Mechanics of Advanced Composite Structures 10 (2023) 333 – 342

Semnan University

Mechanics of Advanced Composite Structures

journal homepage: https://MACS.journals.semnan.ac.ir

Numerical Analysis of Thermal Buckling of Honeycomb Core Sandwich Panel under Thermal Loading

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KEYWORDS

Sandwich panel;
Thermal buckling;
Finite element analysis;
Composites.

ABSTRACT

In the aerospace industry, the usage of sandwich panels in a variety of space structures such as satellites is increasing due to their excellent features like high strength-to-weight ratio, and thermal insulation. These structures are subjected to a variety of thermal loads depending on their working conditions, which cause them to expand or contract. Since these panels are connected simultaneously by twists, buckling is possible due to thermal loads. In this paper, the thermal buckling analysis of a CCCC Aluminum honeycomb core sandwich panel is performed under asymmetric thermal loading, using ABAQUS. The face sheets are attached to the core by adhesive. Thermal loading is assumed to be two heat fluxes of 70 watts and 20 watts, which are asymmetric, and face sheets radiate heat to their surroundings. The modeling results showed the panel does not buckle under the mentioned thermal loading. The critical buckling stress is 750 MPa and 400 MPa where the maximum thermal stress is 95 MPa and 37 MPa for vertical and horizontal edges respectively, which shows a significant difference of 8 to 11 times. The temperature distribution of the various points of the panel was also obtained by calculating the maximum temperature of 84 °C at the 70-watt heat flux location and the minimum temperature of 15.65 °C in the lower-left corner. The influence of various parameters like face sheets and core cell wall thickness on buckling occurrence is also discussed.

1. Introduction

A sandwich panel is made of two highly stiff face sheets, separated by a low-density core. Sandwich plate cases are generally of the same material and thickness, and sandwich plate cores are usually lightweight and thicker than the face sheets. Regarding the load-carrying performance of the sandwich panels, in-plane compressive and shear loadings are carried by the face sheets, while out-off-plane loadings are carried mostly due to the presence of the core. The face sheets and the core are connected by an adhesive to provide proper load transfer between them [1]. Sandwich panels are being increasingly used in the manufacturing process of airplanes and spacecraft, as well as space structures like satellites because of sandwich panels' excellent characteristics such as high strength-to-weight ratio and thermal insulation. Sandwich panels used in space structures are subjected to asymmetric thermal loading, which causes thermal stresses and sometimes buckling. The source of the asymmetric thermal loading is mainly the solar heat radiations and different temperatures of the compartments that are installed close to the sandwich panels. Therefore, thermal analysis of these plates is principal where engineers and researchers have studied its importance with various experimental, analytical, and numerical methods [2–4].

Experimentally, Liu et al. [5] studied the mechanical behavior of sandwich panels with pyramidal cores at high temperatures and different periods. Their study predicts the modulus and the compressive strength of sandwich panels under thermal loading. Their study shows that time and high temperature are both essential factors in the mechanical behavior of sandwich panels. Rajesh and Pitchaimani [6] had an experimental investigation on buckling and free vibration behavior of woven natural

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DOI: 10.22075/macs.2023.29070.1454
Received 2022-11-22; Received in revised form 2023-02-27; Accepted 2023-03-08.

fiber fabric composite under axial compression. Their results showed that by increasing the number of composite layers, the critical buckling load increases. Also, the fiber weaving architecture can affect the critical buckling load where different architectures obtain a different one. The sandwich plate, which is made of a glass-reinforced polymer as its face sheet and natural fiber composite as its core, also achieves a higher critical buckling load. To ensure the validity of the obtained results, some researchers use both experimental and numerical methods [7–9]. An experimental investigation and numerical simulation were done by Zheng et al. [10] to determine the thermal properties of metallic honeycomb sandwich panels. They increased the temperature from 200 to 900° C and considered all three types of heat transfer in the panel. The results, while having good agreement with each other, showed that the heat transfer coefficient increased from 0.446 W/ (m ℃) to 1.52 W/ (m ℃). Rao et al. [11] analyzed the heat insulation of an aluminum honeycomb sandwich structure with different core shapes. By utilizing both experimental and numerical methods, they obtained that the hexagonal core is better insulated than the square core. Experimental methods are often costly; the required equipment and materials are not always available, and human errors in component manufacturing can also cause a difference between numerical and experimental results.

