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# Numerical simulation for the honeycomb core sandwich panels in bending by homogenization method

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**Abstract - Nowadays, with the continuous development of science and technology, computer software has been widely applied and is increasingly popular in many fields such as the automobile, aviation, space, and shipbuilding industries. Numerical simulation is an important step in finite element analysis and product design optimization. However, it is facing challenges of reducing CAD model building time and reducing computation time. In this study, we have developed a homogenization model for the honeycomb core sandwich plate to reduce the preparation of the CAD model as well as the computational times. The homogenization consists of representing an equivalent homogenized 3D-solid obtained from the analysis calculation in-plane properties of honeycomb 3D-shell core sandwich plate. This model was implemented in the finite element software Abaqus. The simulations of tensile, in-plane shear, pure bending, and flexion tests for the case of the 3D-shell and 3D-solid models of the honeycomb core sandwich will be studied in this paper. Comparing the results obtained from the two models shows that the 3D-solid model has close results as the 3D-shell model, but the computation time is much faster. Thereby the proposed model is validated.**

**Keywords: Analytic homogenization, composite sandwich, orthotropic sandwich plates, honeycomb core**

## I. INTRODUCTION

IN industry, reducing the weight of structures is one of the most important challenges. Sandwich is a material that can fulfill this requirement. Because it has a high stiffness-to-weight ratio, so it is widely used in several fields, such as automobile, navy, construction, aviation. The main feature of the sandwich structure is the type of cellular material, which is composed of: The core is the same hexagonal cells, regular repeating and cyclic. It provides shear resistance and rigidity; The shell is two thin and hard plates that serve as the main load-bearing unit (*Figure 1*). To create different sandwich structures, various core shapes have been applied such as solid, foam, truss, web, and honeycomb core. As design engineers in the automotive industry need to reduce fuel consumption and improve safety, composite sandwich structures have become and are becoming an attractive alternative to metal. In the vehicle, reducing the weight of a large structure has a positive effect on other parts. Therefore, the use of a composite sandwich structure helps to reduce weight, improve fuel economy, and increase load capacity. In addition, it also allows the design of aerodynamic, stable vehicles with a low center of gravity [1].

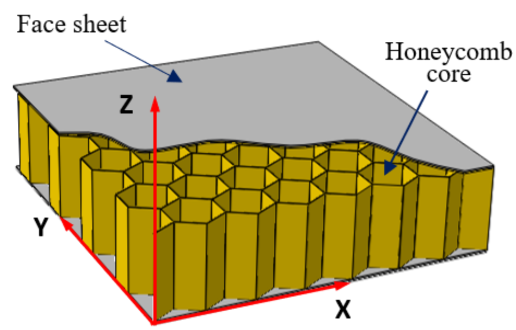


*Fig. 1 Honeycomb core sandwich plate model*

The optimal design of sandwich structures is the goal that must be achieved. Hence, in the same design process, we faced both the difficulty of designing a laminated sheet and the difficulty of designing a complex cellular continuum such as a honeycomb core. Some assumptions or simplified rules are used to achieve, in an easier and faster way [2]. Recent advances in materials and manufacturing techniques have resulted in further improvements and the increased uniformity of the properties of sandwich composites. The optimal design of the sandwich has the help of a computer. However, the generation of CAD models and finite element analyses for the details of these structures remains a major challenge due to the high computational complexity and time. Up to now, many researchers have tried to use different equivalence modeling approaches or homogeneity modeling to overcome this difficulty [3-19]. However, with specific characteristics of geometry and complex conditions in a honeycomb structure, analytical methods are always an effective homogenization method. It is proved by a homogenization model based on the integral method, which has been successfully applied to complete structures with complex cores such as corrugated board boxes [20,21]. Besides, some relevant studies can be found in [22,23].

In this work, we propose a homogenization model to numerically simulate the mechanical behavior of sandwich panels. The homogenization is performed by calculating the in-plane properties for the 3D-shell structure of the honeycomb core and then it is replaced by an equivalent homogeneous 3D solid core. In this paper, the simulations in the case of the Abaqus 3D-solid and 3D-shell model for the honeycomb core sandwich will be performed. The results showed that this 3D-solid homogenization model has close results comparing to the 3D-shell model, while the calculation time and model preparation time are much faster.

## II. METHODOLOGY

To calculate the in-plane properties for honeycomb core sandwich plates, we must use a Representative Volume Elemental (RVE) of the material as shown in Fig.2. In this study, the homogenization method involves the use of a homogeneous fictional material with equivalent macroscopic properties in place of heterogeneous real material. Whereby, it must be large enough for the size of heterogeneity to represent the material and must be the same between regions; It must be small enough for the size of the structure that we can consider as uniform stress or deformation state. The results of homogenization on this RVE will represent the behavior of the whole plate.

