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NUMERICAL STUDY OF SOUND ABSORPTION BY PERFORATED PANELS

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ABSTRACT

Perforated panels are acoustic absorbers typically spaced from a hard surface that attenuate sound due to viscous friction in their holes. An accurate acoustic model of a perforated panel should include viscothermal effects to account for the losses associated with the narrow perforations. In this paper, a numerical finite element model of a perforated panel including these loss mechanisms was implemented. For this purpose, the linearized Navier-Stokes model included in the commercial package COMSOL Multiphysics is used. Simulations allow predicting sound absorption of these types of elements. Results obtained are compared to wellestablished analytical models.

RESUMEN

Los paneles perforados son absorbentes acusticos tipicamente espaciados de una superficie rigida que atenuan el sonido debido a la friccion viscosa que se produce en sus orificios. Un modelo acustico correcto de un panel perforado debe incluir los efectos viscotermicos para tener en cuenta las perdidas asociadas a estas pequenas perforaciones. En este trabajo se ha implementado un modelo numerico en elementos finitos de un panel perforado incluyendo estos mecanismos de perdida. Para ello se ha empleado el modelo linealizado de las ecuaciones de Navier-Stokes incluido en el software comercial COMSOL Multiphysics. Las simulaciones realizadas permiten predecir la absorcion sonora de este tipo de elementos. Los resultados obtenidos se comparan con soluciones analiticas bien asentadas.

1. INTRODUCTION

Perforated panels are sound absorbers commonly used in vibrations and noise control applications. Typically backed by an air cavity and a rigid wall, they can also be found as a complement to the classical porous absorbers. When perforations are reduced in size, sound is attenuated due to viscous friction in their pores, providing higher acoustic resistance and lower mass reactance necessary for a wide-band sound absorber. There are several models and approaches [1-3] for predicting the acoustic response of perforated panels. Most of these models determine the acoustic performance of such devices from their orifice diameter, perforation rate, panel thickness and the depth of the air gap. Atalla and Sgard [4] showed that a perforated plate or screen can be modeled as an equivalent fluid following the Johnson-Allard approach with an equivalent tortuosity and that those classical models can be reobtained using this simple approach. Even though most of these models have been experimentally validated through the years, some uncertainties related to more complex configurations arise. In a recent work by Tayong [5], the holes interaction and heterogeneity distribution effects on the acoustic properties of air-cavity backed perforated plates are discussed. It is particularly shown that in the presence of these effects the sound absorption low-frequency characteristics of such systems are improved. Integrating an expression for the geometrical tortuosity (originally derived for granular materials) in the Atalla and Sgard model these effects are suitably considered. In doing so, a constant that depends on the shape of overall apertures is deduced using an inverse method after measuring the perforated plate effective density. The main drawback of this latter and other inverse characterization procedures are the requirement of test samples that increase early stages product development costs and hinder their widespreading in the sound absorbing manufacturing industry. In this context, complementary modeling and characterization techniques must be developed.

The main aim of this work is to study some perforated plates backed by an air cavity using a finite element methodology (FEM). For this purpose, the thermoacoustics interface included in the acoustics module of COMSOL 4.3 b software is used. Different than isentropic/lossless acoustics, this thermoacoustic linearized Navier Stokes formulation takes the dissipative effects of viscous shear into account, allowing to model acoustic wave propagation through narrow geometries as in the case of perforated panels. The main disadvantage of the finite element discretization of the full viscothermal acoustic formulation used by COMSOL is its high computational cost (a large number of elements is needed to properly model viscous boundary layers). Some authors [6, 7] have proposed the use of more efficient reduced formulations, although most of them can only be applied to a limited range of geometries. In order to validate the proposed characterization methodology, various perforated panels configurations are simulated and then compared to Atalla and Sgard analytical model in terms of sound absorption. This study is focused on rigid panels with circular shape holes and does not consider any motion of the plate or thermal effects.

The structure of the paper is as follows; in section 2 the Atalla and Sgard model for the case of perforated panels backed by an air cavity is reviewed; in section 3 the set of linearized Navier Stokes equations for viscothermal acoustics are briefly introduced and the numerical setup implemented for the simulations described; in section 4 a comparison of the results obtained with FEM and the analytical model for different perforated panels configurations is presented; finally section 5 describes the main conclusions of this paper.

