A TWO-SCALE APPROACH FOR ASSESSMENT OF THE HONEYCOMB CORE SHEAR EFFECTS ON THE TRANSMISSION LOSS

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ABSTRACT

The main purpose of the work reported here is to bring out the effects of the different meso-scale parameters of the honeycomb sandwich panel on their vibro-acoustic response using a meso-macro approach. The present approach is developed using a numerical method known as a wave finite element method (WFEM). The WFE method combines the classical finite element method (FEM) and the periodic structure theory (PST). The main advantage of this method is that it takes into consideration the periodicity of the structure, which allows to model typically just one elementary cell instead of the whole structure. Accordingly, the calculations cost is hugely reduced. In addition, this numerical model keeps the meso-scale parameters of the periodic cell. The obtained results are compared with different analytical methods and commercial tool (Ms-NOVA), showing a very good agreement. A vibro-acoustic parametric analysis of the honeycomb panel with composite face-sheets is done. This analysis showed a great influence of the cell size and of the core material on the transmission loss (TL).

1 INTRODUCTION

In the last decade, composite materials are considered as the most successful and the most promising materials to be used in many advanced industrial fields. Aerospace, transportation, and other branches of civil and mechanical engineering are the major beneficiaries of their tremendous growth. Apart from their considerably low ratio of weight to strength, some composites benefit from other desirable properties, such as corrosion and thermal resistance, toughness and low cost. Yet from an acoustic point of view, decreasing the mass, while keeping a high level of stiffness, could have a significant influence on the vibro-acoustic performance of the honeycomb sandwich panel. As a result, this might lead to unsatisfactory noise reduction efficiency.

The prediction of accurate wave dispersion characteristics in a cellular honeycomb core bonded by two laminated orthotropic face-sheets is a key information for computing the vibro-acoustic indicators. Over the last few decades, various analytical methods have been developed to predict the wave dispersion characteristics. Erickson [1] and Clarkson [2] developed methods of estimating the modal density of typical honeycomb sandwich panels with isotropic face-sheets. These methods take into account the effect of shear of the core on the sandwich panel's deflection. Renji et al. [3] introduce the orthotropic bending properties in both directions and include the core's transverse shear stiffness in a new analytical model. However, these analytical methods do not reveal the meso-scale influence on the acoustic transmission of the sandwich structure. Therefore, a numerical method need to be employed.

The numerical method reported here enables to predict the wave propagation characteristics within a sandwich structure which provides a key to decrypt its vibro-acoustical behavior. The method known as WFE combine the classical finite element method and the theory of periodic structures. A vibro-acoustic parametric analysis is then performed on the transmission loss (TL) in order to study the effect of different parameters of the unit cell.

2 OVERVIEW OF THE 2D WFE METHOD

The Wave Finite Element method (WFE method) is applied for predicting the dynamic behavior of a periodic structure. The method includes the reformulation of the equation of motion by using the dynamic stiffness matrix. This matrix involves the mass and stiffness matrices of a periodic cell of the sandwich structure. Structural wave motion of the sandwich structure is expressed in terms of the eigenvalues and the eigenvectors of the dynamic stiffness matrix (DSM) and these eigenvalues and eigenvectors represent the wavenumbers and the wave modes respectively.

The equation of motion for periodic structural waveguides can be expressed as follows

$$D \begin{pmatrix} U_{bd} \\ U_I \end{pmatrix} = \begin{pmatrix} f_{bd} \\ 0 \end{pmatrix}. \tag{1}$$

Where $D = (1 + j\eta)\, K - \omega^2 M$ is the dynamic stiffness matrix which is obtained from the stiffness and the mass matrices extracted by using a finite element method package like Ansys. U_{bd} and f_{bd} are respectively the displacement and the force of the boundary nodes. While U_I represents the internal nodes.

