Research Article

AN INVESTIGATION INTO STATIC AND DYNAMIC CHARACTERISTICS OF SANDWICH BEAM

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Abstract— Sandwich beams offer designers a number of advantages, as the high strength to weight ratio, flexibility, high bending and buckling resistances. This can be used, particularly in aerospace, civil, and mechanical engineering designs and sustainable-energy applications. In the current study different configurations of sandwich beams were presented to investigate the effect of different sandwich beam elements, such as face and core material and their thicknesses, on the static and dynamic characteristics of sandwich beams such as: sandwich beam with single core, sandwich beam with double cores and sandwich beam with pre crack. Controlling the natural frequency of the sandwich beam was investigated by making holes in the sandwich beam elements. It can be found that: the use of core materials with lower Young’s modulus leads to an increase in the sandwich beam static deflection The natural frequency was increased with higher Young’s modulus and face and core thickness increased. The increase of the core thickness leads to an increase in the bar stiffness (decreasing deflection). From this analysis it is concluded that the optimum selection for the sandwich beam parameters depends on the application. The natural frequency of the sandwich beam can be controlled by making holes in the sandwich beam elements.

Index Terms— Sandwich beams, Core materials, Static and Dynamic Characteristics, Fracture mechanics

I. INTRODUCTION

Modern engineering requires the use of sophisticated and optimized structural design. One way to achieve this goal is to use materials in away that will optimize their inherent properties. An engineering application known as sandwich lamination is very suitable for this purpose. Sandwich materials are frequently used wherever high strength and low weight are important criteria. The most important application are found in the transport industry – such as in the aerospace, aircraft, automobiles, railroad and marine industries – where a high stiffness/weight and strength/weight ratio provides increased pay load capacity, improved performance and lower energy consumption [1-3]. A sandwich structure consists of three elements, face sheets, core and the adhesive interface layers. The faces carry in-plane and bending loads, while the core resist transverse shear forces and keeps the facings in place [4]. The many advantages of sandwich constructions, the development of new materials and the need for high performance and low-weight structures insure that sandwich construction will continue to be in demand [4-6]. The purpose of the core is to maintain the distance between the laminates and to sustain shear deformations [7], by varying the core, the thickness and the material of the face sheet of the sandwich structures; it is possible to obtain various properties and desired performances.

A systematic procedure was presented for comparing the relative performance of sandwich beams with various combinations of non traditional pairs of materials in three-point bending [8]. It was stated that various types of sandwich beams with foam or honeycomb cores are currently used in the industry. There are wide varieties of core material geometries currently in use. Among them, honeycomb, foam, balsa and corrugated cores are the most widely used. Usually honeycomb cores are made of Aluminium or of composite materials as Nomex, glass thermoplastic or glass-phenol. The other most commonly used core materials are expanded foams, which are often the most to achieve reasonably high thermal tolerance, though thermoplastic foams and Aluminium foam are also used. For the bonding of laminate and core materials, normally two types of adhesive bonding are commonly employed in sandwich construction, i.e., co-curing and secondary bonding. Characterization of sandwich materials has been carried out in detail in scientific literature.

Three types of Aluminium honeycomb sandwich beam specimens with different face sheet thicknesses were studied to investigate the effect of the face sheet thickness on the fatigue strength [9]. It was concluded that under the same applied bending loads, no apparent relationship exists between the face sheet thickness and the fatigue life of the studied specimens. Also, the main failure mode of the studied specimens is the debonding at the interface between the adhesive and the face sheet based on observations during the tests. The effect of the

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amount of adhesive for bonding face sheets and cores on four-point bending fatigue strength of Aluminium honeycomb sandwich beams was analyzed [9]. It was experimentally proved that the fatigue strength increases as the amount of adhesive increases. Optimization of sandwich structures for strength to weight under conditions of relatively concentrated loading is surprisingly complex. Effect of unequal face thickness on load resistance of sandwich beams was investigated [10]. The analysis showed that if a sandwich beam is optimized for strength to weight with equal face thickness, increasing the ratio of the thickness of the loaded face relative to the unloaded face does not significantly improve the strength. However, for example, for sandwich beams with other design configurations to increase the stiffness, increasing the relative thickness of the loaded face leads to significant improvements in the strength for a given weight.

