Design of perforated panels for low frequency acoustic correction of rooms for listening to music

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ABSTRACT

The acoustic design of rooms for listening to music or recording is a very difficult subject: in order to improve the acoustic performance of these confined rooms, it may be necessary to absorb noise energy; sometimes all audible frequencies of the spectrum, sometimes at some specific frequencies. The design is especially difficult at low frequencies, where both resonance modes and standing waves are present. For the correction of problems of this kind, a resonance cavity perforated panel can be used. In the technical literature, there are some theoretical models describing the behavior of such a panel, but the results given are scarcely informative. Here we try to develop a simple but accurate approach for the design of these devices. On the basis of a reference FEM simulation of a drilled panel which was discussed in a previous paper [4], an engineering analytical model, which can be simply implemented either in few lines of FORTRAN code or by means of free engineering software tools like OpenOffice.org, has been developed. The dependency of the panel's behavior on the design parameters is here discussed.

1. INTRODUCTION

The correction of the reverberation time is the most important aspect of the interior acoustic design. Usually, to tune the room response both in the time and frequency domain, a part of the sound energy must be absorbed at some particular frequencies.

The sound absorption is quantified by the *sound absorption coefficient* α, defined as the ratio between the sound energy absorbed by a certain material and the total incident sound energy. It is a non-dimensional quantity, frequency dependent, with a range of values between 0 and 1.

There are three physical mechanisms of sound absorption: porosity absorption; cavity resonance absorption; shell resonance absorption. An example of the behavior of these kinds of sound absorption mechanisms is illustrated in Fig. 1: the porous material sound absorption, for usual thickness, is maximum in the middle-high audio spectrum, while in the midde-bass spectrum its performance is insufficient; at low frequencies, shell resonance absorbers or cavity resonators are usually used.

The acoustic response computation of the shell resonance absorbers is usually difficult, because it strongly depends on the interaction between the acoustic medium and the solid structure of the device itself. For this reason, the literature models [1] give a poor approximation of their real behavior.

On the other hand, the cavity resonators are generally simpler to model, because the coupling phenomena are usually negligible. Moreover, since resonance effects like standing waves, which influence both the reverberation time and the sound

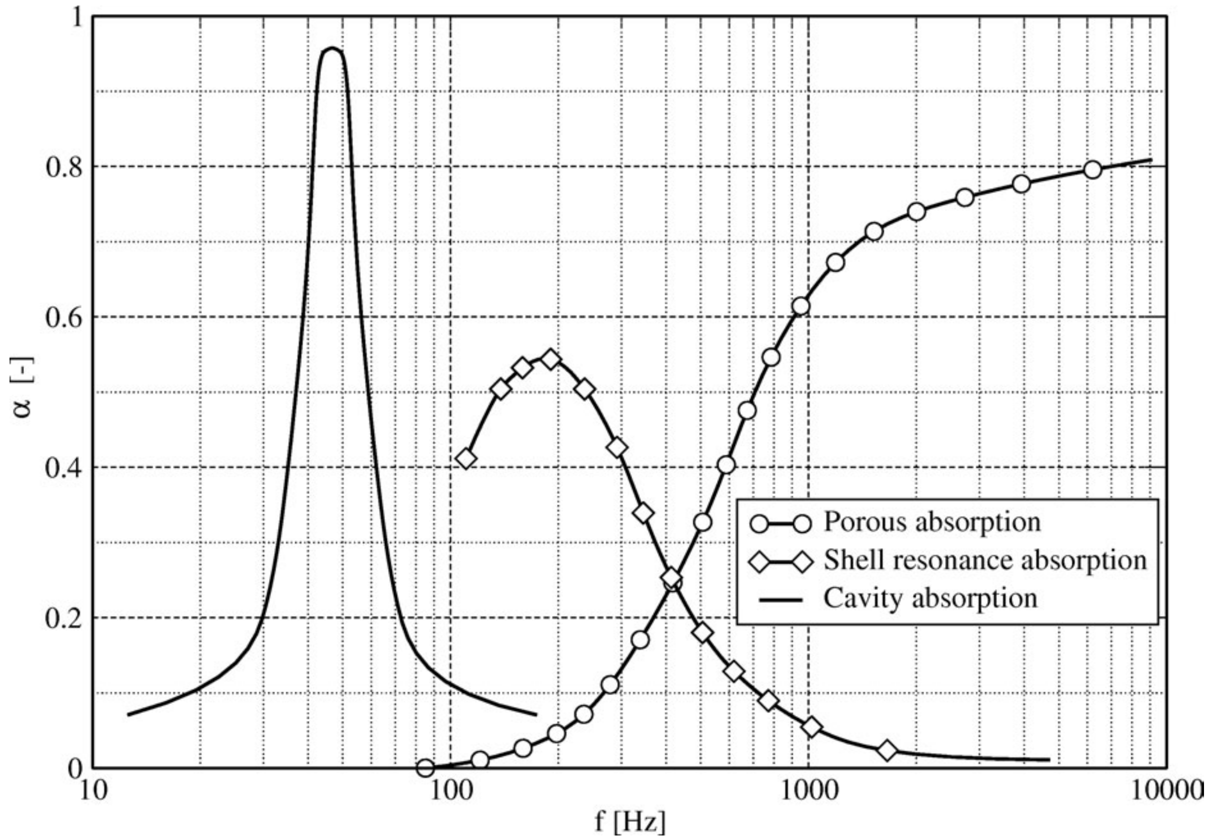

Figure 1. Sound absorption behavior example for three kinds of absorbers.

timbre, are present mainly at low frequency, a more selective absorption is preferable for the acoustic design in this part of the audio spectrum.

2 THE CAVITY RESONATORS

One can describe theoretically a single cavity resonator (also called Helmholtz resonator) as an air volume contained into a cavity with rigid walls, connected with the external ambient through a small duct (see fig. 2).

When a sound wave hits the resonator external surface, the air inside the duct is put in motion, and the air motion is propagated inside the cavity. If one assumes that the air inside the pipe moves rigidly, and that no mass exchange is possible between the resonator and the external ambient, then this acoustic system can be treated analytically as a mechanical mass-spring system, with one degree of freedom, where the behavior of the air inside the duct is analogous to a vibrating mass, and the behavior of air inside the cavity is analogous to a spring.

