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Modeling and experimental measurements of the sound transmission loss for multi-layer core topology systems

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Abstract

In this paper, we focus on multi-layer core topology systems which consists in piling up layers made by different kind of cores and offers innovative solutions to periodic structures design. The main objective of this study is to analyze the phenomena occurring in these multilayer core topologies, and then compare the numerical models with experimental measurements while the effects of the phase shift on the STL will be investigated. The model is implemented using MATLAB and ANSYS apdl. The mass of the unit cell is taken as constraints to eliminate its influence on the STL and the compression modulus is calculated to verify the mechanical properties. To perform the experimental measurements, the samples are designed with CATIA and then made by an industrial 3D printer using the fused deposited model technic. An impedance tube is used to measure the STL and compare with the vibroacoustic phenomena.

1 Introduction

A great challenge of these last decades is to make lightweight structures with efficient acoustical and mechanical performances for many industrial applications (aeronautics, aerospace, building construction, automotive, ...). Sandwich panels with laminates, composite structures as well as porous media have been developed to propose new solutions. Alternatively, periodic structures are widely used in aerospace industry since the stiffness to weight ratio is low. In terms of mechanical properties, the rigidity of the core enhances the tensile and compression modulus of the whole structure. The core geometry creates an anisotropic material with several main directions. Under wave excitations and at some frequencies it will make occur specific characteristics of the structure. Periodic structures can be easily characterized by the unit cell and many models can be applied to investigate their vibroacoustic properties.

The principle of the wave method firstly explained in [1] has been implemented using Finite Element Method and became the Wave Finite Element Method (WFEM) described in [2] for 1D and 2D periodicity. Analytical formulation combined with homogenized formulation can be used to determine the Sound Transmission Loss as it is done in [3] using the k-space obtained with the WFEM. Nevertheless, it is limited in terms of structure complexity due to the use of homogenized method. Besides, another approach using the Transfer Matrix Method ([4]) is proposed in [5] and reduce drastically the computational cost. In [6], the method is based on nodal surfaces and uses a simple formulation to obtain the acoustic properties. Similarly, [7] manipulates the shape functions associated with the displacement of the panel to develop the formulation to obtain the acoustic properties. Recently, the wave-based method has been used in [8] allowing to model any kind of acoustic excitations.

By exploiting these models, the study of design space is done by changing the geometry of the core and by analyzing the influence on the STL. In [9, 10], by computing a parametric survey as well as an optimization, they showed that modifying the shape of the core can improve the sound properties or by altering some

parameters of the core and keeping the same geometry, leading to the possibility to optimize the structure. These effects have been investigated in [11, 12] as well. In [13], several core geometries (rectangular, triangular, hexagonal, ...) are compared to optimize the transition frequency, corresponding to a vibroacoustic indicators.

Concerning multi-layer systems, [14, 15, 16, 17, 18, 19] have exploited the way to pill up several layers made of corrugated cores. These studies were mainly restricted for vibration and mechanical properties. In [20], the experimental measurement indicated the efficiency of a double-layer honeycomb panel compared to a single-layer especially in low frequency.

In this paper, a sandwich panel will be studied. By shifting the unit cell, it will offer the possibility to obtain a multi-layer that will be investigated in terms of mechanical and acoustic properties. Results will be validated by experimental measurements to conclude on the efficiency of making this type of structure.

2 Model

A sandwich panel with a standard rectangular core is used as a reference result. The principle is to make the new geometry by shifting a third of the unit cell in two directions x and y as illustrated in the Fig.1. This layout allows to keep the mass of the unit cell constant. The total thickness of the unit cell is 24 mm (y-direction), 15 mm in the x-direction and 15 mm in the z-direction. The thickness of the skins is 2 mm and the thickness of the core is 0,7 mm.

The following sandwich panels are made of ABS with $E = 1,6 \times 10^9$ Pa, $\rho = 985$ kg/m³ and $\nu = 0,33$. The damping η is evaluated at 2 %. Theses values were obtained by characterizing the material.

