Chapter 10
Industrial Applications II

Acoustic Package Optimization Methods in the Building Industry

Fabien Chevillotte

Abstract This chapter aims at presenting various building applications using porous materials. The role of porous materials and the relevant phenomena will be introduced for four typical applications, namely the acoustical correction, air-borne insulation, solid-borne insulation and ceiling applications. This chapter attempts to help the reader to identify the considered acoustical application, as well as in understanding the associated physical phenomena, in order to easily pinpoint the optimization key levers.

10.1 Introduction

Porous materials are widely used in building applications. Nevertheless, there are different applications and the relevant phenomena as well as their governing parameters differ. The purpose of this chapter is to give an overview of building applications and to attempt to explain how porous materials can help to improve the acoustical performance in these applications. Porous materials such as fibrous materials or foams are known as good "acoustical materials" but there is often a confusion between an absorbent material and an insulating material. Porous materials generally present good visco-thermal dissipation. This is enough to provide a good sound absorption performance but it is not enough to have good insulation properties when they are used alone.

Figure 10.1 compares the sound absorption coefficient in diffuse field (left) and the transmission coefficient (right) of a 46-mm thick glasswool (density $27\,\mathrm{kg{\cdot}m^{-3}}$) and a 12.5-mm thick plasterboard. The single rating number R_w is also indicated for transmission properties [1]. One can note that the porous medium (glasswool) shows a good sound absorption improving with the frequency when compared to the plasterboard which is impervious and has no absorption property. On the contrary, the glasswool has a poor insulation performance compared to the plasterboard one.

F. Chevillotte (✉)
MATELYS - Research Lab., Bât. B, 7 Rue des Maraîchers, Vaulx-En-Velin, France
e-mail: fabien.chevillotte@matelys.com

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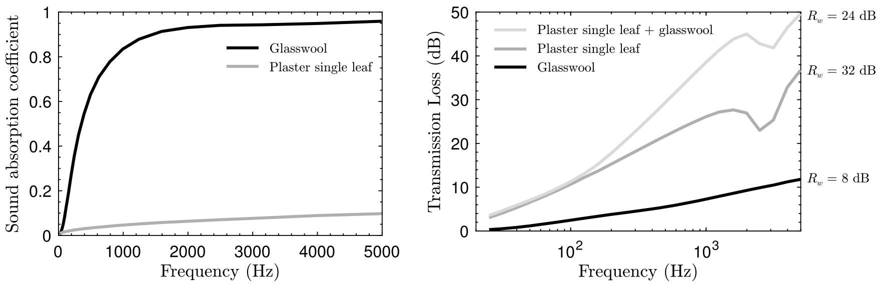


Fig. 10.1 Comparison of sound absorption coefficient (left) and transmission loss in diffuse field (right) of a 46-mm-thick glasswool and a 12.5 mm-thick-plasterboard

Nevertheless, the viscothermal dissipation of the porous medium can be useful to improve the insulation properties when assembled with other materials. The transmission loss of the glasswool added to the plasterboard is shown in the same figure. Note that gluing the porous material on the panel can affect the behaviour of the system and thus its transmission loss.

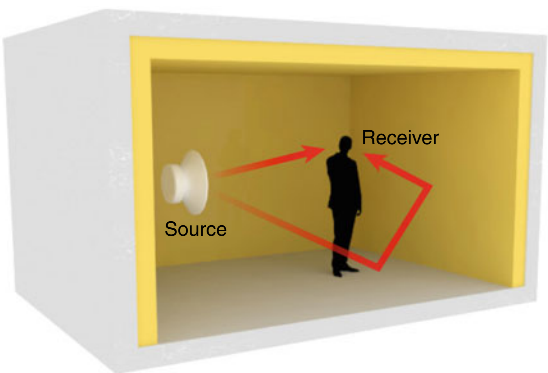
10.2 Acoustical Applications in the Building Industry

10.2.1 Acoustical Correction

The first building application, called "acoustical correction", consists of controlling the reverberation time according to the use of a room (e.g., living rooms, bedrooms, offices, classrooms, lobby, concert halls, etc.). In this application, the source and the receiver are placed in the same room, as shown in Fig. 10.2.

The reverberation time RT_{60} can be approximated by the Sabine's formula:

Fig. 10.2 Acoustical correction: direct propagation and multiple reflections of sound inside a room



$$RT_{60} = \frac{24 \ln 10}{c_0} \frac{V}{A} \approx 0.161 \frac{V}{A}. \tag{10.1}$$

The reverberation time, RT_{60}, is defined as the required time to decrease the sound pressure level by 60 dB from the level of the excitation. Here, c_0 is the speed of sound, V the volume of the room, and A the equivalent absorption area defined as

$$A = \sum_i S_i \times \alpha_i, \tag{10.2}$$

where S_i and α_i are the surface and the sound absorption coefficient of surface element i, respectively.

