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2013 | INNSBRUCK | AUSTRIA
15.-18. SEPTEMBER 2013
NOISE CONTROL FOR QUALITY OF LIFE

Development of electroacoustic absorbers as soundproofing solutions for an industrial ventilation

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ABSTRACT

In many industrial environments, room and cavity modes may severely strengthen the annoyance of low frequency noises, such as in ventilation equipments. Moreover, there is an obvious technological gap in the state-of-the-art relative to low-frequency soundproofing, and the only potential solutions basically take the form of heavy and bulky bodies, that can be practically impossible to implement in real situations.

With a view to reducing the low-frequency annoyance, in the range of 30 Hz, resulting from the general ventilation system of the CHUV (University Hospital of Vaud Canton) in Lausanne, a prototype of electroacoustic absorber has been developed, consisting of closed-box loudspeakers connected to electric shunt resistances, acting as efficient low-frequency sound absorbers. A numerical model has first been developed and challenged with in-situ measurements. Then the electroacoustic absorber design has been optimized, and theoretical performances have been verified.

Keywords: Electroacoustic absorber, modal damping, low-frequency noise annoyance reduction

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 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 selectively4. 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

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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 is considered for application to damp the low-frequency resonances in a ventilation duct, resulting in high sound pressure levels in the very low frequency range (around 30 Hz). First, a presentation of the actual situation, with respect to noise annoyance, is given, followed by a presentation of the methodology for designing the electroacoustic absorbers prototypes. Preliminary results of this design phase are then presented.

2. PRESENTATION OF THE ACTUAL SITUATION

2.1 Description

In the frame of the construction of new buildings at the University Hospital of Vaud Canton (CHUV) in Lausanne, an expertise performed by Ecoacoustique sa, a Lausanne-based acoustic consultants office, raised a significant low-frequency noise issue at the exhaust of a group of ventilation systems. The noise sources considered here are ventilation groups, located inside 4 parallel ventilation ducts of cross section of about 12 m^2 and overall length of about 21 meters, which radiate noise outside through wide metallic vent covers. A scheme of one of the 4 ducts is presented in Figure 1.

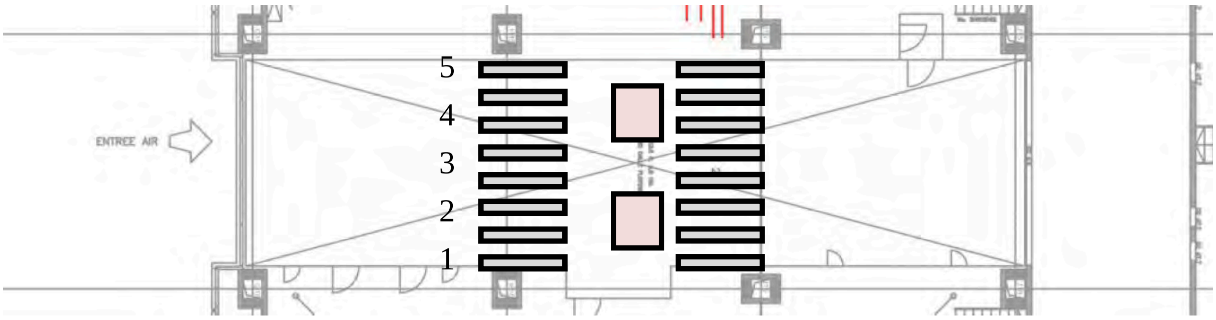

Figure 1: scheme of the ventilation duct, with indicative positions of measurement points 1-5 (grey rectangles represent vertical baffle absorbers, and the two pink squares represent the ventilation groups)

In this situation, the noise sources are composed of two ventilation groups, located nearly in the middle of the duct. Two rows of acoustic baffles are illustrated in Figure 1, which can actually be considered as high-frequency sound absorbers. Different grids and filters are also present along the ventilation duct, but not represented here for sake of simplicity.

The ventilation groups are composed two Centrimaster GT-5[3] centrifugal fans that deliver air flows up to 50 m^3/s and pressure up to 3’300 Pa. The maximum sound power level of a single fan system is specified by the manufacturer as 98.9 dB(A), as illustrated on Figure 2.