Using the Floquet-Bloch theory for a periodic rectangular cell and assuming a time-harmonic response, the displacements of each edge can be written as a function of the displacements at one single edge.

$$\Lambda_L(\lambda_x, \lambda_y) D \Lambda_R(\lambda_x, \lambda_y) \begin{pmatrix} U_1 \\ U_L \\ U_B \end{pmatrix} = 0. \tag{2}$$

By introducing the matrices $\Lambda_L(\lambda_x, \lambda_y)$ and $\Lambda_R(\lambda_x, \lambda_y)$ in the equation (1) a polynomial equation of second-order obtained as written in the equation (3)

$$\frac{1}{\lambda_x^2}\left(A\lambda_x^2 + B\lambda_x + C\right) \begin{pmatrix} U_1 \\ U_L \\ U_B \end{pmatrix} = 0. \tag{3}$$

Where $\lambda_x = e^{i\mu_x}, \lambda_y = e^{i\mu_y}$ and $\mu_x = k_x L_x, \mu_y = k_y L_y$ are the propagation constants of a plane harmonic wave in both x- and y-direction respectively. While k_x, k_y are the wavenumbers along the x- and y-direction respectively.

The sound transmission through an infinite sandwich panel can be calculated by knowing the dispersion curve of the bending wave through the plate. For a flat plate impacted by an acoustic plane wave incidence α and ϕ direction, the acoustic transparency is defined as the ratio of transmitted power through the plate to the incident power. It can be calculated as follows:

$$\tau(\alpha, \phi) = \left\{ \left[1 + \eta \frac{\omega m \cos\alpha}{2\rho_0 c_0} \frac{k_0^4 \sin^4\alpha}{k_{eq,\phi}^4} \right]^2 + \left[\frac{\omega m \cos\alpha}{2\rho_0 c_0} (1 - \frac{k_0^4 \sin^4\alpha}{k_{eq,\phi}^4}) \right]^2 \right\}^{-1}. \tag{4}$$

Where the equivalent wavenumber in the equation (4) is written as follows :

$$\frac{1}{k_{eq,\phi}^2} = \frac{\sin^2\phi}{k_{Bx}^2} + \frac{\cos^2\phi}{k_{By}^2}. \tag{5}$$

The acoustic transparency diffuse field is calculated by averaging all the possible incidents and directions.

$$\tau = \frac{\int_0^{2\pi} \int_0^{\pi/2} \tau(\alpha, \phi) \sin\alpha \cos\alpha \, d\alpha \, d\phi}{\int_0^{2\pi} \int_0^{\pi/2} \sin\alpha \cos\alpha \, d\alpha \, d\phi}. \tag{6}$$

Finally, the sound transmission loss is given by the following expression :

$$TL = 10 \log(\frac{1}{\tau}). \tag{7}$$

The parameters to be studied in this vibro-acoustic parametric analysis are the young modulus of the core E, the cell angle θ (the angle between horizontal cell wall and inclined cell wall), the thickness of the walls t, the thickness of the core h_c, and finally the cell size l and h of the periodic cell (see figure 1).

3 RESULTS

A periodic segment of a honeycomb sandwich panel with orthotropic face-sheets is considered hereby (see figure 1) with L_x, L_y its surface's dimensions and $h_t = h_c + h_f$ its total thickness. The geometrical and material properties of the periodic segment were summarized in Table 1. The vibro-acoustic study is performed in a frequency range between $0\ Hz$ and $5000\ Hz$.

