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Assessment of active electroacoustic absorbers as low-frequency modal dampers in rooms

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In small to medium size rooms, natural room modes may severely strengthen the annoyance of low frequency noises, such as traffic or equipment noise. Moreover, there is an actual technological gap in the state-of-the-art concerning low-frequency treatments, and the only potential solutions basically take the form of heavy and bulky bodies.

With a view to develop such low-frequency solutions (with compactness requirements), the concept of "shunt loudspeaker" has already been demonstrated to provide controllable electroacoustic dissipation to tackle one or a few modes around the loudspeaker resonance (in the low frequency range). The present paper extend this study to the concept of "electroacoustic absorbers", based on the connection of loudspeakers to individual synthetic active electric loads, capable of achieving broadband sound absorption around the loudspeaker resonance, up to almost 200 Hz. This paper will especially investigate the optimization of the design, as well as discussion on practical spatial arrangement of electroacoustic absorbers, in the context of an experimental assessment in a test room, so as to provide significant damping of the low-frequency acoustic resonances in the [20-100 Hz]. The resulting modes attenuations range from -1dB to -14 dB along the bandwidth of interest.

1 INTRODUCTION

The inadequacy of conventional passive soundproofing treatments for the low-frequency range[1], as well as the significant progress beyond the state-of-the-art of active noise control

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observed in the recent years[2], has motivated the development of active loudspeakers with tunable acoustic impedances, such as the electroacoustic absorber concept[3]. Basically, passive means would be preferred since they are robust, cheap, easily available, simple to implement, but their efficiency does not always match with the frequency range of interest. Above all, they are mono-functional, i.e. they can only absorb, or reflect, sound energy. On the other hand, ANC methods are known to be effective in the low-frequency range, more compact than passive means, and able to counteract the noise selectively[4]. Good performances of ANC can be observed in case of simple geometries or with stationary tones, but actual situations with broadband noise, or involving a three-dimensional sound field, where disturbing noise is not easy to predict, are quite difficult to counteract with active means. In addition, the number of required secondary sources quickly becomes prohibitive and the distributed control algorithms may become complicated to implement.

In a recent paper[3], the concept of electroacoustic absorber has been proposed as a practical alternative to low-frequency sound absorption. It consists of an electrodynamic loudspeaker connected to a passive or active shunt electric network, the whole acting as an absorber of sound in the low-frequency range, without any feedback loop on acoustic quantities. The concept was assessed on experimental prototypes that have been thoroughly studied in laboratory conditions, in one-dimensional impedance tubes where the acoustic performances showed almost broadband sound absorption in the [20-200 Hz] range.

In this paper, the concept of electroacoustic absorbers[3] is developed to be applied to damp the low-frequency resonances in an actual room. This practical example aims at showing the feasibility and performance of a set of electroacoustic absorbers controlled by a synthetic electrical load in a three-dimensional situation. The problem of the interaction of such electroacoustic absorbers with the natural modes of the room is not only of practical, but also of theoretical interest. Although it will mainly focus on the practical absorption of low-frequency modes in a room, Kuttruff[1] provides more thorough details on wave theory for room acoustics, that is implicitly referred to in the following study.

2 PROBLEM STATEMENT

2.1 Description and low-frequency acoustic characterization of the experimental room

The present work considers a room with parallel pairs of walls, each pair being perpendicular to the 2 others. Typical living rooms, however, are more or less irregular in shape, partly due to the furniture that forms part of the room boundaries. The studied geometry is however a good example to understand the low-frequency distribution of acoustic energy in enclosed spaces, and to assess the potential performances of active electroacoustic absorber to damp the resulting resonances in the low-frequency range. Moreover, even if the wall are mainly composed of concrete here, the natural acoustic absorption of such reverberant room is finally well representing the general case of actual rooms for the low-frequency range, since none of the state-of-the-art passive treatments are actually absorbing below 100 Hz (α generally of the order of magnitude of 0.1 within this frequency range).

Thus, the experiment is carried out in a technical room used to store containers at EPFL (see Figure 1). This is a hard-walled rectangular room with a total area of 94.3 m2 and a volume of 59.3m3 (3 m x 5.6 m x 3.53 m). From a frequency point of view, the object of modal damping is to reduce the amplitude of mode resonances, by presenting specific acoustic resistance at specific room resonance frequencies. With a view to specifying the low-frequency acoustic treatments that could achieve such room modal damping, a first assessment of room modes has been undertaken, with a sound source located at corner 1 (see Figure 1 for corner identification) delivering a band-limited noise, and the sound pressure level being measured at corners 2 and 8 of the room. Figure 2 allows the identification of the different resonances frequencies in the room.

