4 Acoustics and sound absorption issues applied in textile problems

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4.1 Sound and noise.

Although everyone is familiar with sound, since hearing is one of the traditionally recognized five human senses, it is difficult for a non-specialist to give a definition of sound. The common answer is that "sound is anything can be heard". However, a more appropriate definition could be the following:

"*Sound* is said to exist if a disturbance propagated through an elastic material causes an alternation in pressure or a displacement of the particles of the material which can be detected by a person or by an instrument." (Beranek 1986). From the physics point of view, the audible noise is not different from sound. There is only a psychological definition of *noise*: it is a disagreeable or undesired sound. Clearly, this definition is subjective. Therefore, what is sound to somebody can be noise to another person.

Confining the study to sound propagating in fluids, e.g., in the atmospheric air, sound can be described as a spatiotemporal variation of pressure around the static pressure of the fluid, e.g., the atmospheric pressure, as shown in Fig. 4.1. This pressure variation is called *acoustic pressure*. Acoustic pressure is denoted by the lower case letter p, as opposing to the static pressure denoted by the capital letter P. Both of them are measured in Pascals (Pa), $[P]=[p]=1\text{Pa}=1\text{Nt/m}^2$.

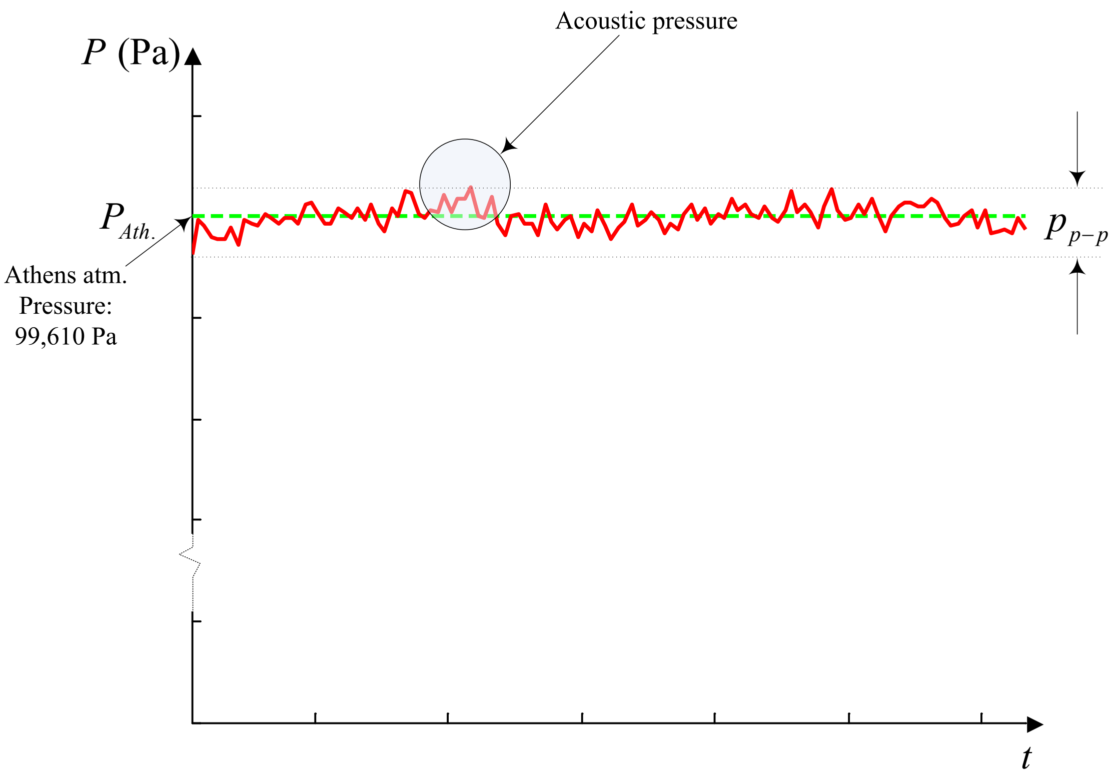

Fig. 4.1: The notion of acoustic pressure vs. static pressure, for example at the city of Athens (Greece). Only the temporal variation is presented here, since the pressure is monitored at a specific location (spatial point).

Acoustic pressure can be described as a sequence of alternating compressions and rarefactions of the particles of the fluid, as shown in Fig. 4.2 for a sinusoidal variation example. In the case of the (ambient) air example, the air particles are disturbed from their thermal random movement under the change of pressure. Therefore, each air particle can be considered to be displaced from its original position due to acoustic pressure. Then, the elastic forces among the air particles tend to restore it to its original position. Because of the inertia of the particle, it overshoots the resting position, bringing into play elastic forces in the opposite direction, and so on. However, the interaction between the specific particle and its neighboring particles disturbs their movement too, propagating the disturbance in space, without actually moving particles from their resting positions.

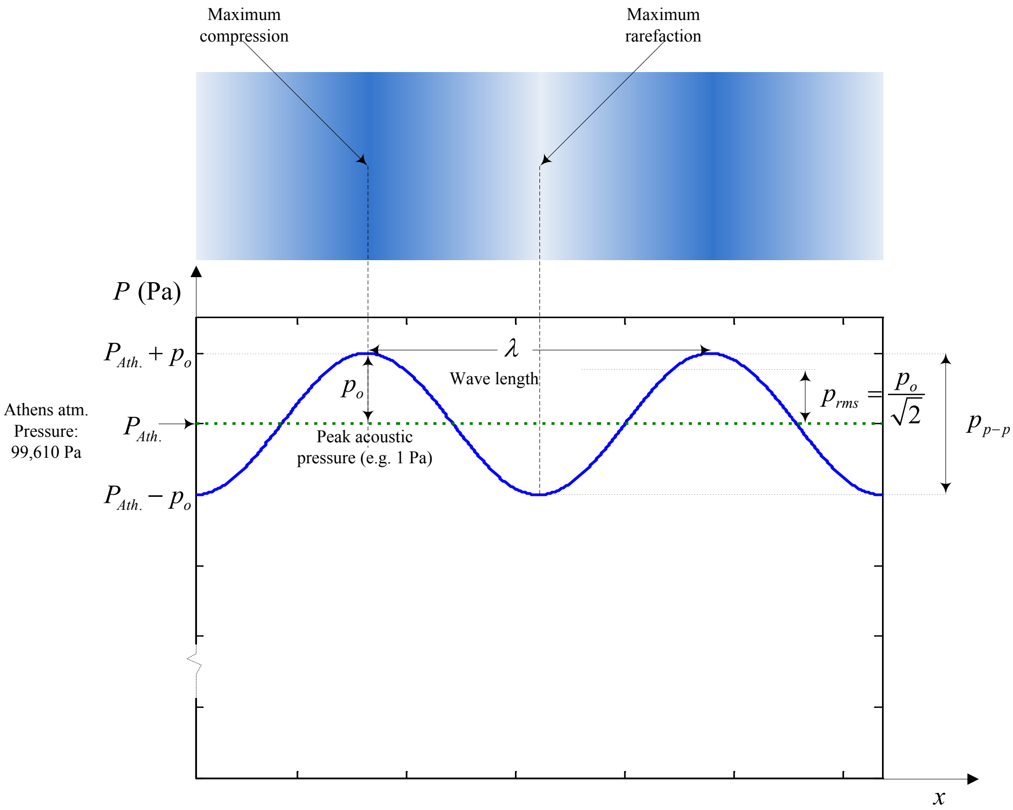

Fig. 4.2: Alternating compressions and rarefactions of the particles of a fluid around its static pressure (upper part), lead to acoustic pressure variations. On this, sinusoidal variation, example some of the characteristics of an acoustic wave are depicted.

It can be proved from basic principles (conservation of mass, 2nd law of Newton and compressibility) that sound as a propagating mechanical spatiotemporal disturbance satisfies the wave equation. The simplest wave case is that of *harmonic plane waves*, which propagate pressure disturbance along one dimension, say x, while being constant in the other two directions. In this case, sound satisfies the one-dimensional wave equation

$$\frac{\partial^2 p}{\partial t^2} = c^2 \frac{\partial^2 p}{\partial x^2}, \tag{4.1}$$

where c is the velocity of acoustic waves, known as *speed of sound*, which is characteristic of a medium for a specific temperature.

Therefore, in a more informative way: *sound* is defined as the mechanical disturbance that propagates with a certain speed in a medium that can develop internal forces (e.g., elastic, internal friction) and has such a character that can excite the hearing mechanism and cause the auditory sense.

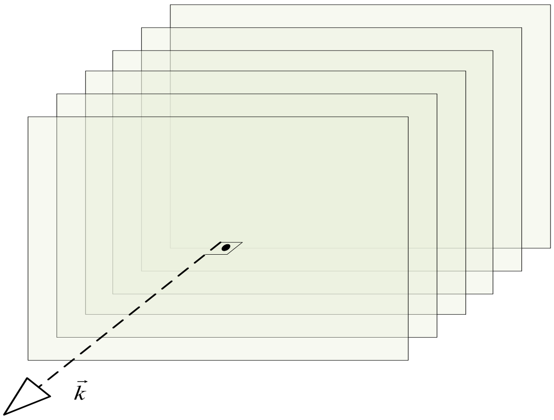

Fig. 4.3: Plane waves propagate along one dimension. Their wavefronts are infinite parallel planes normal to the direction of propagation.

Plane waves are depicted as in Fig. 4.3, by sequential infinite parallel planes perpendicular to the direction of propagation. These planes represent the wavefronts (surfaces of constant phase). Although it is not possible in practice to have a true (infinite extending) plane wave, many waves can be considered approximately plane waves in a confined region of space. The plane waves offer a simplified mathematical framework to study many basic problems in acoustics; they are of outmost importance for a number of applications as for example for the acoustic characterization of materials (see section 4.4).

It is finally noted that sound propagates in the form of *longitudinal waves*, i.e., particle motion is collinear to sound propagation direction, in liquids, plasma, and gaseous media such as air; in contrary to other waves like the mechanical waves propagating on a string or the electromagnetic waves which are *transversal waves*, i.e., the vibration is perpendicular to the direction of propagation. However, in solids sound is propagating through longitudinal, transversal and surface waves.