For most applications, one generally tries to reduce the reverberation time. Looking at Eq. (10.1), this can be done by reducing the volume V, which is often not achievable, or by increasing the equivalent absorption area A. This latter is carried out by increasing the surface of absorbing materials or the absorption performance itself. The absorbing materials are always porous materials or an assembly of porous materials, perforated plates and/or screens.

10.2.2 Air-Borne Insulation

The second building application is the air-borne insulation. It can be insulation to interior noises (TV, radio, vacuum cleaner, etc.) as well as external noises (vehicles, roadworks, airport, etc.). The insulation can be achieved within the same housing or between different premises (housings, commercial surfaces, etc.).

10.2.2.1 Single Wall Partition

When dealing with air-borne insulation, we classically start with a single leaf partition. When considering a plane wave at oblique incidence θ, a drop in transmission loss happens at the so-called coincidence frequency. This decrease is due to the coincidence of the transverse acoustical wavenumber $k_t = k_0 \sin\theta$ of the incident wave and the natural bending wavenumber $k_b = \sqrt{\omega}\sqrt[4]{m'/D}$, with $k_0 = \omega/c_0$ the acoustical wavenumber (Fig. 10.3).

The coincidence frequency is given by

$$f_c' = \frac{1}{2\pi} \frac{c_0^2}{\sin^2\theta} \sqrt{\frac{m'}{D}}, \tag{10.3}$$

with the surface mass density

$$m' = \rho h, \tag{10.4}$$

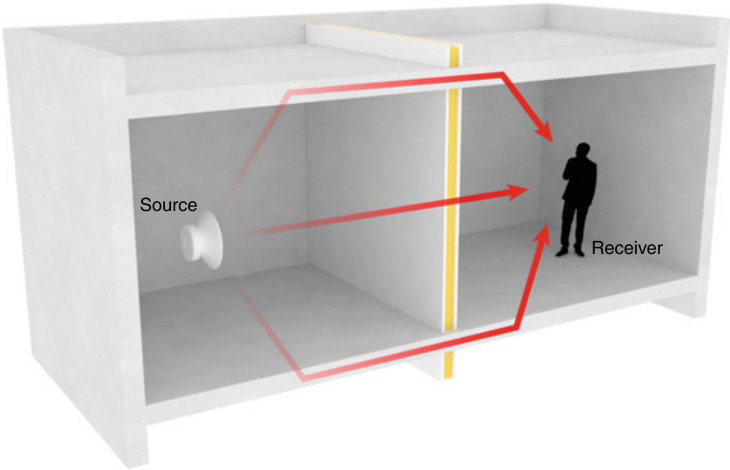


Fig. 10.3 Typical acoustical insulation problem and transmission pathways

and the bending stiffness

$$D = \frac{Eh^3}{12(1-\nu^2)}, \tag{10.5}$$

with ρ the mass density (kg/m^3), E the Young's modulus (Pa), ν the Poisson's ratio and h the thickness of the plate.

This coincidence frequency decreases while increasing the incidence angle (see Fig. 10.4). Under oblique plane wave, the transmission loss can be split in three zones controlled by different parameters:

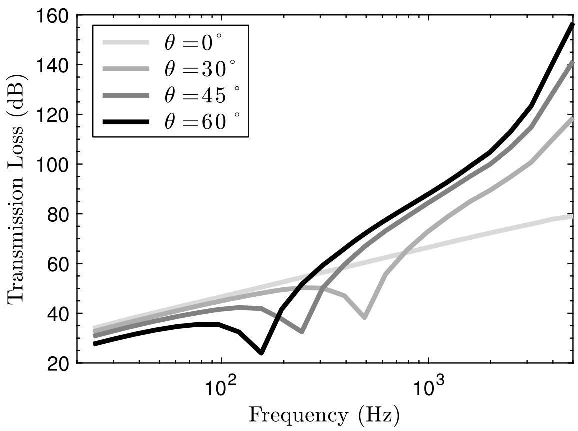


Fig. 10.4 Transmission loss of a 140 mm-thick single wall made of concrete for oblique plane wave excitation

(1) For $f < f_c'$: the mass law zone for frequencies lower than the coincidence one which is controlled by the surface mass density $m' = \rho h$ with a slope of 6 dB/octave.
(2) For $f \approx f_c'$: the coincidence zone which is mainly controlled by the damping loss factor η.
(3) For $f > f_c'$: a third zone for frequencies greater than the coincidence one which is controlled by the stiffness and a slope of 18 dB/octave.