This figure shows the frequency distribution of modes, as well as their properties in terms of quality factor, thus relative amplitudes. Figure 2 especially shows that the main frequency area where sound absorption should be achieved to reduce the main annoyance is mainly the [50-100 Hz] band. This result provides then specification for setting the acoustic performances of electroacoustic absorbers, according to the design methodology detailed in Ref[3].

2.2 Description and setting of the electroacoustic absorbers

In Ref[3], the electroacoustic absorber concept was introduced (although the denomination was already proposed by Bobber[5] in the 1970's), resulting from formal analogies between shunt loudspeakers[6] and direct impedance control[8] concepts. The electroacoustic absorber consists in the connection of an electrodynamic loudspeaker with a synthetic electric network, ensuring a dedicated electric impedance load at the loudspeaker terminals, as illustrated on Figure 3. Basically speaking, when an exogenous sound pressure p impacts the loudspeaker diaphragm, it yields a vibratory movement of the latter, thus the moving coil gains a certain velocity v, resulting in an electromotive force $e=-Blv$ (Bl being the loudspeaker force factor) at the loudspeaker electric terminals. Depending on the electric load at the terminal, at each frequency, a certain current i flows within the coil, inducing in turn a certain feedback force $F=Bli$ that reacts to the acoustic disturbance. There is then a functional relationship between the disturbing acoustic pressure p and current i in the coil, function of the mechanical and electrical impedances of the loudspeaker, and essentially of the load impedance Z_L.

Knowing the desired acoustic impedance to be assigned to the loudspeaker diaphragm (for instance $Z_a=\rho c$, where ρ is the air density and c the speed of sound) yields a target shunt electric impedance Z_L. The concept of electroacoustic absorber takes advantage of this property to identify the correct electric impedance that achieves the desired acoustic impedance. It has been shown in Ref[3] that analog electric circuits could achieve this desired function for several cases. But it has been actually impossible to achieve this with analog solutions for the studied electroacoustic configuration, so the use of a digital synthetic electric load has been preferred.

According to the modal behavior of the test-room, a specific closed-box loudspeaker has been designed, made of a Monacor SPH-300TC driver (see Thiele Small parameters in Table 1) with a cabinet of volume V_b =23 l (entitled "cubes" in the following). Another embodiment has also been realized, made of a column of 4 electroacoustic absorbers (entitled "columns" in the following), in a common volume of V_b=38 l. For each of the prototypes (cubes and columns), a specific shunt electric load has been synthesized after the methodology described by Lissek et al.[3] on a digital platform (based on National Instrument CompactRio FPGA hardware), and the achieved acoustic absorption coefficient has been assessed in an impedance tube, according to ISO 10534-2 standard. The achieved acoustic absorption coefficients are given on Figure 4.

These results show that the designed electroacoustic absorbers are ideally absorbent around the resonance frequency 70 Hz, which is the center of the [50-100 Hz] bandwidth. Then, depending on the volume that is employed, the sound absorption coefficients present different behaviors. There will then be a slight difference in performances between the cubes and the column, each having their own advantages and drawbacks. For instance, a bigger volume (column) allows a good sound absorption down to 50 Hz, whereas small cubes induces a stiffer cabinet, thus less absorption at 50 Hz. On the other hand, small cubes are much more adapted to damp several modes, since they can be readily put at corners where the higher effect of damping is expected. The following experimental study aims at showing the strategy chosen to optimize the mode damping in the room.

3 ROOM MODE DAMPING STRATEGY

A numerical model of the room has been designed on COMSOL Multiphysics, considering a parallelepiped room (dimensions 3 x 5.6 x 3.53 m^3) with ideally reflective treatments on each walls described in the preceding section. This model allows us to characterize the different modes in the room in the range [30-100 Hz]. Thus, the eigenmodes analysis reveals the sound pressure distribution for each room modes, highlighting the position of antinodes where the placement of electroacoustic absorbers should have the most important effect in terms of mode damping. These results are given in Figure 5.

