

Analytical Study on the Rate of Sound Transmission Loss in Single Row Honeycomb Sandwich Panel Using a Numerical Method

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Abstract: Honeycomb structures have recently, replaced with conventional homogeneous materials. Given the fact that sandwich panels containing a honeycomb core are able to adjust geometric parameters, including internal angles, they are suitable for acoustic control applications. The main objective of this study was to obtain a transmission loss curve in a specific honeycomb frequency range along with same overall dimensions and weight. In this study, a finite element model (FEM) in ABAQUS software was used to simulate honeycomb panels, evaluate resonant frequencies, and for acoustic analysis. This model was used to obtain acoustic pressure and then to calculate the sound transmission loss (STL) in MATLAB software. Vibration and acoustic analysis of panels were performed in the frequency range of 1 to 1000 Hz. The models analyzed in this design includes 14-single row-honeycomb designs with angles of -45° , -30° , -15° , 0° , $+15^\circ$, $+30^\circ$, $+45^\circ$. The results showed that a-single row and -45° cell angle honeycomb panel in the frequency range of 1 to 1000 Hz had the highest STL as well as the highest number of frequency modes (90 mods). Furthermore, the panel had the highest STL regarding the area under the STL curve (dB·Hz). The panels containing more frequency mods, have a higher transmission loss. Moreover, the sound transmission loss is more sensitive to the cell angle variable (θ). In other studies, the STL was more sensitive to the number of honeycomb cells in the horizontal and vertical directions, as well as the angle of cells.

Keywords: Finite element model; honeycomb; sound transmission loss

1 Introduction

Noise is an unwanted sound which introduces health in addition to safety risks to workers occupied in various places of work [1]. Noise is one of the most common causes regarding of the loss of hearing in the industrial sectors [2,3]. These types of non-auditory damages hurt the autonomic nervous system, leading to



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increased skin temperature in addition to pulse rate, higher blood pressure, headache, fatigue, constriction of blood vessels, abnormal secretion associated with hormones, and muscle tension [4–6].

Increasing the harmful effect of noise in the industry, as well as exposing a large percentage of workers to the maximum noise standards have made noise control a very important issue. One of the noise control methods is the use of sound barriers. Sandwich panels have a wide range of applications due to their increasing strength-to-weight ratio [7,8]. These panels have been increasingly used in the body of aircraft and ships, cabins of trains, wall of conques, salons, and etc [9].

The Sound Transmission Loss (STL) is one of the important parameters in designing these structures [10]. Cellular materials have been increasingly used in designing and researching due to their macro material properties in recent years [7–10]. These materials are widely used in lightweight, energy absorbing, and packaging structures [11]. Discovering elastic and lightweight features of hexagonal beehive honeycomb has inspired researchers and designers to use the honeycomb structure for structural applications. In 1915, Hugo Junkers examined the sheet structure- honeycomb core for the first time and patented the first honeycomb core [12]. One of the most common orthotropic structures for the core used in these studies is honeycomb structures.

The overall effects of honeycomb cores have been recently studied [9,13,14]. A number of researchers have used honeycomb structures for the core panel in their research study [7,15,16]. Ruzzene studied the effects of the use of honeycomb structures within the sandwich panels on the characteristics of the STL [9]. Griese, examined the effect of the change in angle between the honeycomb structure cell walls on the sandwich panel STL [17]. Griese changed the angle from -45 to $+45$ and observed that the negative angled honeycomb plates had better STL characteristics compared to the positive angled honeycomb for the same mass of the sandwich panel [17]. Galgalikar studied the optimization of the honeycomb sandwich panel to maximize the STL and found that the honeycomb structure with a negative Poisson ratio had better STL characteristics [18]. Shanbag et al. considered the STL by sandwich panels made of viscoelastic cores [19]. Thamburaj et al. [20], Denli et al. [21] and Franco et al. [22] performed an optimization study to achieve maximum the STL of the sandwich panel.

