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SILENCER WITH TUNABLE RESONATORS FOR ACTIVE NOISE CONTROL

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KEYWORDS

Resonators, acoustic impedance, holes profiles, active noise control

ω : angular frequency pulsation

c : sound speed

k : wave number equal to ω/c

ABSTRACT

The article deals with active noise control and presents a silencer made of tunable acoustic resonators to reduce variable tonal noises. The silencer is composed of baffles with acoustic resonators made of two superposed and identically perforated plates associated with cavities. One of the plates is mobile and its displacement is controlled by an electromechanical system, allowing changing the internal shapes of the holes of the perforated layers. Consequently the impedance of the resonators and their resonance frequencies can be controlled. These tunable resonators can be used for noise control of variable tonal noises such as those generated by fans for example. The advantages of the proposed system are the simplicity of the actuation and the compactness of the system.

NOMENCLATURE

l_1 : thickness of plate 1
 l_2 : thickness of plate 2
 L : cavity depth
 $2r$: perforate hole diameter
 δ_e : exterior mass end correction
 δ_i : interior mass end correction
 D : mobile plate displacement
 ϕ : porosity
 Z : normal incidence acoustic impedance ($Z=R+jX$)
 R : normalized acoustic resistance
 X : normalized acoustic reactance
 ρ_0 : density of air
 μ : dynamic viscosity of air
 λ : wavelength

INTRODUCTION

The article focus on silencers with acoustic resonators made of perforated sheets bonded onto honeycomb cavities for noise reduction. The advantage of these resonators is that they can be used in hard conditions. The main drawback is that they are selective in frequency. They are thus efficient for tonal noise reduction.

In case of variable tonal noise, the efficiency is limited if the resonators are not tunable. One common solution is to control the cavities depth so as to tune the resonant frequency to a desired frequency. Another solution is to change the neck geometry of the resonators. Indeed, several works have shown the influence of the shapes of the holes in the perforated layers: Birnbach et al. [1, 2] studied a resonator with two perforated plates and an inlet air gap, Chanaud [3] studied the radiation impedance for geometries of non-circular orifices and Tang [4] studied a resonator with a tapered neck.

This article proposes the study of a baffle silencer with resonators tunable by the change of the neck shape. The resonators are made of two superposed and identically perforated plates associated with cavities [5, 6]. One plate is fixed and bonded to the cavities and the other plate is mobile. The present concept enables to change the internal shapes of the holes of the perforated layers and the resonance frequency of the system. The silencer can thus be used for noise control of variable tonal noises. The advantages of the proposed system are the simplicity of the actuation and the compactness of the system.

The first part of the article concerns the design of tunable acoustic resonators. It will be shown that their resonance

frequencies shifts to lower frequencies as the displacement of the mobile plate increases and that the resonators can be tuned to attenuate tonal noises that vary on a range of 1000 Hz.

In the second part of the article, a silencer composed of baffles with tunable acoustic resonators is presented and experimental results show the efficiency of the system.

The last part of the article describes a control law strategy of the active silencer to reduce variable tonal noises.

TUNABLE RESONATORS

Principle

The acoustic resonators made of perforated sheets bonded onto honeycomb cavities are selective in frequency. The acoustic properties are directly linked to the size, shape and porosity of holes and to the thickness of air gaps.

In order to make the resonance frequency vary, the holes are generated by two perforated plates, side by side, one fixed and one mobile (Figure 1). The fixed plate (plate 1) is taped on cavity and the mobile plate (plate 2), on top, can move by translation in one direction. The perforations of the two sheets are identical: the same orifice shape (circular in this case) and the same porosity. The translation of the mobile plate generates neck geometry with an elliptic profile as shown in figure 2. The translation values vary from zero to the perforate hole diameter. The minimal distance between two holes is more than one diameter to avoid the case of two overlaps for the same perforation.

