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CONSTRAINED LAYER DAMPING OF HONEYCOMB COMPOSITE STRUCTURES

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CONSTRAINED LAYER DAMPING OF HONEYCOMB COMPOSITE STRUCTURES

A Thesis
Presented to
The Graduate School of
Clemson University

In Partial Fulfillment
Of the Requirements for the Degree
Master of Science
Mechanical Engineering

By
Rohit Telukunta
August 2011

Accepted by:
Dr. Lonny L. Thompson, Committee Chair
Dr. Georges M. Fadel
Dr. Gang Li
ABSTRACT

Composite sandwich structures have replaced homogenous dense solids in many applications due to their advantages of high stiffness to weight ratio, and higher damping characteristics. Higher damping in engineering applications is desirable to reduce structural vibrations. The application of a viscoelastic layer between two thin face sheets gives rise to the concept of constrained layer damping which is an effective technique to achieve increased damping in engineering applications.

Honeycomb cellular structures are often used for the core in sandwich construction because of their low density and high stiffness properties. Regular honeycombs are defined by conventional hexagonal geometry, which gives rise to effective transversely isotropic properties. Auxetic honeycombs have cellular geometry defined such that their effective Poisson’s ratio is negative, and have potential for increased shear modulus and nonconventional design compared to their regular counterparts.

In this study, the damping nature of auxetic and regular honeycombs cores within a sandwich plate structure with equal mass density is studied using finite element analysis. A new concept of constrained layer damping is introduced within the honeycomb cell walls, making the honeycomb core, itself, a composite structure. By introducing the composite honeycomb core between two thin face sheets in the macro sandwich structure, further increases in damping can be achieved. The thickness of the constraining layers is defined such that the effective stiffness is increased for the same
mass of a sandwich plate with homogeneous honeycomb core. Comparisons are made for both quasi-static cyclic loading and dynamic analysis subjected to impact loads. The amplitude of loading is defined at a level such that the yield stress within the base materials is not exceeded. Dissipation energy at the end of the loading step in the finite element analysis is used to quantify the structural loss factor.

Results show higher damping is achieved with the novel concept of constrained layer viscoelastic damping in honeycomb cell walls. In the case of out-of-plane loading direction, sandwich plates with composite auxetic honeycomb core gives higher damping over homogeneous honeycomb core sandwich plates and its regular honeycomb counter parts. However, when loaded in the in-plane direction, a condition was found where sandwich plates with homogenous auxetic honeycomb core gave higher damping than with a composite core and its regular counter parts, suggesting that further development is needed to optimize the relative thicknesses of the constraining layer in the honeycomb cell walls.
DEDICATION

This thesis is dedicated to my lovely parents, Mr. T. Anil Kumar and Mrs. T. Vidya Devi, my brothers T. Abhishek and T. Aditya.
ACKNOWLEDGMENTS

First of all, I am thankful to god for his blessings on me throughout my 23 years of journey. I would like to offer my deepest gratitude to my research advisor Dr. Lonny L. Thompson for assigning me this work; offering his support, assistance and guiding me throughout my journey at Clemson University and without whom this thesis wouldn’t have been completed.

Secondly, I would like to thank my committee members Dr. Georges Fadel and Dr. Gang Li for their support. The computational resources provided by the Clemson University also contributed significantly for the completion of this research.

Finally, I would like to acknowledge my friends Shashank Bezgam, Prashanth Polisetty, Sreekanth Reddy Gondipalle and Raviteja Katragadda and my colleagues Nikhil Kumar Seera, Vineeth Kumar Jampala, Mallikarjun Veeramurthy and Nataraj Chandrasekhar who gave me a pleasant Clemson experience through academic and personal collaborations. I have truly enjoyed their friendship and support.
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CHAPTER 1: INTRODUCTION & LITERATURE REVIEW

Many studies have investigated methods to increase the damping capacity of systems by employing either active or passive damping methods [1]. Honeycombs, which are a particular form of cellular solids, have gained importance because of their exceptional mechanical properties like low effective density, high stiffness, and low thermal conductivity compared with conventional dense, solid materials. In addition, damping treatments within the honeycomb structure has potential for improved vibration control. The advantages of honeycombs make them a potential solution for many current design problems.

The construction of honeycomb from the micro level of the base materials to the macro structure of the composite sandwich plate can be described by the hierarchy shown in Figure 1:1. In this framework, the core of the sandwich structure can be considered the Meta structure. The behavior of the Meta structure can be controlled by varying the intermediate Meso properties according to design requirements. The Meso I scale is considered to be the intermediate composite layers for the honeycomb cell walls, while the Meso II middle level scale is defined by the unit cell geometric properties of the cellular honeycomb core structure.
Figure 1:1 Multi scale progression of a honeycomb sandwich plate

The multi-scale progressions from Micro to Meso to Meta and Macro properties are defined with examples of the variables within the context of honeycomb cellular structures in Table 1:1.

Table 1:1 Definitions and Examples of Levels in a Multi-Scale Honeycomb Sandwich Structure

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<th>Definition</th>
<th>Examples</th>
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<td>Micro</td>
<td>Constitutive material properties of host materials</td>
<td>Host material’s moduli, poisson’s ratio and density</td>
</tr>
<tr>
<td>Meso I</td>
<td>Intermediate composite layers for honeycomb cell walls</td>
<td>Effective properties of a composite beam</td>
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<tr>
<td>Meso II</td>
<td>Middle level unit cell geometric structure designed to control the macro properties</td>
<td>Cell height, cell length, cell wall thicknesses, cell angle</td>
</tr>
<tr>
<td>Meta</td>
<td>Effective properties of Honeycomb core structure</td>
<td>Effective In-plane and out-of plane properties of core</td>
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1.1 Damping

Mass, stiffness and damping are the three critical parameters that regulate the dynamic response of the system. One way to quantify the amount of damping properties is by performing a cyclic loadings, and measuring the amount of energy dissipation. Damping in a vibrating system can be defined as the loss of mechanical energy into heat [2]. Damping helps to control the propagation of vibrational waves in the structure. Higher damping can also help to reduce the vibration amplitude of structures at resonance frequencies. Increased damping results in fast decay of vibrations, reduced stresses and lower structural response [3]. As mentioned earlier, damping in a vibrating system can be achieved by various means. The damping methods which are currently employed in engineering applications can be categorized as (a) internal or material damping, and (b) external or system damping. Internal damping is achieved with the conversion of the vibration energy into heat within the volume of the material. One way to control vibration is to use materials with large intrinsic damping properties such elastomers with viscoelastic behavior. Another internal damping method is to use structural or system damping, which uses the orientation of and composition of structures to control vibration. External damping includes acoustic radiation damping, coulomb friction damping, joints and boundary damping. Also, the damping of bonded structures tends to be lower than that of structures with bolted and riveted joints [3,4].
1.2 Measure of Damping

Damping of a system can be measured using a loss factor parameter defined by [2]:

\[ \eta_s = \frac{D_s}{2\pi U_s} = \frac{E'}{E} = \tan \phi \]

Relating \( D_s \), the energy dissipated per cycle of a sinusoidal test, and \( U_s \), the strain energy at peak amplitude. The loss factor can also be related to the ratio of the imaginary and real parts of a complex modulus, denoted by \( E'' \), \( E' \), and the \( \tan \phi \) represents the phase difference between stress and strain. In addition, the loss factor can be related to the damping ratio (\( \zeta \)) for an oscillator, by \( \eta_s = 2\zeta \), where \( \zeta = C / C_c \) and \( C \) and \( C_c \) are the viscous damping and critical damping. In the current study, the loss factor is computed from a finite element analysis computed from the energy dissipation divided by the strain energy.

1.3 Damping in Composite Laminated Plates

Composite laminated beam and plate structures are used in engineering applications because of their high specific stiffness, strength and have attracted interest in methods for improving their damping capacity. Studies have found that the damping nature of composite structures depend on the micromechanical properties of constituent materials, the composite layout schemes. In general, damping of a composite plate or beam can be increased my varying the materials, geometry layout, or by introducing a viscoelastic material. Damping in composites generally exhibits an opposing trend to
stiffness and strength, i.e. damping is a minimum in the direction fibers and maximum in the transverse and shear directions [5]. An advantage of composite plates is the flexibility to design the structure and the material orientation to achieve optimal performance [6].

1.4 Viscoelasticity

Viscoelasticity is defined as a material response that exhibits characteristics of both a viscous fluid and an elastic solid [7]. A viscoelastic material combines these two properties—it returns to its original shape after being stressed, but does it slowly enough to oppose the next cycle of vibration. The degree to which a material behaves either viscously or elastically depends mainly on temperature and rate of loading frequency [7].

As mentioned earlier, the vibration in a dynamic system can be reduced by various means. Studies have been done in introducing a viscoelastic layer in composite structures to enhance damping [2, 8, 9, 10]. Viscoelastic materials are capable of storing strain energy when deformed, and provide damping dissipating a portion of the stored energy through hysteresis [2]. The vibration analysis of a beam with a viscoelastic layer was first conducted by Kerwin and colleagues [8, 9].

For a viscoelastic material under cyclic loading, the strain is not in phase with stress. As discussed earlier, the tangent of the phase angle defines the loss factor ($\eta$) which is a measure of damping. Also discussed earlier, the loss factor is also defined as the ratio of the energy dissipated to that stored in the material [2].
The damping loss factor can also be quantified by the ratio of an imaginary to real modulus. For a material subject to a shear relaxation test, the loss factor is \( \eta = \frac{G_i}{G_s} \) \[11, 48\]. The viscoelastic material properties can be defined by fitting material parameters using a Prony series for the shear relaxation modulus. The Prony series expansion of the shear relaxation modulus is calculated using the Maxwell model with the expression

\[
G_R(t) = G_0\left(1 - \sum_{k=1}^{N} g_k \left(1 - \frac{1}{e^{\frac{t}{\tau_k}}} \right)\right),
\]

where, \( G_0 \) is the instantaneous relaxation modulus, \( g_k \) and \( \tau_k \) are the material constants obtained by curve fitting the shear relaxation modulus within the limits \( t \to \infty, G_R \to G_0 \) and \( t \to \infty, G_R \to G_\infty \)

where,

\[
G_\infty = G_0\left(1 - \sum_{k=1}^{N} g_k \right)
\]

The complex frequency dependent shear modulus with storage and loss modulus is given by,

\[
G^*(\omega) = G_s(\omega) + iG_i(\omega)
\]

where,

\[
G_s(\omega) = G_\infty + G_0 \sum_{k=1}^{N} \frac{g_k (\tau_k \omega)^2}{1 + (\tau_k \omega)^2}
\]

\[
G_i(\omega) = G_\infty + G_0 \sum_{k=1}^{N} \frac{g_k (\tau_k \omega)^2}{1 + (\tau_k \omega)^2}
\]
1.5 Constrained Layered Damping

The concept of a viscoelastic layer to increase the damping of structures is found in many applications, e.g. automobile, aerospace, ships, machine tools, turbines, electrical and optical equipment [3, 10]. The viscoelastic layer is introduced between two face sheets with the effect of producing high damping [13]. Due to shear deformation occurring in the viscoelastic layer, constraint layer damping treatments are known to yield significantly larger system damping compared to unconstrained layer damping, for the same mass of damping material [10].

Energy is dissipated by direct and shear strains when a damping layer is attached to a vibrating structure. The damping capacity of a sandwich plate is increased by altering the material and geometric configurations of the core and the face sheets according to the design requirements. The sandwich plate undergoes flexural vibrations constraining the damped core to undergo shear deformation causing energy dissipation and flexural motion to damp [3, 14]. Ross-Ungar-Kerwin developed a model to study the damping in a sandwich plate [8, 9, 14-16]. Ungar derived an expression to calculate the loss factor for sandwich beams using the shear and structural parameters [16].