The main advantages of the honeycomb structure include their light weight and increase in the STL [23]. The main parameters affecting the transmission loss include the overall thickness of the panel, the thickness of the panel layers, the type of layers (density, Poisson's ratio, energy loss factor and yang coefficient), core shape, angle, and etc. Changing each of these parameters increases or decreases the STL. Finally, by optimizing each of the variables affecting the STL, including the overall thickness of the panel, the thickness of the layers, the type of layers and the core shape, the optimized panel characteristics will be achieved. This research project aimed to present a new design to increase the STL of these structures to create sound barriers in walls of acoustic rooms, conques and industrial salons. In this design, it was tried to have a sandwich panel with high STL by changing the cell angle. The main purpose of this study was to design and evaluate the rate of STL in a single row honeycomb sandwich panel using a numerical method.

2 Methods

2.1 Internal Geometry and Honeycomb Panel Properties

In this study, the in-plane configuration was used. The constant effective density was considered for all models. By keeping the effective density, the size of sandwich panels and, consequently, the overall weight for all models will be the same; therefore, it causes the weight variable to be ineffective in the STL of various sandwich panel models. The thickness of honeycomb unit cells will vary to maintain the effective density of different configurations. The SolidWorks Design software was used to obtain the thickness of new designs. In this regard, at first, the design of new panels was provided in the software and then their thickness was obtained to maintain the effective density of the designs by the software. Eqs. (1) to (6) show the relationship between the sides of these honeycomb cells [17,24].

$$l = \frac{L_x}{2 \cos \theta} \quad (1)$$

$$h = \frac{L_y}{2} - \frac{L_x \sin \theta}{2 \cos \theta} \quad (2)$$

$$L = L_x N_h \quad (3)$$

$$H = L_y N_v \quad (4)$$

L : The whole length of the sandwich panel.

H : The whole thickness of the sandwich panel core

N_h : Number of Honeycomb Cells in Horizontal Alignment

N_v : Number of Honeycomb Cells Vertically

The effective density of each sandwich panel was calculated using the following formula:

$$\rho^* = \rho \frac{\left(\frac{t}{l}\right) \left(\frac{h}{l} + 2\right)}{2 \cos \theta \left(\frac{h}{l} + \sin \theta\right)} \quad (5)$$

ρ^* : The effective density of the honeycomb core (kg/m^3)

ρ : The density of the material from which the honeycomb is made (kg/m^3)

The following formula calculates the thickness of honeycomb unit cells:

$$t = \frac{\rho^* 2l \cos \theta \left(\frac{h}{l} + \sin \theta\right)}{\rho \left(\frac{h}{l} + 2\right)} \quad (6)$$

2.1.1 Study Fixed Parameters

The total thickness of the sandwich panel core (H) was considered 8.66 cm, the total length of the sandwich panel (L) 2 meters, the depth (width) of the sandwich panel core (D) 1 meter, the honeycomb core weight (M) 23.382 kg/m^2 , the total weight of the sandwich panel (M_T) 36.882 kg/m^2 , the thickness of the top plate (tf) 2.5 mm, the weight of the top plate 13.5 kg/m^2 , the density (ρ) and the honeycomb core effective density (ρ^*) were respectively 2700 kg/m^3 , 270 kg/m^3 . Moreover, the features of aluminum material (honeycomb section and top plate) have a mass density of 2777.8 kg/m^3 , a young's modulus of 71.9 GPa, and a Poisson coefficient of 0.3.