The vibration characteristics of sandwich materials have, recently, drawn much attention. The dynamic parameters of a structure, i.e., natural frequency, damping and mode shapes, are determined with the help of vibration testing which provides the basis for rapid and inexpensive dynamic characterization of composite structures [11]. It was shown that modal testing in either a single mode or multiple modes of vibration can be used to determine elastic modules and damping factors of composites and their constituents under various environmental conditions.

The importance of material damping in the design process has increased in recent years as the control of noise and vibration in high precision high performance structures. A viscoelastic core layer, which has high inherent damping, between the two face sheets, can produce a sandwich structure with high damping [12]. Effects of thickness and delamination on the damping in honeycomb foam sandwich beams were analyzed. It was concluded that if the face sheet thickness increases, the damping in the low- and high-frequency ranges becomes lower, but it remains high in the middle-frequency range. If the thickness of the core increases, the damping is increased in the middle- and high-frequency ranges.

The dynamic properties of sandwich beams through experimental tests and using finite element methods were investigated [13]. It was found out that the faces dominate the stiffness of the sandwich beams, the natural frequencies were affected directly by the face materials and decrease with the increasing fibre orientation of the graphite/epoxy face laminates. Increasing the thickness of the cores increases both the frequencies and loss factors of the sandwich beams. Theoretical and experimental investigations on the effect of the sandwich beam elements on its dynamic characteristics are carried out on sandwich beams having different core materials, thicknesses and face thicknesses [14]. From this it can be concluded that, the sandwich beam elements must be carefully selected to increase the sandwich beam equivalent stiffness, damping and to decrease its equivalent mass.

The mechanical behaviour of sandwich structures depends on the properties of the facings, the core and the adhesive bonding of the core to the skins, as well as on the loading conditions and geometrical dimensions. Sandwich beams subjected to a combination of bending, shear and in-plane loading exhibits various failure modes. They include tensile or compressive failure of the facings, debonding at the core/facing interface, indentation failure under localized loads, shear core failure, wrinkling of the compression facing and global buckling. Initiation of a particular failure mode depends on the constituent material properties, geometry and type of loading [15]. A general review of failure modes in sandwich structures [16], were the various failure modes have been analyzed and critical failure loads have been determined. Four point bend tests are performed on sandwich beams with varying geometries to identify competing failure modes [17]. Failure initiation and propagation characteristics of sandwich beams and panels subjected to quasi-static and impact loadings were investigated to improve understanding of the material characteristics and energy absorption modes and facilitate the design of sandwich performance [18].

The value of the beam flexure rigidity, mass, natural frequency and modes of vibrations (static and dynamic performances of the beam) was presented through a model of sandwich beam with hole(s) in the direction, or in the cross direction, of the beam axis [19]. The generalized model can be used for the best design of the sandwich beams.

Debonded between the face sheet and the core is one of the common problems in sandwich structures. A number of studies have been undertaken to understand and characterize the interfacial fracture mechanics of sandwich beams. As shown previously [20], debond failure in sandwich specimens with a low-density foam core tends to occur cohesively in the foam, rather than cohesively in the resin bond layer, or interfacial between the resin layer and core, or resin layer and face sheet. However, for tough high-density cores, the debonded failure may actually occur interfacial between the resin layer and the core [21]. The fracture toughness of PVC foam core sandwich was investigated experimentally [21, 22]. It was stated that the face/core debonded toughness increased with increasing core density but did not depend on face material. Analysis of the influence of core thickness and crack depth on the plastic zone size was performed [23]. It was found that the Tilted Sandwich Debonded (TSD) specimens with core cracks near the F/C interface are highly mode I dominated. Hence, mode mixity is not expected to explain the elevation of the fracture toughness. But there is no clarification of why the interfacial toughness of two different core thicknesses of cracked sandwich beams was different, [24] explained how the core thickness could influence the material constraint and, hence, the measured toughness of the material. Beam models for determination of compliance and energy release rate of the sandwich double cantilever beam (DCB) specimen have been presented where first-order shear deformation laminated beam analysis, elastic foundation analysis, and finite element analysis were applied [25]. It was found that the beam analysis
provides a conservative estimate on the compliance and energy release rate. The elastic foundation model is in agreement with finite element analysis and experimental compliance data. A simple laminated beam analysis is presented for analysis of the propagation path of a core crack in symmetric and asymmetric double cantilever beam (DCB) sandwich fracture specimens with a polymer foam core [26]. The analysis could be useful for the design of DCB type of sandwich specimens to achieve certain desired crack propagation path, through predicting if the crack would propagate self-similarly, or if it would kink upwards or downwards, although it cannot predict the magnitude of the kink angle. The beam analysis can also predict the stability of crack path and equilibrium position of the crack after kinking.