In the case of homogeneous beams and plates subjected to bending, the direct strain increases linearly with the distance from the neutral axis for unconstrained layers. For the same case the shear stress is largest at the neutral axis and zero on the free surfaces for constrained layers. Ungar [15] derived an expression to calculate the system damping for various thickness of viscoelastic layers used. The loss factor is highest when a three layer sandwich structure is symmetric about the neutral axis.
Studies have been conducted on beams with constrained layer treatment which is not applied over the entire length of the beam. Nokes and Nelson [10] stated that stiff viscoelastic layer gives higher loss factor for the case of a partially covered beam, compared to a full one. Plunkett and Lee [17] have carried out an analysis for determining an optimal length of constraining layer, which may give a high value of system damping. Ruzicka [18] has done extensive research on viscoelastic shear damping and structural damping and concluded that the loss factor is independent of stress level for pure viscoelastic materials [18].

Unger and Kerwin proposed strain energy (MSE) model to include the damping capacities of all the elements in the system and the damping of the material was characterized by the ratio of the energy dissipated in each element to the energy stored in the material [19]. Jhonson and Kienholz took the MSE model to the next level and developed a finite element model to predict the damping in structures with constrained viscoelastic layers [20]. Hwang and Gibson studied damping in composite materials and structures at both macro-mechanical and micromechanical levels using the MSE method [21-22]. Lazan was the first person to study the frequency dependence property of viscoelastic damping [23]. The rotation and shear deformation of the sandwich structure play a vital role in the middle frequency core. The behavior of the sandwich structure is determined by pure bending in the low frequency region. The bending of the face sheet plays an important role in the high frequency region. In the middle frequency region, the rotation and shear deformation of the core was important [23].
Viscoelastic materials have been used to enhance the damping in a structure in three different ways, including free-layer damping treatment, and constrained-layer or sandwich-layer damping treatment [1, 7]. In free layer damping, the damping material is either sprayed on the structure or bonded using a pressure-sensitive adhesive. The concept of the free layer damping is applied for undercoating of an automobile, floor panel [7]. The current study focuses on introducing the concept of constrained layered damping (CLD) to honeycomb core when sandwiched. A constrained-layer damping (CLD) system is obtained by laminating a viscoelastic damping layer between two stiff elastic layers as show in Figure 1:2.

A constrained layer damping structure can be formed by various thickness of base, viscoelastic (damping) and constrained layers. The layers are either bolted or rivet and glued. In CLD the damping layer is usually totally covered by the top constraining layer to avoid it from abrasion. The constrained-layer damping is more effective than the free layer design since more energy is consumed and dissipated into heat in the work done by the shearing mode within the viscoelastic layer [1, 7].
Studies have also concluded that, a symmetric configuration in which the base and constraining layers having same the thickness has maximum shear deformation in the core layer, and thus more damping.

Figure 1:3 Symmetry in a constrained layer construction

In the Figure 1:3 $a$ is the distance from the neutral axis to the center of the individual layers. For the symmetric case, the distance is the same for both the top and bottom layers.

Damping in a sandwich plate with a viscoelastic layer depends on the length of the constrained layer. Viscoelastic layer when laid on the surface of structural members under cyclic loading; experience cyclic strains leading to energy dissipation only due to shear deformation leading to higher damping in constrained members than unconstrained members [24]. Studies conducted by Kerwin [8], Parfitt [25] Plunkett and Lee [17] also stated that the amount of damping for a given viscoelastic layer depends on stiffness, length of constraining
layer. There is always an optimum length for the elements of the constraining layer for a given combination of constraining and viscoelastic layers for high system damping [17]. Studies were also conducted on beam with full and partial coverage of constrained layer treatment. Nokes and Nelson [26] from their studies, by varying the coverage of constrained layer said that, a stiff viscoelastic layer gives a higher system loss factor for the case of partially covered beam, compared to a fully covered beam. Studies [27,28] concluded that multi layered structures have higher damping than three layer sandwich beam and plates in terms of shear modulus of the core.

1.6 Honeycombs

As discussed earlier cellular solids allow for design of light weight, stiff components such as sandwich panels used in automobiles and airplanes. The other property of cellular solids which make them attractive to replace dense solids is its low thermal conductivity [29]. The mechanics of honeycombs and their application have been studied over the past four decades [29, 32-38]. A cellular solid made of a base material having a viscoelastic energy loss is studied in [30, 31].

The in-plane strength of honeycomb structures is lower than that for out-of plane loading because bending deformation predominates in the first case, while axial deformation in the second. In the in-plane loading of honeycombs, the cell wall bends. In compression, cell walls collapse once they exceed their critical strain causing buckling. In tension, plastic yielding, creep, or brittle fracture, depending on the material can occur with increased load. On the other hand, when the cell wall is loaded out-of-plane, the cell walls either compress
or extend and the moduli and collapse stresses are much higher than in-plane. Bending of the cell wall when loaded in-plane causes lower stiffness compared to out-of-plane. The advantage of high stiffness in the out-of-plane direction of the cell wall is often utilized in designing honeycomb core sandwich panels [29].

**Regular vs. Auxetic Honeycomb:**

Honeycomb cells are available in many shapes such as rectangle, square, circular, regular hexagon with positive and negative cell angles. The current study focuses only on behavior of honeycombs with positive and negative cell angles and they are termed as regular and auxetic honeycombs correspondingly as shown in Figure 1:4.

![Honeycomb structures](image)

**Figure 1:4 Honeycomb structures**

Regular honeycombs are widely used for engineering applications since they have a good balance of high strength and stiffness with relatively low weight. Regular honeycombs also have a positive effective Poisson’s ratio. Other honeycomb geometries have been studied with the goal to improve their performance for applications such as flexible
structures, energy absorption, negative Poisson’s ratio, and damping, by varying the geometric parameters such as height (h), length (l), thickness (t) and cell angle (θ).

As noted in [29, 41], the value of the in-plane Poisson’s ratio depends on the cell geometry. Auxetic honeycombs are a result of changing the cell angle to be negative which changes the shape of the cellular structure [29, 40, 42]. Auxetic honeycombs also replace regular honeycombs in structural applications which require low cut-off frequency [18, 42].

Auxetic materials, with negative Poisson’s ratio, expand in all directions when pulled only in one direction leading to increase in the total volume. Evans [41] was the first person to explain the concept of auxetic material through his studies. Behavior of conventional (regular) and auxetic honeycombs under tension and compression is illustrated Figure 1:5.

Figure 1:5 Behavior of Honeycombs under tension and compression [51]
1.7 Honeycombs in sandwich plates

A sandwich panel is produced when a low density honeycomb core is sandwiched between two high rigid face sheets and bonded together [3, 45]. Sandwich plate exhibits high bending stiffness (flexural rigidity) for lower mass and has a low shear modulus which makes them a better source of damping [3]. Based on the design requirements the thickness of the core and face sheets are varied [37, 43].

In general, sandwich panels are loaded in the out-of-plane direction. The role of the honeycomb core in a panel is to carry shear. The stiffness of the honeycomb core in bending depends on the direction of loading and deformation of cells and also the material properties applied to the cell wall play a vital role in cell deformation [29, 44]. Sandwich panels used for current applications in majority use regular honeycombs. Study conducted by Evans [41] states that, when the cell bends in the out-of-plane direction it produces a saddle shaped curvature due to in-plane Poisson’s ratio being positive.

1.8 Manufacturing of Honeycombs

Materials like titanium, nickel, stainless steel, aluminum, fiberglass, carbon, polyurethane, polycarbonate, and alloys or base materials are used in honeycomb core manufacturing based on design requirements. The arrangement of honeycomb cells in the honeycomb core manufacturing depends on its application like absorb crushing loads, shear loads, stiffness, and compressive strength. The arrangement of honeycomb cell is generally done in (a) Hexagonal core, (b) Ox-core™, (c) Reinforced honeycomb core, (d) Flex core, (e) Double-Flex™, and (f) Tube core methods [52]. Honeycombs are generally manufactured
using (a) expansion process, (b) corrugated process, and (c) casting process.

**Expansion Process**

The honeycomb manufacturing process by the expansion method begins with the stacking of sheets of the substrate material on which adhesive node lines have been printed. The adhesive lines are then cured to form a HOBE (honeycomb before expansion) block. The HOBE block expands after curing to give an expanded block. The HOBE slices can be cut from the HOBE block to the appropriate dimensions and subsequently expanded. Slices can be expanded to regular hexagons, under expanded to 6-sided diamonds, and overly expanded to rectangular cells. The expanded sheets are trimmed to the desired dimensions in the ribbon direction and transverse to the ribbon [49].

![Figure 1:6 Stacking of layers and adhesives](image)

**Figure 1:6 Stacking of layers and adhesives [52]**

![Figure 1:7 Expanded Honeycomb core](image)

**Figure 1:7 Expanded Honeycomb core [52]**

**Corrugated Process**

The corrugated process of honeycomb manufacture is normally used to produce
products with higher density range. In this process, the corrugated sheets are stacked into blocks, and bonded by welding later this core is sliced to the desired core thickness.

Figure 1:8 Schematic illustration of corrugated process [49]

Casting Process

The casting process uses a wax pattern of the honeycomb structure along with the facesheets manufactured by the process of injection molding. The honeycomb structure is generally made by rapid prototyping process. The slurry along with the binders is poured into the pattern and allowed to solidify [49].

The other manufacturing process are used in the honeycomb manufacturing are sheet metal forming, milling and prismatic topologies methods [52].
CHAPTER 2 : RESEARCH OBJECTIVES

A goal of this study is to design and control material and system damping in the case of honeycomb sandwich composite structures when subjected to out-of-plane, and in-plane, quasi-static and dynamic loading. A new concept is developed in which the traditional homogeneous honeycomb cell walls are replaced with a composite laminate; specifically a viscoelastic material constrained between layers is proposed for the cell walls. The goal of this new concept is to increase damping for vibration control.

The advantages of constrained layer damping (CLD) is taken to a meso-level by introducing the concept of constrained layer damping within the honeycomb cell wall; this is a new design concept which can be considered an invention disclosure. The advantage of having high damping capacity in constrained layer damping over free and tuned viscous damping methods has led to the motivation of this study. On having a composite honeycomb cell wall and when the honeycomb core is sandwiched between high rigid face sheets forms a multi-layered sandwich structure. Previous studies have shown that multilayered beams or plate have higher damping capacity over three layered counterparts [27,28].

In addition, previous studies have shown that auxetic honeycomb having the advantage of negative Poisson’s ratio has potential for increased damage resistance, increased shear modulus, and increased indentation resistance compared to regular honeycombs [45]. Studies conducted by Ju [46, 47] stated that poisson’s ratio helps in avoiding high local cell wall stresses resulting in the advantage of having high effective shear elongation without local cell wall failure. The studies also state that auxetic
honeycombs show high shear flexibility without severe geometric non-linearity compared to their regular counterparts. Auxetic honeycombs show lower effective shear moduli and higher maximum effective shear strains than regular honeycombs [46, 47]. Auxetic honeycombs may also be better candidates for energy efficient structural design as they exhibit low cyclic energy loss under shear loading associated with high shear flexibility for the same shear moduli [31].

2.1 Thesis objectives

Considering the advantages of auxetic honeycombs over regular honeycombs and the advantages of viscoelasticity for constrained layer damping, the current study focuses on the following goals. As mentioned earlier, the main focus of this study is to improve the damping nature of honeycomb core sandwich plates by employing both material and system damping techniques. Specific goals include the following:

1. Compare the damping capacity of sandwich plate structures with regular and auxetic core, when same base material properties are applied for equal mass of the core and total sandwich plate. The damping capacity of regular and auxetic honeycombs is studied independent of material properties applied under dynamic loads. The current study focuses on introducing the viscoelastic behavior to the honeycomb core to study their damping capabilities under quasi-static cyclic and dynamic transient loads.

2. The other objective of this study is to design a honeycomb core such that each individual cell wall in a unit cell is a designed to act as a composite constrained layered damping beam and compare its behavior when the cell wall is made of a
homogenous material. Introduction of the composite honeycomb core makes the sandwich plate act as a multilayered structure with the goal of higher damping capacity.

