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Margaux Regniez[1,2], Adrien Pelat[2], Charles Pézerat[2], François Gautier[2]
[1]CNES, 18 Avenue Edouard Belin, 31400 Toulouse, France
[2]Université du Maine, CNRS UMR 6613, LAUM, Avenue Olivier Messiaen, 72085 LE MANS CEDEX 9, France
emails: margaux.regniez@univ-lemans.fr, adrien.pelat@univ-lemans.fr, nicolas.joly@univ-lemans.fr, charles.pezerat@univ-lemans.fr, francois.gautier@univ-lemans.fr

ABSTRACT: Sandwich honeycomb panels are widely used in aerospace applications because they are very light and stiff. Strong mechanical or acoustical excitations, associated to low damping properties of such panels can lead to high vibration levels, generating fatigue and reliability problems. We propose in this paper to investigate the capability of micro-perforations of honeycomb panels for reducing their vibration levels. Micro-perforations are mostly known and used in acoustics for increasing absorption. A model of the panel vibration damping induced by the acoustic motion in the micro-perforations is proposed here. For this purpose, a lumped element model, based on Maa's results is developped for estimating a viscous damping force at the micro-scale. The resultant force is then homogenized for a group of cells (meso-scale model) and allows us to express a damping term for the global structure (macro-scale model). A perturbation technique is then used to compute the modal damping coefficients of a micro-perforated plate in order to evaluate the performance of the treatment.

KEY WORDS: acoustic impedance, vibration damping, micro-perforations.

1 INTRODUCTION

Sandwich materials composed of two skins glued on both sides of honeycomb cells are widely used in several domains because they offer the possibility to make lightweight and stiff structures. In term of vibroacoustic characteristics, honeycomb sandwich materials have however high acoustic radiation efficiencies, or reciprocally, they are sensitive to acoustic excitations. The aim of the present study is to investigate the capability of micro-perforations to reduce vibrations of honeycomb sandwich materials.

Micro-perforations are well known devices used for the attenuation of acoustic levels. Indeed, micro-perforations provide an acoustic dissipation on the surface of the treated structure due to the viscous effects inside the micro-holes whose dimensions are less than the boundary layer of the fluid. Micro-perforations were particularly studied for rigid structures [1-3], [8-9] and more recently for vibrating structures [10-11], where the objectives are always the improvement of the acoustic absorption. Of course, when the structure is excited by an acoustic field, the decrease of the wall pressure due to the presence of micro-holes implies that the vibration level is also reduced. The reduction of the wall pressure can be quantified by the well known acoustic absorption coefficient quantifying the ratio between the absorbed energy and the incidence energy. This treatment is also particularly effective for high levels, where the dissipation phenomenon becomes non linear [13]. Otherwise, whatever the excitation (airborne or structure-borne excitations), the dynamic comportment of the structure can be changed due to fluid motion (generated by the vibration) inside micro-holes. Viscous effects can then introduce dissipation leading to a non-negligible damping when numerous micro-perforations are considered. This phenomenon was shown on micro-perforated membranes [12]. The goal of this paper is to see if this damping effect is also interesting for flexural motion of honeycomb sandwich panels.

The proposed modeling is divided in three parts, corresponding to models at three different scales: the micro-scale, the mesoscale and the macro-scale. The micro-scale corresponds to the scale of the honeycomb cell, where the physical phenomena are described and modeled to compute the dissipation delivered by one hole. The mesoscale contains several perforated cells, where the damping effect can be homogenized. The macro-scale is the scale of the structure, where the effects on modal properties can be obtained.

2 DYNAMIC MODELING OF THE MICRO-PERFORATED STRUCTURE

All equations of the paper are written at the harmonic regime. The convention chosen is $e^{j\omega t}$, ω being the circular frequency.

2.1 At the micro-scale

The honeycomb sandwich plate is supposed to be modeled by the Kirchhoff's plate theory. As a consequence, at the scale of a single honeycomb cell, the motions of the wall cells (or plate cross-section) can be described by a superposition of two elementary vibratory motions: a translation and a rotation.