From the literature survey, it is found that most of the previous works were focused on the study of static, dynamic, and fracture characteristics of sandwich beam. In the present work, several sandwich beam specimens were analyzed to assess the effect of the different sandwich beam elements, such as face and core materials and their thicknesses, on both static and dynamic characteristics of the sandwich beams. Different configurations of sandwich beams were presented and investigated such as: sandwich beam with single core, sandwich beam with double cores and sandwich beam with pre crack. Controlling the natural frequency of the sandwich beam was investigated by making holes in its elements. Finite element simulation using the finite element code ANSYS 11 aims at evaluating the configuration correction factor for various specimen geometries; particularly those used in the works of [14, and 26].

II. MATERIALS AND METHODS

2-1 Sandwich Beam with Single Core

Different cases of Sandwich beam will be investigated to calculate deflections, and natural frequencies of the sandwich beam. The core materials used are (epoxy, foam, and carbon epoxy) while the used face material is steel for all cases. Table 1 shows the mechanical properties of these materials. All specimens have the same length L=500mm, Figure 1 shows the geometry, and boundary conditions of the sandwich beam with single core and face steel material. The effect of using the different core materials mentioned above were investigated, where the core and the face thicknesses as well as the bar length and width were kept constant. The effect of the face thickness was also investigated to cover a typical range of sandwich faces ranging from (hf = 3mm to 45mm), keeping the core and face materials as well as the bar length, width and core thickness constant. Finally, the effect of the core thickness was also investigated to cover a typical range of sandwich cores (hc=20 mm to 300 mm) while the core and face materials as well as the bar length, width and face thickness were kept constant.

Static and dynamic analyses are carried out, where element type, real constant, material properties, and boundary conditions were all chosen, and the mesh generation was constructed for all cases. For static analysis, the left end of the specimen was constrained from translation along the specimen axes (X & Y axes), load (1 KN) was applied at the other end of the specimen in the opposite direction on the Y axis, and the static deflections were recorded for all cases. For dynamic analysis, the left end of the specimen was constrained from translation along the specimen axes (X & Y axes), and the modal analysis was carried out. The natural frequency, and mode shapes were calculated.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Mechanical properties of different materials of the used sandwich beam [14].</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Poisson’s ratio (ν)</td>
</tr>
<tr>
<td>Face material</td>
<td>Steel</td>
</tr>
<tr>
<td>Core material</td>
<td>Epoxy</td>
</tr>
<tr>
<td></td>
<td>Foam</td>
</tr>
<tr>
<td></td>
<td>Carbon epoxy</td>
</tr>
</tbody>
</table>

Figure 1 Geometry and boundary conditions of sandwich beam with single core
2-2 Sandwich Beam with Double Cores

Sandwich beam with double cores have cores consisting of two different materials upper and lower cores with different mechanical properties. Six cases with double cores of the following materials: epoxy, foam and carbon epoxy were studied. Figure 2 illustrate the boundary conditions and mesh generation of a sandwich beam with double cores. Table 2 shows these different cases where each specimen contains two different core materials from those mentioned above. The core position (upper and lower) is changed to study its effect on deflection in all cases, and natural frequency for two sandwich beams; foam, epoxy cores and epoxy, foam cores respectively. All specimens have the same dimensions presented in Table 3.

Static analysis was performed where the left end of the specimen was constrained from translation along the specimen axes (X & Y axes), and a unit load (1 kN) was applied on the other end of the beam Figure 2. The element type, real constant, material properties, and boundary conditions were all chosen, and the mesh generation was constructed for all cases in static and dynamic analysis. Deflection and natural frequency was calculated and recorded.

