Low-Velocity Impact Behavior of Sandwich Panels with 3D Printed Polymer Core Structures

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LOW-VELOCITY IMPACT BEHAVIOR OF SANDWICH PANELS WITH 3D PRINTED POLYMER CORE STRUCTURES

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

By

ANDREW JOSEPH TURNER
B.S.M.E., Wright State University, 2015

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Wright State University
WRIGHT STATE UNIVERSITY

GRADUATE SCHOOL

April 21, 2017
I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY Andrew Joseph Turner ENTITLED Low-Velocity Impact Behavior of Sandwich Panels with 3D Printed Polymer Core Structures BE ACCEPTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF Master of Science in Mechanical Engineering.

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ABSTRACT

Turner, Andrew Joseph. M.S.M.E., Department of Mechanical and Materials Engineering, Wright State University, 2017. Low-Velocity Impact Behavior of Sandwich Panels with 3D Printed Polymer Core Structures.

Sandwich panel structures are widely used in aerospace, marine, and automotive applications because of their high flexural stiffness, strength-to-weight ratio, good vibration damping, and low through-thickness thermal conductivity. These structures consist of solid face sheets and low-density cellular core structures, which are often based upon honeycomb topologies. The recent progress of additive manufacturing (AM) (popularly known as 3D printing) processes has allowed lattice configurations to be designed with improved thermal-mechanical properties. The aim of this work is to design and print lattice truss structures (LTS) keeping in mind the flexible nature of AM. Several 3D printed core structures were created using polymeric material and were tested under low-velocity impact loads. Different unit-cell configurations were compared to aluminum honeycomb cores that are tested under the same conditions. An impact machine was designed and fabricated following the ASTM D7136 Standard to correctly capture the impact response. The absorption energy as well as the failure mechanisms of lattice cells under such loads are investigated. The differences in energy-absorption capabilities were captured by integrating the load-displacement curve found from the impact response. Similar manufacturing and sandwich-panel-fabrication processes must be used to accurately compare the impact responses. It is observed that selective
placement of vertical support struts in the unit-cell results in an increase in the absorption energy of the sandwich panels. Other unit-cell configurations can be designed with different arrangements of vertical struts into the well-known body centered cubic (BCC) LTS for further improvements in absorption energy capabilities.
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LIST OF NOMENCLATURE

$a$ Acceleration...........................................................................................................ft/s$^2$

$d$ Distance....................................................................................................................in

$\delta$ Displacement.........................................................................................................in

$\delta_D$ Displacement through the Plateau Section..........................................................in

$E$ Energy............................................................................................................................ft-lbs

$\Delta E$ Change in Energy(Absorbed Energy)................................................................ft-lbs

$F$ Force..............................................................................................................................lbf

$F_m$ Constant Force Required to Cause Plateau Stress Section........................................lbf

$g$ Gravitational Constant.............................................................................................(lbm-ft)/(lbf-s$^2$)

$h$ Height..........................................................................................................................in

$KE$ Kinetic Energy.........................................................................................................ft-lbs

$m$ Mass.............................................................................................................................slugs

$n$ Mode Shape................................................................................................................dimensionless

$P_{cr}$ Critical Force Required to Cause Buckling..............................................................lbf

$PE$ Potential Energy.......................................................................................................ft-lbs

$\rho$ Density........................................................................................................................slugs/in$^3$

$t$ Time................................................................................................................................sec

$v_1$ Initial Velocity............................................................................................................ft/s

$v_2$ Final Velocity.............................................................................................................ft/s
\[ \nu_i \quad \text{Initial Velocity} \quad \text{ft/s} \]
\[ \nu_f \quad \text{Final Velocity} \quad \text{ft/s} \]
\[ W \quad \text{Energy/Absorption Energy} \quad \text{ft-lbs} \]
LIST OF ACRONYMS

3D - 3-Dimensional
ABS - Acrylonitrile Butadiene Styrene
AM - Additive Manufacturing
AISI - American Iron and Steel Institute
ASTM - American Society for Testing and Materials
BCC - Body Centered Cubic unit cell
BCCZ - Body Centered Cubic unit cell with Vertical Struts
BCCAV - Body Centered Cubic unit cell with Alternating Vertical Struts
CAD - Computer Aided Design
CFRP - Carbon Fiber Reinforced Plastic
CT - Computerized Axial Tomography
cumtrapz - Cumulative Trapezoidal Integration Matlab Function
DAQ - Data Acquisition
FDM - Fused Deposition Modeling
ICP - Integrated Circuit Piezoelectric
IEPE - Integrated Electronic Piezoelectric
KE - Kinetic Energy
LTS - Lattice Truss Structure
NI - National Instruments
PE - Potential Energy
SS - Stainless Steel
SLM - Selective Laser Melting
STL - Stereo Lithograph
USB - Universal Serial Bus
UV - Ultraviolet
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Dedicated to my son
Mason Joseph Turner
CHAPTER 1: INTRODUCTION

1.1 Background

1.1.1 Sandwich Panels

Sandwich panels consisting of open cell lattice truss cores layered with fiber reinforced face sheets have tremendous potential of becoming common in the automotive industries. These low-density structures allow for improvements in mechanical properties such as strength, stiffness, energy absorption, and a low relative density when compared to the conventional method of solid high-density structures. They may even be comparable to other similar cellular structures such as honeycomb or foams. The strength of the sandwich panels are optimized due to the topologies of the composite structures. Similar to an I-beam in bending, the skins of the structure take the compressive and tensile loads, while the core supports the shear loads and transfers it between the two faces [1]. While improvements and optimization of the material properties of fiber reinforced face sheets, honeycomb and metal foams has been explored, limited research has been conducted on unit cell configurations of polymer lattice truss structures.

1.1.2 Cellular Structures

Cellular structures were first discovered in nature and have since been researched and photographed around the world. Examples of honeycomb and closed cell structures
found in nature are shown in Figure 1.1. These structures, along with more like them, are naturally optimized for their particular applications. For example the honeycomb, shown in Figure 1.1a, produced from honeybees is designed to hold a certain amount of fluid and not fail under the resultant gravitational load caused from the honey, larva, and bees [2]. The natural cork, shown in Figure 1.1b, has been optimized to insulate and protect the core of a tree from injury and/or disease [3].

![Image of natural cellular structures: (a) honeycomb and (b) cork.]

Cellular structures are also commonly used in applications such as aerospace, automotive, and marine. The most commonly used cellular structures are honeycomb or closed-cell foam structures in which cell walls are used to encapsulate a closed volume of empty space. These structures are shown in Figure 1.2. An aluminum honeycomb block (4" x 2" x 2") is shown in Figure 1.2a and a cross-sectional view of laser-formed closed-cell metal foam is shown in Figure 1.2b. Although they do have the high strength and lightweight properties that a low-density structure can provide, they also pose problems due to the closed-cell architecture, including moisture, gas retention, thermal conductivity, and limited improvements in strength versus relative density [4]. A cell wall also increases the stiffness of the global structure, which reduces the amount of possible absorbed energy through impact. On the other hand, a lattice truss structure
(LTS) will reduce the stiffness and increase the energy absorption [5]. A LTS allows for infinite changes to the unit-cell geometry because of the flexibility of additive manufacturing (AM). These small or large changes will result in different impact responses which can lead to improvements in static or dynamic load capabilities.

![Man-made cellular structures: (a)Honeycomb and (b)Laser-Formed Foam][6]

**Figure 1.2**

1.1.3 **Lattice Structures**

Lattice truss structures can also be seen in nature, which coincidentally are most commonly found on some of the most weight-efficient species on the planet [6]. Cross-sectional views of a Hornbill bird beak and an Avian wing bone are shown in Figure 1.3a and 1.3b, respectively [7]. The truss-like members increase the stiffness and strength of the beak while the thin solid faces handle the compressive and tensile loads caused by the Hornbill using its beak to fight and catch prey. To achieve flight, the weight of the large beak must also be kept to a minimum. Similar truss elements can be seen in the section view of the Avian wing bone.
It should be noted that the nodal connectivity of the truss elements has been naturally optimized to reduce the stress concentration caused from a sharp corner by applying a radius at every nodal connection. These naturally evolved lattice-core sandwich panels make way for the design of structures that can potentially have the highest energy absorption capabilities. However, the fabrication of these LTS is not possible with conventional machining techniques. Many attempts [9,10,11,12] have been made to produce simple lattice structures; while some have been effective, most are costly and time intensive. Additive manufacturing processes are able to create complex lattice structures like that shown in the Hornbill beak or Avian wing bone.

1.1.4 Additive Manufacturing

The need for complex geometrical components has driven a market for improving and developing new AM processes. Complex manufacturing capabilities, rapid design-to-fabrication cycle times, and most importantly a minimal amount of waste material
generated has influenced AM to gain respect in the manufacturing industry [8]. Many varieties of AM platforms are available to choose from, but the process used for this work is Fused Deposition Modeling (FDM). A diagram depicting the FDM process is shown below in Figure 1.4.

![FDM Diagram](image)

Figure 1.4: FDM Diagram showing the: (1) extrusion nozzle, (2) extruded molten material and (3) bed plate.

This method works by unwinding a plastic or metal wire from a coil and pushing it through a heated extrusion nozzle by which small flattened strings of molten material are deposited in layers [8]. The quality of the print is dictated by the layer-thickness capability of the 3-Dimensional (3D) printer. The accuracy of the manufactured component compared with the stereo lithograph (STL) file generated by the CAD software increases as the layer thickness decreases. 3D printing software breaks the STL file down into layers and generates a tool path for the extrusion nozzle to follow. Stepper or servo motors drive the extrusion nozzle to move in all three degrees of freedom (x,y,z). The material hardens almost immediately upon exiting the extrusion nozzle. If needed, support material is used to help support extrusions hanging out in space. The support material is liquid (water and/or chemical) soluble so that it can easily be removed after manufacturing. Because of the impressive 3D capabilities of these AM methods,
configurations of LTS can be rapidly designed and tested to work towards improving material properties of the structures such as relative density, stiffness, strength, and energy absorption.

1.1.5 Energy Absorption

During an impact many physical phenomena occur during an impact such as elastic and plastic deformation, shock, fracture and fragmentation, perforation and spallation [9]. Catastrophic failures in aerospace, automotive, and military applications drive the need to mitigate these failures by increasing the amount of absorbed energy caused from an impact. In the aerospace industry, the reduction of damage caused from hail and bird impacts is of high relevance [11]. The automotive industries are utilizing these high-energy-absorbing materials to protect the passengers of vehicles during collisions. Every unit of energy that is absorbed from the structure is one less possible unit that is applied to the passengers of the vehicle. Cellular structures can also reduce the plastic deformation on the interior of the vehicle because of the outer lattice structures' ability to absorb energy and deform under impact. The investigation of energy-absorption capabilities and improvements of a lattice structure is highly desired. The failure mechanisms must be well known to efficiently and accurately design and optimize the energy absorption of these structures.