A finite element analysis is made with the commercial software Ansys apdl. SHELL181 with a size of 1 mm is used to build the finite element model shown in Fig. 1.

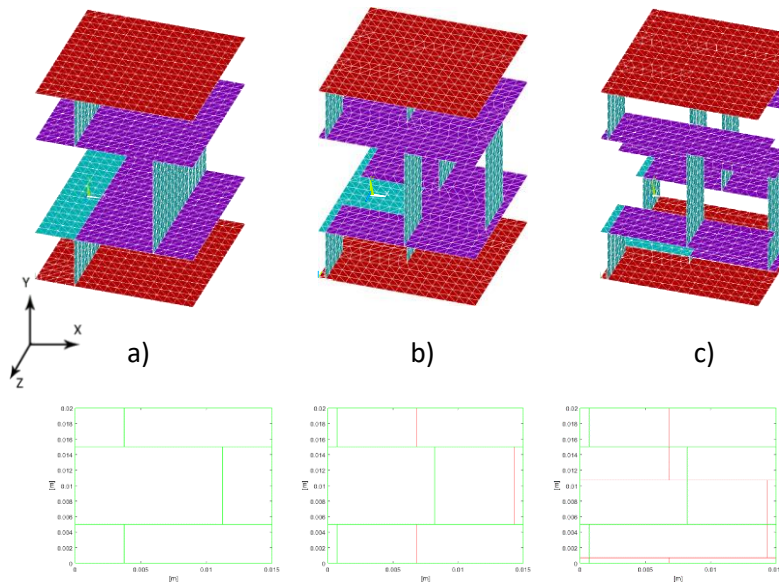


Figure 1 : a) Standard; b) x-Shift c) xy-Shift

2.1 Sound Transmission Loss

The studied structure is excited by normal incident waves considered as plane waves. It is then divided in three kind of acoustic plane waves: the reflected, the absorbed and the transmitted waves. The sandwich panel separates two semi-infinite fluids with $\rho_0 = 1,25 \text{ kg/m}^3$ and $c_0 = 343,6 \text{ m/s}$.

Finally, the Sound Transmission Loss (STL) is obtained using the Eq.1:

$$STL = 10 \log_{10} \left(\frac{W_i}{W_t} \right) \quad (1)$$

The method developed in [6] uses the following assumptions:

- The structure is assumed as an infinite panel.
- Only real wavenumbers are considered.
- The structure is excited by plane waves.
- The angles of incidence and reflection are equal.

A modal analysis is performed with Ansys apdl with free boundary conditions. The stiffness and mass matrix are extracted from the unit cell and allow to calculate the dynamic stiffness. The WFEM is then used to apply the Bloch-Floquet theory. The displacement on the incident and transmitted side and the external forces applied on the structure are linked with the dynamic stiffness and yields to Eq. 2:

$$\begin{bmatrix} b_{ii} & b_{it} \\ b_{ti} & b_{tt} \end{bmatrix} \begin{pmatrix} u_i \\ u_t \end{pmatrix} = \begin{pmatrix} e_i \\ e_t \end{pmatrix} \quad (2)$$

with u_i and u_t the node displacement on the incident and transmitted side whereas e_i and e_t are the external forces applied on the structure, respectively.

Considering the nodal surfaces of the incident and transmitted side (called S_i and S_t) of the unit cell, the external forces applied on the nodes can be rewritten as the product of the nodal surfaces multiply by the pressure evaluated on the node.

The continuity of normal particle velocity at the interfaces gives:

$$\begin{cases} \rho \omega^2 u_i = -ik_y (p_i - p_r) \\ \rho \omega^2 u_t = -ik_y p_t \end{cases} \quad (3)$$

k_y propagative constant in z-direction, p_i , p_r and p_t the incident pressure, the reflected and the transmitted wave, respectively.