Table 2:
Different cases of Sandwich beam with double core and steel face material

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Upper Core material</th>
<th>Lower core material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>Foam</td>
<td>Epoxy</td>
</tr>
<tr>
<td>Case 2</td>
<td>Epoxy</td>
<td>Foam</td>
</tr>
<tr>
<td>Case 3</td>
<td>Foam</td>
<td>Carbon epoxy</td>
</tr>
<tr>
<td>Case 4</td>
<td>Carbon epoxy</td>
<td>Foam</td>
</tr>
<tr>
<td>Case 5</td>
<td>Epoxy</td>
<td>Carbon epoxy</td>
</tr>
<tr>
<td>Case 6</td>
<td>Carbon epoxy</td>
<td>Epoxy</td>
</tr>
</tbody>
</table>

2-3 Sandwich Beam with Pre Crack

The influences of core modulus, face thickness, core thickness, and crack length on the stress intensity factor, \( K_i \), of a typical double cantilever beam (DCB) sandwich specimen were investigated. A DCB specimen labelled "G/H100- Thick" [26] was selected for this parametric investigation. Table 4 shows the mechanical properties of this specimen and Figure 3 shows the geometry and boundary conditions of the DCB with pre crack. The influence of the core modulus, \( E_c \), on the stress intensity factor was investigated first to cover a typical range, \( E_c \) (500 MPa to 10 GPa), where the core and the face thicknesses, crack length as well as the beam length and width were kept constant. Next, the influence of the face thickness, \( h_f \), on the stress intensity factor was investigated; with, \( h_f \), ranging from (1.5 to 4.5 mm), while the core thickness, crack length, as well as the beam length and width were kept constant. The influence of core thickness was also investigated to cover a range from (20 to 60 mm), where the face thickness, crack length, modulus of elasticity and beam length and width were all kept constant. Finally, the influence of crack length, \( a \), ranging from (a=10 to 40 mm) on stress intensity factor was investigated, where the face thickness, core thickness and modulus of elasticity were kept constant.
Two-dimensional plane strain model was constructed using linear interpolation quadrilateral elements with two translational degrees of freedom at each corner node (“Plane 42”), using the commercial code ANSYS 11 [27]. One area was constructed for each face sheet while the core was constructed from two areas to facilitate generation of the crack region. Figure 4a shows a typical finite element model of a DCB specimen “G/H100-Thick”. The deboned was constructed as double (unconnected) nodes in a core layer slightly underneath (0.8mm) the upper face/core interface. For all DCB specimens, the initial crack length was 25mm. The right end of the specimen was constrained from translations and rotation along the specimen axes (x and Y axes), and a unit load) was applied on the other end

2-4 Controlling the Natural Frequency of the Sandwich Beam

Sandwich beams with steel faces and epoxy core material with a circular hole on the axis in the middle faces height upper and lower, in the middle of the core height, in one face at its middle height, or on the upper and lower axes of the interfaces of the beam, and holes with different sizes in one face will be investigated.

This investigation is applied to study the controlling of the sandwich beam natural frequency by making holes in the sandwich beam elements. All specimens have the same dimensions where the beam length= 500mm, beam width=100mm, face thickness= 25mm and core thickness= 50mm. Different hole sizes were considered where the hole diameters (D=10,20mm). Each one of the holes described above was located on the specific axis with different distances from the fixed side (400, 300,200,100 mm).

The modal analysis was applied by Finite Element Analysis (FEA) using commercial code ANSYS 11. The model, element type, boundary conditions and material properties were all chosen, and the mesh generation was constructed for all cases. Figure 4b shows boundary conditions and mesh generation of sandwich beam specimen with a hole on the axis of the core height and on the axis of the upper and lower interfaces with distance (X= 400 mm) from the fixed side.

Modal analysis is carried out while the left end of the specimen was constrained from translation along the specimen axes (X & Y axes). The natural frequency and mode shapes were calculated and recorded for all sandwich beams.

III. RESULTS AND DISCUSSION

3-1 Sandwich Beam with Single Core

3-1-1 Effect of Core Materials

For static analysis, figure 5 shows the deflection of sandwich beam with epoxy core. It is observed that the deflection has a maximum value at the free end of the beam equal to (0.04mm). For sandwich beam with foam and carbon epoxy cores the deflection has a maximum value at the free end of the beam equal to (0.242mm and 0.06 mm) respectively. From figure 6 it is observed that, the value of the static deflection has decreased about 80% from (0.25mm to 0.05mm) when Yang’s modulus changed from (0.34 to 100 GPa). It is concluded from figure 4 and figure 5 that the use of core materials with lower Young’s modulus leads to increasing sandwich beam static deflection and vice versa.
3.1.2 Effect of Face Thickness

Figure 9 shows the effect of face thickness on static deflection of sandwich beam with epoxy core. It can be observed that the value of the static deflection has decreased about 96% from (0.469 to 0.0167 mm) when the face thickness changed from (3 mm to 45 mm). It is concluded that as the face thickness increases the static deflection is decreased. This can be attributed to the increase of the area moment of inertia of the beam.