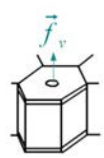

Figure 1: Single microperforated cell and resultant viscous force.

Both structural motions are coupled to the internal fluid motions, inducing the damping effect to be modeled.

Translation of the cell induces an air flow in the hole which is supposed to be be done on one side of the cell. Air flow is also induced by the cell rotation, but is supposed to be very weak. Rotation effects are such ignored in the modeling.

To calculate the equivalent viscous force $\vec{f}_v$ resulting from the microperforated honeycomb cell translation (see figure 1), the fluid motion is described by a mass-spring-damping system as shown on figure 2, for which the equation of motion is:

$$j\omega M_f \dot{x}_f = \frac{-K_f}{j\omega}\left(\dot{x}_f - \dot{w}\right) - C_f\left(\dot{x}_f - \dot{w}\right) \quad (1)$$

where $K_f = \frac{P_0 \gamma S_f^2}{V_{cav}}$ corresponds to the effect of the cell cavity, $M_f = S_f \frac{\Im(Z_{Maa})}{\omega}$ and $C_f = S_f \Re(Z_{Maa})$ are respectively the mass and dissipation of the fluid inside the micro-perforation, computed using respectively the imaginary and real parts of the acoustical impedance given by Maa [1-3]:

$$Z_{Maa} = \frac{32\nu\rho_0 t}{d^2}\left(\sqrt{1+\frac{x^2}{32}} + \sqrt{2}\, x \frac{d}{8t}\right) + j\omega\rho_0 t\left(1 + \frac{1}{\sqrt{3^2 + \frac{x^2}{2}}} + \frac{8}{3\pi}\frac{d}{t}\right)$$

(2)

with, $x = \sqrt{\frac{\omega\rho_0}{\nu}}\frac{d}{2}$, d the hole's diameter, t the hole's thickness, ν the kinematic viscosity, ρ_0 the air density and ω the circular frequency.

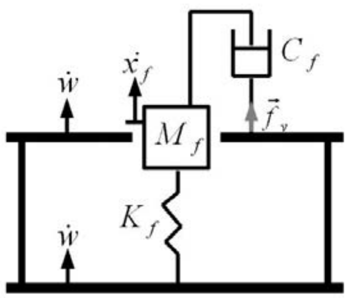

Figure 2: One degree of freedom representation of the fluid movement inside the micro-perforated honeycomb cell.

The applied force on the honeycomb cell due to the fluid dissipation is written as:

$$\vec{f}_v = \left[C_f\left(\dot{x}_f - \dot{w}\right) + \frac{K_f}{j\omega}\left(\dot{x}_f - \dot{w}\right)\right].\vec{z} \quad . \quad (3)$$

Expression (3) is valid only if we consider that the one degree of freedom model described in figure 2 is valid. For an air piston inside a perforation, this is not the case since the stiffness term in (3) is not applied to the plate. This point is not considered in the paper.

It is important to note that the observed reaction is local, which means that each elementary force $\vec{f}_v$ associated to one cell is independent from the forces associated to the neighboring cells, No interaction between adjacent cell is considered.

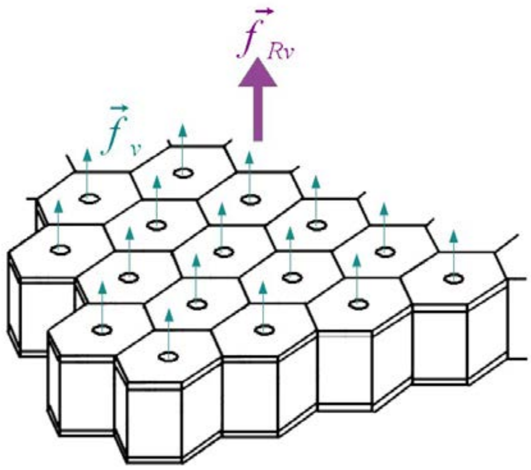

Figure 3: Assembly of microperforated honeycomb cells to form the surface element dS.

