f is the wave frequency.

When a sound wave is incident upon a textile layer, some of the energy is reflected back to the incidence medium and some will be transmitted through the partition or the textile layer. The absorption coefficient at the boundary is the fraction of the incident acoustic power arriving at the boundary that is not reflected, and its value depends not only on the properties of the material but also on the properties of the space behind the material. For a layer of textile, its absorption coefficient when it is backed by a rigid wall is different to that when it is hanged in a room. The value depends on the distance between the layer and the walls as well as the angle of incidence. For the simplest case of normal incidence with a plane wave, its absorption coefficient can be calculated with the reflection coefficient (R) as follows [3],

$$\alpha = 1 - |R|^2 = 1 - \left| \frac{Z_s - \rho_0 c_0}{Z_s + \rho_0 c_0} \right|^2 \quad (1.2.3)$$

where the specific acoustic impedance at the layer's front surface, Z_s, can be formulated with the impedance transfer function for a layer of porous material with a thickness of l by [7] and, [8],

$$Z_s = Z_m \frac{Z_L + jZ_m \tan(k_m l)}{Z_m + jZ_L \tan(k_m l)} \quad (1.2.4)$$

Z_m, k_m and Z_L are respectively the characteristic impedance, propagation constant and the acoustic impedance at the back surface of the porous material. For the configuration shown in Fig. 3, when the wall is rigid and the distance between the layer and the wall is L, Z_L can be calculated by

$$Z_L = -j\rho_0 c_0 \cot(k_0 L) \quad (1.2.5)$$

where $k_0 = \omega/c_0$ is the wavenumber in the air.

For a layer of textile placed against the rigid wall, $L = 0$, so $Z_L \to -\infty$, Eq. (1.2.4) becomes,

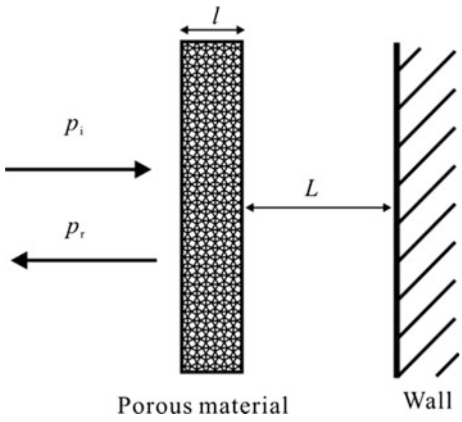

Fig. 3 A layer of porous material with a thickness of l located with a distance of L from a wall

$$Z_s = -jZ_m \cot(k_m l) \tag{1.2.6}$$

The characteristic impedance in Eq. (1.2.1) can be denoted simply as $Z_m = \rho_p c_p$, where ρ_p is the complex density and c_p is the speed of sound in the layer of the porous material. The complex propagation constant in Eq. (1.2.2) can also be denoted simply as $k_m = k_p - j\alpha_p$, where the propagation wavenumber $k_p = 2\pi f/c_p$ is the phase coefficient and α_p is called the attenuation coefficient in the layer of the porous material. Substitute them into Eq. (1.2.6), it has

$$Z_s \approx -j\rho_p c_p \cot(k_p l - j\alpha_p l) \tag{1.2.7}$$

Normalizing this specific acoustic impedance at the layer's front surface with the characteristic impedance of air yields

$$\frac{Z_s}{\rho_0 c_0} = \frac{-j\rho_p c_p \cot(k_p l - j\alpha_p l)}{\rho_0 c_0} = \gamma A + j\gamma B \tag{1.2.8}$$

where $\gamma = \rho_p c_p / \rho_0 c_0$ is the ratio of the characteristic impedance of the porous material to that of the air and

$$A = \frac{\sinh(2\alpha_p l)}{\cosh(2\alpha_p l) - \cos(2k_p l)}, \; B = \frac{-\sin(2k_p l)}{\cosh(2\alpha_p l) - \cos(2k_p l)} \tag{1.2.9}$$

Using Eq. (1.2.3), the absorption coefficient for a layer of porous material placed directly against the rigid wall under normal incidence is

$$\alpha = \frac{4\gamma A}{(1+\gamma A)^2 + (\gamma B)^2} \tag{1.2.10}$$

This equation can be used to illustrate the sound absorption mechanisms of porous material. For example, when the sound attenuation capability of the material is sufficient strong or the thickness of the absorption material is sufficient long, i.e. $\alpha_p l \gg 1$, then $\sinh(2\alpha_p l)$ and $\cosh(2\alpha_p l)$ tend to infinite large, so $A \approx 1$ and $B \approx 0$, the absorption coefficient becomes

$$\alpha = \frac{4\gamma}{(1+\gamma)^2} \tag{1.2.11}$$

It is clear that the absorption coefficient can be 1 only when $\gamma = 1$, i.e. the characteristic impedance of the porous material equals to that in the air. The sound attenuation capability of a porous material is not equivalent to its sound absorption capability. To have perfect sound absorption capability, the characteristic impedance of the porous material needs to match that in the air so that no sound is reflected back first, and then the transmitted sound into the material can be absorbed or dissipated partially or completely by the material.

If $k_p l = \pi/2$, i.e. $l = \lambda_p/4$, λ_p is the wavelength of the sound in the material, then $A = \tanh(\alpha_p l)$ and $B = 0$, the absorption coefficient researches a maximal value,

$$\alpha_{\max} = \frac{4\gamma \tanh(\alpha_p l)}{[1+\gamma \tanh(\alpha_p l)]^2} \tag{1.2.12}$$

If $k_p l = \pi$, i.e. $l = \lambda_p/2$, then $A = 1/\tanh(\alpha_p l)$ and $B = 0$, the absorption coefficient researches a minimal value,

$$\alpha_{\min} = \frac{4\gamma \tanh(\alpha_p l)}{[\gamma + \tanh(\alpha_p l)]^2} \tag{1.2.13}$$

The corresponding frequencies to the maximal and minimal values are called the resonant frequency and anti-resonant frequency.