When considering a diffuse field, the transmission loss can also be split in three zones but separated by the critical frequency f_c. The critical frequency is the lower coincidence frequency, corresponding to the greater incident angle ($\theta = 90°$), given by

$$f_c = \frac{c_0^2}{2\pi}\sqrt{\frac{m'}{D}}. \tag{10.6}$$

The typical transmission loss of a single panel (140 mm of concrete) submitted to a diffuse field is shown in Fig. 10.5. The transmission coefficient under diffuse field is computed thanks to an integration of oblique plane waves from 0 to 90°.

(1) For $f < f_c$: the first zone is still controlled by the mass which is controlled by the surfacic mass $m' = \rho h$ with a slope of 6 dB/octave (it is actually slightly lower around 5.5 dB/oct).
(2) For $f \approx f_c$: the critical zone is still controlled by the damping loss factor η.
(3) For $f > f_c$: a third zone which is controlled by the bending stiffness and the damping loss factor with slope of 9 dB/octave.

The slope goes from 18 dB/octave for a single incident angle to 9 dB/octave under diffuse field because the transmission loss in diffuse field integrates the stiffness effect and the coincidence frequencies of all incident angles.

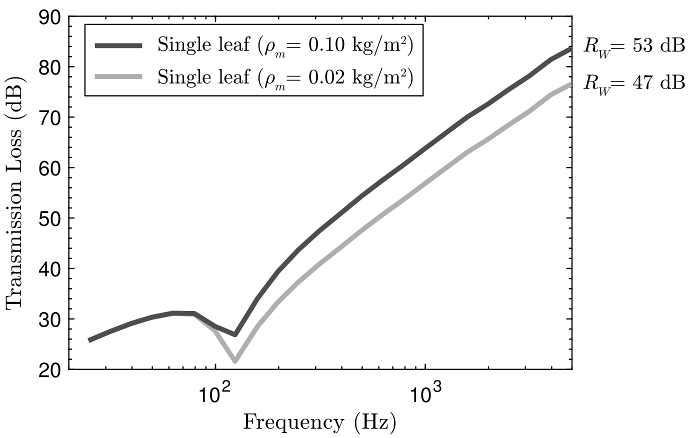


Fig. 10.5 Transmission loss of a 140 mm-thick single wall made of concrete for diffuse field excitation

The first way to enhance the transmission loss is to increase the surface mass density m'. This can be done by increasing the thickness which also implies an increase of the bending stiffness in power of 3 (see Eq. 10.5) or by increasing the mass density ρ. At the same time, the increase of mass density usually implies a stronger increase in Young modulus E. This means that increasing the mass generally implies a greater increase in bending stiffness and thus a decrease in the critical frequency. The decrease of the critical frequency strongly degrades the overall insulation performance. This degradation can be more important than the gain due to the increase of mass. Typical critical frequencies are 12 kHz for 1 mm-thick-steel plate, 2.5 kHz for 12.5 mm-thick-plasterboard 115 Hz for a 140 mm-thick-concrete slab.

10.2.2.2 Double Wall Partition

To get around this limitation, double wall partitions are usually employed (Fig. 10.6). Each panel has is own critical frequency. But thin plates enable to let the critical frequency relatively high. Nevertheless, another resonance appears at lower frequencies. It is a mass-spring-mass resonance and its characteristic frequency is called the breathing frequency, f_0, given by

$$f_0 = \sqrt{\frac{\mathrm{Re}\left(K_{eq}\right)}{4\pi^2 L_c}\left(\frac{1}{m'_1} + \frac{1}{m'_2}\right)}, \tag{10.7}$$

with K_{eq} the bulk modulus of the material filling the cavity, L_c the thickness of the cavity and m'_i the surface mass density of the i-th panel.

This resonance is a limitation, but the slope of the transmission loss is strongly increased right above f_0. The cavity can be filled with a porous material, usually a light glasswool, to increase the sound insulation at medium and high frequencies. The filling porous material mainly adds viscothermal dissipation which strongly improves the transmission loss above the breathing frequency.