Figure 10 shows the effect of face thickness on the natural frequency of the first-mode of sandwich beam with foam cores. It is observed that, the natural frequency is increased about 47% (from 201.37 to 385.88 Hz) when the face thickness changed from (3 mm to 45 mm). It is concluded from Figure 10 that as the face thickness increases the natural frequency is increased. This can be attributed to the increase of the area moment of inertia of the beam.
3.1.3 Effect of Core Thickness

Figure 11 shows the effect of core thickness on static deflection of sandwich beam with epoxy core and it is observed that the value of the static deflection has decreased about 90% (from 8.66 E-2 to 8.16 E-3 mm) when the core thickness changed (from 20 to 300mm). It can be concluded that the increase of the core thickness leads to an increase in the bar stiffness (decrease deflection).

![Figure 11 Effect of core thickness on static deflection of sandwich beam with epoxy core](image1.png)

Figure 12 shows the effect of core thickness on first mode frequency of sandwich beam. It is observed that the value of the natural frequency increased about 60% from (234.4 to 597.24 Hz) when core thickness changed from (20 to 300mm). It can be concluded from figure 12 as the core thickness increases the natural frequency increases. This can be attributed to the increase of the stiffness of the core material.

![Figure 12 Effect of core thickness on the natural frequency of the first mode of sandwich beam with epoxy core](image2.png)

3.2 Controlling of the Natural Frequency of Sandwich Beam

In many cases, it is required to adjust the natural frequency of the sandwich beams to suit the operating conditions of the hole system. The natural frequency can be adjusted either by making holes on its elements or adding a mass on the sandwich beam. In the present study the first method (making holes) will be used. The effect of hole locations, positions, and dimensions will be investigated.

3.2.1 Holes in Upper and Lower Faces

Figure 13 shows the effect of hole position at upper and lower faces on the first mode frequency of the sandwich beam with epoxy core and hole size D= 20mm. It is observed that the natural frequency is increased about 20% from (F= 272.11 to 327.35 Hz) when the hole position changed (from 10 to 400mm) from the fixed end and the hole is located at the middle of upper and lower faces. This behavior can be expressed as shown in equation 1.

\[ y = 269.09 x^{0.1098} \]  

Where \( y \): Natural frequency  
\( x \): Hole position in the beam

![Figure 13 The effect of hole position at the middle of the face height upper and lower on the first mode of the sandwich beam with epoxy core and hole size D= 20mm](image3.png)

3.2.2 Holes in Soft Core

Figure 14 shows the effect of hole position at the middle height of the soft core on the first mode frequency of the sandwich beam with epoxy core and hole size D= 20mm. It is observed from figure 14 that the value of the natural frequency is increased about 0.36% from (F= 315.899 to 317.052 Hz) when the hole position changed (from 10 to 400 mm) from the fixed end and the hole is located at the middle height of the soft core. This behaviour can be expressed as shown in equation 2.

\[ y = 314.26 x^{0.0025} \]  

Where \( y \): Natural frequency  
\( x \): Hole position in the beam

![Figure 14 Effect of hole position at the middle height of the soft core on the first mode frequency of the sandwich beam with epoxy core and hole size D= 20mm](image4.png)
3.2.3 Holes at Interfaces between Upper and Lower Faces and Soft Core

Figure 15 shows the effect of hole position at the interfaces between upper and lower faces and soft core on the natural frequency of the first mode frequency of the sandwich beam with epoxy core and hole size D=20mm. It was observed from figure 15 that the value of the natural frequency is increased about 2.6% from (F=314.2 to 322.5 Hz) when the hole position is changed (from 10 to 400 mm) from the fixed end and the hole is located at the interfaces between upper and lower faces and soft core. This behaviour can be expressed as shown in equation 3.

\[ y = 0.9197x^2 - 4.1094x + 315.44 \]  \hspace{1cm} (3)