With these assumptions, the resonance frequency of this system can be computed as [2]:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{Vh}} \tag{1}$$

where c is the sound velocity (344 m/s in air under standard conditions), S and h are respectively the duct area and length, and V is the cavity volume.

This system transforms a part of the incident acoustic energy to kinetic energy; it is simple to understand that this transformation (and then the acoustic absorption) is maximum in correspondence to the resonance frequency, and it decreases very fast far from the resonance frequency.

Even if this is only a mathematical model, it is useful to understand the behavior of the cavity-resonance acoustical devices and why they are especially effective for pure sound waves (characterized by a single frequency). This is the case for the standing waves at low frequency in the rooms.

If in the cavities of these devices a layer of porous material is also present, the behavior is more difficult to describe: in general, dissipative phenomena are present inside the cavity, the frequency range where the absorber works is generally larger, and the maximum absorption peak is usually lower than in the absence of the porous material. For the design of devices of this type, Eq. (1) is usually not adequate.

3. THE PERFORATED PANELS

Perforated panels are special types of cavity absorbers. In these systems, many cavity resonators work in parallel. The panel discussed here is made of plywood. It is composed of a perforated frontal plate, mounted on wood supports to separate it from the wall of the room. A layer of fibreglass and a layer of air are placed between the wall and the plate; the air layer is connected with the air of the room through the holes of the plate (Fig. 3), which constitute the cavity absorbers ducts. It is not necessary that the volume of each resonator is separated from the others by a material wall: all the space inside the frontal plate can be open.

Using a simple model based on Eq. (1) and assuming that no porous material is present, the resonance frequency of this system can be calculated as [1]:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{\pi\phi^2}{4D_1D_2L_a h'}} \tag{2}$$

where ϕ is the hole diameter, D_1 and D_2 are respectively the distances from the hole centers, measured along the x_1 and x_2 axis, L_a is the thickness of the air layer, i.e., the distance between the internal surface of the frontal plate and the rigid wall, and h' is a corrected length of the duct to take into account the local turbulence effects, defined as [3]:

$$h' = h + 0.8\phi \tag{3}$$

where h is the frontal plate thickness.

Unfortunately, also the results obtained using Eq. (2) are not in perfect accordance with the real behavior of these devices (see for example the experimental measures described in [4]), because the presence of a layer of porous material is neglected. Moreover, Eq. (2) does not show the trend of the sound absorption coefficient in function of the frequency, because it gives only its maximum value.

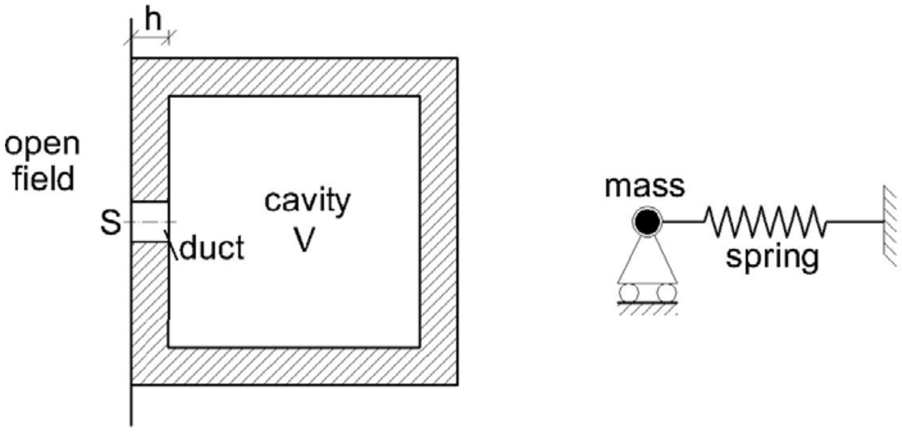

Figure 2. Schematics of a single Helmholtz resonator.

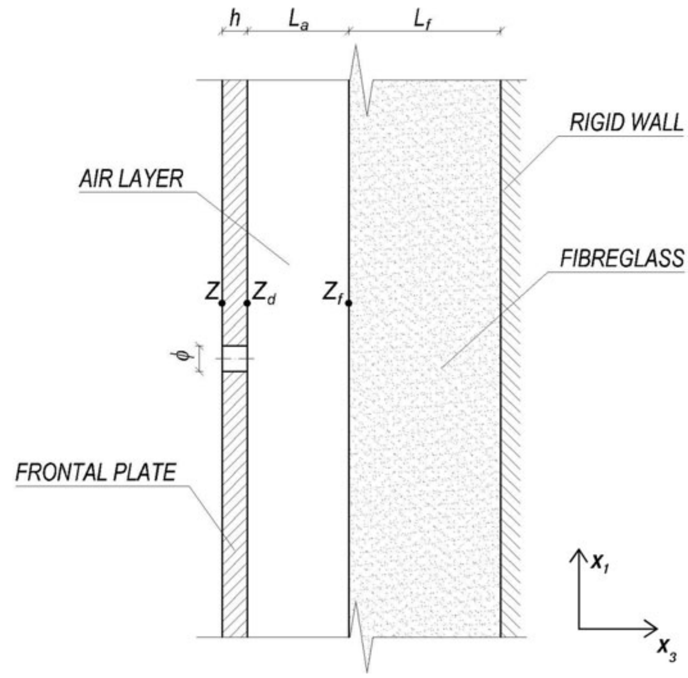

Figure 3. Schematics of the cross-section of a perforated panel.

4. A NUMERICAL MODEL OF THE PERFORATED PANEL

In previous work [5], the acoustic behavior of a perforated panel and the sound field inside the device have been numerically simulated using a commercial finite element code [5]. A coupled fluid-structure simulation has been carried out, in which both the acoustic and the structural behavior of the system are considered.

A comparison between numerical and experimental results is reported in Fig. 4. The simulation results are good agreement with the experimental measures, but the computational cost is considerable: the numerical model has two millions of degrees of freedom, and the solution requires about ten hours on a HP Integrity Server with two 64-bit Itanium-2 processors and 64 GB of RAM. For this reason, starting from the numerical results, a simple analytical model has been developed.