4.2 Sound measurement.

Sound measurement refers to the determination of basic characteristics of sound. Some of them are the following:

Period, T, is the time taken for one full cycle of a harmonic wave to pass a fixed point, $[T] = 1\text{s}$. It is also referred to as cycle time.

Frequency, f, is the number of pressure variation cycles of a harmonic wave in a medium per second, and is expressed in Hertz (Hz), $[f] = 1\text{Hz} = 1\text{s}^{-1}$. It is the reciprocal of the period of vibration, $f = 1/T$.

Angular frequency, $\omega = 2\pi f$, is defined as the rate of change in the phase of a sinusoidal waveform, $[\omega] = 1\text{rad/s}$.

Wavelength, λ, is the distance travelled by the wavefront of a pressure plane wave during one cycle, i.e., during one temporal period, $[\lambda] = 1\text{m}$. Alternatively, wavelength is called spatial period. In a harmonic wave it corresponds to the distance between two successive maximums or minimums of the acoustic pressure (Fig. 4.2).

By analogy with temporal angular frequency, ω, spatial angular frequency is given by $2\pi/\lambda$. This spatial frequency is termed angular *wavenumber* or just *wavenumber* and denoted by k, $k = 2\pi/\lambda$. Therefore, wavenumber may be interpreted as the phase change per unit distance in a pure travelling wave. Wavenumber is the modulus of the *wavevector*, $\vec{k} = k \cdot \vec{n}$, which has the direction of propagation, $\vec{n}$, the direction normal to wavefront (Fig. 4.3).

Particle velocity (or flow per unit area), $\vec{u}$, is the velocity of the elementary quantities, e.g. air particles, of the propagation medium which vibrating around a resting position contribute to the development of a sound (propagating mechanical disturbance). Note that particle velocity (vector) does not relate to the average speed (scalar) of the associated molecules; the square of the latter is characterized by the local temperature of the fluid and it is a measure of the average molecular kinetic energy.

Volume velocity (acoustic volume flow), U, due to a sound wave is the flow rate of the propagation medium through a surface of area, $\vec{S}$, normal to flow direction. It is $U = \vec{S} \cdot \vec{u}$.

Speed of sound, c, is the velocity of the propagating mechanical disturbance in the elastic medium. The speed of sound in ambient air is approximately given as

$$c \approx 331.3 + 0.606 \cdot \Theta , \tag{4.2}$$

where Θ is the temperature in °Celsius.

Phase velocity, $\vec{u}_{ph}$, of a traveling harmonic wave is the velocity of a point of constant phase towards the direction of wavevector. In an isotropic acoustic medium which does not present the phenomenon of acoustic dispersion the phase velocity coincides with the speed of sound.

The speed of sound, the frequency, and the wavelength are related by the following equation:

$$c = \lambda f , \tag{4.3}$$

therefore, the wavenumber may successively be expressed as

$$k = \frac{2\pi}{\lambda} = \frac{2\pi f}{c} = \frac{\omega}{c} . \tag{4.4}$$

Acoustic pressure is of course a central characteristic of sound. However, the instantaneous acoustic pressure, $p(t)$, is seldom the measured quantity, since sound power related quantities are usually pursuit. Therefore the *effective acoustic pressure*, p_{eff}, or *root-mean-square* (rms) *acoustic pressure*, p_{rms} is used instead, since p_{rms}^2 is proportional to energy or power related physical quantities,

$$p_{eff} = p_{rms} = \sqrt{\lim_{T\to\infty} \frac{1}{T} \int_0^T p^2(t) dt} . \tag{4.5}$$

If p_o is the peak value of the acoustic pressure (half of the peak-to-peak, p_{p-p}, value) for a harmonic sound wave, then the rms acoustic pressure (Fig. 4.2) is given by

$$p_{rms} = p_o / \sqrt{2} . \tag{4.6}$$

Sound "strength" (how loud it is) is usually expressed in decibels (dB). Although decibel is not expressing a physical quantity, but only a ratio relative to a reference quantity (e.g., the power gain of an amplifier is $G_{dB} = 10 \log\left(Power_{,out} / Power_{,in} \right)$ and reference quantity is the input power, $Power_{,in}$), if the reference quantity is known a priori, then decibels can be used to express physical quantities.

Therefore, *sound pressure level* (*SPL*), L_p, is defined using $p_{ref} = 20\mu\text{Pa}$ as reference quantity by

$$L_p = 10\log\frac{p_{rms}^2}{\left(p_{ref}\right)^2} = 10\log\frac{p_{rms}^2}{\left(20\mu\text{Pa}\right)^2} = 20\log\frac{p_{rms}}{20\mu\text{Pa}}, \tag{4.7}$$

and is measured in dB_{SPL}, i.e., $\left[L_p\right] = 1\text{dB}_{SPL}$; it is noted that $p_{rms} = 1\text{Pa}$, corresponds to $L_p \approx 94\text{dB}_{SPL}$.

Sound intensity, $\vec{I}$, at a given point of an acoustic field in a specific direction is defined as the ratio of the average acoustic power that is transmitted through a unit area perpendicular to the considered direction at the specific point. It describes the amount and direction of flow of acoustic energy at a given position. $[I] = 1\,\text{W}/\text{m}^2$. For plane waves, the modulus of sound intensity is

$$I = \frac{p_{rms}^2}{\rho c}, \tag{4.8}$$

where ρ is the density of the medium and c the speed of sound in the specific medium. Accordingly, considering $I_{ref} = 1\,p\text{W}/\text{m}^2$ as reference quantity for sound intensity, one can define intensity level (IL), L_I as

$$L_I = 10\log\frac{I}{I_{ref}} = 10\log\frac{I}{1p\,\text{W}/\text{m}^2}, \tag{4.9}$$

and is measured in dB_{IL}, i.e., $[L_I] = 1\mathrm{dB}_{IL}$; it is noted that for specific frequency and density combinations $L_I \approx L_p$. For example for $\Theta = 22^{\circ}\mathrm{C}$ their difference is only 0.1dB. However, this fact should not lead to confusing the two levels, since they represent different physical quantities and not simple numerical factors without units.

The general solution of Eq. (4.1) reads

$$p(t) = p_{01} e^{j(\omega t - kx)} + p_{02} e^{j(\omega t + kx)} \tag{4.10}$$

If one is interested in the propagation along $+x$ direction only it becomes

$$p(t) = p_0 e^{j(\omega t - kx)} = p_0 e^{jk(ct - x)}, \tag{4.11}$$

and since only real quantities are observable, the real part of the above solution is

$$p(t) = p_0 \cos(\omega t - kx), \tag{4.12}$$

Using the definitions of ω, f, T, as well as Eqs (4.3) and (4.4), Eq. (4.12) could be rewritten as

$$p(t) = p_0 \cos 2\pi \left(\frac{t}{T} - \frac{x}{\lambda} \right), \quad (4.13a) \qquad \text{or} \qquad p(t) = p_0 \cos \omega \left(t - \frac{x}{c} \right). \quad (4.13b)$$

Interpreting the solution of the plane wave equation we note the following:

- The instantaneous acoustic pressure $p(t)$ is not a function of distance x, but depends only on time, t.
- In any position the sound has the same frequency, f.
- The only effect of distance is the phase delay, $-kx$. The energy levels that the plane acoustic waves carry (which is proportional to p_{rms}^2) remain unaltered with distance from the source.

4.3 Sound reflection, absorption, refraction.

In an acoustic system, the *cause* (or the stimulus) is the acoustic pressure, while the *effect* (or the result) is the movement of the acoustic medium "particles". Therefore, in direct analogy to the electric circuits, where impedance is defined as the ratio of the voltage (stimulus) to the current (result), one can define the *acoustic impedance*, Z, as the radio of the acoustic pressure, p, to the volume velocity (acoustic volume flow), U,

$$Z = \frac{p}{U}. \tag{4.14}$$

Z is frequency dependent and complex. Being also dependent on the area through which the flow is defined, it is not a property characteristic of the acoustic medium.

On the other hand, the *specific acoustic impedance*, z_s, sometimes referred to as unit area acoustic impedance, can be used to characterize an acoustic medium. It is defined as the ratio of the acoustic pressure, p, to the particle velocity (or flow per unit area), u, at a given point

$$z_s = \frac{p}{u}. \tag{4.15}$$

z_s is also complex and frequency dependent in general, $\left[z_s\right] = 1\,kgr/m^2 s = 1\,rayl$. However, the *characteristic acoustic impedance*, i.e., the specific acoustic impedance under the assumption of harmonic progressive plane wave propagation, is real, frequency independent and given by

$$z_c = \rho c, \tag{4.16}$$

where, ρ is the density c and is the speed of sound in the specific medium. Characteristic impedance is one of the *bulk properties* of an acoustic medium.

Bulk properties are characteristic acoustic properties describing the interaction of a material with sound. They are independent of the material dimensions, e.g. thickness and area; therefore for an absorber they are independent of the absorber size. However, for anisotropic materials the bulk properties are a function of direction. Important bulk properties are the characteristic acoustic impedance, z_c, the *characteristic propagation wavenumber*, k_c, or alternatively the *effective density*, ρ_e, and *bulk modulus* K_e. The term *effective* is used to signify that this density refers to the density experienced by the sound wave and *not* the normal density of the medium (the mass by volume ratio). Bulk modulus is reciprocal to compressibility, i.e., it is defined as the ratio of the pressure applied to a material to the resultant fractional change in its volume. For a porous absorber, the effective density and bulk modulus are interrelated to the characteristic impedance and propagation wavenumber by the following equations

$$z_c = \sqrt{K_e \rho_e}, \tag{4.17}$$

and

$$k_c = \omega \sqrt{\rho_e / K_e}, \tag{4.18}$$

Once these bulk properties are known, the sound wave propagation within the corresponding acoustic medium can be predicted.