2.2 *At the mesoscale*

At the mesoscale the considered surface is noted dS , it is sized to be higher than the cell dimension but small enough compared to the acoustic wavelength.

The relationship between the viscous forces $\vec{f}_v$ (in N) and the resultant force distribution $\vec{f}_{Rv}$ (in N/m²) on dS is:

$$\vec{f}_{Rv}\, dS = N\, \vec{f}_v = \frac{\sigma}{S_f}\vec{f}_v dS \quad (4)$$

where $N = n_{t/c} n_c$ is the number of holes in the surface dS . The number $n_{h/c}$ is the number of holes per cell and n_c the number of honeycomb cell in dS , $\sigma = \frac{n_{t/h} S_f}{S_c}$ is the perforation rate and S_f the surface of a hole. In this study, the number of holes per cell $n_{h/c}$ is fixed to one.

The motion of the elementary surface dS is supposed to be described by the classical bending equation:

$$-\omega^2 \rho h\, w(x,y,\omega)\, dS + D\Delta^2 w(x,y,\omega) dS = \Delta p\, dS + f_{Rv}\, dS \quad (5)$$

where $w(x,y,\omega)$ corresponds to the transverse flexural displacement, ρh is its surface density, D the flexural rigidity and Δp the difference of pressures at each side of the panel. The resultant force distribution $\vec{f}_{Rv}$ given by

$$\vec{f}_{Rv}\, dS = \frac{\sigma}{S_f}\left(C_f + \frac{K_f}{j\omega}\right)\left(\dot{x}_f - \dot{w}\right).\vec{z}\, dS = -C_v\, \dot{w}\,.\vec{z}\, dS \quad (6)$$

allows us to define the damping coefficient C_v which will have sense, as expressed, if it is real and positive.

2.3 *At the macro-scale*

The macro-scale corresponds to the scale of the complete structure. In the following, a rectangular honeycomb sandwich panel is considered . It is considered to be simply supported on its edges. The dimensions are expressed on the figure 4.

Equation of motion (5) is considered on the whole the surface of the structure and established for the harmonic regime:

$$-\omega^2\rho h w(x,y,\omega)+j\omega C(x,y)w(x,y,\omega)+D\Delta^2 w(x,y,\omega)=\Delta p \quad (7)$$

where $C(x,y)=I(x,y)C_v$ (8)

is the damping operator, taking into account the micro-perforation positions, specified thanks to the indicator function $I(x,y)$ ($I(x,y)=1$ if there is a hole at point (x,y), and $I(x,y)=0$ otherwise).

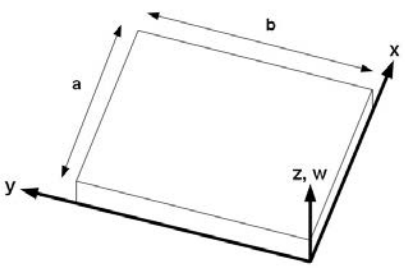

Figure 4: Plate's notations.

The damping properties of the micro-perforated plate are investigated by looking at the modes of the damped plate described by equation (7). Since the damping for the micro-perforated plate is expected to be low, modes of the damped plate are supposed to be closed to the ones of the undamped plates and are computed using a perturbation technique.

The eigen values λ_{0k} and the mode shapes ϕ_{0k} of the undamped plate are the solutions of

$$\left(\rho h \lambda_{0k}^2+D\Delta^2\right)\phi_{0k}=0 \quad , \quad (9)$$

associated to the imposed boundary conditions.

The eigenvalues λ_k and the mode shapes ϕ_k of the damped plate are the solutions of

$$\left(\lambda_k^2\rho h+\lambda_k C(x,y)+D\Delta^2\right)\phi_k=0 \quad (10)$$

and are supposed to be written as

$$\lambda_k=\lambda_{0k}+\delta\lambda \quad (11)$$

$$\phi_k=\phi_{0k}+\delta\phi \quad (12)$$

where $\delta\lambda$ and $\delta\phi$ are correction terms, supposed to be small.