According to these results, the placement of the electroacoustic absorbers has to be carefully addressed, based on the simple observation of the nodes and antinodes: the closer the electroacoustic absorber to a maximum of pressure, the higher the damping. Thus, several geometric distribution of the 4 electroacoustic absorbers have been tested, as illustrated on Figure 6. The first 3 configurations (C_1, C_2 and C_4) employ the column of electroacoustic absorbers, either disposed horizontally along the longer edge, centered in the middle of the edge, and the loudspeaker facing the ground (C_1) or the lateral wall (C_2), either vertically oriented along the diagonal (C_4). The 6 other configurations employ 4 individual cubes, located at the four corners of a same face, with different orientations (C_5, C_6, C_8, C_9).

The different configurations reveal different strategy of modes damping, depending on the type of the mode to be damped (axial, tangential, or oblique, see Table 2). For instance, if the electroacoustic absorbers are oriented along the diagonal of one face, the resulting damping will mainly be optimized for the tangential modes, whereas absorbers oriented against a face would have more effects on axial modes. The different mode attenuations provided on Table 2 illustrate these properties, showing higher mode damping for the modes which correspond to the orientation of the absorbers. For instance, configuration C_6, which present the higher global mode damping (see also Figure 7), is particularly damping the (121) mode with an attenuation of up to 13.7 dB. The other oblique mode (111) is also well attenuated (9.5 dB), and the other modes are then a bit less attenuated, down to the lower attenuation for the axial modes. To conclude, it appears that individual cubes located at the room corners present higher damping performances, and even more when they are oriented against the corner, whereas columns are almost contra-productive on the whole frequency range.

4 CONCLUSIONS

The electroacoustic absorbers represent a straightforward and still efficient solution to lower the room influence on noise at low-frequencies. A method to damp such effects has been presented. Thanks to a dedicated control of acoustic impedance in a test-room, a significant damping of natural room resonances has been achieved. The resonances have been attenuated up to 13.7 dB on a single mode, and globally up to 4.6dB on the frequency range [30-105 Hz]. Moreover, it has been shown that the electroacoustic placement (eg location but also orientation of the diaphragm) has a critical importance on the performances.

5 ACKNOWLEDGEMENTS

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6 REFERENCES

1. Heinrich Kuttruff, *Room acoustics*, Spon Press, 4th ed. edition (2000).

2. Manuel Melon, Philippe Herzog, Azzedine Sitel, and Marie-Annick Galland, “One dimensional study of a module for active/passive control of both absorption and transmission”, *Applied Acoustics*, **73**(3), 234-242, (2012).

3. Hervé Lissek, Romain Boulandet, and Romain Fleury, “Electroacoustic absorbers: bridging the gap between shunt loudspeakers and active sound absorption”, *J. Acoust. Soc. Am.*, **129**(5), 2968-2978, (2011).

4. Philip. A. Nelson and Stephen J. Elliott, *Active Control of Sound*, Academic Press Inc, (1993).

5. Robert J. Bobber, “General reprocity parameter”, *J. Acoust. Soc. Am.*, **39**(4), 680-687, (1966).

6. Andrew J. Fleming and et al., “Control of resonant acoustic sound fields by electrical shunting of a loudspeaker”, *IEEE Transaction on Control System Technology*, **15**(4), 689–703, (2007).

7. Marc Furstoss, Denis Thenail, and Marie-Annick Galland, “Surface impedance control for sound absorption: direct and hybrid passive/active strategies”, *J. Sound and Vibration*, **203**(2), 219-236, (1997).

Table 1 - Thiele-Small parameters of the MONACOR SPH-300TC low-medium loudspeaker.

Parameter name	Notation	Value	Unit
dc resistance	R_e	6.3	Ω
Voice coil inductance	L_e	1	mH
Force factor	Bl	10.3	NA^{-1}
Moving mass	M_{ms}	68	Kg
Mechanical resistance	R_{ms}	3.24	N.s.m^{-1}
Mechanical compliance	C_{ms}	0.85	mm.N^{-1}
Effective area	S	495	cm^2
Natural frequency	f_0	23	Hz

Table 2 – description of the different eigenmodes of the room between 30 Hz and 105 Hz, and levels of attenuation for each electroacoustic absorber configuration in dB (the last line gives also the global sound reduction on the whole range [30 – 105 Hz]).