2. PHYSICAL MODEL

Figure 1 shows a rigid perforated plate composed by a periodic distribution of identical cylindrical perforations of circular cross-sections (thickness t, diameter d, perforation rate ϕ). When backed by a cavity of depth D and under normal incidence (assuming acoustic wavelengths to be larger than the plate characteristics), it is legitimate to assume that the air backing is partitioned (distributed Helmholtz resonator). The classical approaches consist in evaluating the normal surface impedance of such configurations, being the resistive part induced by the viscous effects occurring within the perforation due to viscous boundary layer and the flow distortion effects generated on both sides of the panel. The reactive part accounts for the inertia effect due to the motion of air cylinders thicker than the perforation depth (correction lengths must be used).

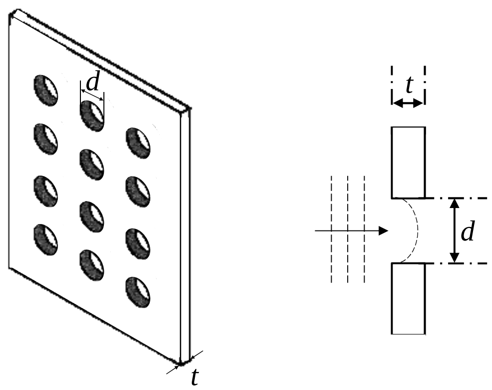

Figure 1 – Schematic of a perforated panel excited by a plane wave.

Following Atalla and Sgard model, the total input impedance of the perforated plate - air cavity combination Z_A can be written as

$$Z_A = \left(\frac{t}{d} + 2\frac{\varepsilon_\varepsilon}{d}\right)\frac{R_S}{\phi} + \frac{1}{\phi}\left(2\varepsilon_\varepsilon + t\right) j\omega\rho_0 - j\rho_0 c_0 \cot\left(k_0 D\right), \quad (1)$$

where ρ_0 is the air density, c_0 is the speed of sound, k_0 the wave number, ω is the angular frequency, $R_S = \sqrt{\eta\omega\rho_0/2}$ denotes the surface resistance, where η is the dynamic viscosity of air, and $\varepsilon_\varepsilon = \varepsilon_0\left(1 - 1.14\sqrt{\phi}\right)$ represents the correction length above mentioned, being $\varepsilon_0 = 0.425d$ the radiation reactance of a circular plane piston baffled in an infinite wall. The correction term $\varepsilon_\varepsilon/\varepsilon_0$ accounts for the so called interaction between the perforations.

In order to assess the absorption performance of air-backed perforated plates, the reflection coefficient R is obtained by

$$R = \frac{Z_A - \rho_0 c_0}{Z_A + \rho_0 c_0}, \quad (2)$$

and the normal incidence absorption coefficient α is given as

$$\alpha = 1 - |R|^2. \quad (3)$$

3. FINITE ELEMENT MODEL

3.1. VISCOTHERMAL ACOUSTICS

As stated above, the standard model for isentropic acoustics (Helmholtz equation) neglects the dissipative viscothermal effects such as viscous friction. Unfortunately, this simplification is not allowed for small geometries as is the case of perforated panels. Viscothermal acoustic models consist of a linearized set of the Navier Stokes equations that can handle the case under study at the expense of requiring a high computational cost. The derivation of the equations describing viscothermal acoustic problems is based on the account of inertia, compressibility, viscosity and thermal conductivity effects.

The finite element discretization of the linearized Navier Stokes model is the viscothermal acoustic model implemented in COMSOL 4.3.b. The governing equations are the linear time harmonic PDEs

$$j\omega\rho_0\mathbf{v} - \nabla\cdot\sigma = 0\,, \tag{4a}$$

$$j\omega\rho_0 C_p T + \nabla\cdot\mathbf{q} - j\omega p = 0\,, \tag{4b}$$

$$\nabla\cdot\mathbf{v} - j\omega\frac{T}{T_0} + j\omega\frac{p}{p_0} = 0\,, \tag{4c}$$

where ρ_0, T_0, p_0 and C_p denote the quiescent density, quiescent temperature, quiescent pressure and the specific heat at constant pressure; $\mathbf{q}$ and σ denote the heat flow perturbation vector and the total stress tensor; $\mathbf{v}$, p and T denote the velocity, pressure and temperature perturbation fields.

The divergence terms are also a function of the chosen degrees of freedom (p , $\mathbf{v}$ and T),

$$\nabla\cdot\sigma = (\lambda+\mu)\nabla(\nabla\cdot\mathbf{v}) + \mu\Delta\mathbf{v} - \nabla p\,, \tag{4d}$$

$$\nabla\cdot\mathbf{q} = -\kappa\Delta T\,, \tag{4e}$$

where λ, μ and κ denote the second viscosity, dynamic viscosity and heat conduction coefficient; ∇ and Δ are the gradient and Laplace operators.

