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STUDY DIFFERENT CORE TYPES OF SANDWICH PLATE ON THE DYNAMIC RESPONSE UNDER IMPACT LOADING

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Abstract

Sandwich structure plates are most widely used in the automotive, aerospace and naval structures. As it gives material with low density and relatively high normal compression and shear properties. In this paper, Finite element method was used with ANSYS APDL (16.0) to analyse the effect of type core in sandwich steel structure of the dynamic response and modal analysis under the action of impact loading. The chief purpose of this work is to get a high reduction in deformation and stress between upper and lower skins. Isolate deflections of sandwich plates are compared and optimized rely on the type of core. The construction of the sandwich composite model consisted of double layers of honeycomb core square type and three types of corrugated cores triangle type, trapezoidal type and Y shaped type, with five cells for all cores, while sheet skins that are connect the cores as a single structure are three layers upper skin, middle skin and lower skin. All of configurations for both core and skin are made from the same material (steel alloy 304) and the structure have a constant weight 13.75 kg with dimensions 500 mm × 500 mm × 106 mm. The properties of materials are isotropic and have an elastic modulus 198 GPa, density 7835 kg/m^3 and Poisson ratio is 0.3. The results observed that the square honeycomb sandwich plate gives maximum separation in stress and deflection with values 99.75 % and 98.51 % respectively.

Keywords: Corrugated cores, Honeycomb cores, Impact load, Sandwich structure, Transient analysis.

1. Introduction

The sandwich structure contains two or multi thin skin layers connected by a thick internal core. When the core and skins are combined together, they function as a one structural component that have all the benefits of each component. Essentially, the skins are stiff, thin and strong, compressive and tensile stresses are mainly carried by the skins, while the core incur transverse shear stresses. Also, the cores provide support for the skins so they are doing not deform [1, 2]. Figure 1 shows the example of sandwich structures.

Finite element method is the most using to analyse the sandwich structures by the researchers. Also, the researchers studied sandwich structure with two aspects; the first one is the free vibration and the second one is the forced vibration.

Havaldar et al. [3] studied the fundamental frequency of (Fibre-reinforced-plastic) core under two boundary conditions; *C-F-F-F* and *C-F-C-F* with different cell sizes and showed that the frequency was higher in small cell size and in *C-F-C-F* boundary condition.

The free vibrations of flat and corrugated sandwich plates with face sheets and a foam filling core had been studied by Kheirikhah et al. [4], the results stated that the natural frequencies of trapezoidal corrugated were higher than sinusoidal, also the effect of trapezoidal shape was more significant on improving the dynamic behaviour with small thickness ratio.

Li et al. [5] analysed by using the finite element and experimental investigations of metallic truss sandwich structures, the investigation showed that the Hourglass sandwich plates have higher natural frequencies (ω_n) than that of the pyramidal type with (free-free) edges and equal relative density of the truss cores.

The numerical analysis and experimental study were done of a composite sandwich structure by Meo et al. [6]. For describing the low velocity impact response of the panels is studied at five energy levels, ranging from 5 to 20 J. The results of this study indicate that significant internal damage occurs at relatively low impact-energy levels, which can significantly reduce the residual strength of the panel. A good agreement was noted between these two methods.

Naresh et al. [7] investigated dynamic properties of sandwich structures with impact load by Finite element, it concluded that the distortion was global and higher for skin while local for sandwich plate, furthermore the stress and distortion were lower for square type than hexagonal core.

Hou et al. [8] examined the effects of key shape, core thickness, unit cell size, and the materials on the crashing behaviours of two corrugated while Mankour et al. [9] studied same effects with honeycomb core. They reached that the shape of core cell has significant influences on the surface impact responses, but a relatively small effect on the responses of low velocity impact. Also found that triangular core has more impact of resistance than the trapezoidal. Lastly the increase of honeycomb core thickness lead to increase the displacements and the frequency response.

Four different models of corrugated sandwich structure (*U* shape, *V* shape, *Y* shape and *X* shape core) had been studied experimentally and FE numerically by Yan-Chang et al. [10]. Results show that the *V* shape core structure has good crashworthiness and absorbed of impact if compared with other shapes core.

However, its behaviour will enhancement when the core is corrugated as *N* shape. Also, using the finite element analysis, Fadhel [11] studied four types of honeycomb cores with a constant volume of sandwich structure; these are hexagonal, rectangular, triangular and circular. The analysis of the three point bending load was conducted and explained that the best shape of core was rectangular which give less deflection than other shapes.

Qin et al. [12] discussed with numerical methods and experimental study the behaviour of damage in metal honeycomb sandwich structures. It showed that the effect of skin thickness was very important for the energy absorption and increase high of core have small an effect, stiffness. Thickness of cell wall core has a notable result on the impact load and structural stiffness of such structures, whereas having a little effect on energy absorption. Mutalib et al. [13] studied by LUSAS program the static behaviour of double and four layers of sandwich plate. The results indicated a rise in shear strength and change in the mode of failure for all beams studied when the corrugation angle was increased. Then again, a little effect was noted for the concrete strength on the resistance of shear and the failure mode with corrugated webs. Zhang et al. [14] aimed to investigate with numerical calculations the low-velocity impact of fully clamped rectangular multilayer sandwich plates with metal foam cores. They showed that the impact location plays important roles in the dynamic response while the effects of the multilayer factor and shape of the striker on the dynamic response are small.