Many researchers choose theoretical and numerical methods to study the buckling of sandwich panels under thermal and mechanical loadings [12–17]. Viscovini et al. [18] presented a conceptual model to investigate the global and local buckling of sandwich panels with orthotropic core and heterogeneous face sheets. The purpose of the mentioned study was to compare the results obtained by their model with previous works to prove their model's validity. Through this investigation, they showed that the higher-order nonlinear buckling and the lower-order linear buckling yield outstanding results. Various cases like foam and honeycomb core under uniaxial and biaxial loading with simply-support boundary conditions were also studied, and the model worked well in all cases. Huang [19] studied the fracture mechanisms as well as the optimal design of metal sandwich panels with truss cores under uniform thermal loading with both theoretical and numerical methods. Han et al. [20] proposed a conceptual model to obtain the free vibration and buckling of sandwich panels with a foam-filled corrugated core under thermal loading. A refined shear deformation theory extended, incorporating two different combinations of hyperbolic and parabolic shear shape functions. Equivalent thermo-elastic properties of the mentioned core were obtained using the method of homogenization based on Gibbs free energy. The results showed that the present model while performing well, facilitates the study of thermal buckling and free vibration behavior of foam-filled corrugated core sandwich panels. Qiu et al. [21] investigated the buckling of a sandwich panel under out-of-plane loading by three experimental, analytical, and finite element methods using ABAQUS software, while another study focused on the performance of an auxetic-core sandwich panel in a thermal environment [22]. Moreover, other studies investigated the torsional buckling behavior of sandwich panels under thermal loading and vibrations [23,24]. Although several studies are carried out to investigate the buckling of sandwich panels under thermal loading, the buckling behavior of sandwich panels under asymmetric thermal loading is not properly studied using numerical and computational methods. A work is done on an analytical study of thermal buckling and post-buckling behavior of composite beams reinforced with SMA by Reddy Bickford theory [25], while another study investigated the buckling and vibration of symmetrically laminated composite elliptical plates on an elastic foundation subjected to uniform in-plane force [26]. In this study, the core shape chose as rectangular, honeycomb, and triangular; the loading was considered as uniaxial compressive and a combination of a compressive and shear load. The purpose of this study was to prove the efficiency of the selected numerical method. In fact, with the results of the three methods and comparison, it was concluded that the chosen numerical method could be used efficiently instead of two other methods. By applying this method, Difficulties in the experimental approach, such as fabrication and component testing, problems in the finite element method such as modeling, and the costly, or inaccessibility of computer systems, are avoided.

In this paper, the numerical analysis of the thermal buckling of an Al sandwich panel with a honeycomb core under asymmetric thermal loading is investigated using Abaqus software. The face sheet thickness is minimal in comparison with the core, so shell elements were chosen for modeling the panel; the boundary conditions were considered as fully clamped in all four edges. Thermal loading is assumed to be two concentrated heat fluxes that are asymmetric, while both face sheets radiate heat to their surroundings. The thermal and critical buckling stress is calculated and compared to determine whether or not thermal buckling occurs. The thermal distribution was also obtained at different points of the panel. Finally, the influence of parameters like cell-wall thickness, face sheet thickness, as well as core thickness on temperature distribution, as well as critical buckling stress, is discussed. In this research work, it is assumed that the thermal load that is applied asymmetrically on the sandwich panels will only cause mechanical stress and finally buckling in these panels. This issue distinguishes the work done from other previous works. In this research work, it is assumed that the thermal load that is applied asymmetrically on the sandwich panels will only cause mechanical stress and finally buckling in these panels. This issue distinguishes the work done from other previous works.

2. Buckling

Buckles are typically caused by a combination of three major factors: high compressive forces, weakened track conditions, and vehicle loads (train dynamics). Since the honeycomb plates used in satellites are constrained by satellite frames from four sides, applying thermal load on the plate causes mechanical stress in the face sheets of the sandwich panels. A plate is a 3-dimensional structure defined as having a width of comparable size to its length, with a thickness that is very small in comparison to its other two dimensions. Similar to columns, thin plates experience out-of-plane buckling deformations when subjected to critical loads; however, contrasted to column buckling, plates under buckling loads can continue to carry loads, called local buckling. This phenomenon is incredibly useful in numerous systems, as it allows systems to be engineered to provide greater loading capacities.

For a rectangular plate, supported along every edge, loaded with a uniform compressive force per unit length, the derived governing equation can be stated by [27]:

$$\frac{\partial^4\omega}{\partial x^4}+2\frac{\partial^4\omega}{\partial x^2\partial y^2}+\frac{\partial^4\omega}{\partial y^4}=\frac{12(1-\upsilon^2)}{Et^3}\left(-N_x\frac{\partial^2\omega}{\partial x^2}\right) \tag{1}$$

where ω is the uniformly distributed compressive load.

