Panel of resonators with variable resonance frequency for noise control

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\textbf{A B S T R A C T}

The article focuses on acoustic resonators made of perforated sheets bonded onto honeycomb cavities. This kind of resonators can be used in adverse conditions such as high temperature, dirt and mechanical constraints. For all these reasons, they are, for example, widely used in aeronautical applications. The acoustic properties are directly linked to the size, shape and porosity of holes and to the thickness of air gaps. Unfortunately, the acoustic absorption of these resonators is selective in frequency and conventional acoustic resonators are only well adapted to tonal noises. In case of variable tonal noise, the efficiency is limited if the resonators are not tunable. One common solution is to control the depth of cavities based on the noise to be attenuated. This article proposes another technology of tunable resonators with only a very small mass and size increase. It consists of two superposed and identically perforated plates associated with cavities. 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. The article describes this system and gives a theoretical model of the normal incidence acoustic impedance that allows to predict the acoustic behavior, in particular the resonance frequency. The model shows that the resonance frequency varies with hole profiles and that the absorption peak moves towards the lower frequencies. The proposed model is validated by measurements on various configurations of resonators tested in an impedance tube. The perspectives of this work are to adapt the hole profiles using an actuator in order to perform active control of impedance.

\textbf{1. Introduction}

The article focuses on acoustic resonators made of perforated sheets bonded onto honeycomb cavities. This kind of resonators can be used in adverse conditions such as high temperature, dirt and mechanical constraints. For all these reasons, they are, for example, widely used in aeronautical applications. The acoustic properties of such resonators are directly linked to the size, shape and porosity of holes and to the shape and thickness of air gaps as shown in many studies \cite{1-11}. Unfortunately, the acoustic absorption of these resonators is selective in frequency and the acoustic resonators are only well adapted to tonal noises. In case of variable noise, the efficiency is limited if resonators are not tunable. The resonators under study in this article are formed by cavities connected to small ducts (called necks). The geometry of necks or cavities can be modified in order to produce tunable resonators.

Many studies deal with the cavity depth change. Konishi et al. \cite{2} and Birdsong and Radcliffe \cite{3} proposed tunable acoustic absorbing systems made of resonators the porosity or air cavity volume of which are controllable so as to tune the resonant frequency to a desired frequency. Kostek and Franchek \cite{4} studied the control of such systems. Kobayashi et al. \cite{5} successfully implemented resonators with tunable cavities in a turbofan.

In this study, the neck geometry of the resonators varies rather than the cavity size. All works about hole geometry show its impact on the acoustic behavior of resonators. The length, shape and section of the neck have an impact on the surface impedance, in particular the reactance. As the sound absorption coefficient is maximum when the reactance is null, the neck has an influence on the frequency resonance of the resonators.

Birnbach et al. \cite{6,7} studied a resonator with two perforated plates and an inlet air gap. They studied different configurations with variable distance between plates. For a very small distance from 0.05 mm to 0.1 mm, the resonators show high acoustic impedance and low absorption. By increasing the distance between the plates, the impedance decreases and the absorption becomes maximum.

Chanaud \cite{8} studied the radiation impedance for geometries of non-circular orifices. He examined the cross orifice made up of two rectangular shapes placed perpendicular to each other. He calculated an equivalent circular orifice because no solution could be found for rectangular radiation piston. He also studied the
interaction of the orifice shape on the end correction. He concluded that the orifice shape did not significantly affect the interior mass end correction.

Ducts' end effects have been also investigated by [9–11]. These studies have shown that edges' end effects generate nonlinearities with high sound pressure levels and that the duct thickness and the duct edge shape have an influence on both resistance and reactance of orifices. Some of these effects can be taken into account in the modeling. The corrections that make models more accurate can be computed for different shapes, in particular for round edges' duct ends.

Tang [12] studied a resonator with a tapered neck. Results show that the resonance frequency increases with the tapering length and that absorption increases with the tapered neck slope.