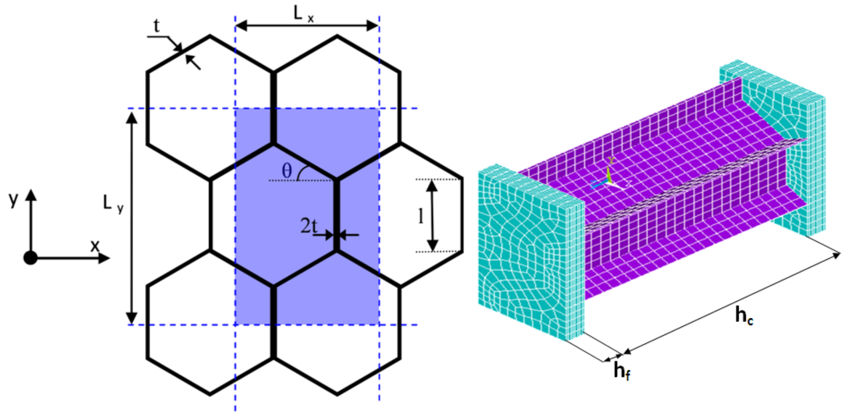

Figure 1. Cell geometrical parameters of the periodic hexagonal cell.

	Core (Nomex material)	Skins (Epoxy resin with carbon yarn)
Material	$E = 5.5\ GPa$ $\rho = 1240\ kg/m^3$ $G = 2.07\ GPa$ $\mu = 0.33$	$E_1 = 133.6\ GPa$ $E_2 = 7.7\ GPa$ $E_2 = 7.7\ GPa$ $G_{12} = 3.1\ GPa$ $\mu_{12} = \mu_{13} = 0.29$
Geometry	$h_c = 12\ mm$ $t = 76.2\mu m$ $l = h = 2.7\ mm$ $\theta = 30^o$	$h_f = 1\ mm$ $[0, 45, 90, -45]_s$ $e_{UD} = 125\ \mu m$

Table 1. The material and geometrical parameters of the periodic cell.

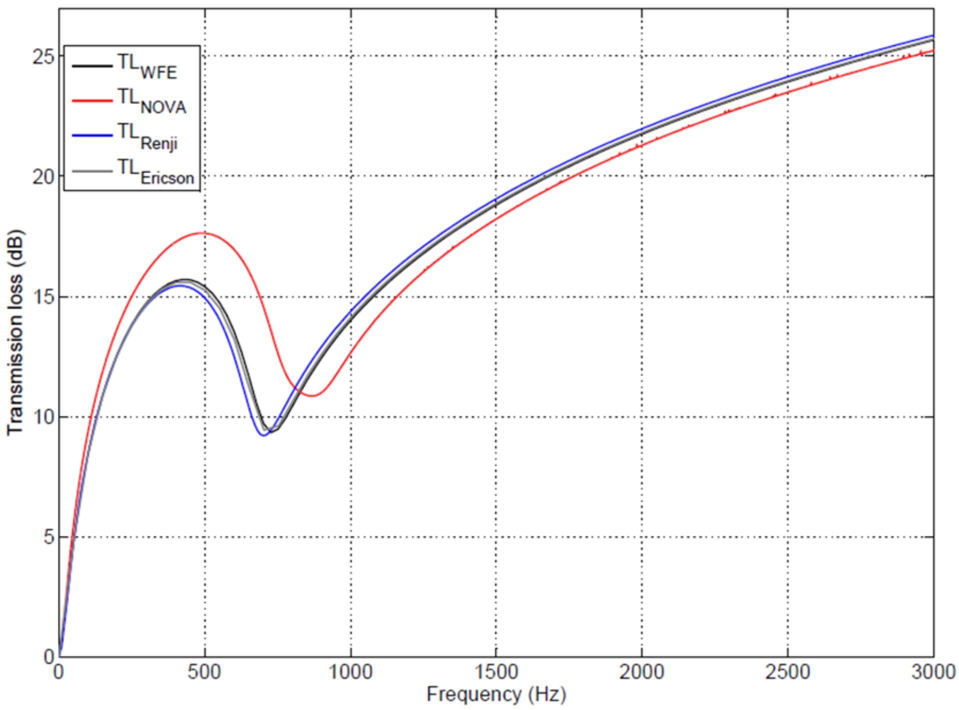

Figure 2. Comparisons of different analytical and numerical models in terms of TL.

Figure 2 present the comparison of the current model with analytical models and commercial tool (Ms-Nova). The comparison of the meso-macro approach with the analytical model shows a good agreement either in the critical frequency or in the rest of the frequency range. However, the comparison with commercial tool exhibits a slight difference, these initial comparisons allow us to validate the present model.