Then, a matrix equation between the acoustic pressures of each side is obtained and becomes:

$$\begin{pmatrix} b_{ii} - \frac{S_i}{Y_0} \\ b_{ii} \end{pmatrix} p_i = \begin{bmatrix} b_{ii} + \frac{S_i}{Y_0} & -b_{it} \\ b_{ti} & -b_{tt} - \frac{S_t}{Y_0} \end{bmatrix} \begin{pmatrix} p_r \\ p_t \end{pmatrix} \quad (4)$$

Consequently, by solving the Eq. 4 the reflected and transmitted pressures are known and allow to calculate the STL. It is easy to implement the methodology with MATLAB since it consists on solving a matrix equation.

This method is applied for the three geometries and compared on the following Fig. 2. The frequency range is chosen from 100 Hz to 1600 Hz corresponding to the valid frequency range of the impedance tube.

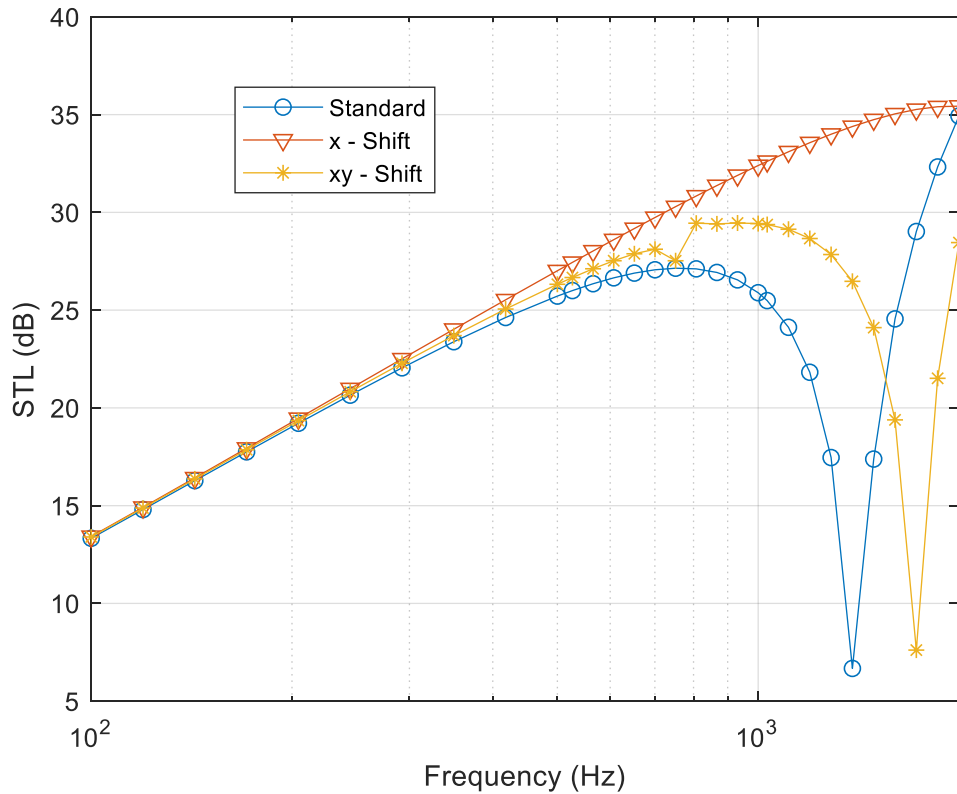


Figure 2 : Comparison of the Sound Transmission Loss of three sandwich panels made of three different unit cell

It is occurred that shifting the core improves the sound transmission loss and shifts the coincidence frequency to higher frequencies. The low mass is kept for a larger frequency band. The x-shift sample turns out being the best layout with the best acoustic performances. Indeed, the more the coincidence frequency will be in higher frequencies the more the structure will be efficient and reliable in terms of sound transmission loss. Finally, it seems that the shift in y-direction decrease the acoustic efficiency of the sandwich panel.