3.2.4 Holes at One Face (Upper or Lower)

Figure 16 shows the effect of hole position at the middle height of one face (upper face) on the natural frequency of the first mode frequency of the sandwich beam with epoxy core and hole size, D=20mm. It was observed from figure 16 that the value of the natural frequency is increased about 9.6% from (F=293.34 to 321.63 Hz) when the hole position changed (from 10 to 400 mm) from the fixed side and the hole is located at the upper or lower face. This behaviour can be expressed as shown in equation 4.

\[ y = -0.8172x^2 + 11.653x + 281.5 \]  \hspace{1cm} (4)

3.2.5 Holes with Different Sizes at Upper or Lower Face

Figure 17 shows the effect of hole position with different hole sizes at the middle height of one face (upper face) on the natural frequency of the first mode frequency of the sandwich beam with epoxy core and hole size, (D=10, and 20mm). It is observed from figure 17 that the value of the natural frequency for hole size, D=10mm is increased about 1.43% from (F=313 to 317.5 Hz), when the hole position changed (from 10 to 400 mm) from the fixed end and the hole is located at the middle height of upper or lower face. This behaviour can be expressed as shown in equation 5.

\[ y = 0.89x + 312.51 \]  \hspace{1cm} (5)
From the above analysis it is concluded that the natural frequency of the sandwich beam can be controlled by making holes in its elements. Table 4 shows concluded remarks on the effect of hole position, location, and size on the natural frequency of the sandwich beam. This difference may due to the value of the Young’s modulus of the face and core materials where the face material is stiffer than the core material.

### 3.3 Sandwich Beam with Double Cores

Six cases were investigated to study the effects when replacing position of the core materials with each other on both free end deflection, and natural frequency of sandwich beam specimens. Figure 18 and figure 19 show sandwich beam with (foam, epoxy), (epoxy, foam) cores. From these figures it is observed that when the core with the higher Young’s modulus is the upper one, the deflection is decreased slightly than the other case (the core with the lower Young’s modulus is the upper one), these are occurred for all cases.

#### Table 4

Effect of hole position, location and size on natural frequency of the sandwich beam

<table>
<thead>
<tr>
<th>Hole Position</th>
<th>F*</th>
<th>Behavior</th>
<th>Effect</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holes in Upper and Lower Faces</td>
<td>20%</td>
<td>(Y=269.1x^{0.198})</td>
<td>large</td>
</tr>
<tr>
<td>Holes at one face</td>
<td>9.6%</td>
<td>(y=-0.81x^2+11.65x+281.5)</td>
<td>moderate</td>
</tr>
<tr>
<td>Holes at interfaces</td>
<td>2.6%</td>
<td>(y=0.919x^2-4.11x+315.44)</td>
<td>slight</td>
</tr>
<tr>
<td>Hole with different size</td>
<td>1.43%</td>
<td>(y=0.89x+312.5)</td>
<td>slight</td>
</tr>
<tr>
<td>Hole in soft core</td>
<td>0.34%</td>
<td>(y=314.26x^{0.0025})</td>
<td>Very slight</td>
</tr>
</tbody>
</table>

F* Natural Frequency of the Sandwich Beam

The effect of exchanging positions of the core materials with each other on natural frequency also was investigated for two sandwich beams. They are: sandwich with foam and epoxy cores and the other having epoxy and foam cores. It is observed from figure 20 that, replacing position of the core material foam and epoxy didn’t make any changes of the values of the natural frequency for the two sandwich beams and this may due to symmetry.

![Figure 18](image1.png)

**Figure 18** Deflection of sandwich beam with foam and epoxy cores.

![Figure 19](image2.png)

**Figure 19** Deflection of sandwich beam with epoxy and foam cores.

![Figure 20](image3.png)

**Figure 20** Effect of replacing position of the core materials on natural frequency for sandwich beam with (foam and epoxy) and (epoxy and foam) cores.
3.4 Sandwich Beam with pre crack

3.4.1 Effect of Core Young’s Modulus

Figure 21 shows the relationship between the core modulus and the stress intensity factor where $E_c$, was varied from (50 MPa to 10 GPa), and the core, the face thicknesses, crack length as well as the bar length, and width were kept constant. It is observed that the value of the stress intensity factor has increased about 86% (from 30.836 to 226.9 $\text{pa m}^{0.5}$) when Young’s modulus is increased (from 50 Mpa to 10 Gpa ). So, increasing the elastic modulus increases the stress intensity factor.