1.1.6 Failure Mechanisms

The failure mechanisms of the polymer lattice-core structure will be analyzed in this thesis. Failure mechanisms are the underlying cause for plastic deformation in a specimen such as bending, buckling, fracture, corrosion, creep etc. [11]. The failure mechanisms of a LTS will be isolated to the failure of a single slender truss element or
the failure at the nodal connections. During an impact from a random projection angle, a single truss element can undergo compressive, tensile, bending, and torsion loads. Depending upon the orientation of the element, these loads can induce both normal and shear stresses. These stresses produce different types of failure depending upon the ductility of the material. Figure 1.5a shows an image of fracture planes caused by internal pressure, bending, torsion, and tension on a brittle material. The ABS (Acrylonitrile Butadiene Styrene) material is more brittle than ductile and should have failures similar to those shown in the figure.

Buckling of individual elements can also occur and different order mode shapes, (like those shown in Figure 1.5b), are possible during complete failure of a column. The geometry and loading conditions of an element strongly dictate whether buckling will occur and the mode shape that will occur during failure. According to Euler's buckling theory, the ratio of the thickness of the element to its length can be optimized to eliminate this buckling phenomena [12]. Understanding all of the failure mechanisms of a polymer LTS will help further improve the strength and stiffness of these structures to achieve the desired performance for each application, while optimizing other properties such as size and weight.

![Figure 1.5: (a) Fracture planes caused by different loading conditions and (b) Mode Shapes induced from Euler Buckling.](image-url)
CHAPTER 2: LITERATURE REVIEW

2.1 Overview

The goal of this literature review is to understand the previous research that has been done on topics related to this thesis, such as lattice-truss structures, low-velocity impact testing, and energy absorption. Manufacturing processes of lattice structures are discussed along with the structural performance of the lattice-cored sandwich panels. Similar low-velocity impact tests are researched. Their experimental procedures are discussed (i.e. specimen properties, test conditions and Data Acquisition (DAQ)). If the test yields the energy absorption of the specimen, then the post-processing methods of the acquired data will be discussed. Finally the goals of this thesis will be listed and explained briefly.

2.2 Lattice Truss Structures

Lattice-truss structures have been proven to have exceptional load capacities when compared to their counterparts. However, the manufacturing of a lattice truss structure has its difficulties. Multiple manufacturing processes of LTS have been developed through the years.

Sypeck et al. [13] developed cellular-metal-truss core sandwich structures from hexagonally perforated 304 Stainless Steel (SS) sheets. The perforated sheets were plastically deformed using a guided press to form the tetrahedral truss cores shown in Figure 2.1a below. The single-layer truss cores were bonded between thin 304 SS face sheets using a transient liquid phase approach, Figure 2.1b. The bending strength,
core modulus, and relative densities were analyzed through a mid-span loading procedure and a quasistatic unload-reload scheme.

![Image](image_url)

Figure 2.1: (a) A tetrahedral truss core after forming and a (b) typical core/facesheet bonding interface.

It was found that the relative properties of the lattice structures significantly exceed that of foams, and show promises as a counterpart to honeycomb. Queheillalt et al. [14] used the same manufacturing approach on a hexagonally perforated sheet of Ti-6Al-4V (Ti64) to create a tetrahedral truss-core structure. Their cores were sandwiched between Ti64 face sheets using diffusion bonding. The stress-strain response was analyzed using compressive and shear loading tests and found to be comparable with other similar lattice-truss-core sandwich panels. While the sandwich panels resisted well in the compression tests, they exhibited extensive node debonding and truss fracture during shear loading. Queheillalt suggests that a mitigation strategy is needed to increase the node fracture strengths and fully optimize the nodal bonding processes and interface geometries.

Wang et al. [15] studied the performance of truss-core sandwich panels with 3D Kagomé cores like those shown in Figure 2.2a. These panels were fabricated by
investment casting using a copper-beryllium alloy and tested in compression, shear, and bending.

Figure 2.2: (a) A truss core sandwich panel with solid face sheets and a 3D Kagomé core and (b) a unit-cell of the 3D Kagomé core.

The casting is manufactured using an ABS model of the structure which is created by 3D printing a model generated from CAD software. Through a series of dips into a ceramic slurry, a ceramic shell is formed around the ABS. The ABS is then exposed and removed through a burnout process and the residual ash is removed through a water rinse and air-jet process. Imperfections in the casting porosity were found to compromise the ductility of the ligaments. This resulted in premature ruptures (or failures) during shear and bending tests. At equivalent core densities the Kagome structure showed a superior performance in terms of a greater resistance to plastic buckling to other truss structures.

Along with the previously discussed processes, many others have been developed to form lattice-truss structures, such as casting, forming, extruding, bonding, and laser welding. Although all of these structures have high relative strengths comparable with honeycomb, the design space is limited by the manufacturing process. Cutting and forming perforated sheets poses problems such as multilayer nodal connectivity and
limits options for the optimization of the geometrical properties of the truss elements. A laser, water jet, or press operation must be used to remove the perforations in a metal sheet. These processes limit the geometry of the truss to a square or rectangle and also result in sharp corners which lead to undesired stress concentrations. Investment casting poses possible geometrical limitations that reduce the design space. The manufacturing process can leave residual ash in the channels of the casting which cause voids in the surface of the casting. Casting can also result in undesired porosity that leads to failure of the elements. The manufacturing process is also time intensive and not cost effective. Williams [8] explored a layer-based AM process for fully designed mesostructures. He found that the conventional methods of manufacturing lattice structures had the following four severe limitations: non-repeatable results, limited materials, limited mesostructure topology, and limited part geometry. He introduced a new classification of cellular structures called designed mesostructures. These are cellular structures that have material selectively placed in locations to specifically achieve the parts design objectives (i.e. low mass, high strength, and high stiffness). Williams states that the use of AM is the most efficient method to fabricate an optimized lattice-truss core. This process gives the designer the ability to place small amounts of material in specific locations of the structure to optimize the desired thermal and mechanical properties.

2.3 Low-Velocity Impact Testing

Mines et al. [16] explored the low-velocity drop-weight-impact behavior of a Selectively Laser Melted (SLM) Ti64 BCC lattice structure with Carbon Fiber Reinforced Plastic (CFRP) skins. A BCC unit cell is shown in Figure 2.3a and the Ti64
lattice truss core that was analyzed is shown in Figure 2.3b. The specimen sizes were 100 x 100 x 20 mm.

![Image](image_url)

Figure 2.3: (a) A BCC unit cell and (b) a 3D lattice structure using a BCC unit cell.

Woven carbon-epoxy prepreg skins were laminated to the core using an epoxy resin and a hot press. The specimen was placed on four hemispherical supports and subjected to an impact from a 10-mm-diameter hemispherical impactor. The impactor height was 2-m and a double guide-rail system was used to align the impactor. A high-speed video camera was used to capture images and obtain the velocity-versus-time curve of the impactor. A Butterworth 700-Hz low-pass filter was applied to the velocity data to remove noise from the measurement system. Acceleration, displacement, and force were derived from the velocity data. The load-displacement curve of the specimens was plotted and found to be repeatable for the testing of identical configurations. The absorption energy of the specimens was not obtained through the analysis. However, Mines [16] determined that the impact performance of the Ti64 is competitive with aluminum honeycomb. They state that there is potential for improvements in the mechanical properties of the lattice core. Using the same machine and test conditions, Shen et al. [17] studied the low-velocity-impact performance of the same BCC Ti64 lattice structure compared with a BCCZ lattice structure. The BCCZ unit-cell and lattice
structure is shown in Figure 2.4a and 2.4b below. They showed that the impact resistance of the lattice structure was dependent upon the unit-cell geometry. Improvements in impact resistances were obtained by increasing the density of the structure through geometrical changes, unit cell configurations, and AM parameters.

Figure 2.4: (a) A BCCZ unit cell and (b) A 3D lattice structure using a BCCZ unit cell.

Hundley et al. [5] studied the low-velocity-impact response of sandwich panels with lattice-core reinforcements. They compared a UV-cured photopolymer lattice to an aluminum alloy cast from an initial polymer template. The photopolymer is a BCC lattice structure adhered to 6061-T6 aluminum face sheets as shown in Figure 2.5. The second sandwich panel consisted of a 6101-T6-aluminum-alloy-casted BCC lattice core with 6951-O aluminum alloy face sheets. The boundary conditions for the 4" x 6" specimen followed the D7136 Standard which are discussed in section 3.1.1 Apparatus Design. The velocity and force histories were obtained through an accelerometer and force transducer, respectively, positioned at the center of the impactor. Through analysis of the velocity and force histories, they showed that the differences in material and architectural combinations could be captured experimentally and analytically. The stress-strain relationships or energy absorption were not discussed.
Figure 2.5: (a) UV-cured photopolymer lattice cored sandwich panel with 6061-T6 aluminum face sheets and (b) 6101 aluminum microlattice-cored sandwich panel with 6951-O aluminum face sheets.

2.4 Energy Absorption

Ashby et al. [18] developed a design guide for metal foams and discuss the energy management for applications such as packaging and blast protection. Ashby characterizes the load-displacement curve shown in Figure 2.6 which is derived from the compression testing of an energy absorber. The curve is split into three different sections: linear, plateau, and densification. The initial linear section of the curve is the elastic region where the material elastically deforms until failure occurs, at which point the plateau region begins. In this region, the absorber collapses plastically at a constant force, $F_m$, to a limiting distance, $\delta_D$. The area under the plateau region is the useful energy, $W$, which can be absorbed.
The longer and higher that the plateau section is, the more energy the structure can absorb. The densification section occurs when the nominal distance is met and the entire structure has plastically deformed and collapsed on itself. They state that the area under the curve, approximately $F_m \delta_d$, is the energy absorption up to the densification section. Although this load-displacement curve is through compression of the absorber (rather than a low velocity impact) it has similarities to that from a dynamic impact. The greatest difference is in the initial peak load and densification section. During an impact, the impactor has a maximum kinetic energy prior to contact where as a static compression test supplies a constant force or displacement. At the initial point of contact the structure will elastically compress until the ultimate stress is reached, at which point initial failure occurs. During this event the kinetic energy of the impactor is reduced until maximum deflection of the specimen occurs (end of the plateau section). At this point the impactor does not have enough energy to generate a densification region as large as that in a static compression test. The asymptotic increase in force is caused by a constant supplied energy from the compression machine and the total collapse of all layers of the absorber. Similar approaches can be considered to characterize the load-displacement curve of a LTS under a low-velocity impact. The mechanical properties of a LTS (i.e.
stiffness, elastic modulus, energy absorption etc.) can be obtained through the analysis of the load-displacement curve captured during a low-velocity impact.