On the interface between two acoustic media, the specific acoustic impedance is usually referred to as *surface impedance* or *wall impedance*, implying that the specific impedance is estimated on the surface of a porous wall. In general, the impedance seen at the surface of an absorber depends on the porous media bulk properties, the geometry of the absorber and the mounting conditions of the absorber. The surface impedance is generally complex. Its real part (*acoustic resistance*) is related to energy losses, while the imaginary part (*acoustic reactance*) is associated with phase changes or energy storage mechanisms. Therefore, surface acoustic impedance provides more information for the absorbing properties of a material compared to the absorption coefficient (defined immediately following, see Eq. (4.19)).

Suppose two acoustic media 1 and 2, interfacing through a plane surface and characterized by surface impedances z_{s1} and z_{s2}, as depicted in Fig. 4.4.

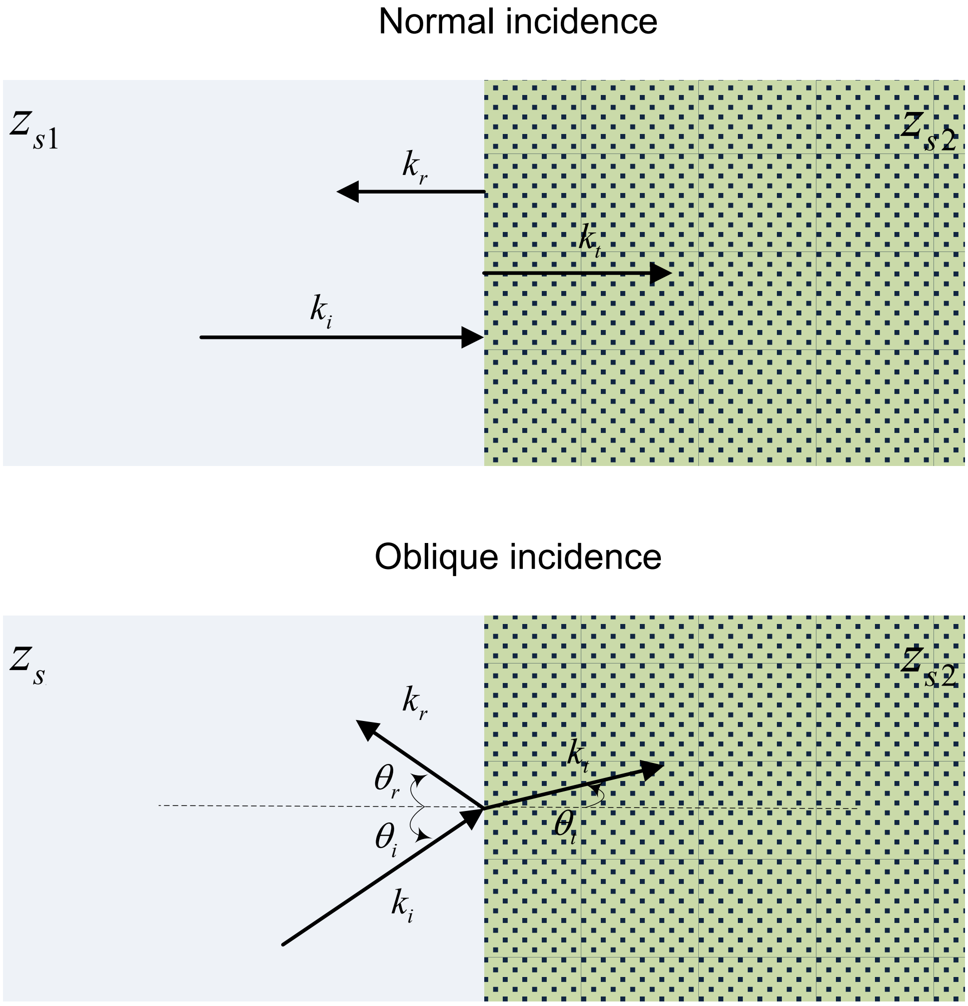

Fig. 4.4: Reflection and refraction in case of normal (upper part) and oblique (lower part) incidence.

A harmonic progressive plane wave incident, $\vec{k}_i$, from medium 1 side to the interface is partly *reflected*, $\vec{k}_r$, back to medium 1, and partly *refracted* (transmitted), $\vec{k}_t$, to medium 2, assuming that interface surface is dimensionless and thus it is not absorbing. If the medium 2 is an absorber itself, then we could refer to the refracted sound as *absorbed* sound.

If the sound intensity moduli and sound energies corresponding to incident, reflected, refracted and absorbed sound are I_i, I_r, I_t, and I_a, and W_i, W_r, W_t, and W_a, respectively, then one can define the *reflection coefficient*, a_r, the *transmission coefficient*, a_t, or τ, and the *absorption coefficient*, a_a, or a, as

$$a_r = \frac{W_r}{W_i} = \frac{I_r}{I_i}, \ \tau = a_t = \frac{W_t}{W_i} = \frac{I_t}{I_i}, \text{ and } a = a_a = \frac{W_a}{W_i} = \frac{I_a}{I_i}, \tag{4.19}$$

For the case of normal incidence (upper part of Fig. 4.4), since from Eqs (4.8) and (4.16) it follows that

$$I = p_{rms}^2 / z_s , \tag{4.20}$$

substituting on the defining Eqs (4.19),

$$a_t = \frac{p_{t,rms}{}^2 z_{s1}}{p_{i,rms}{}^2 z_{s2}} \qquad \text{and} \qquad a_r = \frac{p_{r,rms}{}^2}{p_{i,rms}{}^2}, \tag{4.21}$$

and finally, employing the appropriate boundary conditions, one obtains

$$a_t = \frac{4z_{s1}z_{s2}}{\left(z_{s2}+z_{s1}\right)^2} \qquad \text{and} \qquad a_r = \frac{\left(z_{s2}-z_{s1}\right)^2}{\left(z_{s2}+z_{s1}\right)^2}, \tag{4.22}$$

For the oblique incidence case, since the particle velocity continuation (one of the boundary conditions) at the interface surface, $x = 0$, is expressed as

$$\frac{p_i(0,t)\cos\theta_i}{z_{s1}} - \frac{p_r(0,t)\cos\theta_r}{z_{s1}} = \frac{p_t(0,t)\cos\theta_t}{z_{s2}}, \tag{4.23}$$

one obtains respectively,

$$\frac{p_t(0,t)}{p_i(0,t)} = \frac{z_{s2}\left(\cos\theta_i + \cos\theta_r\right)}{z_{s2}\cos\theta_r + z_{s1}\cos\theta_t}, \quad \text{and} \quad \frac{p_r(0,t)}{p_i(0,t)} = \frac{\left(z_{s2}\cos\theta_i - z_{s1}\cos\theta_t\right)}{\left(z_{s2}\cos\theta_r + z_{s1}\cos\theta_t\right)}, \tag{4.24}$$

and finally

$$a_t = \frac{4z_{s1}z_{s2}\cos\theta_i\cos\theta_r}{\left(z_{s2}\cos\theta_r + z_{s1}\cos\theta_t\right)^2}, \quad \text{and} \quad a_r = \frac{\left(z_{s2}\cos\theta_i - z_{s1}\cos\theta_t\right)^2}{\left(z_{s2}\cos\theta_r + z_{s1}\cos\theta_t\right)^2}, \tag{4.25}$$

Since, on the interface surface no medium change happens, the tangential components of the wavevectors (Fig. 4.4) are equal: $k_i \sin\theta_i = k_r \sin\theta_r = k_t \sin\theta_t$. This means that since the incident and reflected waves are propagated in the same medium, the angle of reflection (angle between the direction in which the reflected wave is traveling and the direction perpendicular to the interface surface) is equal to the incidence angle (angle between the direction in which the incident wave is traveling and the direction perpendicular to the interface surface), i.e.,

$$\theta_i = \theta_r, \tag{4.26}$$

while for the transmitted and incident waves holds sequentially, using $c = \lambda f$ and knowing that frequency does not change when the medium changes, $\sin\theta_i/\lambda_i = \sin\theta_r/\lambda_r$, or

$$\frac{\sin\theta_t}{\sin\theta_i} = \frac{c_2}{c_1}. \tag{4.27}$$

Eqs (4.26) and (4.27) are the well known refraction laws of Snell-Descartes. Quite often Eq. (4.27) is called the *Snell's law*.

It is noted here that the ratio of the reflected by the incident pressure appearing in the right equation of Eq. (4.24), is usually referred to as the *pressure reflection coefficient* (or the *reflection factor*), R, and includes both amplitude and phase information from the reflected sound wave, p_r, and the incident sound wave, p_i,

$$R = \frac{p_r}{p_i}. \tag{4.28}$$

Moreover, for most absorbents, the speed of sound in the absorbent medium is much less than that the speed of sound in the air. Therefore, in the case that the incidence medium is the air, the angle of propagation in the propagation medium is smaller than in the air ($\theta_t << \theta_i$) and it is usually assumed that the propagation is normal to the interface surface, i.e. $\theta_t \to 0$.

