Crashworthiness behavior of aircraft sandwich structure with honeycomb core under bending load.

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Abstract. Sandwich structures have been widely used as lightweight composite parts in the aerospace and shipbuilding engineering for their high capacity of stiffness, strength and energy absorption. There are three different criteria in bending crashworthiness for sandwich structure, namely peak bending load, maximum deflection and energy absorption. In this paper, the crashworthiness criteria of sandwich structure were evaluated theoretically and numerically based on failure mode maps. A failure mode map for the loading under three-point bending was constructed, depicting the reliance of the failure mode and the load upon the ratio of the skin thickness to the span length and the relative density of honeycomb. The finite element models for the sandwich panel with a honeycomb core were developed and analyzed via Ansys soft-ware package. The obtained result elucidated a good agreement between these models and the theoretical solution, where the error ratio was not exceeded 5%. To explore the effect of honeycomb parameters on the crashworthiness criteria of sandwich structure, several parameters have been selected, including the core height, the size of cell and the thickness of cell wall. In order to obtain the optimum solution of crashworthiness, Design of Experiment (DOE) software with the technique of Response Surface Methodology (RSM) was used. Results showed that the optimum value of peak bending load (25310 N) as maximum, deflection as minimum (0.8976 mm) and energy absorption as maximum (9.9949 J) were found at 29.2424 mm core height, 5 mm cell size and 1 mm cell wall thickness. Finally, the present study provides a new basis for more studies upon the optimization of the crashworthiness of sandwich structures.

1. Introduction
Sandwich construction is usually utilized in the structures, where the strength, stiffness, and weight efficiency are needed. Most commonly, sandwich panels are employed in satellites, aircraft, spacecraft, trains, automobiles, and satellites, which reflect on the decreased consumption of power, higher speeds and increment in pay load [1], [2] and [3]. The construction of a sandwich comprises two thin facing layers disintegrated via a thick core, as shown in figure 1. Accordingly, the strength properties of such panels are investigated via numerous researchers. The strength properties of the aluminum honeycomb sandwich panels were investigated via Paik et al. [4] who performed 3-point bending, buckling, and lateral crushing tests. Kuldeep P. Toradmal et al. [5] conducted the analysis of the 3-point bending of the honeycomb sandwich panels experimentally. In such investigation, GFRP was chosen as a face sheet material along with its metallic counterparts, like aluminum (Al) and stainless steel alloys. Polypropylene was utilized as a familiar material of core. Results showed a high value of ultimate load occurred in the GFRP–Polypropylene core. Rao et al. [6] provided a theoretical study on the strength of sandwich structure under a bending load with different face materials. Aluminum, titanium and high tensile steel were used in this study. It was absorbed that the titanium
alloy has better properties of sandwich structure. D. Fadhel et al. [7] presented a numerical and experimental study of the bending behavior for a honeycomb sandwich panel with various core forms (square, hexagonal and circular) each form has two kinds of facing, one of composite and the other of Al. Three-point bending test was conducted in this research.

![Figure 1. Sandwich structure.](image)

V. Matta et al. [8] used Taguchi design of experiment to study the flexural behavior of Al honeycomb core sandwich structure experimentally. Effect of the size of cell, sheet thickness and core height on the bending stress has been concluded. A same method was used by E. S. Arbintarso et al. [9] for investigating the bending stress of glass fiber reinforced polymer (GFRP) sandwich structure for lightweight vehicle. Three types of adhesives (The plastic steel epoxy resin, polyaminoamide-bisphenol-A resin, and thermosetting resin) used to adhesive the face and core. Naseem Abbas et al. [10] improved the mechanical propertied of sandwich panels by using glass fiber composite material for facing with aluminum honeycomb core. A three-point bending load arrangement is conducted to examine the static and fatigue performance of honeycomb sandwich panel. Experimental approach and numerical simulation have been concluded in this research.

Despite of the abovementioned searches, to the best knowledge of author, the investigations on the crushing behaviors for sandwich structures with Al honeycomb core stay fairly bounded so far. The present study aims to fill the gap of the knowledge via giving an analytical investigation into the crashworthiness under 3-point bending and numerical simulation has been concluded to explore the effect of honeycomb parameters (cell size, cell wall thickness and core height) on the crashworthiness of sandwich structure with a honeycomb core. After then, the RSM technique will be employed by DOE software to get the optimum solution for the peak bending load, maximum deflection and energy absorption.