The solution to the deflection can be expanded into two harmonic functions shown:

$$\omega=\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\omega_{mn}\sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi y}{b}\right) \tag{2}$$

In the above relation, m is the number of half-sine curvatures that occur lengthwise, n is the number of half-sine curvatures that occur widthwise, a is the length of the specimen and b is the width of the specimen.

The previous equation can be substituted into the earlier differential equation where n equals N_x can be separated providing the equation for the critical compressive loading of a plate:

$$N_{x,cr}=k_{cr}\frac{\pi^2Et^3}{12(1-\upsilon^2)b^2} \tag{3}$$

where k_{cr} is the buckling coefficient given by:

$$k_{cr}=\left(\frac{mb}{a}+\frac{a}{mb}\right)^2 \tag{4}$$

The buckling coefficient is influenced by the aspect of the specimen, a/b, and the number of lengthwise curvatures. For an increasing number of such curvatures, the aspect ratio produces a varying buckling coefficient; but each relation provides a minimum value for each m. This minimum value can then be used as a constant, independent from both the aspect ratio and m [27]. Given stress is found by the load per unit area, the following expression is found for the critical stress:

$$\sigma_{cr}=k_{cr}\frac{\pi^2E}{12(1-\upsilon^2)(\frac{b}{t})^2} \tag{5}$$

From the derived equations, it can be seen close similarities between the critical stress for a column and a plate. As the width b shrinks, the plate acts more like a column as it increases the resistance to buckling along the plate's width. The increase of a allows for an increase in the number of sine waves produced by buckling along the length but also increases the resistance from the buckling along the width [27]. This creates the preference for the plate to buckle in such a way as to equal the number of curvatures both along the width and length. Due to boundary conditions, when a plate is loaded with critical stress and buckles, the edges perpendicular to the load cannot deform out-of-plane and will therefore continue to carry the stresses. This creates a non-uniform compressive loading along the ends, where the stresses are imposed on half of the effective width on either side of the specimen, given by the following relation:

$$\frac{b_{eff}}{b}\approx\sqrt{\frac{\sigma_{cr}}{\sigma_y}\left(1-1.022\sqrt{\frac{\sigma_{cr}}{\sigma_y}}\right)} \tag{6}$$

where b_{eff} is the effective width and σ_y is the yielding stress.

As the loaded stress increases, the effective width continues to shrink; if the stresses on the ends ever reach the yield stress, the plate will fail. This is what allows the buckled structure to continue supporting loadings. When the axial load over the critical load is plotted against the displacement, the fundamental path is shown. It demonstrates the plate's similarity to a column under buckling; however, past the buckling load, the fundamental path bifurcates into a secondary path that curves upward, providing the ability to be subjected to higher loads past the critical load.

3. Modeling Method

The sandwich panel examined in this research work consists of two face sheets and a honeycomb core with hexagonal cells, which is made by glue and baking process in the oven. In this investigation, a 60×60×3 cm sandwich panel is used. The core is considered a honeycomb with a cell size of 25 mm. The face sheets and the core are made of 7075-T6 aluminum and aluminum 3003, respectively. Also, the face sheet-to-core connection is assumed to be an adhesive joint; the mechanical and thermal properties of the materials can be found in Table 1.

The procedure of modeling the core is that a cell is designed first, and then, some of these cells are duplicated in length and width to obtain the desired dimensions. The core, the face sheet, and the cell wall thickness are assumed 26 mm, 2 mm, and 0.6 mm, respectively. Because of the small thickness of the cell wall as well as the face sheets, the S4R (4-node quadrilateral) shell element was chosen for the mesh type. To obtain the best results, all degrees of freedom of the core, face sheets, and adhesive are coupled together to act as a single set. Fig. 1 shows the designed sandwich panel with the mentioned mesh. Moreover, the schematic view of heat flux locations and radiation with the environment is presented in Fig. 2(a-b).

