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DYNAMIC BEHAVIOUR OF SANDWICH PLATE WITH DIFFERENT CORE CONFIGURATION UNDER ACTION OF IMPULSIVE LOADING

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ABSTRACT
Steel sandwich structures with honeycomb and corrugated cellular cores have demonstrated the capability of supporting significant static bending loads while also enabling effective mitigation of impulse loads, the main objectives to use these structures is weight reduction and isolate or reduce the deflection and stress. This research aims to study the effect of dynamic load on the dynamic properties of various types of sandwich cores then find the best model that withstand high stresses and dissipate loads with less mass was possible. The studied model of sandwich is of dimensions (500x500x100) mm with five cells. Four types of steel sandwich plate (SSP) finite element models of various core types have been created: (1) triangle corrugated core, (2) trapezoid corrugated core, (3) square honeycomb and (4) out-of plane hexagonal honeycomb, the mass of various types was constant with value of 13.75 kg. The SSP types were compared by using ANSYS (15.0) APDL software.The finite element models are examined under the effect of transient concentrated stepped load of (350N) during 10ms. The time history response showed that the minimum von-Mises stress and minimum deflection occur at triangle corrugated SSP with values of stress (12.5Mpa) and deflection ($3.8\mu m$), but in energy absorption the square honeycomb is the best type with reduction of stress (99.65%) and reduction of deflection of (98.95%).
Keywords: SSP, Impulsive Load, Corrugated cores, Honeycomb cores, Sandwich shell.

السلوك الديناميكي لشطيرة صفيحة لنوى مختلفة الترتيب تحت تأثير الاحمال النبضية

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الخلاصة
اثنتت هياكل الشطائر الحديدية بنوى خلية النحل والخلوية المتموجة قدرتها على دعم احمال الانحناء الثابتة الكبيرة مع امكانية التخفيف الفعال للاحمال النبضية، الهدف الاساسي لاستخدام هذه الهياكل هو تقليل الوزن و عزل او تقليل الانحراف والاجهاد. يهدف هذا البحث لدراسة تاثير الاحمال الديناميكية على الخصائص الديناميكية للانواع المختلفة لنوى الشطائر ثم ايجاد افضل نموذج الذي يقاوم الاجهادات العالية و يخمد الاحمال باقل كتلة ممكنة. نماذج الشطائر تمتلك نفس الابعاد الخارجية (500x500x100) mm بخمس خلايا. اربع انواع من نماذج العناصر المحددة لشطيرة صفيحة فولاذية بانواع نوى مختلفة كونت (1) قلب مثلث متموج، (2) قلب شبه منحرف متموج، (3) قلب خلية نحل مربع و (4) قلب خلية نحل سداسي. الكتلة لمختلف الانواع ثابتة بمقدار يساوي (13.75) كغم حيث تتم مقارنة الانواع المختلفة لشطيرة صفيحة فولاذية بواسطة برنامج ANSYS (15.0) APDL ، نماذج العناصر المحددة اختبرت تحت تاثير حمل نبضي مركز تصاعدي بمقدار (350N) خلال (10ms). استجابة الحمل النبضي بينت ان اقل اجهاد von-Mises واقل انحراف يحث عند شطيرة الصفيحة الفولاذية ذات القلب المثلث المتموج بقيم اجهاد (12.5MPa) و انحراف ($3.8\ \mu m$)، لكن بامتصاص الطاقة شطيرة الصفيحة الفولاذية ذات خلية النحل المربعة هي افضل نوع بتقليل الاجهاد بمقدار (99.65%) وتقليل الانحراف (98.95%).

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LIST OF SYMBOLS

Symbol	Definition	Units
$[M]$	Whole Structure Mass Matrix	
$[C]$	Whole Structure Damping Matrix	
$[K]$	Whole Structure Stiffness Matrix	
$\{\delta\}$	Displacement Vector	
$\{\dot{\delta}\}$	Velocity Vector	
$\{\ddot{\delta}\}$	Acceleration Vector	
$\{F(t)\}$	Transient Force Vector	
$\delta(t)$	Displacement response	m
ω_n	Natural Frequency	rad/s
ζ	Damping Ratio	-----
ω_d	Damped Natural Frequency	rad/s

INTRODUCTION

Weight reduction is one of the main design drivers of modern engineering and transport structures for aerospace and automotive applications. In this context, the sandwich design principle is playing a major role, as it allows for much higher weight-specific bending stiffness compared to a monolithic structure. A sandwich structure typically consists of two thin and stiff skins, separated by a lightweight cellular core. The main purpose of the core is to increase the bending stiffness by separating the skins, to carry transverse shear loads and to withstand compressive loads normal to the sandwich surface. In case of transversal impact loads, the core has to support the skins from local bending and it has to prevent major damage or penetration by a high energy absorption capability **Serge Abrate et al [2013]**.

Sandwich structures commonly consists of two thin face sheets made from material with high density, stiff and strong such as fiber composite or metal attached to a thick lightweight material called core. This concept imitate an I-beam, but in two dimensions, where the face sheets support bending loads and the core transfers shear force between the faces in a sandwich structure under excitation of load **Terry C. Domm [1972]**.

Honeycomb cores have a unit cell that is either square, triangular or hexagonal that can be translated and repeated in two or three dimensions. These cores can be manufactured from slotted metal sheets and can be attached to face sheets by one of joining method such as welding, brazing, adhesive bonding and diffusion bonding. However, after attaching to face sheets, honeycomb cores are closed-cell and there is no access into the core region unless the webs are deliberately perforated which can significantly degrade their strength. Corrugated cores have a unit cell that can be triangular, diamond or trapezoidal shaped. These cores can be manufactured by bending metal sheets and sandwich structures are then made by brazing or welding them to face sheets. Unlike honeycomb cores, corrugated cores are an open celled geometry in one direction and do not restrict one directional lateral access into the core region after face sheet bonding. Often times, however, smaller face sheet-core nodal joints make corrugated cores more susceptible to face sheet debonding than honeycomb cores **Wetzel [2009]**.

Many applications for sandwich plates in engineering such as: marine, aerospace, biomedical, mechanical engineering and civil in view of their simplicity of dealing with, great mechanical properties and low creation cost. Sandwich plates are widely used in engineering applications and industrial fields as previously described. Holes and other openings are extensively used as structural members, mainly for practical considerations. Holes are commonly found as access ports for mechanical and electrical systems or simply to reduce weight. Cutouts are also needed to provide access for hydraulic lines, for damage inspection, to lighten the loads, provide ventilation and for altering the resonant frequency of the structures. Also cutouts have wide use with composite material such as in aircraft fuselage, ships, and other high performance structures. In addition, the designers often need to incorporate cutouts or openings in a structure to serve as doors and windows. In some cases holes are used to reduce the weight of the structure **Hatem & Nawal [2014]**.

Fatt and Park [2000], presented a honeycomb sandwich under action of impact load to find the analytical solution for the ballistic limit by blunt and spherical projectiles. The kinetic energy absorbed at the lower face sheet where the load dissipated at the core. **Xue and Hutchinson [2004],** compared between sandwich plate and normal plate made from the same material and both have the same weight. The sandwich plate's cores manufactured with three types as: pyramidal truss, square honeycomb and normal plate. under action of dynamic load it is seen that sandwich plate performance better than normal plate for energy absorption. **McShane et al [2006],** measured the response of sandwich plate and normal plate with circular cross section and clamped from the circumference, both sandwich and normal plate has the same weight and thickness. under action of dynamic loading by applying projectile at the centre of plate and sandwich plate it is seen that the sandwich plate resistance better than monolithic plate to shock load, the square honeycomb sandwich plate core is better than the other core types in shock load resistance . **Ch. Naresh et al. [2014],** The dynamic characteristics of sandwich structures under action of impact loading were studied. Finite element method was used to study the effect of core shape on the behavior of sandwich structure under action of impact loading. The models used in the study are: normal plate, square honeycomb and hexagonal honeycomb sandwich plate. When 1kN applied to the models, the deformation was global and high for normal plate while local for sandwich plate. While the comparison between square sandwich plate and hexagonal sandwich plate showed that the stress and deformation lower in square honeycomb that indicating the square type was better than hexagonal sandwich plate. **Pan Zhang et al [2015],** studied the dynamic response of sandwich plate with trapezoidal corrugated core configuration under action of air blast loading by experimentally. The effects of core height, core thickness, corrugation angle and face sheet thickness on the deflections of the trapezoidal corrugated core sandwich plate were studied. From results the effect of upper face sheet thickness on the deflection is more than the lower face sheet, where the upper face sheet increasing led to deflection decreasing. When core thickness and corrugation angle increased the blast performance of panel improved and deformation lowered, while when core height increasing the localized deformation became larger at the front face and a lower deflection of back face.

NUMERICAL ANALYSIS

The finite element method is used to study the response of the structure under action of impulsive loading. Various core configurations were considered to study the response of the structure. Also, the structural dynamic properties is required to be estimated using (FEM). All of the above was needed to construct different finite element models and then analyzed under action of dynamic impulsive loading.

Shell element is suitable for sandwich plate system. It is defined as an object which, for the purpose of stress analysis may be considered as the materialization of a surface. The thickness of a shell must be small compared with its other dimensions. Most shells are made of a solid material, and generally, it will be assumed that the material is isotropic and elastic**.**

It is difficult to solve shell matrices manually; it must use software in order to simplify the solution, the ANSYS program is used to solve these case studies.

GEOMETRICAL AND MATERIAL PROPERTIES OF SANDWICH PLATE

In the current work, four types of steel sandwich plates were modeled depending on the type of core of sandwich plate. They are square honeycomb, out-of-plane hexagonal honeycomb, triangle corrugated and trapezoid corrugated. The dimensions of face sheet of various types of sandwich plate (500x500x2)mm, the core with five cell with height of 100mm, the square core thickness of 1.48mm, hexagonal, trapezoid and triangle cores thickness equal to 1mm. All types have the same weight 13.75kg figure (1) show the dimension of these models.

The modulus of elasticity assumed to be equal to 198GPa, Poisson ratio equal to 0.3, and density equal to 7855 kg/m^3.Figure (2) shows the finite element models.

INITIAL AND BOUNDARY CONDITIONS

Initial condition (the condition at time = 0) default to zero, boundary condition of all models clamped from both right and left sides (All DOF sets to zero) and applying a concentrated load at the center of sandwich in Y direction. The boundary conditions are shown in figure (3).

ELEMENT TYPES

The element that used to build the steel sandwich plate models is SHELL281 element. Figure (4) shows the geometry, node locations, and the element coordinate system for this element **ANSYS, Release (15.0).**

MESH CONVERGENCE

The sandwich plate models are solved using modal analysis to find the first five natural frequencies and stated that the results stabilized at mesh element area of (1.25x1.25) cm^2.Table (1) shows the mesh convergence of trapezoidal type.

TRANSIENT LOAD EFFECT

Transient load effect (time-history analysis) is a technique used to determine the dynamic response of a structure under the action of any general time dependent loads.

The under-damped structure solution of the equation of motion considers a structure subjected to a unit impulsive loading at zero time as shown in figure (5) **Singiresu S. Rao.**

$$[M].\{\ddot{\delta}\} + [C].\{\dot{\delta}\} + [K].\{\delta\} = \{F(t)\} \tag{1}$$

is given as follows:

$$\delta(t) = e^{-\zeta\omega_n t}\left\{\delta(0)\cos\omega_d t + \frac{\left(\dot{\delta}(0) + \zeta\omega_n\delta(0)\right)}{\omega_d}\sin\omega_d t\right\} \tag{2}$$

Where, $\omega_d = \omega_n\sqrt{1-\zeta^2}$, for under-damping vibration.

If the mass is at rest before the unit impulse is applied$(\delta = \dot{\delta} = 0\ for\ t < 0\ or\ at\ t = 0)$, we obtain, from the impulse-momentum relation,

$$Impulse = f = 1 = [M]\{\dot{\delta}\}(t=0) - [M]\{\dot{\delta}\}(t=0^-) = [M]\{\dot{\delta}\}(0) \tag{3}$$

Thus the initial conditions are given by:

$$\delta(t=0) = \delta(0) = 0 \tag{4}$$

$$\dot{\delta}(t=0) = \dot{\delta}(0) = \frac{1}{[M]} \tag{5}$$

In view of equations (4) and (5), equation (2) reduces to

$$\delta(t) = g(t).\frac{e^{-\zeta\omega_n t}}{[M]\omega_d}.\sin\omega_d t \tag{6}$$

If the magnitude of the impulse is $\{F\}$ instead of unity, the initial velocity

$\dot{\delta}(0)\ is\ \frac{\{F\}}{[M]}$ and the response of the system becomes

$$\delta(t) = \frac{\{F\}e^{-\zeta\omega_n t}}{[M]\omega_d}.\sin\omega_d t = \{F\}.g(t) \tag{7}$$

The response of the system under an arbitrary external force F(t), shown in figure (6).Assuming that at $\tau,$ the force $F(\tau)$ acts on the system for a short period of time $\Delta\tau,$ the impulse acting at $t = \tau$ is given by $F(\tau).\Delta\tau.$ At any time t, the elapsed time since the impulse $t - \tau,$ so response of the system at t due to this impulse alone is given by

$$\delta(t) = \{F\}.g(t-\tau) = \frac{\{F\}}{[M]\omega_d}e^{-\zeta\omega_n(t-\tau)}\sin\big(\omega_d(t-\tau)\big) \tag{8}$$

With$\{F\} = F(\tau).\Delta\tau$, then

$$\Delta g(t) = F(\tau).\Delta\tau.g(t-\tau) \tag{9}$$

The total response at time t can be found by summing all the responses due to the elementary impulses acting at all times τ :

$$\delta(t) = \sum F(\tau).g(t-\tau).\Delta\tau \tag{10}$$

Letting $\Delta\tau \to 0$ and replacing the summation by integration, to obtain

$$\delta(t) = \int_0^t F(\tau).g(t-\tau)d\tau \tag{11}$$

by substituting equation (6) into equation (11), gives, for zero initial conditions **Singiresu S. Rao**

$$\delta(t) = \frac{1}{[M]\omega_d}\int_0^t F(\tau).e^{-\zeta\omega_n(t-\tau)}\sin\omega_d(t-\tau)d\tau \tag{12}$$

There are two methods for transient analysis in ANSYS software:

1. Full method.
2. Mode superposition method.

In this case full method was used to analyze the load applied to structure because it uses full matrices for systems and it more accurate than mode superposition method **ANSYS, Release (15.0)**.

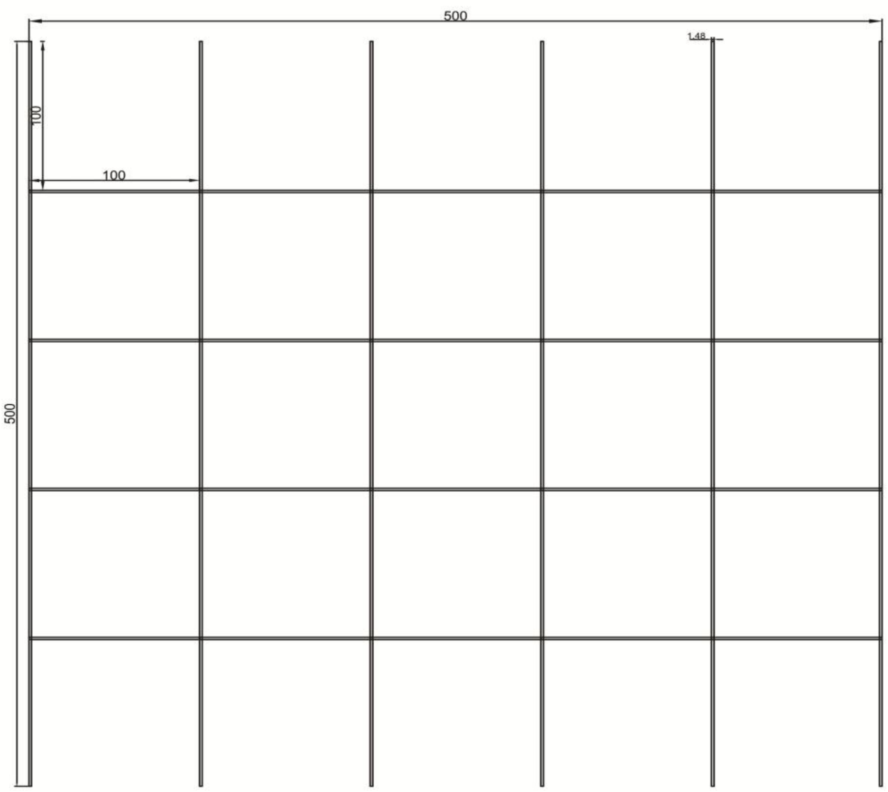
RESULTS AND DISCUSSIONS

The transient load applied to the models as a concentrated load at the center of the structure with value of (350N) during period of (10 ms) as shown in figure (7).

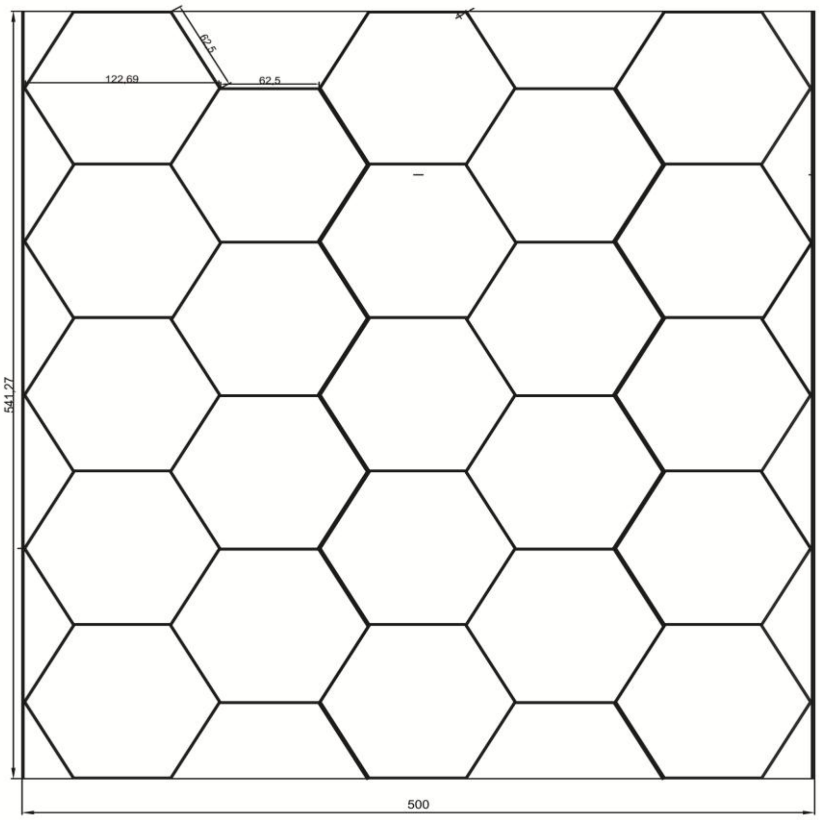
The comparison between sandwich core types under transient load excitation show that square honeycomb core gives maximum stress and displacement reduction of (350N) load between the two surfaces with values of (99.65%) and (98.95%), while minimum values of stress and displacement reduction occur at trapezoidal corrugated core with values of (90.5%) and (0%).Figure (3) shows the point of load that located at upper surface and point (2) that located under point of load at the lower surface.Table (2) shows the results of stress and deflection reduction for all models.

CONCLUSIONS

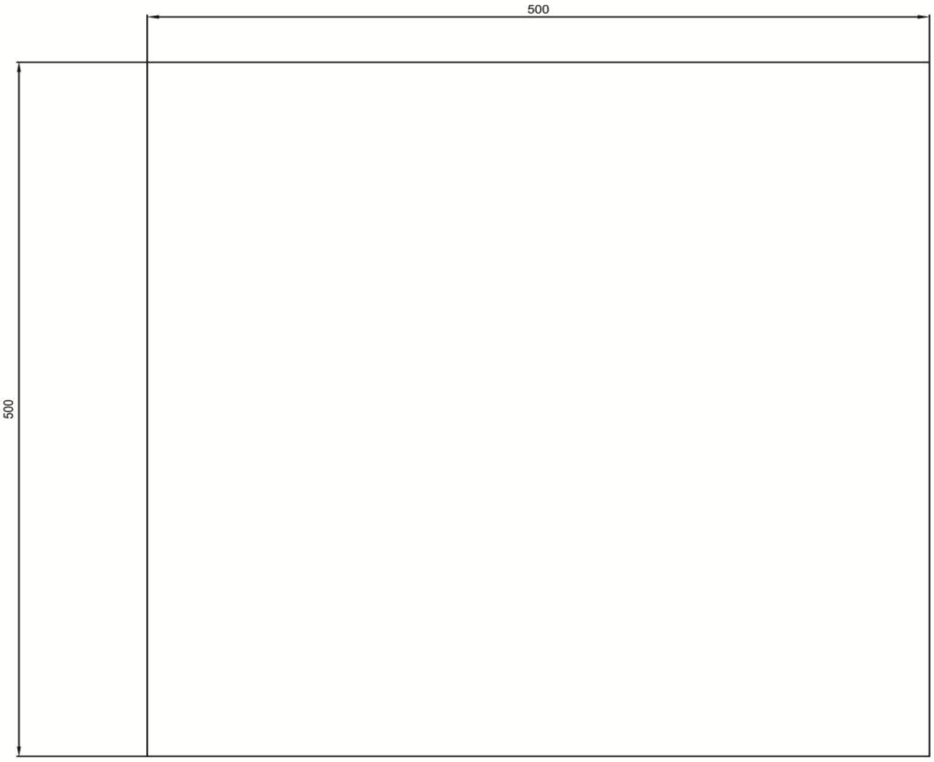
The numerical transient load results showed that the minimum deflection and minimum von-Mises stress occur at the triangular SSP with values of (12.5Mpa) and (3.8μm), but the best one that absorbs load intensity was square honeycomb with reduction of stress (99.65%) and reduction of deflection of (98.95%).



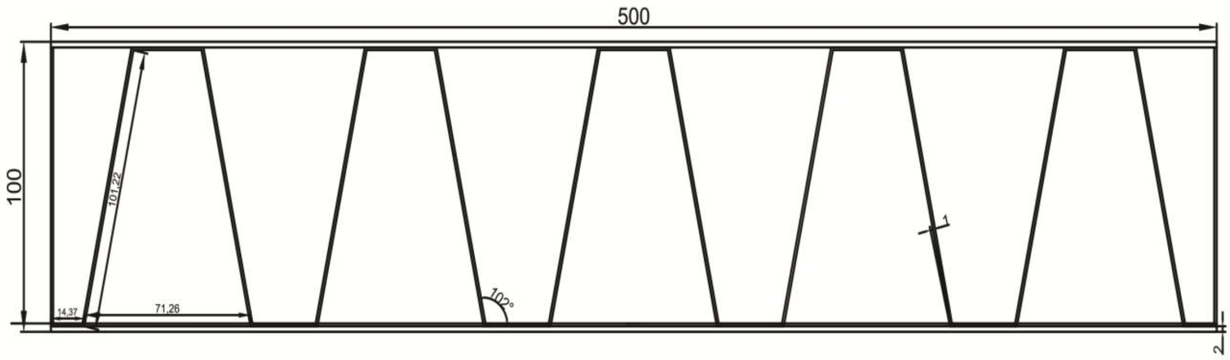
Dimensions of Square Honeycomb Core

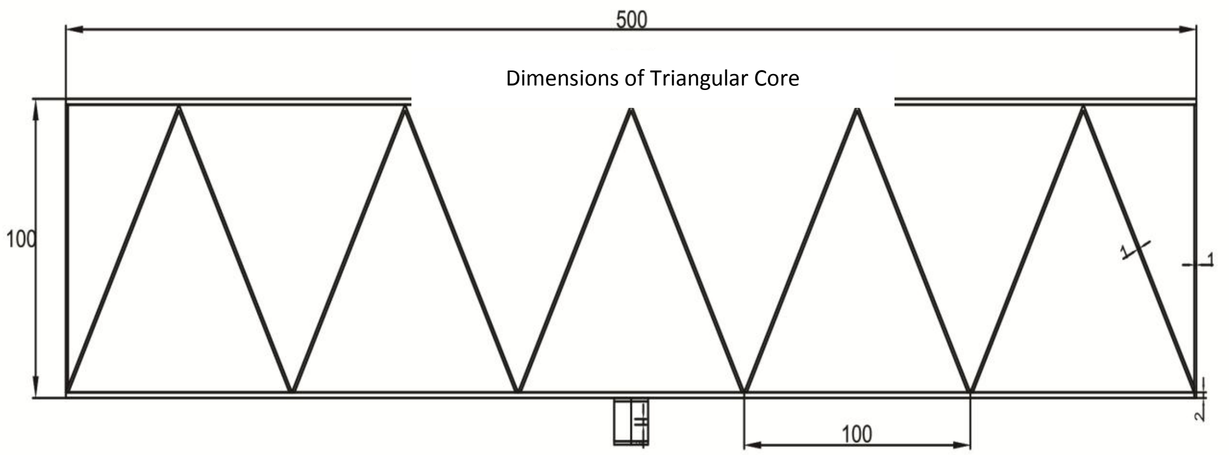


Dimensions of Out-of Plane Hexagonal Honeycomb Core



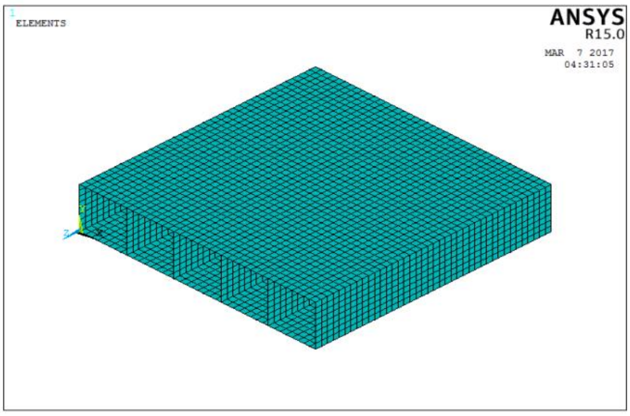
Dimensions of Facesheet Plate



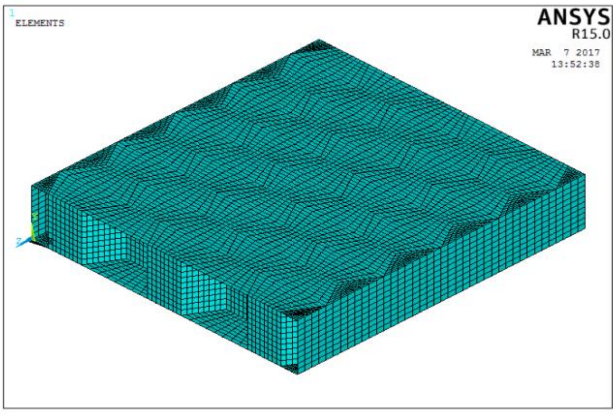


Dimensions of Triangular Core

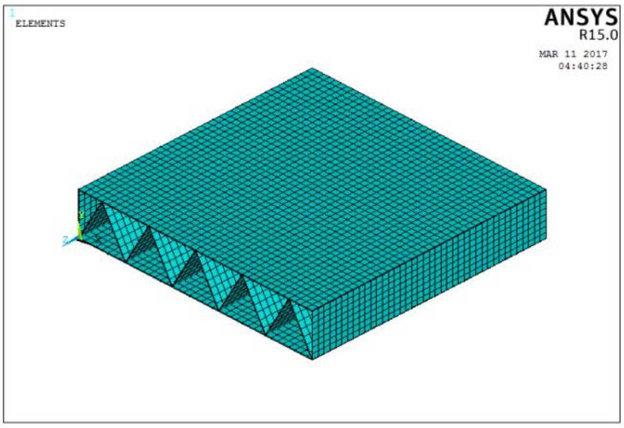
Fig.(1) Dimensions of Geometry of Current Work(all dimensions in mm).



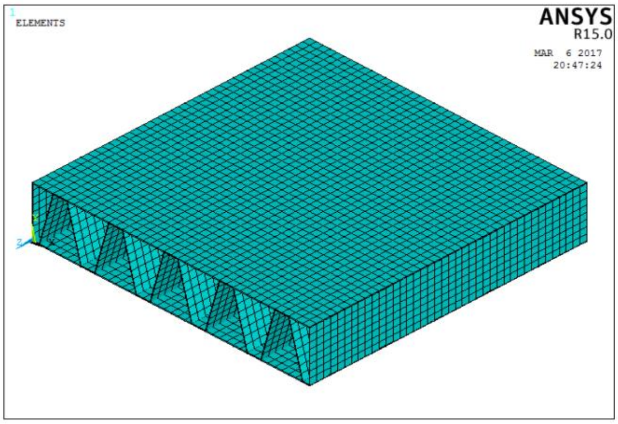
Square Honeycomb Sandwich Plate



Out-of Plane Hexagonal Honeycomb Sandwich Plate

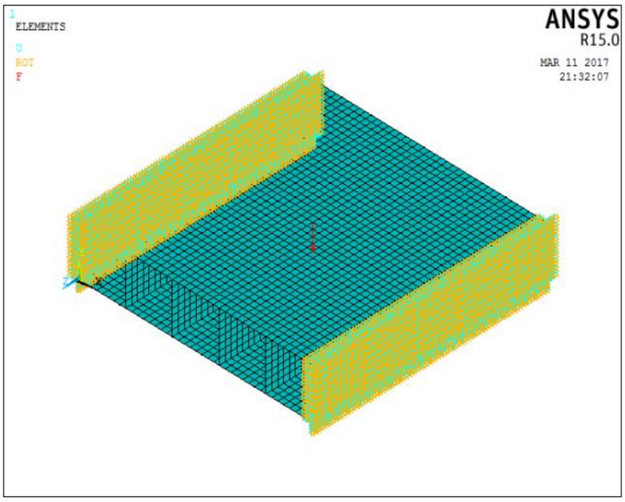


Triangular Corrugated Sandwich Plate

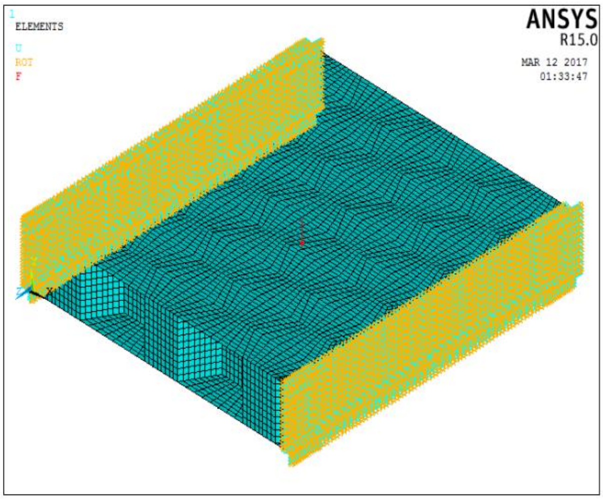


Trapezoidal Corrugated Sandwich Plate

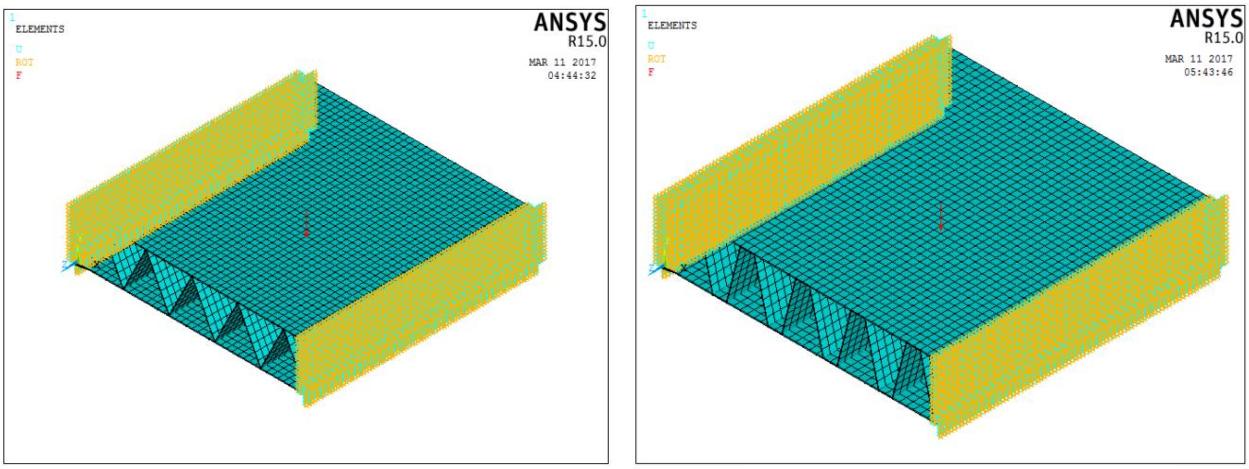
Fig.(2) Finite Element Models of Sandwich Plate.



Square Honeycomb Sandwich Plate



Out-of Plane Hexagonal Honeycomb Sandwich Plate



Triangular Corrugated Sandwich Plate   Trapezoidal Corrugated Sandwich Plate

Fig.(3) Boundary Condition's of All Models of Current Work.

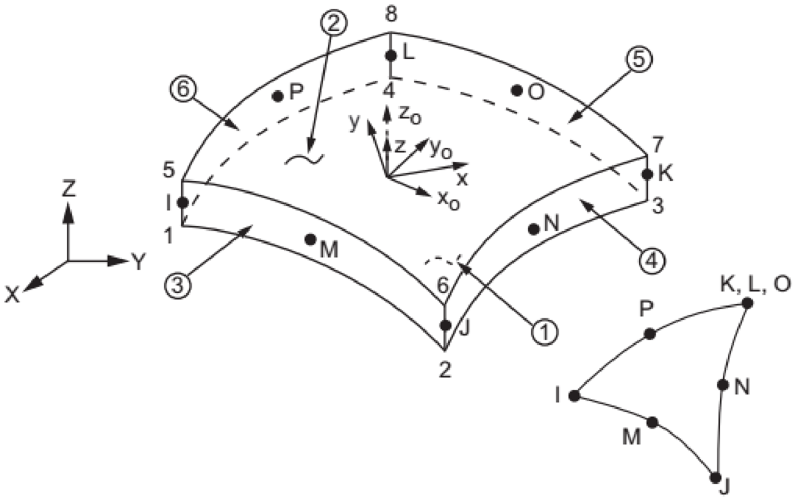


Fig.(4) SHELL281 geometry**.**

Table (3.1) Mesh Convergence.

Mesh Convergence of Trapezoidal Sandwich Plate						
Element Dimension	**No. of Element**	**mode 1**	**mode 2**	**mode 3**	**mode 4**	**mode 5**
(1.75x1.75)cm	**1920**	**130.61**	**228.84**	**302.85**	**371.375**	**418.72**
(1.5x1.5)cm	**3840**	**129.89**	**227.6**	**303.27**	**373.86**	**428.58**
(1.25x1.25)cm	**7840**	**128.22**	**226.88**	**303.55**	**378.40**	**437.62**
(0.75x0.75)cm	**15680**	**129**	**228.22**	**305.74**	**381.54**	**439.8**

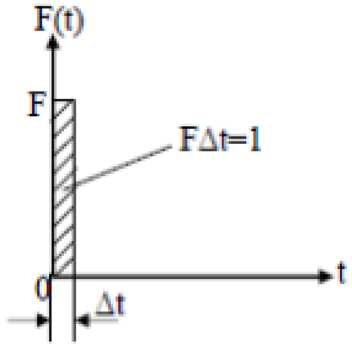


Fig. (5) Impulse Force Excitation**.**

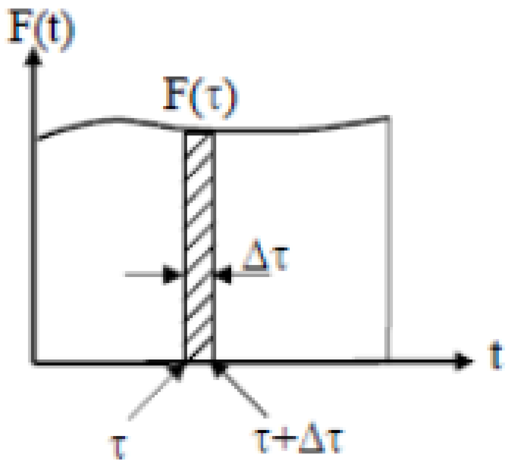


Fig. (6) An Arbitrary (Non-Periodic) Forcing Function.

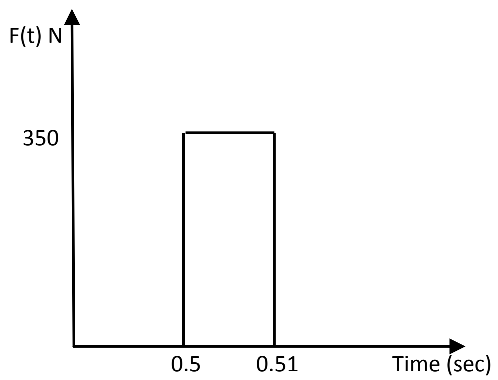
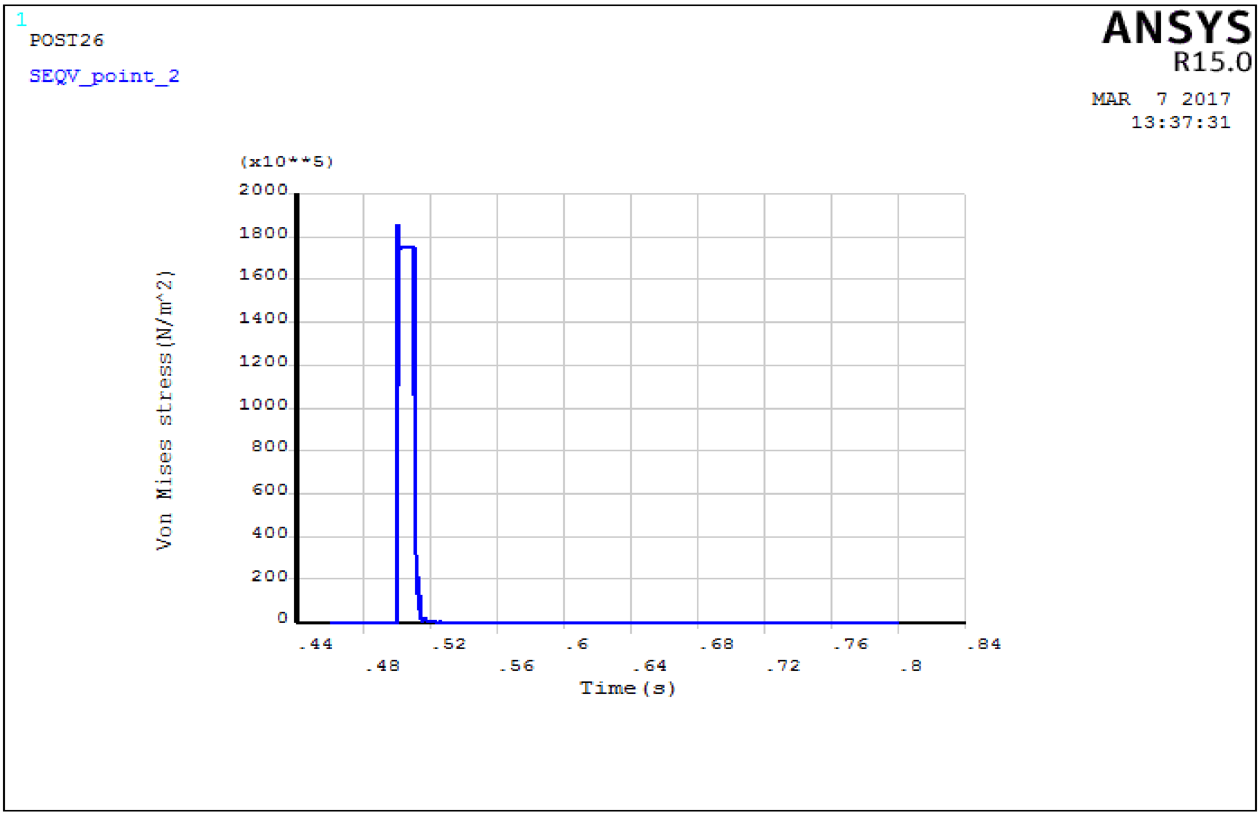
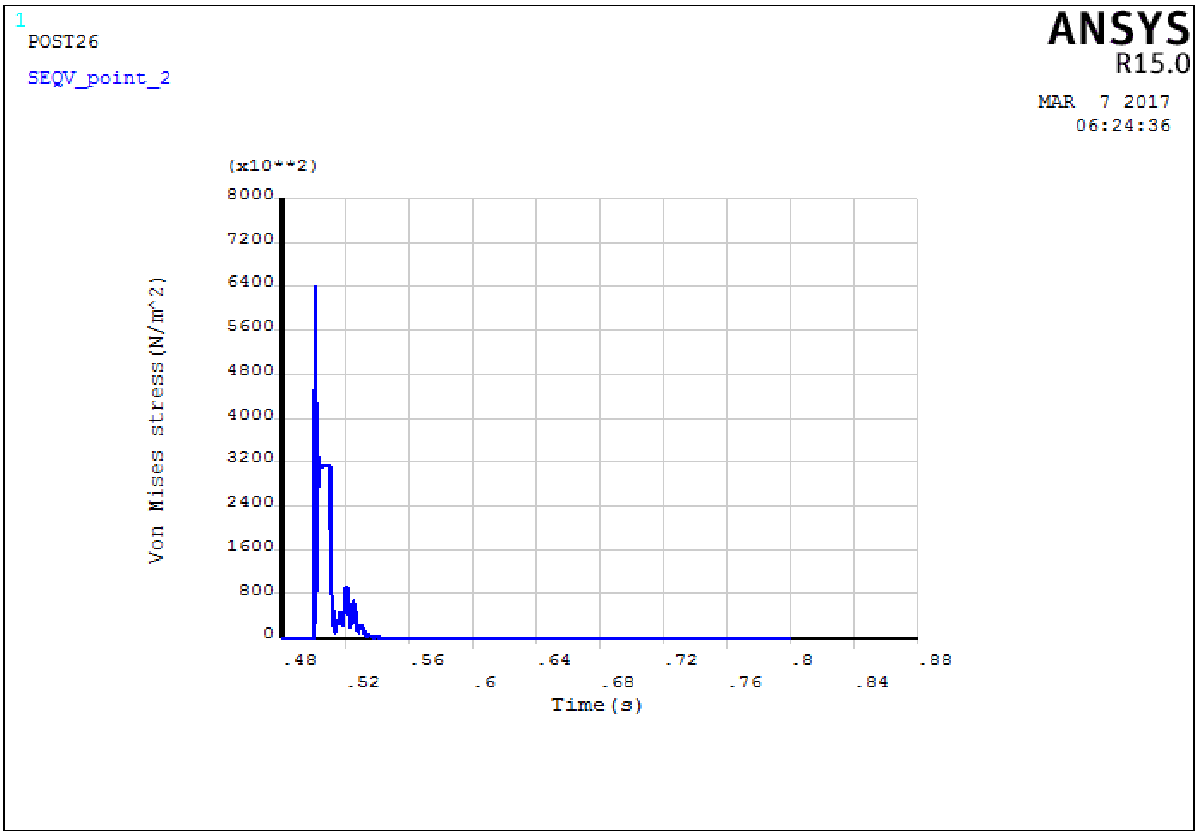


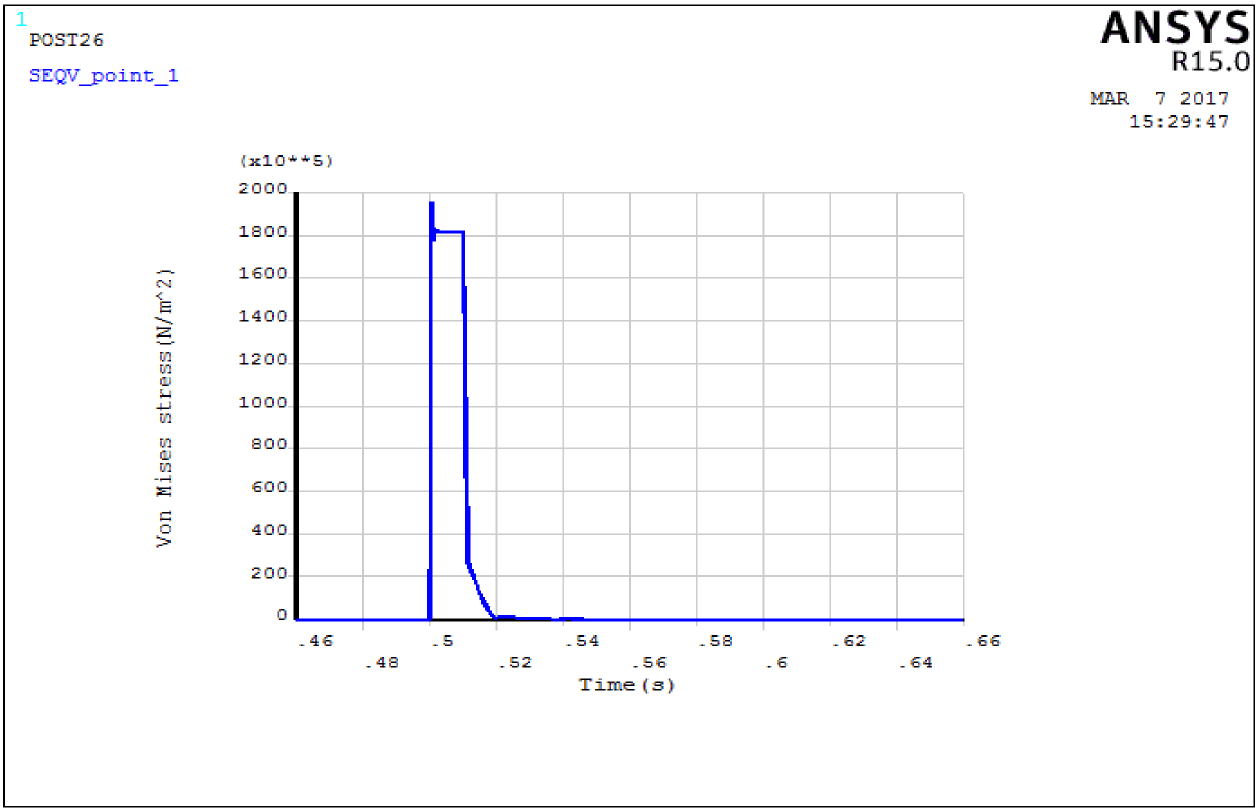
Fig.(7) Transient load.



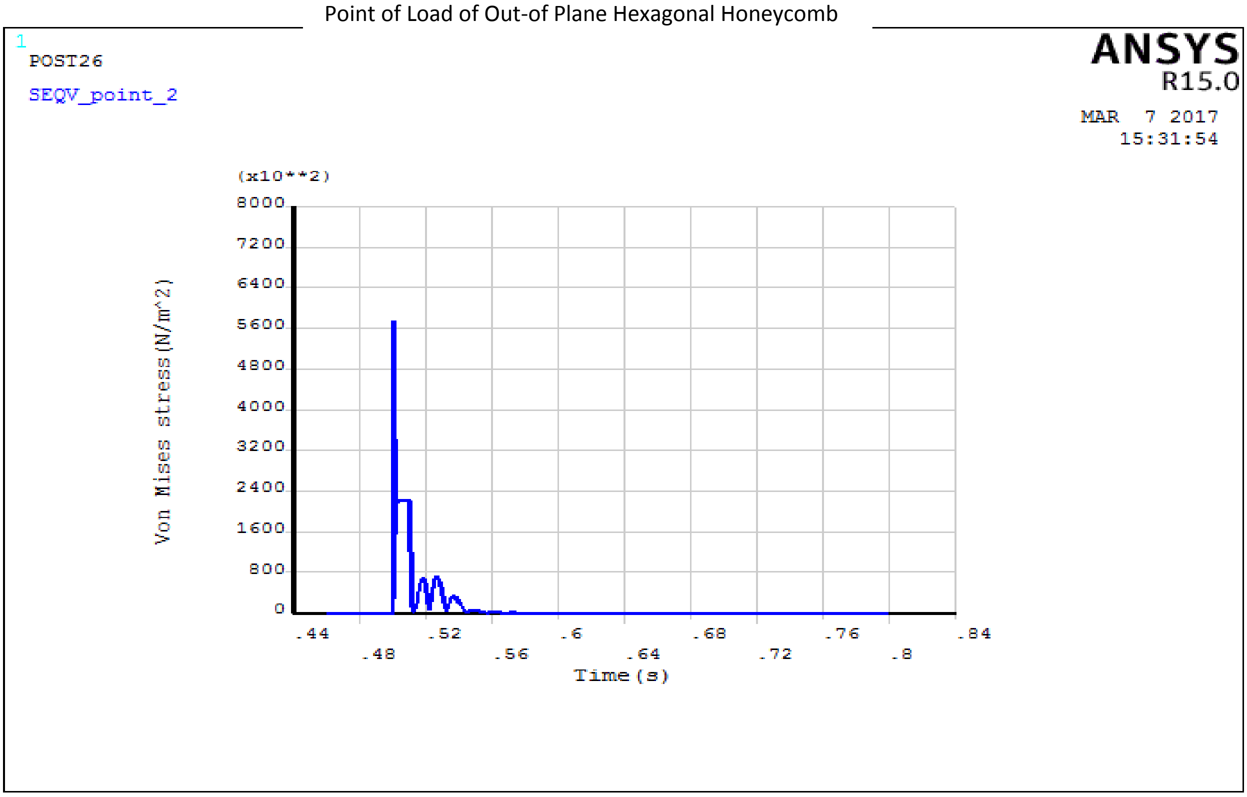
Point of Load of Square Honeycomb



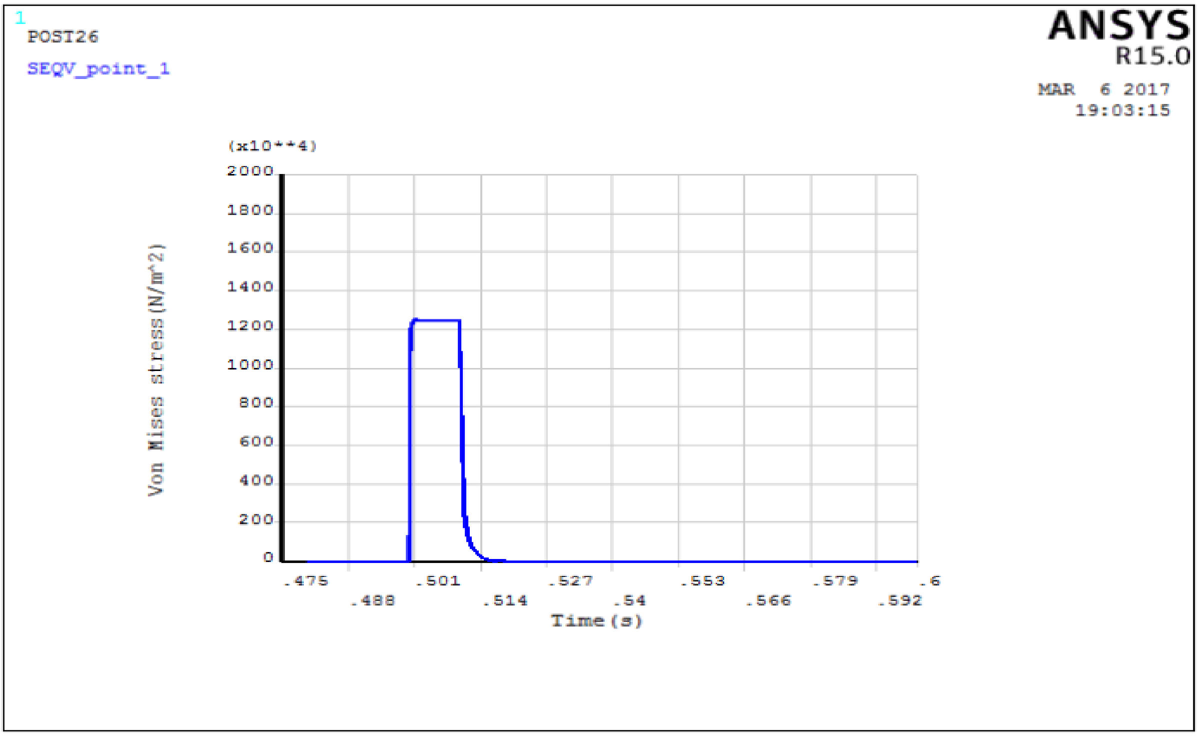
Point 2 of Square Honeycomb



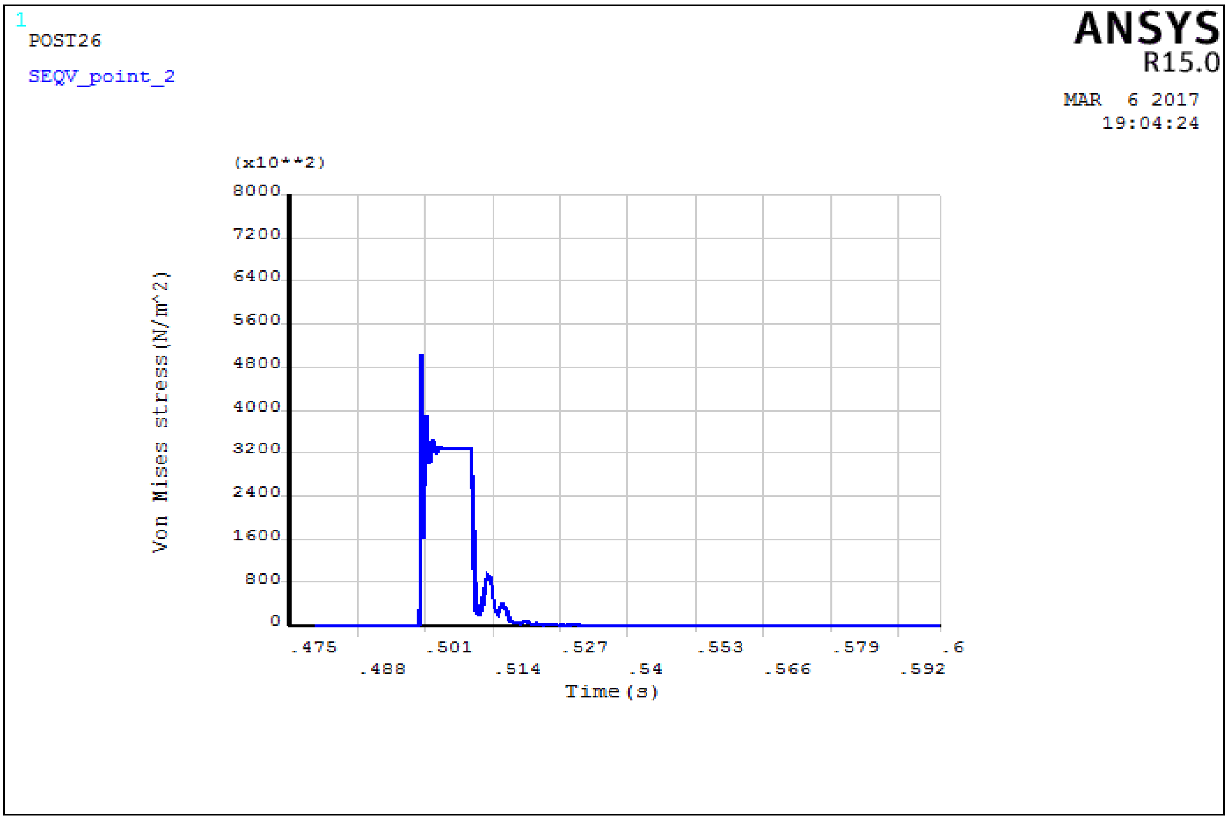
Point of Load of Out-of Plane Hexagonal Honeycomb



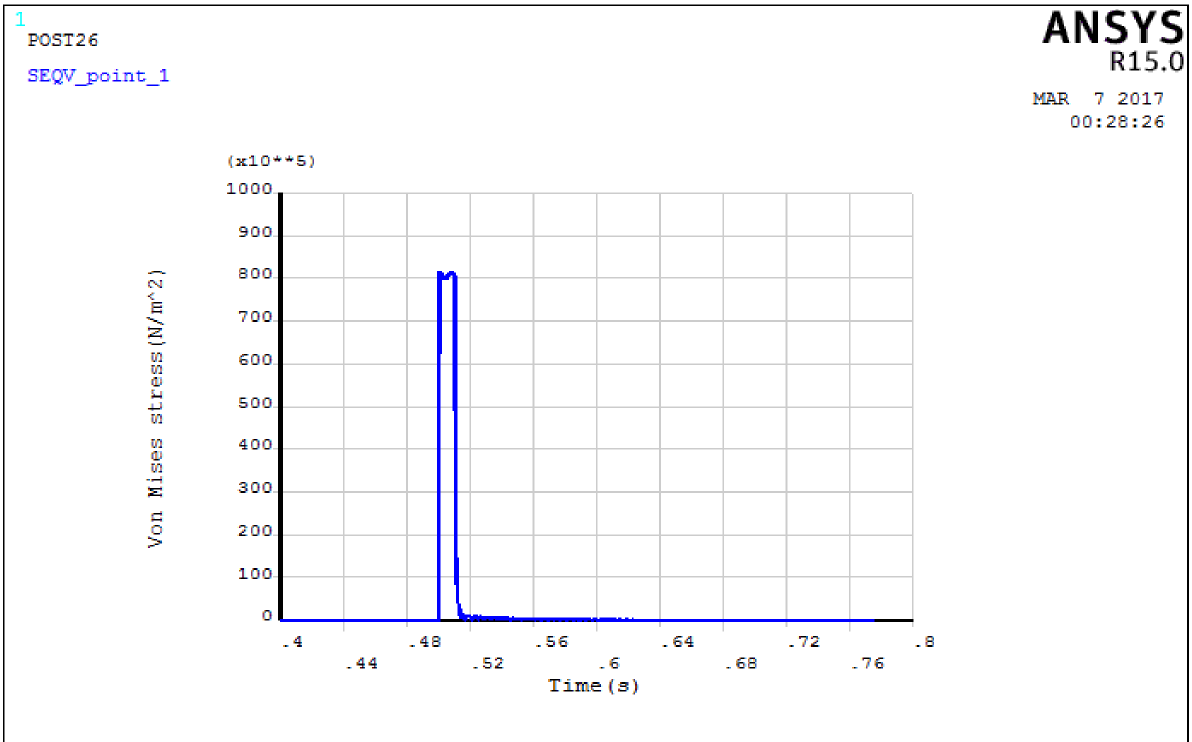
Point 2 of Out-of Plane Hexagonal Honeycomb



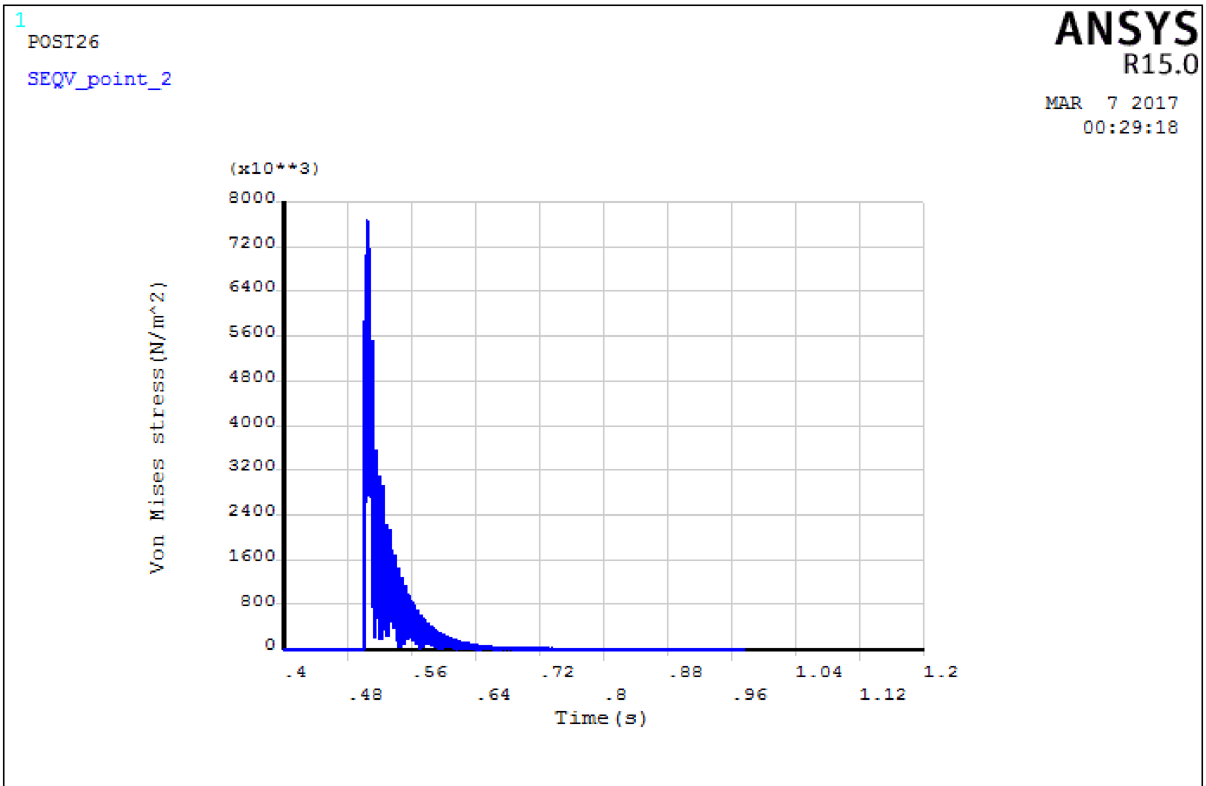
Point of Load of Triangular Corrugated Core