From the above literature, it found that a few researchers deal with double honeycomb and corrugated cores. Therefore, in this paper, it will be studying the optimum core configuration for the sandwich plate with double core and comparative between four types of cores, which gives stiffer structure and mine deflection reduction between upper and lower skins. Also, this paper studied both free and forced vibration where few researchers studied that. This analysis will be done numerically by using ANSYS APDL (16.0) program.

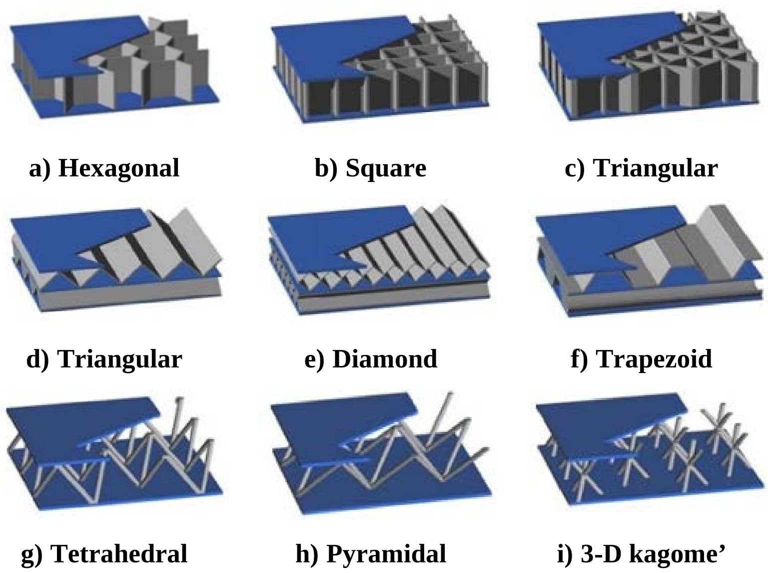


Fig. 1. Various sandwich structures geometries [15].

2. Numerical consideration

The finite-element-solver ANSYS APDL (16.0) configures the dynamic properties to calculate the vibration behaviour under the action of impulsive loading. Furthermore, FE method used to determine natural frequency and five mode shape of sandwich structures. The full interaction was assumed between the core and the skin. Where at first, determining of mode shape, natural frequency and dynamic response. The elements employed in the analysis of all steel sandwich plate models are SHELL (281), this type of element gives a suitable solution because it is containing (8 nodes) with (6 DOF): translations and rotations in three directions. Also, it is very proper for large non-linear strain, large rotation and linear applications. Shell thickness should be small when compared with other dimensions. Generally, the shell is made from solid material and assumed isotropic and elastic. The dynamic load applied to the structure will be impacted loading, the duration of the applied load is 0.01 s. Figure 2 shows location of the nodes, geometry, and the element coordinate for this element [16].

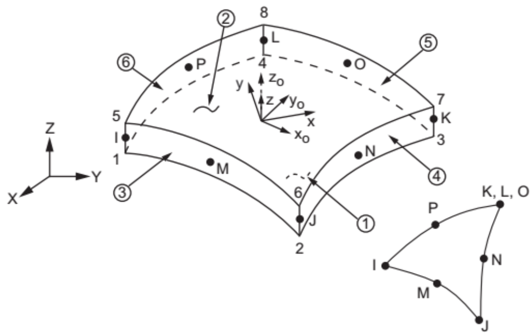


Fig. 2. SHELL (281) geometry [16].

2.1. Creating of models and properties of materials

Four models of sandwich plates with double core were created rely on the type of core. They are triangle corrugated, Y-shaped corrugated, trapezoidal corrugated and square honeycomb. All types have the same of 13.75 kg with dimensions 500 mm × 500 mm × 106 mm and assume that the properties are isotropic. The skins and core materials of sandwich are made from steel. The modulus of elasticity was taken 198 GPa, Poisson ratio is 0.3, and material density is 7835 kg/m^3. The dimensions of these models are illustrated in Table 1.

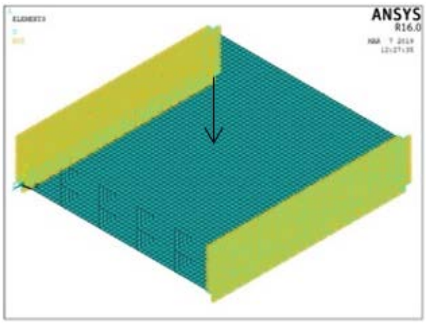
Table 1. Dimensions of models (in mm).