![Image of Figure 21: Effect of core modulus on stress intensity factor](image1.png)

**Figure. 21** Effect of core modulus on stress intensity factor of G/H100-Thick” sandwich beam.

3.4.2 Effect of Face Thickness

The influence of face thickness, $h_f$, on the stress intensity factor was investigated; $h_f$, ranging from (1.5 to 4.5 mm), where the core thicknesses, crack length, beam length and width were kept constant. Figure 22 shows the relationship between the face thickness and the stress intensity factor and it is obvious from this figure that the stress intensity factor has decreased about 73% (from 77.129 to 20.804 $\text{pa m}^{0.5}$) when the face thickness is increased from (1.5 to 4.5 mm) . So, it concluded that as the face thickness increases, the value of the stress intensity factor decreases.

![Image of Figure 22: Effect of face thickness on stress intensity factor](image2.png)

**Figure. 22** Effect of face thickness on stress intensity factor of G/H100-Thick” sandwich beam.

3.4.3 Effect of Core Thickness

The influence of core thickness also was investigated to cover a range from (20 to 60 mm), where the face thickness, crack length, modulus of elasticity, beam length and width were kept constant. Figure 23 shows the effect of core thickness on stress intensity factor. It is observed that the value of the stress intensity factor has decreased about 0.03% (from 43.83to 42.35$\text{pa m}^{0.5}$) when the core thickness is increased from (20 to 60 mm). It is concluded that, as the core thickness increases the stress intensity factor is decreased slightly.

![Image of Figure 23: Effect of core thickness on stress intensity factor](image3.png)

**Figure.23**Effect of core thickness on stress intensity factor of G/H100-Thick” sandwich beam.

3.4.4 Effect of Crack Length

The influence of crack length, “a”, on the stress intensity factor was investigated to cover a range from (10 to 40 mm) where the face thickness, core thickness, beam length, and modulus of elasticity were kept constant. Figure 24 shows the effect of crack length on stress intensity factor, and it is observed that the value of the stress intensity factor has increased about 60% (from 24.192 to 61.812 $\text{pa m}^{0.5}$) when the crack length is increased (from 10 to 40 mm). It was concluded that, as the crack length increases the value of stress intensity factor is increased.

![Image of Figure 24: Effect of crack length on stress intensity factor](image4.png)

**Figure. 24** Effect of crack length on stress intensity factor of G/H100-Thick” sandwich beam.

IV. CONCLUSIONS

From this work it can be concluded that: for sandwich beam with single core, the use of core materials with lower Young’s modulus leads to an increase in the sandwich beam static deflection. Also the use of core materials with higher Young’s modulus leads to an increase of the natural frequency of the sandwich beam. The increase of face thickness leads to decreasing the static deflection. The natural frequency was increased as the face thickness increased. The increase of the core thickness leads to an increase in the bar stiffness (decreasing deflection). As the core thickness increases the natural frequency increases. From this analysis it is concluded that the optimum selection for the sandwich beam parameters depends on the application.
The natural frequency of the sandwich beam can be controlled by making holes in the sandwich beam elements. Changing the position of the hole along the beam axis, in the middle height of the two faces upper and lower, the middle core height, interfaces between upper and lower face and soft core, one face upper or lower and different hole sizes in upper or lower face, has been found to have effect on the natural frequency modes of the sandwich beam. It was concluded that when the hole position changed along the beam length from the fixed end to the free end the natural frequency is decreased. The greater effect on the natural frequency occurs when; the hole is located at the middle height of the faces upper and lower. When the hole is located at one face upper or lower the natural frequency is increased by moderate effect. The minimum effect on the natural frequency was occurred when the hole is located at the soft core of the sandwich beam. This may due to the value of the Young’s modulus of the face and core materials where the face material is stiffer than the core material. For sandwich beams with double cores, replacing the core position upper and lower has minimum effect on deflection and no effect on the natural frequency. For sandwich beams with pre crack, as the core modulus increases the stress intensity factor is increased where the core and face thicknesses, the crack length, the bar length as well as the width were all kept constant. As the face thickness increases the value of the stress intensity factor, $K_t$, is decreased. As the core thickness increases the stress intensity factor is decreased. As the crack length is increased the value of the stress intensity factor is increased. From this analysis it is concluded that the optimum selection for the sandwich beam parameters depends on the application.

REFERENCES