Type	$n_xn_yn_z$	frequency Hz	C_1 dB	C_2 dB	C_4 dB	C_5 dB	**C_6 dB**	C_8 dB	C_9 dB
Axial	010	30.4	-0.6	-1.7	-4.6	-1.5	**-2.2**	-1.8	-0.9
Axial	001	48.2	-1.0	-0.8	-0.8	-0.7	**-0.7**	-0.8	-1.0
Axial	100	56.6	-2.6	-1.5	-2.8	-6.0	**-5.5**	-3.9	-5.0
Tangential	011	56.9	+2.0	+2.0	0	+2.0	**-0.8**	+0.2	+2.0
Axial	020	60.7	-3.0	-4.0	-0.1	-3.0	**-2.9**	+2.2	+1.7
Tangential	110	64.3	-1.8	-3.0	-8.5	-7.0	**-5.3**	-5.6	-4.5
Tangential	101	74.4	-1.8	-1.7	-10.5	-11.7	**-9.1**	-12.2	-8.6
Tangential	021	77.5	-5.5	-5.5	0	-4.0	**-5.7**	-4.3	-6.2
Oblique	111	80.3	0	0	-3.3	-6.8	**-9.5**	-8.6	-9.0
Tangential	120	83.0	-4.8	-4.8	-11.4	-7.9	**-8.9**	-8.5	-9.4
Axial	030	91.0	-2.4	-2.4	-4.1	-3.9	**-3.2**	-5.8	-4.7
Oblique	121	96.0	-5.6	-5.6	-5.6	-7.1	**-13.7**	-10.0	-13.6
Axial	002	96.3	-11.7	-10.3	-3.8	-7.6	**-6.7**	-7.6	-6.0
Tangential	012	100.9	-1.0	0	-2.6	-7.1	**-7.5**	-3.8	-8.2
Tangential	031	103.0	-3.9	-3.2	-1.8	-5.2	**-5.4**	-7.2	-6.9
Global damping			-2.1	-2.0	-2.9	-3.8	**-4.6**	-3.5	-3.3

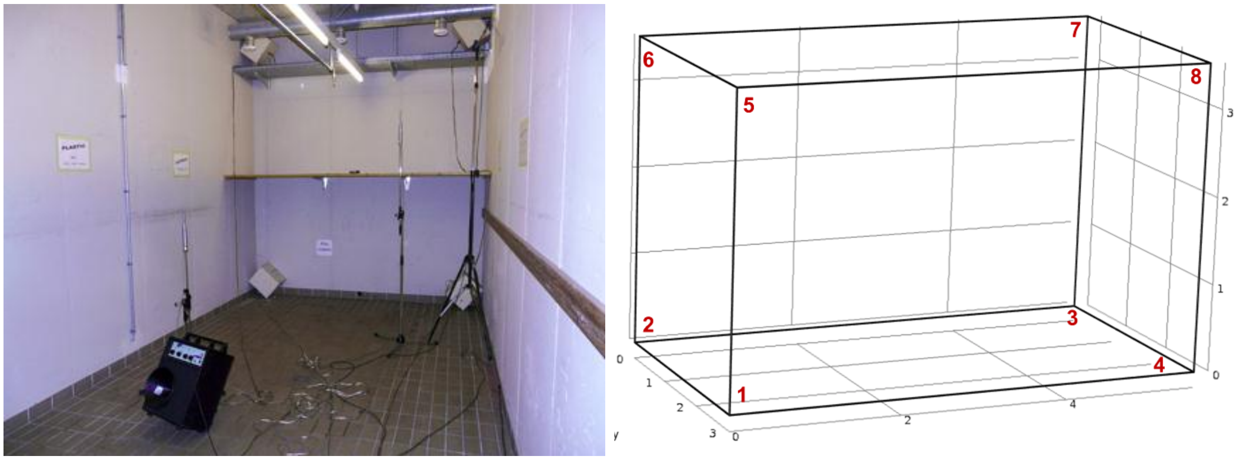


Fig. 1: picture of the experimental test-room and CAD model of the room, with identification of corners.