Zhang et al. [19] investigated the absorption energy of pyramidal lattice sandwich panels under low-velocity impacts. They compared polyurethane-foam-filled pyramidal-lattice-sandwich panels to similar structures with no foam in the core. A Dynatup Model 9250HV impact tester was used for capturing the impact response of the specimens. A force transducer was placed above the impactor to capture the force history, the methods for capturing the velocity were not discussed. The machine captured the load-displacement curve that was integrated to determine the absorbed energy. He found that the foam-filled core did not produce significant differences to the impact response. Even though a small increase in load capacity was observed, the improvement was large enough to justify the increase in weight of the structure.

2.5 Goals of the Thesis

The goal of this thesis is to study/characterize the differences in energy absorption capabilities of LTS sandwich panels. Different unit-cell configurations are designed and fabricated for a comparative study. Aluminum Honeycomb is used to fabricate a sandwich panel using identical Kevlar face sheets and fabrication processes. An Impact Machine will be designed to capture the low-velocity-impact response of the sandwich panels. Post-processing techniques for the acceleration data are explored. The load-displacement curve during the impact event for each specimen is captured and integrated to determine the energy absorbed during impact. The failure mechanisms of the lattice truss elements are then analyzed to help further improve the mechanical properties of the LTS.
CHAPTER 3: EXPERIMENTAL METHODS

3.1 Low-Velocity Impact Testing

3.1.1 Apparatus Design

An impact machine was designed following the ASTM Standard D7136/D7136M-15 [21] to investigate the low-velocity-impact response of the test specimen. SolidWorks®, a computer-aided-design (CAD) software was used to develop a 3D model of the machine. The CAD software uses the model to help create a drawing package in which the individual components and assemblies are detailed. The drawing package is then used for documentation and reference during fabrication and is shown in Appendix B. Figure 3.1a shows a rendered image of the solid model and Figure 3.1b shows an image of the fabricated and fully assembled impact machine. A double-column impactor-guide mechanism was chosen for the design. The impactor assembly is guided along two circular rods to strike the center of the specimen. A latching mechanism, consisting of a high-tension push-button pull pin, is used to adjust the height of the impactor assembly up to 27" above the specimen. Four high-strength locking toggle clamps are used to secure the 4" by 6" specimen to the test plate. Figure 3.2 shows the clamping locations and boundary conditions specified in the D7136 Standard. Four pins are pressed into the test plate to locate the specimen over the 3" by 5" cutout in which a uniform thickness of 0.5" is supported around the perimeter.
Figure 3.1: (a) Rendered image of impact machine and (b) Fully assembled impact machine in AM lab.

Figure 3.2: ASTM D7136 specified clamping locations and top plate dimensions.
AISI 4140 was chosen for the material of the structure because of its high machinability and strength. However, an application of black oxide was needed to increase the steels corrosion resistivity. The hemispherical impactor which is shown in Figure 3.3a was made from Ti64 with a diameter of 0.375". A 32-micron surface finish was applied to the cylindrical and hemispherical surfaces to avoid friction caused by the woven face sheets during impact. After assembly the overall dimensions of the machine are 12" wide x 10" deep x 40" tall.

Toggle clamp risers were designed and fabricated using 1018 low-carbon steel to allow for different thicknesses of specimens. The risers were then painted with a durable black paint to avoid oxidation. The machine can support a specimen up to 1.5" thick. Figure 3.3b shows a close-up view of a toggle clamp mounted on a riser.

Because of the cost and time requirements to additively manufacture a 4" by 6" polymer LTS, it was decided to reduce the specimen dimensions to 2" by 2". A fixture was designed to support the smaller specimen and is shown in Figure 3.3c. The 2x2 fixture uses the cutout on the test plate to locate the specimen at the center of the impactor. The smaller 2x2 specimen sits in a 2.25" x 2.25" recess in the fixture with a 0.125" gap around the specimen. The entire bottom surface of the specimen is bounded from the fixture. A top plate is used to apply a downward pressure across a uniform thickness of 0.125" around the perimeter of the specimen through the use of the four toggle clamps.
The impact energy and final velocity of the impactor can be calculated by solving the following equations below assuming the friction from the guide rails is negligible:

\[
\Delta E = PE - KE = 0 \quad (1)
\]

\[
PE = mgh = wh \quad (2)
\]

\[
KE = \frac{1}{2}mv^2 \quad (3)
\]

Figure 3.4 contains a plot of the theoretical velocities with the machine depending upon the height of the impactor assembly. The velocity upon impact can be adjusted by varying the height at which the impactor is released. Extra mass blocks were fabricated from 1018 low carbon steel to vary the mass of the impactor assembly, which in turn varies the impact energy. The energy levels at which the machine can strike the specimen are shown in Figure 3.5. This plot shows the available energy space with respect to the weight of the impactor. The dashed lines show the available energy levels at different drop heights.
Figure 3.4: Achievable velocities of the impact machine with respect to the height of the impactor assembly.

Figure 3.5: Available Energy levels of the impact machine with respect to the weight of the impactor.
3.1.2 Data Acquisition System

Force and acceleration histories are needed to obtain the absorption energy. Integrated Electronic Piezoelectric (IEPE) load cells were chosen along with an Integrated Circuit Piezoelectric (ICP) accelerometer. Both acronyms refer to the same type of measurement. They require that the dynamic sensors incorporate microelectronics that convert the charge signal from high to low-impedance. This change in impedance allows the dynamic signal to be easily transmitted [22]. These types of sensors were selected because of their high dynamic range and noise immunity.

Four Dytran 1051V5 IEPE Dynamic Force Sensors were selected to measure the force. The sensors were sandwiched between the four columns supporting the ASTM specified bed plate. The locations of the load cells are shown in Figure 3.1b. Each load cell can measure in a dynamic load range of +/- 1,000 lbf, with a bandwidth of 50 kHz, and a resolution of 0.07-lbf. The range of the load cell was selected by calculating the maximum possible force due to the dynamic energy of the impactor. The dynamic energy upon impact is found by substituting the final velocity of the impactor into equation (3). The kinetic energy is equal to the work done by the impact force that slows down the impactor which is shown below in equation (4).

\[ W = Fd \] (4)

where \( F \) is the impact force and \( d \) is the distance required to slow down the impactor (estimated dent depth). The impact force can be derived by setting equations (3) and (4) equal to each other and solving for \( F \).

\[ F = \frac{1/2mv^2}{d} \] (5)
Using the maximum mass of the impactor, including additional mass blocks for increased energy levels, and an estimated worst case scenario for a minimum dent depth of 0.050" yields an impact force of 1,890-lbf. Because of the configuration of the load cells, each sensor only sees one-fourth of the maximum impact force. It is not recommended to collect the data at the maximum range of the sensor, therefore choosing 1,000-lbf load cells allows the measurement to be taken at roughly 50% of the range.

To capture the acceleration of the impact a single PCB 352C23 ICP accelerometer was adhered to the center of the top surface of the impactor assembly. The dynamic accelerometer can operate between a g-force range of +/- 1000 g, at a frequency of 50 kHz, and a resolution of 0.03-g. The maximum g-force, \( a \), seen by the accelerometer can be calculated by using Newton's second law of motion:

\[
F = ma
\]  \hspace{1cm} (6)

where \( F \) is the previously calculated maximum impact force, \( m \) is the mass of the impactor and solving for \( a \) yields equation (7):

\[
a = \frac{F}{m}
\]  \hspace{1cm} (7)

which results in a maximum g-force of 767 g's. To help reduce noise caused by unwanted vibrations upon impact, all cables were strain relieved following the manufactures recommendations.

A National Instruments cDAQ-9174 4-slot USB chassis was selected to collect data simultaneously from multiple sensors and/or sensor types via plug and play modules. Two NI-9234 4-channel IEPE modules, with a sampling rate of 51.2 kS/s/ch, were used to collect the data from both the accelerometer and load cells. NI Signal Express Labview software was used to run the DAQ system and collect the data onto a Windows-
based Dell workstation. Once the experiments were complete the acceleration and force data was post-processed in Matlab to determine and plot the absorption energy of the test specimen (Matlab code is shown in Appendix A). An image of the DAQ system and the entire impact testing setup is shown in Figures 3.6 and 3.7, respectively.

Figure 3.6: DAQ System

Figure 3.7: Impact Testing System.
3.1.3 Data-Processing Methodology

The absorption energy, $W$, can be obtained by integration of the load-displacement curve and is shown in the following equation:

$$ W = \int_{0}^{x} F(x) \, dx \tag{8} $$

where $F$ is the force history recorded from the dynamic load cells and $x$ is the displacement history obtained from the accelerometer. The force history is easily obtainable through a summation of all 4 load cells. This summation is a sufficient method because of the dynamic nature of the IEPE load cells. The sensors are only excited through dynamic loading and the recorded values approach zero over time if no excitation occurs. The displacement however, requires double integration of the acceleration data which results in lost integration constants and necessitates a velocity shift. Data accuracy can be compromised by an incorrect shift. Boundary conditions for the velocity integration can be determined through the following simple dynamics.

Consider an object falling in free fall striking a horizontal surface (Figure 3.8a) where $d_1$ is the free-fall distance and $d_2$ is the maximum height achieved after the first bounce. The free-fall distance is known and the height of the bounce can be obtained through a high-speed camera. The acceleration of this falling object, assuming friction and drag is negligible, is shown in Figure 3.8b. Equation (9) shows the integral required to obtain the velocity curve.

$$ v = \int_{0}^{t} a(t) \, dt \tag{9} $$

This results in a lost integration constant which can be found by determining the velocity immediately before and after impact. The velocity of the falling object, shown in Figure 3.8a, can be determined through equation (10) and the displacement through equation
Combining equations (10) and (11) yield $v_1$ and $v_2$ which are the initial and final velocities, respectively during the impact and are shown in equations (12) and (13).

\begin{align*}
v(t) &= -gt \\ d(t) &= \frac{1}{2} gt^2 \\ v_1 &= -\sqrt{2gd_1} \\ v_2 &= \sqrt{2gd_2}
\end{align*}

These velocities can be used as boundary conditions for correctly shifting the velocity plot. A correctly shifted velocity plot is shown in Figure 3.8c.