5. AN ANALYTICAL MODEL

From the information obtained studying the numerical results, we have developed an analytical model, based on the impedance method, which describes with reasonable accuracy the perforated panel behavior at low frequency, and which allows one to obtain the frequency trend of the sound absorption coefficient.

The numerical results have suggested basing the developed model on the following assumptions:

- a single cell has been considered [4];
- the frontal plate and the wall behind the panel are both assumed to be rigid and perfectly reflective;
- the sound waves incident the panel are considered plane;
- the sound waves in the air layer into the device are considered plane, except near the ducts, because of the diffraction effects.

The acoustic impedance Z is a complex quantity, defined as:

$$Z = \frac{p(\mathbf{x}, f)}{\mathbf{u}(\mathbf{x}, f) \cdot \mathbf{n}} \qquad (4)$$

where p is the acoustic pressure (in complex notation), at a point of coordinates **x**, $\dot{\mathbf{u}}$ is the particle velocity (in complex notation), f is the sound frequency, and **n** is the surface normal. One can prove that the air impedance in the open field is equal to $Z_0 = \rho_0 c$, where ρ_0 is the air density.

According to the numerical results, a representation of the device acoustic behavior based on the impedance model has been developed. The model is schematically represented in Fig. 5, where Z_m, Z_s, and Z_d are respectively the impedances of the panel duct, of the solid portion of the frontal plate and of the inside part of the device (air layer and porous material). One can observe that Z_m and Z_s are connected in parallel: Z_t is their equivalent impedance, which is then in series with respect to Z_d. The impedance of the entire system is called Z (for ease of notation, the frequency dependence has been omitted).

When Z is known, the sound absorption coefficient can be obtained as a function of the frequency according to [2]:

$$\alpha(f) = \frac{4Z_0 Re\{Z(f)\}}{\left(Z_0 + Re\{Z(f)\}\right)^2 + \left(Im\{Z(f)\}\right)^2} \tag{5}$$

In order to calculate the impedance Z_t, we can consider a consequence of the second assumption: Z_s is much larger than the duct impedance Z_m. With some algebraic passages, and considering the expression of the impedance for a translating mass, one can write [2]:

$$Z_t = 2 f \rho_0 h' \frac{4D_1 D_2}{\phi^2} i \tag{6}$$

where i is the imaginary unit.

The inside part impedance Z_d can be computed according to [2]:

$$Z_d = Z_0 \frac{Z_f \cosh\left(ik_0 L'_a\right) + Z_0 \sinh\left(ik_0 L'_a\right)}{Z_f \sinh\left(ik_0 L'_a\right) + Z_0 \cosh\left(ik_0 L'_a\right)} \tag{7}$$

where k_0 is the wave number in air, equal to $k_0 = \frac{2\pi f}{c}$; Z_f is the impedance of the porous material placed against a rigid wall, defined below; L'_a is the corrected thickness L_a in order to take in account the diffraction effects due to the duct presence [4], equal to:

$$L'_a = L_a + \eta\left(h + 0.8\phi\right) \tag{8}$$

where η is a numerical parameter which can be obtained experimentally, or by numerical simulations. We have obtained a good agreement in the whole range of the frequencies of interest for $\eta = 2$.

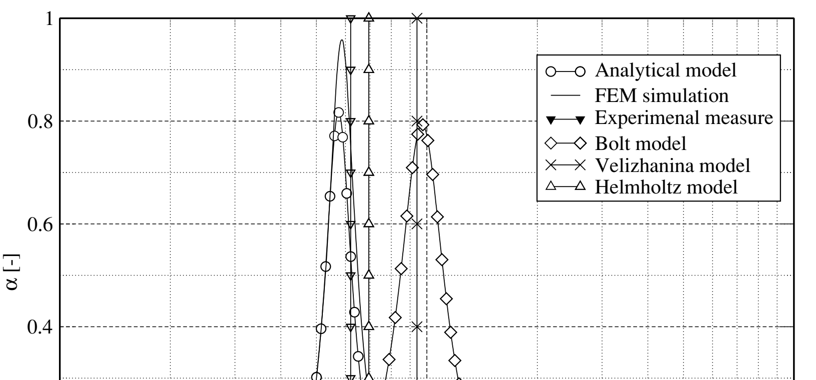

Figure 4. Sound absorption coefficient calculated as a function of the frequency with various method for a panel with D_1 = 107.5 mm, D_2 = 177.14 mm, and ϕ = 8 mm.

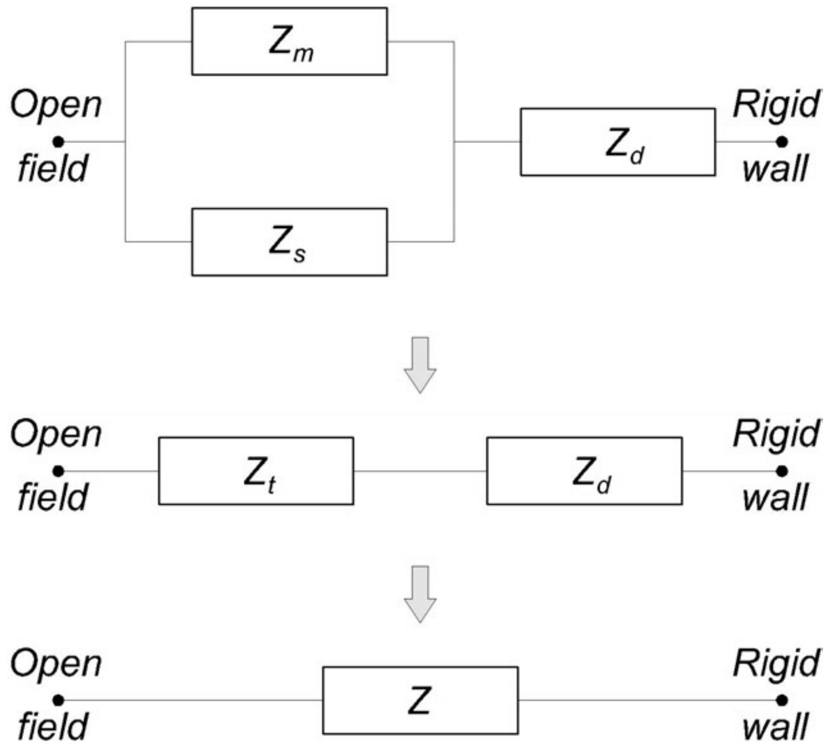

Figure 5. Schematics of the simplifications in the perforated panel impedance model.