The bulk modulus of air is the adiabatic value γP_0, with P_0 the atmospheric pressure and γ the heat capacity ratio ($\gamma = 1.4$ for air). When considering a typical

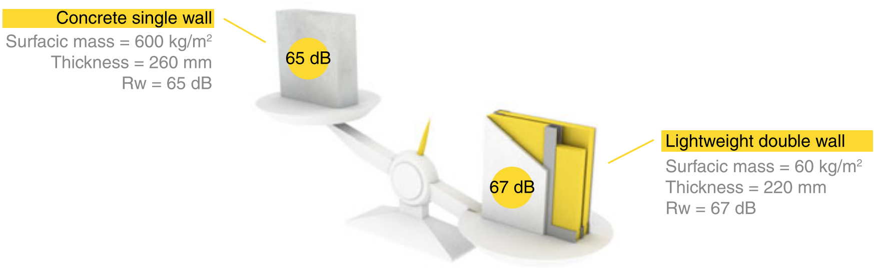


Fig. 10.6 Single leaf versus double leaf

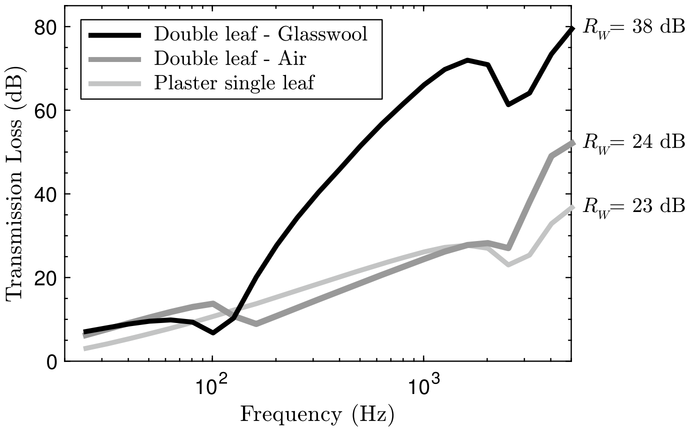


Fig. 10.7 Transmission loss of a double leaf partition (12.5 mm-thick-plasterboards and a 48 mm-thick-cavity filled with air or glasswool, stud-less) compared to a single leaf plasterboard of 12.5 mm

porous medium filling the cavity, the real part of the bulk modulus varies between the isothermal behaviour at low frequencies P_0/ϕ and the adiabatic one $\gamma P_0/\phi$, with ϕ the open porosity of the filling material [2]. Classical porous materials have a high porosity and an isothermal behaviour for frequencies around the breathing frequency. This means that filling the cavity with a porous such as a glasswool, decreases the stiffness of the cavity by a factor $\sqrt{1.4}$.

The improvement of viscothermal dissipation and the decrease of the breathing frequency are illustrated in Fig. 10.7 for a double wall partition made of two 12.5 mm-thick-plasterboards and a 48 mm-thick-cavity filled with air or glasswool. Note that additional phenomena such as double porosity [3] or adsorption/desorption can help to lower this bulk modulus [4].

Due to the self-supporting stiff walls, the Young's modulus of the porous material (glasswool here) is not important in this case, only viscothermal effects and density (so-called "limp" model) have to be taken into account. This situation becomes general whenever the porous material is decoupled from the walls with air gaps. Otherwise, full poroelastic models are compulsory and stiff decoupling porous materials have to be avoided.

10.2.2.3 Effect of Mechanical Links

To ensure the support of the partition, mechanical links have to be used between the two panels. This mechanical links, also called mounts or studs, introduce a solid-borne path which acts as an additional transmission path. The air-borne and the solid-borne paths are often computed separately and then combined [5] (see Fig. 10.8). Note that most of analytical models for solid-borne path computation are limited to two

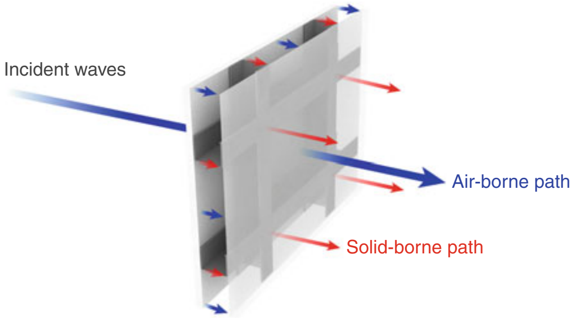


Fig. 10.8 Schematic drawing of a double panel with mechanical links

thin plates coupled by a mechanical link. A recent work has proposed a methodology to get around the thin plate assumption and to deal with more than one mechanical link [6].

The air-borne, solid-borne and total transmission loss coefficient are shown in Fig. 10.9 for the double wall partition used in the previous section (Fig. 10.7) with a mechanical link of stiffness $K_s = 10^6$ N/m and a loss factor $\eta = 0.08$ for the plasterboards. The solid-borne path is usually controlling the total transmission performance for frequencies higher 300 Hz for such a partition. The influence of the stud stiffness on the total transmission loss is illustrated in Fig. 10.10 for a given damping loss factor of the plasterboards ($\eta = 0.08$).