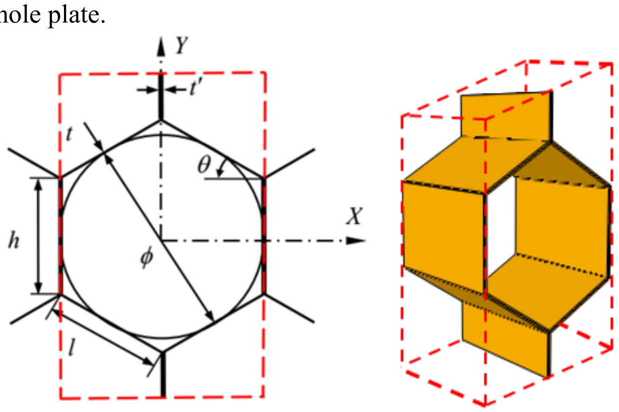


*Fig. 2. Representative volume elemental (RVE) for honeycomb core*

For honeycomb core sandwich plates, by calculating displacements in the x and y directions, we can calculate elastic modules $E_x$ and $E_y$. The traction properties are determined on a single honeycomb cell without the effect of the skins and the properties depend only on the bending behavior of the honeycomb walls [3]. On the assumption that the skins are very hard relative to the honeycomb core walls, the deformation of the honeycomb walls will be determined by the skins [12]. Therefore, in the present study, the effect of traction or compression of thin walls dominates their bending effect. As a result, for a regular hexagonal honeycomb ($t' = 2t$ and $h = l$), the module's Young is quite proportional to ($t/l$) (tensile wall) instead of $(t/l)^3$ (bending wall, in [3]). If the height of the honeycomb core is very small or if we cut the honeycomb core close to the hard skins, then the honeycomb core will deform like the skins (same ν) and it will only behave traction and thus bending, as well as the effect of bending can be ignored, because:

$$\kappa \frac{t}{l} \gg \left(\frac{t}{l}\right)^3 \tag{1}$$

For example, in the case of $t$ = 0.19 mm, $l$ = 4.62 mm, this ratio is 600 times. Based on classical homogeneity theory and related research results [3-14,24], calculations are performed as follows:

*1. Young's modulus $E_x$*

We establish the equation of internal force balance on five walls of EA, AC, CB, CD, and DF from considering the model constructing for slices located far away from the two skins. Perform a displacement at the center of the core from the h/2 to the end position for points A, C, and D, we have an equivalent structure as shown in Fig. 3. The problem becomes as follows: A and B are fixed on the skin; what force must be applied at D to get the displacement $u_0 = 1$? We have:

$$\varepsilon_x = \frac{u_0}{2l\cos\theta} \quad ; \quad \varepsilon_y = \frac{v_0}{h + l\sin\theta} = -\nu\varepsilon_x = -\nu\frac{u_0}{2l\cos\theta} \tag{2}$$

$$\Rightarrow v_0 = -\nu u_0 \frac{h + l\sin\theta}{2l\cos\theta} \tag{3}$$

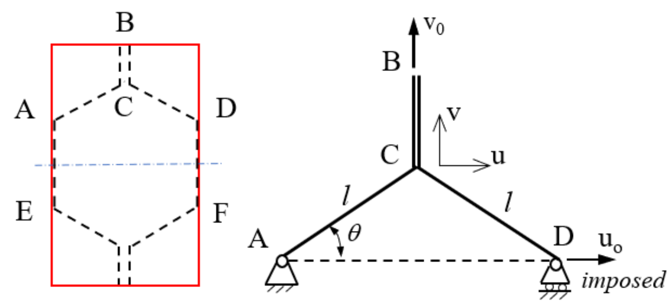


*Fig. 3. Model for calculating elastic modulus $E_x$ for a REV of honeycomb core*

$$\varepsilon_l = \frac{1}{l}\left(u\cos\theta + v\sin\theta\right) \tag{4}$$

$$N_l = \sigma_l .b.t = \frac{Ebt}{l}\left(u\cos\theta + v\sin\theta\right) \tag{5}$$

$$N_h = E\frac{v_0 - v}{h}.2.bt \tag{6}$$

Balancing of node at C, we have:

$$N_h = 2N_l \sin\theta \tag{7}$$

From (5), (6) and (7), we have:

$$\frac{2\,Ebt}{h}\left(v_0 - v\right) = 2\sin\theta\frac{Ebt}{l}\left(u\cos\theta + v\sin\theta\right) \tag{8}$$

$$\Rightarrow l\left(v_0 - v\right) = h\sin\theta\left(u\cos\theta + v\sin\theta\right) \tag{9}$$