2. Definition of Crashworthiness Criteria
As described in [11] and [12], three different criteria, namely peak bending force ($F_{\text{max}}$), maximum deflection ($\delta_{\text{max}}$) and energy absorption ($E_a$) were involved analytically and numerically for evaluating the sandwich beams crashworthiness with various structural factors. The maximum bending force ($F_{\text{max}}$) was determined straightforward from the curve of load-displacement, frequently taking place at the elastic displacement end that characterizes the crushing strength [12]The absorption of energy ($E_a$) can be determined via integrating the curve of load-displacement that depicts how much energy that the sandwich structure is able to absorb through a certain crushing distance.

3. Analysis
In this section, the sandwich beams’ elastic analysis in three-point bending was outlined. This is going to be utilized for evaluating the stresses in the skin or the core and then loads of failure owing to the
different mechanisms. A typical simply-supported sandwich beam, with width \( b \) and span \( L \), loaded in a three-point bending with a central load \( P \) per unit width is illustrated in the figure 2. The faces possess Young's modulus \( E_f \), density \( \rho_f \), and yield strength \( \sigma_{yf} \); the core possesses Young's modulus \( E_c \), shear strength \( \tau_c \), shear modulus \( G_c \), uniaxial compressive yield strength \( \sigma_{cc} \), uniaxial tensile yield strength \( \sigma_{ct} \), and density \( \rho_c \). The solid, from which the core is foamed, possesses yield strength \( \sigma_{ys} \), Young’s modulus \( E_s \), and density \( \rho_s \). Each face thickness is \( t_f \), and the core thickness is \( c \).

**Figure 2.** Simply supported sandwich beam

It is assumed that the skins stay steadily bonded to the core, the beam bends in a cylindrical way without curvature in the \( yz \)-plane, and the cross sections stay plane and normal to the beam longitudinal axis. Then, the sandwich beam’s flexural rigidity \( D \) is given via [13]

\[
D = \frac{E_f t_f b d^3}{2}
\]

(1)

Where, \( d \) represents the distance between the mid-planes of the bottom and upper skins. In three-point bending, the maximum bending moment \( M \) is at the mid span, and the corresponding maximum stress \( \sigma_{fx} \) of the skins is given via Allen [14]

\[
\sigma_{fx} = \frac{P b L}{4} \left[ c + 2 t_f + \frac{P L}{4} \frac{t_f}{2} \frac{1}{\theta} \right]
\]

(2)

Where,

\[
\theta = \frac{L}{c} \left[ \frac{G_c c}{2 E_f t_f} \left( 1 + \frac{3 d^2}{2 t_f^2} \right) \right]^{0.5}
\]

(3)

\[
l = \frac{b t_f^3}{6} + \frac{b t_f d^2}{2}
\]

(4)

\[
G_c = \frac{2 T_c}{L_c \cos \beta (1 + \sin \beta)} G_f
\]

(5)

Where, \( G_c \) is the core out-of-plane shear modulus of the core, \( T_c \) is the cell wall thickness, \( L_c \) is the cell size, and \( \beta \) is the cell angle, as shown in figure 2 [15]. Also, \( l \) is the sandwich second moment of area with regard to its neutral axis, and \( l_f \) is the face plates second moment of area with regard to their centroid axes. Equation (3) reveals that \( \theta \) relies upon the relative stiffness of the core and skin. Eventually, Equation (2) gives
Shear stress changes through the core and face in a parabolic manner under three-point bending. When faces are thinner and too stiffer than the core, shear stress can be considered fixed in the core and a linear through the face. Neglecting the skins’ contribution, the average shear stress in core is given via [16]:

\[ \tau_c = \frac{P}{2d} \]  

### 3.1 Skin failure

Section 3 gives a formula for the ultimate stress (\(\sigma_{fu}\)) of skins. This can be utilized for predicting the failure of beam owing to the modes of the skin failure of face yielding, face wrinkling or intra-cell dimpling, as demonstrated in the figure 3 [16].