The influence of the loss factor on the total transmission loss is illustrated in Fig. 10.11 for a given stiffness of the mechanical links ($K_s = 10^6$ N/m). One can note that the influence of the damping loss factor of the plates is as important as the stiffness of the mechanical links. This means that the optimization of the stud stiffness has to be done for a given damping loss factor of the plates. Therefore, the loss factor of the plates is a major parameter. This can be explained using the wavenumber

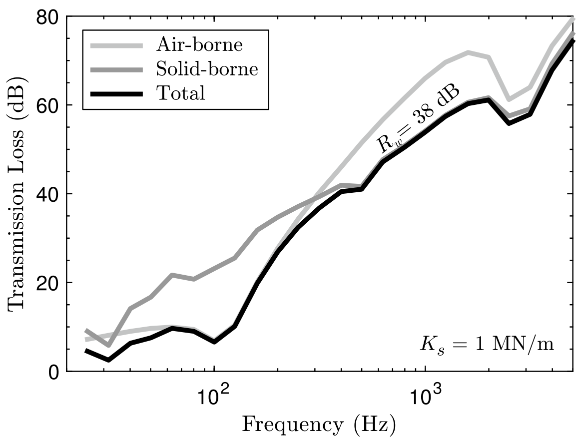


Fig. 10.9 Influence of the mechanical link on the transmission loss of a double wall partition: air-borne, solid-borne and total contribution (12.5 mm-thick-plasterboards ($\eta = 0.08$) and a 48 mm-thick-cavity filled with a glasswool (27 kg.m^{-3}), stud stiffness $K_s = 1e6$ N/m)

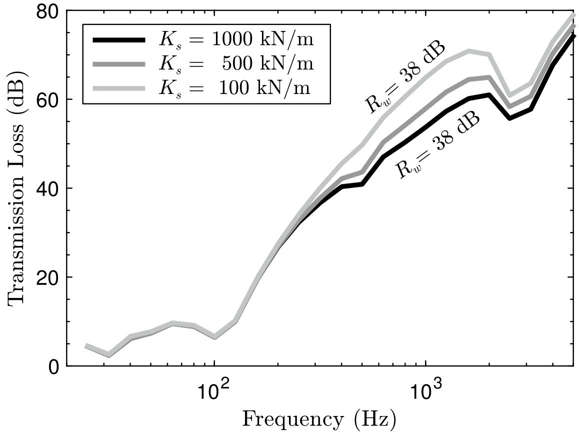


Fig. 10.10 Influence of stiffness of the mechanical link on the transmission loss of a double wall partition ($\eta = 0.08$)

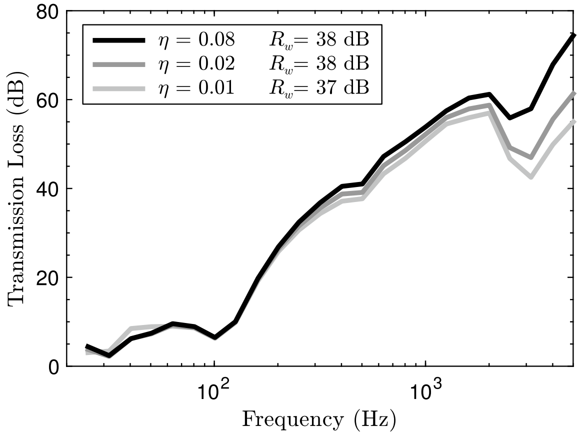


Fig. 10.11 Influence of damping loss factor of plates on the transmission loss of a double wall partition ($k_s = 10^6$ N/m)

analysis [7]. The solid-borne path presents a strong effect of the damping loss factor on the entire frequency range. This can be explained by the fact that the studs are excited by the first plate (emission side) and act as point forces exciting the second plate (reception side) and point forces excite a wide range of wavenumber. In this case there is always energy around the natural bending wavenumber of the second plate and a strong effect of the damping loss factor is observed.

This double wall partition problem is similar to the transmission problem presented in the part dealing with aeronautic applications (see Chap. 12) except that decouplers are employed to avoid the solid-borne path. Nevertheless, as a turbulent boundary layer excitation has to be considered (exciting a wide range of wavenumber [7]), the effect of the damping loss factor of the panels is also of primary importance.