Consequently, if in the case described by the lower part of Fig. 4.4 the left medium (medium 1) is the air, its surface impedance is the characteristic acoustic impedance and $z_{s1} = \rho_o c_o$, with ρ_o, c_o being the density of the (ambient) air and the speed of sound in the air. In the specific case, the right equations appearing in Eqs (4.24) and (4.25) may be rewritten as

$$\frac{p_r(0,t)}{p_i(0,t)} = \frac{\left(z_{s2}\cos\theta_i - \rho_o c_o \cos\theta_t\right)}{\left(z_{s2}\cos\theta_r + \rho_o c_o \cos\theta_t\right)}, \qquad \text{and} \qquad a_r = \frac{\left(z_{s2}\cos\theta_i - \rho_o c_o \cos\theta_t\right)^2}{\left(z_{s2}\cos\theta_r + \rho_o c_o \cos\theta_t\right)^2}, \tag{4.29}$$

furthermore, since $\theta_i = \theta_r$ and $\theta_t \to 0$ they become

$$R = \frac{p_r(0,t)}{p_i(0,t)} = \frac{\left(\frac{z_{s2}\cos\theta_i}{\rho_o c_o} - 1\right)}{\left(\frac{z_{s2}\cos\theta_i}{\rho_o c_o} + 1\right)}, \quad \text{and} \quad a_r = \frac{\left(z_{s2}\cos\theta_i - \rho_o c_o\right)^2}{\left(z_{s2}\cos\theta_i + \rho_o c_o\right)^2} = \frac{\left(\frac{z_{s2}\cos\theta_i}{\rho_o c_o} - 1\right)^2}{\left(\frac{z_{s2}\cos\theta_i}{\rho_o c_o} + 1\right)^2}, \tag{4.30}$$

Therefore,

$$\frac{z_{s2}}{\rho_o c_o}\cos\theta_i = \frac{(1+R)}{(1-R)} \tag{4.31}$$

and

$$a_r = |R|^2 \tag{4.32}$$

Remember that the interface is considered not to be absorbing, therefore $W_a = 0$ and, thus (energy conservation), $W_i = W_r + W_t$, i.e., $a_t + a_r = 1$. However, if medium 2 is an absorber, then the transmitted power is actually absorbed, so one could for convenience call the transmission coefficient, a_t, absorption coefficient, a. Consequently, in this case, the absorption coefficient, a, can be expressed as,

$$a = 1 - a_r = 1 - |R|^2 \tag{4.33}$$

Under the hypothesis of harmonic progressive plane wave propagation, the surface impedance is not frequency dependent; however it can be said to depend on the angle of incidence to the surface, in the sense that only the normal component of particle velocity is effectively contributing to the boundary conditions. If the wall material is locally reacting, the *effective surface impedance* at any angle is related to the surface impedance for normal incidence, the *normal surface impedance*, z_n by the formula

$$z_s(\theta_i) = \frac{z_n}{\cos\theta_i}, \tag{4.34}$$

since $z_s(\theta_i) = p/u_{eff} = p/u \cdot \cos\theta_i = z_n/\cos\theta_i$. This formula can be used to transform normal incidence results to oblique ones by simple substitution of z_n by $z_s(\theta_i)$, using Eq. (4.34). Although the aforementioned analysis regarding the reflection and the transmission coefficients for oblique incidence has been presented differently, one can obtain the same results (Eq. (4.25)), using the above transformation on the normal incidence results (Eq. (4.22)). Therefore, the determination of normal surface impedance is usually pursuit. Nevertheless, in order to characterize, or design, an acoustic medium the characteristic impedance is needed.

Suppose that a porous absorbing medium of thickness d and infinite (very large) area is mounted on a rigid wall as illustrated in Fig. 4.5.

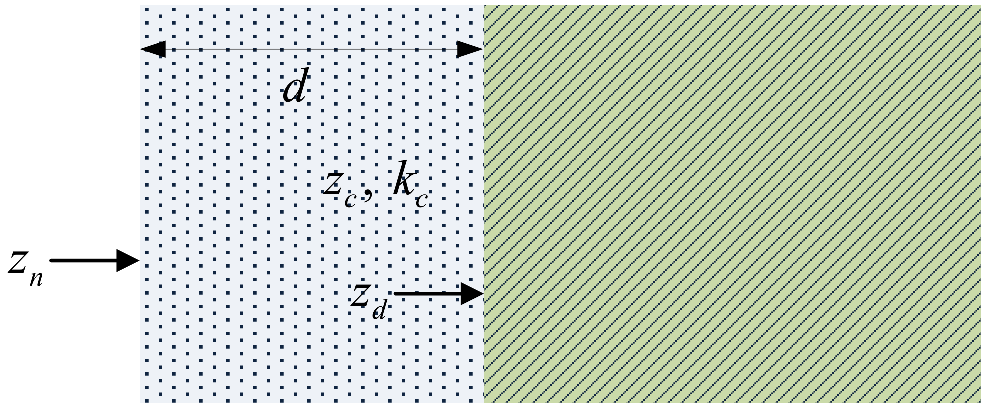

Fig. 4.5: Porous absorbing medium of thickness d mounted on a rigid wall.

The normal surface impedances at the outer interface surface of the absorbing medium (towards the air), z_n, and the inner interface surface (towards the rigid back), z_d, are interrelated through the formula

$$z_n = z_c \frac{z_d + jz_c \tan(k_c d)}{z_c + jz_d \tan(k_c d)}. \tag{4.35}$$

For $z_d \to \infty$, $z_n \to -jz_c \cot(k_c d)$; therefore, knowing z_n for two different thicknesses (d_1, d_2), the characteristic impedance and the characteristic wavenumber can be determined.

On the other hand, if the characteristic impedance and wavenumber of a material have been determined by other means, they can be converted to the surface impedance and absorption coefficient for a particular thickness of the porous material with known boundary conditions (on one side air and on the other side a rigid backing). First the normal surface impedance, z_n, is determined as

$$z_n = -jz_c \cot(k_c d), \tag{4.36}$$

while, since $\theta_i = 0$, R may be obtained from Eq. (4.31) as

$$R = \frac{\left(\frac{z_n}{\rho_o c_o} - 1\right)}{\left(\frac{z_n}{\rho_o c_o} + 1\right)}, \tag{4.37}$$

from which the absorption coefficient can be determined using Eq. (4.33).

4.4 Sound absorption measurement methods.

Various methods have been proposed for the characterization of the sound absorbing capability of different materials. From the application viewpoint, i.e., in order to use sound absorbing materials, it is usually important to know the random incidence absorption coefficient. On the other hand, from the design viewpoint, i.e., in order to be able to design these materials, a non-statistical, material-oriented parameter, like normal incidence absorption coefficient, seems to be more appropriate. Although both approaches can be found, the second one is far more frequently employed in the literature relevant to textile materials. The reason is that most of textile-oriented scientific articles deal with the study of the influence of textile parameters on the sound absorption properties, targeting to the understanding and modeling of the involved absorption mechanisms.

Normal incidence absorption coefficient is usually measured in the controlled conditions of an impedance tube, or Kundt's tube, where standing sound waves are developed. In order for the impedance tube methods to be applied successfully, the plane wave propagation approximation has to be fulfilled. This poses a number of limitations on the test setup:

- Since, ideally, plane waves propagate without attenuation, the losses into and through the tube should be minimized. This leads to heavy (thick) metal structures.
- The tube should have constant cross-section over the measurement section. They are usually circular.

- The loudspeaker should be a few tube diameters from the first microphone position, in order to ensure that the possible cross modes generated by the loudspeaker have not significant presence. Sometimes, absorbent is placed at the loudspeaker end of the tube.
- The microphone positions should not be too close to the sample so that any evanescent waves generated on reflection have had time to die away.
- The tube diameter has to be smaller than sound half wavelength (specifically $D < 0.586\lambda$), while its length longer than $3\lambda/4$.

4.4.1 Standing wave method

The standing wave method is historically the first proposed which is based on the use of impedance tube. The specimen under test is placed at the one end of the tube, while a sound source generates a single frequency sound at the opposite end. Assuming plane wave propagation conditions inside the tube and the specimen placed at the position $x = 0$ before a rigid termination, see Fig. 4.6, the overall resulting pressure after the interference between the two waves traveling in opposite directions (at steady state) is

$$p = A\left(e^{jkx} + \mathrm{Re}^{-jkx}\right), \tag{4.38}$$

where R, is the pressure reflection coefficient, see Eq. (4.28) and A is a complex constant. The first term of Eq. (4.38) corresponds to the incident while the second to the reflected wave. At the position where they are in phase, a maximum pressure, $p_{\max}$, is developed, Fig. 4.6. In contrary, a minimum pressure, $p_{\min}$, is developed at the position where they are completely out of phase. Since the amplitude of the reflected sound wave, is lower than the incident amplitude, the standing wave ratio, s, is defined as

$$s = \frac{p_{\max}}{p_{\min}} = \frac{|p_i| + |p_r|}{|p_i| - |p_r|} = \frac{1 + |R|}{1 - |R|}, \tag{4.39}$$

Rearranging Eq. (4.39) the magnitude of the reflection coefficient is obtained

$$|R| = \frac{s - 1}{s + 1}, \tag{4.40}$$

from which the absorption coefficient, a, results from Eq. (4.33).

Moving a pressure probe (microphone) along the impedance tube, the maximum and minimum pressure can be measured and therefore the standing wave ratio, and finally the absorption coefficient can be determined. For a specific frequency, with wavelength within tube range, the probe is moved on sound source axis (tube axis) starting from specimen's position towards the source. Although it is possible that a maximum of pressure is first detected as the probe is moving away from specimen, see Fig. 4.6, the first recorded value is always the first pressure minimum, and then the following maximum is also recorded. This minimizes the effect of tube losses, i.e., the specific procedure ensures that the recorded maximum is indeed a maximum and not an intermediate value. The same procedure is repeated for all frequencies of interest.

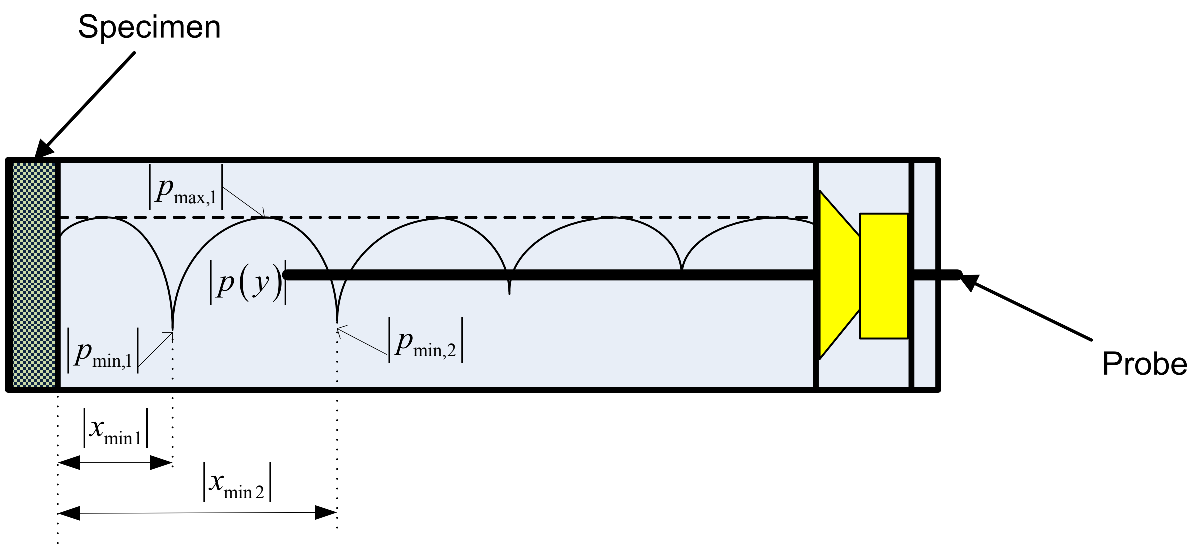

Fig. 4.6: Moving probe setup for the realization of the standing wave method.