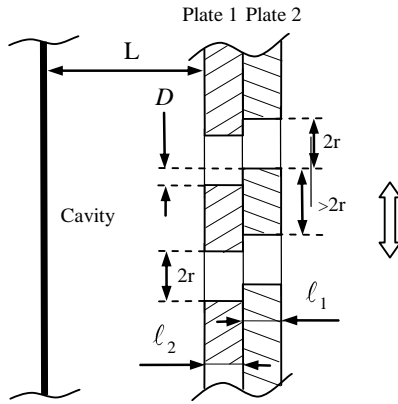


FIGURE 1: PRINCIPLE OF THE TUNABLE RESONATOR WITH TWO PERFORATED PLATES

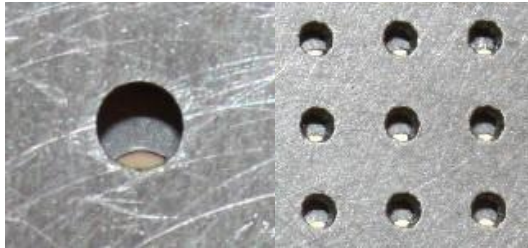


FIGURE 2: VISUALIZATION OF ELLIPTIC PROFILE WITH OVERLAPPED PLATES

Simplified model

In order to describe the acoustic behavior of the variable frequency resonator, a simplified model is presented here, based on the normalized acoustic impedance at normal incidence [7-11]. For a resonator, it can be expressed as follows:

$$Z = \frac{Z_{neck}}{\phi} + Z_{cavity}, \quad (1)$$

with Z_{neck} the normalized acoustic impedance of the neck made of the two perforated plates and Z_{cavity} the normalized acoustic impedance of the opened cavity (without neck),

The normalized impedance of an opened cavity is given by:

$$Z_{cavity} = -j \cot(kL), \quad (2)$$

The normalized impedance of the neck can be expressed by:

$$Z_{neck} = R_{neck} + jX_{neck} \quad (3)$$

where R_{neck} and X_{neck} are respectively the normalized acoustic resistance and reactance of the resonator neck.

R_{neck} is function of:

- the inner surface resistance formed by the acoustic boundary layer in the perforations (plates 1 and 2) during the wave propagation,
- viscous interactions near the orifice lips of the circular holes (on every side) and near the elliptic constriction lips between the two plates,
- the mobile plate displacement.

X_{neck} depends on the length of the cavity, on the plate thickness and on the displacement of the mobile plate..

R_{neck} and X_{neck} can be computed by using results given in [7-13]. When the holes of the two perforated plates are aligned (null displacement of the mobile plate), we get:

$$R_{neck} = \left(\frac{2(l_1 + l_2)}{r} + 2 \right) \sqrt{\frac{\mu\omega}{2\rho_0 c}} \quad (4)$$

and

$$X_{neck} = j \tan(k(l_1 + l_2 + \delta_i + \delta_e)) \quad (5)$$

with

$$\delta_i = \frac{8r}{3\pi} (1 - 1.25\phi) \quad (6)$$

and

$$\delta_e = \frac{8r}{3\pi} \quad (7)$$

Equations (2) and (5) can be linearized by considering that:

$$\lambda \gg \ell \text{ and } \lambda \gg L. \quad (8)$$

Resonance frequency of the resonator $f_{resonance}$ corresponds to the frequency for which the reactance of Z is null. So:

$$f_{resonance} = \frac{c\sqrt{\phi}}{2\pi\sqrt{L(1+l_2 + \partial_i + \partial_e)}} \quad (9)$$

When the mobile plate is in movement, previous works [6] have shown that the frequency resonance decreases and shifts towards the low frequencies.

Experimental results

In this paper, the following configuration is tested:

- thicknesses of the plates: 1mm
- holes diameter: 1.5mm
- cavity depth: 7mm
- porosity: 6%

For such a configuration, the estimated resonance frequency given by relation (9) is equal to 2822 Hz.

Tests are realized in the frequency band 400-3000Hz in an impedance tube. The open end of the tube enables to test samples with a sliding plate (Figure 3). The tube is equipped with two microphones Brüel & Kjær 4187 for pressure measurements. At the opposite side of the tube, the loudspeaker generates a broadband random noise that propagates as plane waves at 120 dB (SPL in linear domain).

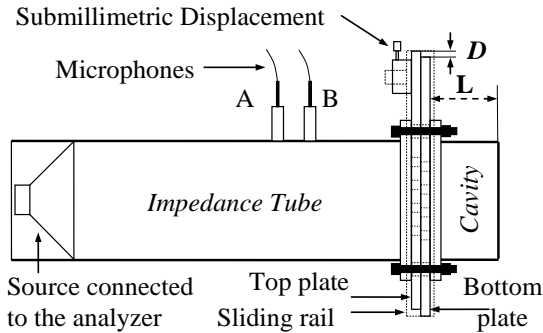


FIGURE 3: SCHEMATIC OF EXPERIMENTAL SET UP

The two microphones standard measurement method is used to compute transfer function H between the two microphones and consequently reflection coefficient R expressed by:

$$R = e^{2jkX_A} \frac{H - e^{-jk(X_A - X_B)}}{e^{jk(X_A - X_B)} - H} \quad (10)$$

where X_A and X_B are the distances between the microphones and the tested sample.