Reporting (11) and (12) into (10) leads to:

$$\begin{aligned}&\left(D\Delta^2+\lambda_{0k}^2\rho h\right)\phi_{0k}+2\lambda_{0k}\rho h\delta\lambda\phi_{0k}+\lambda_{0k}C(x,y)\phi_{0k}+(\delta\lambda)^2\rho h\phi_{0k}\\&+\left(D\Delta^2+\lambda_{0k}^2\rho h\right)\delta\phi+C(x,y)\delta\lambda\phi_{0k}+2\lambda_{0k}\delta\lambda\rho h\delta\phi+(\delta\lambda)^2\rho h\delta\phi\\&+\lambda_{0k}C(x,y)\delta\phi+\delta\lambda C(x,y)\delta\phi=0 \quad . \quad (13)\end{aligned}$$

The third line, the last three terms of the second line and the last term of the first line of equation (13) are second order terms since they correspond to products of small values. The first term of the first line of the same equation is equal to zero because it corresponds to the equation (9) of the conservative associated system. Finally, equation (13) leads to:

$$\left(D\Delta^2+\lambda_{0k}^2\rho h\right)\delta\phi+\left(2\lambda_{0k}\rho h\right)\delta\lambda\phi_{0k}+\lambda_{0k}C(x,y)\phi_{0k}=0 \quad . \quad (14)$$

Projecting equation (14) on the mode ϕ_{0k} gives the correction terms $\delta\lambda$ as

$$\delta\lambda=\frac{-\int_S \phi_{0k}^2 C(x,y)\,dS}{2\int_S \rho h\phi_{0k}^2\,dS} \quad (15)$$

where S is the panel surface. This leads to a modal damping coefficient for the plate equal to

$$\xi_k=\frac{-\delta\lambda}{\omega_{0k}}=\frac{\int_S \phi_{0k}^2 C(x,y)\,dS}{2\omega_{0k}\int_S \rho h\phi_{0k}^2\,dS} \quad . \quad (16)$$

If microperforations are uniformly spread over the structure, we get $C(x,y)=C_v$ and thus,

$$\xi_k=\frac{C_v}{2\omega_{0k}\rho h} \quad . \quad (17)$$

The special case where the hole distribution is uniform leads to the maximum value of the modal damping. In the general case, we have

$$\xi_k=\frac{C_v}{2\omega_{0k}\rho h}-\frac{\int_{\tilde S}\phi_{0k}^2 C_v\,d\tilde S}{2\omega_{0k}\int_{\tilde S}\rho h\phi_{0k}^2\,d\tilde S} \quad (18)$$

where $\tilde S$ corresponds to the non-perforated surface.

Results are presented in the next paragraph for several perforation rates σ .

2.4 *Parametric study*

The analysis of the term C_v (eq. (6)) is made using a parametric study. C_v is written as

$$C_v=\frac{\sigma}{S_f}\left(C_f+\frac{K_f}{j\omega}\right)\left(\frac{j\omega M_f}{\frac{K_f}{j\omega}+C_f+j\omega M_f}\right)=R_v+j\omega I_v \quad (19)$$

where $R_v=\dfrac{\sigma\left(C_f+K_f M_f C_f\right)}{S_f\left(C_f^2-\left(\omega M_f-\frac{K_f}{j\omega}\right)^2\right)}$ and

$I_v=\dfrac{\sigma\left(M_f C_f^2-K_f M_f^2-\frac{K_f^2 M_f}{\omega^2}\right)}{S_f\left(C_f^2-\left(\omega M_f-\frac{K_f}{j\omega}\right)^2\right)}$, and plotted on the figure 5 for three different values of perforation rate, depending on the following holes and cavity dimensions:

Perforation rate σ (%)	Perforations diameter (m)	Honeycomb cell diameter (m)
0,1975	$0,2.10^{-3}$	9.10^{-3}
0,3968	$0,2.10^{-3}$	$6,35.10^{-3}$
0,8928	$0,6.10^{-3}$	$6,35.10^{-3}$

These curves show the classical effect of the resonance of the one-degree-of-freedom oscillator, whose natural frequency is $f_0=\dfrac{1}{2\pi}\sqrt{\dfrac{K_f}{M_f}}$.

