NUMERICAL ANALYSIS OF SMART FOAM FOR ACOUSTIC ABSORPTION

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ABSTRACT
The “smart foam” concept and design originate from the combination of the passive dissipation capability of foam performing in the medium and high frequency range and the active absorption ability of piezoelectric actuator in the low frequency range. This results to a passive/active absorption control device that can efficiently operate over a broad range of frequencies. In this study, the possibilities and limitations of the use of PVDF films embedded in passive absorber (acoustic foam) as actuator for an active control of absorption system is examined. A complete finite element model of smart foam has been developed to examine several basic “smart foam” configurations in low frequencies (under 1000Hz). This modelling tool is used to determine the effects of foam properties, foam shape and PVDF film size, shape and location on the optimal control voltage input that must be imposed on the actuator electrodes. Several minimisation criteria have been investigated. Resulting absorption coefficients and control inputs are presented. These numerical studies are undertaken for sound absorption under normal plane wave incidence. The phase relations between incident wave and optimal actuator voltage are studied in order to analyse active absorption mechanisms and the influence of the configuration (foam/PVDF) on the optimal control input. A promising smart foam configuration especially dedicated to absorption is presented.

INTRODUCTION
Improving low frequency absorption is an important topic in acoustics. Passive materials generally provide adequate absorption at medium and high frequencies whereas active control is efficient to cancel low frequency sound waves. Many devices combining sound absorbing passive materials and active absorption properties have been studied. One can distinguish between two main approaches. The first approach consists in associating a passive porous layer with an active surface separated from the porous layer by an air gap. There are here two different control strategies. The first strategy is to impose a zero pressure on the back surface of the porous layer [1]. The second strategy consists in controlling the surface impedance of the active surface in order to cancel the reflected wave [2]. These two strategies have proved to be effective for a broad frequency range [3] but imply a weight and space penalty that is may limit their application in industrial sectors such as aerospace. To overcome these limitations, a recent approach called smart foam, which consists in a control actuator (generally a PVDF material) directly embedded in a foam layer, has been developed in the last 10 years [4]. Smart foams have been mostly studied for noise reduction but less for absorption. Especially, the mechanisms of acoustic dissipation within a smart foam are still to be clearly established. In this paper, we therefore focus only on absorption. This paper presents a numerical parametric study of smart foam for active sound absorption. A complete finite element modelling tool for smart foam has been developed but is not presented here. Three different smart foam configurations with three different passive materials and three criteria of minimisation have been studied in a closed waveguide in order to conduct a parametric study to find an effective actuator configuration, passive material and criteria of minimisation.
MATERIAL AND METHOD

Methodology
The study takes place in a closed waveguide at frequencies low enough so that plane waves dominate. The primary source is a plane piston with an imposed transverse displacement D fixed at 10 micrometers at one end of the waveguide; the active absorption device consists of a smart foam with an embedded piezoelectric actuator at the other end of the waveguide.

Since all governing equations are linear, it is possible to divide the complete system (piston and smart foam) into the sum of two excitation states: “passive” (piston active and smart foam passive) + “active” (piston passive and smart foam active).

The Frequency Response Function (FRFs) between pressure in the waveguide and voltage applied to the PVDF actuator are calculated for each excitation state at two virtual microphone positions (denoted Mic 1 and Mic 2 in Figure 1). It is therefore possible to calculate the reflected pressure, the reflection coefficient and the surface impedance of the smart foam under the combined effect of the primary source (denoted by the letter $p$) and embedded actuator (denoted by the letter $a$).

Smart foam configurations
Three different configurations are studied (Figure1). The first is the classical configuration that has been proposed by Fuller [5]. The smart foam consists of a curved PVDF film (the active component) between individual layers of sound-absorbing material. In the second configuration, the foam under the PVDF is replaced by air. The third configuration is quite different. The PVDF film is placed around the foam. All the smart foams are supposed to be freely sliding in the tube along their lateral sides. The smart foam of configurations 1 and 2 are fixed on their rear face to the waveguide rigid termination. The rear face in configuration 3 is in contact with an air gap, and the termination of the waveguide is again assumed to be rigid. The PVDF is fixed on the tube on the contact front edge between foam and PVDF.