On the other hand:

$$u = \frac{1}{2}u_0 \quad ; \quad v_0 = -\nu u_0 \frac{h + l\sin\theta}{2l\cos\theta} \tag{10}$$

$$\Rightarrow v = \frac{v_0 - \frac{h}{4l}\sin^2\theta.u_0}{1 + \frac{h}{l}\sin^2\theta} \tag{11}$$

We have:

$$P_x = N_l \cos\theta = \frac{Ebt}{l}\left(\mathrm{u}\cos\theta + \mathrm{v}\sin\theta\right)\cos\theta \tag{12}$$

$$E_x^* = \frac{\sigma_x^*}{\varepsilon_x^*} = \frac{P_x}{\left(h + l\sin\theta\right)b}.\frac{2l\cos\theta}{\mathrm{u}_0} \tag{13}$$

From Eq. (12) and (13), we have:

$$E_x^* = \frac{2E.b.t\left(\mathrm{u}\cos\theta + \mathrm{v}\sin\theta\right).\cos^2\theta}{\left(h + l\sin\theta\right)b.u} \tag{14}$$

*2. Young's modulus $E_y$*

For traction, we impose a displacement along the y-direction $\mathrm{v}_0 = 1$, we have:

$$\varepsilon_y = \frac{\mathrm{v}_0}{h + l\sin\theta} \quad ; \quad \varepsilon_x = \frac{\mathrm{u}_0}{2l\cos\theta} = -\nu\varepsilon_y = -\nu\frac{\mathrm{v}_0}{h + l\sin\theta} \tag{15}$$

$$\Rightarrow \mathrm{u}_0 = -\nu.\mathrm{v}_0\frac{2l\cos\theta}{h + l\sin\theta} \tag{16}$$

$$\varepsilon_l = \frac{1}{l}\left(\mathrm{u}\cos\theta + \mathrm{v}\sin\theta\right) \tag{17}$$

From Eq. (8), we have:

$$\mathrm{v} = \frac{\mathrm{v}_0 - \frac{h}{2l}\sin\theta\cos\theta.\mathrm{u}_0}{1 + \frac{h}{l}\sin^2\theta} \quad ; \quad \mathrm{u} = \frac{1}{2}\mathrm{u}_0 \tag{18}$$

On the other hand, we have:

$$P_y = N_h = \frac{2\,\mathrm{Ebt}}{h}\left(\mathrm{v}_0 - \mathrm{v}\right) \tag{19}$$

$$E_y^* = \frac{\sigma_y^*}{\varepsilon_y^*} = \frac{P_y}{\left(2l\cos\theta\right)b}.\frac{h + l\sin\theta}{\mathrm{v}_0} \tag{20}$$

From Eq. (19) and (20), we have:

$$E_y^* = \frac{\sigma_y^*}{\varepsilon_y^*} = \frac{E.2tb\left(\mathrm{v}_0 - \mathrm{v}\right)}{\left(h.2l\cos\theta\right)b}.\frac{h + l\sin\theta}{\mathrm{v}_0} \tag{21}$$

For sandwich plates with very small core heights, we will use the Poisson's ratio like the Poisson's ratio of two skins. However, if the plate has a large core height, we can use Gibson's formula to calculate the Poisson's ratio [12]:

$$\nu_{yx} = \frac{\left(\frac{h}{l} + \sin\theta\right)\sin\theta}{\cos^2\theta} \tag{22}$$

$$\nu_{xy} = \frac{\cos^2\theta}{\left(\frac{h}{l} + \sin\theta\right)\sin\theta} \tag{23}$$

*3. Shear modulus in the plane $G_{xy}$*

For in-plane shear, we impose a shear angle γ, so tensile and compression deformation will occur in two inclined walls, we have:

$$\gamma = \frac{\mathrm{u}_1}{\frac{h}{2}} = \frac{\mathrm{u}_2}{\frac{h}{2} + l\sin\theta} \tag{24}$$

$$\Rightarrow \varepsilon_s = \frac{\left(\mathrm{u}_2 - \mathrm{u}_1\right)\cos\theta}{l} = \frac{l\sin\theta.\gamma.\cos\theta}{l} \tag{25}$$

On the other hand, we have:

$$\sigma_s = E_s.\varepsilon_s = E_s.\gamma.\sin\theta\cos\theta \quad \& \quad T_x = \sigma_s.2.\mathrm{t.b}.\cos\theta \tag{26}$$

For a homogeneous solid core, we have:

$$\tau_{xy}^* = G_{xy}^*\gamma \Rightarrow G_{xy}^* = \frac{\tau_{xy}^*}{\gamma} = \frac{T_x}{2l\cos\theta.b.\gamma} \tag{27}$$