The general conclusion of all previous studies is that controlling the opening size or the shape of resonators necks can be a way to control the impedance and thus the efficiency of resonators. This article studies a simple system to modify the neck geometry. The system tries to satisfy aeronautical constraints by nearly not increasing the weight and size of conventional resonators. It consists of two stacked perforated plates backed by cylindrical cavities. Plate 1 is fixed and plate 2 is mobile. The fixed plate is bonded onto cavities and the mobile plate, on the top, can move by translation in one direction. The perforations and the porosity of the two plates are identical. The hole diameter is 2mm. The overlap must be performed with accuracy and the plates are thus guided on two parallel sides by two sliding rails. The translation of the mobile plate is performed in one direction by an actuator (Fig. 2). This actuator is a double-row ball bearing linear stage. The stage is fixed on the mobile plate and on the sliding rails through a link rod. Therefore, the mobile plate can be slid in and out with the smooth stage travel by a manual knob control. The translation of the mobile plate generates neck geometry with an elliptic profile as shown in Fig. 3. The translations values (D) vary from zero to the perforate hole diameter. The minimum distance between two holes is more than one diameter to avoid the case of two overlaps for the same perforation. L is the cavity depth, $\ell_1$ and $\ell_2$ are the thicknesses of the bottom fixed plate and of the top mobile plate respectively (Fig. 1).

3. Building a model

3.1. A model of the normal incidence acoustic impedance

The aim of this section is to establish a model of the normal incidence acoustic impedance. The impedance is an important characteristic since it enables to describe the interaction between acoustic incident waves and absorbent materials. In this model, the flow over the perforated interface is not taken into account.

The expression of the acoustic impedance of a duct with a section discontinuity (Fig. 4) can be found in many references [15–17]. The impedance is established by writing the acoustic pressure continuity:

$$P_1' = P_2'$$

and the flow conservation:

$$S_1v_1' = S_2v_2'$$

with $v_1'$ and $v_2'$ the velocities normal to surfaces.
$Z_2$, the normal incidence acoustic impedance after the section change, is expressed as a function of $Z'_1$ (Fig. 4):

\[
Z'_2 = \frac{p'_2}{v'_2} = \frac{S_2p'_1}{S_1v'_1} = \frac{S_2p'_1}{S_1Z'_1}.
\]  

(3)

This impedance model is basic and end corrections, low frequency approximations and resistive aspects can be introduced according to Ingard [13] and Rayleigh and Lindsay [14] to get a more accurate model.

For the system under study, the neck is obtained from the overlapping of the two perforated plates as shown in Fig. 5a. Applying Eqs. (2) and (3) with $S_1 = S_2$ does not show the influence of the restriction in the impedance formulation. This model is too simple because physical phenomena near apertures are not taken into account. A more accurate model must be established.

The first idea to model the neck with the two overlapping plates is to be considered:

- an orifice with a circular surface equivalent to the elliptic surface created by the two overlapping plates,
- a neck with a length equal to the sum of the two plates’ thicknesses.

It will be shown in Section 4 with validation results that this model is not exact.
The model proposed in this paper suggests that the effective mass of air in motion is increased near the orifice lips. It is based on the fact that when an acoustic wave is propagated through an opening, pressure and velocity fluctuations induce shedding of vortices at the opening edge at high pressure level [10–12, 17–19]. The additional mass of air near the orifice lips can be considered as an elliptic piston of volume $V_E$, length $d_E$ and elliptic section $S_E$ (Fig. 5b). The neck model is therefore established by considering 3 pistons in motion with volumes $V_E$, $V_1$ and $V_2$ and two “lost” volumes $V_0^1$ and $V_0^2$ (which means that there is no or nearly no air in vibrating motion in these volumes). In this configuration, the normal incidence acoustic impedances can be expressed as follows:

$$Z_0^E = \frac{p_0^E}{v_0^E}$$

and

$$Z_0^2 = \frac{p_0^2}{v_0^2} = \frac{S_E p_0^E}{S_1 E} = \frac{S_E}{S_1} Z_1$$

With this modeling, even if $S_1 = S_2$, the discontinuity of section appears in the impedance formulation. The proposed model will be implemented to establish the normalized acoustic impedance at normal incidence of the system under study as a function of the geometry and of the mobile plate translation. This impedance will be expressed in the following form:

$$Z = R + jX,$$

where $X$ is the normalized acoustic reactance and $R$ the normalized acoustic resistance.

### 3.2. Normalized acoustic reactance

The normalized acoustic reactance is established using the neck equivalent schematic described in Fig. 6.