![Figure 1.- Description of the three smart foam configurations](image)

In the configuration 1 et 2, the radii of curvature of the PVDF are the same ($R = 0.032$ m). In the configuration 3, the radius of curvature is larger ($R = 0.1$ m).

Material parameters
Three different sound-absorbing materials are used for each configuration. The first is a melamine foam which has good passive absorption performances. The second is a felt-fiber material classically used in the automotive industry. The third material is a polyurethane foam that is widely used. The properties of these materials are given in Table 1, where $\phi$ is the porosity, $\rho$ the density, $\sigma$ the resistivity, $\alpha_\infty$ the tortuosity, $\lambda$ the viscous length, $\lambda'$ the thermal...
length, $E$ the Young’s modulus, $\eta$ the loss factor, $\nu$ the Poisson ratio. These properties have been measured on a melamine foam sample at the LCMA (GAUS). The next part of Table 1 shows the properties of the PVDF actuator and the bonding layer (between PVDF and foam) used in the simulations: $t$ is the thickness of the PVDF and the bonding, $d_{31}$ and $d_{32}$ are the piezoelectric coefficient. $\rho_0$ is the density of the fluid (air) and $C_0$ is the sound velocity in the air. The PVDF data come from literature [6] and the bonding parameters have been experimentally evaluated.

Table 1.- Material properties of the absorbing material, PVDF actuator, bonding layer and air

<table>
<thead>
<tr>
<th></th>
<th>$\phi$</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\sigma$ (N/s/m$^4$)</th>
<th>$\alpha_{\infty}$</th>
<th>$\lambda$ (m)</th>
<th>$\lambda'$ (m)</th>
<th>$E$ (N/m$^2$)</th>
<th>$\eta$</th>
<th>$\nu$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melamine foam</td>
<td>0.96</td>
<td>9</td>
<td>15300</td>
<td>1.02</td>
<td>1.05 10-4</td>
<td>2.05 10-4</td>
<td>108000</td>
<td>0.08</td>
<td>0.4</td>
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<td>Fibrous</td>
<td>0.95</td>
<td>26.5</td>
<td>1500</td>
<td>1.11</td>
<td>1.95 10-4</td>
<td>2.7 10-4</td>
<td>6000</td>
<td>0.23</td>
<td>0</td>
</tr>
<tr>
<td>Polyurethane foam</td>
<td>0.96</td>
<td>22</td>
<td>5000</td>
<td>1.24</td>
<td>1.05 10-4</td>
<td>3.4 10-4</td>
<td>46500</td>
<td>0.14</td>
<td>0.4</td>
</tr>
<tr>
<td>PVDF</td>
<td>5.4 10+9</td>
<td>1780</td>
<td>0.18</td>
<td>0.04</td>
<td>28 10-6</td>
<td>23 10-12</td>
<td>3 10-12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bonding</td>
<td>12 10+7</td>
<td>500</td>
<td>0.4</td>
<td>0.15</td>
<td>150 10-6</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>fluid</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.188</td>
<td>345</td>
<td></td>
</tr>
</tbody>
</table>

Minimisation criteria

We have computed the analytical relationship leading to the cancellation of the reflected sound pressure to determine the optimal voltage $\beta$ to be applied to the PVDF actuator. Other criteria of minimisation have been investigated, such as the reflection coefficient and the difference between the surface impedance and the characteristic impedance of air. As we make the assumption of plane wave propagation in the waveguide, these other criteria give identical results than the cancellation of the reflected pressure. As the minimisation of the reflected sound pressure is linear, easily calculable and easily accessible experimentally, we only present this one.

The $\exp (j \omega t)$ convention is used. The reflected pressure in the tube ($P_r$) can be written:

$$P_r = \frac{P_2 - P_1 e^{-jkd}}{2j \sin(kd)}$$  \hspace{1cm} (Eq. 1)

Where $P_1$ and $P_2$ are respectively the pressure calculated at virtual microphone position 1 and 2, $k$ is the acoustic wave number and $d$ is the inter-microphone separation.