In this study, core weight using effective density and total core volume, was calculated by Formula 7 [17,24].

$$W = \rho^* L H D \quad (7)$$

The total weight of the sandwich panel (WT) by adding the weight of the core and sandwich panel front panel was obtained using Formula 8 [17,24].

$$W_T = W + 2\rho L t_f D \quad (8)$$

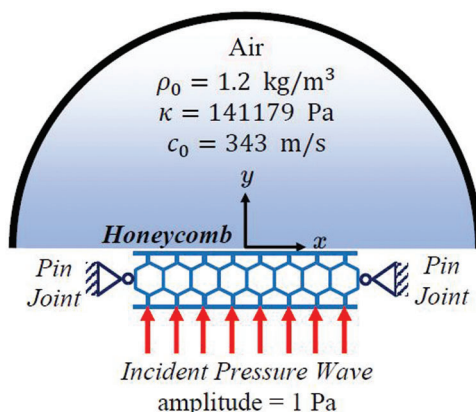
Table 1: Internal geometry and important properties of same weight-single row honeycomb panels

Cell Angle	l (mm)	h (mm)	t (mm)
45°	35.36	18.30	2.43
30°	28.87	28.87	2.50
15°	25.88	36.60	2.45
0°	25.00	43.30	2.32
−15°	25.88	50.00	2.13
−30°	28.87	57.73	1.88
−45°	35.36	68.30	1.56

The internal geometry and the important properties of honeycomb panels with the same weight analyzed in this study are shown in [Tab. 1](#) (Unit cells are 1×40).

2.1.2 Creating Acoustic Ambient Air

For the main analysis of the loss of sound transmission, the air must be modeled in Abaqus software [17,24]. Therefore, a geometry for air must be considered. Suitable candidates for this geometry are rectangles twice the length of the panel or semicircular to the radius equal to the length of the panel. Since pressure is used to model the sound intensity, and only the area very close to the sandwich panel boundary is important, a semicircle can model pressure drop well. If the rectangular geometry is used, the number of elements and, as a result, the degrees of freedom of the elements would increase sharply, resulting in an exponential increase in time. To simulate the air section by Abacus software, a semicircle was drawn according to [Fig. 1](#) and this semicircular space was used as the upper air of the sandwich panel. Modeling this section is necessary to determine the structural noise loss and to simulate any damping effects that air may have on the structure [17].

**Figure 1:** Finite element model with boundary conditions and defined constraints

This section is plotted as a 2D planar deformable shell. This semicircle was attributed to the acoustic properties of air as shown in [Tab. 2](#). The effects of different air properties on the variables of this study were beyond the scope of this study.

Table 2: Air properties used in modeling

Speed of sound (m/s)	Bulk modulus (Pa)	Density (kg/m ³)
343	141179	1.2

2.1.3 Section

A two-dimensional geometry was designed to model the sandwich panel in the Abaqus software environment, when a two-dimensional model is designed in the Abaqus software, it cannot be defined visually for that thickness. Then all the properties including thickness, Young's modulus (elasticity), Poisson's coefficient, etc. must be given in the Properties section. For this reason, it was first designed in the Solidworks sandwich panel software, and given the constant volume and weight of the panel, the apparent properties (side thickness) were calculated. Eventually these numbers were provided as input data to Abaqus software. This section defines the depth (a) and thickness (b) of the sandwich panel as follows (Tab. 3).

Table 3: Profile of the section for the core and plates

Beam section	a (m)	b (mm)
Honeycomb	1	t
Plate	1	2.5

According to Tab. 3, t is the same thickness calculated by the SolidWork software providing input data for the Abacus software.

2.1.4 Meshing

For both parts of the sandwich panel, honeycomb core and plates, the B-22 element is used for beam elements. The B-22 consists of three nodes. This type of element is suitable for pressure drop testing [17,24–28]. The air environment was also modeled using the AC2D3 element, which is two-dimensional and consists of three nodes and is often used to model acoustic environments. The mesh size of the mesh at the sandwich panel contact area is 0.012 m and increases to 0.08 m from the center to the semicircular arch. The reason for the increase in mesh size toward the semicircular arch is to obtain the main result of this study (sound transmission loss) and only a portion of the air directly in contact with the panel is needed.

