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OPTIMIZING NOISE ABSORPTION THROUGH HYBRID ACOUSTIC BRICKS: NUMERICAL SENSITIVITY ANALYSIS OF THE ACTIVE VIBRATION CONTROL LOGIC

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ABSTRACT

Hybrid acoustic absorber have been implemented in order to overcome the limitations of common acoustic panel used for the control of room acoustics parameters. Indeed, the hybrid system allows to enhance the acoustic absorption in a wider frequency range. In this frame, a hybrid absorber block has been developed by combining active Structural Acoustic Control (ASAC) and high frequencies Passive Noise Control (PNC). This innovative smart structure, made of 3D-printed blocks, offers an easy-to- mount and customizable solution. It has potential applications in many fields, such as architecture, engineering, and construction industry, where noise is generated by broadband sound sources. The passive destructive interference proved to be an effective solution for high frequencies, where the wavelengths of interest are quite small, but it is difficult to be implemented at low frequencies and where space is limited. This paper focuses on the sensitivity analysis of the parameters of the active contribution. Mathematical modeling and tests with the Kundt tube demonstrate that the proposed approach can increase the average absorption coefficient in normal incidence condition by more than 80%, in the 125-1000 Hz frequency range.

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1. INTRODUCTION

In terms of effective frequency attenuation range, active and passive noise control can be considered as complementing strategies. Compared to passive control, which is more effective at high frequencies, active control more effectively dampens low frequencies. [1]. Meanwhile active control uses additional sources to create a sound field that destructively interacts with the main source, passive control often uses dampening or mass. [2]. Active noise control has so far only been successfully mass-adopted in enclosed environments, such as vehicle and aircraft cabin interiors, and headphones [3]. Active noise control (ANC) has been shown to be more effective than passive control methods at absorbing low frequency [4]. However, since ANC is now-days an energy-based system, its employment in high energy-consuming fields (such the built environment) [5] is still restricted.
Traditional passive approaches have been the strategy for noise control. Passive materials use sound absorption, diffusion, or reflection to control sound; these three methods are not mutually exclusive. The mass absorption is more effective at grater thicknesses for absorbing lower frequencies [6]. However, this imposes two constraints on costs and interior physical space. Systems with hybrid behaviour have been proposed as a result of these factors. Examples of hybrid strategies that combine passive

performance improvement and active control have been shown to increase the effective height of noise barriers by reducing diffracted waves over the top of the barrier [7]. The enhancement of airflow impact is one benefit of combining active and passive techniques in the built environment [8]. Active control was first used to lessen air flow noise in ducts, and it has more recently been shown to soundproof open windows from outside noises [9]. Unimpeded airflow is crucial for natural ventilation (NV) in the built environment, which is necessary for meeting the Sustainable Development Goals of the United Nations (UN) and ensuring public health [10].
The current study illustrates the construction of the active portion of a block with hybrid noise control behaviour in this setting. The latter, known as a smart acoustic block, improves the hybrid behaviour control by taking advantage of the potential of passive destructive interference (PDI) and active vibrational control (AVC).
The report, in particular, gives the computational and experimental confirmation of the block's behaviour [11], introducing a sensitivity analysis of the logic controlling its behaviour.

2. CASE STYDY

To validate the initial hypothesis experimentally, a study system is built in the laboratory. A piezoelectric patch is attached to the back of a circular aluminum plate that is two millimeters thick and is part of the mechanical system for the active behavior.
The 5.3 cm radius of the circular segment of the impedance tube where the tests are conducted determines the choice of the circular shape and radius (Figure 1).

Figure 1. System compounded by plate and piezoelectric patch in the impedance tube.