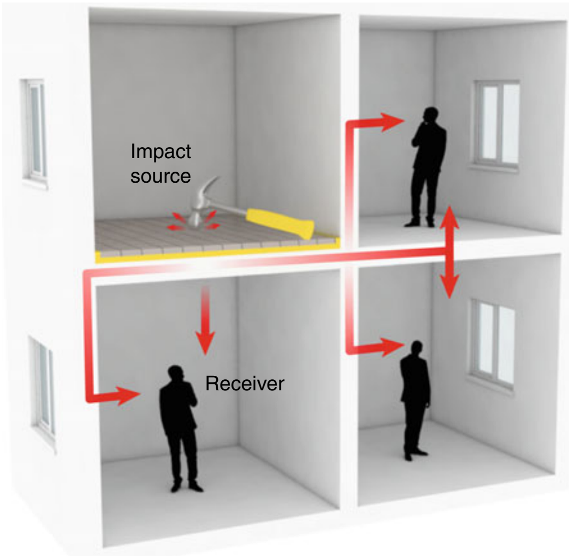


Fig. 10.12 Impact noise pathways in a typical building application. The underlay material (yellow) is added to increase the solid-borne insulation

10.2.3 Solid-Borne Insulation (Impact Noise)

Two floors are generally separated by a concrete slab, with a typical thickness around 140 mm. Its air-borne insulation is relatively good but floors can strongly radiate noise when submitted to impact source, e.g., falling objects, walking, etc. A spring-mass system can be added to increase the structure-borne insulation, as shown in Fig. 10.12. The mass is the surface floor (screed and tiles for instance) and the spring is an underlayer (porous or not). In this case, the porous layer must be sufficiently stiff to support the static load and at the same time sufficiently soft to shift the spring-mass resonance to lower frequencies in order to take advantage of the spring mass insulator. In this case, the performance is defined mainly by the elastic parameters of the porous layer. This is indeed the case presented in Sect. 9.3.3 previously.

Moreover, compared to an air-borne source, a mechanical source excites a wide range of wavenumbers, even at low frequencies, and the loss factor of the receiver panel (concrete slab in this case) is of primary importance [7]. The measurement of the impact noise is carried out thanks to a tapping machine [8, 9]. The tapping machine is placed on the upper floor (Fig. 10.12 top-left) and the sound pressure is measured in the lower room (e.g., at Fig. 10.12 bottom-left). First, one measures the normalized impact sound pressure level L_{0n} of the concrete slab and secondly the one of the concrete slab and the covering or the spring-mass system L_n. Then the reduction of impact sound pressure level is computed $\Delta L = L_{0n} - L_n$[10].

Normalized sound pressure levels are shown in Fig. 10.13 and the reduction of impact sound pressure level is shown in Fig. 10.14. The sound pressure level is strongly reduced at frequencies above the spring-mass resonance ($\approx 150 Hz$ in this configuration) but a slight increase happens at this resonance frequency. As this

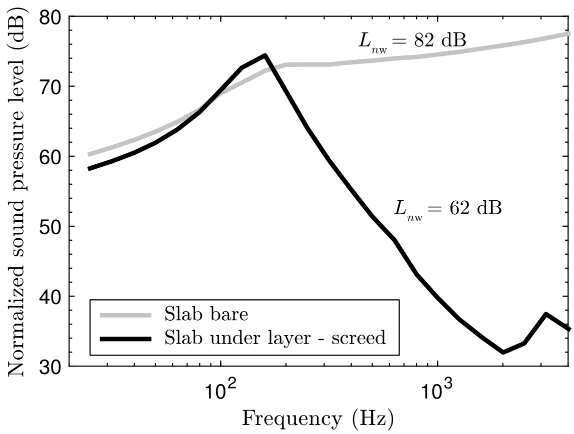


Fig. 10.13 Normalized sound pressure level with L_n and without under layer L_{n0}

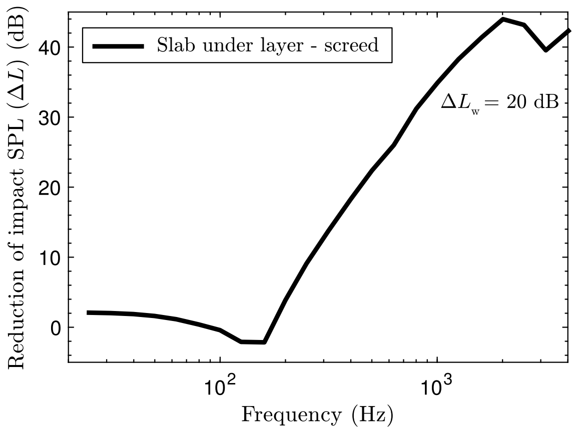


Fig. 10.14 Reduction of impact sound pressure level with under layer ΔL

resonance frequency is in the low frequency range, the single number rating ΔL_w is very sensitive to this phenomenon. When the underlayer is porous, one must be aware of the compressibility of the air which can become predominant when the mechanical stiffness is low.