When there is a space between the layer of the material and the wall, i.e. $L \neq 0$, the variation of the absorption coefficient can be analyzed similarly with Eqs. (1.2.3)–(1.2.5). A general conclusion is that the space between the layer of the material and the wall increases its absorption value at low frequency. The physical explanation is that the particle velocity of sound wave close to rigid walls is usually zero, so placing the material somewhere further from the backing surface will have more absorption because of high particle velocity of sound wave.

Figure 4 shows typical sound absorption coefficient curves of a porous material as a function of frequency, where the solid line corresponds to that of a 5 cm thick porous material with a flow resistivity of 10,000 N s m^{-4} placed directly against a rigid wall, the dot line corresponds to that of the 5 cm thick porous material placed 10 cm away from the rigid wall, and the dash-dot line corresponds to that of the same porous material but with a thickness of 15 cm placed directly against a rigid wall. It is obvious that the absorption coefficient increases with frequency. The

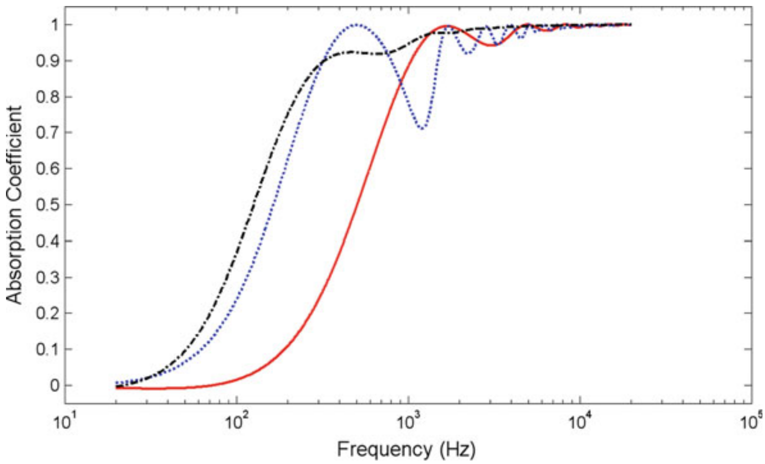

Fig. 4 Typical sound absorption coefficient curves of a layer of porous material with a flow resistivity of 10,000 N s m^{-4} as a function of frequency, *solid line* corresponds to that of a 5 cm thick material placed directly against a rigid wall, the *dot line* corresponds to that of the 5 cm thick porous material placed at 10 cm away from the rigid wall, and the *dash-dot line* corresponds to that of the same porous material but with a thickness of 15 cm placed directly against a rigid wall

mechanism for the low absorption coefficient at low frequency can be illustrated with Eq. (1.2.4). At very low frequency, $k_m l \ll 1$, then $\tan(k_m l)$ tends to be zero, thus Z_s is almost equal to Z_L and the porous material seems not existing. Z_L is the acoustic impedance at the back surface of the porous material, which can be calculated with Eq. (1.2.5) and is usually much larger than the characteristic impedance of the air, so most the incident acoustic energy will be reflected back because of impedance mismatching.

1.3 Resonance Absorbers

As shown in Sect. 1.2, it is often difficult to have large sound absorption at low to mid frequencies with porous absorbers in practice because the required thickness of the material is large and the treatments are often placed at room boundaries where porous absorbers are inefficient due to the low particle velocity. A different kind of sound absorber called resonant absorber can be used to solve the problem, which can tune its resonant frequency to a specified frequency and have maximal sound absorption there [4]. The resonant absorbers work on the principle of dissipating acoustic energy with structure vibration. The most significant difference between resonance absorbers and porous absorbers is that the former is usually only effective in a narrow tunable low frequency band while the latter is effective for broadband from mid to high frequency.

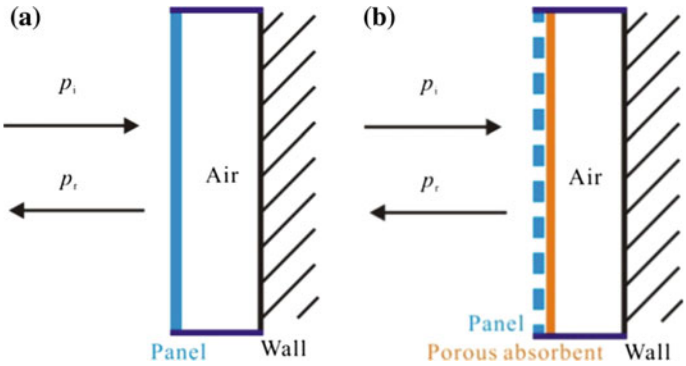

Fig. 5 A schematic diagram of: (**a**) a panel absorber and (**b**) a Helmholtz absorber

There are two common forms of the resonant absorbers: membrane/panel absorber and Helmholtz absorber. For a membrane/panel absorber, the mass is a vibrating sheet of membrane or panel made from various materials, the spring is usually provided by the resilient boundary of the membrane or panel or air enclosed in the cavity; while in the case of a Helmholtz absorber, the mass is a plug of air in the opening of a perforated sheet and the spring is usually provided by air enclosed in the cavity. It is often best to place some porous absorbent in the neck of a Helmholtz resonator or just behind the membrane or panel in a membrane/panel absorber to increase the acoustic resistance of the whole absorber. Figure 5 shows a schematic diagram of a panel absorber and a Helmholtz absorber.