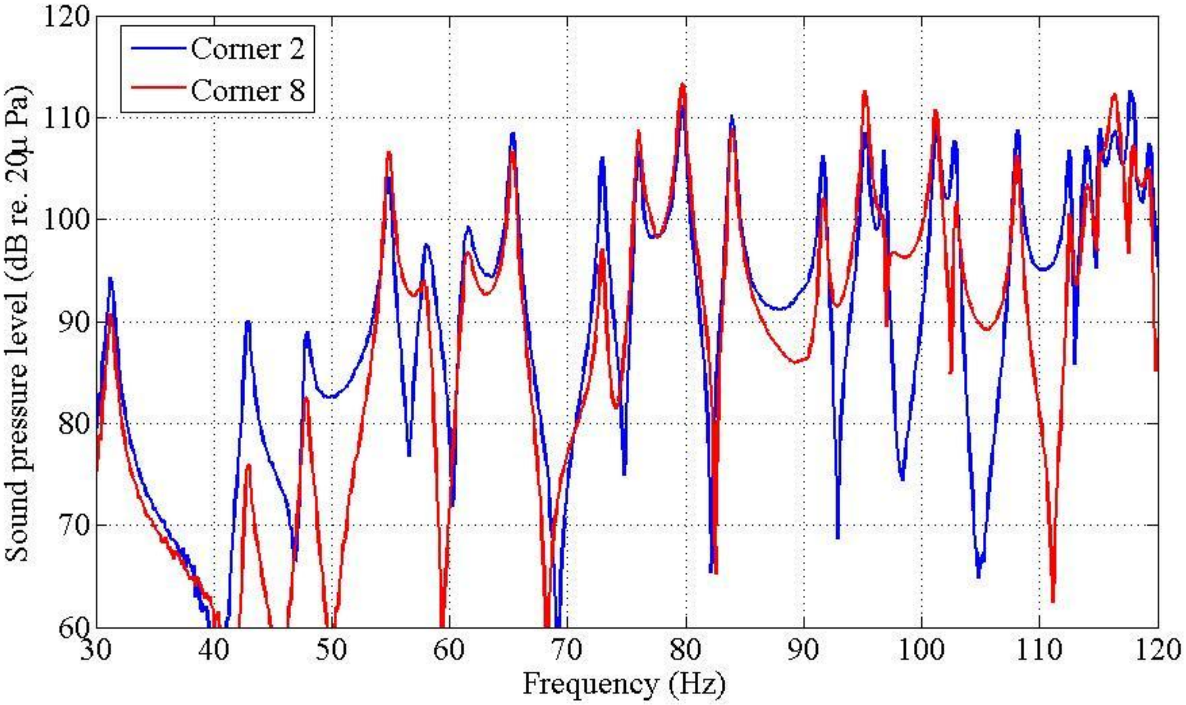


Fig. 2: identification of the resonance frequencies in the experimental test-room

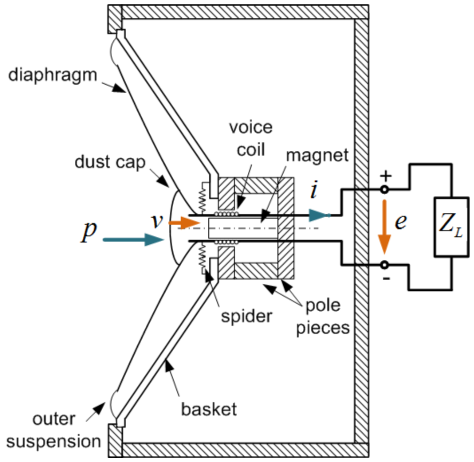


Fig. 3: General description of an electroacoustic absorbers. Z_L denotes any electric load, be it a passive shunt dipole, an active analog network, or a digital synthetic load.