[3] http://www.flaktwoods.com

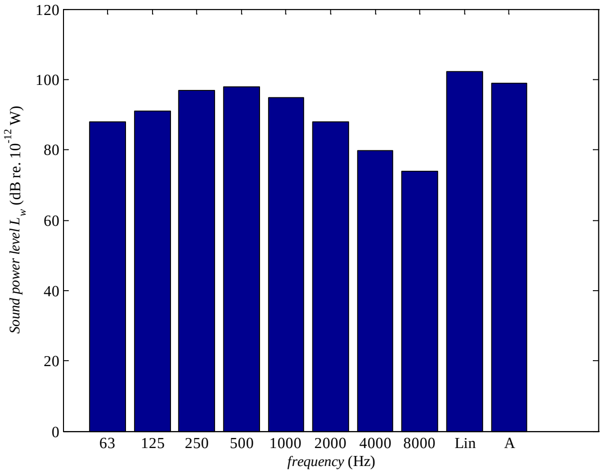

Figure 2: sound power levels of the ventilation group as specified by the manufacturer

2.2 Sound pressure level outside the building

The first measurement step consisted in assessing the noise levels outside the building, to identify the main spectral characteristics of the source. A first measurement of noise outside the building, with a Norsonic Nor118 sound pressure level-meter, gave the results illustrated in Figure 2.

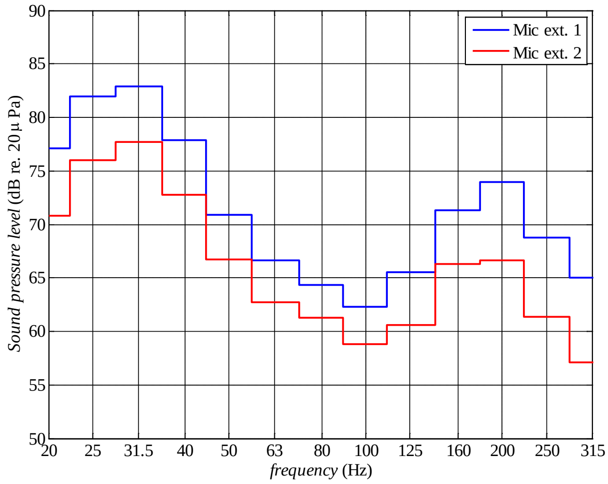

Figure 3: sound pressure levels (by 1/3 octave bands) measured outside the duct, 1 meters in front of the duct cover (blue line) and 5 meters, slightly off the duct cover (red line).

The measurements are given here as linear rather than A-weighted sound pressure levels, with a view to better highlighting the important emergence of the very low-frequency noise. This choice is also legitimated by several recent research outcomes on low-frequency noise assessment, pointing out the weaknesses of usual perceptual weighting to actually account for the effects of low-frequency noise on people [4].

The low-frequency emergence in the third octave band of 31.5 Hz gives an important indication on the nature of the noise annoyance. Assuming that the ventilation groups are not more efficient sound radiators at 31.5 Hz than above (even though the manufacturer does not provide any sound power level below 63 Hz), we consider the hypothesis that this noise emergence could be consecutive to a resonant behavior of the duct itself . Further measurements should then investigate the modal nature of the sound field in the duct in this frequency range.

2.3 Sound pressure levels inside the duct

In this step we did not have the possibility to interrupt the engines for a long period of time, allowing us to envisage a full characterization of the modal behavior inside the duct. We decided then to do the measurement inside the duct, the engines being the actual sources of noise. Also, a limited number of points were accessible for performing the measurement. These points are represented with numbers 1 to 5 in Figure 1.

Figure 4 presents the measurements of sound pressure levels at these 5 points inside the duct. The measurements have been performed with a 01dB Symphonie dual-channel sound and vibration measurement system, and with a Bruel and Kjaer Type 4198 outdoor microphone. The figure to the left represents the narrow-band measurements (with a 1.465 Hz resolution) of sound pressure levels for each of the measurement positions. The figure to the right gives the measured sound pressure levels at a specific frequency f_0=29,3 Hz as a function of the measurement point along the line [1-5].