2.1.5 Boundary Conditions and Loads

To create boundary conditions, both ends of the sandwich panel and the pin boundary conditions were selected to simulate the bonding to a rigid insulator. With this boundary condition, in fact, the displacement of all nodes along the two edges of the sandwich panel is restricted to zero in both X and Y directions. Rotation on off-screen axis is not restricted. This boundary condition is used for both frequency and acoustic analysis. If a jumper head is used instead of a pin connection, it would have had an impact on acoustic analysis; since it prevented the beam from rotating. For acoustic panel analysis, a uniform pressure (1 Pascal in magnitude) is applied to the bottom of the panel. In this study, the impact angle was zero pressure ($\alpha = 0^\circ$).

2.2 Extraction of Panel Natural Frequencies

In the first phase of the analysis, in order to understand the vibrational properties of sandwich honeycomb panels and focus on the STL study, the natural frequency was extracted for each panel model. Initially, the panel was analyzed by the Frequency process to obtain its natural frequencies at ambient temperature. In order to understand the vibrational properties of honeycomb sandwich panels and

to focus on the STL study, in the first analysis phase, the natural frequency extraction was performed for each panel models. This analysis was performed using the Linear Perturbation-Frequency Step in ABAQUS/CAE 6.14. ABAQUS/CAE is a complete ABAQUS environment that provides a simple, consistent interface for creating, submitting, monitoring and evaluating simulations of results [29]. ABAQUS/CAE is divided into modules, where each module defines a logical aspect of the modeling process, such as geometry definition, material properties definition, and mesh generation [29]. Extraction of natural frequencies in Abacus software was done through the Lanczos eigensolver. This phase of the analysis was carried out at the frequency range of 1 to 1000 Hz of the stiffness and resonance regions in the STL curve.

2.3 Acoustic Panel Analysis Method

In this analysis, the panel was analyzed according to the natural frequency conditions obtained in the frequency analysis, and the STL was achieved. The panel frequency range was 1 to 1000 in this phase.

2.4 Drawing the STL Curve

To draw the STL curve, the output of POR (acoustic pressure) software was required.

This output was taken for 168 nodes that were in contact with the top of the sandwich panel. It was taken for various frequencies. For STL calculation at each frequency, formulas number 9 were used:

$$STL = 10\log_{10}\left(\frac{P_i^2}{P_t^2}\right) \quad (9)$$

P_i : The acoustic pressure hitting the panel (N/m^2)

P_t : The acoustic pressure passing through the panel (N/m^2)

The standard method to calculate the used STL was ASTM E-90-09. The STL was achieved using a uniform sound frequency of 100 to 10,000 Hz (1 octave band). Coding was done by using the STL formulas in the MATLAB R2014a software and the STL curves were obtained from the ABAQUS software POR output. To validate the results of this study, they were compared with the results of other studies.

3 Results

The natural frequencies of a single row sandwich honeycomb panel are shown in Tab. 4. In these tables, the natural frequencies up to 25 mode are presented, and for panels with more modes, the number and frequency of the last mode are mentioned (in the range of 1 to 1000 Hz).

The STL curve of all the studied panels are presented in Figs. 2 and 3.