Point 2 of Triangular Corrugated Core

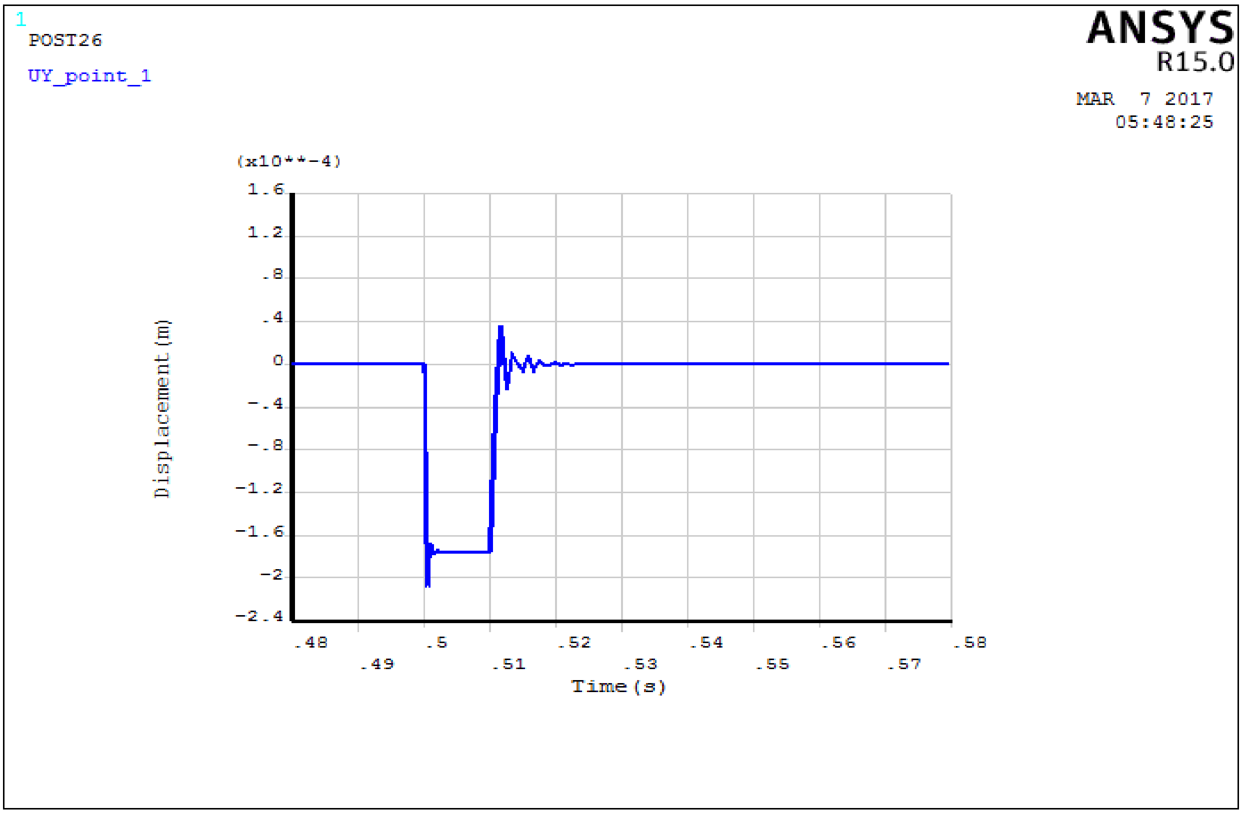


Point of Load of Trapezoidal Corrugated Core

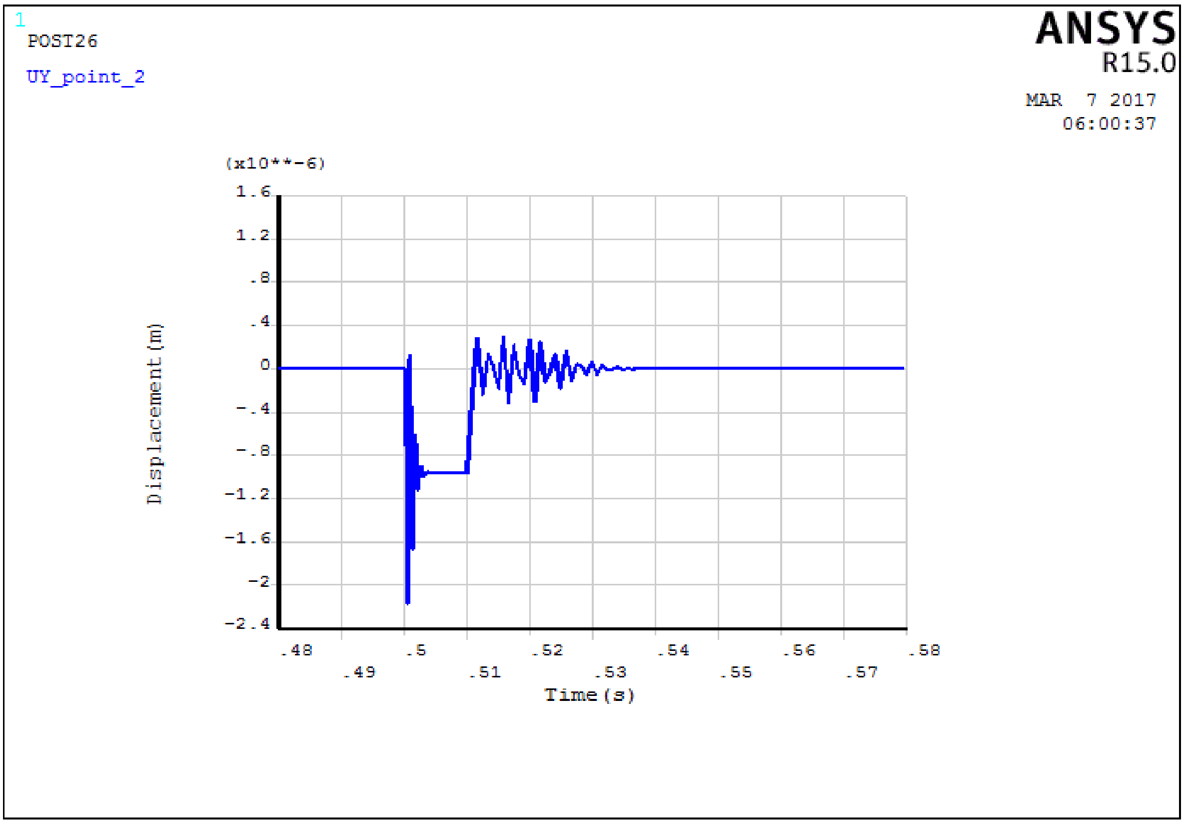


Point 2 of Trapezoidal Corrugated Core

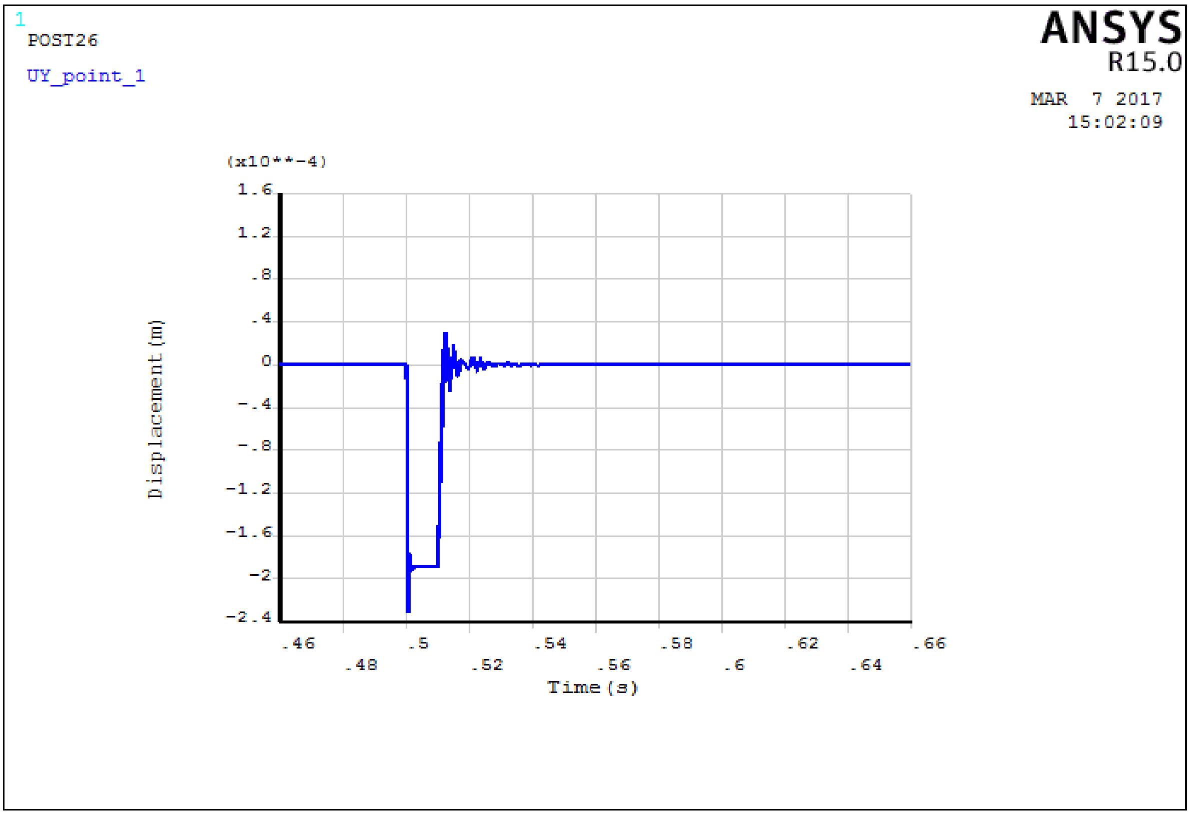
Fig.(8) Von- Mises stress at point of load and point 2 that located at the lower surface under point of load of various types of SSP.