Figure 3.8: (a) Object in free fall striking a horizontal surface (b) Acceleration curve of the object striking the surface (c) Correctly shifted velocity curve during impact.
The integration of this correctly shifted velocity curve yields an accurate displacement curve that has a maximum value where the velocity changes from negative to positive. The maximum value on the displacement curve is the dent depth on the specimen being impacted and can be used to plot the load-displacement/stress-strain curve. Shown in equation (1), the integration of the load-displacement curve will result in the desired absorption energy of the polymer lattice-cored sandwich panel.

Two different methods can be used for validating the absorption energy obtained from the integration of the load-displacement curve. The first method for capturing the absorption energy uses the response from the accelerometer. The Energy of the impactor can be determined by integrating the momentum with respect to the velocity. The integral for the Energy, $E$, is:

$$E = \int_{v_i}^{v_f} \rho v \, dv$$  \hspace{1cm} (14)

where $\rho$ is the momentum and $v_i$ and $v_f$ are the initial and final velocities, respectively. The initial velocity is the velocity at the point of contact and the final velocity is the maximum velocity achieved after the impact. The integration of equation (14) yields the Kinetic Energy (KE) equation (3). To obtain the absorption energy the KE found after the impact must be subtracted from the initial KE at contact.

The second method for validation is done using a high-speed camera to capture the maximum height of the impactor after impact has occurred. The height is used to determine the maximum Potential Energy (PE) achieved after impact. Equation (15) shows the energy absorbed, $\Delta E$, through the impact by subtracting the KE from the PE, while the equations for determining the KE and PE are previously shown in equations (2) and (3).
3.2 Test Specimen

3.2.1 Face sheets

Hexcel Composites' K285-38"-F161 Kevlar fabric was used for the face sheets. The K285 is a Kev. 49 1140 Fiber with a crowfoot weave and has a fiber areal weight of 557 lbs/ft². The Kevlar is pre-impregnated, pre-preg, with a F161 high-temperature, laminate-grade epoxy resin. The properties of the resin are shown in Table 1 [23].

Table 1: Woven Kevlar Pre-impregnated Epoxy Resin Properties.

<table>
<thead>
<tr>
<th>Epoxy Resin</th>
<th>Tensile Strength (ksi)</th>
<th>Tensile Modulus (msi)</th>
<th>Tensile Strain (%)</th>
<th>Fracture Toughness (ksi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>F161</td>
<td>8.7</td>
<td>0.52</td>
<td>2.2</td>
<td>0.394</td>
</tr>
</tbody>
</table>

Four layers of the pre-preg material were bonded together in the hot press shown in Figure 3.9a. A pressure of 3 metric tons was applied to the Kevlar at a temperature of 428°F for 3 hours. The bonded face sheets were allowed to cure for 24 hours before being adhered to the LTS or aluminum honeycomb which is discussed in section 3.2.4 Sandwich Panel Fabrication. Figure 3.9c shows a microscopic image of the 4 ply bonded Kevlar. The 4 layers of Kevlar can be seen in the image along with the 3 layers of cured epoxy. Analysis of the images showed that the interface between the pre-preg Kevlar had uniform adhesion with the cured epoxy resin.
3.2.2 Lattice Core-Design and Fabrication

Solidworks was used to model the LTS shown in Figure 3.10a. The structure consists of linearly patterned unit cells on a base plate with a thickness of 0.087”. A polymer base can be used instead of Kevlar to reduce fabrication time and cost because the energy level of the impact machine was designed to not penetrate through the entire specimen. The pattern consists of 4 unit cells copied in the z-direction (vertical) and 10 unit cells copied in both the x and y-directions. The overall dimensions of the structure shown are 2” x 2” x 1” in thick. Two different unit-cell configurations were considered for this work to capture the differences in absorption energy. In both configurations the circular cross section of the truss elements has a diameter of 0.049” and the unit cells have an overall height of 0.228” and width of 0.200”. The well-studied common BCC unit cell was chosen for the first configuration and is shown in Figure 3.10b. The second unit-cell configuration consists of a BCC unit cell with an alternating vertical strut (BCCAV). An image of this configuration is shown in Figure 3.10c. The BCCZ
structure shown in the literature-review chapter and researched by Shen et al. [17] exhibited a higher relative density and strength than the BCC. A modification to the BCCZ structure is an attempt to increase the absorption capabilities. While the vertical strut increases the strength of the structure, reducing the number of vertical members should in theory increase the amount of absorbed energy when compared with the BCCZ. The overall topological difference of the BCCAV unit-cell configuration can be seen in the side-views shown in Figure 3.11a and 3.11b. The BCC lattice structure produces a 3D diamond lattice when patterned. The BCCAV creates the same 3D diamond lattice with an additional alternating vertical strut. The BCCAV unit cell must be mirrored over the top plane and patterned in all directions to produce the LTS. Figure 3.11a shows the result of this in an orthographic side view, in which a single vertical strut alternates between the layers of diamond lattice. A perspective side view of the BCCAV is shown in Figure 3.11b.
Figure 3.10: (a) Rendered image of the 3D lattice truss core using a BCC unit-cell (b) BCC unit-cell (c) BCCAV unit-cell.
Figure 3.11: (a) Orthographic side view of BCCAV LTS and (b) a perspective side view of BCCAV LTS.

A Stratasys uPrint SE Plus 3D printer (Figure 3.12) was used to manufacture the polymer LTS. The printer has a layer-thickness capability of 0.013". with a maximum build size of 8" x 8" x 6". The bed plate allowed for the printing of 4 lattice structures in one build session after the specimen size was reduced to 2" x 2" x 1". The time required to print 4 specimens was 28 hours.
CatalystEX software was used to convert the STL file created from the CAD software to 3D modeling print paths and included all required support structures. The model head temperature of 590°F was used with the default settings to print the chosen material (ABSplus-P430). The model material is an ivory-colored production-grade thermoplastic; its mechanical properties are shown below in Table 2 [24].

<table>
<thead>
<tr>
<th>Model Material</th>
<th>Ultimate Strength (ksi)</th>
<th>Yield Strength (ksi)</th>
<th>Modulus (Msi)</th>
<th>Strain (%)</th>
<th>IZOD Impact, notched (ft-lb/in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABSplus-P430</td>
<td>5</td>
<td>5</td>
<td>0.32</td>
<td>6</td>
<td>2</td>
</tr>
</tbody>
</table>

After the printing was complete the structures soaked in a heated water bath for 3 to 4 hours to remove the support structure. The bath consists of water and a chemical agent provided by Stratasys. The tank heats and circulates the water to accelerate the support
structure removal process. An image of a BCC unit-cell lattice structure with the un-
removed support structures is shown in Figure 3.13. The face sheet can be laminated to
the lattice core after the support structure is removed and allowed to completely dry. The
unit cell shown in the figure is a BCC. A microscope was used to capture images of the
print quality of the printer shown in Figure 3.14. The microscope was focused on the
center node of the BCC unit cell. The thickness of the center node is 0.050" and can be
used as a reference to compare with the thickness of the layers of molten deposited ABS
seen in the image. The layer thickness matches the manufacture's specifications and the
quality of the print appears to be fairly high.

Figure 3.13: Lattice core with support structure.
3.2.3 Aluminum Honeycomb Core

A 4" x 4" x 1" aluminum honeycomb core was saw cut into 4 equal specimens with a size of 2" x 2" x 1". The hexagonal honeycomb is a 5052 H39 aluminum alloy with a 1/4” cell width and a 1-mm wall thickness. The sandwich panel fabrication processes are the same as that of the LTS core specimens to provide a reference point for load capacities and absorption capabilities. An additional Kevlar face sheet is laminated to the bottom surface of the honeycomb to avoid any undesired damage.

3.2.4 Sandwich Panel Fabrication

Loctite Heavy-Duty Epoxy was used to adhere the Kevlar face sheets to the polymer LTS or Aluminum Honeycomb. The epoxy is a two-part adhesive consisting of an epoxy resin and a hardener. The evenly mixed resin and hardener react with each other to produce a tough, rigid bond. Metals, ceramics, and rigid plastics are the recommended materials for use with this epoxy and when cured for the recommended time of 24 hours can form a water and high-impact resistant bond site. A static pressure of 5-10 lbf was applied to the sandwich panel for 24 hours to allow the epoxy to fully
cure before testing. Images of the epoxy on the 4 ply Kevlar face sheets and the finished polymer LTS sandwich panel are shown in Figures 3.15 and 3.16, respectively.

Figure 3.15: Woven Kevlar face sheet with pre-mixed, two-stage epoxy resin applied to the top surface.

Figure 3.16: Completed sandwich panel with polymer BCC LTS core.
The core-skin interface was analyzed through high magnification images shown in Figures 3.17 to ensure that face sheets were adequately bonded to the core structure. Lack of adequate adhesion could result in delamination which can affect the impact response and lead to inaccurate absorption energy values. The bond site is observed to be uniform across the face sheet and seen to have a high debonding strength [24]. The radii of the cured epoxy seen at the core skin interface increase the contact area and help mitigate localized stress.

Figure 3.17: Microscopic image of core-skin interface.

Computed Topography (CT) scans were also conducted by David Roberts in the Non-Destructive Materials Testing Group at Wright Patterson Air Force Base to analyze the uniformity of the 3D printed material. A uniform print is beneficial due to the mitigation of localized stresses caused from voids in the material. Localized stress can lead to unwanted failure of the individual truss elements. The scans showed the LTS to be a uniform print. The layers of material bonded to one another will little to no defects or voids.
CHAPTER 4: RESULTS

4.1 Overview

Ten polymer lattice core sandwich panels and 4 aluminum honeycomb sandwich panels were fabricated and tested under low-velocity impacts. For the LTS, a BCC and BCCAV were the two unit-cell configurations chosen for the comparative study. Four different energy levels were selected for the experiments. Table 3 below shows the test conditions and specimen properties for each experiment. Specimens 1-5 were BCC unit cells and specimens 6-10 were BCCAV unit cells. Specimens 12-15 were the aluminum honeycomb specimens. All 14 specimens were fabricated using the same methods discussed in section 3.2 Test Specimen. The impactor velocity shown was calculated using the methods discussed in section 3.1.3 Data-Processing Methodology. The energy levels were varied by adding additional mass blocks to the impactor assembly. The energy levels shown in the table were calculated using the kinetic energy (KE) of the impactor where \( KE = \frac{1}{2}mv^2 \).