#### 3.1.1 Yielding of face

The failure will take place in the upper skin owing to the yielding of face if the axial stress in both skins (Equation 2) attains the in-plane strength (\(\sigma_{fy}\)) of face material for the loading along the axis of beam.

\[ \sigma_{fx} = \sigma_{fy} \]  

It’s assumed that for a symmetrical beam, the stress is same in the faces of compression and tension. For the composite face materials, the compressive type, in general, is the critical face.

#### 3.1.2 Intra-cell dimpling

A sandwich having a honeycomb core perhaps fail via face buckling where it’s unsupported via the honeycomb walls (figure 3(b)). The theory of the simple elastic plate buckling can be utilized for deriving a formula for the in-plane stress (\(\sigma_{fi}\)) in skins, at which the intra-cell buckling takes place as:

\[ \sigma_{fi} = \frac{2E_f}{1-v_f} \left( \frac{2t_f}{a} \right)^2 \]  

Where, \((a)\) is the size of cell (i.e. the inscribed circle diameter) of honeycomb, and \((v_f)\) and \((E_f)\) are Poisson’s ratio and modulus of elasticity of skin, respectively for the loading in axial direction. Equations (9) and (10) can be utilized for deriving the cell size value, over which there exists a transition from the yielding of face to the intra-cell buckling as [17]:

\[ P = 4 \sigma_{fx} \xi \frac{t_f}{L} \]  

Where

\[ \xi = \Theta \frac{t_f^5}{9} + \frac{t_f^3d^2}{3} - \frac{t_f^3h(\Theta - 1)}{3} + \frac{t_f^4}{3} + t_f^2d^2 \]
2.1.3 Wrinkling of face

It’s a mode of buckling of skin with a wavelength larger than the width of cell of a honeycomb (figure 3(c)). Buckling may take place either in outwards or towards the core, relying upon the core stiffness in the compression and adhesive strength. Practically, with three-point bending, the inward wrinkling of upper skin takes place in the nearness of the central load. Via modeling the skin as a plate above an elastic base, Allen [14] gave the critical compressive stress ($\sigma_{fw}$), which caused the upper skin wrinkling as:

$$\sigma_{fw} = \frac{3}{12(3 - \nu_{cxy})^2(1 + \nu_{cxy})^2}E_{fx}^{1/3}E_{3}^{2/3} \\quad (12)$$

Where, ($E_{3}$) is the out-of-plane Young’s modulus, and ($\nu_{cxy}$) is the out-of-plane Poisson’s ratio of the core of honeycomb [15].

3.2. Failure of core

The honeycomb sandwich structures that are loaded in bending can fail owing to the failure of core. The relevant modes of failure are the failure by shear or by the indentation via the local crushing in the nearness of loads, as elucidated in the figure 4 [18].

3.2.1. Shear of core

The shear failure of core will take place in a honeycomb having a plastic-yield point if the principal stresses meet the criterion of yield. When the shear stress in core is bigger in comparison with the normal stress, the failure will take place if the shear stress ($\tau_c$) is equal or exceeds the honeycomb yield strength in shear ($\tau_{cy}$). The failure of core is given via:

$$\tau_c = \frac{V}{b\ell_c} = \tau_{cy} \\quad (15)$$

3.2.2. Indentation of core

It’s merely a problem if the loads are too localized and can be prevented when one ensures that this load is spread above a minimum area that is at least
Where, \((\sigma_{yc})\) is the core compressive strength.

### 3.3 Failure mode maps construction for the sandwich beams

Sections (3.1) and (3.2) have described different failure mechanisms that take place with the honeycomb sandwich panels, and the honeycombs mechanics are required for evaluating the loads of failure for each of such mechanisms. Then, the load \(P\) of failure can be manifested as a function of the properties of material and the parameters of beam \(P = f(t_f/L, \rho_c/\rho_s)\). For evaluating such function, the formulas for the core and skin stresses (Equations (2) and (8)) are substituted into different criteria of failure (Equations (9), (10), and (12)) as depicted in section (2) to provide the critical loads as briefed in table 1 [18].