2.2 Thesis outline

Chapter 1 summarizes recent research studies on various damping techniques developed to reduce structural vibrations. The studies include development of composite structures, introduction of viscoelastic layer between two surfaces and application of honeycombs to form a sandwich structure; to increase the damping within the system. It also gives a brief review on different types of honeycombs and their mechanics with concentration on regular and auxetic types.

Chapter 2 summarizes the motivation and objectives of this Thesis.

Chapter 3 presents details regarding the material and geometric features of honeycombs used in this study. This chapter also discusses design a composite honeycomb core from a unit cell. A brief outline on the design of the in-plane and out-of-plane models used in the current study is also presented.

Chapter 4 gives details of the finite element models developed in ABAQUS including the assembly, meshing, constraints and interactions employed in the current study. This chapter also focuses on the analyses procedures developed for quasi-static cyclic and transient dynamics.

Chapter 5 and 6 gives the finite element analysis results of the out-of-plane loading and in-plane loading models, respectively. These chapters also discuss and make observations of the results.
Chapter 7 summarizes the findings and key contributions from the analysis study.

Suggestions for future work are also made.
CHAPTER 3 : MATERIAL AND GEOMETRIC PROPERTIES

For the purposes of this study, the sandwich plates are constructed using Aluminum and Polycarbonate isotropic materials for the base materials of the composite. Symmetry planes refer to the number of axes of rotational symmetry and an isotropic material has infinite number of symmetric panes. An isotropic material is governed by two elastic constants; commonly used constants are Poisson’s ratio and Young’s elastic modulus. A linear elastic material follows Hooke’s law, with a linear relationship between stress and strain below yielding, and upon unloading; the loading curve is reversed, with no residual strain or stress. The Table 3:1 gives the details regarding the elastic properties of the materials used in this study for face sheets and core.

Table 3:1 Elastic Material Properties used in Sandwich construction

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m$^3$), $\rho$</th>
<th>Poisson’s Ratio, $\nu$</th>
<th>Young’s Modulus, E (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum alloy 5052-H39</td>
<td>2700</td>
<td>0.34</td>
<td>68.97</td>
</tr>
<tr>
<td>Polycarbonate</td>
<td>1200</td>
<td>0.37</td>
<td>2.075</td>
</tr>
</tbody>
</table>

Polycarbonate is chosen due to its both moderate stiffness and viscoelastic damping property. Moreover, polycarbonates show high viscoelastic energy dissipation at high temperature.
3.1 Polycarbonate relaxation data

The viscoelastic properties applied for polycarbonate in the current study are defined in time domain and are extracted from a shear relaxation test [11]. Figure 3:1 shows the stress relaxation data for the normalized shear relaxation modulus which $G_R/G_0$ for polycarbonate which is curve fitted in ABAQUS using the least-squares method using the data obtained by Mercier [11].

![Figure 3:1 Normalized shear relaxation modulus $G_R/G_0$ for polycarbonate](image)

The Prony series coefficients used in the current analysis obtained from the above curve fit are tabulated in Table 3:2.

<table>
<thead>
<tr>
<th>$k$</th>
<th>$g_k$</th>
<th>$k_k$</th>
<th>$\tau_k$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.06101089</td>
<td>0</td>
<td>0.0015332</td>
</tr>
<tr>
<td>2</td>
<td>0.84558</td>
<td>0</td>
<td>2.1425</td>
</tr>
<tr>
<td>3</td>
<td>0.0906806</td>
<td>0</td>
<td>19.791</td>
</tr>
</tbody>
</table>
Using the formulas presented earlier for viscoelastic Prony series models in Section 1.4, the ratio of long term to initial relaxation modulus is calculated using the Prony series data as,

\[
\frac{G_\infty}{G_0} = 0.0036
\]

Figure 3.2 shows the frequency dependent loss of the polycarbonate with this data.

Figure 3.2 Normalized Loss Modulus $G_\ell / G_0$ as a function of frequency

Figure 3.3 shows frequency dependence of the storage moduli for the polycarbonate data.
As discussed earlier in Section 1.2, the damping loss factor is defined as the ratio of loss modulus over storage modulus. Figure 3:4 shows the damping behavior of polycarbonate material with the viscoelastic Prony series coefficients in Table 3:2.
As briefly discussed earlier in Section 1.6, various geometry honeycomb structures are available; this study focuses only on regular and auxetic honeycombs. As stated earlier, one goal of the present study is to compare the damping capacity of the two structures when same material properties are applied for equal mass of the core and sandwich plate.

3.2 Effective properties of Honeycombs

The effective properties of honeycombs were first given by Ashby and Gibson [29] using the concept of beam theory with a unit cell to predict the behavior of honeycomb structure. The parameters which describe the honeycomb cell geometry are its vertical length of cell wall (∂-height of cell wall), the inclined length of cell wall (l-length of cell wall), thickness of the cell wall (t), the angle between the vertical and inclined cell wall known as cell angle (θ) and the depth of the cell wall (d). The other factors defining the cell geometry are the thickness to length ratio, β = t/l, and cell aspect ratio, α = h/l.

Conventional honeycombs are defined with cell angle θ =30°, and the vertical height of cell wall, h, equal to inclined length of cell wall, l, such that α = h/l = 1. An auxetic honeycomb is defined with a negative cell angle i.e. θ < 0°. For a comparison study, an auxetic honeycomb is defined with θ = -30°, and the vertical height of cell wall h, is twice the inclined length of cell wall, l, i.e. h=2l, so that conventional and auxetic honeycomb have the same effective cell size. Figure 3:5 shows regular and auxetic honeycomb with same effective cell size.
The effective orthotropic properties of honeycomb structure based on the unit cell beam models given by Ashby are summarized below [29]. In-plane behavior is defined by four independent constants: $E_1^*$, $E_2^*$, $G_{12}^*$ and $\nu_{12}^*$ or $\nu_{21}^*$.

\[
E_i^* = E_s \frac{\beta^3 \cos \theta}{(\alpha + \sin \theta) \sin^2 \theta}
\]

\[
G_{12}^* = E_s \beta^3 \frac{(\alpha + \sin \theta)}{\alpha^2 (1 + 2\alpha) \cos \theta}
\]

\[
\nu_{12}^* = \frac{\cos^2 \theta}{(\alpha + \sin \theta) \sin \theta}
\]

\[
\nu_{21}^* = \frac{(\alpha + \sin \theta) \sin \theta}{\cos^2 \theta}
\]

In the above, $E_s$, is the Young’s modulus of the core material, $E_1^*$, $E_2^*$ are the effective in-plane moduli in the $x_1$ and $x_2$ directions respectively, $G_{12}^*$ is effective in-plane shear modulus, and, $\nu_{12}^*$ and $\nu_{21}^*$ are the in-plane effective Poisson’s ratios.
For out-of-plane loading, five additional moduli are added to the in-plane moduli to describe the behavior:

\[
E_3^* = E \frac{(\alpha + 2) \beta}{2(\alpha + \sin \theta) \cos \theta}
\]

\[
G_{13}^* = G_s \frac{\cos \theta}{\alpha + \sin \theta} \beta
\]

\[
(G_{23}^{*})_{upper} = G_s \frac{(\alpha + 2\sin^2 \theta) \beta}{2(\alpha + \sin \theta) \cos \theta}
\]

\[
(G_{23}^{*})_{lower} = \frac{(\alpha + \sin \theta) \beta}{(1+2\alpha) \cos \theta}
\]

\[
v_{13}^* = \frac{2\beta^2 \cos \theta}{\sin^2 \theta} - v_s
\]

\[
v_{23}^* = \frac{2(\alpha + \sin \theta)^2 \beta^2}{\cos^2 \theta} v_s
\]

\[
v_{31}^* = v_{32}^* = v_s
\]

where \(E_3^*\) is the effective out-of-plane modulus in the out-of-plane (\(x_3\)) direction, \(G_{13}^*\) and \(G_{23}^*\) are the effective out-of-plane shear moduli, \(v_{13}^*\), \(v_{23}^*\) are the out-of-plane Poisson’s ratio, and \(v_s\) is the Poisson’s ratio of the core material.

The effective mass density of the honeycomb is derived from the volume of the unit cell and is given by

\[
\rho^* = \rho_s \frac{(\alpha + 2) \beta}{2(\alpha + \sin \theta) \cos \theta}
\]
where $\rho_s$ is the mass density of the core material.

The current study deals with honeycombs made of viscoelastic material and the equations which define the orthotropic effective properties of the honeycomb with viscoelasticity are obtained from generalizations of the Prony series described earlier in Section 1.4. For example, the viscoelastic moduli for $x_1$ direction and in-plane shear are:

$$E_{11}^*(\theta, t_{wall}, l, t) = \beta^3 \frac{\cos \theta}{(\alpha + \sin \theta) \sin^2 \theta} E_0 \left[ 1 - \sum_{i=1}^{n} g_i \left( 1 - e^{-\frac{t}{\tau_i}} \right) \right]$$

$$G_{12}^*(\theta, t_{wall}, l, t) = \beta^3 \frac{1}{\alpha^2 (1 + 2\alpha) \cos \theta} E_0 \left[ 1 - \sum_{i=1}^{n} g_i \left( 1 - e^{-\frac{t}{\tau_i}} \right) \right]$$

The expressions are developed using the Prony series of the generalized Maxwell model combined with effective properties defining the behavior of honeycomb, and have been used in hysteresis studies in [31]. The other effective orthotropic viscoelastic properties of honeycomb can be developed similarly.

According to the objective of this study, both regular and auxetic honeycomb core are to be designed having equal mass. The effective mass density can be controlled by varying the cell wall thickness. The cell wall thickness of the auxetic honeycomb is reduced by 25% to achieve the cell wall thickness required for regular honeycomb using the effective mass density equation given above.

The geometric dimensions of the host honeycomb cell used in the current study made of homogeneous polycarbonate are given in the Table 3.3 below. Polycarbonate core is chosen as the host and the geometric dimensions of other cores are calculated based on equating to the mass of the host honeycomb cell.
Table 3:3 Geometric Parameters of Regular and Auxetic honeycomb unit cell

<table>
<thead>
<tr>
<th>Geometric Parameters</th>
<th>Regular Honeycomb Cell</th>
<th>Auxetic Honeycomb Cell</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical height of the cell wall (mm)</td>
<td>4.23</td>
<td>8.46</td>
</tr>
<tr>
<td>Inclined length of the cell wall (mm)</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness of the cell wall</td>
<td>0.423</td>
<td>0.31725</td>
</tr>
<tr>
<td>Cell angle (θ)</td>
<td>30°</td>
<td>-30°</td>
</tr>
<tr>
<td>Depth of the cell wall (mm)</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness to length ratio (β)</td>
<td>0.1</td>
<td>0.075</td>
</tr>
<tr>
<td>Cell aspect ratio (α)</td>
<td>1</td>
<td>2</td>
</tr>
</tbody>
</table>

To control the in-plane and out-of plane moduli properties of the honeycombs, the cell wall thickness, length and cell angle play a vital role. The effective properties of homogeneous honeycomb core made of host material polycarbonate are given in the Table 3:4.
Table 3:4 Effective Properties of Polycarbonate Honeycomb cores

<table>
<thead>
<tr>
<th>Core Properties</th>
<th>Regular Honeycomb Cell</th>
<th>Auxetic Honeycomb Cell</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1^*$ (MPa)</td>
<td>4.79</td>
<td>2</td>
</tr>
<tr>
<td>$E_2^*$ (MPa)</td>
<td>4.79</td>
<td>2</td>
</tr>
<tr>
<td>$E_3^*$ (MPa)</td>
<td>238.69</td>
<td>238.69</td>
</tr>
<tr>
<td>$G_{12}^*$ (MPa)</td>
<td>1.2</td>
<td>0.075</td>
</tr>
<tr>
<td>$G_{13}^*$ (MPa)</td>
<td>43.72</td>
<td>32.67</td>
</tr>
<tr>
<td>$\rho^*$ $\rho_s$</td>
<td>0.12</td>
<td>0.12</td>
</tr>
<tr>
<td>$v_{12}^*$</td>
<td>1</td>
<td>-1</td>
</tr>
<tr>
<td>$v_{13}^*$</td>
<td>1</td>
<td>-1</td>
</tr>
</tbody>
</table>