Assuming that linear superposition applies in the waveguide, $P_1$ and $P_2$ can be written as the summation of sound pressures calculated under the effect of the primary source and the embedded actuator The “active” sound pressures generated by the embedded actuator is proportional to the input voltage $\beta$ applied to the smart foam.

$$P_T = \frac{P_2 e^{-jkd}}{2j \sin(kd)} - \beta \frac{P_1 e^{-jkd} - P_2}{2j \sin(kd)}$$  \hspace{1cm} (Eq. 2)

We can easily deduce the optimal voltage $\beta$ that cancels the reflected pressure:

$$P_T = 0 \implies \beta = \frac{p_{2p} - p_1 e^{-jkd}}{p_1 e^{-jkd} - p_{2a}}$$  \hspace{1cm} (Eq. 3)

Description of the finite element modelling tool

A 3D finite element model has been developed to calculate the acoustical and structural response of a smart foam. This model uses five different types of quadratic elements. There are three volume elements with 20 nodes and two shell elements with 8 nodes. The poroelastic element is based on the Biot-Allard equation, using the (u,p) formulation described by Atalla [7]. It has 4 dof by node (3 displacements and 1 sound pressure). It is a general element that can be used to model fibrous as well as foam materials. The elastic element has 3 dof by node (3 displacements). It can be used to model any elastic structure, but is only used in this study to
model the primary piston. The fluid element has 1 acoustical dof by node (sound pressure). The elastic shell element has 6 dof by node (3 displacements and 3 rotations) and is used to model the bonding layer between the PVDF actuator and the foam. The piezoelectric shell element has 7 dof (3 displacements, 3 rotations, 1 electric potential) and is used to model PVDF. It is based on the volumic piezoelectric equations presented by Piefort [8]. This model has been validated numerically for all the elements and for couplings that can be modelled with commercial software (all except the coupling between poroelastic element and piezoelectric shell). The experimental validation of the coupling between poroelastic, piezoelectric shell, and fluid is in progress at the moment.

RESULTS

**Configuration 1**

![Configuration 1 FRF](image1)

![Configuration 1 Absorption](image2)

![Configuration 1 Alpha](image3)

**Configuration 2**

![Configuration 2 FRF](image4)

![Configuration 2 Absorption](image5)

![Configuration 2 Alpha](image6)

**Configuration 3**

![Configuration 3 FRF](image7)

![Configuration 3 Absorption](image8)

![Configuration 3 Alpha](image9)

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Figure 2. - Top: Sound pressure / PVDF voltage Frequency Response Function at the microphone 1 for each configuration and material, with primary excitation off.

Middle: optimal $\beta$ for each configuration and material

Bottom: Absorption coefficient without and with control.

The secondary FRFs are presented on top of figure 3 because they give a first indication of the effectiveness of the configuration in terms of actuation and also reflect the modal response of the closed waveguide.

The optimal $\beta$ are described by three graphs. The top graph gives the amplitude of the control voltage for a displacement of the primary piston fixed at 10 micrometers. It gives information on
the effectiveness of each configuration. The middle graph gives the relative phase of $\beta$ with respect to the displacement of the primary piston. The bottom graph gives relative phase of $\beta$ after compensating the travel time between the piston and the material surface. The last three bottom graphs show the absorption coefficient with and without control. It shows that the passive absorption largely depends on the configuration and the material.

**DISCUSSION**

Considering the passive behaviour, the absorption coefficient shows minor differences between the first and second configuration. The passive absorption is a slightly lower in the second configuration due to the lack of absorbing material under the PVDF. Nevertheless, from a passive point of view, the material under the PVDF has only a minor influence. It is clear that the third configuration exhibits a better passive absorption. Compared to the two others, the third configuration has exactly the same volume, but the part of effective absorbing material is higher. This results from the fact that the PVDF film acts as a screen which cancel the effect viscous dissipation effect of the material behind it.