Knowing the position of the first minimum, since the incident and reflected waves are there out of phase by exactly π, the phase angle of the pressure reflection coefficient can be calculated as

$$\angle R = 2kx_{\min 1} - \pi . \tag{4.41}$$

It is then possible, using Eq. (4.31), to obtain the normal incidence surface impedance as

$$z_n = \rho_o c_o \frac{1+R}{1-R} . \tag{4.42}$$

One of the key advantages of this method is that no calibration is pre-required for the measurement to be performed, provided that the apparatus is not changed during the measurement. The standard methods ISO 10534-1:1996 (1996) and ASTM C384-04 (2011) are based on the standing wave method.

4.4.2 Transfer function method

In the transfer function method, two pressure probes (microphones) are fixed at the positions $x = x_1$ and $x = x_2$ ($x = 0$ is the position of the specimen), see Fig. 4.7. The pressure transfer function between these two positions is

$$H_{21} = \frac{p(x_2)}{p(x_1)} = \frac{e^{jkx_2} + \mathrm{R}e^{-jkx_2}}{e^{jkx_1} + \mathrm{R}e^{-jkx_1}} \tag{4.43}$$

Solving Eq. (4.43) in regard to the complex pressure reflection coefficient R one gets

$$R = \frac{H_{21}e^{jkx_1} - \mathrm{e}^{jkx_2}}{\mathrm{e}^{-jkx_2} - H_{21}\,\mathrm{e}^{-jkx_1}} = \frac{H_{21} - H_I}{H_R - H_{21}} e^{j2kx_1}, \tag{4.44}$$

where $H_I = e^{jk\Delta x}$, $H_R = e^{-jk\Delta x}$, and $\Delta x = x_2 - x_1$

Using Eq. (4.33) and (4.42) we calculate the absorption coefficient and the normal incidence specific acoustic impedance, respectively.

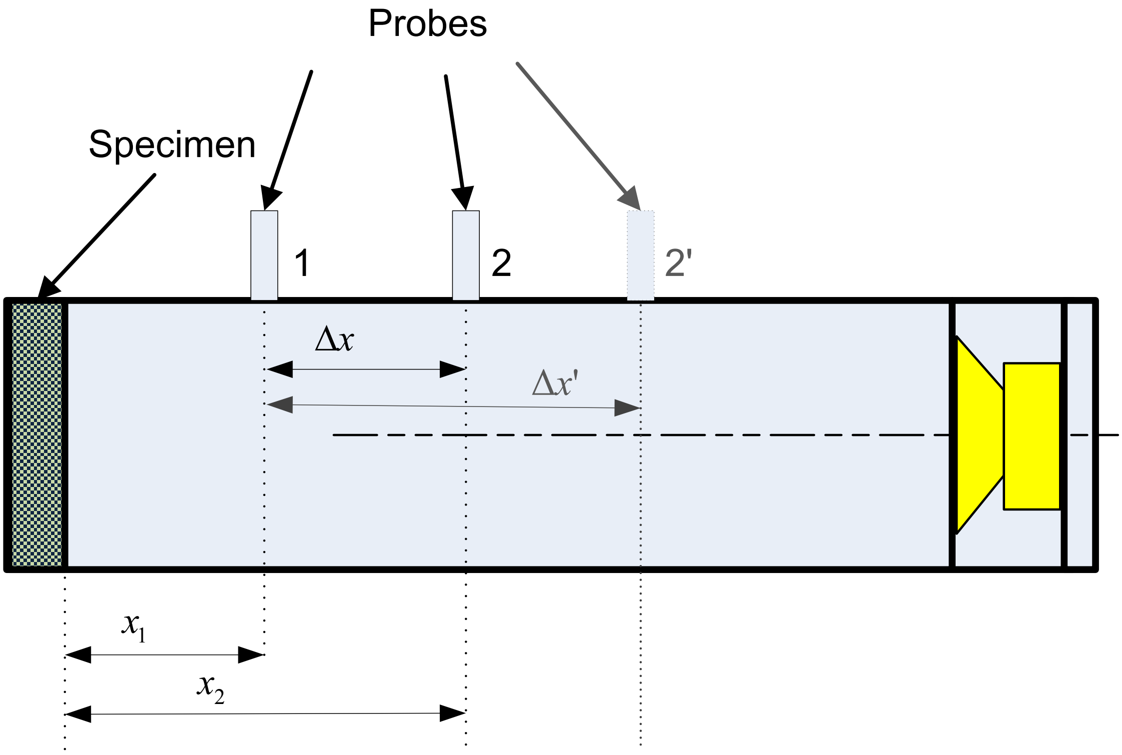

Fig. 4.7: Two microphones setup for the realization of the transfer function method.

In the transfer function method the two microphones used should be carefully positioned, because both a too close and a too distant spacing between them can lead to inaccurate measurements. If they are too close to each other, the pressure difference may be too small to be accurately measured and this leads to a low frequency limit $f_l > c_o/20\Delta x$. On the other hand, if their spacing is too wide there is a possibility to record almost identical pressure values at the microphones, as $\Delta x \to \lambda$. Therefore, an upper frequency limit is posed $f_h < 0.45c_o/\Delta x$. These frequency limitations are usually bypassed using more than two microphones to cover the frequency range of interest. The use of three microphones with appropriate spacing is typical.

The sound source can be reproducing either white noise or a deterministic signal like maximum length sequences (MLS) or log chirp (logarithmically swept sine). However, the first choice is more time consuming and troublesome since it involves an averaging based microphone matching process. Namely, each measurement is followed by a second one with microphones interchanged, to compensate for differences due to the mismatches in their responses.

A comparison between the standing wave method and the transfer function method seems to be in favor of the second, probably due to the accurate positioning of the microphones. The standard methods ISO 10534-2:1998 (1998) and ASTM E1050-10 (2010) are based on the transfer function method.

4.4.3 Transmission loss method

The transmission loss method is a variation on the standard impedance tube using two pairs of microphones. The microphones measure both reflected and transmitted waves to get the normal incident transmission loss, TL_n. The bulk properties characteristic acoustic impedance, z_c, and characteristic propagation wavenumber, k_c, with only one measurement can also be determined by this method. Although there are different variations of this method, the widely accepted variation of Song and Bolton (Song and Bolton 2000; Olivieri et al. 2006) is the one presented here.

As the sample of thickness, d , is placed in the tube, there are four plane acoustic waves with amplitudes, A, B, C , and D, relating to the material sample, as shown in Fig. 4.8. In anechoic conditions, i.e., if $D=0$, it would be $TL_n(f)=10\log_{10}(W_i/W_t)$, or $TL_n(f)=10\log\left(\left|A^{anech.}/C^{anech.}\right|^2\right)$, where W_i and W_t are the airborne sound power incident on the specimen and the sound power transmitted through the specimen and radiated from the other side, respectively. Since different termination conditions would lead to different results, the transmission loss should be expressed in terms of the material under test. The basis for the method lies in the definition of the transfer matrix which is characteristic for an acoustic material.

The matrix relating the forward and backward traveling acoustic waves, $\mathbf{G}$, is defined as

$$\begin{bmatrix} A \\ B \end{bmatrix} = \begin{bmatrix} G_{11} & G_{12} \\ G_{21} & G_{22} \end{bmatrix} \cdot \begin{bmatrix} C \\ D \end{bmatrix}, \tag{4.45}$$

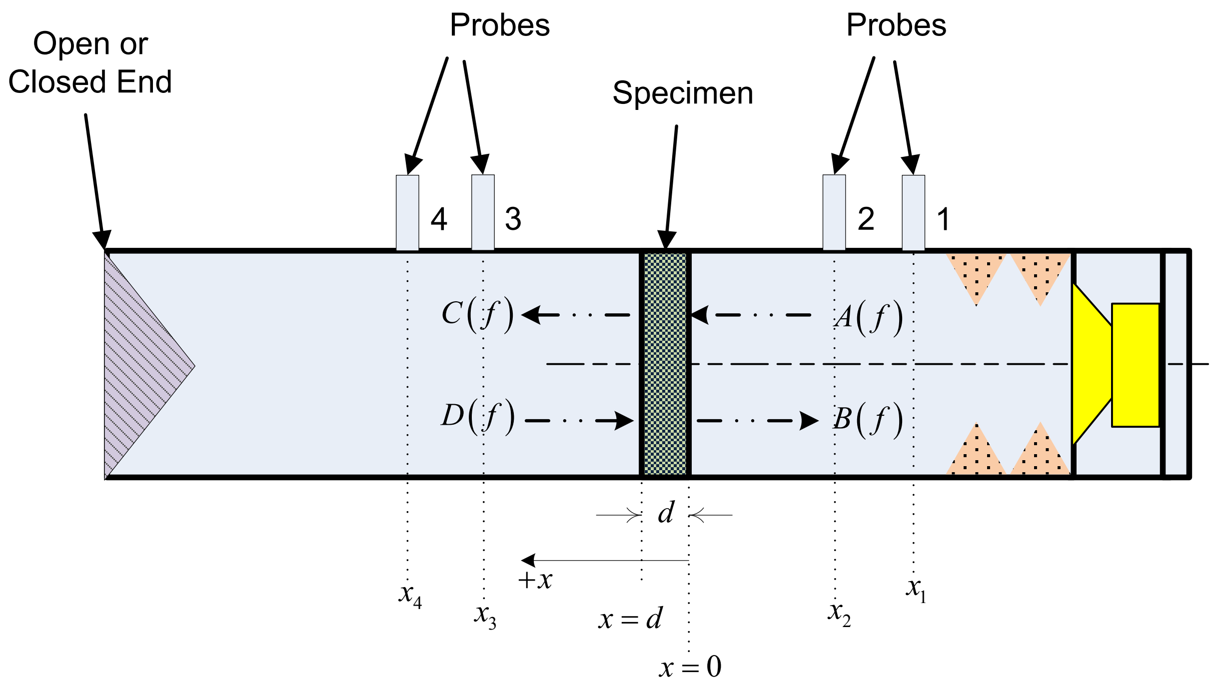

Fig. 4.8: Four microphones setup for the realization of the transmission loss method.

At each position x to the right of the sample, $x \le 0$, the acoustic pressure, $p(x)$, and the modulus of particle velocity, $u(x)$, are

$$p(x) = A(f)e^{-jkx} + B(f)e^{jkx}, \quad \text{and} \quad u(x) = \left(A(f)e^{-jkx} - B(f)e^{jkx}\right)\Big/\rho_o c_o, \tag{4.46}$$

respectively, since pressure is a scalar physical quantity and particle velocity is a vector physical quantity. Similarly, to the left of the sample, $x \ge d$, it is

$$p(x) = C(f)e^{-jkx} + D(f)e^{jkx}, \quad \text{and} \quad u(x) = \left(C(f)e^{-jkx} - D(f)e^{jkx}\right)\Big/\rho_o c_o, \tag{4.47}$$

where ρ_o is the density, c_o is the speed of sound, f is the frequency, and k is the (complex) wave number for the (ambient) air inside the tube. Then, A, B, C, and D, can be expressed as

$$\begin{aligned} A &= \frac{j\left(p(x_1)e^{jkx_2} - p(x_2)e^{jkx_1}\right)}{2\sin k(x_1 - x_2)}, \quad & C &= \frac{j\left(p(x_3)e^{jkx_4} - p(x_4)e^{jkx_3}\right)}{2\sin k(x_3 - x_4)}, \\ B &= \frac{j\left(p(x_2)e^{-jkx_1} - p(x_1)e^{-jkx_2}\right)}{2\sin k(x_1 - x_2)}, \quad & D &= \frac{j\left(p(x_4)e^{-jkx_3} - p(x_3)e^{-jkx_4}\right)}{2\sin k(x_3 - x_4)}. \end{aligned} \tag{4.48}$$

Therefore, the amplitudes A, B, C, and D can be calculated by the four microphone measurements. Once these amplitudes have been determined, the acoustic pressure and particle velocity at $x=0$ and $x=d$, i.e., at both sides of the specimen under test, can be estimated, using Eqs (4.46) and (4.47). Then, *the transfer matrix*, $\mathbf{T}$, can be defined as

$$\begin{bmatrix} p \\ u \end{bmatrix}_{x=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \cdot \begin{bmatrix} p \\ u \end{bmatrix}_{x=d} \tag{4.49}$$

Executing two sets of measurements under different termination conditions at the left end of the tube (Fig. 4.8), one obtains the components of $\mathbf{T}$ solving the linear system of equations

$$\begin{bmatrix} p^r & p^o \\ u^r & u^o \end{bmatrix}_{x=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \cdot \begin{bmatrix} p^r & p^o \\ u^r & u^o \end{bmatrix}_{x=d}, \tag{4.50}$$

where the superscripts o and r indicate the two different termination conditions, i.e., rigid and open end, respectively.

Moreover, the elements of the $\mathbf{G}$ matrix, defined by Eq. (4.45), can be determined from

$$\begin{bmatrix} A^r & A^o \\ A^r & B^o \end{bmatrix}_{x=0} = \begin{bmatrix} G_{11} & G_{12} \\ G_{21} & G_{22} \end{bmatrix} \cdot \begin{bmatrix} C^r & C^o \\ D^r & D^o \end{bmatrix}_{x=d}, \tag{4.51}$$

The element G_{11} corresponds to the transmission loss for normal incidence, therefore solving the system of Eq. (4.51), it is found that normal incidence transmission loss, TL_n, in dB, is

$$TL_n(f) = 20\log_{10}(G_{11}) = 20\log_{10}\left(\frac{A^r D^o - A^o D^r}{C^r D^o - C^o D^r}\right), \tag{4.52}$$

or, equivalently, after some algebraic manipulations employing Eqs (4.45), (4.46), (4.47), and (4.49),

$$TL_n = 20\log_{10}\left(\frac{1}{2}\left|T_{11} + \frac{T_{12}}{\rho_o c} + \rho_o c T_{21} + T_{22}\right|\right), \tag{4.53}$$

Knowing that (Ingard 1994; Allard 1993) the normal incidence transfer matrix for a finite depth, d, layer of a homogeneous, isotropic porous material (either rigid or limp) is

$$\begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} = \begin{bmatrix} \cos k_{c,p} d & j\rho_p c_p \sin k_{c,p} d \\ j\sin k_{c,p} d / \rho_p c_p & \cos k_{c,p} d \end{bmatrix}, \tag{4.54}$$

where $k_{c,p}$ is the characteristic wave number of the acoustic material, ρ_p is the density, and c_p is the speed of sound in the specific material.

Therefore, the characteristic wave number can be evaluated from Eq. (4.54) as

$$k_{c,p} = \frac{1}{d}\cos^{-1} T_{11}, \tag{4.55}$$

or

$$k_{c,p} = \frac{1}{d}\sin^{-1}\sqrt{-T_{12}T_{21}}\ , \tag{4.56}$$

while the characteristic acoustic impedance, $z_{c\ p}$ $\quad {}_{p}c_{p}$, is

$$z_{c,p} = \sqrt{\frac{T_{12}}{T_{21}}}\ . \tag{4.57}$$

Finally, the speed of sound and density can easily be determined as $c_p = 2\pi f / k_{c,p}$ and $\rho_p = z_{c,p} / c_p$, respectively. We remind that since the characteristic impedance and wavenumber of the material have been determined, they can be converted to the surface impedance and absorption coefficient for a particular thickness of the porous material on a rigid backing, using Eqs (4.36), (4.37) and (4.33). The transmission loss method is described in the ASTM E2611-09 (2009) standard. An ISO standard is also under development.

4.5 Sound absorption mechanisms, porous materials.

One of the main sound absorbent classes is the *porous materials*. It encompasses all materials in which sound propagates through a network of interconnected pores in such way that acoustic energy is converted to heat mainly due to viscous boundary layer effects. For the absorption to be effective air should be able to pass through the material. Air is a viscous fluid, and thus sound energy is dissipated via friction with the pore walls. Moreover, flow changes during sound propagation through the irregular pores lead to a loss in momentum. Beyond viscous effects, there are losses due to thermal conduction from the air to the absorber material. All textile materials, nonwoven, woven or knitted, belong to this class. Typical textile materials used as sound absorbers are carpets, curtains, blankets, and cushions.

Porous absorbers are only effective at mid- to high frequencies (see Fig. 4.9). However, the ear is most sensitive at these frequencies and therefore their noise reduction effect is noticeable. Noise reduction is maximized if the maximum of the particle velocity is achieved within the porous material. This implies that in order to extend the noise reduction ability of porous materials to lower frequencies one has to increase their thickness. Another technique to extend to lower frequencies with a given thickness of absorbent is to place the porous material at a distance of quarter wavelength from a rigid back, e.g., from a room wall.

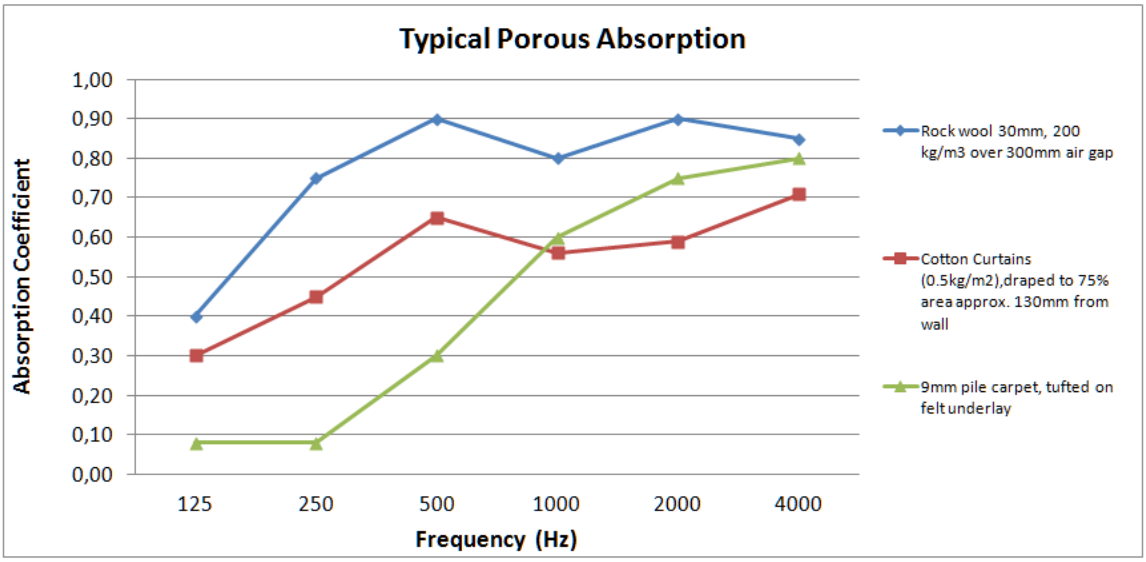

Fig. 4.9: Typical absorption of porous textile absorbers and rock wool fiber assembly.