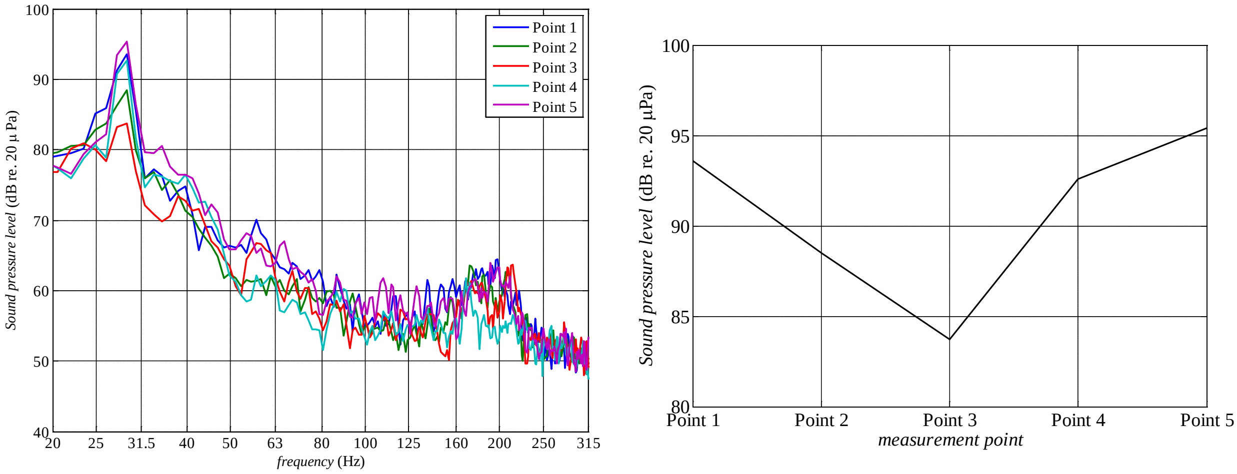

Figure 4: sound pressure levels (dB lin re. 20 μPa) measured inside the duct (left: narrow-band noise spectra at the different measurement positions [1-5] inside the duct; right: distribution of sound pressure levels along the line [1-5] at 29.3 Hz).

These measurements show a sound pressure peak level at f_0=29.3 Hz, which first indicates a potential influence of a duct resonance. It is noticeable that the measured sound pressure level slightly changes with the position. The curve on the right clearly shows a drop of sound pressure levels in the middle of the duct (almost 10 dB of difference at point 3 compared to the extreme points 1 and 5), confirming the hypothesis of a duct resonance.

Based on this observation, the development of a specific low-frequency sound absorption solution is required to envisage the reduction of the corresponding noise problem. The inadequacy of state-of-the-art solutions for such acoustic problems led us to envisage the deployment of electroacoustic absorbers as described in [3]. The design is presented in the following section.

3. DESIGN OF ELECTROACOUSTIC ABSORBERS

Based on the measurements performed in the actual ventilation duct, the design of a set of electroacoustic absorbers has been undertaken, with the primary objective to be as absorbent as possible around the mode in the third octave band of 31.5 Hz. The design of the electroacoustic absorbers is based on the design methodology presented in [3], using loudspeakers in a dedicated cabinet and a shunt resistor of optimal value to achieve ideal sound absorption at the resonance frequency of the loudspeaker.

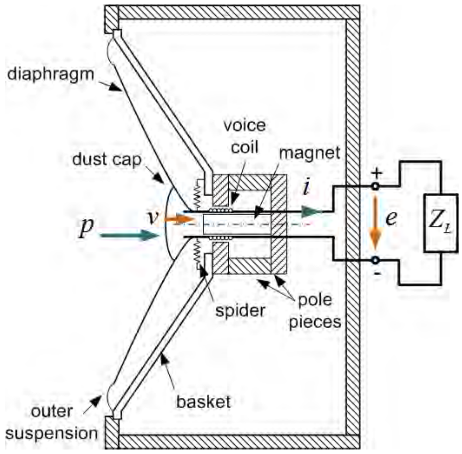

Figure 5: scheme of the electroacoustic absorber principle

In this concept, the specific acoustic impedance (ratio of pressure p over diaphragm velocity v) that the loudspeaker presents to an external sound field can be written as a function of its constitutive electromechanical components, the enclosure volume V_b and the shunt impedance value Z_L:

$$Z(\omega) = \frac{p}{v} = \frac{1}{S_d}\left[j\omega M_{ms} + R_{ms} + \frac{1}{j\omega C_{ms}} + \frac{\rho c^2 S_d^2}{j\omega V_b} + \frac{(Bl)^2}{R_e + j\omega L_e + Z_L} \right] \quad (1)$$

where variable $\omega=2\pi f$ is the angular frequency (f being the frequency), and the parameters of Eq. (1) are defined in Table 1:

Table 1: parameters used in the model of the electroacoustic absorber

Parameter name	Description	Value	Unit
M_{ms}	Loudspeaker moving mass	68.10^{-3}	kg
R_{ms}	Loudspeaker mechanical resistance	3.24	$N.s.m^{-1}$
C_{ms}	Loudspeaker mechanical compliance	$0.85.10^{-3}$	$m.N^{-1}$
S_d	Loudspeaker diaphragm surface	495.10^{-4}	m^2
Bl	Transducer force factor	10.3	$N.A^{-1}$
R_e	Coil dc resistance	6.3	Ω
L_e	Electric inductance of the coil	10^{-3}	H
V_b	Cabinet volume	38.10^{-3}	m^3
r	Medium mass density	1.18	kg/m^3
c	Sound celerity in the medium	343	$m.s^{-1}$

We used here Monacor SPH-300TC subwoofers, the Thiele-Small parameters of which are given in Table 1. The first objective is to design a cabinet capable of making the resonance frequency of the closed loudspeaker fit the targeted modal frequency in the duct (rounded at 30 Hz). Since a too small volume would lead in too much stiffness, yielding a very narrow bandwidth of the acoustic resonator, we decided to glue sets of small lead fishing bowls, giving an additional mass of M_{ma}=200 g on each loudspeaker diaphragm. The cabinet volume for each loudspeaker is then set to 38 liters, and it is filled with mineral wool with a view to increasing the equivalent acoustic compliance by a factor Γ of almost 1,2 [5].

Using a simple electric resistance R_L as shunt, it is possible to achieve optimal sound absorption at the loudspeaker resonance frequency. Since this frequency is defined as:

$$f_s = \frac{1}{2\pi}\sqrt{\frac{\rho c^2 S_d^2 C_{ms} + \Gamma V_b}{\Gamma V_b C_{ms}\left(M_{ms} + M_{ma}\right)}} \approx 29\text{Hz} \tag{2}$$

At this frequency, the optimal value of shunt electric resistance for which Z=ρc is given by:

$$R_L = \frac{(Bl)^2}{\rho c S_d - R_{ms}} - R_e \approx 0\Omega \tag{3}$$

For each electroacoustic absorber, with the identified optimal short-cut configuration, we measured the resulting sound absorption coefficient in an impedance tube, following ISO 10534-2 standard. The achieved performances are given on Figure 7.

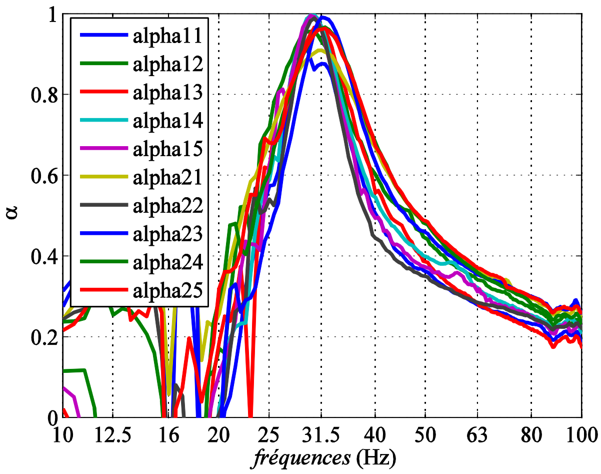

Figure 6: sound absorption coefficient measured on the 10 electroacoustic absorbers prototypes

We see experimentally that the whole electroacoustic absorbers should actually behave as perfect absorber in the 1/3 octave band of 31.5 Hz, as expected.

4. COMPUTED PERFORMANCES

4.1 Presentation of the model

With a view to assessing the acoustic performances of the developed electroacoustic absorbers, and prior to perform the actual assessment in situ, a numerical model of the duct has been designed with COMSOL Multiphysics (4.3a release). The geometry of the duct has been replicated, including the presence of two rows of acoustic baffles at two positions along of the duct.