The purpose of the first experiment is to validate the passive scenario and determine whether or not the plate's reflection coefficient tends to 1. The Impedance Tube (50 Hz–6.4 kHz) Type 4206 of Brüel Kjær was employed.
It exploits a technique called transfer function method using two microphones. The tube has a sound source (a loudspeaker) at one end, which in our case corresponds to $x = 0$, and on the other end, a sample of the material, which in our case corresponds to $x = L$. Broadband, stationary random sound waves are produced by the loudspeaker and travel through the tube as plane waves before striking the sample and reflecting back. The superposition of forward- and backward-moving waves inside the tube causes the propagation, contact, and reflection to produce a standing-wave interference pattern. It is feasible to calculate the sound absorption and complex reflection coefficients as well as the normal acoustic impedance of the material, by monitoring the sound pressure at two fixed places and computing the complex transfer function using a two-channel digital frequency analyser [12].
Due to the constraints of the instrumentation that is currently available and the 5 cm radius of the plate that the sound wave affects, the length of the tube that is available in the preferred configuration is reduced to 29 cm.
The reciprocal relationship between the devices in the full chain set up in the lab to conduct the tests is made clear in Figure 8.
Twenty tests are run, each lasting two seconds, with the sampling frequency set to $fs = 20{,}000\ Hz$. The transfer function method is then used in MATLAB to process the signals and determine the total, incident, and reflected pressure waves as functions of the position inside the tube.
The reflecting behaviour of the plane is shown in Figure 2, and as would be predicted, the retroflected wave is symmetrical to the incident wave.

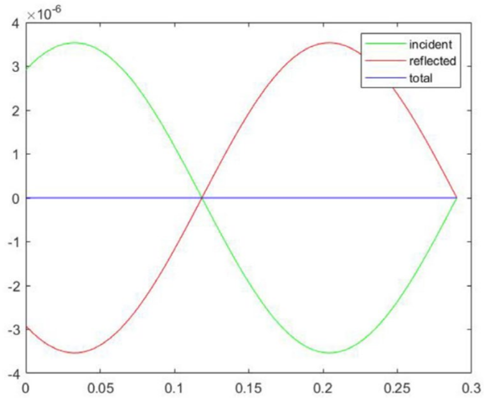


Figure 2. Sound pressure reconstructed inside the impedance tube for the passive case at f = 1000 Hz: real part, with acoustic pressure in Pascal in Y axis and tube length in meters in X axis.

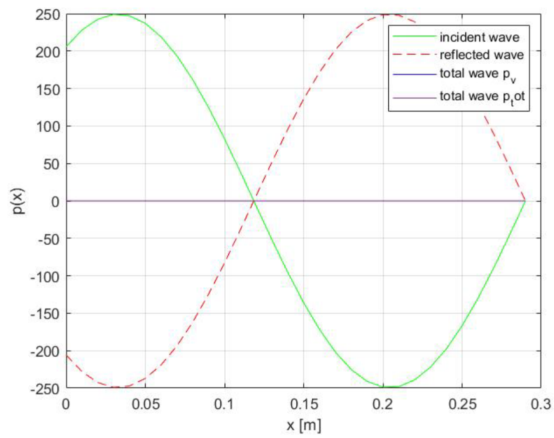
Table 1 contains the evaluated reflection coefficient for several third octave bands between 100 Hz and 2 kHz.

Table 1. Reflection coefficient of the aluminium plate in the passive case.

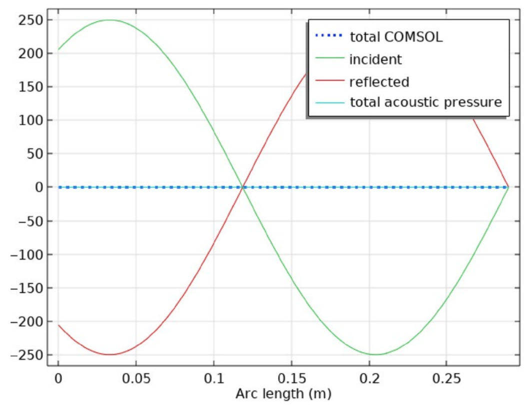
Frequency [Hz]	Reflection Coefficient
100	1
125	1
160	0.85
200	0.94
250	0.95
315	0.97
400	0.87
500	0.91
630	0.91
800	0.93
1000	0.93
1250	0.93
1600	0.95
2000	0.94

Based on that, a numerical model was developed. The plate response to the incident sound pressure is studied using a duct model, which is originally assumed to be 1 m long by default. After mathematical analysis, the length is decreased. The incident plane wave condition is taken into account [13] and the study of the diffuse sound field follows that. By examining the sound pressure wave reflected by the plate, the validation is carried out [14].