Table 3: Specimen Properties & Test Conditions.

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Unit Cell Configuration</th>
<th>Weight of Core (oz)</th>
<th>Impactor Velocity (ft/s)</th>
<th>Weight of Impactor (oz)</th>
<th>Impactor Energy (ft-lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1, 6</td>
<td>BCC, BCCAV</td>
<td>0.47, 0.55</td>
<td>11.25</td>
<td>51</td>
<td>6.3</td>
</tr>
<tr>
<td>2, 7</td>
<td>BCC, BCCAV</td>
<td>0.47, 0.55</td>
<td>11.25</td>
<td>39</td>
<td>4.8</td>
</tr>
<tr>
<td>3, 8</td>
<td>BCC, BCCAV</td>
<td>0.47, 0.55</td>
<td>11.25</td>
<td>51</td>
<td>6.3</td>
</tr>
<tr>
<td>4, 9</td>
<td>BCC, BCCAV</td>
<td>0.47, 0.55</td>
<td>11.25</td>
<td>86</td>
<td>10.6</td>
</tr>
<tr>
<td>5, 10</td>
<td>BCC, BCCAV</td>
<td>0.47, 0.55</td>
<td>11.25</td>
<td>107</td>
<td>13.2</td>
</tr>
<tr>
<td>12, 13, 14, 15</td>
<td>Aluminum Honeycomb</td>
<td>0.38</td>
<td>11.25</td>
<td>107</td>
<td>13.2</td>
</tr>
</tbody>
</table>
The penetration depths were similar between the BCC and BCCAV at the lower energy levels. As the energy level of the impactor increased, the extent of the damage increased as expected. However, the difference in damage was not visually noticeable between the BCC and BCCAV. Images of specimens 5 and 10 after impact are shown in Figures 4.1a and 4.1b, respectively. Images of the aluminum honeycomb specimens are shown in Figure 4.2. The damage of the aluminum honeycomb is much more extensive than that of the polymer LTS. Visible delamination was exhibited on the honeycomb specimens. Pertaining to the polymer LTS, delamination between the lattice core and the Kevlar face sheet occurred on the higher energy level impacts and can be seen in the CT scans, shown in Figures 4.3 through 4.10, where a section view of the model is taken at the center of the specimen. The CT scans reveal a larger penetration depth of the BCC compared with the BCCAV. The penetration depths recorded from the CT scans are shown in Table 5 in section 4.3 Displacement Measurement. It should be noted that the failure mechanisms of the individual truss elements are visible from the CT scans and can be seen in the following images. The failure mechanisms of the LTS are discussed in section 4.7 Failure Mechanisms.

Figure 4.1: Image of (a) specimen 5 (BCC LTS) and (b) specimen 10 (BCCAV LTS).
Figure 4.2: Image of aluminum honeycomb (a) specimen 12 (b) specimen 13 (c) specimen 14 and (c) specimen 15.
Figure 4.3: CT scan of specimen 2 (BCC LTS) impacted at 4.8 ft-lbs.

Figure 4.4: CT scan of specimen 7 (BCCAV LTS) impacted at 4.8 ft-lbs.
Figure 4.5: CT scan of specimen 3 (BCC LTS) impacted at 6.3 ft-lbs.

Figure 4.6: CT scan of specimen 8 (BCCAV LTS) impacted at 6.3 ft-lbs.
Figure 4.7: CT scan of specimen 4 (BCC LTS) impacted at 10.6 ft-lbs.
Figure 4.8: CT scan of specimen 9 (BCCAV LTS) impacted at 10.6 ft-lbs.

Figure 4.9: CT scan of specimen 5 (BCC LTS) impacted at 13.2 ft-lbs.
4.3 Velocity Measurement

The acceleration data is integrated to determine the velocity, as previously discussed. The cumulative trapezoidal integration function in Matlab was used to integrate the acceleration data for all specimens. The integrated data from specimen 10 is shown below in Figure 4.11.

![Velocity vs. Time Plot](image)

Figure 4.11: The integration of the raw acceleration data from specimen 10.

The plot does not accurately represent the velocity of the impactor relative to time. The velocity must be shifted in the negative direction. The methods for correctly shifting the velocity plot are discussed in section 3.1.3 Data-Processing Methodology. The correctly
shifted velocity curve for specimen 10 is shown in Figure 4.12. It should be noted here that the point at which the correctly shifted velocity curve crosses the zero of the velocity axis is when the impactor's velocity becomes zero and no more damage to the specimen is possible. The final velocity can be validated by using the height of the impactor after the first bounce. The maximum height of the impactor after the first bounce is recorded with a high-speed camera. The images taken for Specimen 2 are shown in Figure 4.13a and 4.13b. Figure 4.13a shows an image of the impactor at its furthest depth into the specimen while Figure 4.13b shows the impactor at its maximum height achieved after the first bounce.

![Figure 4.12: Correctly shifted velocity curve for specimen 10.](image-url)
All velocities were shifted and validated with the same methods shown in the *Data Processing Methodology* section. Table 4 below shows all of the heights recorded from the high-speed camera for the polymer test specimens. The table also compares the final velocity measurements to the calculated numerical values. Equation (13) can be used to calculate the velocity, \( v \), using the maximum height, \( h \), achieved from the first bounce after the impact.

**Table 4: Height and velocities achieved after impact for Polymer LTS Sandwich Panels.**

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Height of Impactor after Impact (in)</th>
<th>Theoretical upward Velocity of Impactor using Eq. 12 (ft/s)</th>
<th>Experimental Velocity (ft/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.0</td>
<td>4.0</td>
<td>3.7</td>
</tr>
<tr>
<td>2</td>
<td>3.5</td>
<td>4.3</td>
<td>3.8</td>
</tr>
<tr>
<td>3</td>
<td>3.4</td>
<td>4.3</td>
<td>4.5</td>
</tr>
<tr>
<td>4</td>
<td>1.6</td>
<td>3.0</td>
<td>2.8</td>
</tr>
<tr>
<td>5</td>
<td>1.3</td>
<td>2.6</td>
<td>1.9</td>
</tr>
<tr>
<td>6</td>
<td>2.9</td>
<td>4.0</td>
<td>3.6</td>
</tr>
<tr>
<td>7</td>
<td>3.9</td>
<td>4.6</td>
<td>5.2</td>
</tr>
<tr>
<td>8</td>
<td>2.9</td>
<td>3.9</td>
<td>3.9</td>
</tr>
<tr>
<td>9</td>
<td>1.5</td>
<td>2.8</td>
<td>2.7</td>
</tr>
<tr>
<td>10</td>
<td>0.9</td>
<td>2.2</td>
<td>2.0</td>
</tr>
</tbody>
</table>
Figures 4.14-4.18 show the comparison of velocities between the two unit-cell configurations at different energy levels. The lower energy levels of 4.8 and 6.3 ft-lbs produced noticeably similar damage to both configurations. The BCC and BCCAV have similar topologies for the first two layers of diamond lattice. The vertical strut in the BCCAV will not have a significant effect on the impact response until the deformation reaches the strut. The velocity curves obtained for the lower energy levels appear to have more scatter. This can be explained through the analysis of a low-plastic-deformation impact, in which the displacement of the impactor into the specimen is close to zero and the duration of the impact response, $\Delta t$, is small (1-3 ms). These issues cause an asymptotic increase in both the impact force and maximum g-force experienced by the structure. These increases can cause high-frequency vibrations in the structure which results in unwanted noise. If low-deformation impacts are desired, then increasing the sampling rate, relocating the sensors, and providing damping to the mechanical structure where possible is recommended. Although the data seems to be scattered in the lower-energy-level impacts, both configurations follow a similar trend line and the final velocities closely match the calculated numerical values. The higher energy levels yield a well-sampled velocity curve. Both configurations follow the curve for the first millisecond of impact. This can be the response of the Kevlar face sheet in combination with the elastic deformation of the first layer of lattice. The BCCAV causes the velocity to decrease faster when compared with the BCC. The single strut in the BCCAV structure offers higher stiffness which resists more deformation. This resistance also causes an increase in the impact force which will be shown in the analysis of the force histories.
While the duration of the response is shorter for the BCCAV relative to the BCC, they both reach the same velocity on the bounce. A second integration operation to determine the displacement will highlight the differences in the velocity curves. It can be seen from the plots that the velocity of the impactor on the BCCAV lattice structures consistently decreases earlier than the BCC.

Figure 4.14: Velocity curve for specimens 1 and 6 at 6.3 ft-lbs.
Figure 4.15: Velocity curve for specimens 2 and 7 at 4.8 ft-lbs.

Figure 4.16: Velocity curve for specimens 3 and 8.
Figure 4.17: Velocity curve for specimens 4 and 9 at 10.6 ft-lbs.

Figure 4.18: Velocity curve for specimens 5 and 10.
The velocities for specimens 1-10 are plotted together in Figure 4.19. It can be seen from the plot that the impact velocity takes longer to approach zero as the energy level increases. This is expected because of the higher energy levels causing higher plastic deformation, which results in a larger penetration depth and longer impact response. The plots also show that the velocity reached upon the first bounce decreases as the energy level increases. This is a result of more energy being absorbed from the deeper penetrations which reduces the maximum height or velocity that can be achieved on the first bounce.

The velocity curves for the Aluminum Honeycomb specimens are plotted in Figure 4.20. To adequately compare the velocities between aluminum honeycomb and the polymer
LTS, only the velocity curves for the specimens impacted at the same energy levels are shown. All four Honeycomb specimens were identical and tested under the same conditions. All four honeycomb specimens achieved roughly the same velocity of 1.0 ft/sec after the first bounce. The velocity curves shown for the Aluminum Honeycomb have a longer impact response and a lower final velocity than the polymer LTS. The lower final velocity will result in more energy that has been absorbed from the system. If all of the energy is absorbed in the impact then the velocity of the impactor after the impact will be zero.