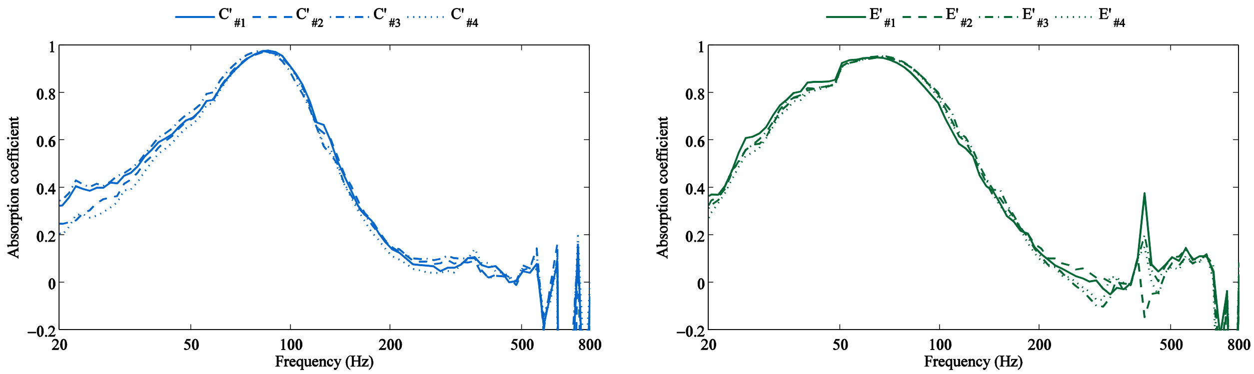
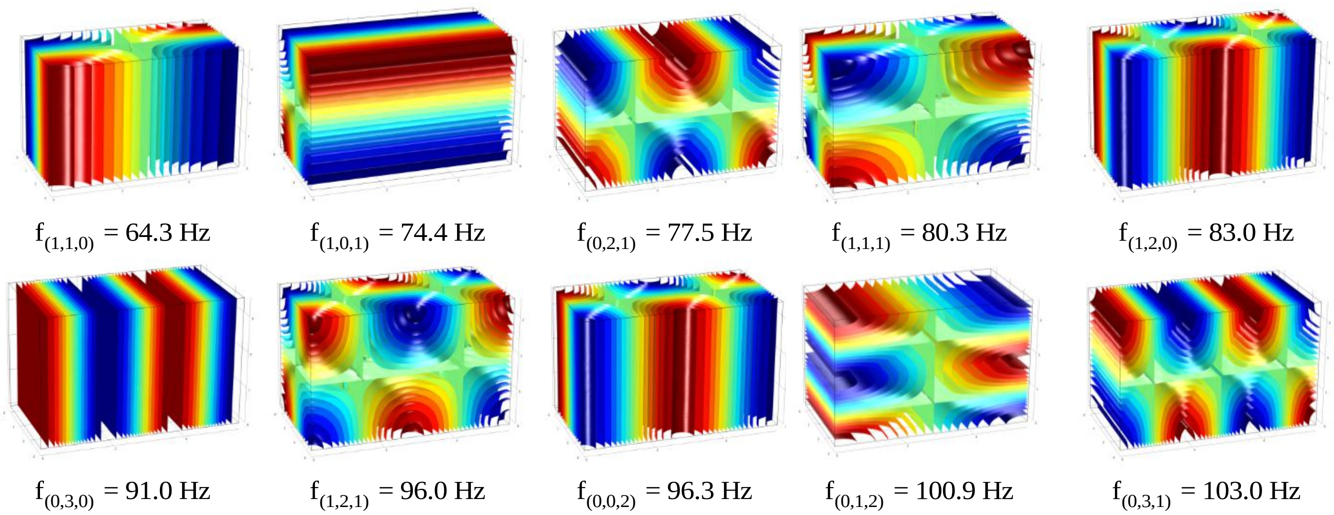
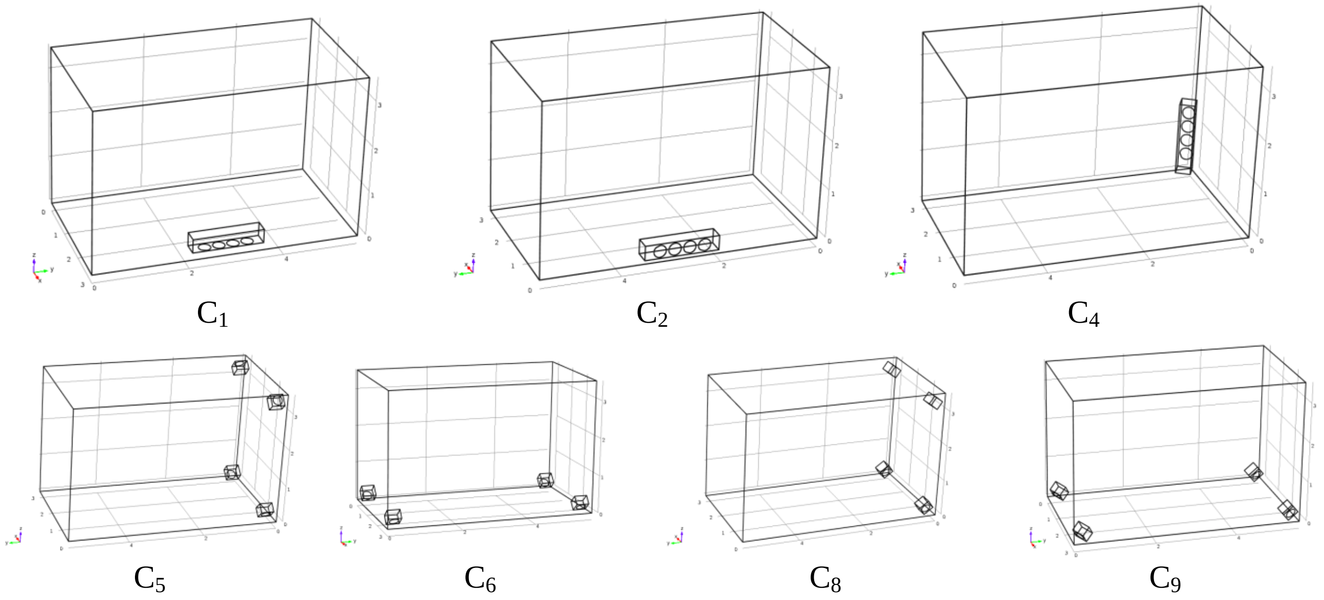