The impedance of the porous material placed against a rigid wall Z_f can be computed as:

$$Z_f = Z_1 \coth\left(ik_1 L_f\right) \quad (9)$$

where k_1 and Z_1 are respectively the wave number and the specific impedance of the porous material, and L_f is the porous material thickness. If the material is fibreglass these two quantities can be calculated when its flow resistivity r is known, using the Mechel model (see [2]). In this case, defining

$$b = \frac{\rho_0 f}{r} \quad (10)$$

they are equal to:

$$Z_1 = Z_0\left(1 + 0.0563\, b^{-0.725} - i\, 0.127\, b^{-0.655}\right)$$
$$k_1 = k_0\left(1 + 0.103\, b^{-0.716} - i\, 0.179\, b^{-0.663}\right)$$

$$\text{if}\; b \geq 0.025 \quad (11)$$

$$Z_1 = Z_0\left(1 + 0.081\, b^{-0.699} - i\, 0.191\, b^{-0.556}\right)$$
$$k_1 = k_0\left(1 + 0.136\, b^{-0.641} - i\, 0.322\, b^{-0.502}\right)$$

$$\text{if}\; b < 0.025 \quad (12)$$

6 RESULTS OF THE ANALYTICAL MODEL

We first show the accuracy of the analytical model described above by applying it to a panel studied both experimentally and numerically in [4], and comparing the results with those given by the models of Bolt [6] and Velizhanina [7].

The parameters of the panels we have studied are:

- fibreglass layer thickness $L_f = 60$ mm;
- fibreglass resistivity $r = 20273$ kg/(m^{3}s);
- air layer thickness $L_a = 40$ mm;
- frontal plate thickness $h = 10$ mm;
- distance between holes in x_1 direction $D_1 = 107.5$ mm;
- distance between holes in x_2 direction $D_2 = 177.14$ mm;
- hole diameter $\phi = 8$ mm.

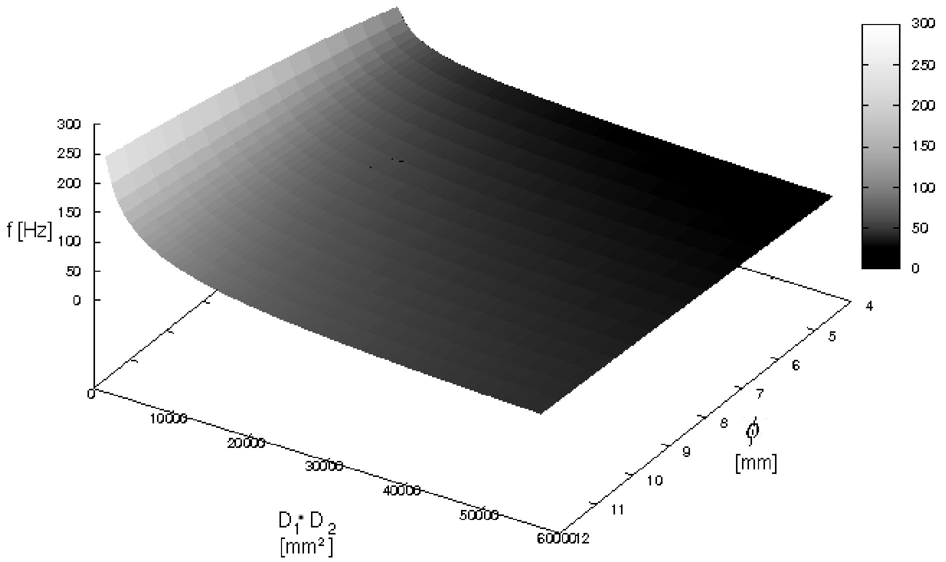

Figure 6. 3D Plot of the peak frequency in function of cell area and of the duct diameter for a panel with L_f = 60 mm and L_a = 40 mm.

The comparison is presented in the graph of Fig. 4. One can see a good agreement of the analytical model for a frequency range between 40 and 150 Hz.

A study of the perforated panels behavior as a function of the hole diameter and of the cell area is reported in the graph of Fig. 6: the resonance frequency is reported for a range of the geometrical parameters D_1, D_2, and ϕ.

7. ANALYSIS OF THE MODEL BEHAVIOR AND DESIGN GRAPHS

Figures 7, 8, and 9 summarize the panel behavior predicted by the analytical model as a function of its design parameters. The effective absorption width l_b is defined as the frequency width of the portion of the absorption curve in which the absorption coefficient is greater or equal of 0.1.

Figures 10, 11, and 12 are examples of design graphs for perforated panels. In these figures we have assumed fixed values for h, L_a, and L_f; given a frequency of interest, from Figure 10 one obtains the cell size, and from Figure 11 and 12, respectively, one obtains the effective absorption width and the maximum absorption coefficient. The equations described in section 5 allow one to construct similar graphs for any desired values of h, L_a, and L_f.

8 CONCLUSIONS

The analytical models that one can find in the literature to design perforated panels give no adequate results at low frequencies. A new model for the design of this kind of device at low frequency (40-150 Hz) has been presented, and it has been compared with a coupled FEM simulation.

While the FEM analysis has a high computational cost, requiring powerful and expensive software, the proposed model is very simple to implement into a code (less than 140 code lines are necessary in FORTRAN), or into an OpenOffice.org spreadsheet, and only some seconds are necessary to obtain numerical results by using this model. From the analytical model, three easy to use design graphs are presented. They allow one to design a panel knowing the frequency that must be corrected.