Figure 4.20: Velocity curves for the polymer LTS and aluminum honeycomb.
4.3 Displacement Measurement

A second cumulative-trapezoidal-integration method in Matlab is used to determine the displacement of the impactor during contact. Once the velocity value changes from negative to positive, the impactor is no longer in contact with the specimen. The integration will start at the first excitation point on the velocity curve and end at the point at which the velocity changes direction. The cumulative trapezoidal (cumtrapz) method automatically smoothes the data by breaking the area under the curve into infinitesimally small trapezoids. This method eliminates the need for a filter on the displacement data. The displacement curves for all specimens are plotted together in Figure 4.21. It is very obvious that the BCCAV has less deformation than the BCC. It can also be seen that the deformation occurs more rapidly in the BCC structure. The honeycomb has the most deformation at the same energy levels. Although the displacement appears inconsistent between the honeycomb specimens, the scale is relatively small and only seeing a difference of 1/16”. The orientation of the honeycomb, relative to the center of the impact tip is uncontrollable. The impactor could initially make contact with a vertical honeycomb cell wall or in the middle of the hexagonal cell, which can help explain differences in the impact response. The maximum penetration depth obtained experimentally is shown in Table 5 and compared with the value recorded from a depth gauge. The depth gauge was tared at the top of the un-deformed section of the Kevlar face sheet and then taken to the maximum deformation point in the damaged zone. It has been concluded that the depth gauge measurement is inaccurate due to the elastic nature of the Kevlar face sheet and epoxy resin used for lamination. Depth measurements taken from the CT scans are also compared with the experimental values
in Table 5. Once the force history is obtained, a load-displacement curve for each specimen can be plotted using the displacement data shown in Figure 4.21.

![Figure 4.21: Displacement curves for specimens 1-10.](image)

Table 5: Different Displacement Measurements.

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Energy Level (ft-lbs)</th>
<th>Displacement from Acceleration History (in)</th>
<th>Depth Gauge (in)</th>
<th>CT Scan (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6.3</td>
<td>0.249</td>
<td>0.094</td>
<td>N/A</td>
</tr>
<tr>
<td>2</td>
<td>4.8</td>
<td>0.184</td>
<td>0.093</td>
<td>0.191</td>
</tr>
<tr>
<td>3</td>
<td>6.3</td>
<td>0.251</td>
<td>0.097</td>
<td>0.263</td>
</tr>
<tr>
<td>4</td>
<td>10.6</td>
<td>0.353</td>
<td>0.185</td>
<td>0.371</td>
</tr>
<tr>
<td>5</td>
<td>13.2</td>
<td>0.401</td>
<td>0.227</td>
<td>0.410</td>
</tr>
<tr>
<td>6</td>
<td>6.3</td>
<td>0.226</td>
<td>0.091</td>
<td>N/A</td>
</tr>
<tr>
<td>7</td>
<td>4.8</td>
<td>0.186</td>
<td>0.089</td>
<td>0.195</td>
</tr>
<tr>
<td>8</td>
<td>6.3</td>
<td>0.225</td>
<td>0.125</td>
<td>0.246</td>
</tr>
<tr>
<td>9</td>
<td>10.6</td>
<td>0.313</td>
<td>0.177</td>
<td>0.328</td>
</tr>
<tr>
<td>10</td>
<td>13.2</td>
<td>0.371</td>
<td>0.229</td>
<td>0.380</td>
</tr>
</tbody>
</table>
Figure 4.22 shows a comparison of the displacement measurements made on the BCC sandwich panels. The experimental data derived from the acceleration response closely resemble that from the CT scans. Similar results for the BCCAV are shown in Figure 4.23. A comparison of the experimentally captured penetration depths of the BCC and BCCAV is shown in Figure 4.24. The BCC clearly has a larger penetration. Deeper penetration resulting in more plastic deformation usually leads to a higher amount of absorbed energy. However, the BCC capability to withstand an impact from a higher energy level object is lower than that of the BCCAV.

![Figure 4.22: Comparison of displacement measurements of the polymer BCC LTS.](image-url)
Figure 4.23: Comparison of displacement measurements of the polymer BCCAV LTS.

Figure 4.24: Comparison of displacement measurements of the polymer BCC LTS vs BCCAV LTS.
4.4 Force History

Four load cells were used to capture the force history. A summation of all four load cells was done in Matlab to capture the overall force being applied to the specimen. When vibration of the test plate occurs during impact, each load cell can theoretically see a different value. If the load cells are all compression then a summation would be sufficient. If some are compression and others tension then a summation will give an incorrect force history. The response of each load cell during the impact of specimen 10 is plotted in Figure 4.25 to ensure that the load cells are synchronized and balanced. The responses are all positive (compression) and follow the same trend. Identical methods discussed for validating the force history was used for all specimens. All of the recorded force data appear to be uniform and synchronized.

Figure 4.25: Response of all four load cells for specimen 10 at 13.2 ft-lbs.
The total force history for specimen 10 is plotted in Figure 4.26. A 700 Hz low pass Butterworth filter was applied to eliminate noise and smooth the data for visual representation of the load curve. The plot shown represents well known load curves for energy absorbers, like those shown in section 2.4 Energy Absorption. The constant increase in force represents the elastic region. The maximum peak is the point of failure or plastic deformation. Plastic deformation occurred, on Specimen 10 with an energy level of 13.2 ft-lbs, at a maximum load of 760 lbf. The structure plastically deforms and reduces the load until the plateau begins. The plateau section is the result of the impactor plunging through the specimen with a constant force. Densification occurs at the end of the load curve, in which the compaction of broken elements increase the localized density under the impactor tip. At this point in time the impactor does not have enough energy to cause further deformation and the recorded force rapidly increases.

Figure 4.26: Force vs. time plot for specimen 10 at 13.2 ft-lbs.
Figures 4.27-4.31 show the load curves of the BCC structures compared with the BCCAV at different energy levels. To ensure that the load curve only represents the time when the impactor is in contact with the specimen, the data are clipped when the velocity becomes zero. This is done in Matlab by applying the same length of the velocity vector to the force vector. The first three energy levels shown in Figures 4.27-4.29 produced similar results for BCC and BCCAV and showed very little differences at low energy levels. This is due to the similar topologies for the first 2 layers of diamond lattice in the BCC and BCCAV. The load-cell responses appeared to be more scattered at lower energy levels. This is caused by the rapid impulsive force. A higher sampling rate would be needed to analyze these lower energy levels. The lower energy levels do not produce a plateau section in the load curves because they are not high enough to plunge the impactor deep into the specimen. The highest tested energy level of 13.2 ft-lbs produced the smoothest results. Both the BCC and BCCAV follow the same trend in the elastic section of the curve. This elastic section is the impact response from a combination of the Kevlar facesheet, epoxy resin, and stiffness of the lattice core. Although they seem to be similar, small differences in the overall stiffness of the structures can be seen in the elastic region of the curve. It can also be seen that the BCC fails at a smaller peak load than the BCCAV because of the lower stiffness value of the BCC. The vertical struts in the BCCAV permit a higher max load prior to failure. Both load curves also reflect a plateau section. The constant load required to penetrate the layers of lattice is lower in the BCC and can be seen in Figure 4.31 which shows the result of this higher stiffness. When comparing lattice structure configurations, an energy level should be chosen where the impactor plunges to a depth over 50% of the specimens thickness. These load curves
can be plotted with respect to the previously determined displacement data to produce the load-displacement curve.

Figure 4.27: Force vs. time plot of BCC and BCCAV at an energy level of 6.3 ft-lbs.

Figure 4.28: Force vs. time plot of BCC and BCCAV at an energy level of 4.8 ft-lbs.
Figure 4.29: Force vs. time plot of BCC and BCCAV at an energy level of 6.3 ft-lbs.

Figure 4.30: Force vs. time plot of BCC and BCCAV at an energy level of 10.6 ft-lbs.
Figure 4.31: Force vs. time plot of BCC and BCCAV at an energy level of 13.2 ft-lbs.

To help compare the different energy levels and configurations, all load responses are plotted together in Figure 4.32. A 700-Hz Butterworth filter was applied to the data to smooth the curves for visual purposes only. The raw load data will be used for the absorption energy calculation in the following sections. The plot shows that the plastic failure of the specimen occurs around 2 ms, regardless of the energy level. It is obvious that the BCCAV structure can withstand a higher impact load prior to deformation. The plot also shows the maximum load for failure increasing as the energy level decreases. Once again, this is because of the reduced impact time at lower energy levels which exponentially increases the impact force. Specimens 1 and 6 do not follow this trend compared with the others, probably as a result of uncontrolled sandwich panel fabrication processes or a rare irregularity in the lattice core material caused by the 3D printer.
Figure 4.32: Load curves for specimens 1-10.

Figure 4.33 shows a plot of the load curves for the high-energy-tested LTS versus the aluminum honeycomb specimens. The BCCAV LTS exhibits a much higher load capacity than both the BCC and the honeycomb. The load capacity of the BCC appears similar to that of the honeycomb. The plateau stress for the polymer LTS is higher than that of the honeycomb. The mechanical properties of aluminum are much greater in terms of structural performance than that of ABS. If an identical LTS was manufactured from the same Aluminum alloy, then it would yield much greater results than the honeycomb. Therefore, the geometry and topology of the lattice truss structure is superior to the honeycomb when designing for load carrying capacities.
Figure 4.33: Load curve of specimens 5 and 10 (BCC and BCCAV) versus specimens 12 - 15 (aluminum honeycomb) impacted at 13.2 ft-lbs of energy.

4.5 Load-Displacement

The load-displacement curve can be used to capture the absorption energy through integration. A plot of the load on a specimen with respect to the displacement can yield material properties such as the elastic modulus, yield stress, ultimate stress, and strain rate. Figure 4.34 shows the load-displacement curves for specimens 4, 5, 9, and 10.
Figure 4.34: Load-displacement curve for specimens 1-10.

Figure 4.35: Load-Displacement curve for higher energy level specimens (4, 5, 9 and 10).
The load data captured from impacting specimen 1 and 6 was much lower than expected. This can be caused from differences in the fabrication process such as the curing temperature of the Kevlar facesheet, uniformity of the skin-core interface, and the quantity of epoxy resin for lamination to the lattice core. All other specimens appear to follow the same trend. It’s obvious that the displacement is increasing as the energy level increases. It is obvious that the BCCAV can withstand a greater load with less displacement when compared with the BCC. The BCCAV appears to have absorbed more energy, or more area under the load-displacement curve. However, the BCC has a higher displacement which lengthens the load-displacement curve, adding more absorbed energy. The lower energy levels do not allow the full development of the load-displacement curve. Although enough plastic deformation did not occur, the captured load-displacement curve can still be used to validate the absorption-energy calculation. Figure 4.35 shows the load-displacement curve for the polymer LTS impacted at 13.2 ft-lbs compared with the aluminum honeycomb.
The honeycomb had more displacement than the LTS. This higher displacement stretches the load-displacement curve and permits for more energy absorption. However, the BCCAV did not penetrate to its full potential. Therefore, the BCCAV can endure a higher-energy-level impact than the honeycomb which will allow for more energy absorption. The integration of these load-displacement curves will yield the absorption energy with respect to time.