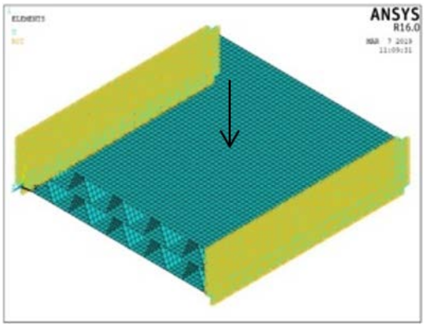
Model	Dimension of skin	Thickness of core	High of core	Cell length
Square	500 × 500 × 1.754	0.8774	50.37	100
Trapezoidal	500 × 500 × 1.439	0.7197	50.84	100
Triangle	500 × 500 × 1.521	0.7606	50.72	100
Y-shaped	500 × 500 × 1.537	0.7686	50.69	100

2.2. Boundary and initial conditions with loading

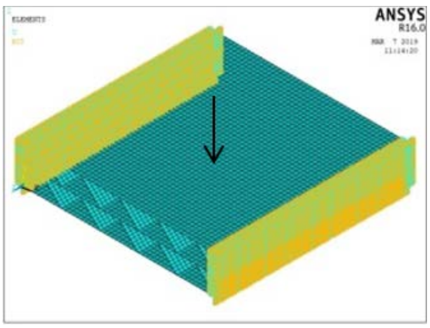
Initial conditions (where time is zero) assume to be zero, while boundary conditions of all models fixed from right and left edges (node movement of *x*, *y*, *z* are equal to be zero, and node rotation of the *x*, *y* and *z* are zero), and free for the other sides. The boundary conditions are shown in Fig. 3.

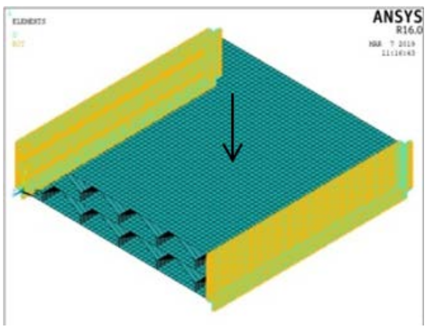
A central concentrated load was applied at a front face sheet of sandwich plate for all models in y direction with coordinates (*x*=0.25, *y*=0.1 and *z*=-0.25) with a value of 350 N during period of 0.01 s as shown in Fig. 4. This value of impact was chosen because it provides a clear and effective response and offers a good range of impact, causing failure in one type and safe in another.





(a) Square honeycomb sandwich plate. **(b) Trapezoidal corrugated sandwich plate.**





(c) Triangular corrugated sandwich plate. **(d) Y-shaped corrugated sandwich plate.**

Fig. 3. Load and boundary conditions of model.

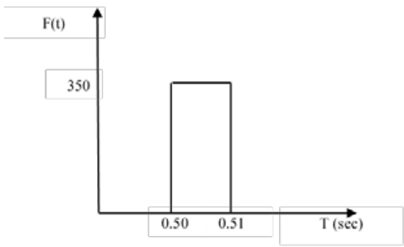


Fig. 4. Transient load.

2.2. Mesh generation

To find the mode shapes and natural frequencies for sandwich models, modal analysis has been resolved for the first five modes. It was ended that the results steadied at No. of elements 30720 and generate other models depending on this elements number. The mesh convergence is done for five modes of trapezoidal type as listed in Table 2.

Table 2. Convergence of mesh.

No. of Element	Natural freq. at mode 1	Natural freq. at mode 2	Natural freq. at mode 3	Natural freq. at mode 4	Natural freq. at mode 5
960	135	218.78	300.67	367.3	415.86
1920	130.61	228.84	302.85	371.375	418.72
3840	129.89	227.6	303.27	373.86	428.58
7680	128.22	226.88	303.55	378.4	437.62
15360	129	228.22	305.74	381.54	439.8
30720	129.1	228.2	305.6	381.49	439.6

3. Modal analysis (natural frequency and mode shape)

Analysis of free vibration is significant to show an impression about the fluctuation of the structure versus time. Modal analysis can be applied to any resonant structure has single-DOF system, where required the natural frequency, damping coefficient and the shape of the single mode for any structure such as aircraft, spacecraft or land vehicle. At a more advanced level, modal analysis can be used to find the natural frequencies, damping coefficients and vibration modes, of large, complex, structures. All aircraft, spacecraft, and many other structures, are designed mathematically, usually using the finite element method, followed by switching to normal modes [17].

Suppose that the structure, of which the modes are to be measured, is linear, and could be represented by a hypothetical set of equations of the form:

$$[M]\{\ddot{X}\} + [K]\{X\} = \{0\} \tag{1}$$

Supposes harmonic motion in the differential equation that govern the motion of the free vibration [17]:

$$\{X_i\} = \{\varphi_i\} \sin \omega_i t \qquad ; i = 1,2, \ldots . k \tag{2}$$

where, k is the number of (DOF) [16].

From the concept of homogeneous equation, if the determinant of the coefficient matrix is zero non-trivial solution exists. Therefore:

$$[K] - \omega_i^2 [M] = [0] \tag{3}$$

In the analysis, Lanczos method is adopted to estimate the eigenvalues of the system [17].

4. Effect of transient load

Transient analysis or time-history-analysis is a way that is used structure act to loads that is dependent of any general time to calculate the dynamic response. Equation of motion was solved under-damped structure when a structure submitted to a unity impact load at the initial state, Fig. 5 shows this impulse force excitation [18].

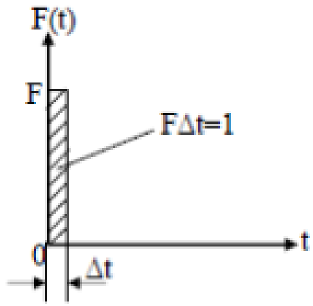


Fig. 5. Impulse force excitation [18].

$$[M].\{\ddot{X}\} + [C].\{\dot{X}\} + [K].\{X\} = \{F(t)\} \quad (4)$$

The response of this equation is given as below [18]:

$$X(t) = exp^{(-\zeta\omega_n t)}\left\{X(0) * \cos(\omega_d t) + \frac{\left(\dot{X}(0) + \zeta\omega_n X(0)\right)}{\omega_d} * \sin(\omega_d t)\right\} \quad (5)$$

wherever, $\omega_d = \omega_n\sqrt{(1-\zeta^2)}$, for under damping system [18].