The interaction of sound and the porous material is governed by the structural (geometric) details of the material. There are some material properties, known as *the microstructural properties*, which are closely related to the bulk properties of the porous material. The microstructural properties are used as design parameters for designing porous absorber with specific bulk properties. The key parameters for development or modeling are flow resistivity and porosity.

Flow resistivity, $\sigma = \Delta P / v_{fl} d$, where v_{fl} is the mean steady flow velocity, ΔP is the resultant pressure drop, and d is the thickness of the absorbent, $[\sigma] = 1 kgr / m^3 s = 1 rayl / m$, gives a measure of the resistance to air flow that the porous absorber poses. It provides some sense of how much sound energy may be lost due to boundary layer effects within the material. The porous material should have a low enough flow resistivity so the sound enters the absorbent, but not too low otherwise no dissipation occurs.

Porosity, ε, gives a measure of the amount of open air volume in the absorber available to sound wave propagation. It is a ratio of the total pore volume to the total volume of the absorbent. However, only the open connected pores, that allow air flow, should be included in the total pore volume. For many good porous absorbents it is possible to assume a porosity of $\varepsilon \approx 1$ (values near to 0.98 are usual).

It is noted that flow resistivity is related by empirical relationships to either bulk density, ρ_m, or to porosity of a fibrous porous material, $\varepsilon = \rho_m / \rho_f$, where ρ_f is the density of fibers. Although porosity and flow resistivity are the most important parameters in determining the sound absorption, other secondary parameters such as the shape factor and the tortuosity can be important.

Tortuosity, T_s, is defined as the ratio between the sound route through the pores and the thickness of the porous material. It is a parameter representing how the orientation of pores relative to incident wave affects sound propagation. The air that is forced to follow a tortuous path suffers accelerations which cause momentum transfer from air to the material. It is partly indicating the degree of complexity of the sound path through the material; the more complex is the propagation path, the higher is the absorption.

Characteristic lengths are parameters which refer to geometrical characteristics of the pores. Usually, pores have complicated shapes and this is why at least two geometric parameters (lengths) are needed to express shape influence to the sound propagation and consequently the absorption. Different pore shapes have different surface areas and hence have different thermal and viscous effects. The effective density of real porous absorbers is mainly determined by parts of the pores with smaller cross-sections, whereas the bulk modulus is more determined by larger cross-sectional areas.

Once the microstructural properties are known, it is then possible to calculate the characteristic acoustic impedance, z_c, and the characteristic propagation wavenumber, k_c, via appropriate models. The phenomenological theoretical models need more microstructural properties to be known than the macroscopic empirical models which consider a macroscopic view of the propagation, ignoring the details of propagation through each pore. This is the reason why the second ones, like the Delany and Bazley model, are more popular.

The macroscopic empirical model of Delany and Bazley offers very good accuracy under certain conditions. The empirical relations giving the characteristic acoustic impedance and the characteristic propagation wavenumber as functions of flow resistivity for a fibrous porous material are:

$$z_c = \rho_o c_o \left(1 + 0.0571 X^{-0.754} - j0.087 X^{-0.732}\right), \tag{4.58}$$

and

$$k_c = \frac{\omega}{c_o}\left(1 + 0.0978 X^{-0.700} - j0.189 X^{-0.595}\right), \tag{4.59}$$

where $X = \frac{\rho_o f}{\sigma}$, while c_o and ρ_o are the speed of sound in air and the air density, respectively.

The conditions under which the above model gives reasonable accuracy are: (a) The porosity should be close to 1, which most purpose built fibrous absorbers achieve. (b) $0.01 < X < 1.0$, which means that the validity is restricted within a defined frequency range. (c) The limits of the flow resistivity should be $1000 \le \sigma \le 50{,}000$ (in $rayl/m$).

Different phenomenological theoretical models are also used in the literature, like, for example, the model of Zwikker and Kosten (1949), which has been recently employed in textile materials (Shoshani and Yakubov 2000; Dias et al. 2007b). These models can also lead to the estimation of the characteristic acoustic impedance, z_c, and the characteristic propagation wavenumber, k_c, of the medium, via the effective density and the bulk modulus, using Eqs (4.17) and (4.18).

We remind that since the characteristic impedance and wavenumber of a material have been determined, they can be converted to the surface impedance and absorption coefficient for a particular thickness of the porous material on a rigid backing, using Eqs (4.36), (4.37) and (4.33).

4.6 Applications on textiles.

In a recent review article (Seddeq 2009, and references therein) the influence of various factors of a fibrous material on sound absorption have been summarized in the following points:

- Sound absorption coefficient increases with a decrease in fiber diameter, micro denier fibers with less than 1 denier per filament (dpf) provide a dramatic increase in acoustical performance.
- One of the most important qualities that influence the sound absorbing characteristics of a fibrous material is the air flow resistivity. In general, it can be inferred that, higher airflow resistivity always gives better sound absorption values but for airflow resistance higher than a limit the sound absorption is decreasing because of the difficulty of sound wave to pass through the materials.

- Tortuosity mainly affects the location of the quarter-wavelength peaks, whereas porosity and flow resistivity affect the height and width of the peaks. It has also been said that the value of tortuosity determines the high frequency behavior of sound absorbing porous materials.
- Fiber surface area and fiber size have strong influence on sound absorption properties; higher surface area and lower fiber size increases sound absorption.
- Less dense and more open structures absorb sound of low frequencies; denser structures perform better for frequencies above 2000 Hz.
- The creation of air gap increases sound absorption coefficient values in mid and higher frequencies. At the same time, creation of air gap presents minima at various frequencies for various air gap distances.
- Covering films such as PVC attachment increase sound absorption at low and mid frequencies at the expense of higher frequencies.

Textiles are typical examples of fibrous porous absorbers. Of course, different manufacturing techniques, different design, different raw materials and different treatments lead to different sound absorption characteristics. Several research articles have been published on the sound absorption properties of textile structures. The interest begun in the early 70s (e.g., Nute and Slater 1973; Dunlop 1974; Slater 1974, and references of these papers) when the focus was in evaluating common home textile items like curtains and carpets. However, the last years an increasing number of researchers, both from acoustics and textile side, focus their efforts on the acoustics of textiles. This trend is, to a great extent, caused by the evolution of new textile production methods and new textile materials. The textile products constitute an interesting sound absorbing alternative because of their two important advantages: low production cost and light weight. They are widely used in aviation, automotive and buildings. Further to the achievement of a high sound absorption performance the aesthetic aspect is also pursuit.

One could suggest two different classifications of the corresponding scientific literature. On one hand, the research works could be divided to the ones trying to achieve physical models of specific textile configurations and the ones trying to attain "black-box" models linking textile properties and acoustical properties. Both approaches involve sound absorption evaluation measurements, either to verify, or to lead to the models, respectively. On the other hand, the research works could be classified according to the textile structure under study. In this classification too, the acoustic evaluation measurements play a central role. Most of the publications use one of the widely accepted impedance tube methods for the involved acoustic evaluation.

Following the second classification scheme, one could identify four main categories corresponding to the study of: (a) raw materials in the form of fiber assemblies, (b) nonwoven fabrics, (c) woven fabrics, and (d) knitted fabrics. However, it should be noted that the majority of the published material deal with either nonwoven or knitted fabrics, mainly due to the associated lower production cost. The trend is to improve the acoustic performance of knitted fabrics and accordingly shift interest from nonwoven to knitted, which further provide a better aesthetic result.

In the case of *raw material* studies, the interest is focused on the comparison to standard porous fibrous absorbents like mineral wool and glass wool. The objective is usually to suggest eco-friendly (recyclable, biodegradable, natural) materials, not causing environmental pollution or danger for human health, with sound absorbing ability comparable to the one of the standard porous fibrous absorbents. (Ersoy and Küçük 2009; Yang et al. 2011).

Nonwoven fabrics are currently the most widely used textile materials in acoustic insulation products. This is due to their high total surface area, which is directly related to the denier and cross-sectional shape of the fiber. Nonwovens' fiber network structure provides enough degrees of freedom to permit acoustic design. Although they have only fibers and voids, which are filled by air, their structure is very complex. Voids, which are covered by fibers, are called pores in nonwoven structures and their size, number and shape is very important for the sound absorptive properties of nonwovens (Süvari and Ulcay 2011). The smaller the denier, the more fibers for the same material density, the higher the total fiber surface area is. This results to a higher chance for interaction between the sound wave and the fibers in the fabric structure. Another important parameter is the density of the nonwoven material, which affects the geometry and the volume of the voids in the structure. Ultra fine fibers enhance the mass per unit area, which contributes to an increase in the density and provides more chances for contact with the sound energy in the sound absorption nonwoven (Lee and Joo 2004). Moreover, web orientation, although marginally, is affecting sound absorption with unoriented web being more efficient (Lee and Joo 2003).

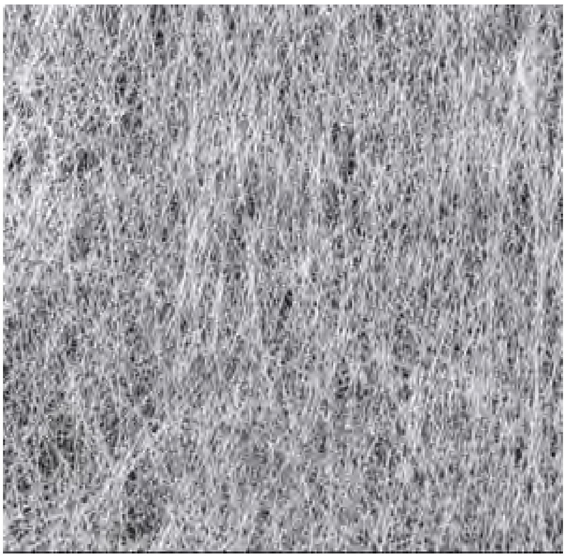

Fig. 4.10: Typical nonwoven fabric with sound absorption capability.