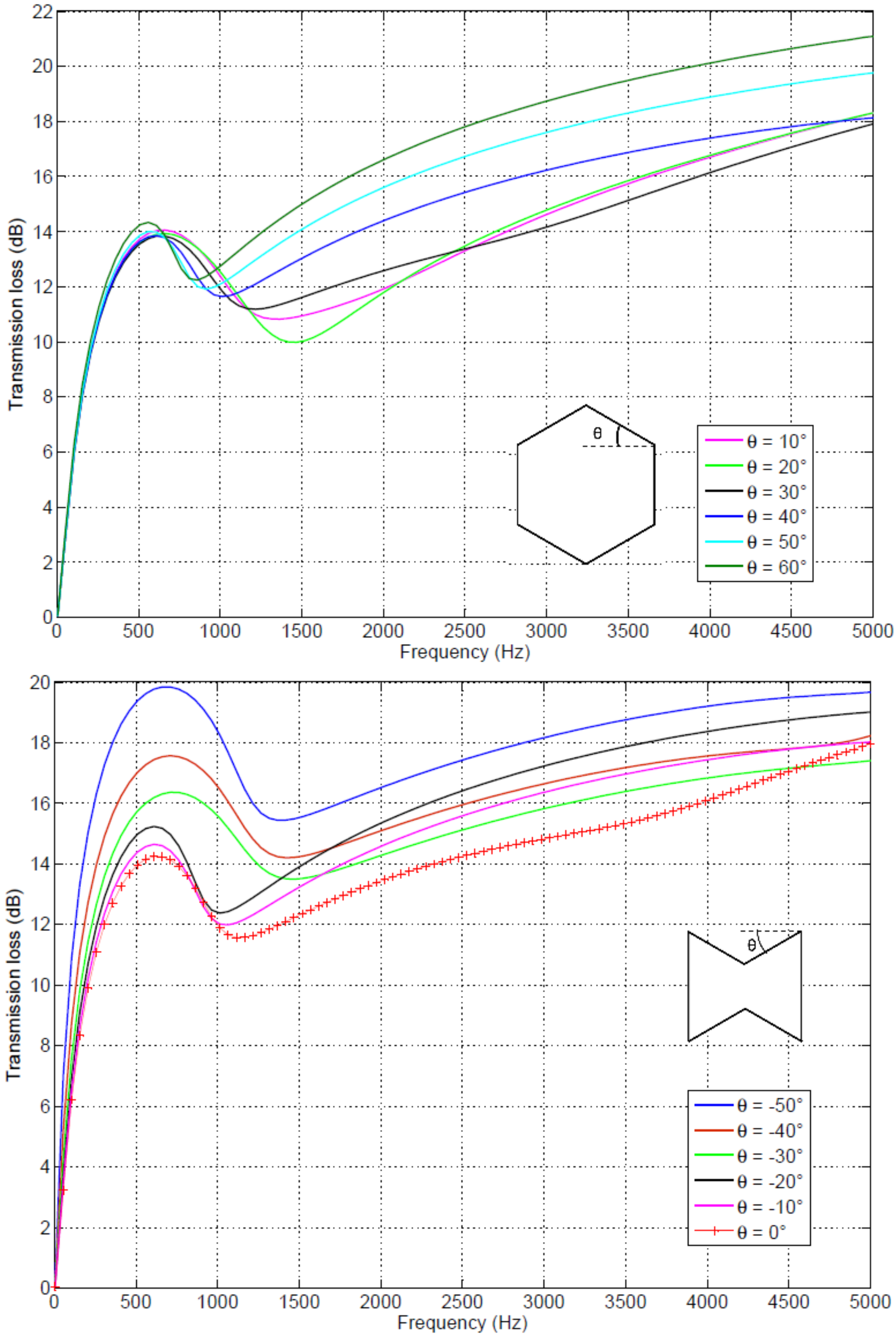

Figure 3. Effect of the cell angle θ on the transmission loss.