Transient analysis in ANSYS program contains two ways:

- The first is the full theory.
- The second is the mode superposition theory.

In this paper the full theory employs to analyse the response of the structure when load applied since its usages full matrices for systems also, full theory had high accuracy if it compared with the mode superposition method [16].

In this study, damping ratio was calculated with (Logarithmic decrement method) through experimental test as 1×10^{-4}.

5. Results and discussion

Optimization and comparative study of the four types of sandwich plate are done under transient load excitation in the centre of the upper skin (point (1)), the response curve (time-history analysis) were captured at the time when impact load applied until the deflection reach near zero. The reduction of deflection and stress between point load and centre points of middle and lower face sheet (point (2)) and (point (3)) were calculated for all models as shown in Table 3.

Point (1) of load that positioned on the center of upper skin, point (2) that's positioned below the point of load at the middle skin and point (3) that's positioned on the center of lower skin.

Table 3. Reduction of stress and deflection between upper and lower skins.

Model Type	U_{y1} upper skin (m)	U_{y2} middle skin (m)	U_{y3} lower skin (m)	Reduction of U_y (%)
Triangle Corrugated	8.85E-06	3.79E-07	7.46E-07	91.57
Trapezoid Corrugated	2.66E-04	2.40E-04	2.69E-04	-1.13
Square Honeycomb	3.24E-04	1.34E-06	4.94E-06	98.52
Y-Shaped Corrugated	1.10E-03	4.62E-04	3.72E-04	66.08

Model Type	σ_{v1} (MPa)	σ_{v2} (MPa)	σ_{v3} (MPa)	Reduction of σ_v (%)
Triangle Corrugated	3.64E+07	1.23E+06	105871	99.71
Trapezoid Corrugated	5.49E+07	3.65E+06	9.35E+06	82.97
Square Honeycomb	2.62E+08	601887	649076	99.75
Y-Shaped Corrugated	3.56E+08	1.22E+07	3.75E+06	98.95

5.1. Effect of free vibration

Five modes of shape are obtained by Lanczos method for all models. Table 4 shows the values of these modes, it noticed that the maximum natural frequency for mode 1 is 573.45 rad/s that occurred in square honeycomb core while minimum value is 89.44 rad/s occurred in Y-Shaped.

Table 4. First five natural frequencies for all models in (rad/s).

Model	Mode	Value	Model	Mode	Value
Triangle	First	425.28	**Square**	First	573.45
	Second	428.73		Second	573.53
	Third	434.91		Third	587.09
	Fourth	438.26		Fourth	587.18
	Fifth	469.88		Fifth	621.23
Trapezoidal	First	107.96	**Y-Shaped**	First	89.442
	Second	195.75		Second	184.59
	Third	264.55		Third	281.99
	Fourth	363.39		Fourth	375.13
	Fifth	420.34		Fifth	444.67

The best stiffnesses and high natural frequency is a square model, and this is due to the high thickness of cores and skins than all other models, also the angle of core is large and perpendicular on the skins that give good resistance in case of bending loads. Figure 6 illustrates the five modes for the square honeycomb structure.

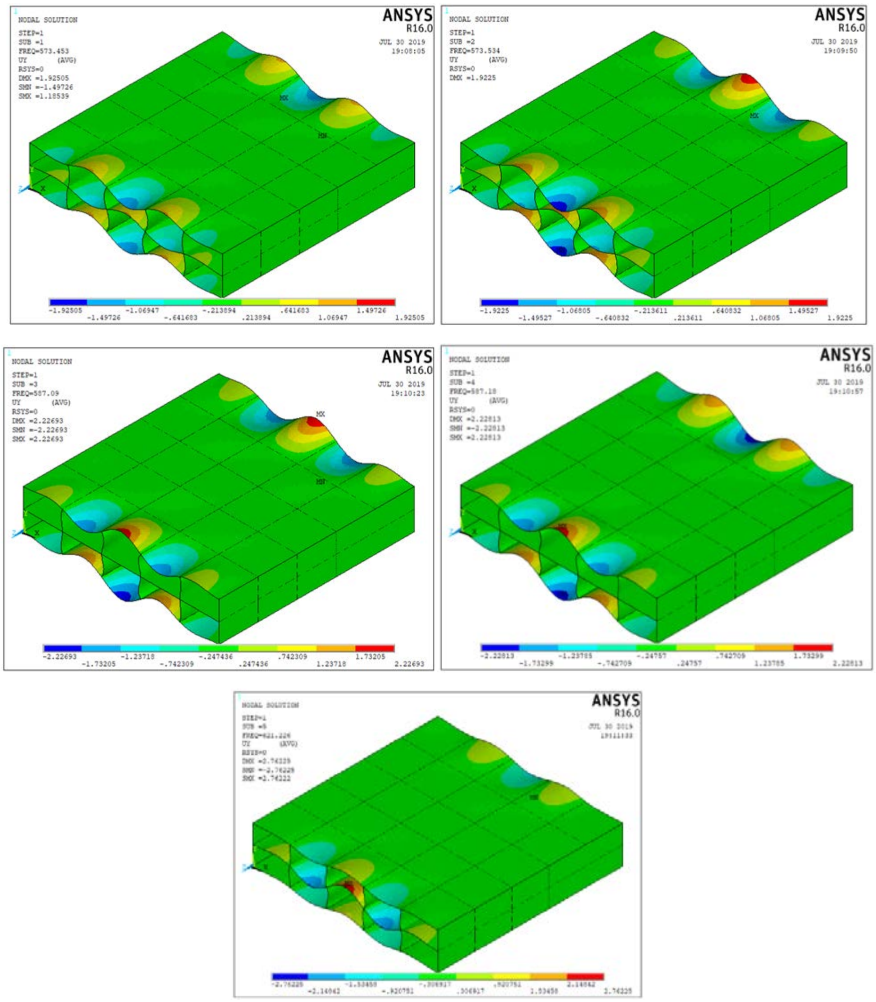


Fig. 6. Mode shape for square honeycomb sandwich structure.

5.2. Effect of forced vibration

The response of sandwich plate is calculated under perpendicular impact load with the magnitude of 350 N for Von-Mises stress and deflection in (y) direction to predict the behaviour of type core on sandwich plate. The main purpose of this work is to get a high reduction in deformation and stress between upper and lower skins. The time of load is 0.01 s, from 0.5 s to 0.51 s.

5.3. Square honeycomb model

The results of square honeycomb sandwich plate show that the maximum stress in point of load at time 0.50054 s with a value of 262.259 MPa and maximum deflection occurred also in point of load with a value of 0.333 mm as shown in Fig. 7 and deflection in Fig. 8.

This type of core produces maximum reduction in stress and displacement between the upper and lower surfaces with values of 99.75 % and 98.47 % respectively.

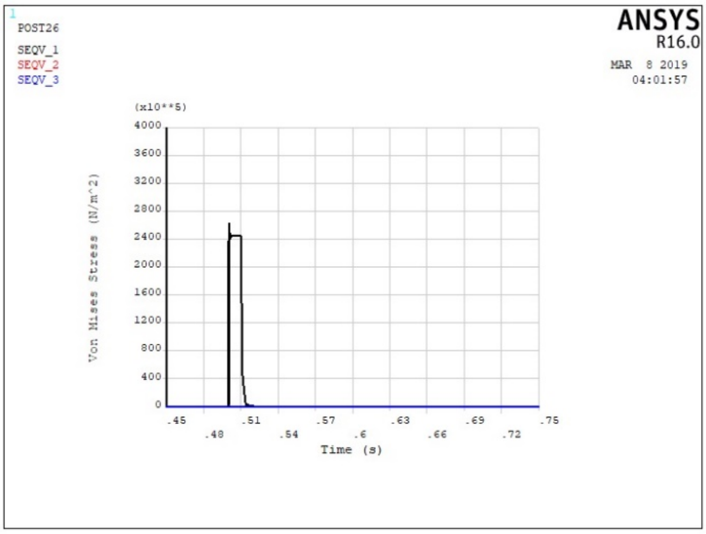


Fig. 7. Stress response for square honeycomb sandwich plate.

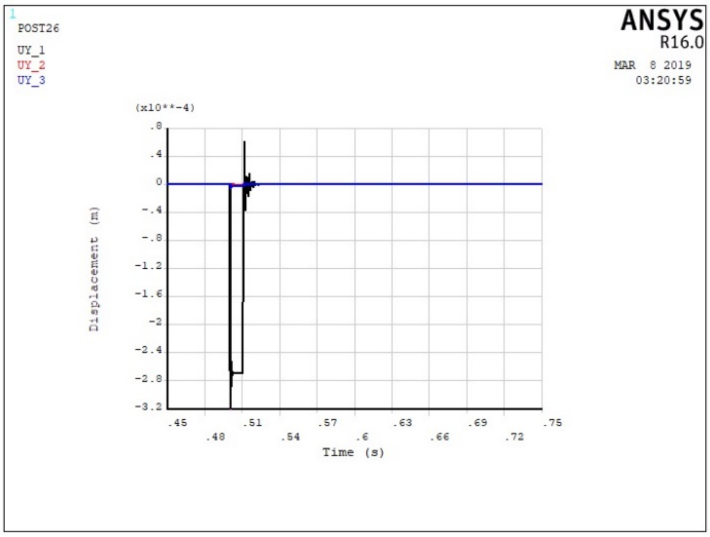


Fig. 8. Deflection response for square honeycomb sandwich plate.

5.2.2. Trapezoidal corrugated model

The results of Trapezoidal corrugated sandwich plate show that the maximum stress in point of load at time 0.5006 s with a value of 59.3 MPa while, maximum deflection occurred in the centre of lower face because the point load has high thickness (core and skin thicknesses) and the point maximum deflection occurred in the skin only with a value of 0.297 mm as shown in Fig. 9 and deflection in Fig. 10.

Also, Trapezoidal corrugated sandwich structure state that the reduction in stress and displacement between the three surfaces is 82.97 % and -1.13 % respectively. The response of this model as shown in Fig. 10 gives high damping than other models, this was occurred because large contact area between skin and core, where the sheet skin has damping more than core.