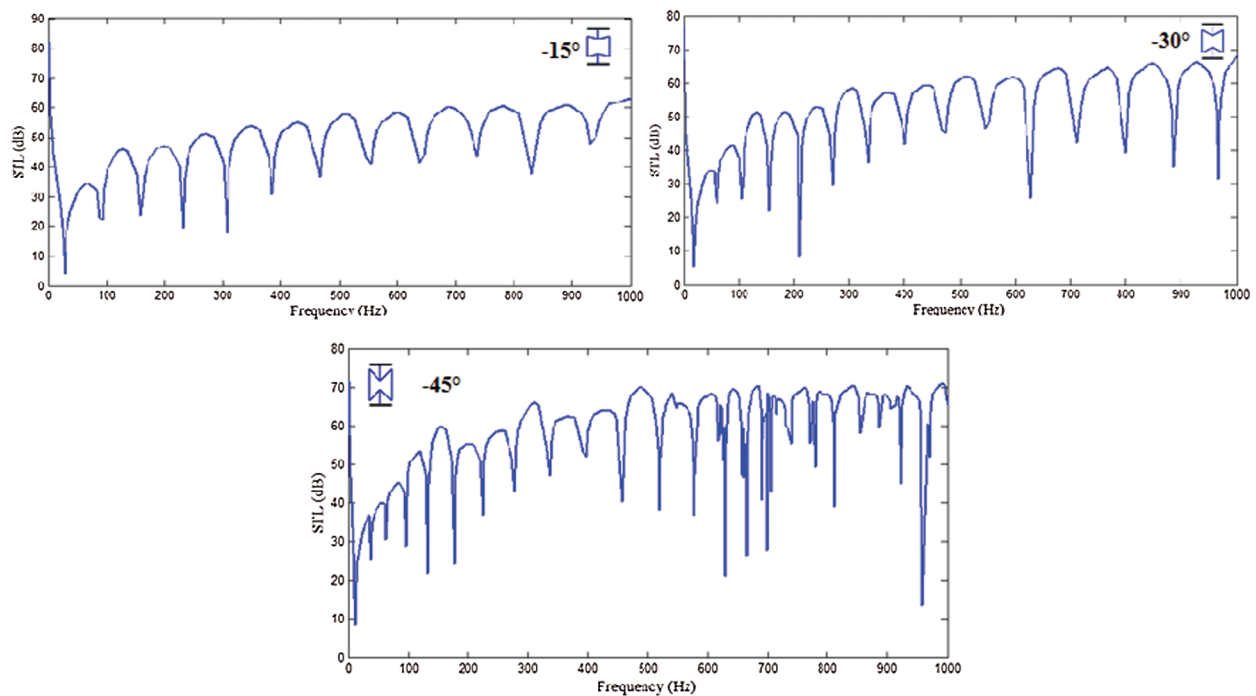
The -45° curve shows the STL curve of a single row sandwich panel with angle of -45° in the frequency range of 1 to 1000 Hz. This panel has the most vibrational mode in this frequency range (90 vibrational modes) compared to other panels. Furthermore, the panel has the highest STL in terms of the area under the STL curve ($dB \cdot Hz$).

As shown in the $+30^\circ$ curve, the odd frequency modes correspond to minimum relative points of the curve, and the even frequency modes correspond to maximum relative points of the curve. This figure shows that the STL in the odd frequency modes is very low due to the fact that the sound pressure hitting the panel is close to the sound pressure passing through the panel. The area below the STL curve ($dB \cdot Hz$) of the honeycomb panel with a configuration of $+30^\circ$ in the stiffness region has the highest amount compared to other designs, due to the high frequency of the first mode in this panel.

The comparison between a single row $+15^\circ$ and -15° honeycomb panel is shown in Fig. 4. As Fig. 3 reveals, in the same number of rows, positive and negative honeycomb panels have a significant

Table 4: Natural frequencies of a single row sandwich honeycomb panel with the same weight (Unit: Hz)

	45°	30°	15°	0°	−15°	−30°	−45°
1	84.29	62.79	49.89	40.18	29.23	19.83	11.77
2	184.59	129.75	101.93	82.26	59.39	40.04	23.84
3	301.13	205.87	160.33	128.76	92.41	61.59	36.62
4	422.69	282.97	219.69	176.14	126.56	84.02	50.21
5	545.69	360.81	280.31	224.10	161.92	107.61	64.81
6	668.22	438.98	341.70	272.29	198.14	132.35	80.56
7	767.77	517.78	403.86	320.84	235.16	158.24	97.54
8	789.97	597.19	466.63	369.77	272.87	185.26	115.81
9	910.79	677.32	529.99	419.22	311.31	213.40	135.41
10	-	758.16	593.92	469.24	350.50	242.64	156.35

**Figure 2:** The STL curve of a single row sandwich panel with angles of -15° , -30° and -45°

difference in the STL curve and the height of the STL curve is higher in panels with a negative angle; therefore, they have more area under the curve.