Replace (26) with (27) we have:

$$G_{xy}^* = E_s\left(\frac{t}{l}\right)\sin\theta\cos\theta \tag{28}$$

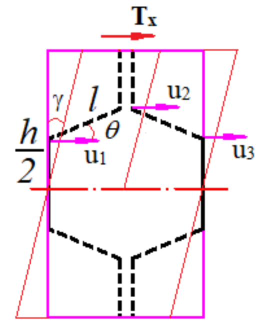


*Fig. 4. Calculation model of shear modulus $G_{xy}$ for a REV of honeycomb core*

*4. Young's modulus $E_z$*

The elastic modulus $E_z$ is calculated by multiplying the modulus $E_s$ of the honeycomb core by the ratio "the cross-sectional area of the honeycomb to the total surface area of the REV figure" [1]:

$$E_z^* = E_t\frac{4ht + 4ht}{\left(2h + 2l\sin\theta\right)2l\cos\theta} = E_t\left(\frac{t}{l}\right)\frac{h + l}{\left(h + l\sin\theta\right)\cos\theta} \tag{29}$$

Poisson's coefficients $\nu_{zx}$ and $\nu_{zy}$ are assumed to be equal to the Poisson's ratio $\nu_{12}$ of the paper layer forming the honeycomb core, that is:

$$\nu_{zx} = \nu_{zy} = \nu_{12-core} \tag{30}$$

The reciprocal relationship allows to determine the remaining 2 Poisson's ratio:

$$\nu_{xz} = \frac{E_x}{E_z}\nu_{zx} \tag{31}$$

$$\nu_{yz} = \frac{E_y}{E_z}\nu_{zy} \tag{32}$$

*5. Shear modulus out of the plane $G_{xz}$*

Considering the cross-section (out-of-plane) model as shown (Figure 5), we impose a shearing angle γ, so shear strain will occur in 4 inclined walls, we have:

$$\tau_s = \frac{\mathrm{P}_z}{t.b} = G_s\gamma_s = G_s\frac{\mathrm{w}_A}{2l} \tag{33}$$

$$\Rightarrow \mathrm{w}_A = \frac{\mathrm{P}_z 2l}{G_s t.b} \tag{34}$$

For a homogeneous solid core, we have:

$$\tau_{xz}^* = G_{xz}^*\gamma = G_{xy}^*\frac{\mathrm{w}_A}{2l\cos\theta} \tag{35}$$

On the other hand, we have:

$$\tau_{xz}^{*} = \frac{P_z}{(h + l\sin\theta).b} \tag{36}$$

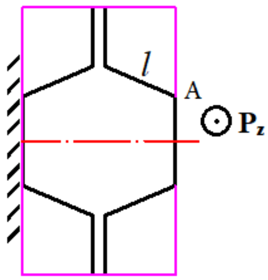


*Fig. 5. Calculation model of shear modulus $G_{xz}$ for a REV of honeycomb core*

From Eq. (34), (35) and (36), we have:

$$G_{xz}^{*} = G_s\left(\frac{t}{l}\right)\frac{\cos\theta}{\left(\frac{h}{l} + \sin\theta\right)} \tag{37}$$

*6. Shear modulus out of the plane $G_{yz}$*

Considering the cross-sectional (out-of-plane) model as shown (Figure 6), in this model we only need to consider a half REV is enough. We impose a shear angle γ, so shear strain will occur in all walls. For 4 vertical walls in the y direction (length h/2), we have:

$$\tau_s^h = \frac{P_z}{2t.b} = G_s\gamma_s^h = G_s\frac{\Delta w_3}{h/2} \tag{38}$$

$$\Rightarrow \Delta w_3 = \frac{P_z}{2t.b}.\frac{h}{2G_s} = \Delta w_1 \tag{39}$$

For two inclined walls (length $l$), we have:

$$\tau_s^l = \frac{P_z}{2t.b} = G_s\gamma_s^l = G_s\frac{\Delta w_2}{l} \tag{40}$$

$$\Rightarrow \Delta w_2 = \frac{P_z.l}{2t.b.G_s} \tag{41}$$

Thus, we have the total displacement is:

$$w = 2\Delta w_3 + \Delta w_2 = \frac{P_z.(h+l)}{2t.b.G_s} \tag{42}$$

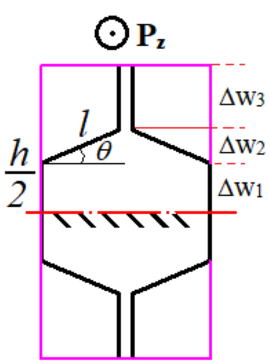


*Fig. 6. Calculation model of shear modulus $G_{yz}$ for a REV of honeycomb core*