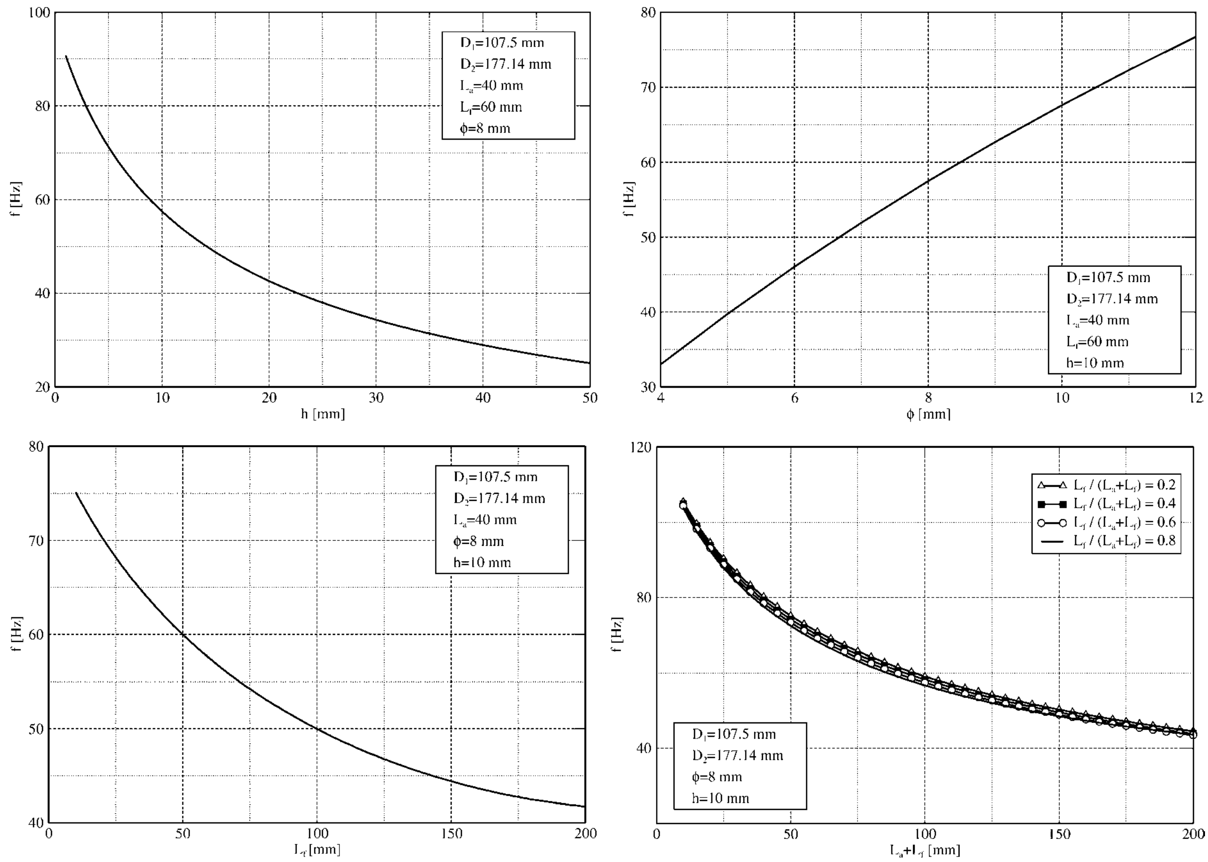

Figure 7. Resonance frequency as a function of design parameters

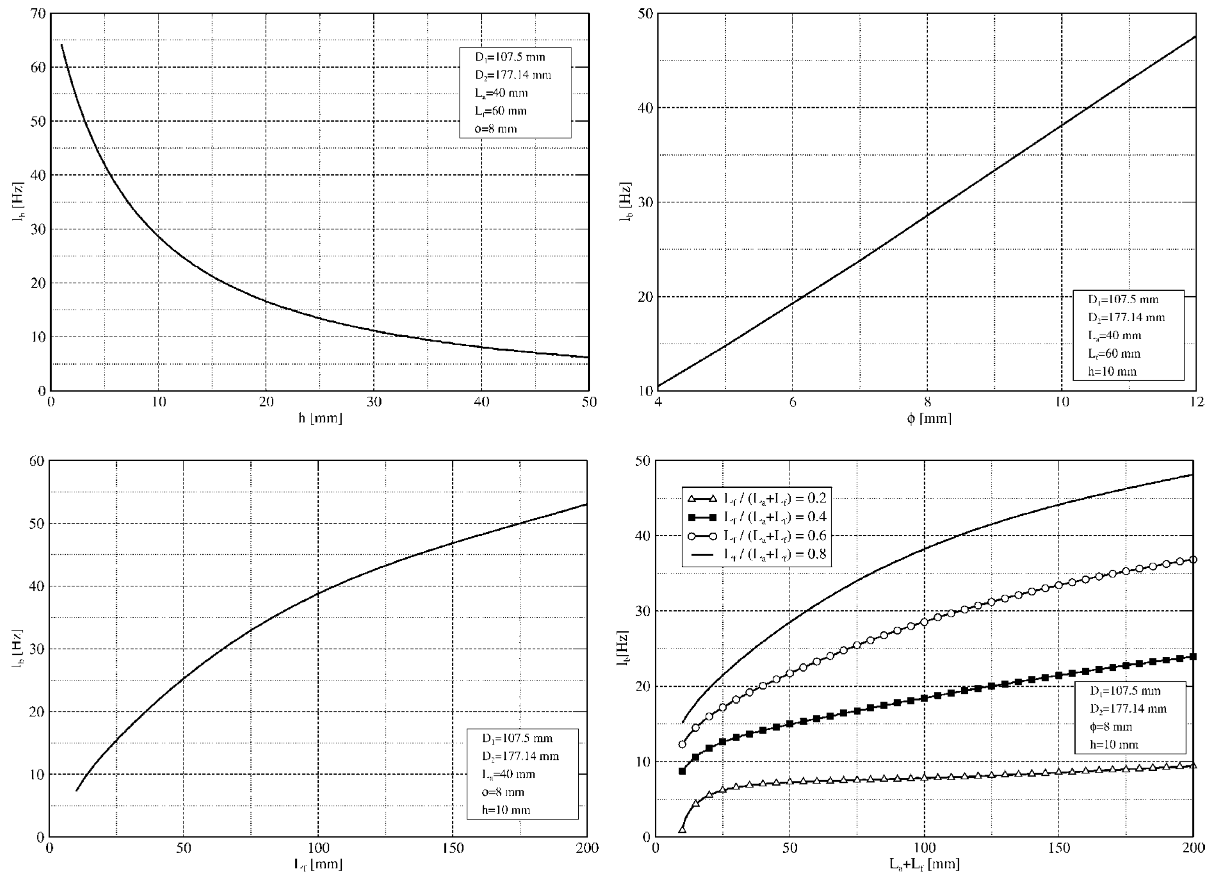

Figure 8. Effective absorption width as a function of design parameters.