The mechanism of sound absorption for membrane or panel absorbers is energy dissipation by vibration of the membrane or panel. Whether the flexible membranes or panels are mounted over an air space or are mounted on a suspended ceiling, the membranes or panels must couple with and be driven by the sound field. Acoustic energy is then dissipated by flexure of the membrane or panel. Additionally, if the backing air space is filled with a porous material, energy is also dissipated in the porous material. Maximum absorption occurs at the first resonance of the coupled membrane/panel-cavity system [3].

It should be noted that this mechanism of resonant absorbers is different to that of porous absorbers. For the porous absorbers, the absorption depends only upon local properties of the material, while for the membrane/panel absorbers the absorption is dependent upon the response of the panel as a whole. Furthermore, as the panel absorber depends upon strong coupling with the sound field to be effective, the energy dissipated is very much dependent on the sound field and thus on the rest of the room in which the panel absorber is used. This latter fact makes prediction of the absorptive properties of membrane/panel absorbers difficult. Two design methods (one is empirical based upon data measured in auditoria and concert halls and the other is based upon analysis) can be found in the Ref. [3] for estimating the Sabine absorption of panel absorbers. The two main steps in these methods are tuning the resonant frequency and adjusting the sound absorption peak value and band width; however, they will not be introduced here because they are quite complicated.

1.3.1 Helmholtz Resonant Absorber

The sound absorption mechanism of Helmholtz resonators is to form an acoustic resonant system so that the acoustic energy can be better dissipated with the acoustic resistance, thanks to the amplified particle velocity. The acoustic resistance can be obtained by having some porous material near the neck where the particle velocity is the maximal or to make the opening very small to have high acoustic resistance. Figure 6 shows a schematic diagram of a single Helmholtz absorber, which is an individual cell of Fig. 5b. The Helmholtz resonator consists of an air cavity with a volume of V acting like spring in a mechanical resonant system, a neck with a length of l_0 and a diameter of d. The mass of the air in the neck acts like the mass in the mechanical resonant system. Because there is no physical subdividing in the volume in Fig. 5b, the model in Fig. 6 is an approximation of that in Fig. 5b. However, when a porous absorbent is placed in the cavity, the need for physical subdividing is less critical [4].

For a single Helmholtz absorber shown in Fig. 6, the cavity walls are usually rigid, the volume of neck is smaller than that of the volume, and the dimension of the device is usually much smaller than the wavelength because the resonant frequency is quite low. The acoustic impedance of the absorber can be expressed as

$$Z_a = R_a + jX_a \tag{1.3.1}$$

where R_a is the acoustic resistance of the absorber, which is related to the porous material used in the cavity and/or the viscous loss along the neck. X_a is the acoustic reactance of the absorber and can be calculated from the mechanical reactance of the system by

$$X_a = \left(\omega M_a - \frac{K_a}{\omega}\right) \tag{1.3.2}$$

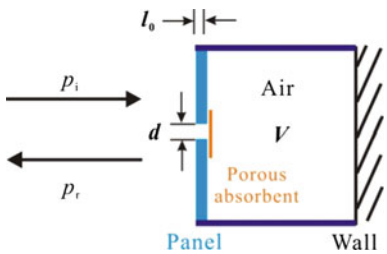

Fig. 6 A schematic diagram of a single Helmholtz absorber

where $\omega = 2\pi f$ is the angular frequency and f is the frequency, $M_a = \rho_0 l_e/S_0$ is the mass of the air in the neck and $l_e = l_0 + 0.8d$ is the equivalent length of the neck, $S_0 = \pi d^2/4$ is the cross-section area of the neck, $K_a = \rho_0 c_0^2/V$ is the acoustic stiffness of the air cavity as a spring. The local specific acoustic impedance of the absorber, $Z_s = R_s + jX_s$, can be obtained by

$$Z_s = Z_a S \tag{1.3.3}$$

where S is the cross-section area of the whole absorber to the incident sound. Similar to that for porous absorbers, the absorption coefficient can be calculated with the reflection coefficient as follows [3],

$$\begin{aligned} \alpha &= 1 - \left|\frac{Z_s - \rho_0 c_0}{Z_s + \rho_0 c_0}\right|^2 \\ &= \frac{4R_s/\rho_0 c_0}{(R_s/\rho_0 c_0 + 1)^2 + (X_s/\rho_0 c_0)^2} \end{aligned} \tag{1.3.4}$$

If $X_a = 0$, the absorption coefficient will be maximum, and the resonant frequency can be calculated by setting Eq. (1.3.2) to zero. The solved resonant frequency depends on the geometry of the Helmholtz absorber as shown in Eq. (1.3.5).

$$f_r = \frac{1}{2\pi}\sqrt{\frac{K_a}{M_a}} = \frac{c_0}{2\pi}\sqrt{\frac{S_0}{VL_e}} \tag{1.3.5}$$

To have low resonance frequency, the volume of the cavity should be large, the length of the neck should be long and the cross-section area of the neck should be small. Even at the resonant frequency, the acoustic resistance of the absorber needs to match the characteristic impedance of the air so that $R_s = \rho_0 c_0$ to have perfect absorption.

Unlike porous absorbers, resonant absorbers sometimes are not distributed uniformly against the boundaries but used as a single separate object. Under this situation, absorption coefficient which refers to a uniform surface might not be appropriate for describing their sound absorption performance. Another terminology, absorption cross section (or equivalent absorption area), which is defined as the ratio of sound power being absorbed and the intensity which the incident sound wave would have at the place of the absorber if it were not present, can be used to describe the sound absorption performance of individual absorbers [5]. For a Helmholtz absorber, its absorption cross section is approximately $\lambda^2/2\pi$, where λ is the wavelength corresponding to the resonant frequency.

