Study the Effect of Face Sheets Material on Strength of Sandwich Plates with Circular Hole

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ABSTRACT

This study aims to investigate the effect of changing skins material on the strength of sandwich plates with circular hole when subjected to mechanical loads. Theoretical, numerical and experimental analyses are done for sandwich plates with hole and with two face sheet materials. Theoretical analysis is performed by using sandwich plate theory which depends on the first order shear deformation theory for plates subjected to tension and bending separately. Finite element method was used to analyse numerically all cases by ANSYS program.

The sandwich plates were investigated experimentally under bending and buckling load separately. The relationship between stresses and the ratio of hole diameter to plate width (d/b) are built, by studying the effect of hole size on strength of sandwich plates. The maximum stress were developed at the hole region in sandwich plates clarified the dropped in their strength. So, the experimental maximum stress was found by means of multiplying the experimental nominal stress obtained from Stress-strain curve by the stress concentration factor.

All results which obtained, theoretically, numerically and experimentally are compared to find that the hole weaken the strength of sandwich plates because of the stress concentration and that weakness is depending on the hole size and the face sheets materials.

Keywords: sandwich plate, hole size, sandwich plate theory, stress concentration.
الكلمات الرئيسية: الصفائح الشطيرية, حجم الثقب, نظرية الصفائح الشطيرية, تمركز الإجهادات

دراسة تأثير مادة القشرة على مقاومة الصفائح الشطيرية المثقوبة

الخلاصة

تهدف هذه الدراسة إلى بحث تأثير مادة القشرة على مقاومة الصفائح الشطيرية المثقوبة عند تعرضها الى الأحمال الميكانيكية. تم تحليل الصفائح الشطيرية المثقوبة وباستعمال نوعين من المعادن كقشرة تحليلا نظريا وعمليا. تم التحليل النظري بطريقتين: الاولى استعمال نظرية الصفائح الشطيرية التي تعتمد على نظرية تشوه القص من الدرجة الأولى للصفائح المعرضة الى حملي الشد و الانحناء بشكل منفصل. اما الثانية فهي التحليل العددي حيث تم باستخدام طريقة العناصر المحدودة لتحليلها عدديا ولجميع الحالات عن طريق استعمال برنامج ANSYS .
وقد تم بحث الصفائح الشطيرية تجريبيا تحت تاثير احمال الانحناء و الانبعاج بشكل منفصل. تم ربط العلاقة ما بين الاجهادات و نسبة قطر الفجوة على عرض الصفيحة a/b. من خلال دراسة تأثير حجم ثقب على قوة الصفائح الشطيرية . ان االاجهاد الأقصى للاجهاد الذي تم الحصول عليه من التجارب العملية في منطقة الثقب في الصفائح الشطيرية أوضح الانخفاض في قوتها. لذلك ، تم العثور على أقصى إجهاد التجريبية عن طريق ضرب الإجهاد الاسمي التجريبية التي تم الحصول عليها من منحنى الإجهاد والانفعال في عامل تركيز الإجهاد.

وتمت مقارنة جميع النتائج التي حصلنا عليها، نظريا ، عدديا و تجريبيا وتبين أن الثقوب تضعف من مقاومة المواد الشطيرية بسبب تركيز الاجهادات و هذا الضعف يعتمد على حجم الثقب وكذلك المادة التي تم استعمالها كقشرة للسطوح .

1. INTRODUCTION

A sandwich structure results from the assembly by bonding -or welding- of two thin facings or skins on a lighter core that is used to keep the two skins separated. Sandwich is built up of three elements as shown in **Fig.1**: two faces, core and joints**.**

Every part has its specific function to make as a unit. The aim is to use the material with a maximum of efficiency. The two faces are placed at a distance from each other to increase the moment of inertia, and thereby the flexural rigidity, about the neutral axis of the structure.

The faces carry the tensile and the compressive stresses in the sandwich. The core has several important functions. It has to be stiff enough to keep the distance between the faces constant. It most also be so rigid in shear that the faces do not slide over each other.

To keep the face and the core co-operating with each other the adhesive between the faces and the core, must be able to transfer the shear forces between the faces and the core. The adhesive must be able to carry shear and tensile stresses. It is hard to specify the demands on the joints. A simple rule is that the adhesive should be able to take up the same shear stress as the core.

The quality of the bond is fundamental for the performance and life duration of the piece. In practice we have**, Daniel, et al., 2003.**

$$0.025 \text{ mm} \leq \text{adhesive thickness} \leq 0.2 \text{ mm}$$

Many applications for sandwich plates in many engineering fields namely: aerospace, biomedical, civil, marine, and mechanical engineering because of their ease of handling, good mechanical properties and low fabrication cost.

Sandwich plates and sandwich beams 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.

The study here is compared between two groups of sandwich plate: one consists of Low carbon steel as face sheets and polyvinylchloride as a core. The other group consists of aluminum alloy 7075-T6 sheets (AA7075-T6) and polyvinylchloride as a core. The sandwich plate was either solid or had a central circular hole with diameter (10, 15 or 20mm) and subjected to tension, bending and buckling loads to study the effect of hole size on its strength. **Qing-Sheng,** and **Wilfried, 2004,** modeled laminated plates with holes by an inclusion problem with anisotropic matrix. The effective stiffness's are calculated by different homogenization methods and the microscopic deformation of a RVE is modeled by the finite element method for the plate with arbitrarily shaped holes. All of the effective stiffness coefficients, especially stretching–shear coupling coefficients are evaluated. **Podruzhin,** and **Ryabchikov, 2004,** studied distribution of bending stresses in anisotropic plates with stress concentrators. Stresses in the vicinity of the tips of defects of the type of a crack or rigid inclusion are determined. The effect of holes and interaction between the defects on the stress intensity factors is analyzed, **Ali,** and **Masood, 2010**. The aim of the work presented in this research is to deal with some of the aspects in the FEM with some of the aspects in the FEM analysis of sandwich panels containing holes which comprised with foam core. In this research, the FEM modeling was produced, analyzed and computed considering laboratory conditions. An extensive parametric study was investigated under different load conditions; different geometrical parameters, such as; dimensions, face thickness, core thickness, size and location of the opening.

2. STATIC ANALYSIS OF SANDWICH PLATE STRUCTURES

2.1 Sandwich Plate Theory

The theory of sandwich plates is based on the following basic hypotheses**, Berthelot, 2010.**:

1. The thickness of the core is much greater than that of the skins: $h \gg h_1, h_2$.
2. The in-plane displacement in the core u_c and v_c in the x and y directions are linear functions of the z coordinate.
3. The in-plane displacements u and v in the x and y directions are uniform through the thickness of the skins.
4. The transverse displacement w is independent of the z coordinate: the strain ε_{zz} is neglected.
5. The core transmits only the transverse shear stresses σ_{xz}, σ_{yz}: the stresses σ_{xx}, σ_{yy}, σ_{xy} and σ_{zz} are neglected in the core.
6. The transverse shear stresses σ_{xz} and σ_{yz} are neglected in the skins. Lastly, the theory considers the elasticity problems of small deformations.

By using these assumptions the governing equations are derived for isotropic symmetric sandwich plates for the in-plane and flexural field Eq.(1) and for transverse shear field Eq.(2):

$$\begin{bmatrix} N_x \\ N_y \\ N_{xy} \\ M_x \\ M_y \\ M_{xy} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{16} & B_{11} & B_{12} & B_{16} \\ A_{12} & A_{22} & A_{26} & B_{12} & B_{22} & B_{26} \\ A_{16} & A_{26} & A_{66} & B_{16} & B_{26} & B_{66} \\ C_{11} & C_{12} & C_{16} & D_{11} & D_{12} & D_{16} \\ C_{12} & C_{22} & C_{26} & D_{12} & D_{22} & D_{26} \\ C_{16} & C_{26} & C_{66} & D_{16} & D_{26} & D_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_{xx}^0 \\ \varepsilon_{yy}^0 \\ \gamma_{xy}^0 \\ \mathcal{K}_x \\ \mathcal{K}_y \\ \mathcal{K}_{xy} \end{bmatrix} \quad (1)$$

$$\begin{bmatrix} Q_y \\ Q_x \end{bmatrix} = \begin{bmatrix} F_{44} & F_{45} \\ F_{45} & F_{55} \end{bmatrix} \begin{bmatrix} \gamma_{yz}^0 \\ \gamma_{xz}^0 \end{bmatrix} \quad (2)$$

In the case of isotropic symmetric sandwich plates (a sandwich plate is isotropic when the core of the sandwich plate is made of an isotropic (such as foam) or transversely isotropic material (such as honeycomb) and the face-sheets are made of identical isotropic materials or quasi-isotropic laminates**, Springer,** and **Kollar, 2003**, hence:

$$A_{ij}^{1} = A_{ij}^{2}, \quad C_{ij}^{1} = -C_{ij}^{2}$$
$$A_{ij} = 2A_{ij}^{2}, \quad D_{ij} = hC_{ij}^{2}$$
$$B_{ij} = C_{ij} = 0 \qquad (3)$$
$$A_{16} = A_{26} = D_{16} = D_{26} = 0$$
$$F_{45} = 0$$

$$A_{ij}^{s} = 2\left(Q_{ij}\right)_{k} h$$
$$D_{ij}^{s} = \frac{1}{2}\left(Q_{ij}\right)_{k} h h_1 (h + h_1) \qquad (4)$$
$$F_{44} = F_{55} = hG^{c}, F_{45} = 0$$

and

$$\left[Q_{ij}^{sk}\right] = \frac{E}{(1-\upsilon^2)}\begin{bmatrix} 1 & \upsilon & 0 \\ \upsilon & 1 & 0 \\ 0 & 0 & \frac{1-\upsilon}{2} \end{bmatrix} \qquad (5)$$

In the case of statics problems, the fundamental equations of sandwich plates are:

$$A_{11}\frac{\partial^2 u_o}{\partial x^2} + A_{66}\frac{\partial^2 u_o}{\partial y^2} + (A_{12} + A_{66})\frac{\partial^2 v_o}{\partial x \partial y} = 0$$
$$(A_{12} + A_{66})\frac{\partial^2 u_o}{\partial x \partial y} + A_{66}\frac{\partial^2 v_o}{\partial x^2} + A_{22}\frac{\partial^2 v_o}{\partial y^2} = 0$$
$$D_{11}\frac{\partial^2 \varphi_x}{\partial x^2} + D_{66}\frac{\partial^2 \varphi_x}{\partial y^2} + D_{12}\frac{\partial^2 \varphi_y}{\partial x \partial y} - F_{55}\left(\frac{\partial w_o}{\partial x} + \varphi_x\right) = 0 \qquad (6)$$
$$(D_{12} + D_{66})\frac{\partial^2 \varphi_x}{\partial x \partial y} + D_{66}\frac{\partial^2 \varphi_y}{\partial x^2} + D_{22}\frac{\partial^2 \varphi_y}{\partial y^2} - F_{44}\left(\frac{\partial w_o}{\partial y} + \varphi_y\right) = 0$$
$$F_{55}\left(\frac{\partial^2 w_o}{\partial x^2} + \frac{\partial \varphi_x}{\partial x}\right) + F_{44}\left(\frac{\partial^2 w_o}{\partial y^2} + \frac{\partial \varphi_y}{\partial y}\right) + q = 0$$

2.2 Tension

A sandwich plate consists of two identical skins made of an isotropic material with thickness h_1 and of an isotropic core with thickness h. The plate is clamped along the edges $x = 0$ and free at $x = a$. This plate is subjected to axial load in x-direction at the free end and there is no coupling between in-plane and flexural behaviors so the stress equation is**, Berthelot, 2010.**

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix}\begin{bmatrix} \varepsilon_{xx}^{0} \\ \varepsilon_{yy}^{0} \\ \gamma_{xy}^{0} \end{bmatrix} \qquad (7)$$

By substituting Eq.(1) in Eq.(7):

$$\begin{bmatrix}\sigma_{xx}\\ \sigma_{yy}\\ \sigma_{xy}\end{bmatrix} = [\boldsymbol{Q}][\boldsymbol{A}^*][\boldsymbol{N}] \tag{8}$$

where the coefficients A^*_{ij} are the components of the inverse matrix of $[A_{ij}]$. In the case of tension along the x direction, the tension and twisting results N_y and N_{xy} are zero:

$$\sigma_{xx} = Q_{11}A^*_{11}N_x \tag{9}$$

where Q_{11} is defined in Eq.(5) and:

$$A^*_{11} = \frac{1}{2*Q_{11}h_1(1-\upsilon^2)} \tag{10}$$

$$N_x = P/b \tag{11}$$

By substituting Eqs.(5, 10 and 11) in Eq.(9):

$$\sigma_{xx} = \frac{1}{2*h_1(1-\upsilon^2)b}P \tag{12}$$

2.3 Bending

The square sandwich plate, see **Fig.2**, having two identical skins constituted of an isotropic material with thickness h_1 and of an isotropic core with thickness h. The plate is simply supported along the edges $x = 0$ and x = a while the other two edges y = a/2 and y = - a/2 may be simply supported. By Levy Solutions, this plate is subjected to the transverse load:

$$p(x) = \sum_{m=1,2,\ldots}^{\infty} p_m \sin\frac{m\pi x}{a} \tag{13}$$

where:

$$p_m = \frac{2}{a}\int_0^x p(x)\, sin\frac{m\pi x}{a}\,dx \tag{14}$$

When the plate is subjected to a line load $p(x) = q$ along $x = \frac{a}{2}$, see **Fig.2**, Eq.(14) will be**, Ansel, 1999.**:

$$p_m = \frac{2q}{a}sin\frac{m\pi}{2} \tag{15}$$

The fundamental bending relations are given by Eq.(6), the coefficients A_{ij}, D_{ij} and F_{ij} being defined by Eq.(4) imply for a symmetric sandwich plate:

$$u_o = 0, \quad v_o = 0 \tag{16}$$

These conditions are satisfied by functions of the form [3]:

$$\varphi_x = A\cos\frac{\pi x}{a}$$
$$\varphi_y = B\sin\frac{\pi x}{a} \tag{17}$$
$$w_o = C\sin\frac{\pi x}{a}$$

By substituting Eq.(16) and Eq.(17) in Eq.(6):

$$B = 0 \Rightarrow \varphi_y = 0 \tag{18}$$

From Eqs.(15, 17 and 18), we derive that the case of bending is cylindrical bending and the deformation state of the sandwich plate is described as**, Berthelot, 2010.**

$$u_o = 0, \qquad v_o = 0,$$
$$\varphi_x = \varphi_x(x), \quad \varphi_y = 0 \tag{19}$$
$$w_o = w_o(x)$$

By substituting Eq.(19) into Eq.(6):

$$hG\left(\frac{d^2 w_o}{dx^2} + \frac{d\varphi_x}{dx}\right) + q = 0 \tag{20a}$$

$$D_{11}\frac{d^2\varphi_x}{dx^2} - hG\left(\frac{dw_o}{dx} + \varphi_x\right) = 0 \tag{20b}$$

By considering the case of a plate simply supported along the edges $x = 0$ and $x = a$:

$$w_o = 0, \quad M_x = 0, \quad \frac{d\varphi_x}{dx} = 0 \tag{21}$$

Integration of Eq.(20a) with respect to x and substituting the result in Eq.(20b), then integration of the final result with respect to x again leads to:

$$\frac{d\varphi_x}{dx} = -\frac{q}{2D_{11}}x^2 + \frac{C}{D_{11}}x + A \tag{22}$$

Associated with condition Eq.(21) for the supports, leads to:

$$\frac{d\varphi_x}{dx} = -\frac{q}{2D_{11}}x(x-a) \tag{23}$$

For the symmetric isotropic sandwich the coefficients and there is no coupling between in-plane and flexural behaviors, then:

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{xy} \end{bmatrix} = \frac{h}{2}\begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix}\begin{bmatrix} \dfrac{\partial\varphi_x}{\partial x} \\ \dfrac{\partial\varphi_y}{\partial y} \\ \dfrac{\partial\varphi_x}{\partial y}+\dfrac{\partial\varphi_y}{\partial x} \end{bmatrix} \tag{24}$$

The substitution of Eq.(23) into Eq.(24) leads to:

$$\sigma_{xx} = \frac{h}{2}Q_{11}\frac{q}{D_{11}}x(x-a) \tag{25}$$

For Q_{11} and D_{11}defined in Eq.(5) and Eq.(4) respectively and:

$$q = \frac{P}{a} \qquad p_m = \frac{2q}{a} \text{ for } m = 1{,}3{,}5 \tag{26}$$

Then Eq.(25) for the maximum bending stress at $x = \frac{a}{2}$ will be:

$$\sigma_{xx} = \frac{P}{4(h+h_1)h_1} \tag{27}$$

2.4 Stress Concentration Factors

The stress concentration factor, listed in **Table 1** *K* can be defined as the ratio of the peak stress in the body (or stress in the perturbed region) to some other stress (or stress like quantity) taken as reference stress:

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \tag{28}$$

where the stresses σ_{max} represent the maximum stresses to be expected in the member under the actual loads and the nominal stresses σ_{nom} is reference normal stress. In the case of the theory of elasticity, a two-dimensional stress distribution of an elastic body under known loads is a function only of the body geometry and is not dependent on the material properties as shown in Eq.(27).
Suppose that the thickness of the plate is *t*, the width of the plate is *b*, and the diameter of the hole is *d*. The reference stress could be defined in two ways**, Walter, 1997.**

1. Use the stress in a cross section far from the circular hole as the reference stress. The area at this section is called the gross cross-sectional area. Thus define:

$$\sigma_{nom} = \frac{P}{ht} \tag{30}$$

so that the stress concentration factor becomes

$$K_{tg} = \frac{\sigma_{max}}{\sigma} = \frac{\sigma_{max} ht}{P} \tag{31}$$

Use the stress based on the cross section at the hole, which is formed by removing the circular hole from the gross cross section. The corresponding area is referred to as the net cross-sectional area. If the stresses at this cross section are uniformly distributed and equal to σ_o:

$$\sigma_n = \frac{P}{(h-d)t} \tag{32}$$

2.2 F.E Static Modeling of Sandwich Plate

Four nodes element (SHELL181) is used to analyze rectangular sandwich plates under tension and buckling loads respectively and square sandwich plates under bending load.

SHELL181 used for layered applications for modeling laminated composite shells or sandwich construction. The accuracy in modeling composite shells is governed by the first order shear deformation theory. To define the thicknesses and materials properties of the three layers of the sandwich plates, section definition can use.

2.2.1 **In plane loads (tension and buckling loads)**

. The sandwich plates are built-in at edge (x=0) and free at edge (x=a, y=0 and y=b) and the in plane loads load is applied at the free end (x=a). the modeling and meshing of sandwich plates under tension and buckling is same, while the solution of each case is different.

The best meshed method for solid plate is that 20 elements along the vertical edges of plate (x=0 and x=a) by interring the element edge length is 5, while 30 elements along horizontal edges of plate (y=0 and y=b) by interring the element edge length is 5, as shown in **Fig.3.**

The best meshed method for plate with central circular hole is that**, Erdogan,** and **Ibrahim, 2006**, as shown in **Fig.4**:

- Draw square area has edge length equal to double of hole diameter **(A_1)**.
- Draw rectangular area represented the plate **(A_2)**, then glue the two areas.
- Draw circular area in the middle **(A_3)**, then subtracting it from the other areas to obtain the finally shape (plate with central hole).
- The outer edge are meshed as in solid plate (lines 1 & 2 have 20 elements while lines 3 & 4 have 30), but the edges of square area (lines 5, 6, 7 and 8) and the curves of circle are meshed by interring the element edge length is 0.1 as shown in**.**

2.2.2 **Bending**

The sandwich plates are simply supported at edges (x=0 and x=a) while free at edge (y=0 and y=a) and loaded by transverse line load at x=a/2.

The solid plate meshed as in tension **Fig.3,** while the plates with central circular hole as shown in **Fig. 5**:

- Draw square area represented the plate **(A_1)**.
- Draw circular area in the middle the square area **(A_2)**, then subtracted it from the square area.
- The vertical edges of plate(x=0 and x=a) have the element edge length is 1 while the horizontal edges of plate(y=0 and y=a) have the element edge length is 5.
- The curves of the circle have the element edge length is 0.1**.**

2.3 Experimental Method for Size Effect-Related Static Analysis

The experimental analysis will be done by several steps:

2.3.1: **Selection basic materials and manufacturing the sandwich plates**

Tensile test used to find the mechanical properties of the basic materials which represented by Low Carbon Steel and AA7075-T6 for face sheets and PVC for core of sandwich plates. The stress-strain curves of the tensile for these materials are shown in **Fig.6** and mechanical properties obtained from them are listed in **Table 2**.

After selecting the basic materials, these materials are cutting to the suitable dimensions depended on the thin plate theory.

Tensile test is done again for three sandwich specimens each one is bonded by different adhesive (Polyester, Epoxy and Titan). The load-deformation curves show that the sandwich specimen bonded by Epoxy adhesive has the highest load as shown in **Fig.7.**

Depending on the results the Epoxy adhesive will be used to bond the sandwich plates.

2.3.2 **Tensile test of the sandwich plates**

Tensile tests are passes in room temperature at maximum load 200KN and 2mm/min for all specimens. The results of the tensile test are the maximum elastic loads from load-deformation curves which using to obtain the theoretical stress by Eq.(12) and in ANSYs program input data and the maximum elastic stresses from stress-strain curves.

2.3.3 **Bending test of the sandwich plates**

The bending tests are passed through the room temperature under maximum load 10KN and speed 3mm/min for all sandwich plate specimens.

The results of the bending test are the maximum elastic loads from load-deformation curves which using to obtain the theoretical stress by Eq.(27) and in ANSYs program input data and the maximum elastic stresses from stress-strain curves.

2.3.4 **Buckling test of the sandwich plates**

The buckling tests are passed through the room temperature under maximum load 10KN and speed 2mm/min for all sandwich plate specimens. The results of the buckling test are the maximum elastic loads from load-deformation curves.

3. RESULRS AND CONCLUSION

The main conclusion from the results of this study is that the hole in the plate is weaking its strength under mechanical loads. The weakness in strength of plate appears as decreasing in nominal stress of sandwich plate because of concentration stresses around it as shown in **Fig.8** for analytical nominal stress of sandwich plates with hole under tension as well as **Fig.9** and **Fig.10** for analytical and experimental nominal stress of sandwich plates with hole under bending

Fig.8 shows that the AA7075-T6/PVC/AA7075-T6 can be had strength more than ST/PVC/ST sandwich plates. The effect of the hole in the plate under tensile makes strength be dropped at (d/b = 0.1), then the curves can be risen at (d/b = 0.15). The behavior can be explained by the increase in diameter of hole may be reduced the stress concentration but did not eliminate the influence. The second drop of the curves can be clarified by that the hole diameter at this point (20 mm) was approximately equal to the half of plate width (100 mm) and that will reduce the stress concentration effect as compared with other hole dimensions.

This discussion can be applied to both of **Fig.9** and **Fig.10,** but it can be noted that the ST/PVC/ST sandwich plate were had strength more than the AA7075-T6/PVCAA7075-T6 plate sandwich. After (d/b = 0.15), this difference in strength between the two sandwich materials was decreased as well as the effect of increasing in a hole size and can be stabled for each materials.

Fig.11 and **Fig.12** were represented the relationship between the analytical and numerical maximum stress of sandwich plates under tensile load respectively. The two figures can be shown two important things. The first thing, the use of (K_{tg}) stress concentration factor with gross nominal stress or (K_{tn}) stress concentration factor can be obtained same results of the maximum stress with maximum difference (5.88%). The second thing, the maximum strength in the hole can be caused the weakness of sandwich plates and beams.

Fig.13, **Fig.14** and **Fig.15** can be showed the relationship between the analytical, numerical and experimental maximum stress of sandwich plate under bending load. In these figures, it can be noted the obvious difference between the two maximum stresses obtained from ($\sigma_{nom}*K_{tg}$) and (σ_n*K_{tn}) for each sandwich materials, because of the studied sandwich plates were square and the effect of the width in Eq.(27) can be canceled.

The numerical values of stress concentration factors can be shown in **Fig.16** and **Fig.17.** From these figures, it can be noted that the curves of K_{tn} have same behavior while the curves of K_{tg} appears different behavior. Where K_{tn} is proved the fact of reducing the stress concentration with increasing of hole size, K_{tg} can be behaved randomly with increasing of hole size.

The buckling load decreased when the hole size is increase because of the hole became region to concentrate the stresses and weaken the plates. ST/PVC/ST sandwich plate is undergoing buckling load more than AA7075-T6/PVC/AA7075-T6 sandwich plate as shown in **Fig.18** and **Fig.19** for numerical and experimental buckling load of sandwich plate respectively.

When the deformation shape modes are discussed, the solid plates have different forth deformation shape mode for the two sandwich materials shown in **Fig.15 a and b**.

While the deformation shape modes are differed and changed for sandwich plates with hole but they remain the same in each sandwich materials as shown in **Fig.16 a and b**. The hole not only causes a decrease in resistance but is changing the deformation shape modes of the sandwich plate since each sandwich materials varies in response the deformation shape modes are differed for each one because of the different in faces materials.

The comparisons between the theoretical, numerical and experimental results are shown in **Fig.17** and **Fig.18.**

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NOMENCLAUTURE

γ_{xy}^{0}: shear strain components of middle-plane in (z) directions respectively.
$\varepsilon_{xx}^{0}, \varepsilon_{xx}^{0}$: strain components of middle-plane in (x, y) directions respectively.
υ: Poisson′s ratio.
$\mathcal{K}_x, \mathcal{K}_y, \mathcal{K}_{xy}$: curvatures components of the middle-plane.
σ_n: net stresss, MPa.
σ_{nom}: nominal stress, MPa.
$\varphi_x, \varphi_{y:}$: rotation of the cross section in the x-y and x-z planes respectively.
$A_{ij}, B_{ij}, C_{ij}, D_{ij}$: extensional stiffness, the coupling stiffness, and the bending stiffness.
A_{ij}^{*}: inverse extensional stiffness.
a: length of plate , mm.
b: beam and plate width, mm.
D_{ij}^{*}: inverse extensional stiffness.
E_f, E_c: Young modulus of skin and core respectively, Pa.
F^c: transverse shear stiffness.
G_f, G_c: shear modulus of skin and core respectively, Pa.
h: thickness of core, mm.
h_1, h_2: thickness of lower and upper skins respectively, mm.
h_t: total thickness of each plate, mm.
K_{tn}, K_{tg}: stress concentration factors.
M_x, M_y, M_{xy}: bending and twisting moments $N.mm$.
m: no. of half wavelengths in x and y directions.
N_x, N_y, N_{xy}: in − plane force resultant, N.
P: external applied load, N.
p_m: the transverse load coefficient of Levy Solutions.
Q_{ij}: transformed stiffness.
Q_x, Q_y: transverse shear resultants.
q: external load.
u_o, v_o, w_o: middle-plane displacement components along (x, y and z) directions respectively.
x: distance in x − direction, mm.
z: distance from neutral axis in $z_{\text{direction}}$.

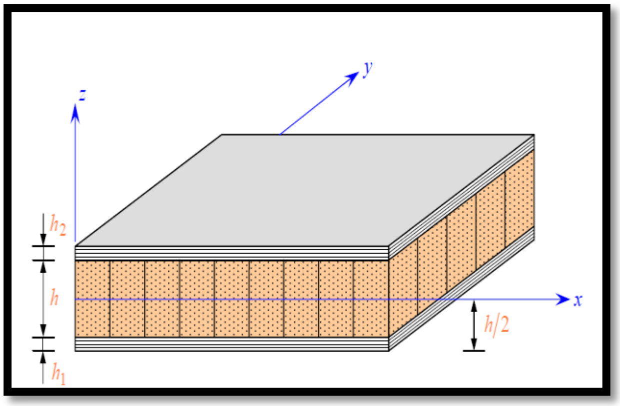


Figure 1. Sandwich materials.

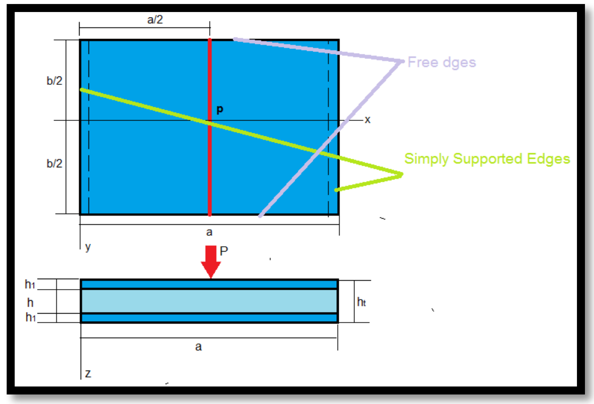


Figure 2. Sandwich plate under bending load.

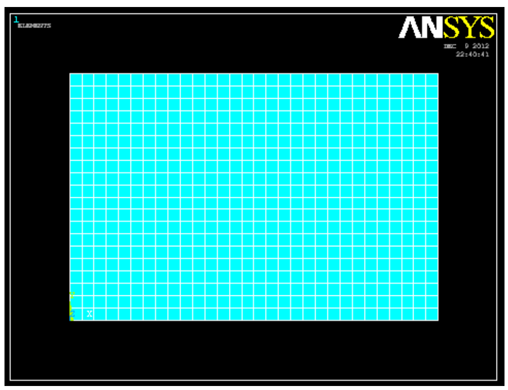


Figure 3. Mesh of solid sandwich plate.

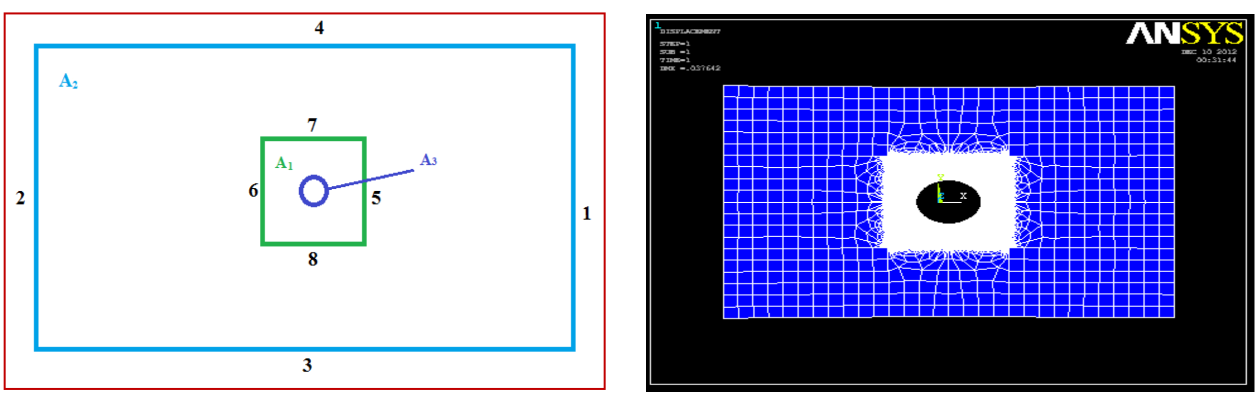


Figure 4. Mesh of sandwich plate with hole under tension and buckling.

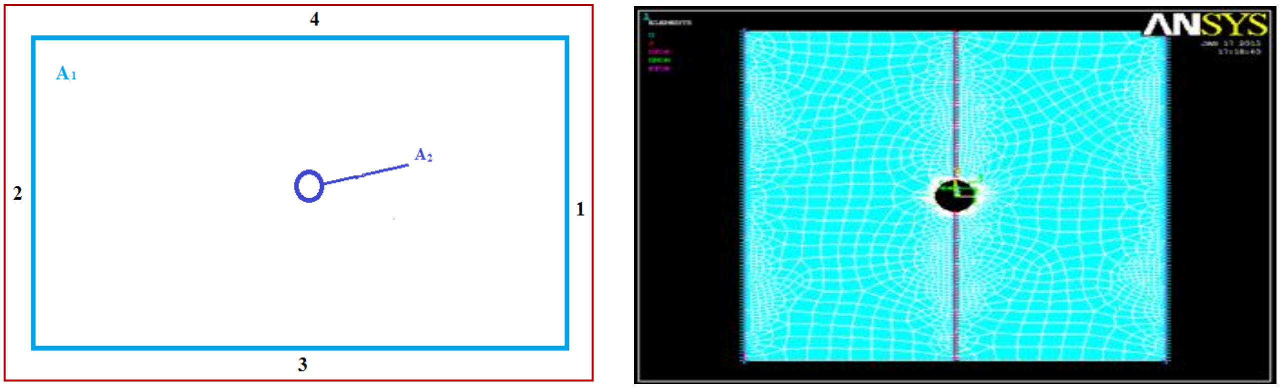
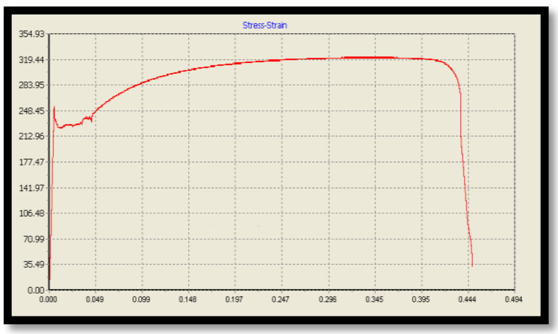
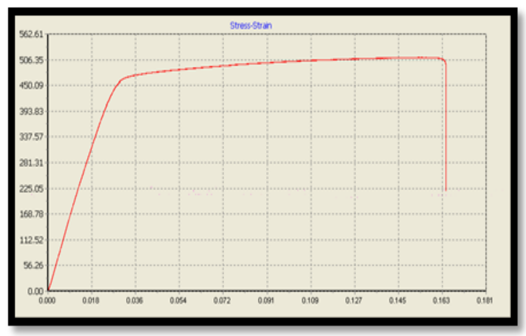


Figure 5. Mesh of sandwich plate with hole under bending.



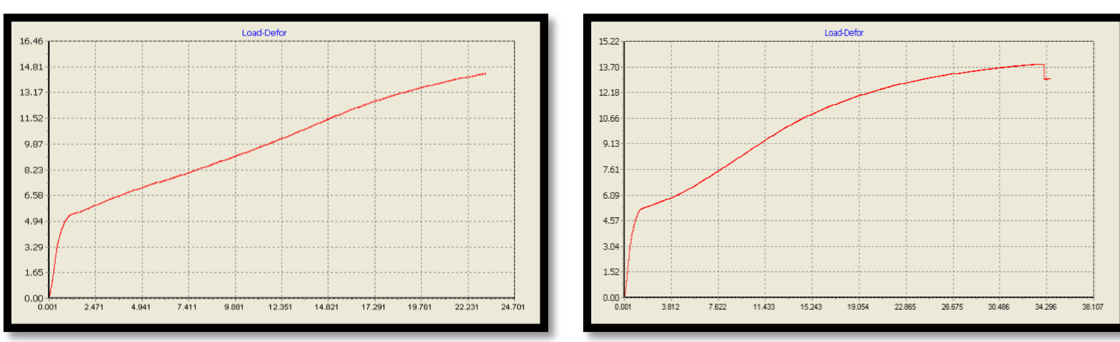
(a)



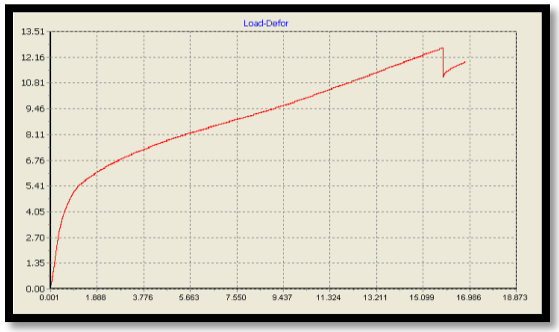
(b)

(c)

Figure 6. Tensile test curves: (a) Steel, (b) AA7075-T6, (c) PVC.



(a) Epoxy (b) Polyester



(c) Titan

Figure 7a. Tensile test curve of specimens to adhesive selection.

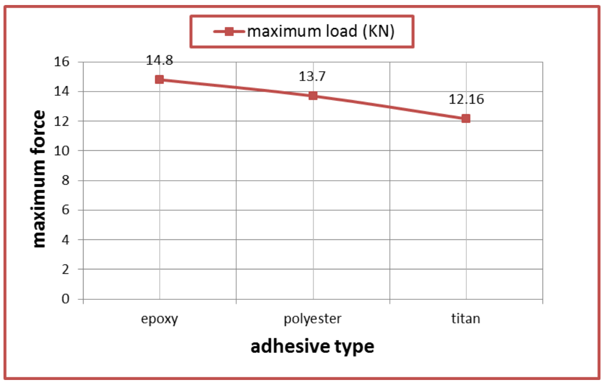


Figure 7b. Maximum load (KN) of adhesive selected Specimens tensile test.

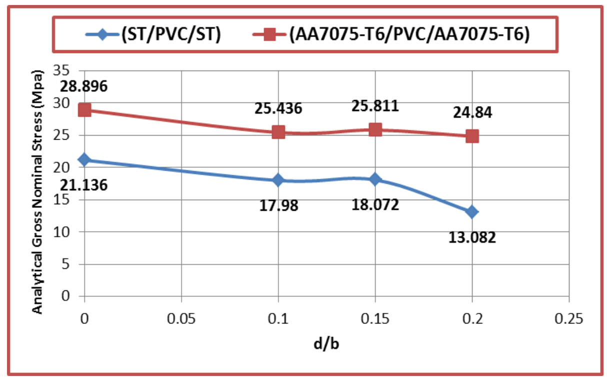


Figure 8. Analytical elastic nominal maximum stress of sandwich plates subjected to tensile load.

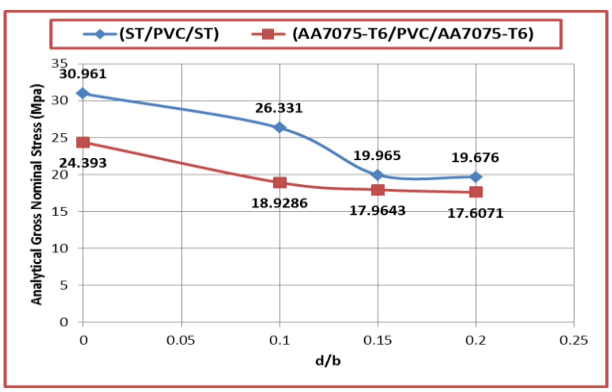


Figure 9. Analytical elastic nominal maximum stress of sandwich plates subjected to bending load.

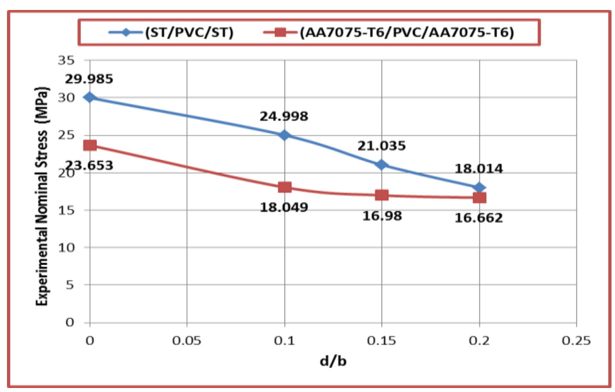


Figure 10. Experimental nominal stress of sandwich plates under bending load.

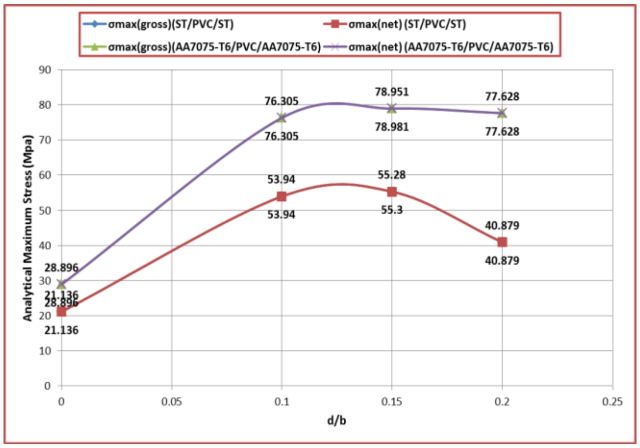


Figure 11. Analytical maximum stresses of sandwich plates under tensile load.

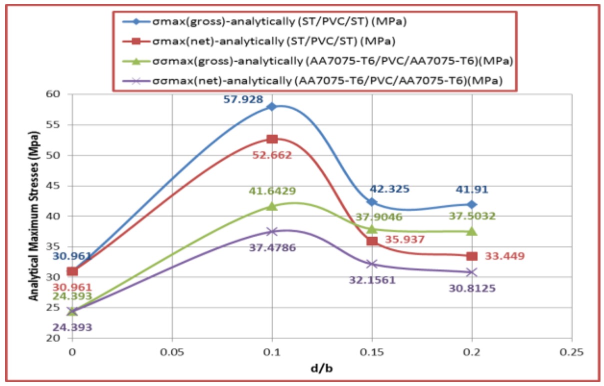


Figure 13. Analytical maximum stresses of sandwich plates under bending load.

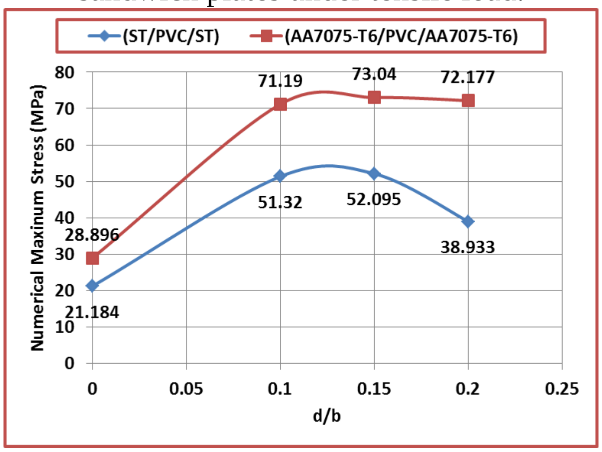


Figure 12. Numerical maximum stress of sandwich plates under tensile load.

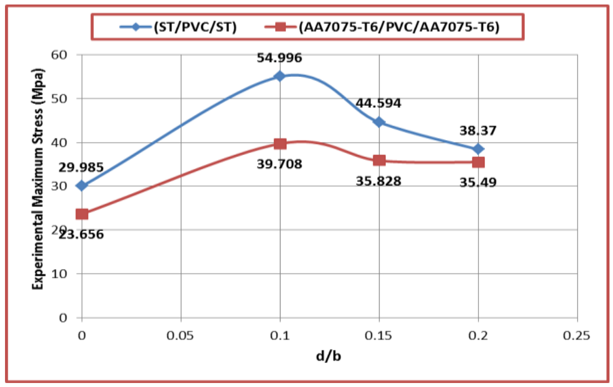


Figure 14. Experimental maximum stresses of sandwich plates under bending load.

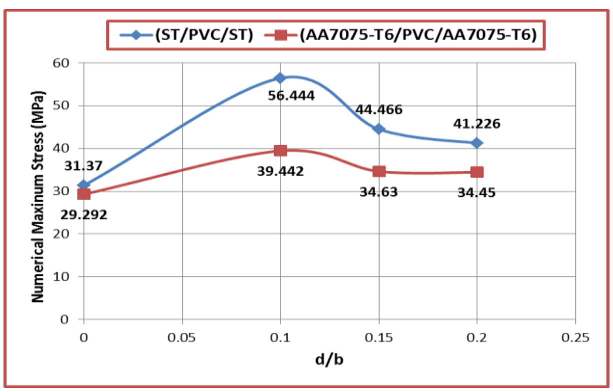


Figure 15. Numerical maximum stress of sandwich plates under bending load.

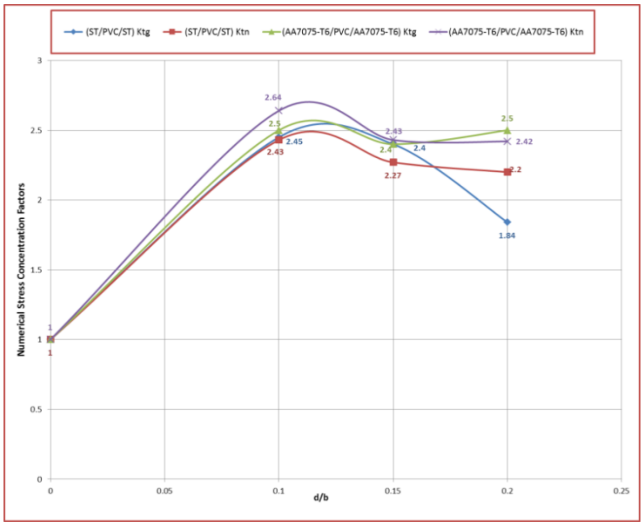


Figure 16. Numerical stress concentration factors of sandwich plates under tensile load.

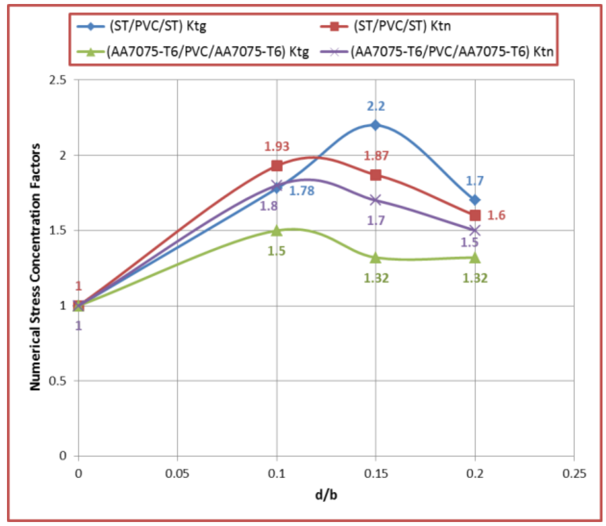


Figure 17. Numerical stress concentration factors of sandwich plates under bending load.

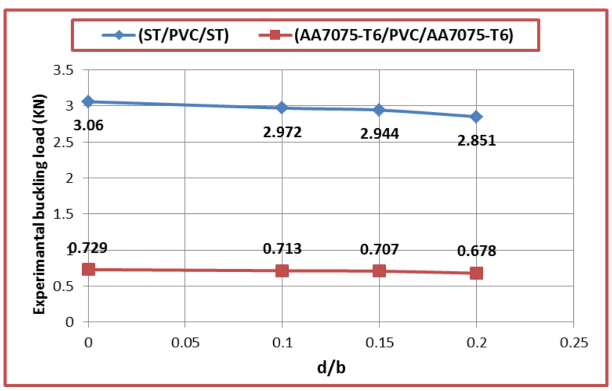


Figure 18. Experimental buckling load of sandwich plates under compression load.

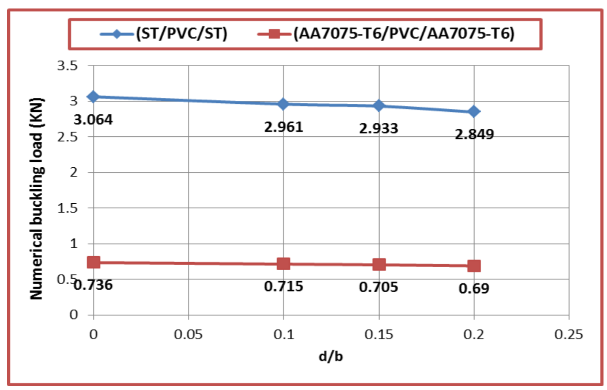
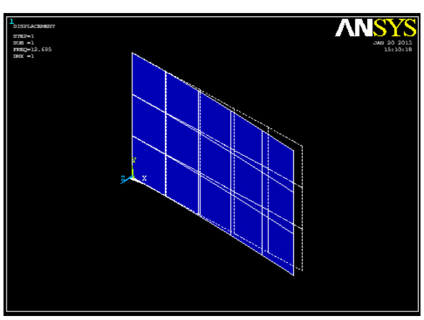
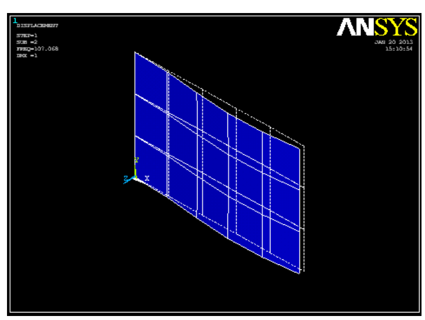


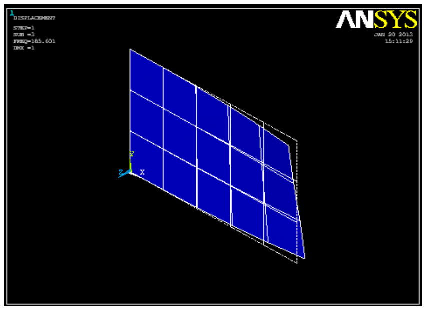
Figure 19. Numerical buckling load of sandwich plates subjected to compression.



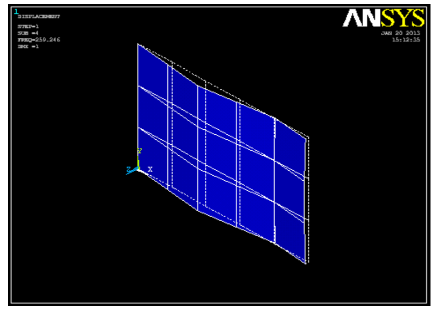
First mode



Second mode

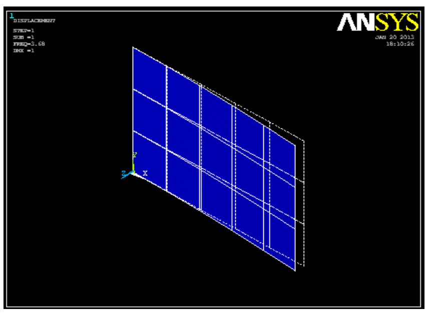


Third mode

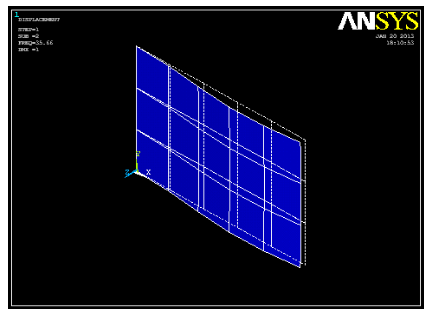


Fourth mode

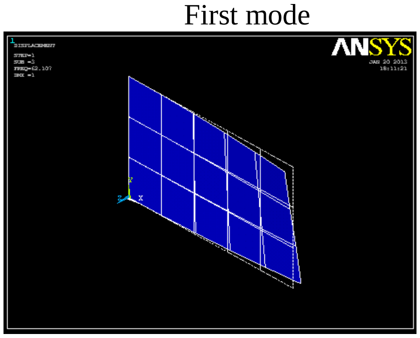
Figure 20a. Deformation shape modes of ST/PVC/ST solid sandwich plates under buckling.



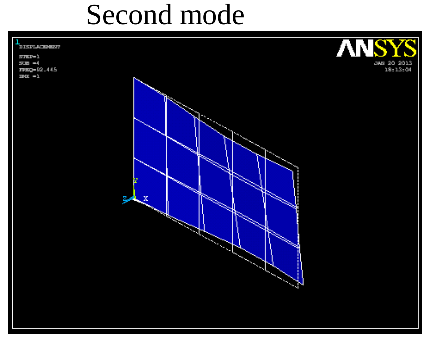
First mode



Second mode

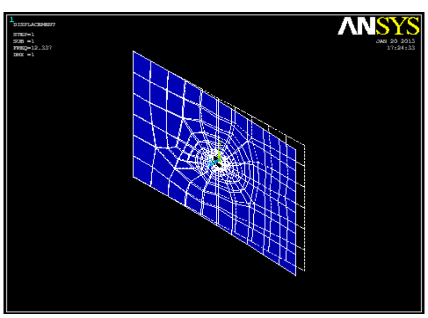


Third mode

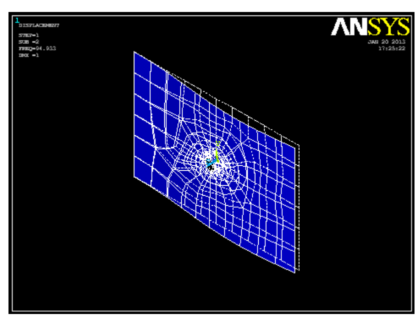


Fourth mode

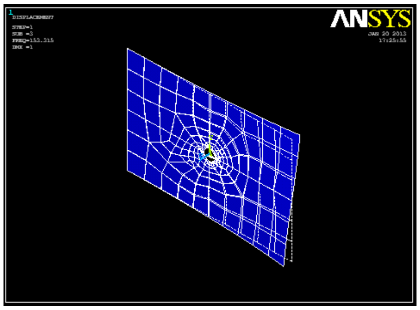
Figure 20b. Deformation shape modes of AA7075-T6/PVC/AA7075-T6 solid sandwich plate under buckling.

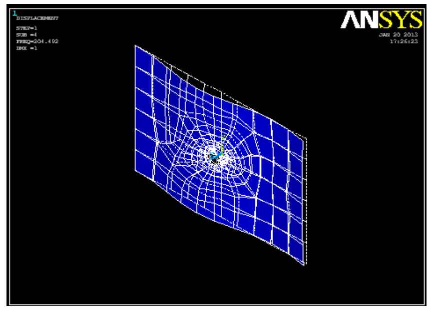


First mode



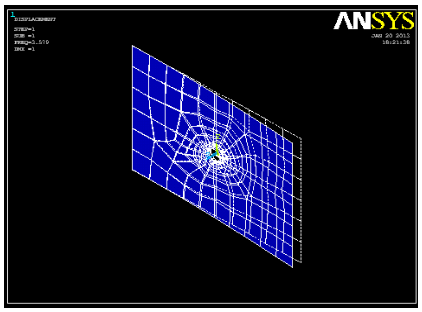
Second mode

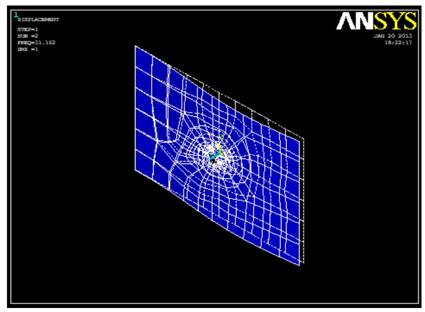




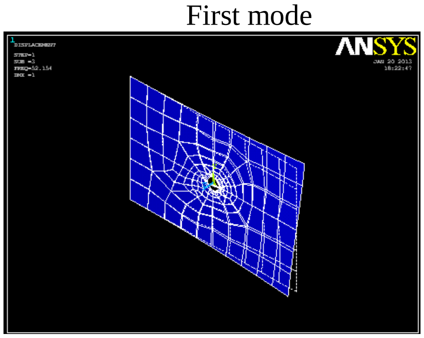
Third mode Fourth mode

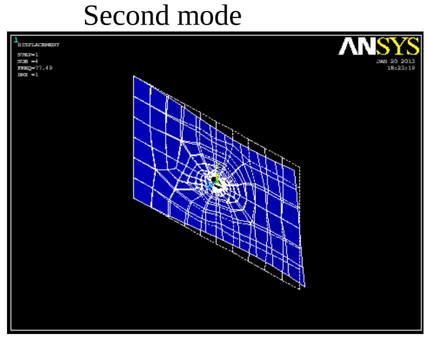
Figure 21a. Deformation shape modes of ST/PVC/ST sandwich plate with hole under buckling.





First mode Second mode





Third mode Fourth mode

Figure 21b. Deformation shape modes of AA7075-T6/PVC/AA7075-T6 sandwich plate with hole under buckling.

Table 1. Stress concentration factors for beams and plates under tension and bending loads.

Tension				
ST/PVC/ST PLATES and AA7075-T6/PVC/AA7075-T6 PLATES	**D**	**10**	**15**	**20**
	K_{tg}	**3**	**3.06**	**3.125**
	K_{tn}	**2.7**	**2.6**	**2.5**
Bending				
ST/PVC/ST PLATES	**D**	**10**	**15**	**20**
	K_{tg}	**2.2**	**2.12**	**2.13**
	K_{tn}	**2**	**1.8**	**1.7**
AA7075-T6/PVC/AA7075-T6 PLATES	**d**	**10**	**15**	**20**
	K_{tg}	**2.2**	**2.11**	**2.13**
	K_{tn}	**1.98**	**1.79**	**1.79**

Table 2. Mechanical properties of constitutions materials.

Mechanical Properties	Steel	AA7075-T6	PCV
Young Modulus(GPa)	212	72	4
Yield Stress(MPa)	255.128	447.84	7.746
Maximum Stress(MPa)	325.118	515.352	11.93
Possin's Ratio	0.3	0.33	-