If we compare the passive effect of the different materials, it appears that the melamine and polyurethane foam are better than the fibrous material. The melamine foam is generally better than the polyurethane foam except for the compression modes of the melamine foam. Moreover the melamine foam is much lighter than the polyurethane foam or the fibrous material which can be a great advantage for application in the transport industry. Considering now the active aspect of the results, several observations can be done. The optimal beta decreases with the frequency for every materials and configurations. So the more efficient the passive absorption, the less important the beta amplitude is. The materials have little importance on the actuator of the different configurations. By analysing the FRF and the optimal beta graphics, it is however possible to see some interesting differences. In the low frequencies, the FRF of the first configuration with the fibrous material is higher than with the other material. In addition, the magnitude of the optimal beta is less important with the fibrous material in this frequency range. This is not true for the configuration 2. In this configuration the magnitude of the optimal beta is the same for every material under 400Hz and the same than the first configuration with the fibrous material. This indicates that the stiffness of the material under the PVDF film has a non negligible influence and tends to reduce the actuation efficiency. It is also possible to see this phenomenon with the configuration 3. Indeed, in this configuration, the PVDF compress all the absorbing material. So the stiffer the material, the less effective the actuation of the PVDF is. The FRFs show that the melamine foam is better than the other material above 800 Hz for the configuration 1 and 2. It can be explained by the fact that melamine foam is lighter than the other ones so the inertial forces are less important. It tends to show that for the configuration 1 and 3, the stiffness of the absorbing-sound material has a relative importance in the low frequencies and that the density of material has an importance for the configuration 1 and 2 in the high frequency range.

One of the most important observations for the active aspect is that the configuration 3 is much efficient than the other ones regarding the amplitude of the optimal beta. The optimum voltage to be applied to the PVDF is roughly three times lower than for the other configurations. This is due to two main facts. First, the passive absorption is better. Second, the surface of PVDF involved in the configuration 3 is more important because there is PVDF film on the side of the foam. The curvature radius of the PVDF is also greater which increases the displacement of the foam [6].

For all configurations, the compensated angle of beta is close to zero ($\pi$ for the configuration 3 due to the orientation of the PVDF actuator). The control signal is in phase with the perturbation displacement. It shows that the control mechanism is to dissipate in the actuator the incident acoustic energy that is not passively dissipated. The variation of the compensated angle beta with the frequency is due to the propagation in the foam which is not taken into account in the compensation.

The modal behaviour of the smart foam is very simple with the fibrous material because this material has no stiffness and no compression resonance. The modal behaviour of the polyurethane foam seems to be more complicated than the melamine foam. Indeed the polyurethane foam is heavier and less stiff than the melamine, which involves more resonance at lower frequencies. This can be a disadvantage for control.
The optimal absorption coefficient is perfect (unity) for all the configuration and material, but this is only a theoretic result. Cancelling the reflected sound pressure at a position in the waveguide with plane wave assumption involves a cancellation everywhere in the waveguide. The boundary conditions are ideal and hardly feasible in an experimental setup. These results are thus only considered to show the general tendencies.

CONCLUSIONS
This theoretic study is an exploratory one aiming to understand main tendencies and to select suitable prototypes of smart foam. The geometrical configuration has a great importance on the optimal voltage. The configuration 3 seems to be very efficient for active and passive sound absorption. It is however certainly possible to optimize the design of smart foam further. It is theoretically already possible to absorb a wave with a particular displacement of 10 micrometers with less than 100 Volt applied to the PVDF.

The displacement of the actuator is almost in phase with the incident wave particular displacement. The acting mechanism results in transferring in the actuator the energy of the incident sound pressure that is not passively dissipated. The melamine foam is efficient for smart foam application, because of its low density and its passive absorption efficiency. The important stiffness and low density prevent the melamine foam of having too much resonance mode at low frequencies, which can be an advantage for active control. If the PVDF has to compress the foam, a lower stiffness can also be an advantage for the efficiency of the actuator. The control criteria of reflected pressure, reflection coefficient and difference between surface impedance and characteristic impedance give identical results for plane waves in normal incidence.

Future work will be to model more realistic boundary conditions and perform experimental measurements to validate the structural and acoustical behaviour of the model. We will compare the optimal voltage given by a classical control algorithm with the theoretic voltage determined in this study. We will only build the prototype with melamine foam that seems to be the best material choice for smart foam application.

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References: