Structural-acoustic optimization of 2-D Gradient Auxetic Sandwich Panels

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ABSTRACT

This paper investigates the vibroacoustic behavior of sandwich structures made of 2-dimensionally gradient auxetic hexagonal core. Homogenized finite element model has been used to determine the mechanical properties of the auxetic structures. Then this model is used to find the natural frequencies and radiated sound power level of sandwich panels made by the auxetic gradient cores. The radiated sound power level of the structure over the frequency range of 0–200 Hz is minimized by modifying the core geometry of the 2- dimensionally gradient auxetic sandwich panels. To do the optimization, Genetic algorithm method has been applied establishing an interactive link between MATLAB and ANSYS software. The results of this research present significant insights into the design of auxetic structures with respect to their vibroacoustical properties

Keywords: *auxetic; hexagonal; 2-D gradient; sandwich panel; vibroacoustic; genetic algorithm optimization*

1. INTRODUCTION

Honeycombs strcutures or hexagonal periodic cells have gained considerable attention in recent years as they possess outstanding out of plane mechanical properties [1]. Recently, a new types of honeycombs have been introduced which possess negative Poison's ratio [2]. In contrast to conventional materials, these materials called “auxetic”, expand in all direction when subjected to uniaxial loading [3]. One of the iconic examples of auxetic materials is the hexagonal center-symmetric re-entrant configuration. They show an increase in the bending stiffness which can be useful in vibration[4]. Representative unit cells have been widely used to model mechanical properties of composite materials and sandwich structure [5]–[7]. Auxetic structure can also be considered as periodic repetition of the unit cell and their homogenized mechanical properties of the can be defined in terms of their geometrical parameters.

As the application of the auxetic materials are widespread they have been used in various areas of research. Auxetic cellular structures have been used to prototype morphing wings [8]. In another work, the wave propagation in sandwich panels having periodic auxetic core was investigated [9]. The above cited micro-structure configurations tessellate periodically in the plane. Therefore, the cellular structure is made of cells having same geometry in any part of the structure. However, this cellular structure can be produced with a gradient configuration in which the structure is made of a continuous distribution of unit cells with compatible geometry but a single variable parameter like the internal cell angle [10]. Several researches have investigated vibrational and acoustic behavior of sandwich structures with normal or gradient cellular core [4]. More recently, Ranjbar et al. [10], studied the effect of geometrical parameters of a 1-dimensionally gradient auxetic core on radiated sound power level of sandwich panel structures.

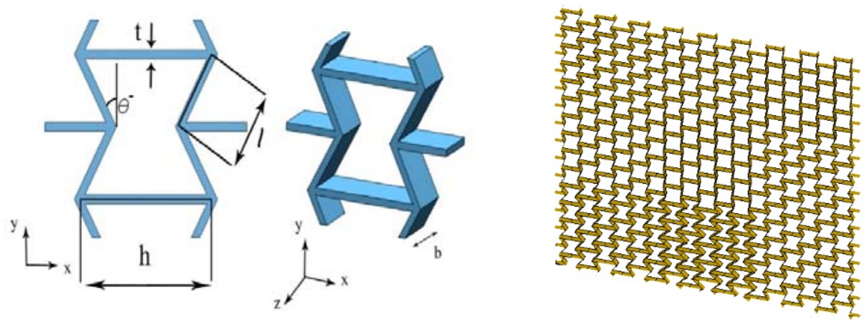
In this study the effect of geometrical parameters for a 2-dimensionally gradient auxetic honeycomb core on the radiated sound power level will be investigated. Furthermore, genetic algorithm (GA) optimization will be applied to minimize the radiated sound power level from the sandwich structure. To do so, an interactive link between MATLAB and ANSYS software has been established.

2. HOMOGENIZED MECHANICAL PROPERTIES OF AUXETIC CORE

An analytical model has been used to calculate the mechanical properties of auxetic hexagonal honeycombs. This models which have been previously used by Gibson and Ashby [11] and Lira et al [12] defines mechanical properties of hexagonal honeycombs based on three non-dimensional parameters $\alpha = \frac{h}{L},\ \beta = \frac{t}{L},\ \gamma = \frac{b}{L}$ and the angle θ. Figure 1.a., shows a representative unit cell of auxetic hexagonal honeycomb structure and its geometrical parameter. An orthotropic equivalent material have been used to model the mechanical properties of the honeycomb core plate. The compliance matrix [S] for an orthotropic material is defined, in which the engineering constants E_x, E_y, E_z, G_{xy} and G_{xz} can be found from Gibson and Ashby[11] or Lira et al [12]. Figure 1.b shows a 2-D gradient hexagonal cellular configuration. To generate the gradient core configuration, the unit cells with different internal angle θ have been assembled next to each other. The geometrical parameters of the original auxetic hexagonal unit cell have been shown in Table 1.

Table1. Geometrical parameters of auxetic hexagonal sandwich plate

h (mm)	l (mm)	t (mm)	b (mm)	θ (degree)
33.55	17.02	1	20	-30



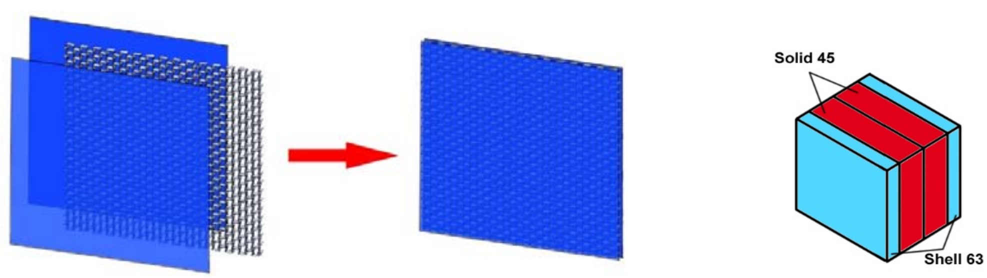
(a) (b)

Figure 1. a. A representative unit cell (RUC) of auxetic hexagonal honeycomb, b. 2-Dimensional hexagonal gradient core

3. FE MODELLING OF AUXETIC HEXAGONAL SANDWICH PANEL

The FE modeling for the auxetic hexagonal honeycomb sandwich panel with constant angle cell distribution has been performed using the ANSYS Rel. 14.0 commercial FE analysis package. The geometrical parameters of the auxetic hexagonal honeycomb core are listed in Table 1. The core is covered with two 960×996×2 mm skins plates. Both core and skins are made of ABS plastic with elastic properties listed in [12].

The homogenization of the sandwich panel is performed using shell elements to represent the skins, while the homogenized core of the auxetic hexagonal cells is represented by two solid element per gauge thickness. The mechanical properties of the skin are same as the mechanical properties of the ABS plastic and the mechanical properties of the homogenized core can be defined using the compliance matrix [S]. Figure 2.a. shows a full scale model of sandwich panel and Figure 2.b. demonstrates the homogenized single unit cell having a core and two skins. This homogenized unit cell is then reproduced along both x and y directions to make the sandwich panels with overall dimensions mentioned above.



(a) (b)

Figure 2, a. Full scale demonstration of the sandwich panel with auxetic hexagonal core, b. Finite element model of a homogenized auxetic sandwich panel unit cell.

To compare the behavior of full scale FE model with detailed geometry of the core detailed model and the homogenized model, a modal analysis have been performed. Simply supported boundary condition (SSBC) is considered for the modal analysis. The natural frequencies of both models have been shown in Table 2. The first six mode shapes in both full scale detailed and homogenized model are alike. Moreover, the natural frequencies find by homogenized model are in good agreement with the ones calculated by the full scale detailed model.

To do the harmonic analysis a pressure loading has been applied on the sandwich panel in z direction with frequency range of 0-200 Hz. Figure 3 shows the applied pressure and the distribution of nine different homogenized auxetic hexagonal unit cells with different angles($\theta_1, \theta_2, \theta_3, \theta_4, \theta_5, \theta_6, \theta_7, \theta_8, \theta_9$). The main objective is to minimize the radiated noise from the panel. To do so, the geometry of model in each region will be modified to reduce the radiated sound power level.

Table 2. Modal Analysis result of the auxetic honeycomb sandwich panel on the frequency range of 0 to 200 Hz for the homogenized and full scale FE models

	Model type	1	2	3	4	5	6
Frequency (Hz)	Homogenized FE	37.75	94.73	98.35	143.13	187.90	193.85
	Full Scale FE	34.80	83.84	87.31	133.17	160.88	170.90

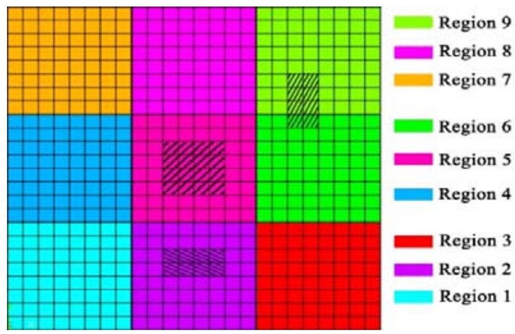


Figure 3. Applied pressure on the homogenized FE model

4. VIBRO-ACOUSTIC OPTIMIZATION OF THE SANDWICH PANEL

The minimization process of radiated sound power level over the frequency range of 0 to 200 Hz is discussed in this section. The objective function for the optimization process is the root mean square level of radiated sound power level (RMSL) of sandwich panel. The design variables of this optimization problem are the angles of unit cells in the different regions. The methodology to calculate RMSL can be find in [10].

The original design, is a non-gradient sandwich panel with core geometries given in Table1 and skin thickness of 2mm. At first, the effect of change in the core thickness on the RMSL of the sandwich panel with a constant cell angle of -30^{o} is evaluated. Figure 4 shows the change of RMSL with variation of thickness. The increase in the thickness makes a reduction in RMSL. The optimum thickness of 30 mm is identified.

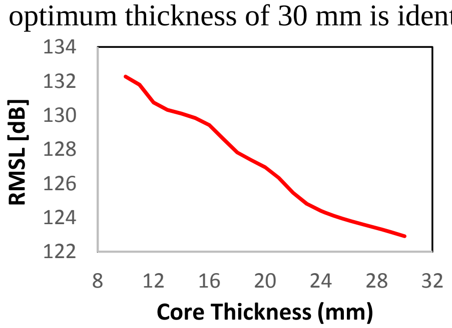


Figure 4. Variation of RMSL with respect to the core thickness

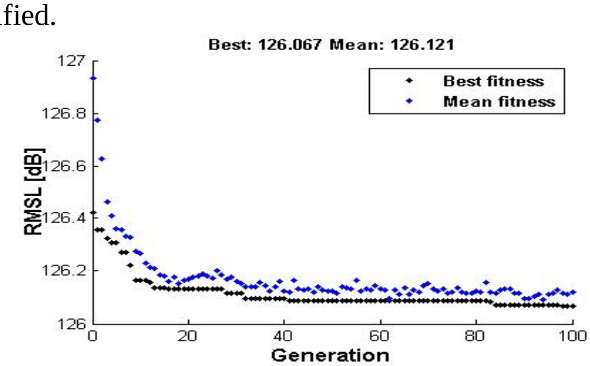


Figure 5.Variation of RMSL versus generation as a result of GA optimization

Next, nine different regions with nine different angles have been considered to compose the gradient sandwich plate. A fixed original value of 20 mm is considered for the core thickness. To do the optimization process Genetic algorithm optimization toolbox in MATLAB has been used. In order to have a better exploration of the feasible region of this optimization problem which has nine design variables and population size of 50, the first generation of genetic algorithm, has been created by Latin hypercube sampling (LHS) method. The objective function of this problem, the root mean square of radiated sound power level, then is minimized over 100 generations. Figure 5 shows the variation of best and mean of the objective function in each generation. After about 60 generation the objective function converges. The each function evaluation in this problem takes about 4.5 seconds and the total computation time was about 6 hrs and 15 minutes.Table 3 shows detailed geometry, optimization variables and total mass of the sandwich panel for the original design and after genetic algorithm optimization design with original thickness of 20 mm. As it is clear in Table 3 the genetic algorithm reduced root mean square of sound power level which is objective function of this problem to 126.07 dB and the mass of sandwich structure after GA optimization has increased about 3.2%.

Table 3. The optimization variables for the auxetic hexagonal sandwich panel

Design Set	θ_1	θ_2	θ_3	θ_4	θ_5	θ_6	θ_7	θ_8	θ_9	RMSL (dB)	Mass
Original Design	-30	-30	-30	-30	-30	-30	-30	-30	-30	126.95	5.635
GA Design	-48.84	-48.61	-45.83	-33.93	-49.32	-49.66	-10.36	-10.59	-10.78	126.07	5.815

Figure 6 illustrates the radiated sound power level over the frequency range of 0-200 Hz, for the original design and GA optimized design sets. The figure clears that the first natural frequency for the optimum designs is not substantially changed, while for the 2nd and 5th natural frequency shifted to a lower value and for the 4th and 6th natural frequency in moved to a higher value. The 6th natural frequency for the optimized design shifted to a 206.51 Hz. Figure 7 shows, the radiated sound power level over the frequency range of 0-200 Hz for the original design, design with original angles and optimum thickness and the design with optimum angles and optimum thickness. The figure depicts that for the optimum thickness designs, the first natural frequency is increased by 38 % for the optimum thickness and angles. Besides, all other five natural frequencies are shifted to a higher value. For the optimum thickness and angles design just the first four natural frequency remained in the range 0-200 Hz and the 5th and 6th natural frequencies are shifted to 248.42 Hz and 273.86 Hz respectively.