Table 1. Mechanical and thermal properties of the panel materials

Properties	Symbol	Al. 3003 [29]	Al. 7075-T6 [28]	Adhesive
Young's modulus (GPa)	E	70	71.6	2.3
Poisson's ratio	ν	0.33	0.33	0.36
Density (Kg/m^3)	ρ	2730	2810	1100
Thermal expansion coefficient ($\mu m/m$°C)	α	23.2	25.2	90
Conductivity (W/m-K)	k	162	130	0.1883
Specific heat (J/Kg°C)	c	900	960	1046

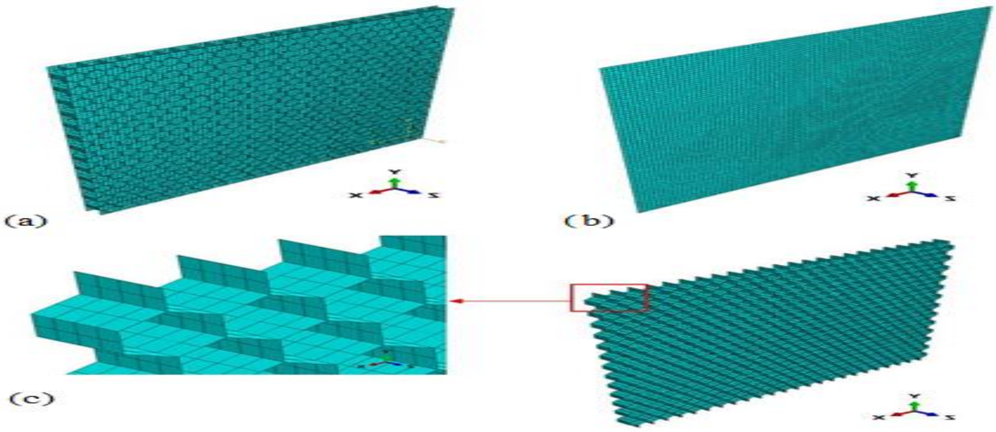

Fig. 1. Meshing the sandwich panel with S4R elements; (a) The panel, (b) The face sheet, (c) The core

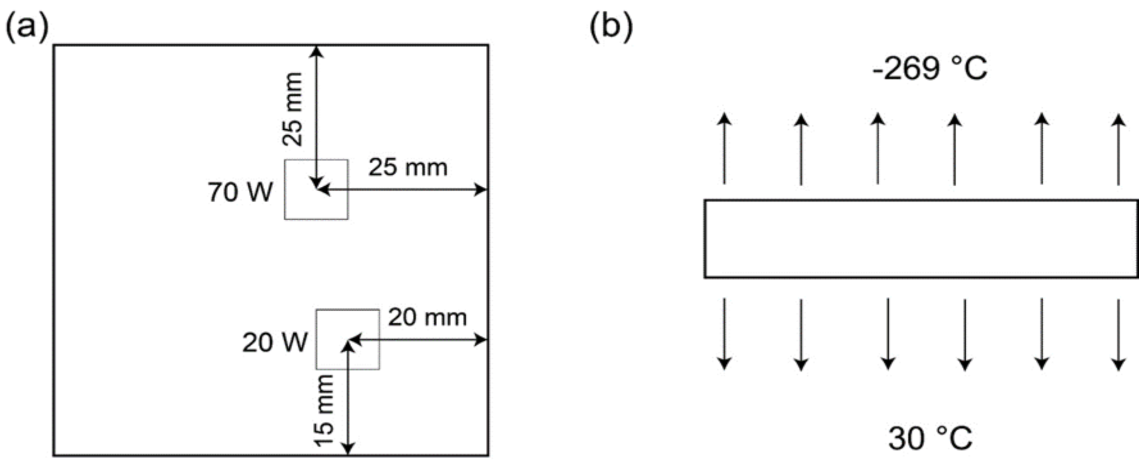

Fig. 2. Asymmetric thermal loading; (a) Heat flux locations, (b) Radiation with the environment

4. Results and Discussion

4.1. Validation

Benthouhami and Keskes [30] investigated the Buckling capabilities of a pressurized sandwich panel in both the experimental method and finite element modeling using Abaqus software. While reviewing several cases, they presented the critical buckling load for different cell numbers, which was used to validate the numerical method used in this study as stated in the previous section. Figure 3 shows the comparison between the results presented by this reference and the results of the present modeling, which show an error of less than 2%.

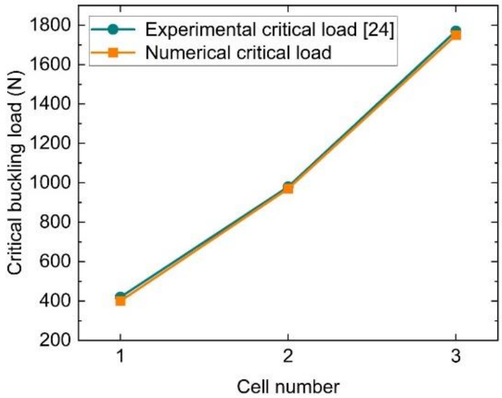

Fig. 3. Comparison between the results presented by the reference [30] and the present modeling

4.2. Occurrence or Non-occurrence of Global and Local Buckling

To investigate whether or not global buckling occurs, the thermal stresses at the face sheet edges are compared with the stresses at the moment of buckling. Fig. 4 shows the stresses generated at the moment of buckling. Because of the existing symmetry, the stress diagrams are the same for the horizontal edges and the vertical edges. Also, stress is uniform at the horizontal edges, while stress variations for the vertical edges show severe fluctuations. The reason for this is the distortion in the panel along the vertical edges when buckling occurs. The phenomenon is called Wrinkling, which Frosting et al. [31] have discussed in their research.