The parametric study on the cell angle θ presented in Figure 3 indicates that the cell angle θ generally influences on the sound transmission loss (TL) whether in the low, medium or high frequency ranges. On the one hand, with the positive cell angle θ, the more the angle increases the more the critical frequency decreases and the sound transmission loss is enhanced throughout the whole frequency band. On the other hand, for the negative cell angle θ, the more the angle decreases the more the sound transmission loss rises as well as the critical frequency is shifted.

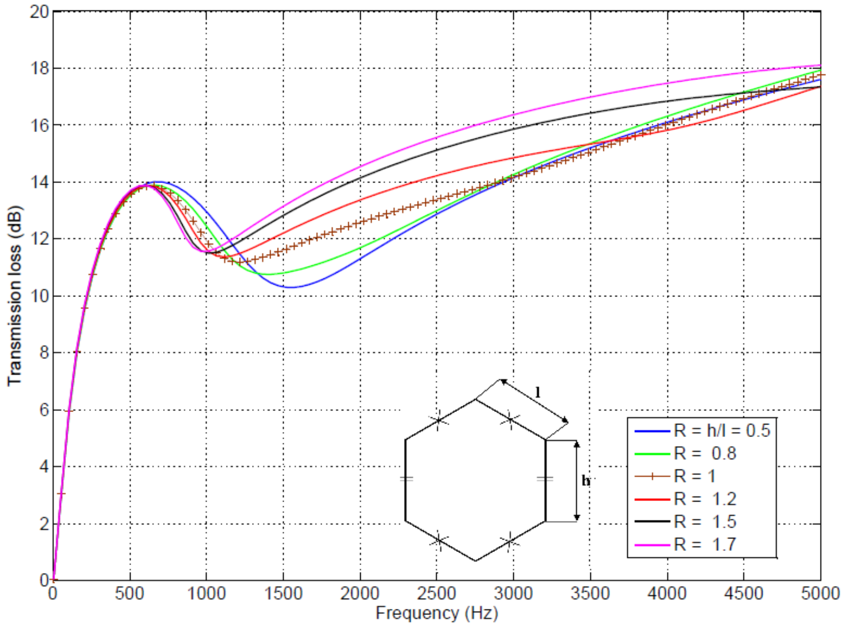

Figure 4. Effect of cell size h and l on the transmission loss.

Figure 4 exhibits the comparison of the different curves of the sound transmission loss while varying the cell size (h and l).The comparison showed that the more the parameter h increases the more the sound transmission loss improves. However, the critical frequency of the sandwich panel decreases.

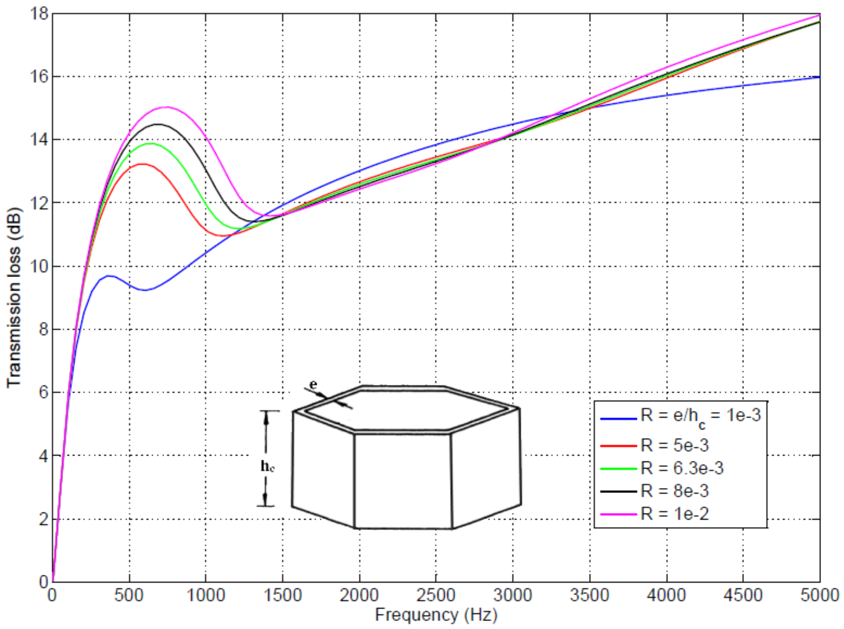

Figure 5: Effect of the thickness of the core h_c and the thickness of the walls e on the transmission loss.