As discussed earlier in Chapter 2, a goal of this thesis is to compare damping properties of composite honeycomb core sandwich plate to homogeneous honeycomb core sandwich plate. For comparisons, the effective mass of the core and effective mass of the total sandwich plate will be held constant for all cases considered. The constraint of equal mass is obtained by varying the thickness of the cell walls. Both in-plane and out-of plane loading are studied. Each sandwich plate model is constructed of either auxetic or regular honeycomb cores of equal mass. The auxetic and regular honeycomb core sandwich plate models are subdivided into four sub models based on the core material used; a summary of the different cases studied is shown in Figure 3:6.
3.3 Design of Composite Honeycomb Core

In the homogeneous cases the honeycomb core is made of either aluminum or polycarbonate material. The composite core is developed based on the concept of constrained layer damping theory such that each individual cell wall of the honeycomb core acts as a constrained composite beam. In the composite honeycomb core each cell wall is made of three layers, the inner and outer layers of the cell wall are made of aluminum and middle layer is made polycarbonate with viscous behavior having a symmetric configuration as shown in Figure 3:7. The effect of this three-layer cell wall creates a constrained layer damping within the cell wall.
As mentioned earlier, one objective of this thesis is to compare the behavior of honeycomb cores, when cell wall is made of homogeneous material with when the cell wall is made to act like a composite beam for equal mass. The design of composite cell wall of equal mass as of homogeneous cell wall is discussed below. Figure 3:8 shows the modeling of composite cell wall having equal mass as of homogeneous cell wall.

Figure 3:8 Design of composite cell wall
The thickness of the homogeneous cell wall is kept constant, in this study polycarbonate honeycomb is chosen as the host core and the individual thicknesses of the composite wall as calculated accordingly. The individual thicknesses of the composite cell wall are calculated based on the equivalent cell wall mass equation given below.

\[ \rho_h t_{wall} = \rho_p t_p + 2\rho_a t_a \]

where, \( \rho_h \), is the density of homogeneous cell (polycarbonate in this study) , \( t_{wall} \), is the thickness of the homogeneous cell wall, \( \rho_p \), is the density of polycarbonate , \( t_p \) is the thickness of polycarbonate layer in the composite wall, \( t_a \) is the thickness of aluminum in the composite cell wall and \( \rho_a \), is the density of aluminum. In this study, the polycarbonate in the composite core was 40% of the total cell wall thickness, with the aluminum layers totaling 60%.

3.4 Geometric properties of Regular honeycombs

Table 3:5 shows the geometric properties used in the study the regular honeycomb core for homogeneous and composite cell walls. For the composite core the thickness of the cell wall is split into three layers with a symmetric configuration with a viscoelastic polycarbonate layer, sandwiched between aluminum base and constraining layers having equal thickness. The thicknesses for the cases of aluminum and composite cores are based on equating the effective mass the core to the effective mass of polycarbonate core.
Table 3:5 Geometric Parameters of various Regular Honeycomb core models

<table>
<thead>
<tr>
<th>Cell Parameters</th>
<th>Homogeneous Polycarbonate Honeycomb Cell</th>
<th>Composite Honeycomb Cell</th>
<th>Homogeneous Aluminum Honeycomb Cell</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical height of cell wall (h)-mm</td>
<td>4.23</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Inclined length of cell wall (l)-mm</td>
<td>4.23</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness of the cell wall (t)-mm</td>
<td>0.423</td>
<td>0.2435556</td>
<td>0.188</td>
</tr>
<tr>
<td>Cell angle (θ)</td>
<td>30°</td>
<td>30°</td>
<td>30°</td>
</tr>
<tr>
<td>Depth of the cell wall (d)-mm</td>
<td>4.23</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness to length ratio (β)</td>
<td>0.1</td>
<td>0.057578156</td>
<td>0.044</td>
</tr>
<tr>
<td>Cell aspect ratio (α)</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 3:6 gives details regarding the thicknesses of different layers used in composite honeycomb cell walls to achieve equal mass as of homogeneous honeycomb cores.
Table 3:6 Thicknesses defined for a regular honeycomb composite cell wall

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum (Base layer)</td>
<td>0.0717778</td>
</tr>
<tr>
<td>Polycarbonate (sandwiched layer)</td>
<td>0.1</td>
</tr>
<tr>
<td>Aluminum (Constraining layer)</td>
<td>0.0717778</td>
</tr>
</tbody>
</table>

3.5 Geometric properties of Auxetic honeycombs

As mentioned earlier, for the auxetic honeycomb to have the same effective mass density as regular honeycomb the thickness of the cell wall is reduced by 25% to calculate the thickness of cell wall for auxetic core. Table 3:7 shows the geometric properties used in the study for auxetic honeycomb core. Similar to the regular honeycomb cases, the thicknesses mentioned in Table 3:7 for the cases of aluminum and composite auxetic cores are based on equating the effective mass of the core to the effective mass of polycarbonate core. In addition, the mass of the auxetic honeycomb core is equated to regular honeycomb irrespective of the material.
Table 3:7 Geometric Parameters of various Auxetic Honeycomb core models

<table>
<thead>
<tr>
<th>Cell Parameters</th>
<th>Homogeneous Polycarbonate Honeycomb Cell</th>
<th>Composite Honeycomb Cell</th>
<th>Homogeneous Aluminum Honeycomb Cell</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical height of the cell wall (h)-mm</td>
<td>8.46</td>
<td>8.46</td>
<td>8.46</td>
</tr>
<tr>
<td>Inclined length of the cell wall (l)-mm</td>
<td>4.23</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness of cell wall(t)-mm</td>
<td>0.31725</td>
<td>0.1826668</td>
<td>0.141</td>
</tr>
<tr>
<td>cell angle (θ)</td>
<td>-30°</td>
<td>-30°</td>
<td>-30°</td>
</tr>
<tr>
<td>Depth of cell wall (d)-mm</td>
<td>4.23</td>
<td>4.23</td>
<td>4.23</td>
</tr>
<tr>
<td>Thickness to length ratio (β)</td>
<td>0.075</td>
<td>0.04232</td>
<td>0.0333</td>
</tr>
<tr>
<td>Cell aspect ratio (α)</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 3:8 gives the thicknesses of the composite cell wall based on equating the mass to the effective mass of the host core.

Table 3:8 Thicknesses defined for Auxetic honeycomb composite cell wall

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum (Base layer)</td>
<td>0.0538334</td>
</tr>
<tr>
<td>Polycarbonate (Sandwiched Layer)</td>
<td>0.075</td>
</tr>
<tr>
<td>Aluminum (Constraining layer)</td>
<td>0.0538334</td>
</tr>
</tbody>
</table>
3.6 Analytical expressions to design a sandwich plate

The equations to measure the dimensions of a unit cell use basic trigonometry formulae. The analytical expressions for regular honeycomb are given below. Similar equations can be developed for auxetic honeycomb by replacing the positive cell angle with a negative cell angle.

- The effective height (H) of a unit cell: \( H = 2(h + l \sin \theta) \)
- The effective length (L) of a unit cell: \( L = 2l \cos \theta \)

![Regular Honeycomb](image)

**Figure 3:9 Dimensions of regular honeycomb unit cell**

To develop a series of unit cells the overall dimensions of the core can be calculated using the analytical expressions given below:

- Length of the core: \( L_{\text{eff}} = n(2l \cos \theta) \)
- Breadth of the core: \( H_{\text{eff}} = (n+1)(h + l \sin \theta) \)
In the above,

- \( n \) = number of cell in the series in the core,
- \( l \) = inclined length of the cell wall of a unit cell,
- \( h \) = vertical length of the cell of a unit cell and
- \( \theta \) = cell angle (positive for regular honeycomb and negative for auxetic honeycomb).

The above equations are used to calculate the number of unit cells required to be modeled for the given dimensions of the core and given dimensions of a unit cell or vice versa.

3.7 Honeycomb Sandwich plate

In the current study, for the sandwich construction the face sheets are made of aluminum and the honeycomb core is made of polycarbonate or aluminum or a combination of aluminum and polycarbonate (composite) to obtain the desired core material.

The in-plane and out-of-plane behavior of honeycombs when sandwiched is analyzed in this study. Accordingly, sandwich plate models for studying in-plane and out-of-plane properties have been developed. The equations in Section 3.6 are using in developing the out-of-plane and in-plane models for analyses.

Figure 3:10 shows the loading directions on honeycomb core. Loading in \( x_3 \) direction is referred as out-of-plane loading or core wise loading and loading along \( x_1 \) and \( x_2 \) directions is termed as in-plane loading or edge wise loading.
3.7.1 Out-of-plane sandwich model

The sandwich plate is termed as out-of-plane plate model when the loading on the honeycombs is done in the out-of-plane direction, which is the $x_3$ direction as shown in Figure 3:10.

In the following descriptions, sandwich plate models with regular and auxetic honeycomb core structure are developed separately. The honeycomb core for both regular and auxetic cases have same number of unit cells in vertical ($x_2$) and horizontal ($x_1$) direction, respectively. The models are developed such that their effective mass density and the overall geometric dimensions of the sandwich plate are equal.

Honeycomb Core:

Figure 3:11 shows the out-of-plane honeycomb core model, a symmetric honeycomb core is developed so as to resemble a square. Accordingly the honeycomb core is drawn with 11 unit cells in the $x_1$ direction and 6 unit cells in $x_2$ direction for both regular and auxetic cases. The overall dimensions of the core are calculated using the
equations developed in Section 3.6 which is helpful in defining the dimensions of the face sheet.

Figure 3:11 Top views of auxetic and regular honeycomb cores used in out-of-plane model

Face sheet:

The geometric dimensions of the face sheets are defined by its length (L), breadth (B) and depth (d). The dimensions of the face sheet are: length $L = 80.592774916$ mm, thickness $t = 0.2$ mm and breadth $B = 76.14$ mm as shown in Figure 3:12.
Figure 3:12 Top view of face sheet used in out-of-plane model

Figure 3:13 shows the assembly of both regular and auxetic honeycomb core sandwich plates developed for out-of-plane loading.

Table 3:9 gives a mass break-down of core and face sheets of regular and auxetic sandwich plate irrespective of the core material. The data shows that the mass of regular and auxetic cores from the ABAQUS model are approximately the same.
Table 3.9 Mass of individual components of sandwich plates for out-of-plane loading

<table>
<thead>
<tr>
<th>Mass (kg)</th>
<th>Regular Honeycomb Core Sandwich Plate</th>
<th>Auxetic Honeycomb Core Sandwich Plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Core</td>
<td>0.0036511830</td>
<td>0.003678359</td>
</tr>
<tr>
<td>Facesheet</td>
<td>0.003313620</td>
<td>0.003313620</td>
</tr>
<tr>
<td>Sandwich Plate</td>
<td>0.010278387</td>
<td>0.010305585</td>
</tr>
</tbody>
</table>

3.7.2 In-plane sandwich plate model

The sandwich plate is termed as in-plane plate model as the loading on the honeycombs is done in the in-plane direction, which is the $x_2$ direction. Sandwich plate models with regular and auxetic honeycomb core structure are developed separately. Similar to the out-of-plane models, the in-plane models are developed such that there effective mass density and the overall geometric dimensions of the sandwich plate are equal.

Honeycomb Core:

Figure 3.14 shows the in-plane honeycomb core model developed. The honeycomb core is designed with 11 unit cells in the $x_1$ direction and 2 unit cells in $x_2$ direction for both regular and auxetic cases. The overall dimensions of the core are calculated using the equations in Section 3.6 which are helpful in defining the dimensions of the face sheet.
Figure 3:14 2D views of auxetic and regular honeycomb cores used in in-plane model

The dimensions of the face sheet are: length, $L = 80.592774916\text{mm}$, depth, $d = 4.23\text{mm}$, and thickness, $t = 0.2\text{mm}$. Figure 3:15 shows the assembly of sandwich plate model developed for both regular and auxetic honeycomb cores for in-plane loading.