If the Helmholtz absorbers are distributed uniformly as that shown in Fig. 5b, its sound absorption coefficient can be calculated with Eq. (1.3.4). Figure 7 shows sound absorption coefficient curves of a typical perforated sheet (a 6.3 mm thick panel with a perforation rate of 6 %, the diameter of the holes is 5 mm, and the air

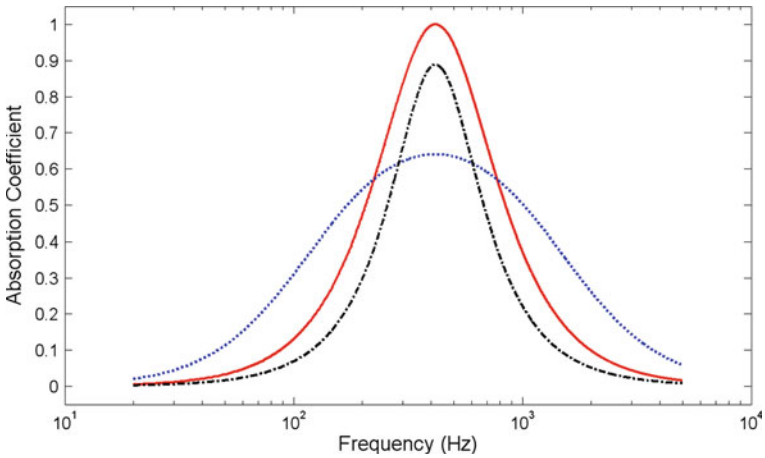

Fig. 7 Sound absorption coefficient curves of a typical perforated sheet (a 6.3 mm thick panel with a perforation rate of 6 %, the diameter of the holes is 5 mm, the air layer thickness between the panel and the wall is 0.1 m) as a function of frequency, where the *solid line* corresponds to that with $R_s = \rho_0 c_0$, the *dot line* corresponds to that with $R_s = 4\rho_0 c_0$, and the *dash-dot line* corresponds to that with $R_s = 0.5\rho_0 c_0$, the resonant frequency of the absorber is approximately 418 Hz

layer thickness between the panel and the wall is 0.1 m) as a function of frequency, where the solid line corresponds to that with $R_s = \rho_0 c_0$, the dot line corresponds to that with $R_s = 4\rho_0 c_0$, and the dash-dot line corresponds to that with $R_s = 0.5\rho_0 c_0$. It is clear that the acoustic resistance of the absorber needs to be finely tuned to have perfect absorption. The resonant frequency of the absorber is approximately 418 Hz under the current configuration, but can be reduced by increasing the air layer thickness between the panel and the wall or the length of the neck. It can also be reduced by reducing the size of the neck cross-section area as shown in Eq. (1.3.5).

1.3.2 Micro-Perforated Panel Resonators

A micro-perforated panel (MPP) consists of a thin sheet panel perforated with a lattice of sub-millimetre apertures which create high acoustic resistance and low acoustic mass reactance necessary for broadband sound absorption without further using additional porous material [9]. Because the light and fibreless MPPs can be made of various recyclable materials, they are becoming more and more widely used in sound field control today, especially in clean situations where strict hygiene is required or in industrial environments where porous materials deteriorate. Furthermore, MPPs can be made of transparent or colourful plates or membranes, so they are also in demand by architects for sound quality control in auditoriums [10]. MPP absorbers are tagged as the “next generation” absorbing materials due to their huge potential in comparison with conventional porous materials [11].

The mechanism of MPP absorption is related to the resonance effect, where the air inside the apertures of the MPP vibrates like a mass and the air inside the backing cavity acts like a spring, so the effective sound absorption frequency band is around the resonance peak. Overcoming this drawback (narrow absorption frequency range) has been the subject of a significant amount of recent work [12–21]. For example, additional MPPs can be used in the backing cavity to form multiple-layer absorbers and thus extend the width of the absorption frequency band. The backing cavity can be partitioned into parallel cells using a honeycomb structure or it can be transformed to an irregular shape to increase the coupling between the acoustic modes in the backing cavity, thereby extending the sound absorption bandwidth. Even optimizing the coupling effect between the cavity acoustical response and the vibration of the MPP has been taken into account to expand the frequency bandwidth. Additional mechanical components have been proposed to be added to the simple panel-hole-cavity system to improve its performance using a flexible panel to substitute the rigid backing wall of a normal MPP system to produce a finite-sized micro-perforated panel-cavity-panel partition.

Despite all these efforts, the frequency range of MPP absorbers is still not satisfactory in many practical applications, especially for low frequency sound absorption applications where a larger cavity has to be used. Active control techniques have been introduced recently into MPP systems to address the low frequency limitations of the passive systems. In these cases, a loudspeaker and an error microphone have been placed behind an MPP layer to obtain the desired acoustic performance using "pressure releasing" on the back of the MPP or "impedance matching" of the active surface of the hybrid noise control system [22–25]. A hybrid passive-active system using flexible MPPs has been studied, where it has been shown that the panel vibration can increase the absorption at the structural resonance frequencies when the impedance matching approach is used [26].