A transition in the mechanism of failure will occur if two or several mechanisms possess the similar load. Such information can be viewed as diagram or map (failure-mode map). The highly significant transitions that one gets from equating the pairs of the equations of failure-mode include face yield–core shear, face yielding–face wrinkling, and face wrinkling–core shear.

#### Table 1. Summary of failure load for each mode

| Mode                  | Formula                                                                 |
|-----------------------|------------------------------------------------------------------------|
| Face yielding         | \(P = 4 \sigma_{fy} \xi \frac{t_f}{L}\)                               |
| Intra-cell dimpling   | \(P = \frac{8}{1 - v_f^2} \left(\frac{t_f}{a}\right)^2 E_f \times \frac{t_f}{L} \times \xi\) |
| Face wrinkling        | \(P = \frac{12 E_{fx}^{1/3} E_3^{2/3}}{12(3 - v_{cxy})(1 + v_{cxy})^2} \left(\frac{t_f}{L}\right)^{1/3} \xi \left(\frac{\rho_c}{\rho_s}\right)^{2/3}\) |
| Core share            | \(P = 3.4 E_f d \left(\frac{\rho_c}{\rho_s}\right)^3\)                  |
| Indentation           | \(P = 3.25 \sigma_{fy}^2 \left(\frac{\rho_c}{\rho_s}\right)^{5/3} \delta\) |

Where, \(\delta\) is the contact length between the upper skin and the central roller.

#### 3.3.1 Equation of transition between face wrinkling and face yielding

The face yielding/fracture takes place if

\[
\sigma_{fx} = \sigma_{fy}
\]  

(17)

Thus, from table (1) the load \((P)\) of failure is given via:

\[
P = 4 \sigma_{fy} \xi \frac{t_f}{L}
\]  

(18)

The face wrinkling (local buckling) takes place if

\[
\sigma_{fx} = \sigma_{fw}
\]  

(19)

Hence, from table (1) the load \((P)\) of failure can be stated as:
Putting Equations (18) and (20) equal to each other, one obtains that
\[
4 \sigma_{fy} \xi \frac{t_f}{L} = \frac{12 E_{fx}^{1/3} E_3^{2/3} t_f}{12(3 - v_{cxy})^2 (1 + v_{cxy})^2} \xi \left( \frac{\rho_c}{\rho_s} \right)^{2/3}
\]
(21)
Hence, the transition between face wrinkling and face yielding is given via this expression:
\[
\left( \frac{\rho_c}{\rho_s} \right) = \sqrt[3/2]{\frac{\sigma_{fy} (12(3 - v_{cxy})^2 (1 + v_{cxy})^2)^{-1/3}}{3 E_{fx}^{1/3} E_3^{2/3}}}
\]
(22)

3.3.2. Transition equation between face yielding and core share
The share failure of core takes place if
\[
\tau_c = \tau_{cy}
\]
(23)
Thus, from table (1) the load (W) of failure is given via:
\[
P = 3.4 E_f d \left( \frac{\rho_c}{\rho_s} \right)^3
\]
(24)
The face yielding/fracture takes place if
\[
\sigma_{fx} = \sigma_{fy}
\]
(25)
Thus, the load (W) failure is given via
\[
P = 4 \sigma_{fy} \xi \frac{t_f}{L}
\]
(26)
Putting Equations (24) and (26) equal to each other, one gets that
\[
3.4 E_f d \left( \frac{\rho_c}{\rho_s} \right)^3 = 4 \sigma_{fy} \xi \frac{t_f}{L}
\]
(27)
Hence, the transition between core share and face yielding is given via this expression:
\[
\frac{\rho_c}{\rho_s} = \sqrt[1/3]{\frac{4 \sigma_{fy} \xi}{3.4 E_f d}} \left( \frac{t_f}{L} \right)
\]
(28)

3.4.3 Equation of Transition between core share and face wrinkling
The face wrinkling (local buckling) takes place if
\[
\sigma_{fx} = \sigma_{fw}
\]
(29)
Hence, from table (1) the load (P) failure can be stated as
\[
P = \frac{12 E_{fx}^{1/3} E_3^{2/3} t_f}{12(3 - v_{cxy})^2 (1 + v_{cxy})^2} \xi \left( \frac{\rho_c}{\rho_s} \right)^{2/3}
\]
(30)
The share failure of core takes place if
\[
\tau_c = \tau_{cy}
\]
(31)
Thus, the load \((P)\) of failure is given via:

\[
P = 3A E_f d \left( \frac{\rho_c}{\rho_s} \right)^3
\]