The comparison performed in Figure 5 by changing the ratio of e/h_c, shows that the more the thickness of the core's sandwich panel h_c decreases and the thickness of the walls of the periodic cell e increases, the more the sound transmission loss enhances and the critical frequency shifts.

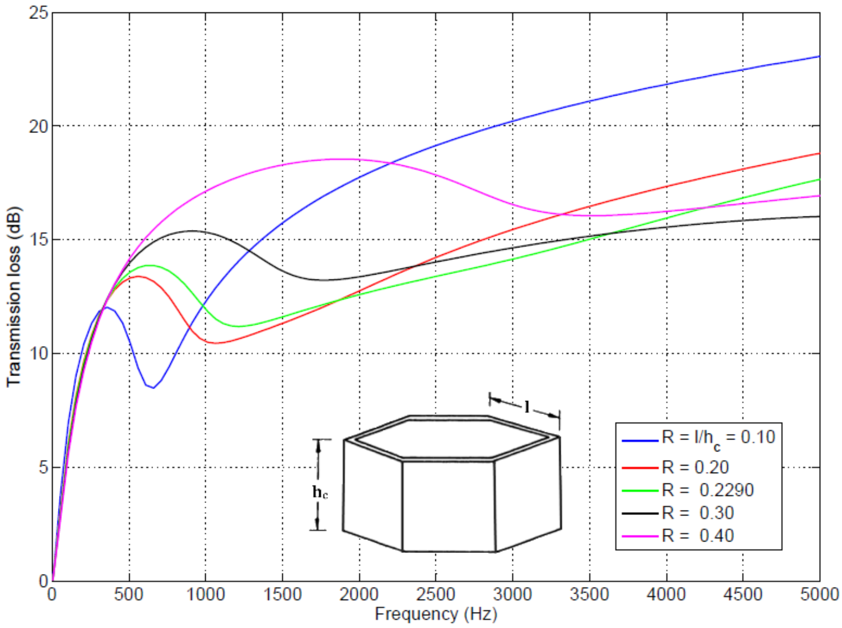

Figure 6. Effect of the cell size l and the thickness of the core h_c on the transmission loss.

The next comparison is carried out between the thickness of the core's sandwich panel h_c and the cell size l. Figure 6 shows that the more the cell size l increases with respect to the thickness h_c the more the critical frequency increases until the transmission loss curve becomes smooth. However, the sound transmission loss curve decreases in the different frequency range.