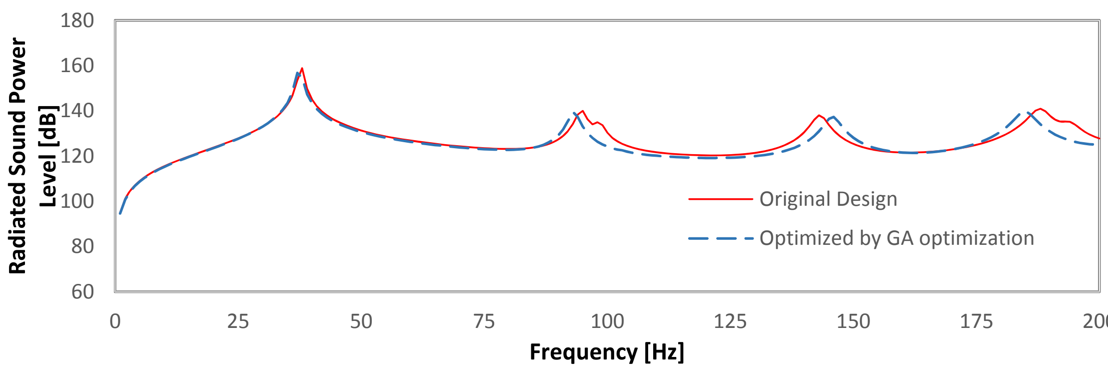


Figure 6. Effect of GA optimization methods on sound power level reduction

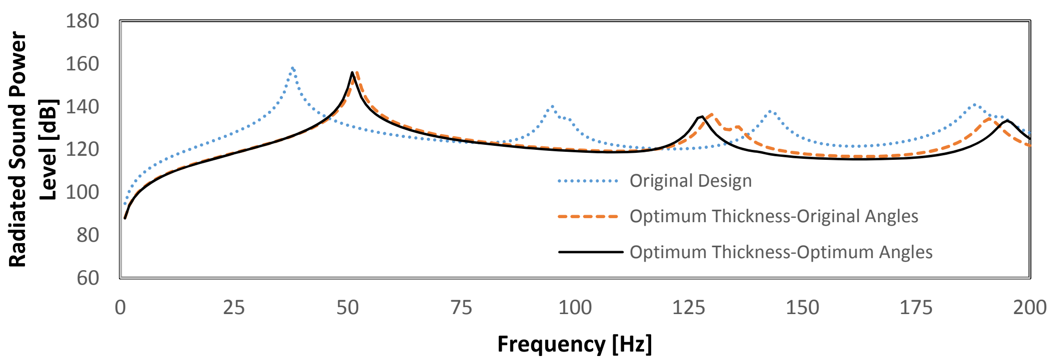


Figure 7. Sound power level reduction for the optimum thickness designs

5. CONCLUSION

This research has investigated the vibro-acoustic behavior of sandwich panels with 2-dimensionally gradient auxetic hexagonal honeycomb core. First, to determine the mechanical properties, the natural frequencies and the radiated sound power level of the hexagonal honeycomb sandwich panels made by the 2-D gradient auxetic cores, a homogenized finite element modeling approach has been implemented. The geometrical parameters of the core have effect on the radiated sound power level. The radiated noise level of the structure is significantly dependent to the location of the excitations. Therefore, it is recommended to use the gradient core geometries to cover the excitations areas.

For the optimization process an interactive link between MATLAB and ANSYS software was based. Genetic algorithm method has been used to optimize the root mean square of sound power level. The root mean square of radiated sound power level is reduced by about 1 dB. This reductions in RMSL is accompanied by an increase in total mass of the panel by 3.2 %. This designs behaves significantly better than the original design both in low and high frequency ranges. This feature could be considered for structures applications, in which weight reductions or control are paramount.

REFERENCES

[1] A. Lorato *et al.*, “The transverse elastic properties of chiral honeycombs,” *Compos. Sci. Technol.*, vol. 70, no. 7, pp. 1057–1063, 2010.

[2] J. Huang, X. Gong, Q. Zhang, F. Scarpa, Y. Liu, and J. Leng, “In-plane mechanics of a novel zero Poisson’s ratio honeycomb core,” *Compos. Part B Eng.*, vol. 89, pp. 67–76, 2016.

[3] R. Lakes, “Advances in negative Poisson’s ratio materials,” *Adv. Mater.*, vol. 5, no. 4, pp. 293–296, Apr. 1993.

[4] F. SCARPA and G. TOMLINSON, “Theoretical Characteristics of the Vibration of Sandwich Plates With in-Plane Negative Poisson’S Ratio Values,” *J. Sound Vib.*, vol. 230, no. 1, pp. 45–67, 2000.

[5] M. M. Shokrieh and M. S. Mazloomi, “An analytical method for calculating stiffness of two-dimensional tri-axial braided composites,” *Compos. Struct.*, vol. 92, no. 12, pp. 2901–2905, 2010.

[6] M. M. Shokrieh and M. S. Mazloomi, “A new analytical model for calculation of stiffness of three-dimensional four-directional braided composites,” *Compos. Struct.*, vol. 94, no. 3, pp. 1005–1015, 2012.

[7] Y. J. Chen, F. Scarpa, Y. J. Liu, and J. S. Leng, “Elasticity of anti-tetrachiral anisotropic lattices,” *Int. J. Solids Struct.*, vol. 50, no. 6, pp. 996–1004, 2013.

[8] P. Bettini, A. Airoldi, G. Sala, L. Di Landro, M. Ruzzene, and A. Spadoni, “Composite chiral structures for morphing airfoils: Numerical analyses and development of a manufacturing process,” *Compos. Part B Eng.*, vol. 41, no. 2, pp. 133–147, 2010.

[9] M. Ruzzene, L. Mazzarella, P. Tsopelas, and F. Scarpa, “Wave Propagation in Sandwich Plates with Periodic Auxetic Core,” *J. Intell. Mater. Syst. Struct.*, vol. 13, no. 9, pp. 587–597, 2002.

[10] M. Ranjbar, L. Boldrin, F. Scarpa, S. Neild, and S. Patsias, “Vibroacoustic optimization of anti-tetrachiral and auxetic hexagonal sandwich panels with gradient geometry,” *Smart Mater. Struct.*, vol. 25, no. 5, p. 54012, 2016.

[11] L. J. Gibson and M. F. Ashby, *Cellular solids. Structure and properties*. 1997.

[12] C. Lira, F. Scarpa, and R. Rajasekaran, “A Gradient Cellular Core for Aeroengine Fan Blades Based on Auxetic Configurations,” *J. Intell. Mater. Syst. Struct.*, vol. 22, no. 9, pp. 907–917, 2011.