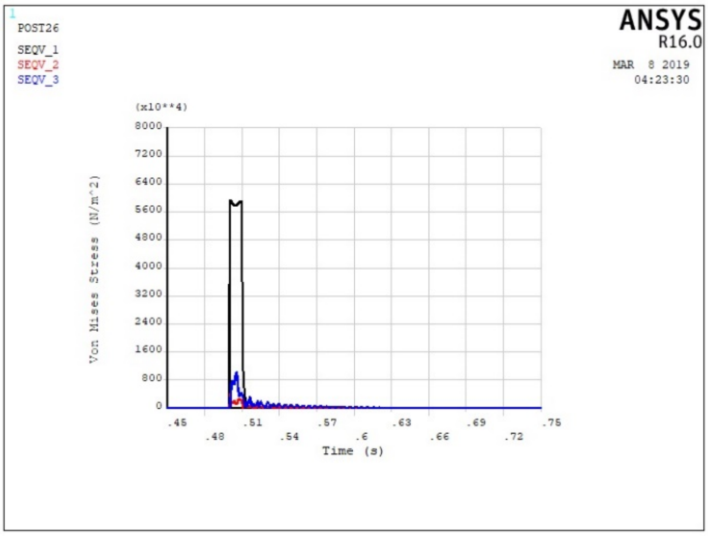


Fig. 9. Stress response for trapezoidal corrugated sandwich plate.

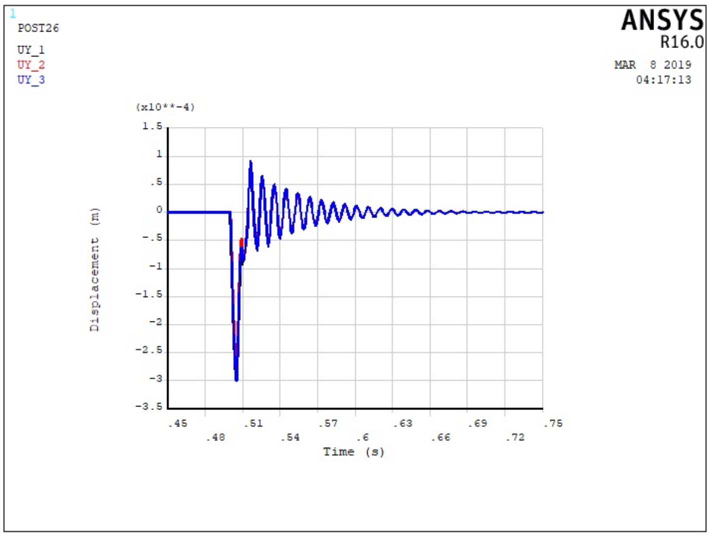


Fig. 10. Deflection response for trapezoidal corrugated sandwich plate.

5.2.2. Triangular corrugated model

The results of triangular corrugated sandwich plate show that the maximum stress in point of load at time 0.50075 s with a value of 36.4 MPa and maximum deflection occurred also in point of load with a value of 0.00885 mm as shown in Fig. 11 and deflection in Fig. 12.

Also, Triangular corrugated sandwich structure state that the reduction in stress and displacement between the three surfaces is of 99.71 % and 91.57 % respectively.

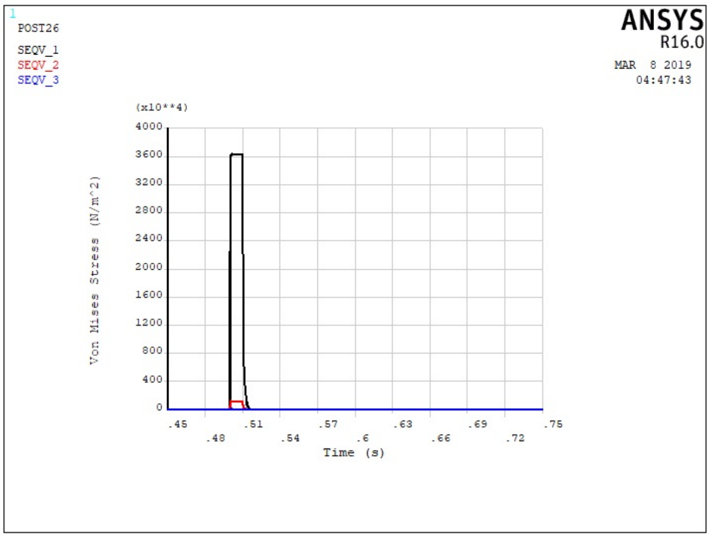


Fig. 11. Stress response for triangular corrugated sandwich plate.

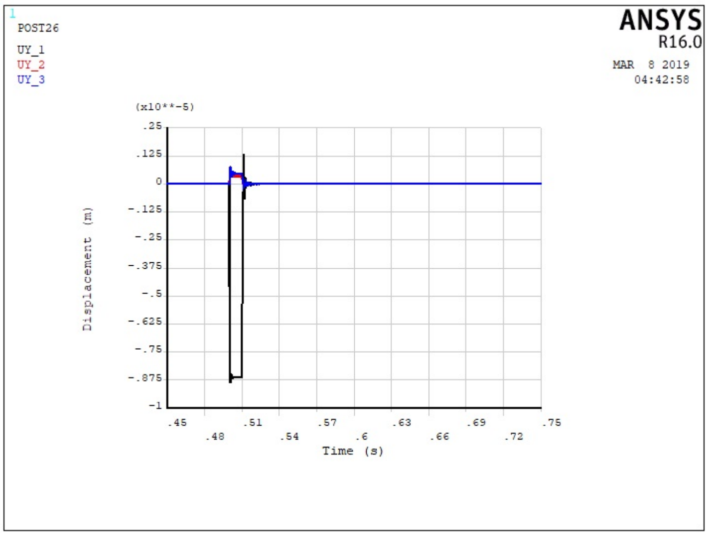


Fig. 12. Deflection response for triangular corrugated sandwich plate.