To a limit, the thicker a material, the more absorbent it becomes, particularly for low-frequency sounds (Rettinger 1968; Lee and Joo 2003; Na et al. 2007). Thinner materials, generally, require higher flow resistivity than thicker materials. One common method of increasing flow resistivity is the addition of a flow resistant scrim or film layer, which increases the air flow resistivity without adding too much weight or thickness. It is also possible to increase the flow resistivity by increasing the surface density of the material (adding density without changing the thickness); however, this method adds weight, which may be an issue in specific applications (Zent and Long 2007). Nevertheless, although the flow resistivity of a material may be increased to improve absorption at lower frequencies this is at the cost of lower absorption at higher frequencies (Lee and Joo 2003; Lee and Joo 2004; Zent and Long 2007).

Increasing the level of needle punching, using finer diameter or lower modulus fibers, and applying various coatings or sizing on nonwoven cellulosic composites improved their sound absorption (Parikh et al. 2001; Na et al. 2007). The surface roughness of nonwovens is also effective for increasing noise reduction performance. Roughness on the surface leads to resonance, and so appropriate roughness gives an increase in the sound absorption (Lee and Joo 2004). Nonwoven fabrics with a greater number of layers yielded higher noise absorption values (Rakshit et al. 1995; Shoshani 1990; Na et al. 2007). Heat-bonded materials are able to provide an equal sound absorption value at lower thickness than the needle-punched fabrics (Genis et al. 1990; Na et al. 2007). Acoustic barriers made of thermally bonded nonwovens can have several interesting applications, such as fillings inside walls separating neighboring apartments in wooden houses, noise shelters in the transfer industry, and acoustic enclosures for noise equipment in factories and workshops (Lee and Joo 2004).

An appropriate treatment, like carbonization, can also improve sound absorbing performance. For example, nonwoven composites with activated carbon fiber (ACF) nonwoven as a surface layer had significantly higher values of average transmission loss than glass fiber-surfaced composites above 600Hz (Chen and Jiang 2009). Furthermore, recent studies are considering the possibility of using natural fibers, like coir fibers, blends of bamboo, banana, and jute fibers with polypropylene (PP) staple fibers, or kenaf, waste cotton, and flax in blends with polypropylene and polyester (PET) as raw materials for nonwoven (Ayub et al. 2009; 2011; Thilagavathi et al. 2010; Parikh et al. 2006).

Concerning *woven fabrics*, the intrinsic parameters of the woven fabrics have a very little effect on their noise absorption coefficients in the low frequency domain ($f < 500Hz$) but a significant impact at the higher examined frequency (e.g., $f > 2000Hz$). It has been found (Shoshani and Rosenhouse 1990) that out of the intrinsic parameters examined, the cover factor had the most significant effect on sound absorption. The air gap behind the fabric has a very significant effect on the functional relations between absorption coefficient and the frequency (Shoshani and Rosenhouse 1990). The acoustic characteristics of woven hoses have been exploited in pipe noise reduction applications, like automotive intake systems, ventilation and air-handling systems (Dokumaci 2010). Interesting new ideas, like the construction of woven microperforated sound absorbers, have been possible since the introduction of woven three-dimensional fabrics (Onen and Caliskan 2010).

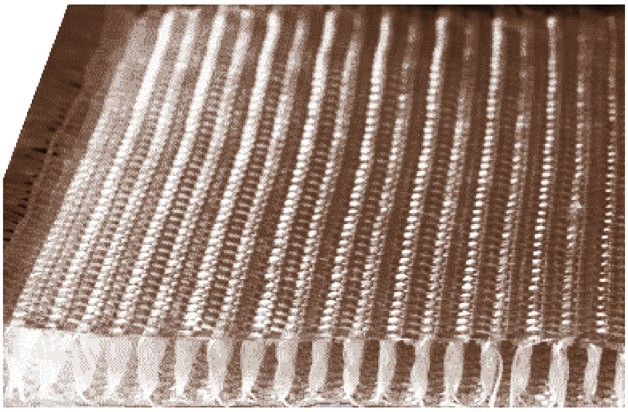

Fig. 4.11: Typical woven fabric with sound absorption capability.

Carpets, although usually referred to as a distinct category different from woven and knitted, could be seen as early three-dimensional woven fabrics. Carpets have been important sound-absorbing materials. Their effectiveness varies according to backing material, pile structure, yarn weight, pile thickness and underlay (Rossing and Fletcher 2003, pp. 256–270; Na et al. 2007). Pile characteristics including fiber content, pile density, pile height and air gap were reviewed with sound absorption of carpets (Shoshani and Wilding 1991; Na et al. 2007). The sound absorption of a carpet depends on its pile height, pile weight, type of backing material and whether the backing is coated with latex, loop or cut pile, and type and thickness of pad. Cut pile provides greater sound absorption than loop pile of otherwise identical construction. The type of pile fiber seems to has no significant effect on absorption (Na et al. 2007).

Nevertheless, the three-dimensional fabrics are mostly found in the form of *knitted spacer fabrics*, mainly due to their considerably lower cost compared to the corresponding woven fabrics. It has been shown that the fabric surface structure and thickness, spacer yarn type and their connecting ways, fabric combinations and their arrangement methods have significant effects on the sound absorbability (e.g., Öztürk et al. 2011). Sound absorption coefficient of fabrics increases with the increase in the number of miss stitches in knitted structure, because the total thickness of fabric increases. The results showed (Öztürk et al. 2010) that better noise absorption coefficient values can be achieved by using mini-jacquard knit instead of plain knit structure due to thicker face layer and consequently thicker spacer fabric. Besides, density has positive effect on sound absorption property of fabric. The sound absorbency of the fabric increases with the reduction in its porosity.

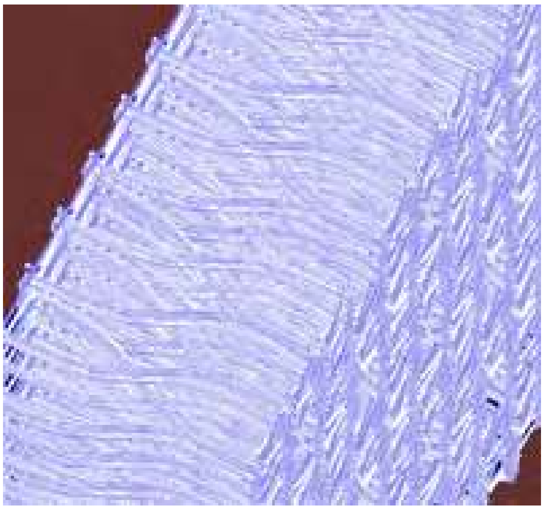

Fig. 4.12: Typical knitted spacer fabric with sound absorption capability.

The results also demonstrate that good sound absorbability could be achieved by using knitted spacer fabrics if suitable fabric structures and combinations are used. A recent research (Liu and Hu 2010) on several weft- and warp-knitted spacer fabrics yield interesting results. While the weft-knitted spacer fabrics exhibit the typical sound absorption behavior of porous absorber, the warp-knitted spacer fabrics exhibit the typical sound absorption behavior of microperforated panel absorber. The noise absorption coefficient increases with increase of the frequency for both kinds of fabrics. Both the weft-knitted and the warp-knitted spacer fabrics backed with the air-back cavity exhibit frequency-selected sound absorption with a resonance form. The sound absorbance can be improved by laminating different layers of fabrics. For the weft-knitted spacer fabric, the absorption increase by adding layers up to a number beyond which no more improvement is observed. However, for the warp knitted spacer fabrics, the absorption seems to continuously improve with increase of the fabric layers, but with a shift of the resonance region towards the lower frequency side. The combinations of weft-knitted and warp-knitted spacer fabrics can significantly improve their sound absorption, but their arrangement sequence has an obvious effect. At higher frequencies, the absorption coefficients of the warp-knitted spacer fabrics backed with weft knitted fabrics are much higher than those of the weft-knitted spacer fabrics backed with warp-knitted fabrics. However, at lower frequencies, the absorption of the warp-knitted spacer fabrics backed with weft-knitted fabrics is much lower than this of the warp-knitted spacer fabrics backed with weft-knitted fabrics. The air-back cavity can be replaced by multilayered warp-knitted spacer fabrics to achieve high absorption at low and middle frequencies.

Tuck spacer fabrics consists of top and bottom plain knitted layers. These two layers are interconnected with a mesh of yarn. This kind of fabric is a suitable alternative to utilizing several layers of plain knitted fabrics for achieving better sound absorbency. The sound absorption of these fabrics increases with both air flow resistivity and thickness. However, the effect of density is more predominant in terms of sound absorbency than thickness. The fabrics can be made denser by having more rows of the interconnecting yarn between a plain knitted course of the front and back beds (Dias et al. 2007a).

Another possibility investigated is that of thick knitted spacer fabrics. The results show better noise absorption when there is a thicker air gap between the front and back fabric layers of the spacer fabric and/or a thicker face layer. This type of a knitted structure is considered to be a promising material since it provides wide band sound absorption (Dias et al. 2007b).

Beyond knitted spacer fabrics, 2D knitted fabrics are also considered as a sound absorbent alternative, since their pros include, an aesthetic look and drapability compared with nonwovens. Therefore, if appropriately applied, they can supply a 3D seamless fabric with an attractive appearance (Honarvar et al. 2010). The sound absorption of plain knitted fabrics has recently been investigated (Dias and Monaragala 2006). It was found that knitted structures with smaller pore sizes (and a reduced porosity) with increased thickness, i.e. a thicker and denser knitted fabrics, have better sound absorbent properties (Dias and Monaragala 2006). Moreover, weft knitted fabrics produced in rib gating that is knitted on double jersey machines have also been studied (Honarvar et al. 2010). The samples with different numbers of knit and tuck stitches provide higher noise absorption than the samples with different numbers of knit and miss stitches (Honarvar et al. 2010).

Finally, a separate characteristic example is *microfiber fabrics*, which can be either woven or knitted. In any form, micro-fiber fabrics present an increased sound absorption because their fibers have a higher surface area than those of regular fiber fabrics, resulting in higher flow resistance (Na et al. 2007; Lee and Joo 2003).

4.7 References

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