The acoustic impedance of the hole of the micro-perforated panel can be approximated as [9]

$$Z_{\mathrm{MPP}} \approx \frac{32\eta t}{d^2}\left(\sqrt{1+\frac{K^2}{32}}+\frac{\sqrt{2}Kd}{32t}\right) - \mathrm{j}\rho_0\omega t\left[1+\left(1+\frac{K^2}{2}\right)^{-\frac{1}{2}}+0.85\frac{d}{t}\right] \tag{1.3.6}$$

where t is panel thickness, d is diameter of the hole, η is the dynamic viscosity coefficient, $\omega = 2\pi f$ is angular frequency and f is the frequency of interest, ρ_0 is the air density, and $K^2 = d^2\rho_0\omega/4\eta$. After obtaining the effective specific normal acoustic impedance Z_{MPPS} at the front surface of the MPP, and assuming that the MPP is local reactive, the complex amplitude reflection coefficient for a plane wave with an incidence angle of θ can be expressed as [19]

$$R(\theta) = \frac{Z_{\text{MPPS}} \cos\theta - \rho_0 c_0}{Z_{\text{MPPS}} \cos\theta + \rho_0 c_0} \tag{1.3.7}$$

Then the statistical absorption coefficient for random incidence can be obtained using

$$\alpha_{\text{st}} = 1 - 2 \int_0^{\pi/2} |R(\theta)|^2 \cos\theta \, \sin\theta \mathrm{d}\theta \tag{1.3.8}$$

1.4 Factors Affecting Sound Absorption

As introduced in the last two sections, the main mechanisms of sound absorbers are acoustic impedance matching on the absorbers' surface and acoustic energy dissipation inside the absorbers. If the acoustics impedance at the surface of the sound absorbers does not match that of the medium where the incident sound comes from, some or most of the sound is reflected back to the medium. When the acoustics impedance at the surface of the sound absorbers matches that of the medium, the sound is not reflected back to the medium, and this makes it possible to dissipate acoustic energy in the absorbers. To maximize the acoustic energy dissipation inside the absorbers, various physical effects can be used, such as viscous along the boundaries, heat exchanges, mechanical vibration, magnetic and electrical damping. For the textile materials, although a lot of factors such as fibre type, material thickness, fibre size, porosity, density, tortuosity and compression can affect the sound absorption of the material, there is no direct simple relationship between the value of the sound absorption coefficient and these factors [27].

Because the viscous and thermal effects are affected by the pore diameter, network shape and layout, density and other physical properties of the material in very complicated ways, the micromodels for understanding the underneath physical mechanisms of sound absorption in porous materials are quite complicated. Although there is no direct relationship between the physical and geometrical properties of textiles and the sound absorption, these micro effects for sound absorption of a uniform textile layer can be approximately represented by a macro quantity, i.e. the specific flow resistance. As introduced in Sect. 1.1.4, the flow resistivity of a textile layer is an intrinsic property of the textile and is a measure of how easily air can enter a porous textile material and the resistance that the air flow meets through the material. Equation (1.1.11) shows that the flow resistivity of certain fibreglass and rockwool is proportional to the dynamic gas viscosity linearly and the ratio of the porous material bulk density to the fibre material density nonlinearly, but is inversely proportional to the squared fibre diameter. The relationship between the flow resistance and sound absorption are very complicated. As shown by Eqs. (1.2.1)–(1.2.2) in Sect. 1.2, the characteristic impedance and propagation constant can be calculated with the flow resistivity in the nonlinear complex domain, which can then be used to obtain the specific acoustic impedance of the layer surface with Eq. (1.2.4) using complicated triangular functions. Finally, the absorption coefficient can be calculated from the ratio of the specific acoustic impedance of the layer surface to the characteristic impedance of the medium. In practice, the absorption performance of absorbers also depends on the sound field where it is located.

1.4.1 Material Thickness

Thickness of textile structures is one of the important parameters affecting the sound absorption. If the acoustics impedance at the surface of the textile structures matches that of the medium, the sound is not reflected back to the medium, then the thicker the structure is, the larger its sound absorption will be. To have effective sound absorption in the structure under this condition, the thickness of the structure should be at least one-tenth of the wavelength of the incident sound wave. This implies that thicker structures are required for absorbing low frequency sound due to its long wavelength. If the acoustics impedance at the surface of the textile structures partially matches that of the medium, thicker structures would absorb more sound that is not reflected back to the medium, especially in the low frequency range. If the acoustics impedance at the surface of the textile structures differs significantly with that of the medium, most sound is reflected back to the medium. Under this situation, the thickness of structures hardly affects sound absorption.

1.4.2 Fibre Size

As shown in Eq. (1.1.11), the flow resistivity is inversely proportional to the squared fibre diameter. This implies that finer fibres would result in high flow resistivity if all other parameters of the structure remain the same. Unfortunately, the relationship between the flow resistance and sound absorption is not straightforward but in a very complicated way. As shown by Eqs. (1.2.1)–(1.2.2), if the flow resistivity of the structures is very small, the characteristic impedance and propagation constant of the structure tend to be those in the medium. Under this situation, if there is no rigid wall against the structure and the structure is infinite thick and can dissipate the sound energy somehow, the sound absorption of the structure can be large. However, large fibre size does not necessarily result in small flow resistivity of the structures, and small flow resistivity of the structures does not necessarily result in large sound absorption. On the other hand, if the flow resistivity of the structures is very large, the characteristic impedance and propagation constant of the structure can be very different to those in the medium, so the sound absorption of the structure might be small.

1.4.3 Porosity

The porosity of a material indicates the amount of empty space or void present in the structure. Porosity is expressed as the ratio of amount of void present in the structure to the total volume of the sample

$$P_{\text{porosity}} = \frac{V_e}{V_t} \tag{1.4.1}$$

where V_e is the volume of the empty space and V_t is the total volume of the material. In the case of porous sound absorbers, the type, size and number of pores influence the sound absorption. Higher number of pores in a structure or large porosity usually means small porous material bulk density. As shown in Eq. (1.1.11), the flow resistivity is proportional to the porous material bulk density nonlinearly, which implies that large porosity would result in small flow resistivity if all other parameters of the structure remain the same. Unfortunately, the relationship between the flow resistance and sound absorption is very complicated. As shown in Eqs. (1.2.1)–(1.2.2), if the flow resistivity of the structures is small, the characteristic impedance and propagation constant of the structure tend to be those in the medium. Under this situation, if there is no rigid wall against the structure and the structure is infinite thick and can dissipate the sound energy, the sound absorption of the structure can be large. However, large porosity does not necessarily result in small flow resistivity of the structures, and small flow resistivity of the structures does not necessarily result in large sound absorption. On the other hand, if the porosity of the structure is small, the flow resistivity of the structures might be large, the characteristic impedance and propagation constant of the structure might differ from those in the medium significantly, so the sound absorption of the structure might be small. The methods of porosity measurement have been discussed in Chapter "Acoustical Test Methods for Nonwoven Fabrics".