4 Discussion and Conclusion

In the single row panels, the maximum and minimum points of the STL curve was slightly moved to higher frequencies. The single row -45° honeycomb panels with the highest STL had the highest number of frequency modes. The single row -45° panel had 90 frequency modes. A study conducted in 2005 entitled “The Evaluation of Sandwich Models to Predict the single row Sound Transmission Loss” by

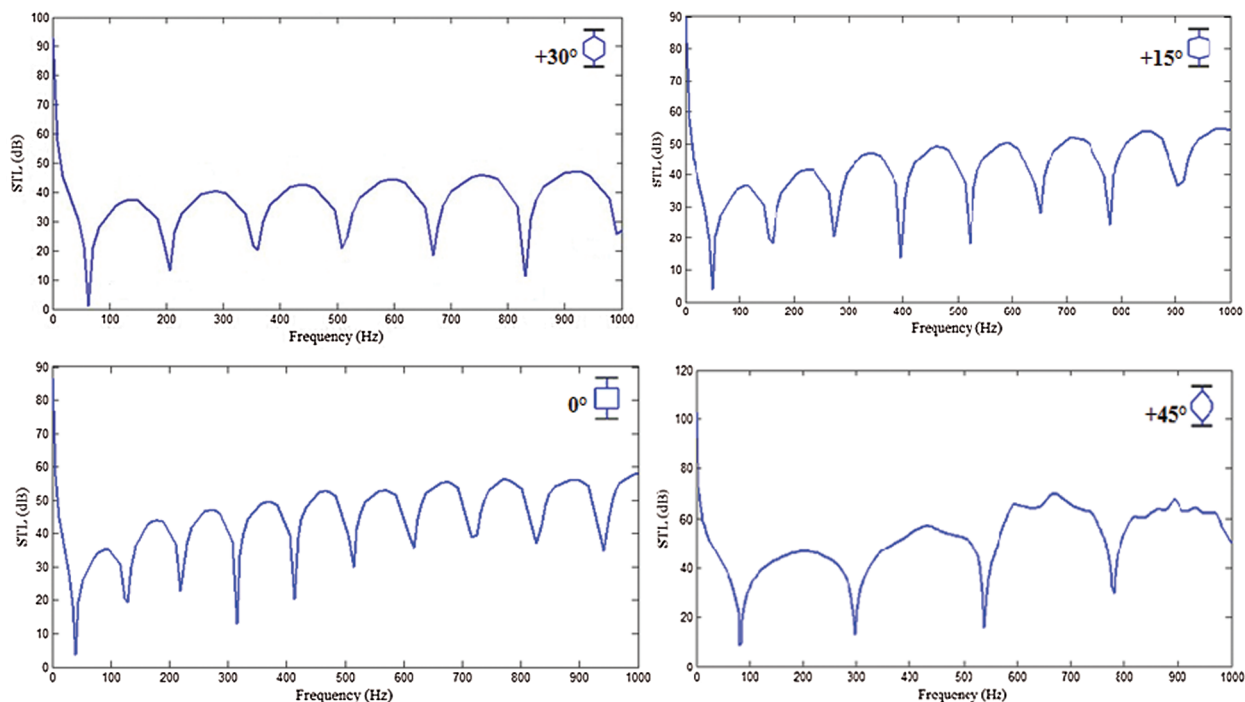


Figure 3: The STL curve of a single row sandwich panel with angles of $+15^\circ$, $+30^\circ$, $+45^\circ$ and 0°