Note that the linearized Navier Stokes implementation is computationally costly since it contains five coupled fields and the mesh must be refined near boundaries to accurately solve for the resulting acoustic damping. Otherwise, is widely applicable given that practically any geometry can be modeled with this model.

3.2. NUMERICAL SETUP

A finite element procedure is established to simulate the acoustic behavior of the air-cavity backed perforated panel under normal incidence. The modeling procedure is based on a 3D configuration which consists of a backing cavity and a perforated panel covering on the open side of this cavity, as shown in Figure 2. Following the ISO 10534 standard [8] on determination of sound absorption coefficient, a duct is connected to the perforated panel in the numerical model, representing the impedance tube in the experimental setup. Such a configuration is primarily for the simulation of the acoustic properties of the air-cavity backed perforated panel under normal incidence.

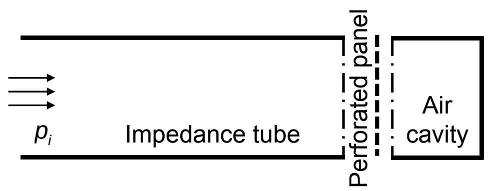

Figure 2 – Numerical model for the air-cavity backed perforated panel.

A plane incident wave p_i comes from the left-hand side of the duct with unit amplitude, being part of the acoustic energy reflected and the rest absorbed in the resonator system. The acoustic field in the fluid regions is governed by the set of equations (4). Likewise, on each boundary location a mechanical and thermal condition must be prescribed. Since the perforated panel is assumed to be rigid and no thermal effects are considered, acoustically rigid and isothermal conditions are prescribed in remaining boundaries of the numerical scheme.

Thereby, evaluating the transfer function H_{12} given by the relation between acoustic pressures in two positions spaced from the perforated panel front surface, the reflexion coefficient R can be obtained with the aid of Eq. (5) and normal incidence absorption coefficient of the air-cavity backed perforated panel can be then evaluated using Eq. (3).

$$R = \frac{H_{12} - e^{-jks}}{e^{jks} - H_{12}} e^{j2kx_1} , \quad (5)$$

where s is the separation distance between evaluated points and x_1 the separation of the most distant point from the sample.

Various perforated panels with different thickness, hole diameter and perforation rates are simulated. Table 1 summarizes the geometrical characteristics of those whose results are going to be shown. The narrowest perforations are greater than 1 mm, which is in the order of the viscous length scales. The simulations were carried out using a mesh with tetrahedral elements with reduced size close to and inside the panel perforations. The frequency range from 50 to 500 Hz with 10 Hz frequency steps was calculated.

Table 1 – Characteristics of the perforated panels.

Sample	t (mm)	d (mm)	ϕ (%)
PP 1	1	3	0.36
PP 2	1	3	0.72
PP 3	1	2.5	0.25

4. RESULTS

Figure 3 shows a comparison of the analytical and numerical results of the normal incidence absorption coefficient for perforated panels of Table 1 with an air cavity depth of 50 mm. A good agreement can be observed for the numerical model and the analytical solution. Some discrepancies can be noted close to the resonance for the sample PP2, likely due to the finite dimension of the perforated panel in the numerical model that implies a non equality of the baffling conditions of each one of the holes in the panel. In any case, the numerical model predicts the viscothermal effects accounting for the dissipation occurring at the discontinuities. It is observed that increasing porosity holding other parameters constant, forces the central frequency of the absorption coefficient band higher in frequency (PP2 regarding PP1).

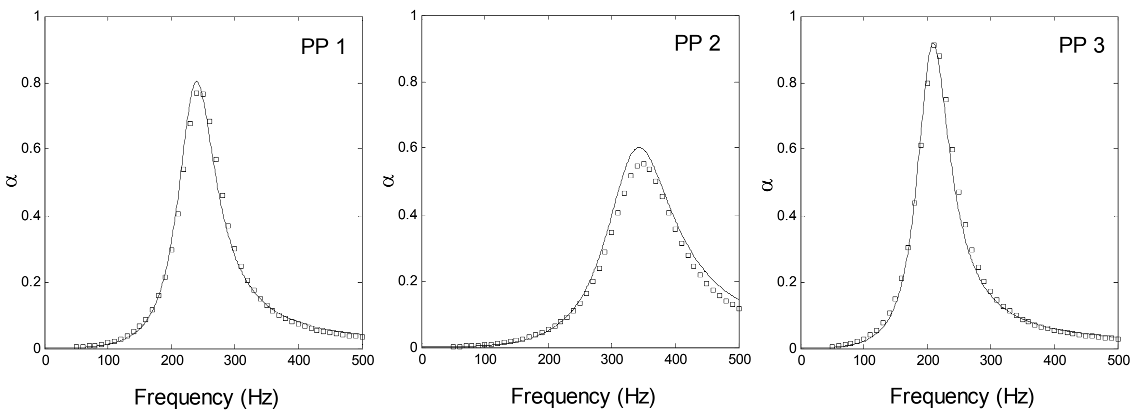

Figure 3 – Numerical (□) and analytical (continuous line) results comparison. Absorption coefficient for perforated panels of Table 1 with an air cavity depth of 50 mm.