For a homogeneous solid core, we have:

$$\tau_{yz}^{*} = G_{yz}^{*}\gamma = G_{yz}^{*}\frac{w}{h + l\sin\theta} \tag{43}$$

On the other hand, we have:

$$\tau_{yz}^{*} = \frac{P_z}{2l\cos\theta} \tag{44}$$

From Eq. (42), (43) and (44), we have:

$$G_{yz}^{*} = G_s\left(\frac{t}{l}\right)\frac{\left(\frac{h}{l}+1\right)}{\left(\frac{h}{l} + \sin\theta\right)\cos\theta} \tag{45}$$

## III. NUMERICAL VALIDATION OF HOMOGENIZATION MODEL

To validate the proposed homogenization 3D-Solid model, the tests finite element simulations are performed with the 3D-shell structure of the sandwich plate and the homogenized 3D-solid sandwich plate. According, the dimensions of the two plates are the same (the length L = 176 mm and width B = 222 mm). To better apply force, a rigid plate is attached to the edge opposite the fixed edge of the honeycomb sandwich plate as shown in Figure 7. The skins of the sandwich are made from unidirectional non-woven linen and combined with Acrodur® resin forming a multi-layer plate (includes three layers oriented 0°, 90° and 0°). The mechanical properties are given in Table 1 for the skin and in Table 2 for the core. For the homogenized 3D-solid sandwich plate, the mechanical properties of the core are calculated by using a homogeneous model as shown in Table 3.

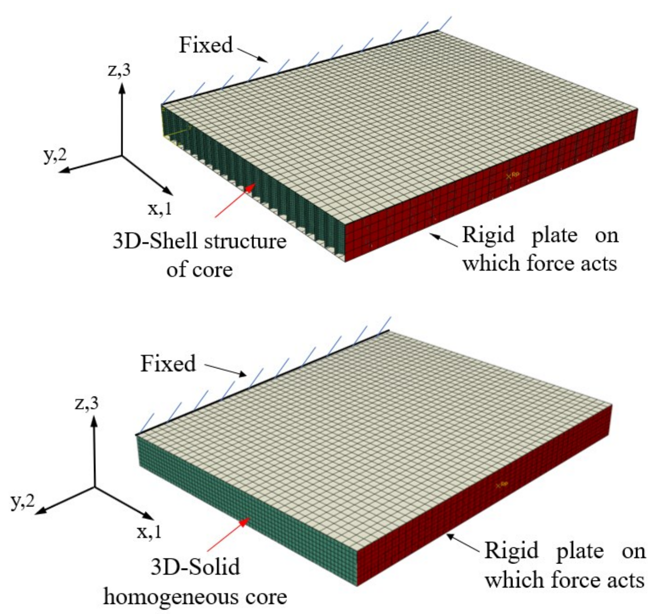


*Fig.7. Boundary conditions of 3D-shell plate and 3D-solid plate used for the simulations.*

*Table 1. Parameters of the layers forming the skins of honeycomb core sandwich plate.*

| $E_1$ (MPa) | $E_2$ (MPa) | $\nu_{12}$ | $G_{12}$ (MPa) | $G_{13}$ (MPa) | $G_{23}$ (MPa) | Thickness (mm) |
|---|---|---|---|---|---|---|
| 18000 | 2000 | 0.4 | 8500 | 10 | 10 | 0.2 |

*Table 2. Parameters of the paper forming the honeycomb core of sandwich plate.*

| $E_1$ (MPa) | $E_2$ (MPa) | $\nu_{12}$ | $G_{12}$ (MPa) | $G_{13}$ (MPa) | $G_{23}$ (MPa) | Thickness (mm) |
|---|---|---|---|---|---|---|
| 3292 | 1594 | 0.42 | 788 | 10 | 10 | 0.19 |

*Table 3. Parameters of the paper forming the honeycomb core of sandwich plate*

| *ϕ (mm)* | *θ (°)* | *l=h (mm)* | *t (mm)* | *t' (mm)* | *Height (mm)* |
|---|---|---|---|---|---|
| 8 | 30 | 4.62 | 0.19 | 0.38 | 20 |

The skins of the 3D-shell plate and the 3D-solid plate mesh with reduced-integration four-node shell elements (S4R) with 1540 elements and 1620 nodes. For the core of the 3D-shell plate, we meshed with 212340 elements S4R and 208992 nodes. We meshed by 6336 solid elements C3D8R with 8325 nodes for the solid core of the homogenization model. Several simulations were carried out with different types of loading such as tensile, in-plane shear, pure bending, and flexion. The comparison of the obtained simulation results allows evaluating the efficiency and accuracy of the proposed homogenous model. The obtained results are shown in Table 4 and Figure 8-11. We can see that the calculations using the 3D-solid model are very fast, while the 3D-shell calculations take a lot of time. The relative difference between the 3D-shell model and the 3D-solid model is less than 2% for the displacement along direction x and direction y. However, the CPU time is reduced by more than 30% for the 3D-solid model. Therefore, the 3D-solid model can be used to replace the 3D-shell model in computation by finite element simulation.