By comparing the maximum and minimum buckling and thermal stresses at each edge, it was deduced that under mentioned asymmetric thermal loading, the vertical edges are up to 80% and the horizontal edges up to 90% distant from the buckling condition. So, global buckling will not happen. Considering various thermal loading, the temperature in some areas (like heat sources) may be so high that the thermal stress exceeds the buckling stress. In this condition, called that the local buckling happened at that particular location. Given the asymmetric thermal loading mentioned above, there are two areas of 70- and 20-watt heat fluxes. So, the local buckling may occur at the location of these fluxes as the heat centers, which must be considered. Table 2 presents the values of the thermal and buckling stresses of the two heat flux points. By comparing these stresses in both X and Y directions, observed that the areas with 70- and 20-watt heat fluxes are approximately 88% and 94% far from buckling conditions. Therefore, the occurrence of local buckling also exclude.

4.3. Temperature Distribution at Various Points

Based on the thermal analysis, the maximum and minimum temperatures obtained are 84 and 15.65 °C at the 70-watt heat flux location and in the lower-left corner, respectively. Fig. 5 shows the temperature over time graphs.

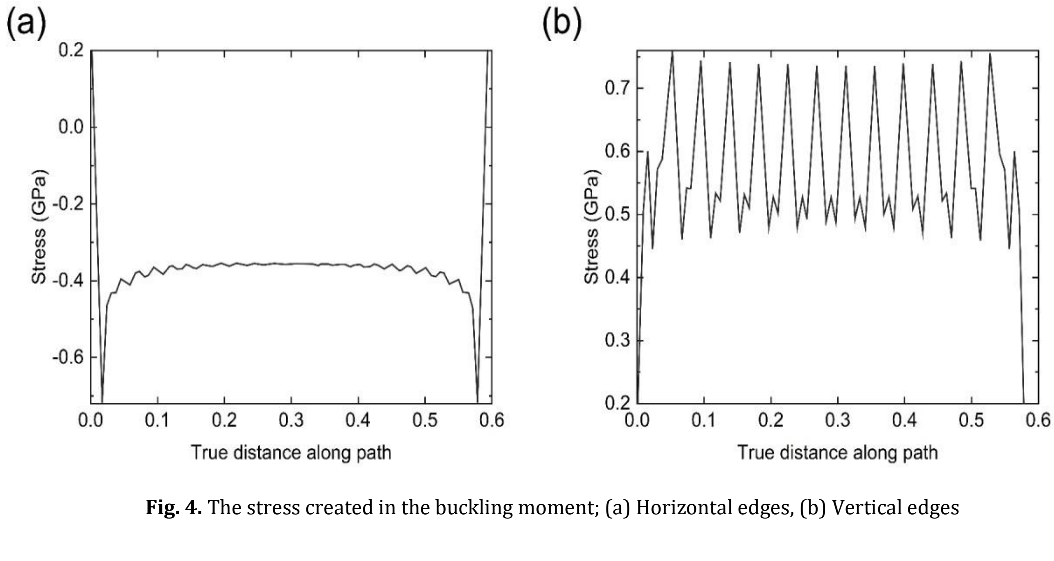

Fig. 4. The stress created in the buckling moment; (a) Horizontal edges, (b) Vertical edges

Table 2. Values of the thermal and buckling stresses of the two heat flux points

	Buckling Stresses (MPa)		Thermal Stresses (MPa)		Difference %	
	X direction	Y direction	X direction	Y direction	X direction	Y direction
70-watt flux region	−641.17	−647.12	−74.42	−81.29	88.4	**87.94**
20-watt flux region	−598	−765	−35	−50	94.15	**93.46**

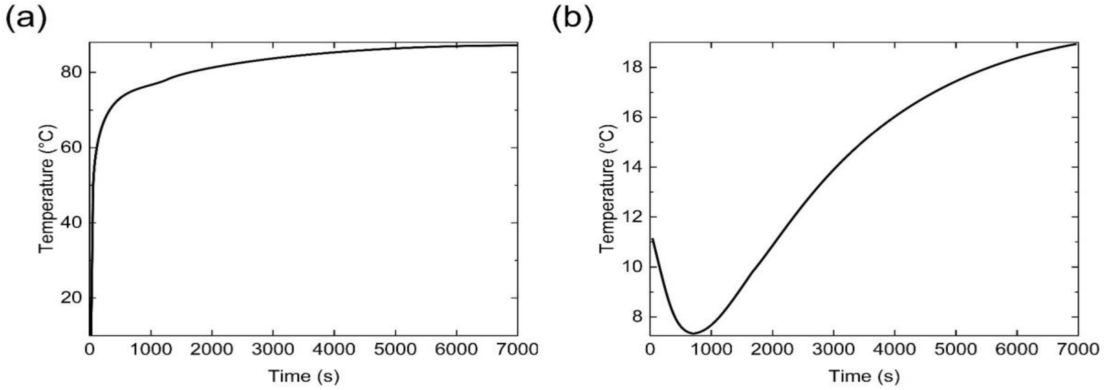

Fig. 5. Temperature distribution over time; (a) 70-watt location, (b) Left lower corner

It is observed that the panel has reached thermal equilibrium within two hours. The initial temperature of the panel was assumed to be 10°C, so the temperature-time diagram started from this point. In figure 7b, the temperature first decreased and then increased. This behavior was created due to the asymmetric thermal loading as well as the structure of the honeycomb sandwich panel. Figure 6 shows the temperature distribution and the heat movement from the heat centers to the different directions of the panel. Depending on the position of the heat sources, the heat reaches the left side later than the rest. As mentioned earlier, the panel radiates heat to its surroundings, with outside and inside temperatures of -269 °C and 30 °C, respectively. The outside is much cooler than the inside. The panel loses some temperature while the heat reaches the heat centers and begins to warm after that. The process takes about 10 minutes. Examination of the temperature at similar points on the front and back of the panel showed that after thermal equilibrium, a temperature difference of 1.9 °C was created between the panel face sheets, indicating heat dissipation in the vacant space in the hexagon core.

4.4. Effect of the Cell-wall Thickness

As the cell wall thickness increases, the core becomes stiffer, resulting in a higher critical buckling load, which has a positive effect. Nevertheless, this increase in thickness increases the mass of the core, which has a negative effect. Because, as mentioned earlier, engineers and designers always seek to reduce structural mass while maintaining design goals. According to Figure 7, the critical buckling load increased by 79%, with a 35% increase in cell wall thickness. This increase in thickness increased the panel's mass by 20%. However, if the panel mass increases by 10% (cell wall thickness of 0.8 mm), the critical buckling load is increased by 45%.

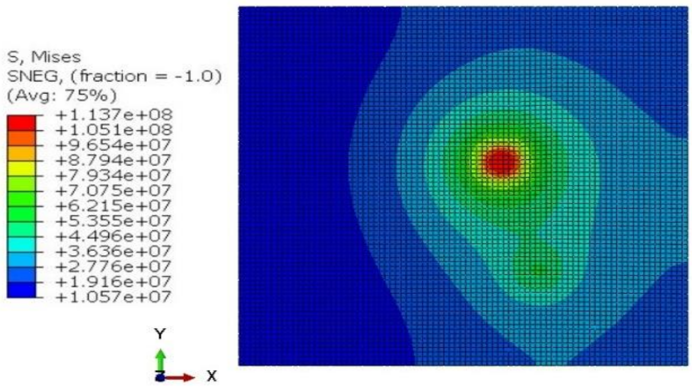

Fig. 6. Heat movement from the sources to various directions

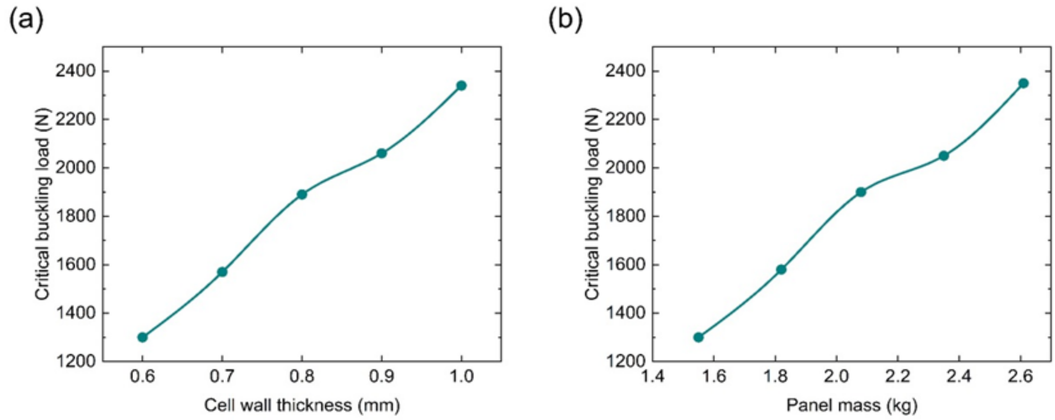

Fig. 7. Effect of cell wall thickness increasing on the critical buckling load

Increasing this thickness had little effect on the temperature distribution and as a result heat transfer. With a 35% increase of this thickness from 0.6 to 1 mm, the temperature difference at similar points on the inner and outer face sheet is approximately 0.2°C and 0.3°C, respectively. The remarkable point is the decrease in temperature difference between the two face sheets with the increasing cell wall thickness, which means faster heat transfer in the panel (Figure 8). As the thickness of cell wall thickness increases, the volume of material increases, resulting in a faster conduction heat transfer. Also, the space in the hexagonal cells reduces, which results in less heat loss. Increasing this parameter by 35% did not have much effect on the thermal stress. This stress at the mentioned points and the edges increased by about 4 MPa and 10 MPa, respectively, which is due to the increased surface area of contact sheets with the core. Due to the significant increase in the critical buckling load and the slight increase in thermal stress, buckling will not occur under these conditions.

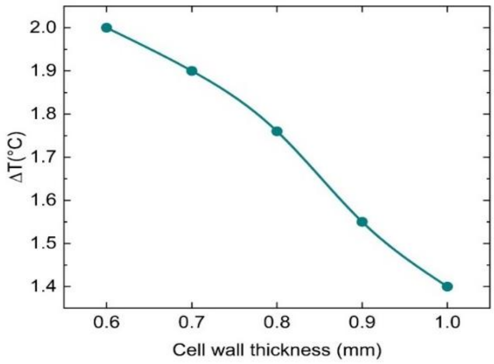

Fig 8. Cell wall thickness effect on the temperature difference between the face sheets

4.5. Effect of Face Sheet Thickness

Increasing the surface thickness of the sandwich panel increases the critical buckling load. It can see from Fig. 9 that this increase is linear. It should note that the mass of the panel and the cost of its production also increase. Due to the constant core thickness, the temperature difference between the two face sheets remains almost constant (about 1.9 ° C). The vital issue is the faster spread of heat due to the increased thickness of the face sheet. This case is especially evident in the left midpoint as well as the minimum temperature point. Previously stated that due to the location of heat fluxes and heat coupling, as well as the form of heat transfer (Fig. 6), the heat transfer to the minimum temperature in the lower-left corner is too late and too wasted before it arrives. By increasing the face sheet thickness from 1 to 3 mm, the minimum temperature increased by more than 5 degrees, so it can conclude that as the thickness of the face sheet increases, heat transfers more rapidly, so that there is less waste heat. As the face sheet thickness increased and the clamped boundary condition was used, the stress at the edges was expected to increase as more areas were fixed. However, since the thermal loading is asymmetric, this stress increase must be asymmetric. According to the results, the upper and lower edge stresses increased by about 110% and 25%, respectively. There was a 50% increase in stress on both the right and left edges. Due to the three-time increase in the critical buckling stress, which is higher than the increase in thermal stress, buckling does not occur.

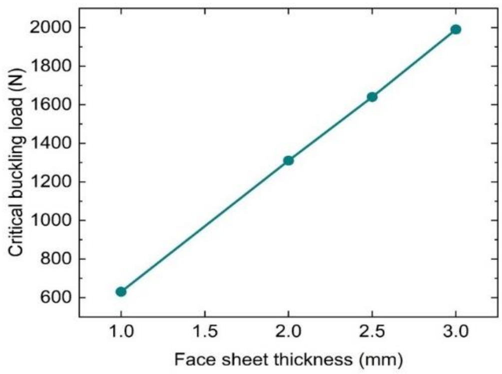

Fig. 9. Effect of face sheet thickness on the critical buckling load

4.6. Effect of Core Thickness

Figure 10 shows the effect of increasing the core thickness on the critical buckling load. As the two-time increase in core thickness from 15 mm to 30 mm, the critical buckling load is reduced by about 45%. The heat flux's locations are on the inner surface. Since the thickness of the face sheet is constant; therefore, as the thickness of the core increases, there is no change in the amount of heat generation and its transfer to the inner face sheet. However, it will take longer for the outer face sheet to heat up as the empty cell space increases. Therefore, the heat dissipation increases, and the temperature difference between the two plates increases. It can conclude that this enhancement enhances the thermal insulation property of this sandwich panel.