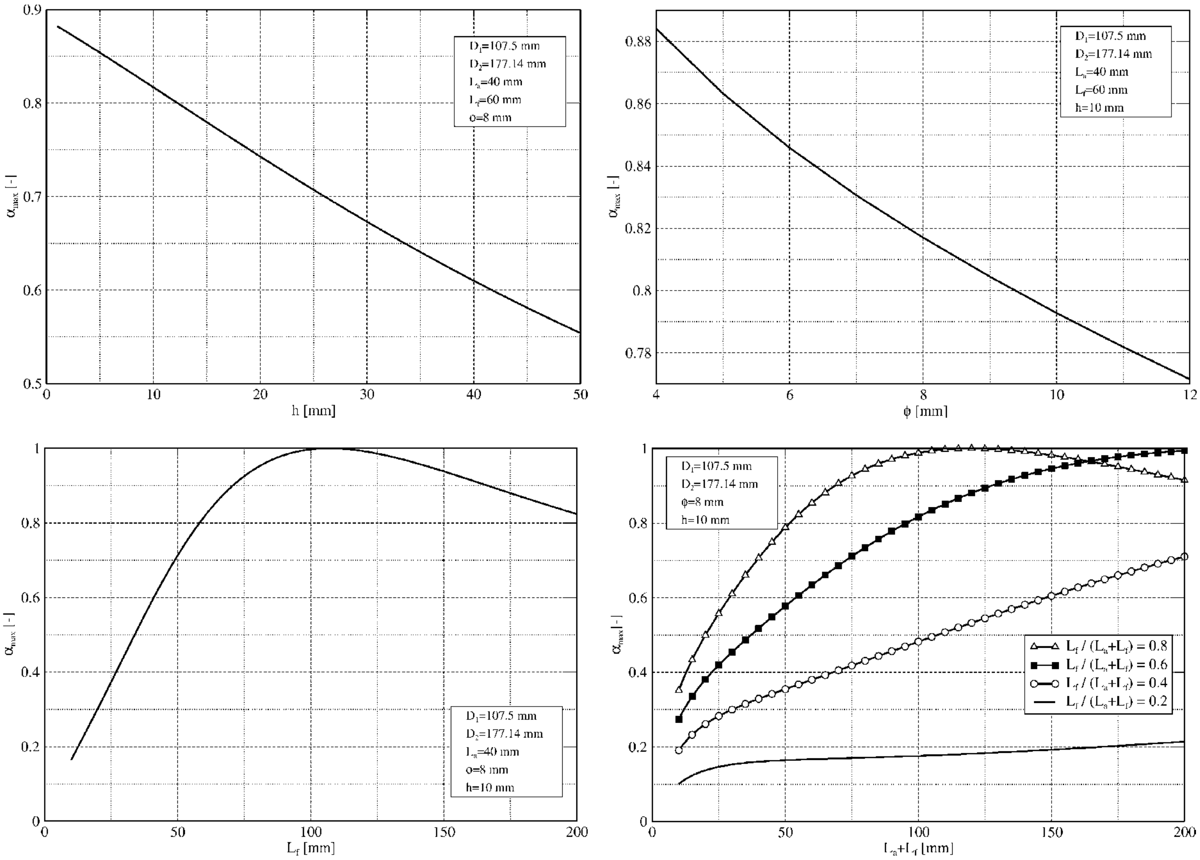

Figure 9. Max absorption coefficient as a function of design parameters

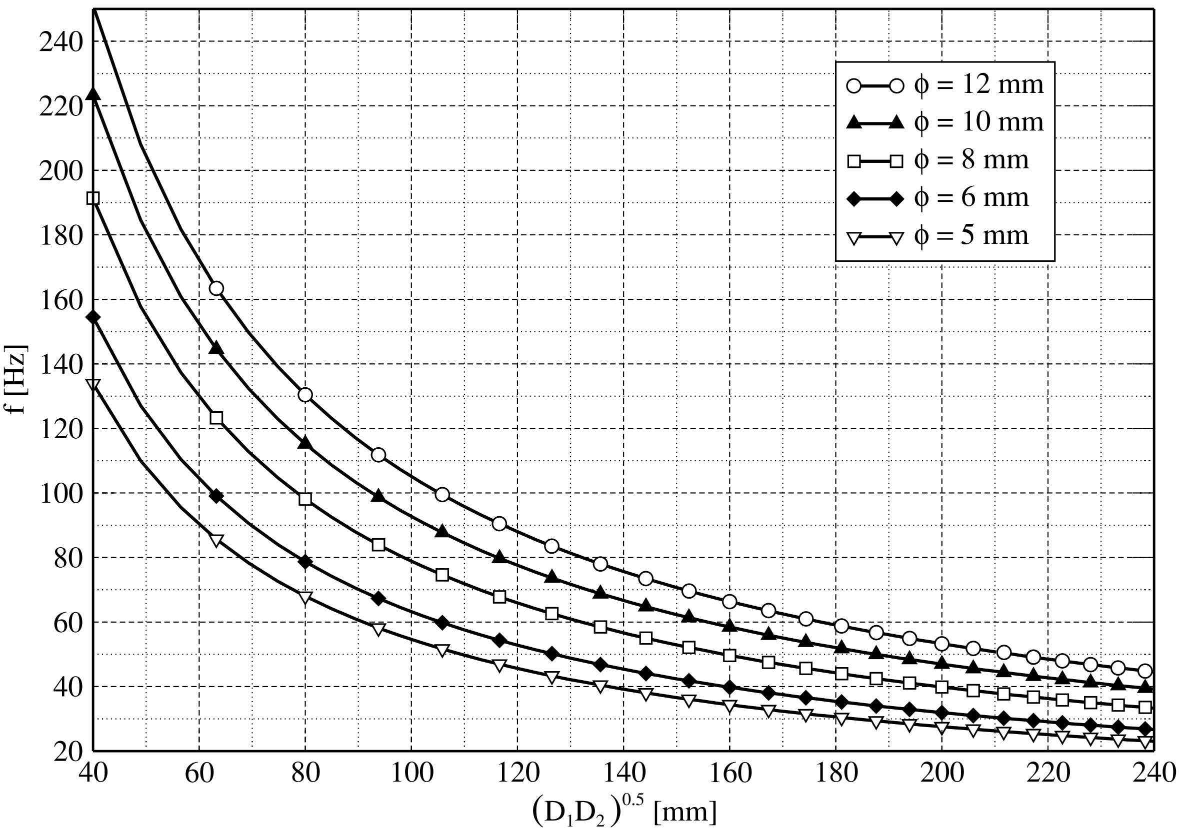

Figure 10. Resonance frequency as a function of the square root of the cell area