1.4.4 Density

The density of a material indicates the mass concentration of the material, which is measured as mass per unit volume. In the case of porous structure, the bulk density plays an important role in acoustic absorption. As shown in Eq. (1.1.11), the porous material bulk density affects the flow resistivity nonlinearly. Usually, large density results in large flow resistivity if all other parameters of the structure remain the same. Unfortunately, the relationship between the flow resistance and sound absorption is very complicated. As shown in Eqs. (1.2.1)–(1.2.2), when the structure density is large, the flow resistivity of the structures is large, the characteristic impedance and propagation constant of the structure might differ from those in the medium significantly, and this results in small sound absorption of the structure; however, small density does not necessarily result in small flow resistivity of the structures, and small flow resistivity of the structures does not necessarily result in large sound absorption. For example, it has been found experimentally that the normal incidence sound absorption coefficient of a bamboo fibre material increases with its density [28].

1.4.5 Tortuosity

Tortuosity refers to the sinuosity and interconnectivity of void spaces in a porous structure. It describes the difference between the actual distance travelled by a fluid in a porous structure and the macroscopic travel distance. Tortuosity has several definitions depending on the field of study. Generally, tortuosity indicates the diffusion in porous media such as fibrous structures. Geometric tortuosity expressed as the ratio of the shortest path of interconnected points in pore fluid space to the straight distance between these points. In the case of acoustic materials tortuosity describes the elongation of the pathway through the pores. The methods used to measure tortuosity are described in Chapter “Acoustical Test Methods for Nonwoven Fabrics”.

1.4.6 Compression

Textile structures used for acoustic absorption are compressible, which can alter the sound absorption property. When the textile structure or a porous material is compressed, the porosity and thickness values decrease, but the density increases. As shown in Eqs. (1.2.1)–(1.2.2), large structure density results in large flow resistivity of the structures if the other parameters of the structure remain the same. With a large flow resistivity, the characteristic impedance and propagation constant of the structure might differ from those in the medium significantly, and this results in low sound absorption of the structure; however, compression does not necessarily result in small sound absorption. Usually, compression increases sound absorption in the low frequency range at the cost of reducing sound absorption in the high frequency range. The methods used to measure compression are described in Chapter “Acoustical Test Methods for Nonwoven Fabrics”.

1.4.7 Airflow Resistance

The acoustic absorption of a porous material is greatly influenced by the airflow resistance. The interlacement of threads in a woven structure or intermeshing of fibres in a nonwoven structure provides a resistance to the passage of air, which is an intrinsic property of the textiles. The airflow resistance is measured in terms of the resistance provided by the material per unit thickness. The details of flow resistivity have been described in Sect. 1.1.4.

1.4.8 Surface Impedance

The surface impendence of a structure is the short name of the specific boundary impedance of the structure, which is defined as the specific acoustic impedance that the boundary surface presents to an adjacent sound field, and its value is equal to the complex ratio of the acoustic surface pressure to the normal velocity of the fluid at the surface with the positive direction into the boundary [2]. As shown in Eq. (1.2.3), the absorption coefficient of a structure can be calculated from the ratio of the specific acoustic impedance of the layer surface to the characteristic impedance of the medium.

1.4.9 Position of Sound Absorbers

The effectiveness of sound absorbing materials depends on their relative placement. For acoustic materials used in buildings with various sizes and shapes, it is imperative to select the best locations for positioning of the acoustic material. Usually, the acoustic absorbers are placed along the edges or surfaces of the room for architecture or cost reasons; however, these positions might not be best locations for the porous material absorbers because the particle velocity of sound on the surfaces is usually very small. The details have been described in Sect. 1.1.2.

1.5 Applications of Sound Absorbers

Sound absorbers are commonly used in rooms to control sound field properties such as sound pressure level and reverberation time. To determine whether it is necessary to treat surfaces in a room with acoustically absorbing materials, the first step is to determine whether the reverberant sound field dominates the direct sound field at the point where it is desired to reduce the overall sound pressure level because treating reflecting surfaces with acoustically absorbing material can only affect the reverberant sound field [5]. At locations close to the sound source (for example, a machine operator's position) it is likely that the direct field of the sound source dominates, so it might be little improvement in treating a factory with sound absorbing material to protect operators from noise levels produced by their own machines. However, if an operator is affected by noise produced by other machines some distance away then treatment may be appropriate.

At any location in a room, the sound field is a combination of the direct field radiated by the source and the reverberant field. The acoustic energy due to the direct field at a location with a distance of r and in a direction θ from the source can be expressed by [3],

$$p_{\mathrm{d}}^2 = \frac{W\rho_0 c_0 D_\theta}{4\pi r^2} \tag{1.5.1}$$

where ρ_0 is the air density, c_0 is the speed of sound, W is the sound source power, D_θ is the directivity factor of the sound source. The acoustic energy due to the reverberant field can be expressed by [3],

$$p_{\mathrm{r}}^2 = \frac{4\rho_0 c_0 W(1-\alpha)}{S\alpha} \tag{1.5.2}$$

where S is the room surface area, and α is the total mean absorption coefficient (including air absorption) of the room. Thus the total acoustic energy measured at the location can be obtained by summation of the direct acoustic energy and the reverberant acoustic energy as

$$p^2 = \frac{W\rho_0 c_0 D_\theta}{4\pi r^2} + \frac{4\rho_0 c_0 W(1-\alpha)}{S\alpha} \tag{1.5.3}$$

It is obvious that the sound absorption only affects the reverberant acoustic energy. Written in the form of sound pressure level in dB, the above equation becomes

$$L_{\mathrm{p}} = L_{\mathrm{w}} + 10\log_{10}\left[\frac{D_\theta}{4\pi r^2} + \frac{4(1-\alpha)}{S\alpha}\right] + 10\log_{10}\frac{\rho_0 c_0}{400} \tag{1.5.4}$$

where L_{w} is the sound power level of the sound source.