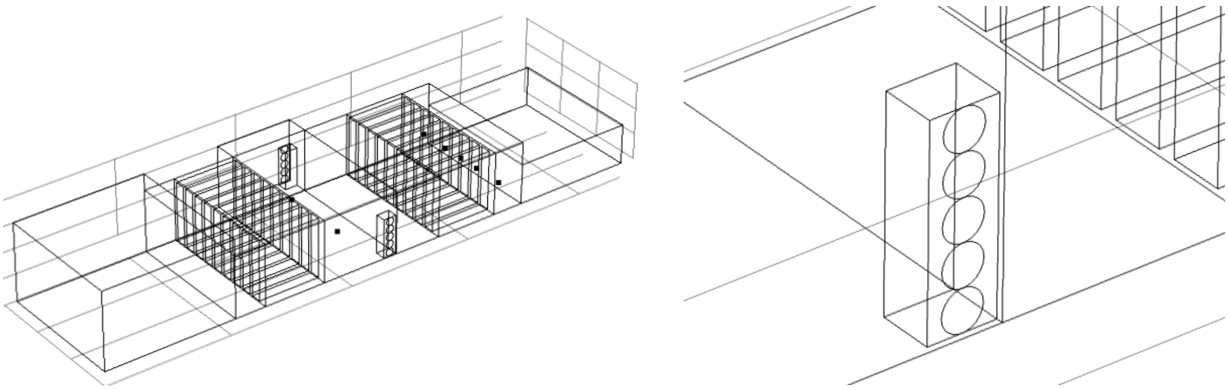

Figure 7: isometric view of the duct with 2 electroacoustic absorbers columns (left: whole model; right: zoom on the electroacoustic absorber)

In this numerical model, the duct lateral walls are assumed perfectly reflective acoustically, underestimating a potential dissipation of sound through vibration mechanisms, and two pressure release conditions have been assigned to the two remaining faces. The acoustic baffles have been modelled to consider some acoustic dissipation in the duct. Thus they have been considered slightly absorbing, the corresponding acoustic impedance being defined after [5]:

$$R = \rho c\left(1 + 0.0571.X^{-0.754} - i.0.087.X^{-0.732}\right) \tag{4}$$

where $X = \dfrac{\rho f}{\sigma}$ and $\sigma = 3.2\eta \dfrac{(1-\varepsilon)^{1.42}}{a^2}$ is a function of the air dynamic viscosity $\eta = 1.84.10^{-5}$, a being the diameter of the fibres within the porous filling (we considered here 1µm diameter) and ε the porosity (here we considered 92% porosity).

Concerning the ventilation groups, they are considered in this model as point sources of acoustic power P_0. The value of sound power is set so as to correspond to the specifications of the manufacturer (see Figure 2).

Last, we designed two columns of dimensions 0.3 m x 0.3 m x 1.5 m, with 5 circular disks distributed along one vertical face, representing 5 electroacoustic absorbers, as defined in the previous section. In COMSOL, we assigned to each circular surface a boundary condition corresponding to an acoustic impedance as a function of frequency, as defined in Eq, (1), with the values given in Table 1, and with Z_L=R_L=0 Ω.

4.2 Results

With the numerical model, a first eigenmode identification has been performed, highlighting a mode at 28.5 Hz (slightly different from 29.3 Hz, yet acceptable for this study), which can be seen on Figure 9 (left side).

Then a simulation of the room frequency response to a stationary excitation by two monopoles of constant acoustic power has been performed, the duct being first empty (apart from the noise sources and the two rows of acoustic baffles), and then with two columns of 5 electroacoustic absorbers located at two extremities of the median line of the duct. The sound pressure level distributions at the targeted modal frequency of 28.5 Hz, in the two simulated cases are given in Figure 8, and the comparison of sound spectra at a same virtual measurement position (corresponding to point 1 in the measurement phase) is given in Figure 9 (zoom in the range of 28.5 Hz).

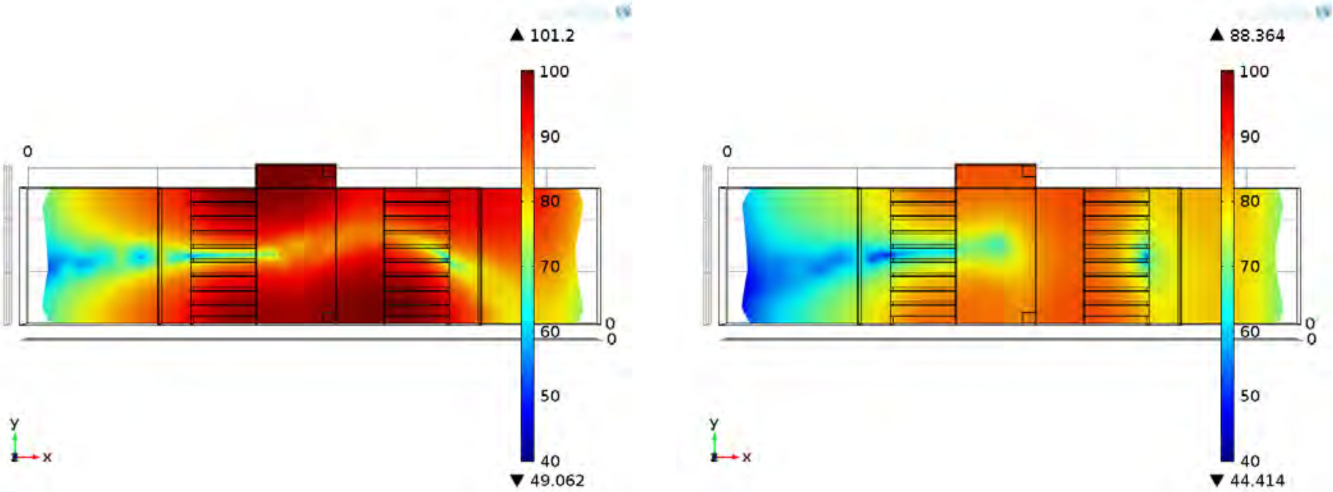

Figure 8: computed distribution of sound pressure levels in the duct at 28,56 Hz (left: without the electroacoustic absorbers; right: with two columns of 5 electroacoustic absorbers)

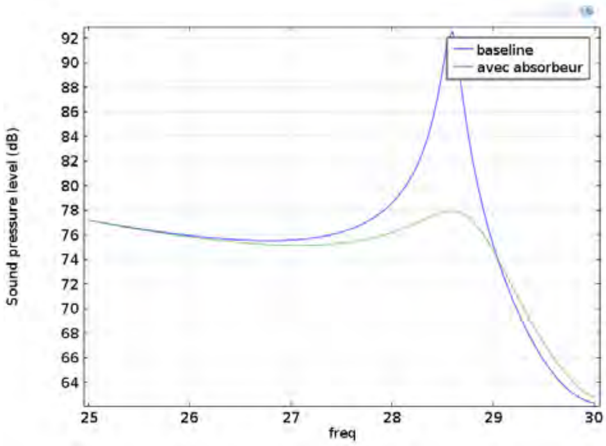

Figure 9: simulated sound pressure levels at a virtual measurement point corresponding to point 1 in the former measurement step (in blue: without electroacoustic absorbers; in green: with 2 columns of 5 electroacoustic absorbers)

Based on these simulations, the electroacoustic absorbers seem to be capable to achieve a significant sound reduction inside the duct, thus potentially yielding significant noise reduction outside the building in the very low-frequency range. In these simulations, a damping of 14 dB of the mode at 28.5 Hz is achieved, corresponding to the order of magnitude of the achievement of former experiments in a reverberant chamber [6].

5. CONCLUSION

Following the design phase of the electroacoustic absorbers, and due to a limited available time of the facility for such measurements, a very quick assessment of the performances of the electroacoustic absorbers has been undertaken at the CHUV, unfortunately with poor results that we don't present here in the paper. Although we see some reasons for such disappointing results, there remain some positive outcomes and perspectives, such as the possibility to actually damp a non-negligible part of sound energy in such a big volume (roughly 250 m^3) with less than 1m^2 surface of sound absorbers (representing even less than 1m^3 of volume).

ACKNOWLEDGEMENTS

The authors wish to warmly thank Dr. Sami Karkar, Etienne Rivet, Cédric Monchâtre and Louis Kaeppeli for their help in the experimental phases of the work.

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