With R , one can compute the normalized acoustic impedance:

$$Z = \frac{1+R}{1-R} \quad (11)$$

and the absorption coefficient in normal incidence:

$$A = 1 - |R|^2 \quad (12)$$

Figure 4 shows the measured sound absorption coefficient versus the plate displacement. Results are only given for three plate positions to get the figure clear. However, the tests have been achieved for more displacements (every 0.1mm) and the results are similar whatever the displacement.

When the mobile plate displacement is null, the measured resonance frequency is 2780 Hz, which is close to the computed value. A part of the difference can be explained by the accuracy in the positioning of the mobile plate displacement. When the mobile plate is moving and the orifice restriction becomes smaller, the resonance frequency of tunable resonators shifts to lower frequencies. The frequency shift is of 1000 Hz for the tested configuration. One can also notice that the absorption band widens with the orifice closing because of the increase of viscous interactions.

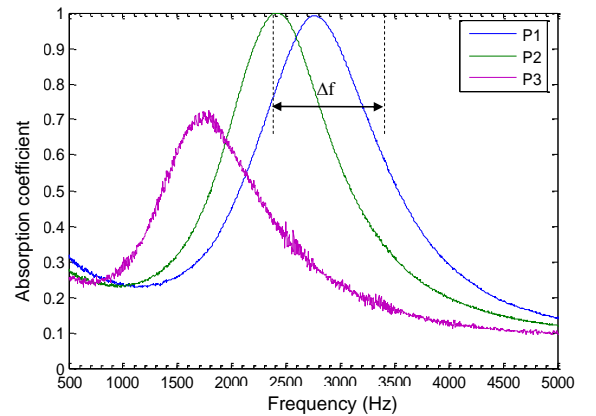


FIGURE 4: MEASURED ABSORPTION COEFFICIENT FOR THREE MOBILE PLATE DISPLACEMENTS

P1: mobile plate displacement = 0mm

P2: mobile plate displacement = 0.8mm

P3: mobile plate displacement = 1.4mm

BAFFLE SILENCER WITH TUNABLE RESONATORS

Description of the silencer

The silencer is made of three baffles with tunable resonators on each side (Figures 5 and 6). There are so six absorbent surfaces. The mobile parts of the three baffles are actuated by a single electromechanical linear actuator (Part B). The link between the actuator and the mobile parts of baffles is rigid, so the displacement of the actuator is the same as the ones of the baffles. A loudspeaker (Part A) is connected to one extremity of the silencer to generate noise.

The characteristics of the baffles are as following:

- baffle sizes: 222mm*148mm
- gap between 2 baffles: 22mm
- variable resonance resonators of same sizes as in section 2

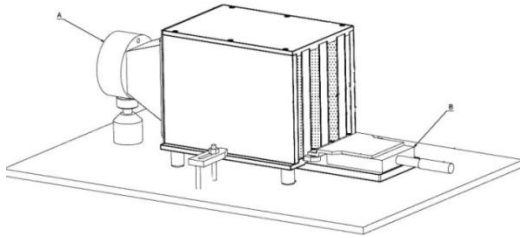


FIGURE 5: SCHEME OF THE SILENCER

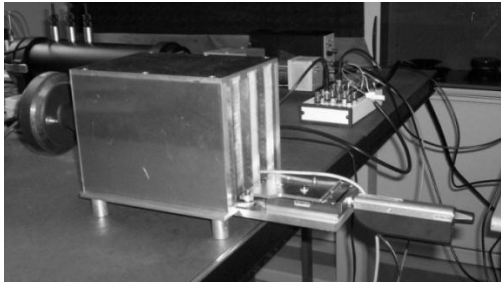


FIGURE 6: PICTURE OF THE SILENCER

Experimental results

The measure of the acoustic attenuation consists in measuring the acoustic sound power of the source with and without the silencer. The acoustic attenuation is the difference between these two acoustic sound powers. These measurements are made on the frequency range 1600-3000 Hz using a sound level meter of type Bruel & Kjaer model 2250. The acoustic attenuation is measured for different values of the mobile plate displacement, from zero to the maximal value (1.4mm) (silencer in active mode). The acoustic attenuation is also measured for no displacement of the mobile plate (silencer in passive mode). Results are synthesized in Figure 7 and show the efficiency of the silencer to better attenuate the lower frequencies than the silencer with no active system. The maximal gain is at 2200Hz: the attenuation is

increased by 17 dB with the silencer in active mode compared with the silencer in passive mode.

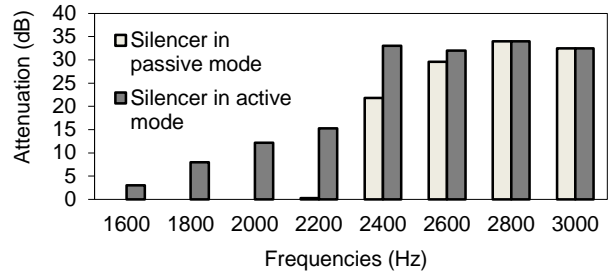


FIGURE 7: ATTENUATION OF THE SILENCER

CONTROL STRATEGY

The plant to control is composed of the electromechanical actuator linked to the moving part of the silencer. The objective of the control is to minimize the noise at the output of the silencer.

Control in position of the electromechanical actuator

The actuator must be controlled in position in order to ensure a good positioning of the mobile plates of the resonators. The model of the actuator with its load (the moving part of the silencer) is identified by tests in open-loop. It can be described by transfer function $H_A(s)$:

$$H_A(s) = \frac{D(s)}{U(s)} = \frac{0.23}{s+1} \quad (13)$$

with D the displacement and U the input voltage of the actuator.

In order to ensure accuracy, the actuator will be controlled by a proportional-integral controller $C(s)$:

$$C(s) = 10 + \frac{1}{s} \quad (14)$$

The final scheme for the actuator control is given in Figure 8.

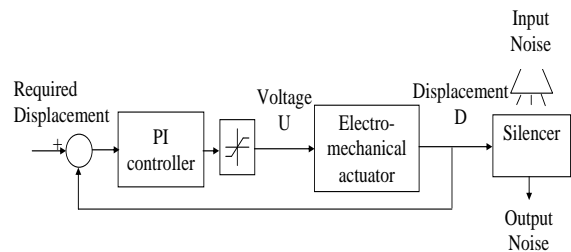


FIGURE 8: CONTROL OF THE ELECTROMECHANICAL LINEAR ACTUATOR

Static model of the silencer

Acoustic measurements are performed to establish a static model of the silencer. For frequencies between 2000 and 3000 Hz with a step of 50Hz, the attenuation produced by the silencer is measured for different displacements of the actuator and the displacement for which the attenuation is the highest is identified. A curve (displacement versus frequency) is extracted from these tests (Figure 9) and a polynomial expression of the acoustic behaviour of the silencer is established:

$$D(f) = 2.2028e-15f^4 - 1.9119e-11f^3 + \dots + 6.0139e-8f^2 - 8.2061e-5f + 0.042786 \quad (15)$$

This expression gives the displacement of the actuator that best attenuates an input tonal noise. It is a static model and it is assumed that the mechanical dynamic behaviour of the silencer is taken into account in the actuator model since this model is established from tests of the actuator connected to the silencer. The dynamic of the acoustic phenomena is negligible compared to the mechanical dynamic.

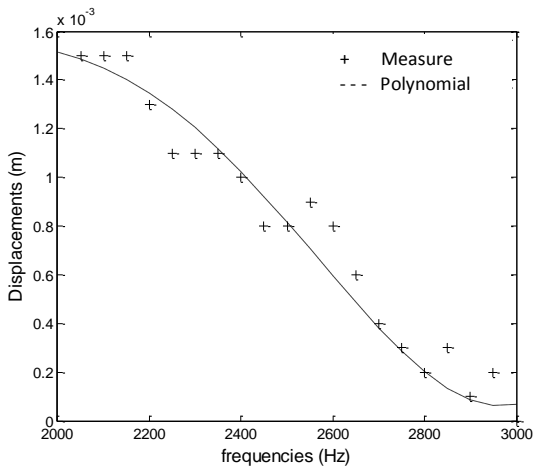


FIGURE 9: ACOUSTIC BEHAVIOUR OF THE SILENCER

Active control of the silencer

The proposed control strategy is as following (Figure 10):
 1 - Measurement of the input noise with a microphone
 2 - Computation of the tonal noise to be attenuated by a FFT analysis of the measured noise
 3 - Computation of the displacement of the actuator that best attenuates this tonal noise by using the polynomial of relation (15)

In order to consider changes in the model of the silencer, the polynomial expression of the silencer behaviour can be up-dated regularly. This strategy has been tested and is efficient with tonal noises that vary between 2000 and 2800Hz.

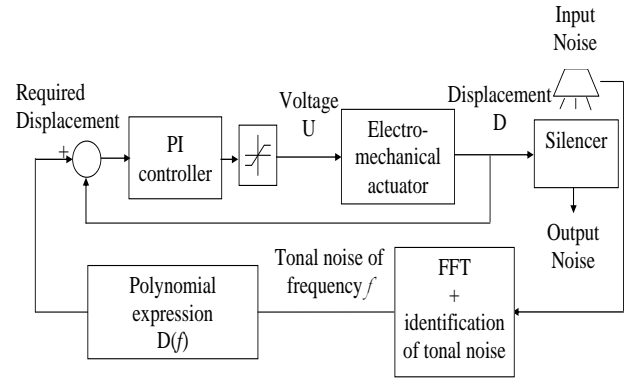


FIGURE 10: CONTROL STRATEGY OF THE SILENCER

Experimental results

Measurements are performed with the silencer to validate the control strategy. A tonal noise varying between 2000Hz and 2800 Hz is sent in the silencer via a loudspeaker (Figure 11(a)). The noise is measured at the output of the silencer by a microphone. Figure 11(b) shows the voltage measured by the microphone when the silencer is in passive mode (no displacement of the mobile plate). Figure 11(c) shows the same voltage when the silencer is in active mode and when the mobile plate translates to track the tonal noise.

In passive mode, as the displacement of the mobile plate is null, the silencer is configured to attenuate mostly the frequency around 3000Hz. So the output noise is important when the input tonal noise is at 2000Hz and decreases as the frequency of the input noise increases and becomes closer to 2800Hz.

In active mode, the displacement of the mobile plate starts from zero (initial condition imposed by the system) and has to increase to become maximal as the frequency of the tonal noise is 2000Hz (see Figure 9). So, at the beginning of the measure, there is a transition phase for which the level of the noise is the same than in passive mode because the optimal mobile plate displacement is not yet reached. Then Figure 11(c) shows that the noise is much more attenuated than in passive mode as the displacement of the mobile plate is regulated with success (Figure 11(d)). For example, at 5s and for a frequency of 2300Hz, the input noise level is 0.5725V with the silencer in passive mode and 0.115V with the silencer in active mode, so there is so a gain of 14 dB. At the end of the measure, the noise measured in passive mode and in active mode is the same since the mobile plate is in the optimal position, i.e. null, in both cases.

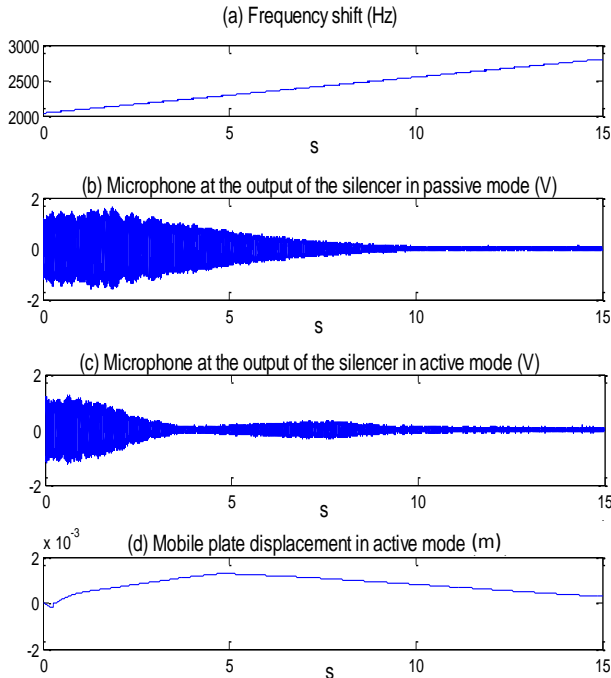


FIGURE 11: EXPERIMENTAL RESULTS OF THE SILENCER WITH TUNABLE RESONATORS

CONCLUSION

The article has shown that resonators made of two superposed and identically perforated plates associated with cavities can be tuned by moving one of the two perforated plates. The resonance frequency then shifts to lower frequencies when the profile of the resonators' neck is modified to be restricted. A baffle silencer made with this kind of resonators can be efficient to attenuate tonal noise that varies on a frequency range of 1000 Hz. A control strategy based on the position control of the silencer and on the identification of the acoustic behavior of the silence can be implemented to achieve active noise control. Improvements could be made to the system by optimizing the actuator.

The proposed active noise control system is easy to implement and is different of conventional active solutions based on the interference with secondary noise sources. It can be used for machines with variable tonal noises such as fans of aircraft engines and rotating machines controlled at variable speed.

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