First the reactance of the neck will be expressed and then the reactance of the whole system with the cavity will be computed.

$$X = \frac{\tan k \ell_1}{1 + j X^1 \tan k \ell_1}, \quad \ell_1 = x_1 - x_0$$

The change of section ($S_1$ to $S_E$) gives reactance $X_1$ at position $x_1$:

$$X_1 = \frac{S_E}{S_1} X_1^1 = \frac{S_E}{S_1} (\frac{X_1^1 + j \tan k \ell_1}{1 + j X^1 \tan k \ell_1})$$

In the duct of length $\ell_2 = x_2 - x_1$, reactance $X_2$ at position $x_2$ is expressed as a function of reactance $X_1$ at position $x_1$:

$$X_2 = \frac{X_1 + j \tan k \ell_2}{1 + j X_1 \tan k \ell_2}$$

Length $\ell_2$ of volume $V_E$ depends on the elliptic section $S_E$ and on the Rayleigh conductivity [14,20]. For a duct with an ellipsoid shape [13], $\ell_2$ is equal to:

$$\ell_2 = \frac{1}{\tan k \ell_2}$$

Fig. 5. Two overlapping perforations (a), model configuration with piston $V_E$ (b).

Fig. 6. Equivalent schematic for reactance computation.
where $b$ and $\varepsilon$ are the semi-major axis and the eccentricity of the ellipse respectively. For the system under study, these two parameters depend on the mobile plate displacement $D$ (Fig. 7):

$$b = r \sqrt{1 - \left(\frac{D}{2r}\right)^2}$$

and

$$\varepsilon = \frac{\sqrt{b^2 - a^2}}{b}$$

where $a$ is the semi-minor axis expressed as follows:

$$a = r - \frac{D}{2}$$

In Fig. 6, the change of section ($S_e$ to $S_2$) gives reactance $\chi_2$ at position $x_2$: 

$$\chi_2 = \frac{S_2}{S_e} \arctan k \lambda_2 \left(1 + \frac{j \tan \lambda_2}{1 + j \lambda_2} \tan k \ell_2 \right).$$

In the duct of length $\ell_2 = x_2 - x_s$, reactance $\chi_2$ at position $x_2$ is expressed as a function of reactance $\chi_1$ at position $x_1$:

$$\chi_2 = \frac{\chi_1 + j \tan \lambda_2}{1 + j \lambda_2} \tan k \ell_2$$

Finally, reactance $\chi_2$ can be expressed as a function of reactance $\chi_1$. In order to get a more accurate model, the interior and exterior mass end corrections, respectively $\delta_i$ and $\delta_o$, are added respectively to $\ell_1$ and $\ell_2$ (Fig. 6). For circular openings, and if porosity $\phi$ is lower than 0.4, usual formulations [14,21] are:

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

$$\delta_o = \frac{8}{3\pi} r$$

As mentioned previously, the cavity impedance is added to the neck impedance to get a complete model of the system. For sound waves at normal incidence and for $L \gg L$, the normalized reactance of the cavity air layer is:

$$X_{cavity} = -j \cot(kL).$$

Finally, the expression of the normalized reactance of the two plates with all the perforations (characterized by porosity $\phi$) combined with the cavity is:

$$X_{total} = X_{cavity} + \chi_2/\phi.$$
Hence, the system total normalized resistance is:

\[ R_{\text{total}} = 2R \left( \frac{\ell_1}{r} + \frac{\ell_2}{r} + \frac{S_k}{2r} + 2 - \frac{S_k}{\ell} \right). \]  

(26)

4. Experimental and modeling results

This section describes the experimental set up and gives different results stemmed from measurements and from the model.

4.1. Experimental set-up and devices

The experimental set-up based on a home-made impedance tube is schematized and shown in Figs. 9 and 10.