for a panel with L_f = 60 mm, L_a = 40 mm, h = 10 mm

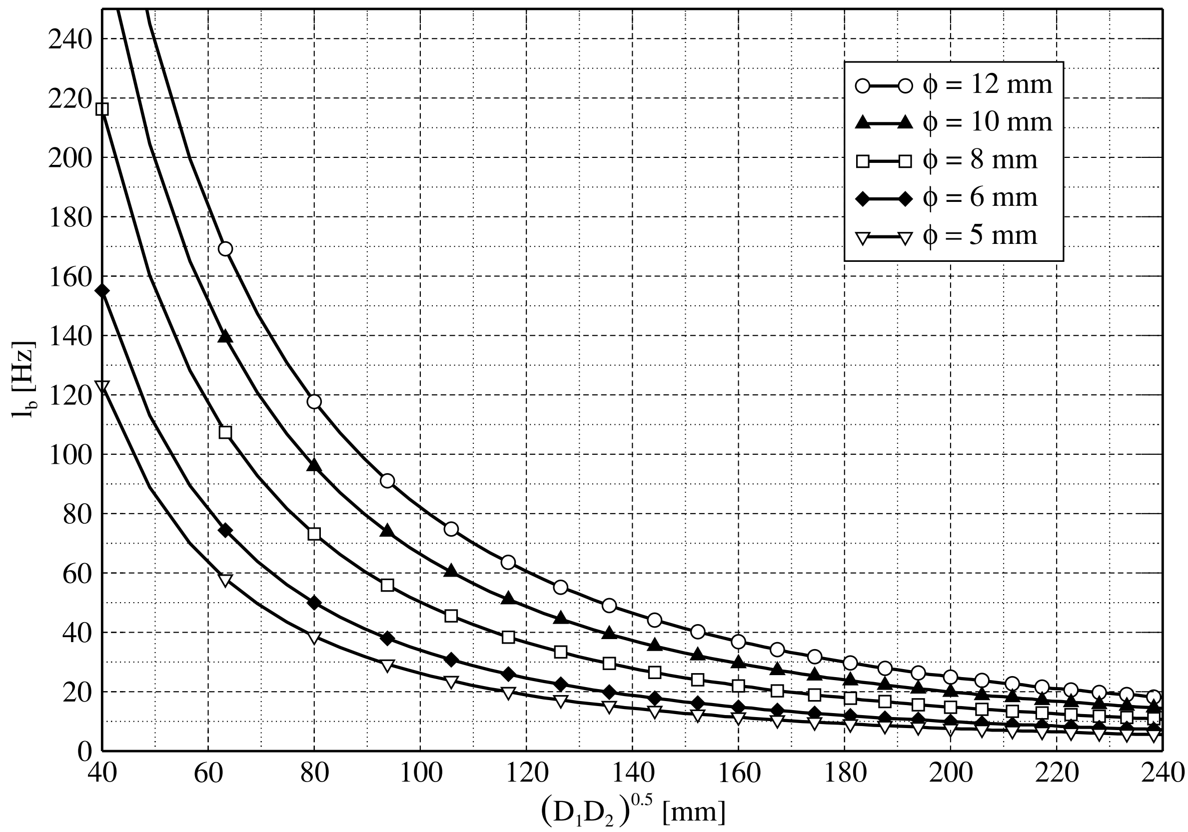

Figure 11. Effective absorption width as a function of the square root of the cell area for a panel with L_f = 60 mm, L_a = 40 mm, h = 10 mm