Figure 4.36: Load-displacement curve of specimens 5 and 10 (BCC and BCCAV) versus specimens 12 - 15 (Aluminum Honeycomb) impacted at 13.2 ft-lbs of energy.
4.6 Absorption Energy

Integration of the load-displacement curve which is conducted in Matlab using the cumulative trapezoidal method is used to determine the absorption energy. This integration yields the absorption energy of the test specimen with respect to time. Figure 4.37 shows the absorption energy with respect to time for all 10 polymer lattice truss core specimens. The solid lines are the BCC structures and the dashed lines are BCCAV. A difference in absorption energy between configurations is apparent from the higher energy levels. Even though the BCC had a deeper penetration and longer impact time, the amount of absorbed energy was less than the BCCAV. As expected, the lower energy levels produced similar results between configurations because of the low plastic deformation of the core. The capture of similar absorption energies at lower impact energy levels, is indicative of the repeatability of the experiment. The captured absorption energy at low-impact-energy levels should be relatively close because of the similar lattice-core topology between configurations. The overall stiffness of the BCCAV is higher and it exhibits a slightly higher absorption-energy capability. Specimens 1 and 6 show a low absorption energy because of the irregular force history shown in the previous section. while Specimens 3 and 8 show an abnormally large absorption energy. The calculation of the absorption energy at lower energy levels yields inaccurate results because of an under-sampled impact response. Table 6 lists all of the absorption energies of the BCC compared with the BCCAV.
Figure 4.37: Absorption Energy with respect to time for Specimens 1-10 found from integration of the load-displacement curve.

Table 6: Comparison of Absorption Energies

<table>
<thead>
<tr>
<th>Unit-Cell Configuration</th>
<th>Specimen</th>
<th>Impact Energy (ft-lbs)</th>
<th>Absorption Energy (ft-lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>BCC</td>
<td>2</td>
<td>4.8</td>
<td>3.6</td>
</tr>
<tr>
<td>BCCAV</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>10.6</td>
<td>9.4</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>13.2</td>
<td>11.9</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>13.2</td>
<td>10.5</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 4.38: Absorption energies found from integration of the load-displacement curve for BCC and BCCAV.

The absorption energies of the LTS compared with the aluminum honeycomb are plotted in Figure 4.39. The location of the impact site relative to the honeycomb unit cell was uncontrollable and had large effects on the absorbed energy. However, the absorption energy of the polymer LTS was surprisingly comparable with aluminum honeycomb. Improvements in the geometry of the lattice-unit-cell could potentially lead to a polymer lattice structure that surpasses an aluminum honeycomb structure in terms of absorption energy.
4.7 Failure Mechanisms

The failure mechanisms of the lattice truss elements was observed from the CT scans which were previously shown in Figures 4.3-4.10. This non-destructive imaging process allowed for the analysis of the damaged structure at the impact site without disturbing or causing any further damage to the lattice truss elements. In addition to the CT scans, a 3D model was generated to help analyze the failures. The software for the CT scanner compiles 1400 images to generate a stereo lithograph (STL) model, which can be rotated and viewed from any direction. A few of these models are shown in Figures 4.40 - 4.44. Nodal failure outside of the impact site occurred on specimen 4. This failure is shown in the squared region in the upper left quadrant in the figure.
Delamination of the Kevlar face sheet from the lattice core at the impact site occurred on every specimen and is best seen below in Figure 4.41. The layer of hardened epoxy resin was left behind at the bottom of the impact as the Kevlar attempted to stretch back when the impactor was no longer in contact with the specimen. All the images show the densification region, where the broken elements gathered to increase the overall density of the structure. This appears at the end of the load-displacement curves when the force asymptotically increases because of the increase in stiffness. Figure 4.42 clearly shows buckling occurring in the vertical struts. Failure planes caused from buckling were observed at both the nodal connections and the center of the vertical strut. It should be noted that the fracture planes caused by buckling are parallel to the 3D printed layers of material. Failure of the 45° truss elements can also be seen in the image below. These fracture planes are perpendicular to the axis of the truss elements and shown in Figure 4.43. This perpendicularity is caused by the ductility of the material which results in a 90° shear plane during bending. This proves that the 45° truss elements failed from reaching the ultimate stress value during bending and that the failure is not related to the orientation of the lattice core during the FDM process. Similar results are seen on specimen 10 shown in Figure 4.44.

![Image of a cross-sectional view of a generated 3D STL model of specimen 4 (BCC) impacted at 10.6 ft-lbs.](image)

Figure 4.40: Cross-sectional view of the generated 3D STL model of specimen 4 (BCC) impacted at 10.6 ft-lbs.
Figure 4.41: Cross-sectional view of the generated 3D STL model of specimen 5 (BCC) impacted at 13.2 ft-lbs.

Figure 4.42: Cross-sectional view of the generated 3D STL model of specimen 9 (BCCAV) impacted at 10.6 ft-lbs.
Figure 4.43: 90° shear planes on specimen 9 (BCCAV) impacted at 10.6 ft-lbs.

Figure 4.44: Cross-sectional view of the generated 3D STL model of specimen 10 (BCCAV) impacted at 13.2 ft-lbs.
CHAPTER 5: CONCLUSION

5.1 Summary

Ten polymer-lattice-core sandwich panels and four aluminum-honeycomb-sandwich panels were fabricated and tested under low-velocity impacts. BCC and BCCAV where the two unit-cell configurations for the LTS chosen for the study at four different energy levels. Specimens 1-5 were of BCC unit cells and specimens 6-10 were BCCAV. Specimens 12-15 were aluminum honeycomb specimens. All 14 specimens were fabricated using the same methods discussed in section 3.2.4 Sandwich Panel Fabrication. Microscopic images of the fabricated sandwich panels were taken to confirm adequate bonding at the core-skin interface and also lamination of the 4-ply pre-impregnated woven Kevlar face sheets.

An impact machine was designed and fabricated following the ASTM D-7136 standard. The machine was able to capture absorption energy of the test specimens, the penetration depth, and the entrance and exit velocity of the impactor. The measurements were validated by determining the absorption energy from integrating the momentum of the impactor with respect to time and subtracting the leftover potential energy after impact from the initial value. CT scans of the damaged specimens were conducted to validated the penetration depth and show the critical failures of the truss elements.
5.2 Conclusions

Impacting at lower energies did not produce meaningful results. The quick impact time caused from a low deformation impact requires a higher sampling rate. At higher energy levels, the BCCAV absorbed more energy than the BCC. The penetration depth of the BCCAV was substantially lower than that of the BCC. Because of the alternating vertical struts increasing the overall stiffness of the lattice core. This larger stiffness was manifested in the load-displacement curves of the BCCAV structures. Which allowed for a higher load with less plastic deformation. The impact machine was able to capture small differences in lattice topology. New lattice core designs can be tested with the impact machine to capture the differences in absorption energy, load capacity, and penetration depths. The machine is also capable of capturing the differences between new lattice designs and well known energy absorbers such as aluminum honeycomb and metal foams.

The polymer lattice structures exhibited surprisingly close absorption energies to the aluminum honeycomb. While the honeycomb absorbed more energy, the polymer LTS had a smaller penetration depth. Aluminum honeycomb has been studied for years and optimized to its fullest potential. Polymer lattice truss structures are relatively new and will require years of research to reach full optimization of the structures. Due to the flexibility of AM, polymer LTS show feasibility in surpassing a metallic honeycomb.

5.3 Recommendations for Future Work

Addition of roller bearings is recommended for the sliders that guide the impactor along the two guide rails. To reduce friction between the sliders and the guide rails and eliminate noise in the accelerometer and load-cell responses. To further improve the
load-cell response, it is recommended that the load cells be relocated to the impactor. A new impactor tip can be fabricated to accommodate the mating threads of the load cells and the range of the load cell must be increased by a factor of 4.

It is also recommended that specimens be tested at higher energy levels in the future. The best results will be achieved when the specimen size is increased to 4 in x 6 in and the energy level is set to penetrate a minimum of 50% of the thickness of the lattice core. Removing the additional 2x2- fixture will allow the impactor to pierce through the specimen if the energy level is too high and not cause damage to the machine or impactor tip. The larger specimen will also help eliminate delamination at the edges.

Manufacturing the LTS in different orientations should be investigated. Improvements in strengths and stiffness of the structures could be seen if the printed layers are deposited parallel to the axis of the vertical struts. A stronger epoxy for lamination of the face sheet to the core structure should be investigated to reduce delamination at the impact site.
REFERENCES


Appendix A

Matlab Code

Matlab was used to post-process the acceleration and force histories. The following code extracts the data from preset folders. The acceleration history is integrated twice to get the displacement. The load-cell responses are summed together and clipped where the velocity equals zero. The absorption energy is calculated by integration the load-displacement curve. The load, acceleration, velocity, displacement, and absorption energy is plotted.

```matlab
% clc
% clear all
% close all
format long

%% Custom
file1 = 'Specimen10';

f_S = 50000;  % frequency of Strain Gauges(Hz)

%% Reference
items_ACC1 = {'cDAQ1Mod2_ai0'};
items_LC1 = {'cDAQ1Mod1_ai0'};
items_LC2 = {'cDAQ1Mod1_ai1'};
items_LC3 = {'cDAQ1Mod1_ai2'};
items_LC4 = {'cDAQ1Mod1_ai3'};

%% Upfront - Acceleration
TMDS Function was provided by:
Dr. Ryan Merritt at Ahmic Aerospace Solutions

filePath1 = 'Acceleration.tdms';
addpath('TMDS_function')
params = { ...'
'UTC_DIFF' -5 ...%Refer timestamps to U.S. Central Time (UTC - 6)
```
'USE_INDEX' false ... %index may be corrupted, use tdms file, not
tdms_index
'MAX_NUM_OBJECTS' 1100 ... %We have about 1000 objects, +100 for padding
'DATE_STR_FORMAT' 'dd-mmm-yyyy' ... %Only return data info for properties
%NOTE: timestamp data is returned in such a format that datestr() works
%on it

%% BREAK RUN APART
	namel = strcat('RAW',file1,'\',filePath1);
filePath = namel;
tempOutput = TDMS_readTDMSFile(filePath,params{:});
output = TDMS_dataToGroupChanStruct_v1(tempOutput);
list = fieldnames(output);
filename = char(list{2});
filename = filename(1:23);
c2 = cellstr(filename);