*Table 4. Comparison between Abaqus-3D and H-2D-Model for the plate under traction, flexion, and in-plane shear loading*

| | | | *3D-Shell Model* | *3D-Solid Model* | *Error (%)* |
|---|---|---|---|---|---|
| *Traction F = 200 kN* | *Along x* | *Displacement $U_1$* (mm) | 10.18 | 10.12 | +0.59 |
| | | *CPU time (s)* | 399.7 | 12.7 | 31.47 times |
| | *Along y* | *Displacement $U_2$* (mm) | 27.25 | 26.96 | +1.06 |
| | | *CPU time (s)* | 470.4 | 13.8 | 34.09 times |
| *In-plane shear F = 10 kN* | *Along x* | *Displacement $U_1$* (mm) | 3.603 | 3.577 | +0.72 |
| | | *CPU time (s)* | 364.6 | 12.7 | 28.71 times |
| | *Along y* | *Displacement $U_2$* (mm) | 1.215 | 1.211 | +0.33 |
| | | *CPU time (s)* | 348.3 | 13.1 | 26.59 times |
| *Pure bending M = 200 kN.mm* | *Along x* | *Displacement $U_3$* (mm) | 9.022 | 9.018 | +0.04 |
| | | *CPU time (s)* | 451.1 | 13.2 | 34.17 times |
| | *Along y* | *Displacement $U_3$* (mm) | 30.84 | 30.37 | +1.52 |
| | | *CPU time (s)* | 380.5 | 12.7 | 29.96 times |
| *Flexion + bending F = 1 kN* | *Along x* | *Displacement $U_3$ (mm)* | 7.433 | 7.441 | -0.11 |
| | | *CPU time (s)* | 388.9 | 13.2 | 29.46 times |
| | *Along y* | *Displacement $U_3$ (mm)* | 25.07 | 24.63 | +1.76 |
| | | *CPU time (s)* | 390.7 | 13.8 | 28.31 times |

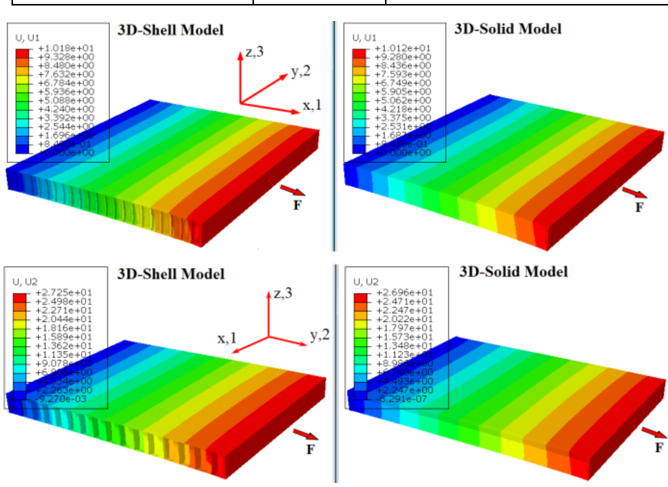


*Fig.8. Simulation of 3D-Shell model and 3D-Solid model in traction for the honeycomb core sandwich*

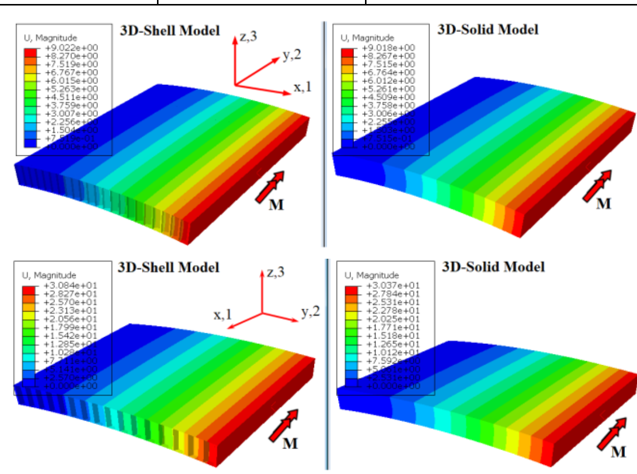


*Fig.9. Simulation of 3D-Shell model and 3D-Solid model in pure bending for the honeycomb core sandwich*