Figure 4 shows a cross-sectional detailed view of the magnitude distribution of the velocity components in the region of geometric discontinuities (the perforations) for one of the perforated panels under study at 350 Hz. It can be seen that the axial component in between the discontinuities has a constant velocity far from the wall, increasing as it gets closer to the wall and then decaying to zero on the wall. On the other hand, the radial velocity at the discontinuities illustrates dominant dissipation mechanisms. In both plots it can be appreciated the interaction effect with other nearby perforations (in this case, located on the left) and the non equality of the baffling conditions of each hole aforementioned is verified.

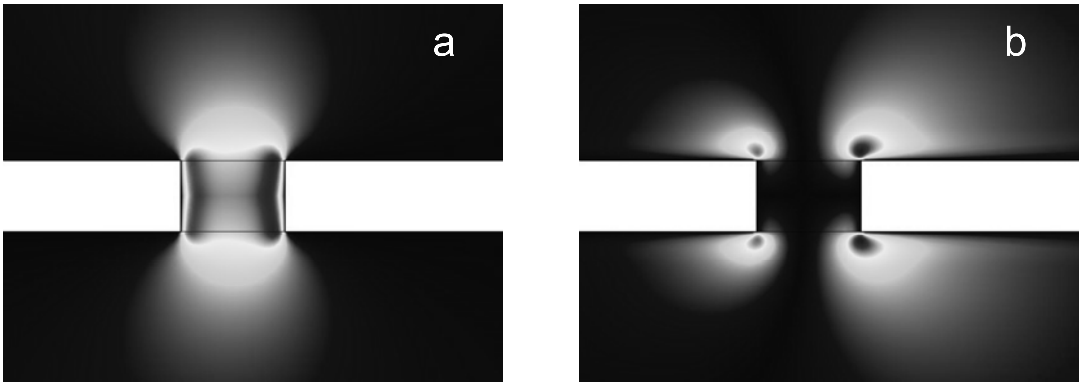

Figure 4 – Cross-sectional detailed view of the velocity distribution for the axial (a) and radial (b) components of a perforated panel hole at 350 Hz.

5. CONCLUSIONS

In this paper, the finite element numerical modeling by discretized full viscothermal acoustic formulation of air cavity backed perforated panels with different characteristics has been presented. From the simulations data, the normal incidence absorption coefficient has been calculated and then compared to well-established analytical solutions like the Atalla and Sgard equivalent fluid approach. The results obtained corroborate the validity of the proposed methodology and offer a useful alternative to traditional design procedures. The main advantage of this methodology is that it can handle more complex configurations (i.e. non homogeneous holes distribution or variable perforation diameter), becoming a useful tool for sound absorbing manufacturing industry.

REFERENCES

[1] K. U. Ingard, Notes on Sound Absorption Technology, *Noise Control Foundation*, New York, 1994.

[2] L. L. Beranek, *Noise and Vibration Control Engineering*, Wiley, New York, 1992.

[3] D. Y. Maa, *Microperforated wideband absorbers*, Noise Control Engineering Journal, 29 (3), 77-84, 1987.

[4] N. Atalla, F. Sgard, *Modeling of perforated plates and screens using rigid frame porous models*. Journal of Sound and Vibration, 303, 195-208, 2007.

[5] R. Tayong, *On the holes interaction and heterogeneity distribution effects on the acoustic properties of air-cavity backed perforated plates*, Applied Acoustics, 74, 1492-1498, 2013.

[6] W. M. Beltman, *Viscothermal wave propagation including acousto-elastic interaction, part I: Theory*, Journal of Sound and Vibration, 227, 555-586, 1999.

[7] W. R. Kampinga, Y. H. Wijnant, A. de Boer, *An Efficient Finite Element Model for Viscothermal Acoustic*s, Acta Acustica united with Acustica, 97, 618-631, 2011.

[8] ISO 10534-2, *Determination of Sound Absorption Coefficient and Impedance in Impedance Tubes* (International Organization for Standardization, Geneva, Switzerland, 1998).