The tube has a 50 mm inner diameter and is 1-m long. It is used for measurements in the frequency range 400–3000 Hz. The open end of the tube enables to mount the system with the sliding plate. The two-plate system described in Section 2 is locked tightly between the cavity and the tube. The tube is equipped with two microphones Bruel & Kjær 4187 for pressure measurements. On the opposite side of the tube, the loudspeaker generates a broadband random noise that propagates plane waves from 400 to 3000 Hz at 120 dB (SPL in linear domain). The devices are mounted as shown in Figs. 9 and 10. The standard measurement method for the two microphones is used in accordance with the International ISO 10534-2 standard [23] and enables to calculate the transfer function \( H \) between the two microphones and consequently the reflection coefficient \( R \) expressed as follows:

\[ R = e^{j\phi} \frac{X_B H - e^{-j\phi(X_A - X_0)}}{e^{j\phi(X_A - X_0)} - H}, \]  

where \( X_A \) and \( X_B \) are the distances between the microphones and the tested sample. With \( R \), the normalized acoustic impedance can be calculated:

\[ Z = \frac{1 + R}{1 - R}, \]  

(28)

and the absorption coefficient in normal incidence:

\[ A = 1 - |R|^2. \]  

(29)

Several variable neck resonators have been manufactured and tested in the impedance tube. The tested resonators were designed for plane wave propagation in the tube. Four system configurations with different porosities (5% or 10%), different plate thicknesses (1 mm or 2 mm), different perforate hole diameters (1 mm or 2 mm) and different cavity depths (10 mm or 20 mm) will be tested. The configurations are described in Table 1.

4.2. Results and discussions

Figs. 11–14 show the measured normalized reactance and the measured sound absorption coefficient extracted from Eqs. (20) and (29) respectively. They are plotted versus the plate displacement for the four experimental devices. Results are only given for four displacements to get clear figures. However, the tests have been achieved for more displacements (every 0.1 mm) and the results are similar whatever the displacement.

For each case, the normalized reactance \( X_{\text{total}} \) is also calculated from Eq. (19) and the sound absorption coefficient is extracted from relation (29) with the computed reflection coefficient equal to:

\[ R = \frac{Z_{\text{total}} - 1}{Z_{\text{total}} + 1}, \]  

(30)

with

\[ Z_{\text{total}} = R_{\text{total}} + jX_{\text{total}}. \]  

(31)

They are both plotted in Figs. 11–14 to compare measured and computed data.

It can be pointed out that the acoustic behavior of the developed devices is as expected. The resonance frequency shifts to lower frequencies when the orifice restriction gets smaller. The frequency shifts range from 200 Hz to 800 Hz for the tested configurations. For configurations with 10% of porosity (configurations 2 and 4), the frequency shift is lower than for configurations with lower porosities. At last, also note that the absorption band widens with the orifice closing due to the increase of viscous interactions.

Table 2 synthesizes the results and gives the resonance frequencies obtained from measurements and predicted by the proposed model (columns 2–6). Some errors are to be noticed. The first error source is the actuator minimal resolution of 0.02 mm that can generate a maximum dispersion of +/- 50 Hz on the resonance frequency. For small openings that correspond to large displacements, errors are more important and can be explained by the measurement dispersion that is significant in these cases. However, the results show that the proposed model predicts correctly the resonance frequency shift of the resonator and gives a good estimation of the useful frequency range.

Table 2 also gives the resonance frequencies predicted by the model with equivalent circular orifices. For this model, the differences between the measured and computed frequencies are significant, in particular for large displacements. These differences are

![Fig. 9. Schematic of experimental set up.](image-url)
Fig. 10. The resonator with the two plates and the actuator mounted in the impedance tube.

Table 1
Characteristics of the experimental devices.

<table>
<thead>
<tr>
<th>Screen thickness</th>
<th>Hole diameter</th>
<th>Porosity (%)</th>
<th>Cavity depth</th>
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<tr>
<td>Configuration 4</td>
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Fig. 11. Normalized reactance and sound absorption coefficient $A$ of a system made of two 1-mm thick plates perforated at 5% with 1-mm diameter holes, backed by a 10-mm air gap (configuration 1). $D = 0$ mm, – computed data, $\circ\circ\circ$ measurement; $D = 0.5$ mm, -- computed data, +++ measurement; $D = 0.7$ mm, -- computed data, $\bullet\bullet\bullet$ measurement; $D = 0.9$ mm, -- computed data, +++ measurement.