The above equations can be used for designing sound pressure level control system in a room when the reverberant sound field dominates the direct field. Under this condition, the sound pressure level can be reduced or increased by adjusting the absorption added to the room. Assume the original total mean absorption coefficient of the room is α_0, but is changed to α_1 after control. The sound pressure level change due to the reverberant field can then be expressed as

$$\Delta L_{\mathrm{p}} = 10\log_{10}\frac{\alpha_1(1-\alpha_0)}{\alpha_0(1-\alpha_1)} \tag{1.5.5}$$

The reverberation time in a room can also be changed by adjusting its absorption as shown in the Sabine equation below [3],

$$T_{60} = \frac{55.25\,V}{c_0 S\alpha} \tag{1.5.6}$$

where S is the room surface area, V is the volume of the room, c_0 is the speed of sound and α is the total mean absorption coefficient of the room. The reverberation time T_{60} in a room can be increased by reducing the absorption in the room so the sound does not feel too dry or the reverberation time T_{60} in a room can be reduced by adding more sound absorbers in the room so the speech can be more clearly heard.

For example, a room of 8 m × 6 m × 3 m has a reverberation time of 0.64 s, and the sound source is located in the center of the room (not on ground) with a sound power level of 70 dB. It will be illustrated below, whether it is possible to reduce the reverberation sound level to 45 dB by putting some sound absorption material on the walls. The first step is to calculate the volume of the room and the room surface area, which are V = 144 m^3 and S = 180 m^2, respectively. Then the total mean absorption coefficient of the room can be calculated with the Sabine equation (1.5.6) as

$$\alpha = \frac{55.25V}{c_0 S}\frac{1}{T_{60}} = 0.2 \tag{1.5.7}$$

The sound pressure level of the direct sound 1 m away from the source, the reverberation sound, and the total sound 1 m away from the source can be calculated with Eq. (1.5.4).

$$L_{\text{pd}} = L_{\text{w}} + 10\log_{10}\frac{D_\theta}{4\pi r^2} + 10\log_{10}\frac{\rho_0 c_0}{400} = 59.2\ \text{dB} \tag{1.5.8}$$

$$L_{\text{pr}} = L_{\text{w}} + 10\log_{10}\frac{4(1-\alpha)}{S\alpha} + 10\log_{10}\frac{\rho_0 c_0}{400} = 59.6\ \text{dB} \tag{1.5.9}$$

$$L_{\text{p}} = 70 + 10\log_{10}\left[\frac{1}{4\pi} + \frac{4(1-0.2)}{180\times 0.2}\right] + 10\log_{10}\frac{415}{400} = 62.4\ \text{dB} \tag{1.5.10}$$

The reverberation sound level is 59.6 dB now and 14.6 dB is needed to be reduced to reach 45 dB. Substituting this number to Eq. (1.5.5), it has

$$10\log_{10}\frac{\alpha_1(1-\alpha_0)}{\alpha_0(1-\alpha_1)} = 14.6 \tag{1.5.11}$$

Substitute α_0 = 0.2 to the equation and solve the equation, α_1 = 0.88 can be obtained. Therefore, it is feasible to reduce the reverberation sound level to 45 dB by laying sound absorption material on the walls until the total mean absorption coefficient of the room reaches 0.88. However, this sound pressure level reduction is only significant at the locations where the reverberant sound dominates. As can be observed from Eqs. (1.5.8) and (1.5.10), the maximal total sound pressure level reduction at 1 m away from the source is approximately 3 dB because of the dominated direct sound.

1.6 Specific Discussions on Acoustic Textiles

A textile is a flexible woven material consisting of a network of natural or artificial fibres often referred to as thread, which is produced by spinning raw fibres of wool, flax, cotton or other material to produce long strands (https://en.wikipedia.org/wiki/Textile). Textiles are formed by weaving, knitting, crocheting, knotting or felting. The acoustic properties of textiles can be characterized in three categories as propagation, absorption, and scattering and those properties can be represented by flow resistance, transmission loss, absorption coefficient and scattering coefficient. The propagation and absorption properties of a textile depend on its flow resistance. Acoustic textiles are kinds of porous absorbers or resonant absorbers, so their acoustic properties can be described or predicted with what introduced in sections from 1.1 to 1.3.

Although theories for both membrane absorbers and micro-perforated plate absorbers have been well established, there is little method developed for their combination such as a fibreglass textile. Kang and Fuchs presented a theoretical method for predicting the absorption of such a structure [29]. The idea was to treat an open weave textile or a micro-perforated membrane as a parallel connection of the membrane and apertures. The predictions for both normal and random incidence showed reasonable agreement with measurements. It is found that the absorption coefficient of a fibreglass textile mounted at 100 mm from a rigid wall can exceed 0.4 over 3–4 octaves, and this range can extend to 4–5 octaves with two layers of the material over the same total air space of 100 mm.