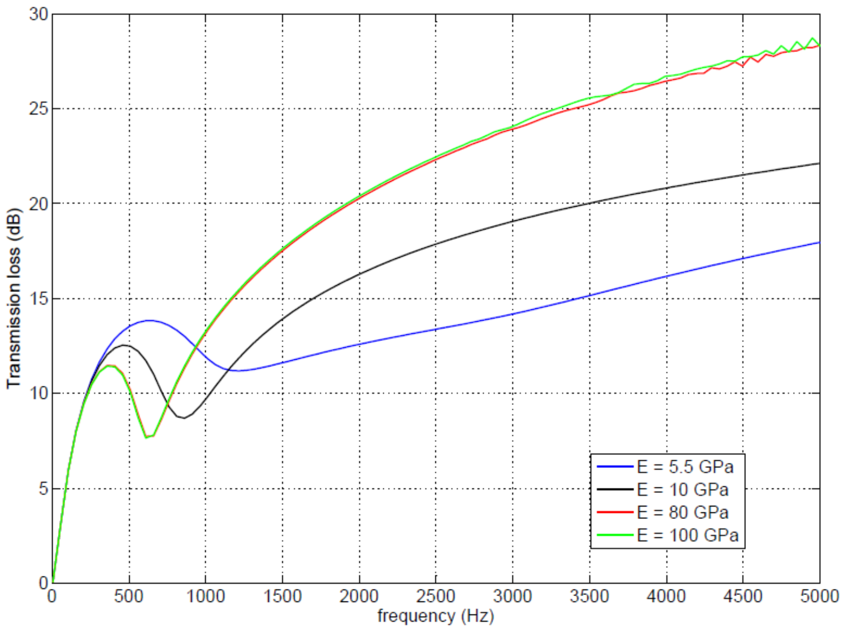

Figure 7. Effect of the young modulus E of the core on the transmission loss.

In Figure 7, a comparison of the sound transmission loss (TL) with respect to the young modulus E of the core is presented. The comparison exhibits that the more the young modulus E increases the more the sound transmission loss enhances up to a certain value at which the sound transmission loss curve will not increase. However, the critical frequency decreases when the young modulus increases.

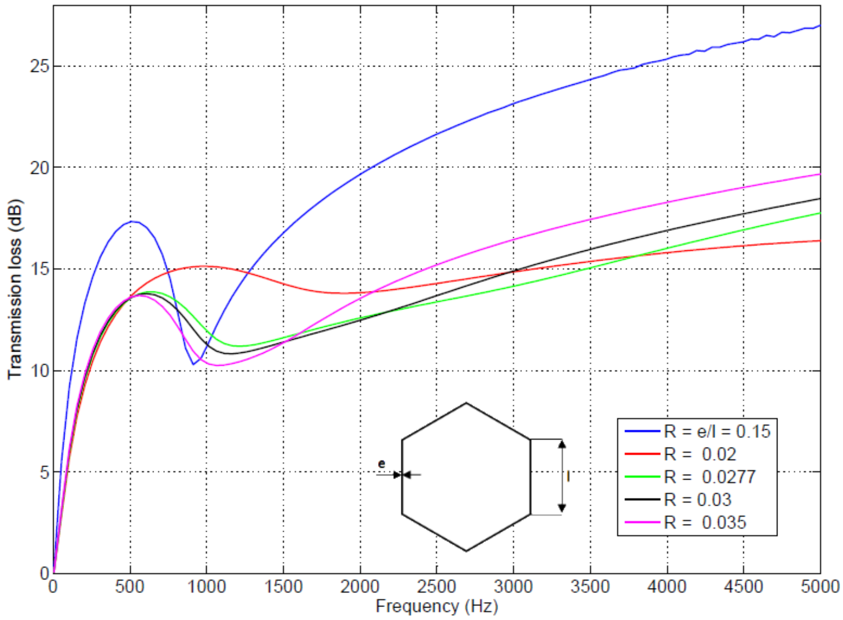

Figure 8. Effect of the cell size l and the thickness of the walls e on the transmission loss.

In Figure 8, a parametric study of the ratio e/l the walls' thickness e with respect to the cell size l is presented. the study indicates that the more the walls' thickness of the cell increases the more the critical frequency shifts until the curve becomes smooth. However, The sound transmission loss curve decreases when the ratio e/l increases.

4 CONCLUSION

The presented vibro-acoustic parametric analysis used the meso-macro approach, based on the wave finite element method (WFE), showed clearly that the geometrical and material properties of the periodic unit cell of the panel has a significant influence on the sound transmission loss (TL) as well as on the shifting of the critical frequency. In the present vibro-acoustic study, when changing two parameters at the same time, the criteria was to maintain the mass constant. Subsequently, This vibro-acoustic parametric analysis will facilitate the next step which is the optimization study. This later will permit to define the optimal design parameters of the honeycomb sandwich panel.

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