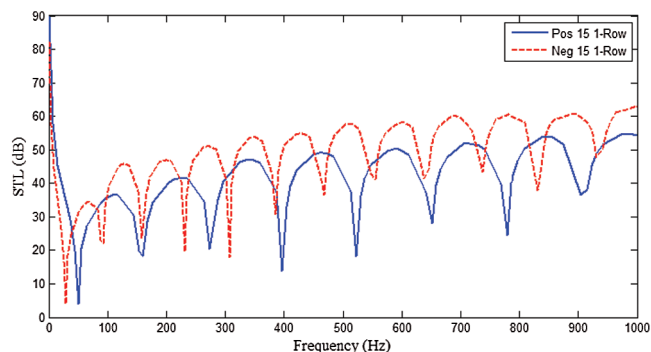


Figure 4: Comparison of the STL curve of a single row honeycomb panel with angles of $+15^\circ$ and -15°

Wang et al. showed that if the honeycomb cells are parallel to the panel screen, instead of being perpendicular, the transmission loss of the panel will increase [30].

A study carried out in 2007 entitled “Optimizing the Acoustic Properties of Cellular-Cell Sandwich Structures to achieve Minimum Sound Frequency” by H. Denli and J. Q. Sun indicated that sandwich structures create a significant transmission loss in the narrow and wide frequency band [21]. To validate the results of this study, they were compared with the results of other studies. The natural frequencies found by David Griesse, Rohan Galgalikar, Xin Wang, and Xiao Gong had a difference of less than 1 percent [17,18,23,31]. The comparison of the natural frequencies of the $+30^\circ$ single row honeycomb panel with other studies is shown in Tab. 5.

The base design STL (the $+30^\circ$ single row Honeycomb) was compared with the base design STL of Griesse [17] and Gong’s [31] research. The difference between the area under the STL curve was about 8%. Comparing honeycomb sandwich panel passage loss of a $+30^\circ$ row with other studies, the present

Table 5: Comparison of natural frequencies of the +30° single row honeycomb panel with other studies (Unit: Hz)

	Peresent work	Griese [17]	Galgalikar [18]	Wang [23]	Gong [31]
1	62.789	62.8	63.43	63.18	63.82
2	129.747	129.7	131.16	130.56	132.1
3	205.871	205.9	208.22	207.17	209.7
4	282.965	283.0	286.29	284.75	288.5
5	360.806	360.8	365.18	363.08	368.0
6	438.976	439.0	444.45	441.75	447.9
7	517.776	517.8	524.40	521.04	528.5
8	597.185	597.2	605.01	600.95	609.6
9	677.321	677.3	686.37	681.6	691.5
10	758.155	758.2	765.90	762.95	766.0

study is below the STL curve in the stiffness area of 2430.5 dB z Hz, in the resonance region of 34435 dB·Hz and in the whole region up to a frequency of 1000 Hz 36865 dB·Hz. David Griese, in his study [17], obtained 2649.4 dB·Hz in the stiffness region, 37642.9 dB·Hz in the resonance region, and 40292.3 dB·Hz in the whole region up to a frequency of 1000 Hz. Also, Gong [31] obtained the area under the STL curve in the stiffness region of 2659 dB·Hz, in the resonance region of 35765 dB·Hz and in the whole region up to the frequency of 1000 Hz 40304 dB·Hz.

The results showed that a-single row and -45°cell angle honeycomb panel in the frequency range of 1 to 1000 Hz had the highest STL as well as the highest number of frequency modes (90 mods). Furthermore, the panel had the highest STL in terms of the area under the STL curve (dB·Hz). The panels containing more frequency mods, have a higher transmission loss. Moreover, the sound transmission loss is more sensitive to the cell angle variable (θ). In other studies, the STL was more sensitive to the number of honeycomb cells in the horizontal and vertical directions and the angle of cells. Suggestions for this study are to investigate the sound transition loss in the multilayer sandwich panels and different shapes.

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