2.2 Compression Modulus

Since the geometry of the core changes, it has been chosen to calculate the compression modulus as a relevant indicator to verify the mechanical properties of the sandwich panel. The same boundary conditions as a compression machine are applied:

1. only the displacement in y-direction is free
2. A unit force is applied on the top skin
3. All nodes of the top skin are constraint to have the same displacement.

The commercial software Ansys apdl and CATIA are used to model the unit cell. SHELL181 is still used with Ansys apdl whereas Solid Elements are chosen for CATIA. The finite element models are illustrated in Fig. 3 and the result obtained shown in Table 1.

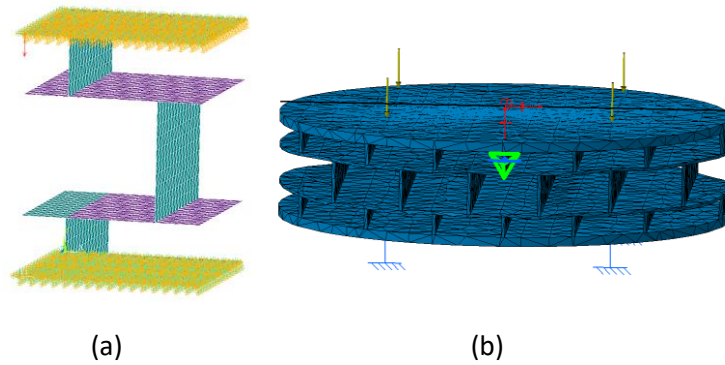


Figure 3 : Finite element model a) Ansys apdl b) CATIA

	Ansys apdl (MPa)	CATIA (MPa)
Standard	$2,3 \times 10^5$	$5,7 \times 10^7$
x - Shift	$6,77 \times 10^6$	$1,08 \times 10^8$
xy - Shift	$7,3 \times 10^5$	$6,94 \times 10^7$

Table 1 : Calculation of the compression modulus

The differences between the two software could be mainly due to the element used since with solid elements the physics is better captured and closer to the reality. Nevertheless, the conclusion remains the same for both simulations: the x-Shift sample occurs to have the highest compression modulus. Indeed, adding the phase shift lead to well distribute the load on the skins and increase the rigidity of the whole sandwich panel. Shifting the unit cell in the y-direction creates nodes contact at some locations and drastically decrease the compression modulus.

Finally, the x-Shift sample appears to be the optimize structure to obtain the best acoustic and mechanical performances and keeping the mass constant. These results still need to be validated by measurements.

3 Experimental Measurements

3.1 Impedance tube

The experimental measurements were performed to verify the efficiency of the optimize structure as expected from the modeling part. A four microphones B&K standard wave tube (Fig. 4) with the two loads method was used. The loudspeaker is located at the end of the tube generating a random noise signal in the frequency range from 200 Hz to 1600 Hz. The B&K 4206 large tube with a diameter of 100 mm was chosen. The frequency response functions are measured with the four microphones and all transfer functions are computed. Two measurements are needed to calculate the sound transmission loss:

1. Measurement with an open tube termination
2. Measurement with an anechoic termination made using 4 standards sound absorbing samples with a 100 mm depth in total.



Figure 4 : Impedance tube (B&K)

3.2 Manufacturing process using the 3D printing technic

Both samples were made using a the FORTUS 250, a 3D printer using the fused deposited model technic making the structure layer by layer. The 3D printer can add two kinds of matter: the model material, corresponding to the material of the structure, and the support material helping to build the structure, but which will be removed after immerging the structure inside a solution.

The samples were firstly designed with CATIA and then preprocessed using INSIGHT as illustrated in Fig. 4. To improve the quality of the structure it is needed to add the support material. The thickness of the deposited rob is 0.3556 mm while the thickness of each layer is 0.1778 mm.