Fig. 4: Measured absorption coefficient of the loudspeakers when connected to the synthetic loads. The results for the individual cubes (V_b=23 l) are shown on the left and for the column (V_b=38 l) on the right.



$f_{(1,1,0)}$ = 64.3 Hz
$f_{(1,0,1)}$ = 74.4 Hz
$f_{(0,2,1)}$ = 77.5 Hz
$f_{(1,1,1)}$ = 80.3 Hz
$f_{(1,2,0)}$ = 83.0 Hz
$f_{(0,3,0)}$ = 91.0 Hz
$f_{(1,2,1)}$ = 96.0 Hz
$f_{(0,0,2)}$ = 96.3 Hz
$f_{(0,1,2)}$ = 100.9 Hz
$f_{(0,3,1)}$ = 103.0 Hz

Fig. 5: structure of the rooms eigenmodes between 65 Hz and 105 Hz. The nodal surfaces are colored in green and the antinodes in blue or red



C_1 C_2 C_4

C_5 C_6 C_8 C_9

Fig. 6: different configurations of electroacoustic absorbers studied. Configurations C_1, C_2 and C_4 correspond to "columns" of electroacoustic absorbers (V_b=38 l), whereas configurations C_5, C_6, C_8 and C_9 correspond to individual "cubes" of electroacoustic absorbers (V_b=23 l).

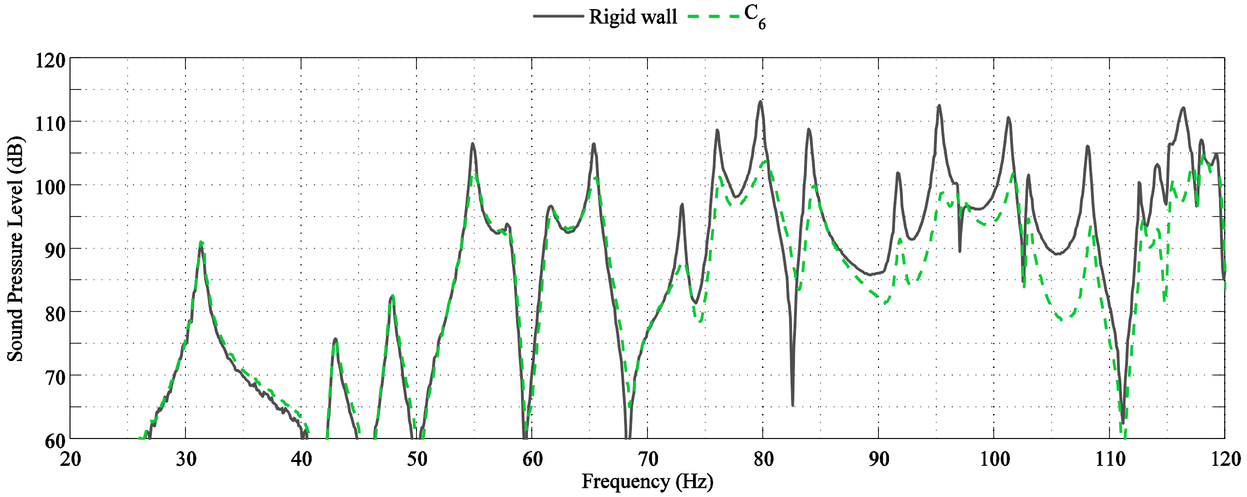


Fig. 7: achieved sound mode attenuation in configuration C6 compared to the baseline configuration (no electroacoustic absorber)