Figure 3 shows the plate's simulated reflecting behaviour; both in Matlab (a) and Comsol (b), the reflected wave is symmetrical to the incident one.



(a)



(b)

Figure 3. Passive case in Matlab (a) and Comsol (b).

3. CONTROL LOGIC

The plate-patch system's vibration under the influence of the control law provides an active response. The operation of the software code is significantly regulated by the control law.

In this instance, the validation of the control rule is built up in three stages, moving from a simplified case study to a more intricate and realistic model.

In the first example, a 2D duct is implemented in MATLAB and solved in the frequency domain.

The dimensions of the rectangular duct are 100 cm long by 10 cm broad.

A uniform harmonic motion (u_0) is imposed at boundary $x = 0$ (Figure 4). The control law in terms of vibration velocity is imposed on the other boundary $x = L$. In order to maximize the absorption coefficient of the totally reflective right boundary, a displacement is generated at the controlled boundary:

$$u_L = u_0 e^{-jkL}, \tag{1}$$

where L is the length of the tube, k the wave number, u_0 the velocity imposed on the other boundary and j the imaginary part.

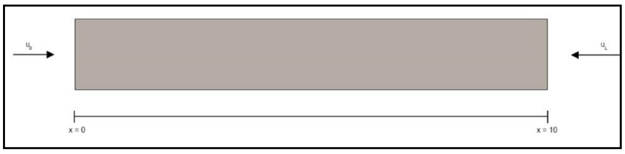


Figure 4. 2D duct model.

In this case, a totally absorption of the incident pressure wave is achieved, as shown in Figure 5a, where the sound pressure field is split in incident and reflected contribution:

$$p_i(x) = p_0 e^{-jkx} \tag{2}$$

and

$$p_r(x) = r p_0 e^{-jkx} \tag{3}$$

with arbitrary amplitude p_0 and reflection factor r.

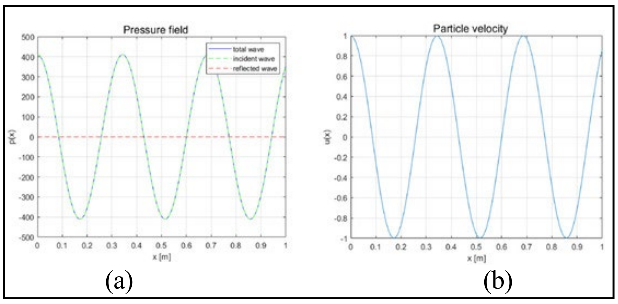


Figure 5. Closed–closed duct with control law maximizing the absorption coefficient as boundary condition in the 2D MATLAB model: (a) real part of the pressure field in Pascal and (b) real part of the particle velocity field at $f = 1000$ *Hz* in m/s.

The first 2D variant is necessary to validate the analytical formula implemented for the control law. The second one highlights the applied boundary conditions, validated also in case of diffuse incident field. For the sake of brevity, these two variants are not reported in this paper, but they were necessary to ensure that the MATLAB equation (1) is also valid in the case of a diffuse field and that it is also applicable in COMSOL. The third model considers a 3D tube with a clamped 2-mm thick aluminium shell placed at the $x = L$ position. This makes the model similar to the reality. The COMSOL numerical model evaluate the first resonance of the shell at 1979.6 Hz. This ensures the first mode of vibration. Furthermore, it is verified that the plane wave assumptions are valid up to about 2000 Hz, which corresponds to the first acoustic mode in 3D.