Figure 3:15 3D view of sandwich plates of in-plane model
Table 3:10 shows the mass properties of core and face sheets of regular and auxetic sandwich plate obtained from the Abaqus model, showing that they are essentially the same.

**Table 3:10 Mass of individual components sandwich plates in in-plane loading**

<table>
<thead>
<tr>
<th>Mass (kg)</th>
<th>Regular Honeycomb Core Sandwich Plate</th>
<th>Auxetic Honeycomb Core Sandwich Plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Core</td>
<td>0.0012170461</td>
<td>0.00122612</td>
</tr>
<tr>
<td>Facesheet</td>
<td>0.000184090</td>
<td>0.000184090</td>
</tr>
<tr>
<td>Sandwich Plate</td>
<td>0.0015852261</td>
<td>0.0015943064</td>
</tr>
</tbody>
</table>
CHAPTER 4: FINITE ELEMENT MODELING OF HONEYCOMB SANDWICH PLATES

This chapter describes details of the finite element models developed for honeycomb sandwich plates and analysis procedures. Previous finite element models for honeycomb structures are reported in [19, 30, 31, 37, 38, 44]. In these studies, for out-of-plane loading, three-dimensional models have been used with shell elements, while for in-plane loading, some of the models used two-dimensional models with beam elements. In the present study, 3D models are developed in ABAQUS v.6.8.1 with shell elements for both out-of-plane and in-plane loading. For the composite honeycomb cell wall models, shell composite elements which allow the implicit definition of multiple layer thicknesses and materials are used. For analysis, a standard simulation procedure is developed to perform Quasi-static, Natural Frequency Response, and Dynamic analyses to understand the behavior of the sandwich plate models under cyclic and impact loading.

A shell element model is chosen over a solid model for the thin structures making up the honeycomb sandwich plate to save computational time. Shell elements are defined using section points which help to capture both bending and membrane behavior. The application of shell elements is valid for structures falling under classical thin shell theory. Shell elements carry extra degrees of freedom (dof) at each node bypassing the necessity of modeling physical thickness. In particular, the face sheets and honeycomb core are meshed using S4R shell elements. The S4R shell elements are 4-node doubly curved thin or thick shell, reduced integration, hourglass control with finite membrane
strains. These elements allow transverse shear deformation; they use “thick shell theory” and become “Kirchhoff’s thin shell elements” as thickness decreases [48].

4.1 Assigning composite section in Abaqus

The use of shell elements makes it easier to define composite sections bypassing the necessity to draw physical thickness as in the case of solid elements. Composite section for a regular honeycomb core for the given thickness is assigned in ABAQUS. Figure 4:1 shows the arrangement of layers in the cell-walls and assigned sections which define thickness layers and associated material properties. The auxetic core is modeled in the similar fashion.

![Figure 4:1 Sectional view of composite section regular honeycomb in Abaqus](image)
4.2 Analyses conducted on sandwich plates

To understand the behavior of the honeycombs when loaded in both in-plane and out-of-plane directions different analyses have been conducted: (1) Natural Frequency Response, (2) Quasi Static Analysis, and (3) Dynamic-Implicit Analysis.

For the cases of Quasi-static and Dynamic-Implicit analyses “Long-term” moduli scale is turned on to define viscoelasticity in the time domain in ABAQUS.

**Natural Frequency Response:**

The natural frequency response analysis is conducted using Abaqus/Standard to calculate the mode shapes and associated resonance frequencies for given boundary conditions. The natural frequency of a system depends on the stiffness and mass of the system, and the geometric configuration. For a simple oscillator, the natural frequency ($\omega$) is proportional to the square root to the ratio of stiffness and mass. In general, the natural frequencies of the system increase with the increase in the stiffness of the system for the same mass.

**Quasi Static Analysis:**

Quasi-Static Analysis is conducted to analyze the response to time-dependent material behavior (creep, swelling, viscoelasticity and viscoplasticity); mass inertia effects are not included. Quasi-static analysis can be used for both linear and non-linear systems [48]. The quasi-static analysis is conducted to capture the hysteresis behavior of the system and is done using a sinusoidal amplitude using Abaqus/Standard.
Dynamic – Implicit Analysis:

Dynamic analysis is used to capture the dynamic behavior of the system. The numerical damping control parameter available in the dynamic step in ABAQUS is changed to -0.01 from the default to decrease the effect of external damping. The dynamic analysis is conducted with very small time increments using Abaqus/Standard.

4.3 Properties of out-of-plane sandwich plate model

4.3.1 Mesh properties

Honeycomb Core:

Figure 4:2 shows the meshed honeycomb cores for the cases of regular and auxetic cores. The honeycomb core is meshed to have 4 elements along each cell wall and together the core has 7488 elements for regular core and 7488 elements in the case of an auxetic core.

![Figure 4:2 Isometric views of meshed auxetic and regular honeycomb cores](image)
Face sheet:

Figure 4:3 shows the meshed face sheet developed in ABAQUS. The face sheet is discretized into a total of 1936 shell elements such that there are 44 elements along each edge to maintain symmetry and capture the center node where maximum displacement occurs.

![Figure 4:3 Meshed face sheet](image)

4.3.2 Assembly and constraints

The honeycomb core is sandwiched between two face sheets to obtain a sandwich plate and the assembled plate is shown in Figure 4:4. The auxetic honeycomb core sandwich plate is meshed and assembled in the similar manner.
Constraint:

After assembly the instances have to be constrained together to make it operate as a single part, and to perform this operation, tie-constraints are used in ABAQUS. A surface-based tie constraint ties two surfaces together for the duration of a simulation. It constrains each of the nodes on the slave surface to have the same motion as the point on the master surface to which it is closest [48]. The tie constraint eliminates the degrees of freedom of the slave surface nodes that are constrained. The relative stiffness of the surfaces that are being tied help in determining the master and the slave surface. The stiffer surface is considered to be the master surface.
4.3.3 Boundary and load conditions

The sandwich plate for the out-of-plane model is fixed on all the four sides with all the 6 degrees of freedom constrained to resemble a clamped plate as shown in Figure 4:6. A uniform pressure loading of 1MPa is applied normal to the top face of one of the aluminum face sheets in the out-of-plane direction x3 of the honeycomb core as shown in Figure 4:7. The uniform pressure loading is applied similarly for the sandwich plate with auxetic core.
4.4 Properties of In-plane sandwich plate model

4.4.1 Mesh properties

Honeycomb core:

The figure below shows the meshed honeycomb cores in the case of regular and auxetic cores. The honeycomb core is meshed to have 4 elements along each cell wall.
Figure 4:8 Meshed 3D auxetic and regular honeycomb core for in-plane loading

Face sheet:

Figure 4:9 shows the meshed face sheet developed in ABAQUS. The face sheet is discretized into a total of 176 elements such that there are 44 elements across the length and 4 elements along the depth to maintain symmetry.

Figure 4:9 Meshed 3D face sheet

4.4.2 Assembly and constraints

The honeycomb core is sandwiched between two face sheets to obtain a sandwich plate and the assembled plate is shown in Figure 4:10. The auxetic honeycomb core sandwich plate is meshed and assembled in the similar fashion.
Constraint:

The face sheets and core are constrained using tie-constraints using a similar procedure as the out-of-plane model.

4.4.3 Boundary and load conditions

The sandwich plate for the in-of-plane model is fixed on two sides with all the 6 degrees of freedom constrained to resemble a short beam structure clamped on both ends and a uniform pressure loading 0.1Mpa is applied in the in-plane direction as shown in Figure 4:11.
4.5 Analyses Procedure

A uniform pressure of 1Mpa is applied for out-of-plane model and uniform pressure load of 0.1Mpa is applied for the in-plane model on the top face sheets. The procedure employed to conduct each individual analysis in detail is given as below and similar procedure is carried for both in-plane and out-of-plane models.

**Natural Frequency Response**

The natural frequency response analysis is conducted with no load applied and the boundary conditions being active. The first ten natural frequencies or mode shapes are extracted.

**Quasi-Static Analysis:**

The quasi static analysis is conducted using a visco step for a time period of 0.25sec. Sinusoidal cyclic pressure amplitude loading is applied to study the hysteresis
behavior of the sandwich structures. The resultant force applied is the pressure multiplied by the surface area of the face sheet as shown in Figure 4:12.

![Sinusoidal loading equation](image)

**Figure 4:12 Sinusoidal loading equation for cyclic analysis**

*Step 0:* The boundary conditions applied to the system are defined in this step.

*Step 1:* A uniform sinusoidal cyclic pressure load is applied for a step time of 0.25sec.

The center node of the face sheet is selected based on symmetry conditions as a marker node for later post-processing of center deflection. The boundary conditions defined in step 0 are kept constant throughout the computation. The above computational procedure is employed to calculate the corresponding displacement, stresses and energies of the model. The displacement in the model is extracted at the center node in Step1 which is the location of maximum displacement in the model. The energy stored in the system is also extracted for the entire model in Step 1.
Dynamic-Implicit Analysis:

The dynamic analysis is conducted using a series of two dynamic-implicit steps. The boundary conditions defined in Step 0 are kept constant throughout the computation.

*Step 0:* The boundary conditions applied to the system are defined in this step.

*Step 1:* A ramp loading is applied for a uniform pressure of 1Mpa for a step time of 0.001sec.

*Step 2:* An instantaneous pressure load is applied and held constant for the remaining time period.

The out-of-plane model is run for a time period of 0.1sec and the in-plane model is subjected to loading for 0.3sec. The above computational procedure is employed to calculate the corresponding displacement and energies of the dynamic model.
CHAPTER 5 : RESULTS FOR OUT-OF-PLANE SANDWICH PLATE MODEL

5.1 Loss factor

As discussed earlier in Section 1.2, for the quasi-static sinusoidal loading analysis, the loss Factor ($\eta_s$) in defined as [2]:

$$\eta_s = \frac{D_s}{2\pi U_x}$$

where $D_s$ is the damping energy dissipated/cycle, and $U_x$ is the internal energy due to material loading. According to Lazan [2], damping is defined as the ratio between energy dissipated to the energy stored in the system due to material loading, i.e. energy released by the material during unloading over a cycle ranging from zero force to maximum force to the energy stored from zero to maximum force. The energy stored in the system is always greater than the energy released by the indicating loss of energy by the system. The energy lost by the system is termed as loss factor or the damping capability of the system [2].

In the current study, the internal energy or the input energy of the system is termed as strain energy and the energy released is defined as the dissipated energy. In the above equation for loss factor,
The sinusoidal loading equation is plotted for a time period ($t$) of 0.25sec and the time period ($T$) between 0.05sec and 0.2 sec is termed as one cycle of loading and unloading and is used to calculate energy dissipated per cycle. $U_s$ is the magnitude of the strain energy at the end of the 1st amplitude peak of the sinusoidal loading where maximum loading takes place from zero force to maximum force at time period $t_0$ as shown in Figure 5.1 i.e. $U_s = ALLSE_{t=t_0}$ in the ALLSE (strain energy of the system) computed with ABAQUS. $D_s$, is calculated from the ALLCD (creep dissipation energy dissipation) per one cycle, $D_s = ALLCD_{t=T+t_0} - ALLCD_{t=t_0}$.

For dynamic analysis, a dynamic loss factor ($\psi$) is defined as

$$\psi = \frac{D_s}{U_s}$$
where $D_s$ is the energy dissipated at the end of dynamic step time, $U_s$ is the strain energy stored in the model at the end of the dynamic time step. The dynamic loss factor is an extension of the loss factor calculated in quasi-static analysis.