5.2.3. Y-Shaped corrugated model

The results of Y-Shaped corrugated sandwich plate show that the maximum stress occurred in point of load at time 0.50540 s with a value of 356 MPa and maximum deflection occurred also in point of load with a value of 1.10 mm as shown in Fig. 13 and deflection in Fig. 14**.**

Also, Y-Shaped corrugated sandwich structure state that the reduction in stress and displacement between the three surfaces is 98.95 % and 66.08 % respectively.

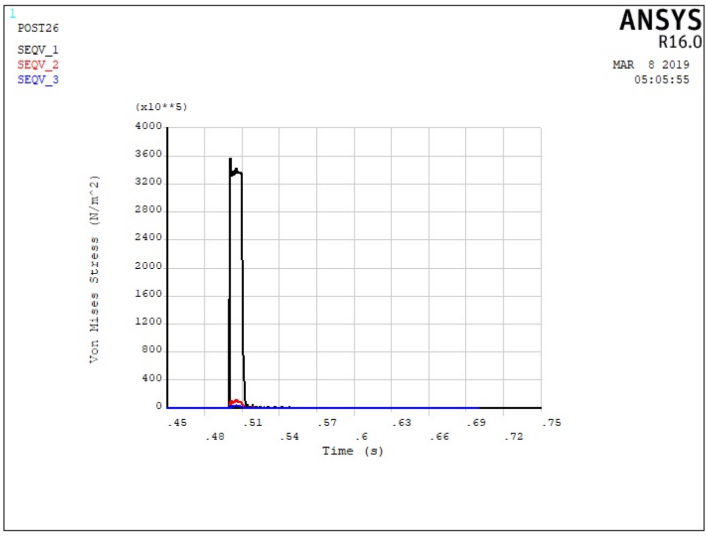


Fig. 13. Stress response for Y-shaped corrugated sandwich plate.

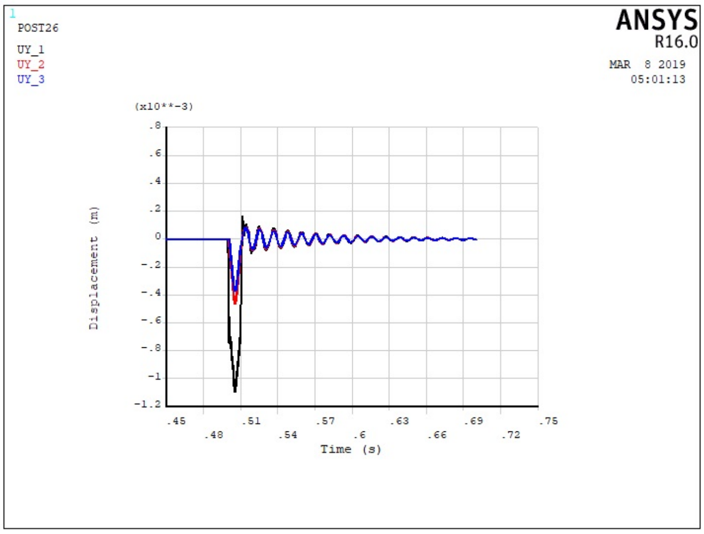


Fig. 14. Deflection response for Y-shaped corrugated sandwich plate.

5.3. Verification case study

The maximum deflection of plate with different thickness is calculated numerically and compared with analytical solution by Vanam et al. [19] for verifying case study, isotropic rectangular aluminum plate as shown in Fig. 15. is taken from Vanam et al. [19], with dimensions and material properties illustrated in Table 5. In the present work, the model is solved numerically by FE approach using ANSYS program. Then comparison was shown in Fig. 16 for the static analysis by applying distributed load of 500 N/m^2 with plate thickness varying in the range of 10 mm to 180 mm, with boundary condition all sides are clamped. It is found that the maximum difference between them estimated by 4.1095 % and these differences the maximum deflection is shown in Table 6.

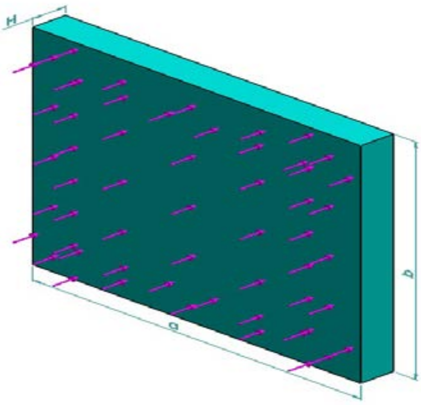


Fig. 15. Geometry of plate.

Table 5. Dimensions and material properties of reference [19].