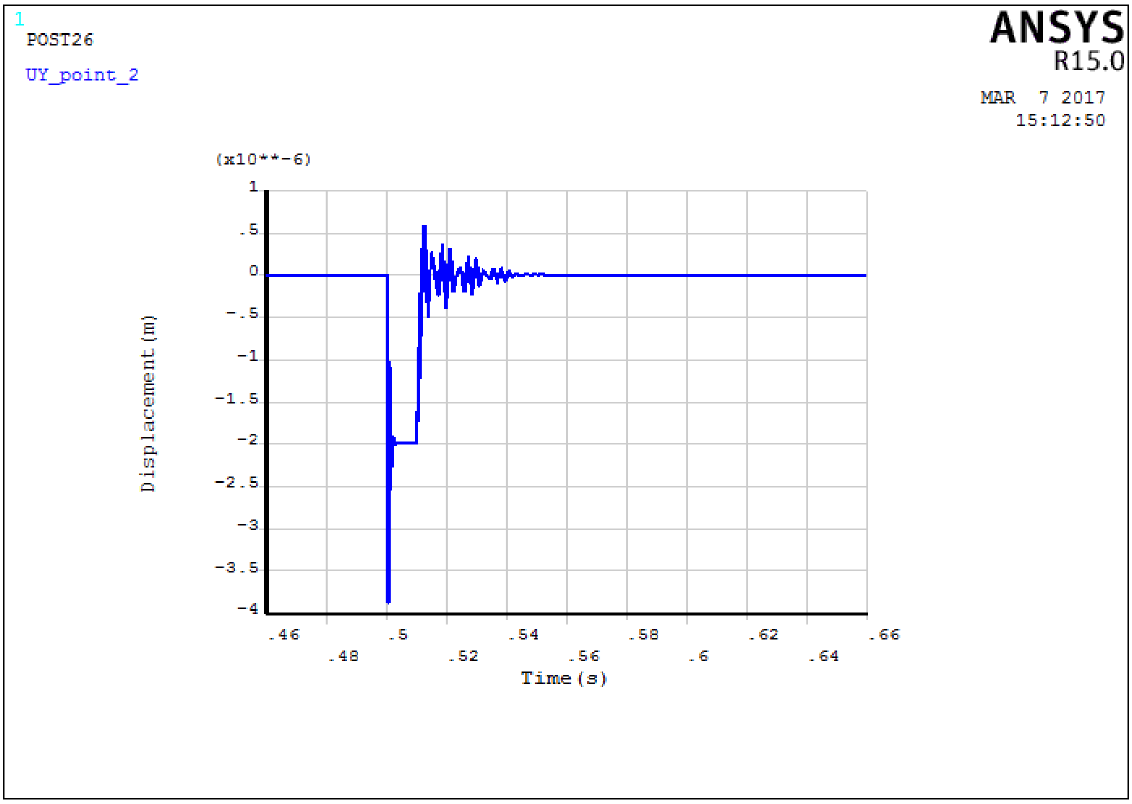
Point of Load of Square Honeycomb



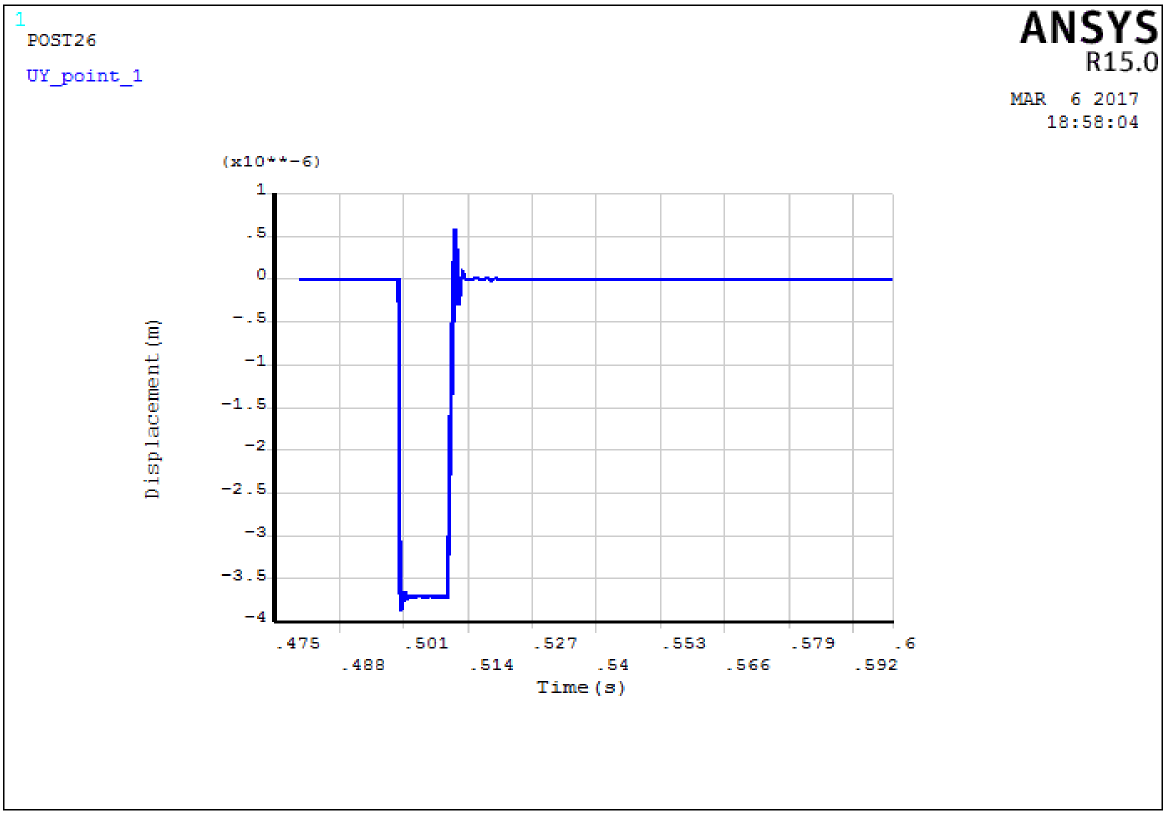
Point 2 of Square Honeycomb



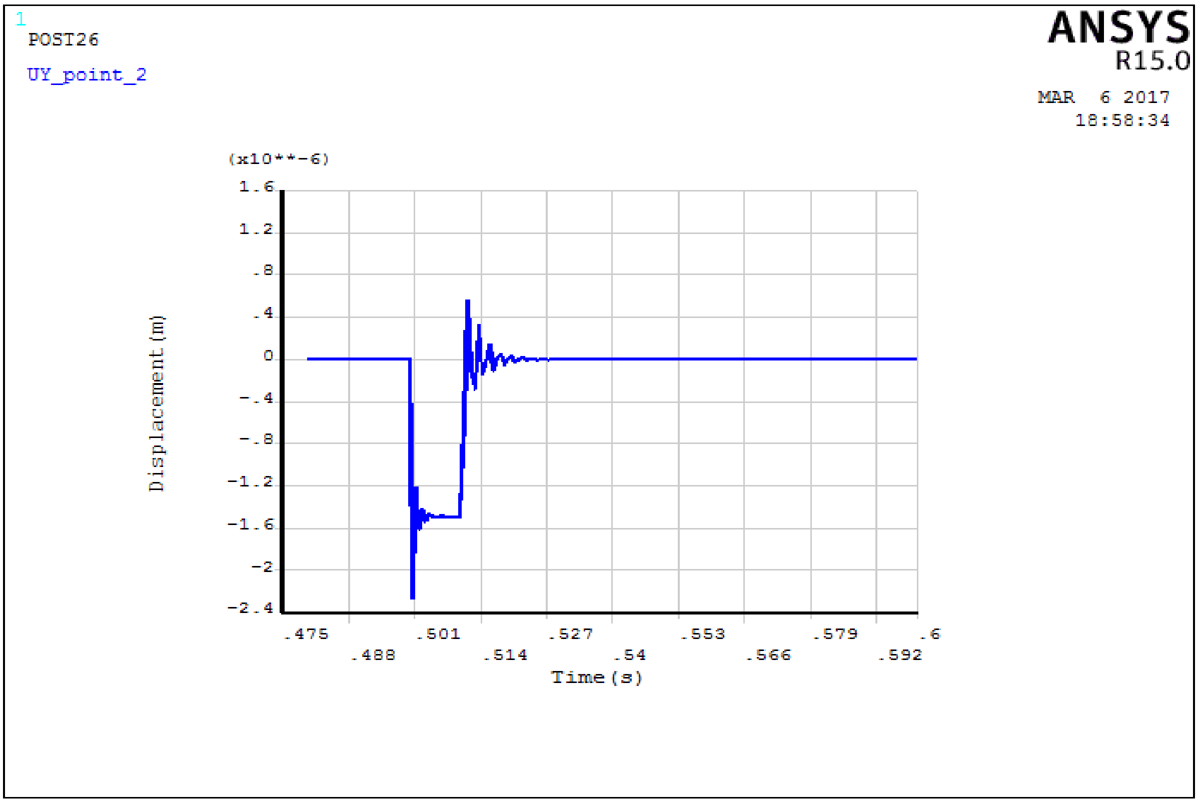
Point of Load of Out-of Plane Hexagonal Honeycomb



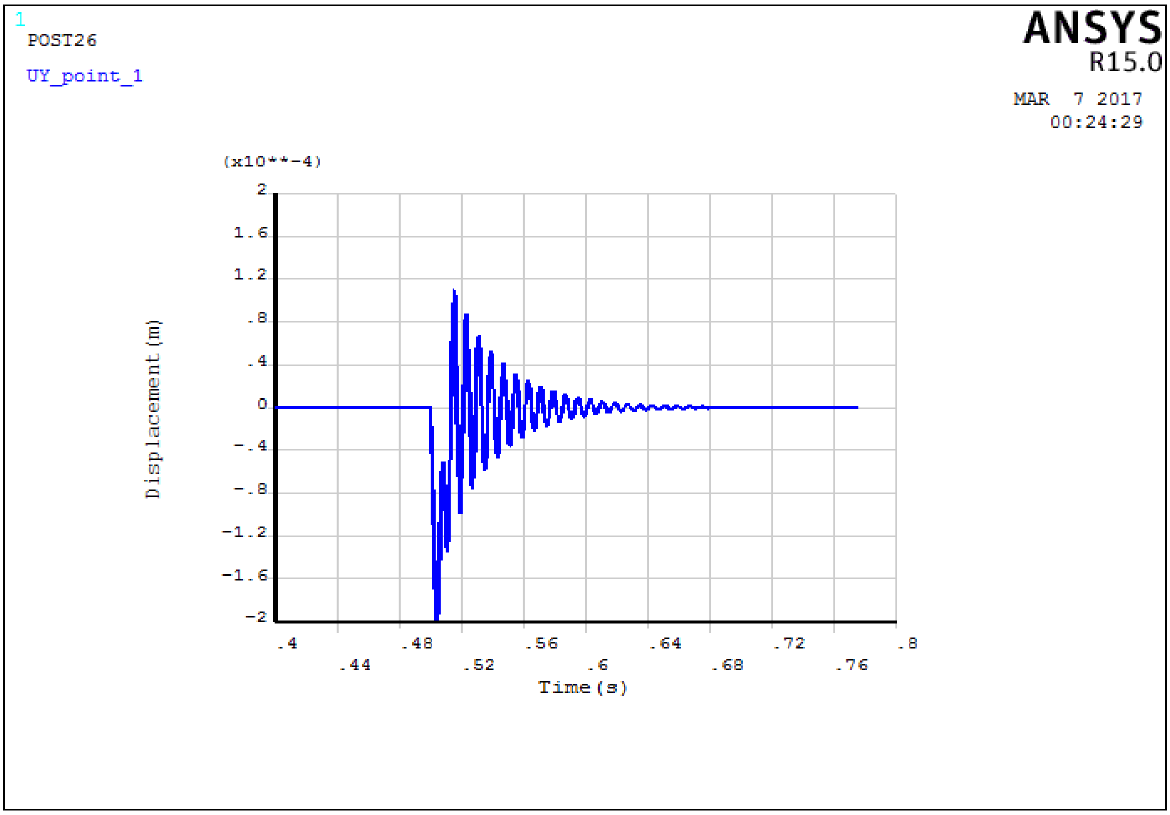
Point 2 of Out-of Plane Hexagonal Honeycomb



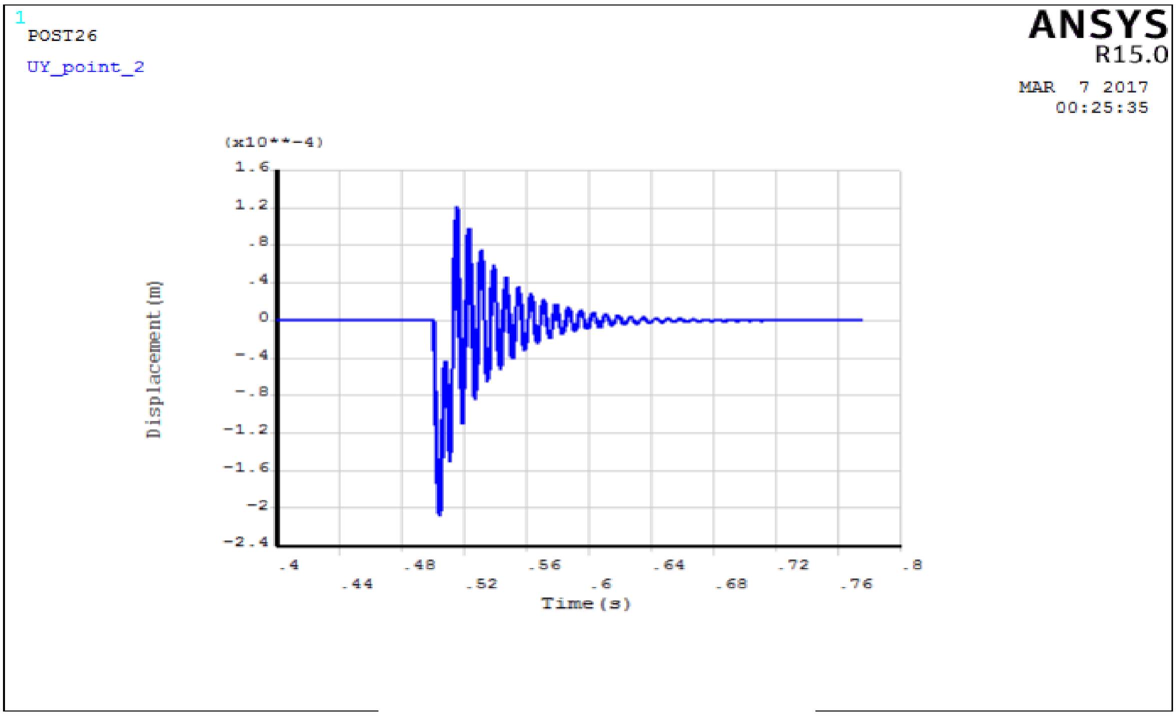
Point of Load of Triangular Corrugated Core



Point 2 of Triangular Corrugated Core



Point of Load of Trapezoidal Corrugated Core



Point 2 of Trapezoidal Corrugated Core

Fig.(9) Deflection at point of load and point 2 that located at the lower surface under point of load of various types of SSP.

Model Type	U_{y1} (m)	U_{y2} (m)	σ_{v1} (MPa)	σ_{v2} (MPa)	Reduction of U_y %	Reduction of σ_v %
Triangular Corrugated Core	$3.8x10^{-6}$	$2.2x10^{-6}$	12.5	0.51	$\frac{3.8-2.2}{3.8} = 42.1$	$\frac{12.5-0.51}{12.5} = 95.92$
trapezoid Corrugated Core	$2x10^{-4}$	$2x10^{-4}$	82	7.75	$\frac{2-2}{2} = 0$	$\frac{82-7.75}{82} = 90.5$
Square Honeycomb	$2.1x10^{-4}$	$2.2x10^{-6}$	184	0.64	$\frac{210-2.2}{210} = 98.95$	$\frac{184-0.64}{184} = 99.65$
Out-of Plane Hexagonal Honeycomb	$2.31x10^{-4}$	$3.8x10^{-6}$	195	0.85	$\frac{231-3.8}{231} = 98.35$	$\frac{195-0.85}{195} = 99.56$

Table (2) Reduction of stress and deflection.

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