5.2 Results of natural frequency response

The natural frequency response analysis aids in determining a stiffer system for the same mass of the system. The natural frequencies are extracted for the out-of-plane sandwich model made of different honeycomb core material for both regular and auxetic honeycomb cores. Table below shows the magnitudes of first ten natural frequencies for a sandwich plate made of regular honeycomb core for different core materials.

Table 5:1 Comparison of first ten natural frequencies, of regular honeycomb core sandwich plate sub models

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Homogeneous Polycarbonate Honeycomb Core (Hz)</th>
<th>Homogeneous Aluminum Honeycomb Core (Hz)</th>
<th>Composite Honeycomb Core (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2923.2</td>
<td>6767.0</td>
<td>6434.8</td>
</tr>
<tr>
<td>2</td>
<td>4577.2</td>
<td>11732</td>
<td>11006</td>
</tr>
<tr>
<td>3</td>
<td>4752.0</td>
<td>12243</td>
<td>11470</td>
</tr>
<tr>
<td>4</td>
<td>6001.8</td>
<td>16097</td>
<td>15010</td>
</tr>
<tr>
<td>5</td>
<td>6582.7</td>
<td>18013</td>
<td>16744</td>
</tr>
<tr>
<td>6</td>
<td>6914.0</td>
<td>18927</td>
<td>17589</td>
</tr>
<tr>
<td>7</td>
<td>7655.8</td>
<td>21361</td>
<td>19819</td>
</tr>
<tr>
<td>8</td>
<td>7837.0</td>
<td>21830</td>
<td>20260</td>
</tr>
<tr>
<td>9</td>
<td>8672.3</td>
<td>24634</td>
<td>22799</td>
</tr>
<tr>
<td>10</td>
<td>9118.4</td>
<td>25710</td>
<td>23797</td>
</tr>
</tbody>
</table>

Table 5:2 shows the first ten natural frequencies for a sandwich plate made of auxetic honeycomb core for different core materials.
Table 5:2 Comparison of first ten natural frequencies of auxetic honeycomb core sandwich plates sub models

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Homogeneous Polycarbonate Honeycomb Core (Hz)</th>
<th>Homogeneous Aluminum Honeycomb Core (Hz)</th>
<th>Composite Honeycomb Core (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2533.5</td>
<td>6079.2</td>
<td>5725.2</td>
</tr>
<tr>
<td>2</td>
<td>3996.9</td>
<td>10384</td>
<td>9927.9</td>
</tr>
<tr>
<td>3</td>
<td>4084.6</td>
<td>10668</td>
<td>9936.8</td>
</tr>
<tr>
<td>4</td>
<td>5198.1</td>
<td>14061</td>
<td>13004</td>
</tr>
<tr>
<td>5</td>
<td>5736.5</td>
<td>15594</td>
<td>14495</td>
</tr>
<tr>
<td>6</td>
<td>5924.5</td>
<td>15762</td>
<td>15198</td>
</tr>
<tr>
<td>7</td>
<td>6650.3</td>
<td>16472</td>
<td>17180</td>
</tr>
<tr>
<td>8</td>
<td>6769.3</td>
<td>18703</td>
<td>17559</td>
</tr>
<tr>
<td>9</td>
<td>7571.4</td>
<td>19093</td>
<td>19453</td>
</tr>
<tr>
<td>10</td>
<td>7899.6</td>
<td>21670</td>
<td>19815</td>
</tr>
</tbody>
</table>

Table 5:1 From Table 5:1 and Table 5:2, sandwich plate made of aluminum core and composite core have natural frequencies which are nearly the same, and approximately two times larger than the case of homogeneous polycarbonate core for the same mass. Similar trend is observed for the cases of sandwich plates made of regular and auxetic honeycomb cores suggesting that for the same mass, the aluminum core is slightly stiffer than the polycarbonate and composite cores. But in the case of composite core, the stiffness can be varied by changing the ratio of the aluminum or polycarbonate present in the cell wall of the honeycomb core accordingly to maintain the same mass.

Table 5:3 shows the mode shapes of sandwich plate with homogeneous polycarbonate core for the first ten natural frequencies. The mode shapes of sandwich plates with any of the above mentioned cores look similar, as mode shapes are irrespective of their core material for a square plate for the given boundary conditions.
Table 5:3 First ten mode shape of regular polycarbonate honeycomb core sandwich plate

<table>
<thead>
<tr>
<th>Mode shape:1</th>
<th>Mode shape:2</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="Image" /></td>
<td><img src="image2" alt="Image" /></td>
</tr>
<tr>
<td>Mode shape:3</td>
<td>Mode shape:4</td>
</tr>
<tr>
<td><img src="image3" alt="Image" /></td>
<td><img src="image4" alt="Image" /></td>
</tr>
<tr>
<td>Mode shape:5</td>
<td>Mode shape:6</td>
</tr>
<tr>
<td><img src="image5" alt="Image" /></td>
<td><img src="image6" alt="Image" /></td>
</tr>
<tr>
<td>Mode shape:7</td>
<td>Mode shape:8</td>
</tr>
<tr>
<td><img src="image7" alt="Image" /></td>
<td><img src="image8" alt="Image" /></td>
</tr>
<tr>
<td>Mode shape:9</td>
<td>Mode shape:10</td>
</tr>
<tr>
<td><img src="image9" alt="Image" /></td>
<td><img src="image10" alt="Image" /></td>
</tr>
</tbody>
</table>
The mode shapes displace symmetry as expected with peak amplitudes at the center node occur for modes 1, 5, 6.

5.3 Results for quasi-static analysis

Quasi-static analysis is conducted to capture the hysteresis behavior of the plate under cyclic loading using the sinusoidal loading equation shown in Figure 4:12. The main objective of conducting this analysis is to calculate the damping capacity of the model when visco properties are applicable. Calculation of creep dissipation energy is vital in measure the damping nature of a system.

5.3.1 Results of regular honeycomb core sandwich plate

Figure 5:2 shows the applied force resultant as a function of the center point z-component of displacement, during the cyclic loading. The results show that during loadings and unloading during the sinusoidal cycle, very little hysteresis is found, indicating a small damping loss factor.
Figure 5:2 Force vs. Displacement curve for regular honeycomb core sandwich plate subjected to cyclic loading for out-of-plane model

Figure 5:3, Figure 5:4 and Figure 5:5 show energy and displacement plots for regular honeycomb core sandwich plates. The energies are used to measure the damping of the sandwich plates for a given step time of 0.25sec.

Figure 5:3 Comparisons of creep dissipation energy of regular core sub models under cyclic loading for out-of-plane model
From Figure 5:3 it can be observed that the composite honeycomb core sandwich plate has higher creep dissipation when compared to the homogeneous polycarbonate honeycomb core plate.

![Regular Core-Strain Energy](image)

**Figure 5:4 Comparisons of strain energy regular core sub models under cyclic loading for out-of-plane model**

From the above plot it can be inferred that composite honeycomb core sandwich plate has higher internal strain energy stored than the homogenous polycarbonate core sandwich plate indicating it has higher stiffness.
Figure 5:5 Comparisons of out-of-plane displacement regular core sub models under cyclic loading for out-of-plane model

The above graph shows that both composite honeycomb core plate and homogeneous polycarbonate plate have the similar displacement.

5.3.2 Results of auxetic honeycomb core sandwich plate

Figure 5:6 below shows the Force vs. Displacement hysteretic curves for the cyclic analysis of regular honeycomb core.
Figure 5:6 Force vs. Displacement curve for auxetic honeycomb core sandwich plate subjected to cyclic loading for out-of-plane model

Figure 5:7, Figure 5:8 and Figure 5:9 show the creep energy dissipation, internal strain energy as and z-component of displacement at the center node as functions of time for the auxetic honeycomb core sandwich plate models.
From the above graph it can be observed that the composite honeycomb core has higher creep dissipation energy when compared to homogeneous honeycomb core sandwich plate.
From the above internal strain energy plot it can be stated that, the homogeneous viscous polycarbonate core sandwich plate has lower strain energy when compared to the composite core plate.
From Figure 5: it can be observed that the sandwich plate with homogeneous polycarbonate core has the similar displacement when compared to the composite honeycomb core plate.

The loss factor for each model is calculated from the dissipation and internal energies as discussed earlier, and reported in Table 5:4. It can be observed that the composite core has higher loss factor indicating that more energy is lost by the system relative to the stored strain energy, indicating increased damping. The loss factor of homogeneous aluminum and polycarbonate cores with no viscoelastic properties assigned is zero, as expected.

**Table 5:4 Loss factors of sandwich plates under cyclic loading for out-of-plane model**

<table>
<thead>
<tr>
<th>Core Material</th>
<th>Loss Factor – Cyclic loading</th>
<th>Regular</th>
<th>Auxetic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Homogeneous Polycarbonate</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>- viscous</td>
<td></td>
<td>0.0014</td>
<td>0.0017</td>
</tr>
<tr>
<td>Composite</td>
<td></td>
<td>0.0021</td>
<td>0.0030</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>- non-viscous</td>
<td></td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Homogeneous Aluminum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>- non-viscous</td>
<td></td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

From the conducted quasi-static analyses, comparing the magnitude of the loss factor as mentioned in the Table 5:4, it can be concluded that, Auxetic Composite Honeycomb core sandwich plate has 43% more damping than Auxetic Homogeneous
Polycarbonate viscous Honeycomb core sandwich plate. In the similar fashion, regular composite honeycomb core sandwich plate has 33% higher damping than Regular Homogeneous Polycarbonate viscoelastic Honeycomb core sandwich plate.

5.4 Results of Dynamic-Implicit analysis

5.4.1 Results of regular core sandwich plate

The graphs below show the energy and displacement plots of the regular core sandwich plate subject to dynamic loading used in measuring the damping.

![Graph showing creep dissipation energy comparison](image)

**Figure 5:10 Comparison of creep dissipation energy of regular core sub models under dynamic loading for out-of-plane model**

From Figure 5:10, it can be observed that the composite honeycomb core has much higher creep energy dissipation when compared to the homogeneous polycarbonate core plate.
Figure 5:11 Comparison of strain energy of regular core sub models under dynamic loading for out-of-plane model

From the above graph it can be observed that homogeneous non viscous polycarbonate core plate with behavior has the highest strain energy stored followed by homogeneous aluminum, composite core and homogeneous viscous polycarbonate honeycomb core plate.
Figure 5:12 Comparison of out-of-plane displacement of regular core sub models under dynamic loading for out-of-plane model

Figure 5:12 shows the out-of-plane displacement of the sandwich plates and it can be inferred that the plated having viscous behavior have lower displacement when compared to the plate with non-viscous behavior due to the creep behavior.

Table 5:5 shows a comparison of the dynamic properties of regular core sandwich plate at the end of the step time.

**Table 5:5 Comparisons of response parameters of regular honeycomb core sandwich plates for sub models under dynamic loading for out-of-plane model**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Homogeneous Polycarbonate Honeycomb Core</th>
<th>Homogeneous Aluminum Honeycomb Core</th>
<th>Homogeneous Polycarbonate Honeycomb core-non visco</th>
<th>Composite Honeycomb Core</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center Displacement (m)</td>
<td>0.0003364</td>
<td>0.000512793</td>
<td>0.00214131</td>
<td>0.0003662</td>
</tr>
<tr>
<td>ALLCD (J)</td>
<td>0.005648</td>
<td>0</td>
<td>0</td>
<td>0.00925325</td>
</tr>
<tr>
<td>ALLSE (J)</td>
<td>0.332105</td>
<td>0.593381</td>
<td>2.78571</td>
<td>0.375601</td>
</tr>
</tbody>
</table>
5.4.2 Results of auxetic core sandwich plate

The energy and displacement values are plotted as a function of time for the auxetic core sandwich plate subject to dynamic loading.

![Graph showing creep dissipation energy comparison](image)

**Figure 5.13 Comparison of creep dissipation energy of auxetic core sub models under dynamic loading for out-of-plane model**

The above graphs conclude that the composite core plate has higher dissipation energy when compared to the homogeneous polycarbonate core plate.
Figure 5:14 Comparison of strain energy of auxetic core sub models under dynamic loading for out-of-plane model

Figure 5:14 shows a comparison of the internal strain energy stored by different sandwich plates subjected to dynamic loading.