The theoretical investigation into the sound propagation through flexible porous media is of prime importance for evaluating the noise absorption capacities of foam materials or textiles, such as woven fabrics or nonwoven fibrewebs [30]. The Zwikker and Kosten model [31] for sound propagation through porous flexible media has been used for numerical calculations of some intrinsic characteristics of nonwoven fibrewebs so that it can yield the highest sound absorption coefficients in the audible frequency range. These results can serve as guidelines for the optimal design of acoustic elements made of textile assemblies.

Micro-fibre fabric has fine fibres and a high surface area and it has been used in such applications as wipers, thermal insulator, filters or breathable layers, as well as for sound absorption. The feasibility of using micro-fibre fabrics as sound absorbent materials has been investigated [32]. The test results of five micro-fibre fabrics and one regular fibre fabric showed that the micro-fibre fabrics' sound absorption is superior to that of conventional fabric with the same thickness or weight, and the micro-fibre fabrics' structure was found to be important for controlling sound absorption according to sound frequency. Furthermore, fabric density was found to have more effect than fabric thickness or weight on sound absorption, and the Noise Reduction Coefficient increases to its highest value at a fabric density of about 0.14 g/cm^3, and it decreases thereafter.

A review paper was published to describe how the physical prosperities of materials like fibre type, fibre size, material thickness, density, airflow resistance and porosity can change the absorption behaviour [27]. The effect of surface impedance, placement of sound absorptive and compression, on sound absorption behaviour of materials was also considered. The results showed the relationship among the sound absorption and airflow resistance, material thickness, air gap and attachment film, and it was found that higher airflow resistance usually gives better sound absorption values under a certain value, the creation of air gap behind the absorptive material increases sound absorption coefficient values, as discussed in Sects. 1.2 and 1.3.

Nonwoven fabrics can be used in car interior components (head liners, doors, side panels and trunk liners) to prevent noise from reaching the passenger compartment and so achieving comfort in the car interior [33]. Two kinds of fibres (polyester and hollow polyester fibres, both six denier) were used to produce three different fabrics (100 % polyester fibres, 75 % polyester/25 % hollow polyester fibres and 55 % polyester/45 % hollow polyester fibres) with four fabric weights (300, 400, 500 and 600 g/m^2). It was found that the samples produced with high percentage of hollow fibres had high sound absorption, whereas samples produced with 100 % polyester fibres had the lowest rates. It was also found that there are direct relationship between weight per m^2 and sound absorption efficiency. The sample produced with 55 % polyester/45 % hollow polyester fibres and 600 g/m^2 has the best absorption.

The sound absorption properties for recycled fibrous materials have been investigated, which include natural fibres, synthetic fibres and agricultural lignocellulosic fibres [27]. The results indicated that nonwoven samples had high sound absorption coefficients at high frequencies (2000–6300 Hz), low sound absorption coefficients at low frequencies (100–400 Hz) and better sound absorption coefficients at mid (500–1600 Hz) frequencies, and could be improved by increasing the thickness of nonwovens. The rice straw and sawdust composite samples have low sound absorption at low and mid frequencies. The sound absorbing performance can be improved by adding perforation of 6 % for the tested sample, increasing the thickness of nonwoven samples and adding air spaces behind the tested composite systems.

1.7 Conclusions and Future Directions

As shown in this chapter, sound absorbers can be made from porous materials or resonant structures, and the main mechanisms are acoustic impedance matching on the absorbers' boundary and acoustic energy dissipation inside the absorbers. If the acoustics impedance at the boundary of the sound absorbers does not match that of the medium where the incident sound propagates, some or most of the sound will be reflected back to the medium. Under this situation, no matter how efficient the absorption mechanism inside the absorbers is, the reflected acoustic energy cannot be dissipated, so the sound absorption coefficient cannot be large. When the acoustics impedance at the boundary of the sound absorbers matches that of the medium, the sound will not be reflected back, and this makes it possible for dissipating acoustic energy in the absorbers. To maximize the acoustic energy dissipation inside the absorbers, various physical effects can be used, such as viscous along the boundaries, heat exchanges, mechanical vibration, magnetic and electrical damping.

Even with all the progress and so many kinds of sound absorbers, there are still many aspects that can be improved for better sound absorption. One of the challenges is to reduce the size of the absorbers for low frequency absorption. For example, a micro-perforated panel needs to be backed by a 0.92 m deep cavity to have good absorption for 100 Hz sound because the wavelength of the 100 Hz is about 3.4 m in air. An integrated mechanical and electrical sound absorber based on Micro-Perforated Panels and Shunted Loudspeakers (MPPSL) has been proposed to reduce the size of the absorber [34]. It is expected a 0.1 m thick MPPSL should be able to have better sound absorption performance than the micro-perforated panel backed by a 0.92 m deep cavity for 100 Hz sound.

The idea of the MPPSL is to integrate mechanical and electrical components into acoustics absorbers to reduce the size of devices for low frequency sound control. Because the wavelength of low frequency sound is quite long, the size of normal Micro-Perforated Panels cannot be reduced significantly if only the acoustical energy dissipation mechanism is engaged. Mechanical energy dissipation components such as thin panels are useful; however, they are not as versatile and powerful as electrical energy dissipation components. It is the powerful energy storing and dissipating capability and design flexibility of the electrical components that make it possible to use compact devices to control long wavelength sound.

Another challenge for future sound absorbers is smart absorbers that can change its absorption coefficient under different situations. Active absorbers might be a solution to the problem, which uses microphones and loudspeakers to interact with the original sound field with active control technologies [4]. Active noise control is a method for reducing existing noise via the introduction of controllable secondary sources to affect generation, radiation, transmission and reception of the original primary noise source. It can provide better solutions to low frequency noise problems than current passive noise control methods when there are weight and volume constraints. As an active sound absorber, the system can adjust the sound absorption coefficient according to the need, and it offers the possibility of bass absorption or diffuse reflection from relatively shallow surface as well as the possibility of variable acoustics.

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