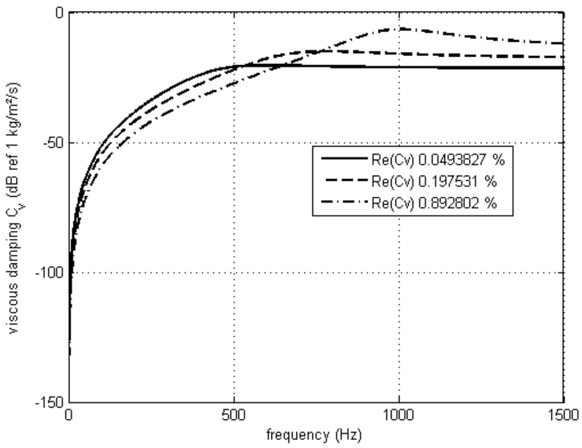

Figure 5: Real part of the viscous damping as a function of frequency, for different perforation rates.

Using equation (17), the modal damping of a rectangular simply supported sandwich plate (dimensions 0,5m x 0,33m x 0,02m and bending stiffness 150 kg.m².s^{-2}) is computed for the first five modes (between 200 and 1000 Hz). Figure 6 plots the results as a function of perforation rate σ .

Eigenfrequencies of the plates are found to be 210 Hz, 405 Hz, 648 Hz, 729 Hz and 842 Hz.

The order of magnitude of the modal damping is rather low and is found to be 1.10^{-8} for first and second modes. For modes of higher frequency and for higher perforation rates, the modal damping is found to be 2.10^{-6}. A simple computation allows us to estimate the value that such modal damping should have to obtain a mobility difference of 1 dB between the non-perforated and the perforated panels. This value is found to be 10^{-3}. Hence, the current treatment as dimensioned here, leads to a too low damping to reach this mobility difference.

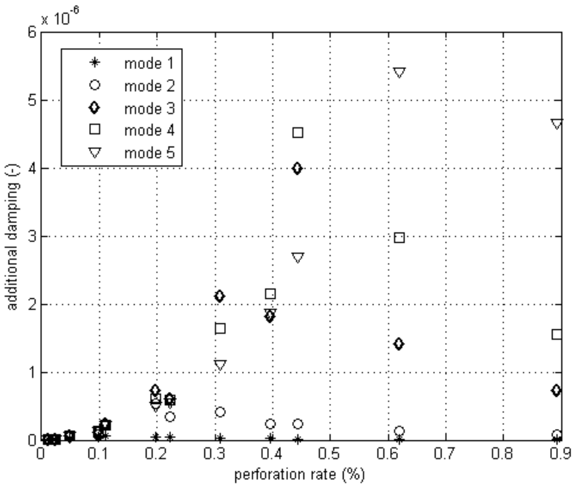

Figure 6: Modal damping as a function of perforation rates, plotted for the first 5 modes.

3 CONCLUSION

In this paper, we propose a methodology for computing the modal damping induced by micro-perforations performed on one side of a honeycomb sandwich panel. This method is based on the Maa model for the fluid motion inside each hole, which allows us to estimate an equivalent viscous force (micro-scale model). The resultant force is then computed considering a group of cells (meso-scale model) and allow us to express a term, whose real part expresses apparent damping for the global structure (macro-scale model).

A perturbation technique is used to estimate the modal damping coefficient. The model can take into account a non-uniform distribution of microperforations and the upper bound result for the modal damping has been found. Indeed, it corresponds to the case where the distribution of microperforations is homogeneous on the structure.

The order of magnitude observed for such modal damping is found to be rather low for the proposed geometry.

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