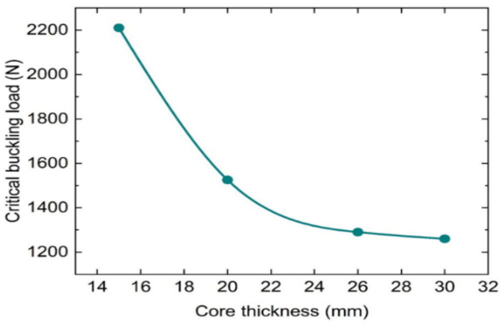

Fig. 10. Effect of core thickness on the critical buckling load

On the other hand, increasing the core mass increases the total panel mass and increases production costs. That is what has always been the focus of Sandwich panel engineers and designers. Figure 11 shows that by a two-time increase in core thickness from 15 mm to 30 mm, the temperature difference between the inner and outer surfaces reaches from 1.6 °C to 2.1 °C, which means a 30% improvement in the insulation properties of the panel. At the same time, a 21% increase in panel weight should be considered (Figure 11(b)).

5. Conclusions

In this paper, the thermal buckling of the sandwich panel with the kernel core under asymmetric thermal loading is studied by the finite element method. Based on finite element analysis on sandwich panels in different geometric parameters and loads, the following can be briefly mentioned:

- Critical buckling stress analysis at the edges of the panel showed that the panel, in the buckling moment, had a distortion in one direction. At the same time, there was a uniform stress perpendicular to it.
- The results showed that due to the thermal coupling of heat fluxes and the structure of the honeycomb core panel, the heat reaches the left edge later than the other edges while the minimum temperature occurs in the lower-left corner of 15.65 degrees Celsius.
- A comparison of thermal and buckling moment stresses explained that thermal stress is about 0.1 times the critical buckling stress, and thus, buckling will not occur.
- The thermal insulation properties of the sandwich panel with a honeycomb core are also investigated, which obtained that a temperature difference of 1.9 ° C is created between the inner and outer face sheets.
- Examination of different thicknesses of cell wall thickness revealed that by increasing this parameter by 35%, the critical buckling load increased by 79% but also by a 20% increase in panel mass, which was a negative point.
- Increasing face sheet thickness reduces heat generation. By increasing three-times thickness from 1 to 3 mm, reducing the heat center temperatures reduced by 40%, but also heat transfer happened more rapidly. The midpoint of the left edge, as well as the minimum temperature point, warmed to 5.8 ° C. This warming also happened elsewhere but was insignificant.
- The change in core thickness is also discussed as an essential parameter. A two-times increase in core thickness from 15 mm to 30 mm reduced the critical buckling stress by 45% but also reduced thermal stresses as well.

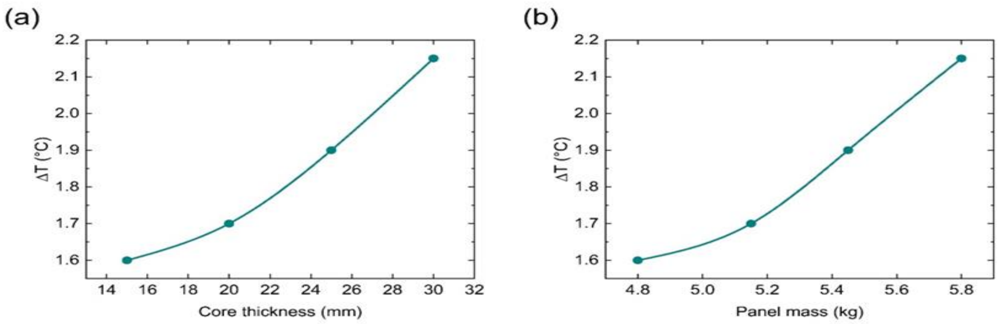

Fig. 11. (a) Effect of core thickness on the temperature difference; (b) Effect of panel mass on the temperature difference

Nomenclature

E	Young's modulus
υ	Poisson's ratio
ρ	Density
α	Thermal expansion coefficient
c	Specific heat
k	Thermal conductivity
w	Out-of-plane deflection
t	Thickness

Conflicts of Interest

The author declares that there is no conflict of interest regarding the publication of this manuscript.

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