Geometry and material properties	Value
Length a (m)	1
Width b (m)	1
Thickness h (mm)	10 to 180
Distributed load (N/m^2)	500
Modulus of elasticity (GPa)	70
Poisson's ratio ($\boldsymbol{\nu}$)	0.33

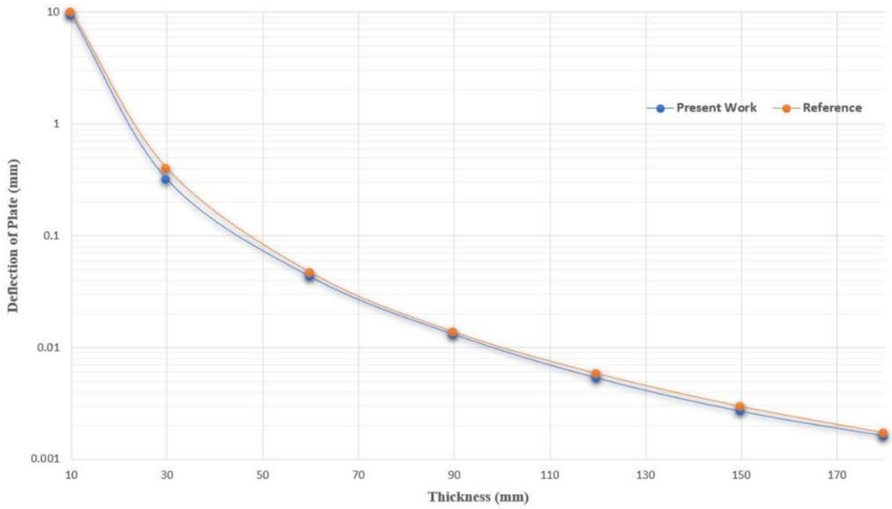


Fig. 16. Comparison of maximum deflection of plate of both works.

Table 6. Maximum deflection, all dimensions in (mm).

Thickness	Present work	Analytical solution [19]	Error (%)
10	9.69	9.86	1.7241
30	$3.50 * 10^{-1}$	$3.65 * 10^{-1}$	4.1095
60	$4.47 * 10^{-2}$	$4.56 * 10^{-2}$	1.9736
90	$1.33 * 10^{-2}$	$1.35 * 10^{-2}$	1.4814
120	$5.56 * 10^{-3}$	$5.70 * 10^{-3}$	2.4736
150	$2.86 * 10^{-3}$	$2.92 * 10^{-3}$	2.0547
180	$1.66 * 10^{-3}$	$1.69 * 10^{-3}$	1.7751

5.4. Recommendation and limitation

This study has revealed to us several important things that researchers in this field should be interested in. Also, we adhered to some limitations because we were unable to cover all factors affecting in the results.

- Studying the optimization of models to get the optimum design.
- Using composite materials to get better mechanical properties.
- Studying the effect of change the boundary condition and nature of impact.
- Take into considered the plastic deformation and non-linear analysis.
- The creation of the mesh network was specified by the number of specific elements in order to be intersected and avoiding nodes sliding.

6. Conclusion

The results of the transient analysis concluded that:

- Maximum stress and maximum deformation occurred at the point of load.
- Maximum stress occurred at the Y-Shaped SSP with values about 356 MPa.
- Maximum deformation occurred at the Y-Shaped SSP with values about 1.10 mm.
- Maximum stress separation between upper face and lower occurs at square honeycomb with reduction of stress 99.75 %.
- Maximum deflection separation between upper face and lower occurs at square honeycomb with reduction of deflection of 98.51 %.
- The model that withstands higher stresses with minimum deflection was the triangle corrugated SSP, so it can be the best model of the various SSP types.
- The optimum shape of stiffnesses is the square honeycomb, which has higher natural frequencies to give stiffer structure with 573.45 rad/s for first mode and 621.23 rad/s for fifth mode.
- The region that located near the edges have very low stress and deformation so, it is recommended in the future studies to reduce the thickness of metal at this region for obtained low weight and saving metal.

Nomenclatures

$[C]$	Matrix of damping structure, N.s/m
F(t)	Force respect to time, N
$\{F(t)\}$	Force response vector, N

$[K]$	Matrix of stiffness structure, kg/s^2
$[M]$	Matrix of mass structure, kg
t	Time, s
U_y	Deflection in (Y) direction, m
$X(t)$	Displacement response, m
$\dot{X}(0)$	Initial velocity, m/s
$\{X\}$	Displacement vector, m
$\{\dot{X}\}$	Velocity vector, m/s
$\{\ddot{X}\}$	Acceleration vector, m/s^2
$\{X_i\}$	Displacement response vector for the i^{th} mode of vibration, m

Greek Symbols

φ_i	Mode shape vector for the i^{th} mode of vibration
σ_v	Von-Mises stress, N/m^2
ω_i	Angular frequency of mode *i* of vibration, rad/sec
ω_n	Natural frequency, rad/s
ω_d	Damped frequency, rad/s
ζ	Damping ratio
ν	Poisson's ratio

Abbreviations

APDL	ANSYS Parametric Design Language
C-F-C-F	Clamped-Free-Clamped-Free
C-F-F-F	Clamped-Free-Free-Free
DOF	Degree of Freedom
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Method
FRP	Fibre Reinforced Plastic
SSP	Sandwich Structure Plate

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