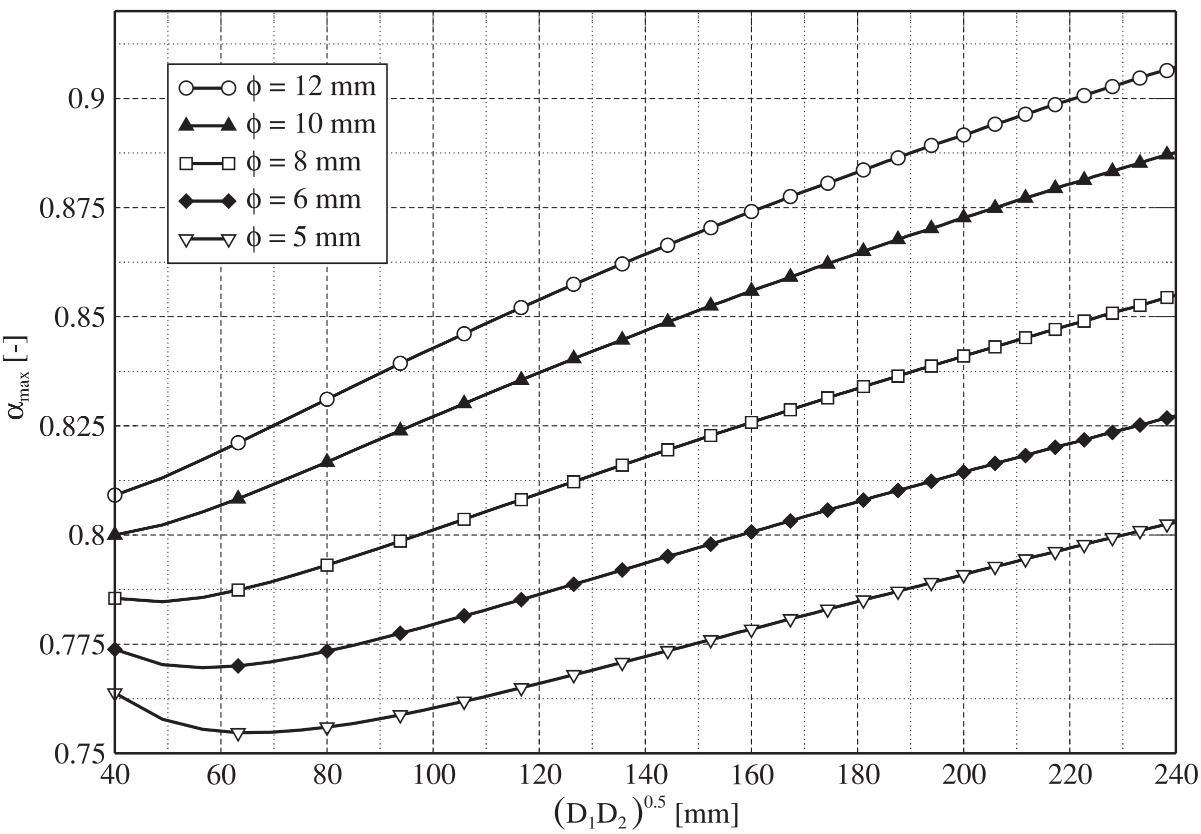

Figure 12. Maximum absorption coefficient as a function of the square root of the cell area for a panel with L_f = 60 mm, L_a = 40 mm, h = 10 mm

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SHOT FOR COMPLAINING

In Sydney a man has been shot dead in the chest after confronting a neighbour who was blasting loud music from his house. Stephen Holmes, 41, reportedly went to the neighbour's house to complain about the noise – something that has apparently been a constant problem for the neighbours. The heated exchange of words then took a violent turn as Holmes took a single shot to his chest.

NOISE MITIGATION SUIT SETTLED FOR $127 MILLION

The Metropolitan Airports Commission in Minnesota has had FAA approval to spend $127 million to settle its noise-mitigation lawsuit with the communities of Eagan, Minneapolis and Richfield. Eagan's city council voted unanimously to settle a lengthy lawsuit against the MAC, allowing hundreds of city residents to insulate their homes against noisy overhead traffic to and from Minneapolis-St. Paul International Airport. City leaders from Minneapolis and Richfield also voted in favour of the settlement, according to the Pioneer Press. Homeowners can have central air conditioning and $4,000 for a specific list of noise-dampening improvements including air conditioning.

NO NONSENSE APPROACH TO NOISY CHILDREN

Police say a Berkeley (California) man who routinely screams at children playing in a park crossed from free speech into “malicious” territory when he was arrested on a charge of disturbing the peace. Art Maxwell, who lives next to the Becky Temko Tot Park on Roosevelt Avenue, has engaged in a months-long battle with children and their parents over the noise they make while using the park. Parents say Maxwell intimidates children and their parents, by screaming at them, threatening them with arrest, videotaping them and playing obscene rap music when they use the park. At the end of November a man using the park called police and made a misdemeanour citizen’s arrest after Maxwell started throwing and breaking bottles in his backyard and swearing at him in the park through the fence, said Berkeley police Lt. Wes Hester. “(Maxwell’s) level of behaviour exceeded the norm,” he said. “The difference between free speech and being arrested for disturbing the peace has to do with the maliciousness of the words he was using, the profanity.

LEAF BLOWER BANS COST MONEY

Another US city is about to vote on a completely banning leaf blowers. This time it is Cambridge, Massachusetts. While silence lovers are for it, there are costs attached. A spokesman from the city’s Public Works Department pointed out, if there’s a total ban, “You have potential workers compensation claims from injuries due to strain from continuous raking … the costs of contracts would increase … you couldn’t maintain the expected level of maintenance at a golf course without this equipment.”

GOOD NEIGHBOURS MAKE GOOD FRIENDS

An upstairs-downstairs noise row in Chicago has simmered for months, and finally ended in court. In an apartment block, Emerick and Saenz live above Sago and Pourmelid. From below, accusations of excessive noise, from above, counter claims that the accusations are ‘extreme’ and ‘outrageous’. Verbal disputes over noise, police involvement, over an eight month period. Now Emerick and Saenz want the court to stop their downstairs neighbours complaining and making unreasonable demands, such as, they should not do laundry after 9pm, nor play loud video games, nor wear hard soled shoes, nor drop things on the floor. And, because its America, Emerick and Saenz would also like $150,000 for ‘humiliation and emotional distress.’

NO NIGHT-TIME BAN IN BANGKOK

A Thai court has rejected a petition from residents living near Bangkok’s new Suvarnabhumi Airport to seek a ban on night time flights to ease noise pollution. The court dismissed the request from 359 people who demanded the year-old airport be shut from 10:00 pm to 5:00 am. It said the seven-hour ban on night flights would affect 100,000 passengers and nearly 170 flights on average per day, arguing it would go against the kingdom’s commitment to becoming a major aviation hub. “It will affect Thailand’s global aviation commitment, and could trigger retaliation” from airliners and travellers, the court ruling said. With capacity to handle 45 million passengers a year, Suvarnabhumi opened in September 2006 with Thailand hoping it would establish Bangkok as Southeast Asia’s pre-eminent air hub. But the three-billion-dollar facility has been plagued by problems ranging from cracks in the runways to complaints about safety and sanitation. Original assurances reported in NVWW, that local residents would be protected from and/or compensated for, noise pollution, appear to have come to nothing.