10.2.4 Ceilings

Ceilings have a particular role since they are employed as sound absorbers but they can also introduce a lateral air-borne path between two rooms, as shown in Fig. 10.15. These parts have thus to be designed to control both the sound absorption in the emission room and to insure the air-borne sound insulation between two rooms. The

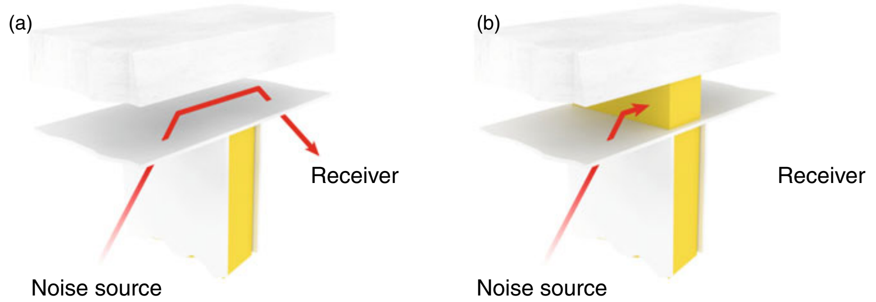


Fig. 10.15 Noise pathways in ceilings. **a** Noise propagating in the plenum introduces a lateral air-borne path. **b** Acoustic treatment of the plenum

treatment of the plenum is thus of primary importance to avoid lateral leaks and to attenuate the resonances of the plenum.

Acoustical treatments used in the automotive industry such as dash panels (see Chap. 11) are also designed to increase the sound absorption and the air-borne sound insulation. In both applications, a multilayer made of porous media, screens (or perforated plates) and/or impervious heavy layers can be employed.

10.3 New Trends in Building Acoustics

10.3.1 Thin and Aesthetic Absorbers

The new trends in building industry are varied. The first one is to use thin absorbers using textile or thin coatings. The idea is not to reach a sound absorption coefficient of 100% but rather to use thin treatments with lower absorption coefficient on large surfaces, such as the entire walls of a hotel lobby.

Another trend is to use additional suspended panels on ceiling or wall to increase the absorption area, e.g., in restaurants, offices, open-spaces, etc. Some panels can also include lighting. Absorption panels are used more and more often as aesthetic parts. Green materials and recycled ones are also often considered.

10.3.2 Low Frequency Performances and Non-conventional Phenomena

Similarly to other domains, the goal is to increase the low frequency performance of the solutions, for both absorption and insulation with a limited space, weight and cost. Acoustic metamaterials are of course studied. These materials attempt

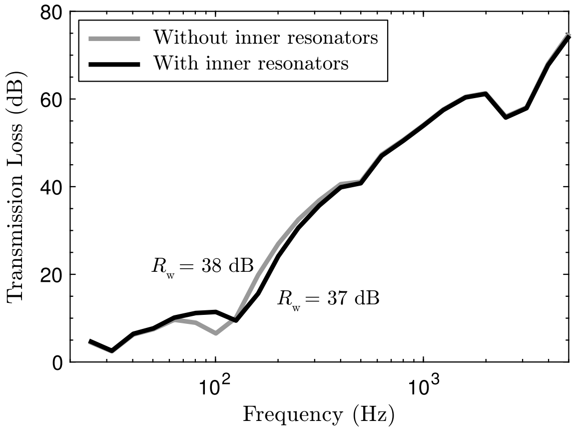


Fig. 10.16 Influence of inner resonators on the transmission loss of a double wall partition

to use additional physical phenomena to improve the sound absorbing or insulating performances. Resonating (acoustical, spring-mass or membrane-type resonators) or diffusion phenomena (pressure diffusion or multiple-scattering effects) are among them. In addition, the effect of the periodicity (Bragg's effect) can be used, but it is not required to take advantage of other non-conventional phenomena such as local resonances.

An example of porous medium with inner acoustical resonators (Helmholtz-type) in a double wall partition is studied here. The double wall partition is the same as the one studied in Sect. 10.2.2.3 with two plasterboards, mechanical links and a 48 mm-thick-cavity filled a glasswool and 20% of embedded resonators targeted around the breathing frequency.

The transmission loss values with and without resonators are compared in Fig. 10.16. One can note that the inner resonators enable to increase the transmission loss at the breathing frequency, which is shifted toward higher frequencies.

This can be explained looking at the bulk moduli shown in Fig. 10.17. Without inner resonators, the real part of the bulk modulus varies from the isothermal value P_0/ϕ (with $\phi \approx 1$ for a glasswool) at low frequencies to the adiabatic value $\gamma P_0/\phi$ at high frequencies. The imaginary part increases around the visco-inertial frequency (600 Hz). Looking at the bulk modulus of the glasswool with inner resonators, one can observe an increase of the imaginary part of the bulk modulus at the resonance frequency, which results in additional dissipation. On the other hand, one can notice a decrease of the real part of the bulk modulus below the resonance frequency and its increase right above it.