Figure 5:15 Comparison of out-of-plane displacement of auxetic core sub models under dynamic loading for out-of-plane model
From Figure 5:15 it can be observed that the sandwich plate with homogeneous polycarbonate core has a low displacement when compared to the plates because of the creep behaviour due to visco in nature.

The table below shows a comparison of the dynamic properties of auxetic core sandwich plate at the end of the step time.

Table 5:6 Comparisons of response parameters of auxetic honeycomb core sandwich plates for sub models under dynamic loading for out-of-plane model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Homogeneous Polycarbonate Honeycomb Core</th>
<th>Homogeneous Aluminum Honeycomb Core</th>
<th>Homogeneous Polycarbonate Honeycomb core-non visco</th>
<th>Composite Honeycomb Core</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center Displacement (m)</td>
<td>0.0003523</td>
<td>0.0006325</td>
<td>0.002357</td>
<td>0.0003892</td>
</tr>
<tr>
<td>ALLCD (J)</td>
<td>0.00657</td>
<td>0</td>
<td>0</td>
<td>0.013707</td>
</tr>
<tr>
<td>ALLSE (J)</td>
<td>0.339455</td>
<td>0.740231</td>
<td>2.85747</td>
<td>0.349486</td>
</tr>
</tbody>
</table>

The dynamic loss factor is calculated for all the sandwich plates at end of the step time of 0.1 sec are tabulated and shown below.

Table 5:7 Dynamic loss factors for out-of-plane models under dynamic loading

<table>
<thead>
<tr>
<th>Core Material</th>
<th>Dynamic Loss Factor –Dynamic loading</th>
<th>Auxetic</th>
<th>Regular</th>
</tr>
</thead>
<tbody>
<tr>
<td>Homogeneous Polycarbonate - visco</td>
<td></td>
<td>0.01935632</td>
<td>0.01700667</td>
</tr>
<tr>
<td>Composite</td>
<td></td>
<td>0.03922046</td>
<td>0.02463585</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate non-viscous</td>
<td></td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Homogeneous Aluminum non-viscous</td>
<td></td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>
From Table 5:7, it can be inferred that the Auxetic Composite Honeycomb core sandwich plate has 51% more damping than Auxetic Homogeneous Polycarbonate viscous Honeycomb core sandwich plate. In the similar fashion, Regular Composite Honeycomb core sandwich plate has 31% higher damping than Regular Homogeneous Polycarbonate viscous Honeycomb core sandwich plate. The loss factors of homogeneous aluminum honeycomb core and homogeneous polycarbonate core having no viscous behavior are zero because they do not dissipate and energy as expected.
CHAPTER 6 : RESULTS FOR IN-PLANE SANDWICH PLATE MODEL

6.1 Results of Natural frequency response

Table 6:1 shows the first ten natural frequencies of the sandwich plate made of regular honeycomb core for different core materials.

Table 6:1 Comparisons of first ten natural frequencies of regular honeycomb core sandwich plate for in-plane model

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Homogeneous Polycarbonate Honeycomb Core (Hz)</th>
<th>Homogeneous Aluminum Honeycomb Core (Hz)</th>
<th>Composite Honeycomb Core (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>634.3</td>
<td>1073.1</td>
<td>1311.8</td>
</tr>
<tr>
<td>2</td>
<td>1285.1</td>
<td>2007.3</td>
<td>1946.2</td>
</tr>
<tr>
<td>3</td>
<td>1440.7</td>
<td>2162.4</td>
<td>2448.6</td>
</tr>
<tr>
<td>4</td>
<td>1966.2</td>
<td>2459.0</td>
<td>2637.9</td>
</tr>
<tr>
<td>5</td>
<td>2193.2</td>
<td>3279.9</td>
<td>3994.3</td>
</tr>
<tr>
<td>6</td>
<td>2254.1</td>
<td>3730.7</td>
<td>4607.2</td>
</tr>
<tr>
<td>7</td>
<td>2312.9</td>
<td>3941.7</td>
<td>4946.2</td>
</tr>
<tr>
<td>8</td>
<td>2542.5</td>
<td>4238.5</td>
<td>5360.1</td>
</tr>
<tr>
<td>9</td>
<td>2686.0</td>
<td>4419.0</td>
<td>5375.2</td>
</tr>
<tr>
<td>10</td>
<td>2700.1</td>
<td>4816.8</td>
<td>5402.4</td>
</tr>
</tbody>
</table>

Table 6:2 shows first ten natural frequencies of the sandwich plate made of auxetic honeycomb core for different core materials.
Table 6:2 Comparisons of first ten natural frequencies of auxetic core sandwich plate

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Homogeneous Polycarbonate Honeycomb Core (Hz)</th>
<th>Homogeneous Aluminum Honeycomb Core (Hz)</th>
<th>Composite Honeycomb Core (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>201.36</td>
<td>317.72</td>
<td>418.22</td>
</tr>
<tr>
<td>2</td>
<td>430.44</td>
<td>659.96</td>
<td>861.37</td>
</tr>
<tr>
<td>3</td>
<td>705.46</td>
<td>1047.5</td>
<td>1351.9</td>
</tr>
<tr>
<td>4</td>
<td>970.03</td>
<td>1452.6</td>
<td>1449.7</td>
</tr>
<tr>
<td>5</td>
<td>1023.2</td>
<td>1485.3</td>
<td>1893.5</td>
</tr>
<tr>
<td>6</td>
<td>1037.7</td>
<td>1753.3</td>
<td>2227.2</td>
</tr>
<tr>
<td>7</td>
<td>1037.7</td>
<td>1964.4</td>
<td>2334.3</td>
</tr>
<tr>
<td>8</td>
<td>1346.9</td>
<td>2219.3</td>
<td>2490.9</td>
</tr>
<tr>
<td>9</td>
<td>1365.6</td>
<td>2229.2</td>
<td>2546.9</td>
</tr>
<tr>
<td>10</td>
<td>1394.0</td>
<td>2485.7</td>
<td>3088.6</td>
</tr>
</tbody>
</table>

The stiffness of the homogeneous polycarbonate and aluminum core plates for both regular and auxetic models cannot be altered, as the thickness of the cell wall of the honeycomb core is fixed to maintain the same mass. But in the case of composite core, the stiffness can be varied by varying the ratio of the aluminum or polycarbonate present in the cell wall of the honeycomb core accordingly to maintain the same mass. Composite has twice the stiffness when compared to homogeneous polycarbonate and slightly higher frequency range than homogeneous aluminum cores sandwich plates. Regular honeycomb core sandwich plate has higher frequencies than auxetic honeycomb core plate indicating it is much stiffer.

Table 6:3 shows the corresponding mode shapes of sandwich plate with homogeneous polycarbonate for the first ten natural frequencies. The mode shapes of sandwich plates with any of the above mentioned cores look similar as mode shapes are irrespective of their core material for the given boundary conditions.
Table 6:3 First ten mode shapes for regular polycarbonate honeycomb core sandwich plate

<table>
<thead>
<tr>
<th>1st mode</th>
<th>2nd mode</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="Mode shape:1" /></td>
<td><img src="image2" alt="Mode shape:2" /></td>
</tr>
<tr>
<td><img src="image3" alt="Mode shape:3" /></td>
<td><img src="image4" alt="Mode shape:4" /></td>
</tr>
<tr>
<td><img src="image5" alt="Mode shape:5" /></td>
<td><img src="image6" alt="Mode shape:6" /></td>
</tr>
<tr>
<td><img src="image7" alt="Mode shape:7" /></td>
<td><img src="image8" alt="Mode shape:8" /></td>
</tr>
<tr>
<td><img src="image9" alt="Mode shape:9" /></td>
<td><img src="image10" alt="Mode shape:10" /></td>
</tr>
</tbody>
</table>
6.2 Results for quasi-static analysis

Quasi static cyclic analysis is computed by applying a uniform pressure load using a sine equation as shown in Figure 4:12.

6.2.1 Results of regular core sub sandwich plate

The graph below shows the Force vs. Displacement hysteresis curves for a complete cycle for composite and homogeneous polycarbonate cores. The energy dissipated by the sandwich plate is normalized by strain energy to calculate the loss factor.

![Force vs Displacement](image)

**Figure 6:1 Force vs. Displacement curve for regular honeycomb core sandwich plate subjected to cyclic loading for in-plane models**

The figures below show the energy plots and displacement plots correspondingly of the sandwich plates for a regular core for a given step time of 0.25sec.
From the above figure it can be observed that the homogeneous polycarbonate has higher creep dissipation than the composite honeycomb core plate.
From the above plot, it is seen that composite core plate has higher stored strain energy than homogeneous viscous polycarbonate honeycomb core plate indicating it is much stiffer.

![Graph showing displacement in the vertical direction of sandwich plates made of different core materials.](image)

**Figure 6:4 Comparison of vertical displacement of regular core sub models under cyclic loading for in-plane models**

The above graph shows the displacement in the vertical direction of sandwich plates made of different core materials indicating the homogeneous non-viscous polycarbonate core has the highest displacement and the homogeneous viscous polycarbonate core has the lowest displacement.

6.2.2 Results of auxetic core sandwich plate

The graph below shows the hysterisis plot for different sandwich plates subject to cyclic loading.
Figure 6:5 Force vs. Displacement curve for auxetic honeycomb core sandwich plate subjected to cyclic loading for in-plane models.

Figure 6:6, Figure 6:7 and Figure 6:8 below show the energy and displacement plots correspondingly of the sandwich plates for an auxetic core for a given step time of 0.25sec.

Figure 6:6 Comparison of creep dissipation energy of auxetic core sub models under cyclic loading for in-plane models.
From the above plot it is observed that the homogeneous polycarbonate has higher dissipation energy than the composite core plate.

Figure 6:7 Comparison of strain energy of auxetic core sub models under cyclic loading for in-plane models

From the above figure it is seen that the composite has higher strain energy stored the homogeneous viscous polycarbonate core sandwich plate.

Figure 6:8 Comparison of vertical displacement of auxetic core sub models under cyclic loading for in-plane models
From Figure 6:8, it can be observed that the sandwich plate with homogeneous polycarbonate core displaces low when compared to the other plates.

The table below shows the loss factors calculated for the sandwich plates subject to cyclic loading.

**Table 6:4 Loss factors of sandwich plates under cyclic loading for in-plane model**

<table>
<thead>
<tr>
<th>Core Material</th>
<th>Loss Factor for Cyclic Loading</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Auxetic</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate - visco</td>
<td>0.0161276921</td>
<td>0.0093869</td>
</tr>
<tr>
<td>Composite</td>
<td>0.002711299</td>
<td>0.0029612</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate non-viscous</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Homogeneous Aluminum non-viscous</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

From the above table, it can be seen that the Auxetic homogeneous Polycarbonate viscous core sandwich plate has 81% more damping than Auxetic Composite Honeycomb core sandwich plate. In the similar fashion, Regular homogeneous polycarbonate viscous core sandwich plate has 68% higher damping than Regular composite honeycomb core sandwich plate. The loss factors of homogeneous non-viscous aluminum and polycarbonate cores are zero as they don’t dissipate any energy.
6.3 Results for Dynamic-implicit analysis

The dynamic analysis is performed to calculate the dynamic loss factor to study the damping behavior of the honeycomb core.

6.3.1 Results for regular core sandwich plate

Figure 6:9, Figure 6:10 and Figure 6:11 below show the energy plots of the regular core sandwich plate subject to dynamic loading. The graphs show comparison of energy plots of the 4 sub models.

![Figure 6:9 Comparison of creep dissipation energy of regular core sub models under dynamic loading for in-plane models](image)

From the above graph it is observed that the composite core plate dissipates higher energy when compared to the homogeneous viscous polycarbonate honeycomb core sandwich plate.
From the above plot it can be inferred that homogeneous non-viscous polycarbonate core has the highest stored internal strain energy than the homogenous aluminum followed by composite core plate and homogeneous viscous polycarbonate honeycomb core plate.