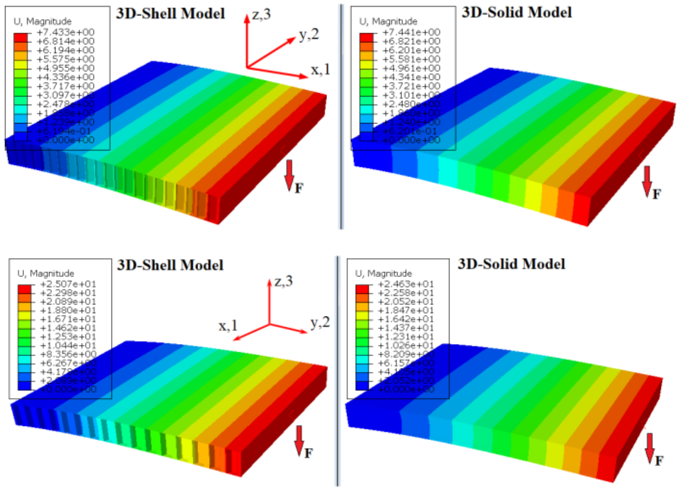


*Fig.10. Simulation of 3D-Shell model and 3D-Solid model in bending and shearing coupling for the honeycomb core sandwich*

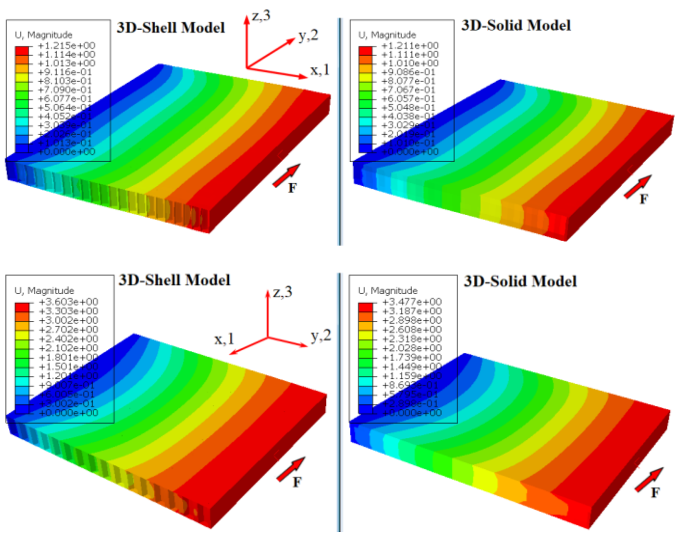


*Fig.11. Simulation of 3D-Shell model and 3D-Solid model in in-plane shear for the honeycomb core sandwich*

## IV. CONCLUSION

In this paper, we have proposed a homogenization model for the honeycomb core of a sandwich panel. This model is proven to reduce computation time by comparing the results obtained by the 3D-Shell and 3D-Solid models. In addition, the preparation time for the CAD model is also significantly reduced when using the 3D-solid model. The comparison of this result has proved the precision and effectiveness of the 3D-solid model. From this model, it is possible to implement homogenous models for other loading cases considering the effect of the skins and the effect of honeycomb core height. This homogenization model can be used not only for honeycomb core sandwich plates but also for other composite structures.