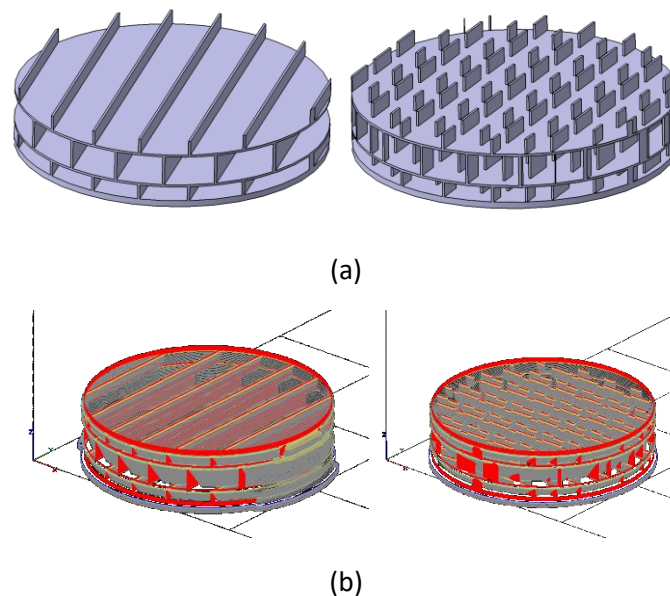


Figure 5 : (a) Samples designed using CATIA (b) Preprocessing using INSIGHT

The final samples are shown in Fig. 6. Since the structure is rigid and it is difficult to accurately reach 100 mm to fit inside the tube, it has been chosen to manufacture the sample with a diameter of 97 mm to overcome the uncertainty of the manufacturing process. Then, to set up the sample inside the tube, a circular joint made of rubber and scotch tape (Fig. 7) was made to avoid air leakage.

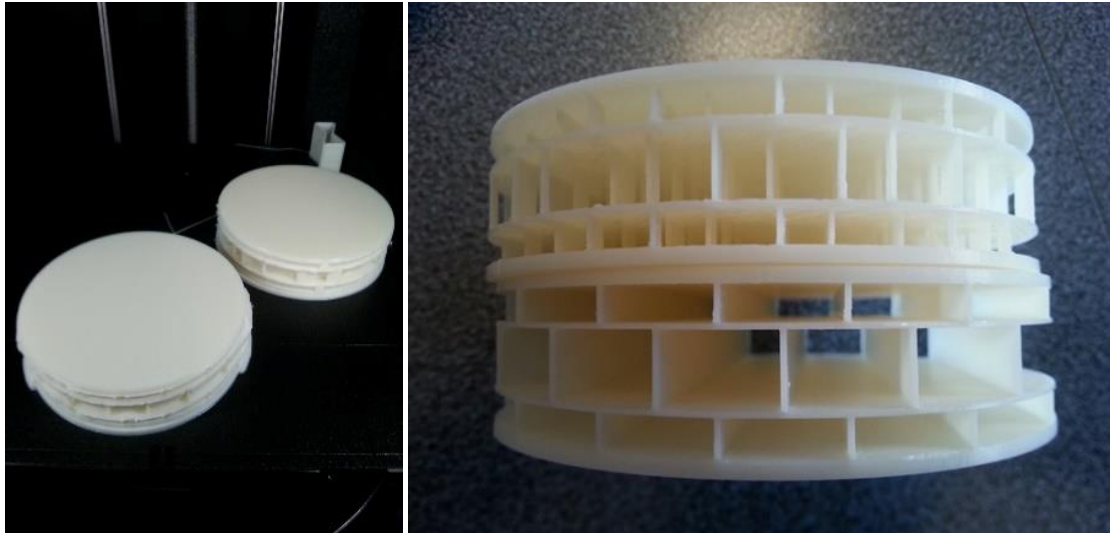


Figure 6 : Printed samples



Figure 7 : Joint made of rubber and scotch tape

The weight of both samples was measured and listed in Table 2.

	Without rubber joint	With rubber joint
Standard	47,29 g	69,93 g
x - Shift	47,80 g	68,72 g

Table 2: Samples weighth

Without the joint the difference between both samples is 1 % while adding the joint it reaches 1,7 %. These differences are neglectable and will not affect the mass low corresponding to low frequencies.