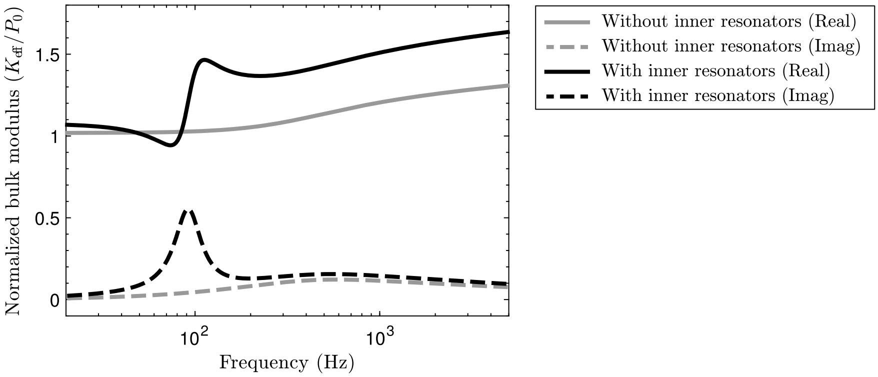


Fig. 10.17 Influence of inner resonators on the normalized Bulk modulus ($K/P0$)

As mentioned in Sect. 10.2.2.2, the breathing frequency increases with the real part of the bulk modulus (Eq. 10.7). Unfortunately, this phenomenon cancels the benefits of the additional dissipation of the resonators and the single number rating of the treatment with the resonators R_w is 1 dB lower that for the treatment without resonators. Obviously, different designs can be investigated but this type of drawback will always exist when the resonant phenomena are involved. This, however, does not happen for treatments based on diffusion phenomena.

10.3.3 Rolling Noise

Another trend is to divide buildings into commercial and habitable areas. Commercial shops typically occupy the ground floor, with private residencies on the upper floors. The delivery carts generate vibrations at low frequencies (100 Hz) that propagate easily throughout the building structure and on the upper floors, disturbing the habitants therein. Figure 10.18 shows that the sound pressure level due to rolling noise is far from the one generated with the tapping machine used for measuring impact noise according ISO standards [8, 9]. Moreover, one can note that typical decoupling spring-mass systems usually employed (see Sect. 10.2.3) are not suitable for rolling noise and can even amplify the rolling excitation when the spring-mass resonance coincides with the characteristic lobe of the rolling excitation. This problem is being addressed in current research using simulations and experiments [11].

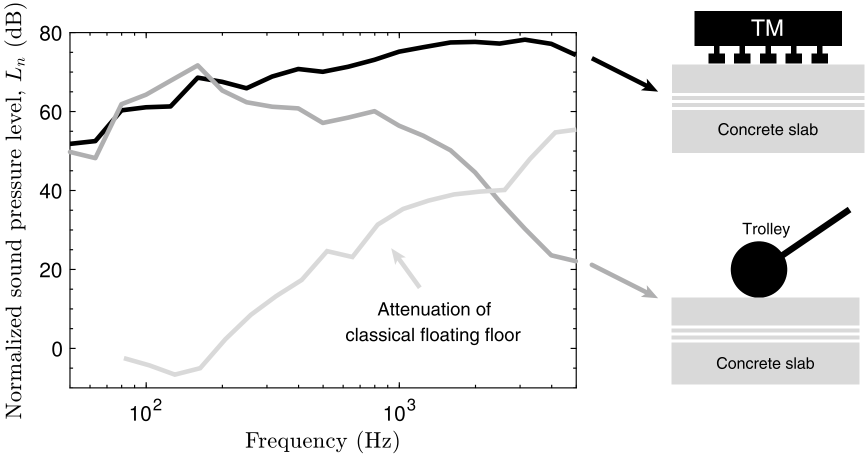


Fig. 10.18 Comparison of the sound pressure levels of tapping noise and rolling noise of a classical floating floor, as well as the attenuation of a typical floating floor

10.4 Conclusions

The porous media are employed to dissipate acoustical energy for absorption or insulation purposes. In double partition, porous media allow for an additional viscothermal dissipation at mid and high frequency and a decrease of the bulk modulus lowering the breathing frequency. When dealing with an acoustical problem in the building industry, we first have to identify which application is considered (acoustical correction, air-borne insulation, solid-borne insulation, ceiling) in order to understand which phenomena and therefore which parameters are important for optimization purposes.

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