Figure 6:10 Comparison of strain energy of regular core sub models under dynamic loading for in-plane models

Figure 6:11 Comparison of vertical displacement of regular core sub models under dynamic loading for in-plane models
The above plot stated that the homogeneous viscous polycarbonate honeycomb core sandwich plate has the lowest displacement.

The table below shows the comparison of the response parameters calculated for the regular honeycomb core sandwich plate models subjected to dynamic loading.

**Table 6:5 Comparisons of response parameters of regular core sub models under dynamic loading**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Homogeneous Polycarbonate Honeycomb Core</th>
<th>Homogeneous Aluminum Honeycomb Core</th>
<th>Homogeneous Polycarbonate Honeycomb core-non visco</th>
<th>Composite Honeycomb Core</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center Displacement (m)</td>
<td>0.0000368</td>
<td>0.00059598</td>
<td>0.0012773</td>
<td>0.000247125</td>
</tr>
<tr>
<td>ALLCD (J)</td>
<td>0.000076525</td>
<td>0</td>
<td>0</td>
<td>0.000189778</td>
</tr>
<tr>
<td>ALLSE (J)</td>
<td>0.000379156</td>
<td>0.006484345</td>
<td>0.0120791</td>
<td>0.00262404</td>
</tr>
</tbody>
</table>

6.3.2 Results for auxetic core sandwich plate

Figure 6:12, Figure 6:13 and Figure 6:14 below show the energy and displacement plots correspondingly of the regular core sandwich plate subject to dynamic loading.
Figure 6:12 Comparison of creep dissipation energy of auxetic core sub models under dynamic loading for in-plane models

From the above graph composite core sandwich plate has higher dissipation energy when compared to the homogeneous viscous polycarbonate core sandwich plate.

Figure 6:13 Comparison of strain energy auxetic core sub models under dynamic loading for in-plane models

From the above graph it is observed that composite has higher internal strain stored in the sandwich plate than homogeneous viscous polycarbonate core sandwich plate.
Figure 6.14 Comparison of vertical displacement of auxetic core sub models under dynamic loading for in-plane models

From Figure 6.14, it can be observed that the sandwich plate with homogeneous polycarbonate core has a lower vertical displacement when compared to the other plates.

Table 6.6 shows a comparison of the dynamic properties of auxetic core sandwich plates at the end of the step time.

Table 6.6 Comparison of response parameters of auxetic core sub models under cyclic loading

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Homogeneous Polycarbonate Honeycomb Core</th>
<th>Homogeneous Aluminum Honeycomb Core</th>
<th>Homogeneous Polycarbonate Honeycomb core-non visco</th>
<th>Composite Honeycomb Core</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center Displacement (m)</td>
<td>0.0003348</td>
<td>0.00197335</td>
<td>0.0020037</td>
<td>0.00169088</td>
</tr>
<tr>
<td>ALLCD (J)</td>
<td>0.00104562</td>
<td>0</td>
<td>0</td>
<td>0.00203359</td>
</tr>
<tr>
<td>ALLSE (J)</td>
<td>0.00309293</td>
<td>0.016852</td>
<td>0.015311</td>
<td>0.016852</td>
</tr>
</tbody>
</table>
Table 6:7 Dynamic loss factors of auxetic and regular core sandwich plate under dynamic loading for in-plane models

<table>
<thead>
<tr>
<th>Core Material</th>
<th>Dynamic Loss Factor – Dynamic loading</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Auxetic</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate - visco</td>
<td>0.33806779</td>
</tr>
<tr>
<td>Composite</td>
<td>0.12067351</td>
</tr>
<tr>
<td>Homogeneous Polycarbonate non-viscous</td>
<td>0</td>
</tr>
<tr>
<td>Homogeneous Aluminum non-viscous</td>
<td>0</td>
</tr>
</tbody>
</table>

From Table 6:7, it can be inferred that the loss factor for Auxetic Homogeneous Polycarbonate viscous Honeycomb core sandwich plate has 64% more damping than Auxetic Composite Honeycomb core sandwich plate. In the similar fashion, Regular Homogeneous Honeycomb core sandwich plate has 64% higher damping than Regular Composite Polycarbonate viscous Honeycomb core sandwich plate. The non-viscous homogeneous core plates have zero loss factors as expected.
A goal of this thesis was to improve the damping capacity of a sandwich honeycomb structure by studying their behavior using different materials. The analyses have been performed on both the regular and auxetic honeycombs to identify which geometry gives higher damping. The first sub-objective of this thesis was to compare the composite honeycomb core sandwich plate to homogeneous honeycomb core sandwich plate and study their behavior and identify a better damping model. The second sub-objective was to study the characteristics of regular honeycomb to auxetic honeycomb core for both the composite and homogenous core plates.

Analyses have been conducted on both the in-plane and out-of plane directions of the honeycomb core when sandwiched between two face sheets.

7.2 Summary of results of out-of-plane model

7.2.1 Stiffness of sandwich plate with auxetic and regular honeycomb cores

For equal mass of the sandwich plates irrespective of the honeycomb core material, stiffness of regular honeycomb is greater than auxetic and has higher natural frequencies.

It was observed that, increase in the ratio of polycarbonate in the composite wall decreased the stiffness of the sandwich plate and decrease in the polycarbonate ratio of the composite wall increases the stiffness of the sandwich plate for both regular and auxetic cases. From the out-of-plane displacement graphs plotted for different core sandwich plates it is observed that the regular honeycomb core plate has a lower displacement when compared to its counter part i.e. auxetic honeycomb core sandwich
plate for the same material indicating that the regular honeycomb core is stiffer than auxetic honeycomb core plate.

7.2.2 Damping of sandwich plates with regular and auxetic honeycomb cores

7.2.2.1 Results for quasi-static analysis

From the calculated loss factors it mentioned in it can be observed that, for equal mass of the sandwich plates, Auxetic composite honeycomb core sandwich plate has the highest damping capability with 30% more damping capacity than its counterpart i.e. regular composite honeycomb core sandwich plate.

7.2.2.2 Results for Dynamic-implicit analysis

Comparing the magnitude of the dynamic loss factors from Table 5:7, the damping in the structure for equal mass of the sandwich plate, Auxetic composite honeycomb core sandwich plate has the highest damping with 37% more damping than its counterpart.

Similar trend is observed in both dynamic and quasi-static analyses. Sandwich plates with composite honeycomb core have the highest damping irrespective of the shape of honeycomb core. It can also be concluded that auxetic honeycomb core sandwich plate has the highest damping irrespective of the honeycomb core material for the given boundary conditions.

In the case of out-of-plane loading of honeycomb, axial deformation plays a vital role in the core behavior. In the case of constrained layer damping axial or shear forces between the layers help in generating more damping of the structure and this when applied to honeycomb core loaded in out-of-plane direction causes higher damping than the honeycomb core made of homogeneous material.
7.3 Summary of results of in-plane model

7.3.1 Stiffness of sandwich plates with auxetic and regular honeycomb cores

For the same mass of the sandwich plate made of different honeycomb cores, based on the natural frequencies it can be inferred that composite core is stiffer than homogeneous aluminum and polycarbonate cores. The similar pattern is observed in the case of both regular and auxetic honeycomb core models. For equal mass of the sandwich plates irrespective of the honeycomb core material, stiffness of regular honeycomb is higher than auxetic honeycomb core sandwich plates and has higher natural frequencies.

From the displacement graphs plotted for various models under different loading conditions it can be observed that the regular honeycomb core plate has lower displacement than the auxetic honeycomb core sandwich plate for the same material inferring that regular honeycomb core is stiffer than auxetic honeycomb core irrespective of the material.

The stiffness of the composite honeycomb core sandwich plate increases with increase in the polycarbonate ratio as the aluminum layers get separated by higher distance from the neutral axis resulting the increase of stiness. Increase in the aluminum ratio in the composite honeycomb core causes decrease in natural frequencies.

7.3.2 Damping of sandwich plates with auxetic and regular honeycomb core

7.3.2.1 Results for quasi-static analysis

On comparing the magnitude of the loss factors from the Table 6:4, damping in the structure for equal mass of the sandwich plate, Auxetic homogeneous polycarbonate
viscous honeycomb core sandwich plate has the highest damping with 42% more damping than its Regular counterpart.

7.3.2.2 Results for Dynamic Analysis

Comparing the loss dynamic loss factors from Table 6:7, Auxetic homogeneous polycarbonate viscous honeycomb core sandwich plate has the highest damping with 37% more damping than its counterpart.

For the conducted dynamic and quasi-static analyses, an auxetic honeycomb core sandwich plate has the highest damping irrespective of the honeycomb core material for the given boundary conditions.

7.4 Key observations

It is observed that the damping loss factors are higher for the in-plane loading when compared to out-of-plane loading. Also the out-of-plane stiffness was significantly higher with lower stored internal strain energy, as compared to in-plane loading, as expected since the out-of-plane plane effective elastic moduli are generally larger than in-plane.

Another key observation was that in the case of the out-of-plane model, the damping loss factor was higher for the composite honeycomb core compared to the homogeneous honeycomb core. Conversely, for the in-plane loading model, the homogeneous honeycomb core showed higher damping compared with the composite honeycomb structure. In the case of the composite honeycomb core, even keeping the mass equivalent to the homogeneous case, there is some variability in the choice of
relative constrained layer damping thicknesses used for the polycarbonate and aluminum layers in the cell walls. The current study used a composite cell wall thickness composed of 40% polycarbonate with viscoelastic properties, with 60% aluminum. The ratio of polycarbonate to aluminum can be changed to improve the performance of the constrained layer damping effect; especially in-plane loading case. The interplay between the stiff aluminum layers and sandwiched polycarbonate in the cell walls may play an important role in controlling the amount of damping in the honeycomb structure. For the case of the in-plane loading of honeycombs, bending of the cell wall plays an important role in the core behavior. The in-plane loading causes bending deformation and which ideally would cause the polycarbonate to shear between the aluminum constraining layers, producing more damping. By adjusting the relative stiffness of aluminum to polycarbonate, the damping effect could be improved.

7.5 Key contributions

An innovative design of composite honeycomb core is developed having stiffness similar to or greater than that of normal metallic or polymer honeycomb cores. A key finding is that for the sandwich plate of equal mass, with Regular or Auxetic core; sandwich plate with composite honeycomb core provides higher stiffness and good damping for out-of-plane loads when compared to plates made of homogeneous metallic honeycomb cores. The other important contribution of the thesis is that, for the loading and boundary conditions studied, auxetic honeycomb has high damping capacity on compared to regular honeycomb irrespective of the material used and irrespective of type of loading. Other key results are that the Auxetic core being flexible in both in-plane and
out-of plane conditions generate lower natural frequencies than Regular core. It is hypothesized that Auxetic core being more flexible may be one of the important reasons it dissipates higher energy than Regular core.

7.6 Suggestions for future work

In the present work, the damping of honeycombs is measured using quasi-static and dynamic analyses by calculating the loss factor. The advantages of constrained layer damping led to the proposed novel design of composite honeycomb core to replace the homogeneous honeycomb core. For comparison purposes, the mass of all the models studied was made equal by varying the thickness of cell walls in regular and auxetic cells for different materials accordingly.

As mentioned earlier, the geometry of the honeycomb has a vital role in determining the behaviour of the honeycomb.

1. The ratio of polycarbonate to aluminum in the composite honeycomb core can be changes to improve the performance of the constrained layer damping effect; especially in-plane loading case.

2. The present work can be extended by varying the cell parameters: thickness, height, length and cell angle, to other honeycomb geometries to study their effect on energy dissipation.

3. The analysis of damping was conducted in the current study in the time domain; in future analyses can be conducted in the frequency domain with damping measured with quality factor and the half power band width method.
4. Develop an analytical model and perform physical experiments to validate the FEA results for sandwich structures under quasi-static and dynamic loads.
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