In the third step, the control law is applied as a point load to the plate installed at the end of the tube, simulating the force that is exerted by the piezoelectric patch. A frequency-dependent gain is identified in COMSOL, by considering the frequency response function (FRF) of the shell. The final expression of the point force F in the frequency domain becomes:

$$F = \frac{j u_L}{H(j\omega)} = jH(j\omega)\frac{u_0}{\sqrt{2}} e^{-jkL} \tag{4}$$

where H(jω) is the FRF of the shell, in terms of the RMS velocity of the entire shell per unitary point force at the centre. Figure 6, as an example, plots the simulation results at 1000 Hz. In particular, the figure plots the incident and the reflected pressure waves are obtained with the transfer function method [14]. The models are tested in the frequency range between 100 Hz and 2000 Hz at intervals of 10 Hz. Figure 5a plots the cancellation of the reflected wave, confirming the ability of the control law in maximizing the absorption coefficient.

4. RESULTS AND DISCUSSION

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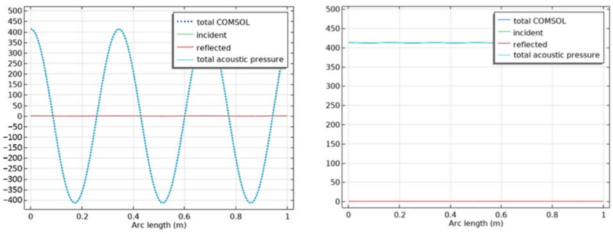
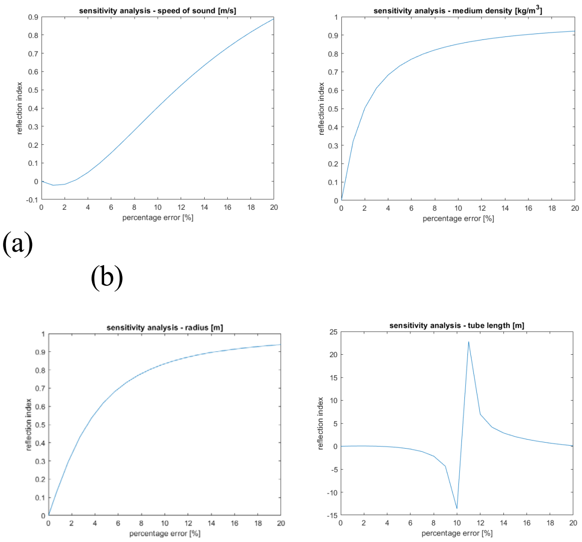


(a) (b)

Figure 6. Simulation results of the 3D duct model in COMSOL: (a) real part of the pressure field in Pascal and (b) amplitude of the pressure field at $f = 1000\ Hz$ in Pascal.

5. CONCLUSIONS

In order to change a brick's hybrid acoustic behavior and alter its sound absorption capabilities, this work offers a numerical validation of the active portion. The goal of the smart block is to provide a component that can increase the frequency range and have a high sound absorption coefficient. This enables it to be used inside of buildings and more effectively ensure high levels of acoustic comfort.
In order to create the control logic, an analytical model in MATLAB was first used to assess and describe the acoustic field in a duct. A preliminary evaluation of the results was carried out numerically using COMSOL. Three versions of the system were implemented for the 2D duct, 2D duct in the chamber, and 3D duct systems for the specific examination of the problem; only the third one was given in this paper.
In this research, the purpose was to minimize reflection. However, the same logic may be used to manage the absorption coefficient and the reflection coefficient with the aim of both minimizing and increasing them. The verification of the entire system must then be demonstrated in a real-world scenario in future research.



(a) (b)

(c) (d)

Figure 7. Percentage error the parameters: (a) variation of the speed of sound caused by the medium, (b) variation of the medium density, (c) variation of the radius of the disc, (d) variation of the length of the tube.

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