## ACKNOWLEDGEMENTS

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## REFERENCES

[1] I. Dayyani, S. Ziaei-Rad, and M. I. Friswell, “The mechanical behavior of composite corrugated core coated with elastomer for morphing skins,” J. Compos. Mater., vol. 48, no. 13,2014, pp. 1623–1636.
[2] S. N. Huang and D. W. Alspaugh, “Minimum weight sandwich beam design,” AIAA J., vol. 12, no. 12,1974, pp. 1617–1618.
[3] L. G. et M. Ashby, Cellular Solids: Structure and Properties, Cambridge. Cambridge, 1999.
[4] L. A. Carlsson, T. Nordstrand, and b. o. Westerlind, “On the Elastic Stiffnesses of Corrugated Core Sandwich,” J. Sandw. Struct. Mater., vol. 3, no. 4, 2001, pp. 253–267.
[5] N. Buannic, P. Cartraud, and T. Quesnel, “Homogenization of corrugated core sandwich panels,” Compos. Struct., vol. 59, no. 3,2003, pp. 299–312.
[6] B. M. Aboura Z., Talbi N., Allaoui S., “Elastic behaviour of corrugated cardboard: experiments and modelling,” Compos. Struct., vol. 63, 2004, pp. 53–62.
[7] M.E. Biancolini, “Evaluation of equivalent stiffness properties of corrugated board,” Compos. Struct., vol. 69, 2005, pp. 322–328.
[8] Q. H. Cheng, H. P. Lee, and C. Lu, “A numerical analysis approach for evaluating elastic constants of sandwich structures with various cores,” Compos. Struct., vol. 74, no. 2, 2006, pp. 226–236.
[9] A. Zinno, A. Prota, E. Di Maio, and C. E. Bakis, “Experimental characterization of phenolic-impregnated honeycomb sandwich structures for transportation vehicles,” Compos. Struct., vol. 93, no. 11, 2011, pp. 2910–2924.
[10] I. Dayyani, S. Ziaei-Rad, and H. Salehi, “Numerical and experimental investigations on mechanical behavior of composite corrugated core,” Appl. Compos. Mater., vol. 19, no. 3–4, 2012, pp. 705–721.
[11] Y. Q. G. P.T.M. Duong, B. Abbès, Y.M. Li, A.D. Hammou, M. Makhlouf, “An analytic homogenization model for shear-torsion coupling problems of double corrugated-core sandwich plates,” J. Compos. Mater., vol. 27, 2013, pp. 1327–1341.
[12] L. J. Gibson and M. F. Ashby, Cellular solids: Structure and properties, second edition. England: Pergamon press, 2014.
[13] J. Arbaoui, Y. Schmitt, J. L. Pierrot, and F. X. Royer, “Numerical simulation and experimental bending behaviour of multi-layer sandwich structures,” J. Theor. Appl. Mech., vol. 52, no. 2, 2014, pp. 431–442.
[14] H. Ijaz, W. Saleem, M. Zain-Ul-Abdein, T. Mabrouki, S. Rubaiee, and A. Salmeen Bin Mahfouz, “Finite Element Analysis of Bend Test of Sandwich Structures Using Strain Energy Based Homogenization Method,” Adv. Mater. Sci. Eng., vol. 2017, 2017.
[15] J. Wolff, M. Brysch, and C. Hühne, “Validity check of an analytical dimensioning approach for potted insert load

introductions in honeycomb sandwich panels," Compos. Struct., vol. 202, 2018, pp. 1195–1215.

[16] A. Farshidi, C. Berggreen, and R. Schäuble, "Numerical fracture analysis and model validation for disbonded honeycomb core sandwich composites," Compos. Struct., vol. 210, 2019, pp. 231–238.

[17] A. Aborehab, M. Kamel, A. F. Nemnem, and M. Kassem, "Experimental modal testing of a honeycomb sandwich plate," IOP Conf. Ser. Mater. Sci. Eng., vol. 973, no. 1, 2020.

[18] S. Chen, O. P. L. McGregor, A. Endruweit, L. T. Harper, and N. A. Warrior, "Simulation of the forming process for curved composite sandwich panels," Int. J. Mater. Form., vol. 13, no. 6, 2020, pp. 967–980.

[19] T. A. Mae Oiwaa,Toshio Ogasawaraa, Hajime Yoshinagab, Tsuyoshi Ogurib, "Numerical analysis of face sheet buckling for a CFRP/Nomex honeycomb sandwich panel subjected to bending loading," Compos. Struct., vol. 270, 2021.

[20] V. D. Luong et al., "Finite element simulation of the strength of corrugated board boxes under impact dynamics," in Lecture Notes in Mechanical Engineering, 2018, vol. PartF3, pp. 369–380,

[21] V. D. Luong, A. S. Bonnin, F. Abbès, J. B. Nolot, D. Erre, and B. Abbès, "Finite Element and Experimental Investigation on the Effect of Repetitive Shock in Corrugated Cardboard Packaging," J. Appl. Comput. Mech., vol. 7, no. 2, 2021, pp. 820–830.

[22] M. Lin and P. Papadopoulos, "Formula-one 3D vehicle aerodynamic modeling, geometric idealization and mesh refinement strategies," WSEAS Trans. Fluid Mech., vol. 14, 2019, pp. 40–61.

[23] L. Setlak and R. Kowalik, "Dynamics of the designed robotic manipulator in the CAD program," WSEAS Trans. Appl. Theor. Mech., vol. 14, 2019, pp. 66–74.

[24] T. G. Garbowski, Tomasz, "Determination of Transverse Shear Stiffness of Sandwich Panels with a Corrugated Core by Numerical Homogenization," Mech. Corrugat. Compos. Mater., vol. 14, no. 8, 2021, p. 1976.

## Contribution of individual authors to the creation of a scientific article (ghostwriting policy)

Luong Viet Dung carried out the simulation for two models.

Dao Lien Tien has built models and written manuscripts.

Duong Pham Tuong Minh has implemented the homogenization model.