Fig. 12. Normalized reactance and sound absorption coefficient $A$ of a system made of two 1-mm thick plates perforated at 10% with 1-mm diameter holes, backed by a 20-mm air gap (configuration 2). $D = 0$ mm, -- computed data, $\circ\circ\circ$ measurement; $D = 0.5$ mm, -- computed data, +++ measurement; $D = 0.7$ mm, -- computed data, $\bullet\bullet\bullet$ measurement; $D = 0.9$ mm, -- computed data, +++ measurement.
Fig. 13. Normalized reactance and sound absorption coefficient $A$ of a system made of two 2-mm and 1-mm thick plates perforated at 5% with 1-mm diameter holes, backed by a 10 mm air gap (configuration 3). $D = 0$ mm, — computed data, $\bigcirc\bigcirc\bigcirc$ measurement; $D = 0.5$ mm, — computed data, $$ measurement; $D = 0.7$ mm, — computed data, $\bigcirc\bigcirc\bigcirc$ measurement; $D = 0.9$ mm, — computed data, $$ measurement.

Fig. 14. Normalized reactance and sound absorption coefficient $A$ of a system made of two 2-mm and 1-mm thick plates perforated at 10% with 1-mm diameter holes, backed by a 10 mm air gap (configuration 4). $D = 0$ mm, — computed data, $\bigcirc\bigcirc\bigcirc$ measurement; $D = 0.5$ mm, — computed data, $$ measurement; $D = 0.7$ mm, — computed data, $\bigcirc\bigcirc\bigcirc$ measurement; $D = 0.9$ mm, — computed data, $$ measurement.

Table 2
Comparisons of measured and predicted resonance frequencies.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Displacements (mm)</th>
<th>Ellipse area (mm²)</th>
<th>Predicted resonance frequency (proposed model) (Hz)</th>
<th>Measured resonance frequency (Hz)</th>
<th>Predicted/measured error (%)</th>
<th>Equivalent radius of circular orifice (mm)</th>
<th>Predicted resonance frequency (equivalent circular orifices) (Hz)</th>
<th>Predicted (equivalent circular orifices)/measured error (%)</th>
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represented in Fig. 15 where the measured resonance frequencies and the predicted resonance frequencies obtained with the two models are plotted versus the mobile plate displacement. These results show that the system with the two overlapping plates cannot be modeled by equivalent cylindrical orifices.

5. Conclusions

This paper presents an absorber made of resonators with two perforated plates (a fixed one and a mobile one). The idea behind this design is to obtain resonators the resonance frequency of which varies with the mobile plate displacement. The paper proposes a model of this absorber using a phenomenological approach. Three pistons in motion are taken into account to model the shape of the neck created by the translation of the mobile plate and to describe the inertial behavior in the neck. This model is used to compute the normal incidence acoustic reactance and the absorption coefficient without flow.

Several absorbers with different configurations have been manufactured and tested from 400 to 3000 Hz in an impedance tube. The computed and measured normalized reactances indicate a shift of the first frequency resonance. The lower the porosity, the larger the resonance frequency shift. For porosities exceeding 10%, the frequency shift is limited. The perforation diameter is also an important parameter since, for the tested configurations, the frequency shift increases with the hole diameter. Results show that the developed model gives a good estimation of the resonator behavior, in particular of the useful frequency range. This model can thus be used to design tunable resonators with variable resonance frequencies.

The first advantage of the system proposed in this article is that it is a simple system for active noise control. Indeed, by controlling the displacement of the mobile plate with an electromechanical actuator and a control loop, active control of slowly variable tonal noise could be performed. The actuator would not need to be as large as speakers used for active noise control techniques with anti-noise source since its only function is to move slowly one plate. However, in case of applications in hot operating conditions, the actuator should not be exposed to heat constraints.

The second advantage is that the resonance frequency of the absorber shifts towards lower frequencies. The system is thus more...
In order to improve the proposed model, a better estimation of viscous effects could be performed by integration of the kinetic energy in the vicinity of the restriction between the two overlapping plates. This computation requires preliminary tests to estimate the acoustic boundary layer in the vicinity of restriction.

The model proposed in this paper does not take into account the flow over the perforated interface. As some of the foreseen applications (e.g. fans controlled at variable speed) are exposed to flow, it would also be useful to build a model that includes the sound propagation conditions.

**Appendix A**

The appendix presents the normalized reactance of the resonator neck made of two plates (without cavity).

Figs. 17–20.

**References**