3.3 Results

The obtained result with the impedance tube are shown in Fig. 8. Three measurements were done with the standard case to verify the repeatability. As it was expected the x-Shift sample reveals the best acoustic performance. Before 1000 Hz, the difference is mainly due to the boundary conditions that are not perfectly respected and experimental conditions including measuring errors by microphones. After 1000 Hz, the x-Shift shows a great efficiency and reaches more than 7 dB better than the standard sandwich panel.

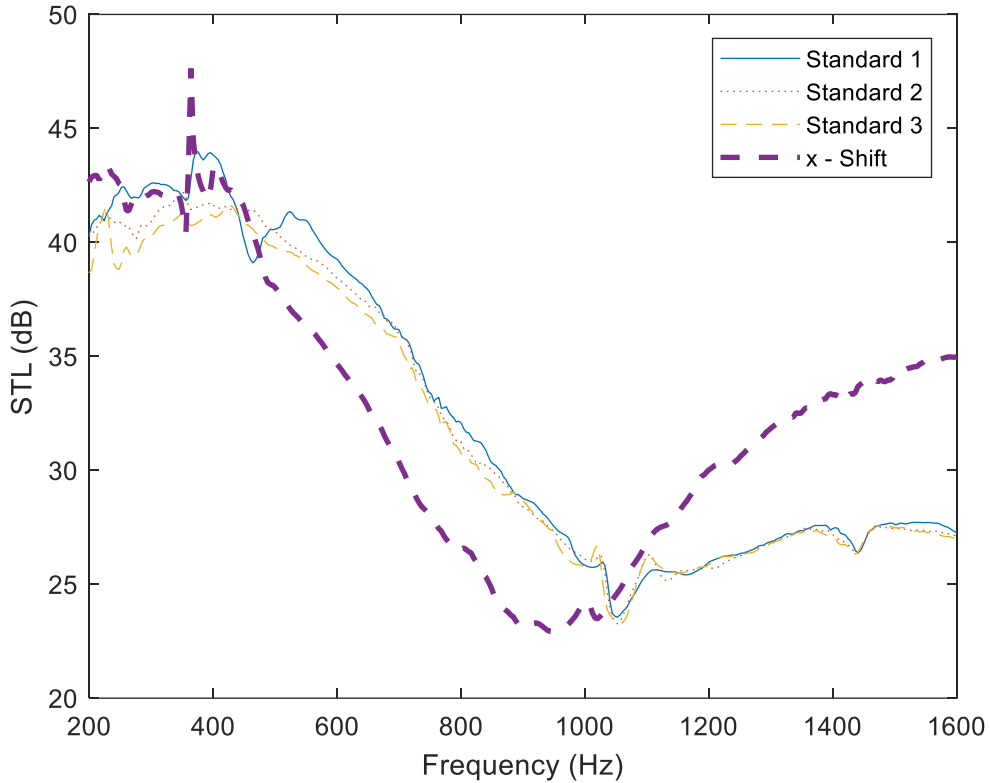


Figure 8 : Comparison of the sound transmission loss with the impedance tube (B&K)

4 Conclusion

This work proposes a new way to design sandwich panels creating multi-layer core topology systems. Three samples were designed by shifting a third of the unit cell in two directions. They were modeled under normal incident waves using the Wave Finite Element Method and then compared in terms of acoustic and mechanical properties. The sound transmission loss and the compression modulus were chosen as acoustic and mechanical relevant indicators.

The model revealed the possibility to increase the compression modulus of the sandwich panel as well as the sound transmission loss by shifting the unit cell in the x-direction and keeping the mass constant.

A 3D printer was used to manufacture the samples and then, a joint made of rubber was added to ensure the samples to fit inside the impedance tube without air leakage. The measurements were done using a Tube de Kundt from B&K with a diameter of 100 mm and shown the efficiency of shifting a part of the unit cell.

Further studies can be carried out to predict the efficiency of any type of unit cells and an optimize structure could be obtained.

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