Putting (30) and (32) equal to each other, one obtains that

\[
3A E_f d \left( \frac{\rho_c}{\rho_s} \right)^3 = \frac{12 E_{fx}^{1/3} E_3^{2/3}}{(12(3 - \nu_{cxy})^2(1 + \nu_{cxy})^2)^{-1/3}} L \xi \left( \frac{\rho_c}{\rho_s} \right)^{2/3}
\]

Hence, the transition between core shear and face wrinkling is given via this expression:

\[
\left( \frac{\rho_c}{\rho_s} \right) = \frac{3/7}{\sqrt[3]{\frac{12 E_{fx}^{1/3} E_3^{2/3}}{(12(3 - \nu_{cxy})^2(1 + \nu_{cxy})^2)^{-1/3}} L \xi \left( \frac{t_f}{L} \right)^{3/7}}}
\]

The transitions of failure modes in Equations (22), (28) and (34) are evinced in the failure mode map in figure 5 for Al [19].

![Figure 5. The failure mode map for an Al sandwich beam in 3-point bending](image)

3.5 Deflection of sandwich beam

Generally, the deflection \((\delta)\) of a simply-supported sandwich beam is the sum of the shear bending and the shear components [19] as:

\[
\delta = \delta_b + \delta_s = \frac{pL^3}{48EI_{eq}} + \frac{pL}{4(AG)_{eq}}
\]

3.6 Energy absorption of sandwich beam

Energy absorption depicts how much energy that the sandwich structure is able absorb through a certain crushing distance, \(d\). It can be mathematically determined via integrating the curve of load-displacement [20].

\[
E_a = \int_0^d P(\delta) d\delta
\]

From equation (35),

\[
\delta = p \left( \frac{L^3}{48(EI)_{eq}} + \frac{L}{4(AG)_{eq}} \right)
\]
Substituting Equation (37) in Equation (36) and integrating with respect to $\delta$ give

$$E_a = \frac{\delta^2}{2 \left( \frac{L^3}{4B(EI)_{eq}} + \frac{L}{4(AG)_{eq}} \right)}$$

(38)

4- Finite Elements Models for Three-point Bending

For better understanding the characteristics of the crashworthiness of honeycomb sandwich structures, a numerical model was generated to validate the theoretical solution in the present study. The finite element analysis was conducted in ANSYS/Static structure, as displayed in the figure 6. The mesh of the skins and the core were done individually and the entire model of honeycomb plate was assembled. The whole elements and nodes of FEM models were (28520) elements and (160360) nodes for the honeycomb sandwich plate. The boundary condition in the finite element model simulation was a simply supported.

According to failure mode map (figure5), the face yield failure mode is more likely. The range of static loads was applied at the center of sandwich beam gradually to determine the peak load. Also, von mises stress has been evaluated at each value of static loads and compared it with yield stress of upper facing (equation 9). The static load represents the peak load if von mises stress at upper facing equals to yield stress corresponding. Then, this peak load will be reapplied to calculate the maximum deflection and energy absorption.

Figure 6. Finite element model

5- Comparison Study

Figure 7 shows the comparison between theoretical and numerical results of crashworthiness (peak load, maximum deflection and energy absorption) of honeycomb beam structure with different honeycomb parameters. To establish this comparison 6 cases with different value of core high for each criterion were conducted. It is clear that the suggested analytical solution of honeycomb beam gives a good agreement compared with numerical results evaluated by using finite element technique, ANSYS program ver. 18. The maximum percentage discrepancy between the analytical and numerical results is about (5%).

a. Peak bending load  b. maximum deflection  c. Energy absorption

Figure 7. Comparison study
6- Results and Discussions

For investigating the honeycomb parameters effect upon the sandwich beam crashworthiness, theoretical solutions are used here. Honeycomb parameters and its variation were listed in table 2. Failure modes maps (Figure 5) were used to identify the failure load equation (table 1). The materials of facing and honeycomb are aluminum alloys AA3003 with mechanical properties as list in table 3. The length and width of sandwich beam are (l=210 mm) and (b=35 mm), respectively.

| Parameter               | Value (mm)                          |
|-------------------------|-------------------------------------|
| Core high (c)           | 5,10,15,20,25 and 30                |
| Cell size (Lc)          | 5,10,15,20,25 and 30                |
| Cell wall thickness (Tc)| 0.03 , 0.05 , 0.1, 0.5, 0.7, 1     |

| No | Specification   | Value |
|----|----------------|-------|
| 1  | Elastic modules| 71 GPa|
| 2  | Poisson ratio  | 0.33  |
| 3  | Density        | 2700 kg/m³ |
| 4  | Shear modules  | 26 GPa|
| 5  | Yield stress   | 280 MPa |

6.1 Honeycomb parameters effect

The results evaluated are included the peak bending load, energy absorption and maximum deflection of the honeycomb sandwich beam with various sizes of cell, thicknesses of cell and heights of core. Figure 8 shows the variation of the peak bending load with core height for different values of cell size and cell wall thickness. The sandwich structure bending stiffness was highly affected via the core height of honeycomb. This is due to that the sandwich beam second moment of inertia can be highly influenced via the core height of honeycomb; therefore the peak load of bending is increasing with the increase of core height and cell wall thickness. This is in consistent with the reference [20]which achieved an experimental study on the failure mechanics of sandwich panel under three-point bending.

On the other hand, the increasing of cell size leads to a decrease in the core density, which reduces the Young’s modulus and shear rigidity of honeycomb core. As result, the peak bending load is reduced.

Figure 9 reveals the variation of the maximum deflection with core height for different values of size of cell and thickness of cell wall. An increase in the core height and cell size results a decrease in the maximum deflection value, while the increase in the cell wall thickness leads to an increase in the maximum deflection.

To understand the influence of honeycomb parameters on the energy absorption, figure 10 shows the variation of the energy absorption with core height for different values of size of cell and thickness of cell wall. An increase in the core height and cell size results a reduction in the value of the energy absorption, while the increase in the cell wall thickness leads to an increase in the energy absorption.

To explore a wide range of variations of honeycomb parameters, the results are also corroborated via 3D surface plot evinced in figures 11, 12 and 13 for peak bending load, maximum deflection and energy absorption, respectively.
Figure 8: Variation of peak bending load with the core height and cell size for different cell wall thicknesses

Figure 9: Variation of peak bending load with the core height and cell size for different cell wall thicknesses
Figure 10: Variation of energy absorption with the core height and cell size for different cell wall thickness

Figure 11: 3D graph of peak bending load as a function of honeycomb parameter
6.2 Optimization Solution

The main goal of this study is to find the optimal parameters that give the maximum bending load, minimum deflection and maximum energy absorption, thus avoiding the failure of this structure due to static load. DOE software with RSM was utilized to achieve the analytical optimization and to obtain the optimum factors combinations for completing the needs as wanted. Accordingly, DOE software
was employed for the optimization goal developed on the results of theoretical solution (Equations 6, 35 and 38) for crashworthiness criteria, as a function of three input factors: cell size, cell wall thickness and core height. For establishing a new predicted model, an objective function, which is named “Desirability” to allow for a proper combining the goals, was estimated. This desirability must be maximized via the theoretical optimization, and its range is from (0) to (1) [21]. Figure 14 illustrates the optimum parameters for peak bending load, deflection and energy absorption, respectively.

![Figure 14. Optimum parameters](image)

### 7- Conclusions

1. According to DOE software with RMS, the optimal solution for maximum bending load, minimum deflection and maximum energy absorption were found at 29.2424 mm core height, 5 mm size of cell and 1 mm thickness of cell wall. Where, the optimal value of maximum bending load, minimum deflection and maximum energy absorption were 25310 N, 0.8976 mm and 9.9949 J, respectively.
2. Peak bending load is directly proportional with the cell wall thickness and core height but inversely proportional with cell size.
3. The increasing of cell size and core height leads to increase the deflection and energy absorption, but the increasing in cell wall thickness results a decrease in the deflection and energy absorption.
4. Core height has the largest effect on the crashworthiness properties of sandwich panels.
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