ACCI = output.(list{2}).(items_ACC1{1}).data;

%% Upfront - Force

filePath1 = 'Force_(IEPE).tdms';
addpath('TMDS_function')
params = {
'UTC_DIFF' -5 ...%Refer timestamps to U.S. Central Time (UTC - 6)
'USE_INDEX' false ... %index may be corrupted, use tdms file, not
tdms_index
'MAX_NUM_OBJECTS' 1100 ... %We have about 1000 objects, +100 for padding
'DATE_STR_FORMAT' 'dd-mmm-yyyy' ... %Only return data info for properties
%NOTE: timestamp data is returned in such a format that datestr() works
%on it

%% BREAK RUN APART
	namel = strcat('RAW',file1,'\',filePath1);
filePath = namel;
tempOutput = TDMS_readTDMSFile(filePath,params{:});
output = TDMS_dataToGroupChanStruct_v1(tempOutput);
list = fieldnames(output);
filename = char(list{2});
filename = filename(1:23);
c2 = cellstr(filename);

LC1 = output.(list{2}).(items_LC1{1}).data;
LC2 = output.(list{2}).(items_LC2{1}).data;
LC3 = output.(list{2}).(items_LC3{1}).data;
LC4 = output.(list{2}).(items_LC4{1}).data;

%% SOLVE
MA = 10001;

l1 = length(LC1);
t1 = 0:1/f_S:l1/f_S-1/f_S;
t2 = 0:1/f_S:l1/f_S-1/f_S;

LC_sum = LC1 + LC2 + LC3 + LC4;

%% ISOLATE Impact
acc_interval1 = 10.38096;
acc_interval2 = 10.38585;
delta_t = acc_interval2 - acc_interval1;
lc_interval1 = 10.3812;
lc_interval2 = lc_interval1 + delta_t;
a = lc_interval1*f_S;
b = lc_interval2*f_S;
c = acc_interval1*f_S;
d = acc_interval2*f_S;
t2 = t2(c:d);
t1 = t1(a:b);
lt = length(t1);
t3 = 0:1/f_S:lt/f_S-1/f_S;
LC_sum = LC_sum(a:b);
ACC1 = ACC1(c:d);

%% filter acceleration
n = 6;  
% Filter Order  
f1 = 1001;  
% low frequency  
wn = f1/(f_S/2);  
% Normalized cutoff frequency (Wn = 1 is half of sample Hz)
[b1,a1] = butter(n,wn,'low');
ACC1 = filtfilt(b1,a1,ACC1);

%% filter load
n = 2;  
% Filter Order  
f1 = 700;  
% low frequency  
wn = f1/(f_S/2);  
% Normalized cutoff frequency (Wn = 1 is half of sample Hz)
[b1,a1] = butter(n,wn,'low');
LC_sum = filtfilt(b1,a1,LC_sum);

%% CALCULATE - VELOCITY/DISPLACEMENT
%without filtering
ACC1g = ACC1*32.2;

ACC1_vel = cumtrapz(t2, ACC1g); %Adjust Velocity to Zero Axis
%% Shift Velocity
ACC1_vel = ACC1_vel-11.25;

ACC1_disp = -12*cumtrapz(t2, ACC1_vel);

%% Plot Load
figure(1)
set(gca,'FontSize',12)
hold on
h = plot(t3,LC_sum,'bl', 'LineWidth',2);
h = plot(t3,f_LC_sum,'g', 'LineWidth',2);
legend('Raw Data','700-Hz Filter')
ylabel('Force (lbf)','fontsize',14);
xlabel('Time (sec)','fontsize',14);
grid on;

%% Plot Acceleration
figure(2)
hold on
h = plot(t3,ACC1, 'bl', 'LineWidth',2);
h = plot(t3,f_ACC1, 'g', 'LineWidth',2);
ylabel('Acceleration (g)')
xlabel('Time (sec)');
grid on;

%% Plot Velocity
figure(3)
hold on
h = plot(t3,ACC1_vel, 'bl', 'LineWidth',2);
ylabel('Velocity (ft/s)')
xlabel('Time (sec)');
grid on;

%% Plot Displacement
figure(4)
hold on
h = plot(t3,ACC1_disp, 'bl', 'LineWidth',2);
ylabel('Displacement (in)')
xlabel('Time (sec)');
grid on;

%% Plot Load vs Displacement
figure(5)
hold on
h = plot(ACC1_disp,LC_sum, 'bl', 'LineWidth',2);
h = plot(ACC1_disp,f_LC_sum, 'g', 'LineWidth',2);
xlabel('Displacement (in)')
ylabel('load (lbf)','FontSize',14);
grid on;

%% Absorption Energy
A_E = cumtrapz(ACC1_disp, LC_sum)/12;
max_AE = max(A_E);

%% Plot Absorption Energy vs time
figure(6)
hold on
h = plot(t3,A_E,'b1','LineWidth',2);
xlabel('Time (sec)')
ylabel('Absorption Energy (ft-lb)',['FontSize',14]);
grid on;

To plot all of the data together a separate script file is used. The following code runs all of the script files for the individual specimens and plots all of the data. The code can also write the data to an excel file for other post-processing methods.

close all
clear all
clc
max_A_E = [];
run Test3_Specimen1

Test1_time = t3;
Test1_Acc = ACC1;
Test1_Vel = ACC1_vel;
Test1_Disp = ACC1_disp;
Test1_Load = f_LC_sum;
Test1_AE = A_E;
max_A_E = [max_A_E; max_AE];

run Test3_Specimen2

Test2_time = t3;
Test2_Acc = ACC1;
Test2_Vel = ACC1_vel;
Test2_Disp = ACC1_disp;
Test2_Load = f_LC_sum;
Test2_AE = A_E;
max_A_E = [max_A_E; max_AE];

run Test3_Specimen3

Test3_time = t3;
Test3_Acc = ACC1;
Test3_Vel = ACC1_vel;
Test3_Disp = ACC1_disp;
Test3_Load = f_LC_sum;
Test3_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen4

Test4_time = t3;
Test4_Acc = ACC1;
Test4_Vel = ACC1_vel;
Test4_Disp = ACC1_disp;
Test4_Load = f_LC_sum;
Test4_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen5

Test5_time = t3;
Test5_Acc = ACC1;
Test5_Vel = ACC1_vel;
Test5_Disp = ACC1_disp;
Test5_Load = f_LC_sum;
Test5_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen6

Test6_time = t3;
Test6_Acc = ACC1;
Test6_Vel = ACC1_vel;
Test6_Disp = ACC1_disp;
Test6_Load = f_LC_sum;
Test6_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen7

Test7_time = t3;
Test7_Acc = ACC1;
Test7_Vel = ACC1_vel;
Test7_Disp = ACC1_disp;
Test7_Load = f_LC_sum;
Test7_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen8

Test8_time = t3;
Test8_Acc = ACC1;
Test8_Vel = ACC1_vel;
Test8_Disp = ACC1_disp;
Test8_Load = f_LC_sum;
Test8_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen9
Test9_time = t3;
Test9_Acc = ACC1;
Test9_Vel = ACC1_vel;
Test9_Disp = ACC1_disp;
Test9_Load = f_LC_sum;
Test9_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen10

Test10_time = t3;
Test10_Acc = ACC1;
Test10_Vel = ACC1_vel;
Test10_Disp = ACC1_disp;
Test10_Load = f_LC_sum;
Test10_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen12

Test11_time = t3;
Test11_Acc = ACC1;
Test11_Vel = ACC1_vel;
Test11_Disp = ACC1_disp;
Test11_Load = f_LC_sum;
Test11_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen13

Test12_time = t3;
Test12_Acc = ACC1;
Test12_Vel = ACC1_vel;
Test12_Disp = ACC1_disp;
Test12_Load = f_LC_sum;
Test12_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen14

Test13_time = t3;
Test13_Acc = ACC1;
Test13_Vel = ACC1_vel;
Test13_Disp = ACC1_disp;
Test13_Load = f_LC_sum;
Test13_AE = A_E;
max_A_E = [max_A_E; max_AE];
run Test3_Specimen15

Test14_time = t3;
Test14_Acc = ACC1;
Test14_Vel = ACC1_vel;
Test14_Disp = ACC1_disp;
Test14_Load = f_LC_sum;
Test14_AE = A_E;
max_A_E = [max_A_E; max_AE];

clc

%% WRITE DATA TO EXCEL FILE
%% Specimen_Numbers = [1,2,3,4,5,6,7,8,9,10]
%% file = 'Specimen_Data'
%%
%% for n = 1:length(Specimen_Numbers)
%%    sheet = ['Specimen',num2str(Specimen_Numbers(n))];
%%    Header = {'Time','Displacement','Load','Absorption Energy'};
%%    xlswrite(file,Header,sheet)
%%
%%    %write time vector
%%    array = ['Test',num2str(n),'_time'];
%%    array = genvarname(array);
%%    array = eval(array)';
%%    xlswrite(file,array,sheet,'A2')
%%
%%    %write Displacement vector
%%    array = ['Test',num2str(n),'_Disp'];
%%    array = genvarname(array);
%%    array = eval(array)';
%%    xlswrite(file,array,sheet,'B2')
%%
%%    %write Load vector
%%    array = ['Test',num2str(n),'_Load'];
%%    array = genvarname(array);
%%    array = eval(array)';
%%    xlswrite(file,array,sheet,'C2')
%%
%%    %write Absorption Energy vector
%%    array = ['Test',num2str(n),'_AE'];
%%    array = genvarname(array);
%%    array = eval(array)';
%%    xlswrite(file,array,sheet,'D2')
%%
%% end
%%
%% clc

%% Plot Displacements
figure(1)
hold on
h = plot(Test1_time, Test1_Disp,'bl','LineWidth',2);
h = plot(Test2_time, Test2_Disp,'g','LineWidth',2);
h = plot(Test3_time, Test3_Disp,'cy','LineWidth',2);
h = plot(Test4_time, Test4_Disp,'r','LineWidth',2);
h = plot(Test5_time, Test5_Disp,'k','LineWidth',2);
h = plot(Test6_time, Test6_Disp,'bl--','LineWidth',2);
h = plot(Test7_time, Test7_Disp,'g--','LineWidth',2);
h = plot(Test8_time, Test8_Disp,'c--','LineWidth',2);
h = plot(Test9_time, Test9_Disp,'r--','LineWidth',2);
h = plot(Test10_time, Test10_Disp,'k--','LineWidth',2);
legend('Specimen 1-\text{BCC-6.3 ft-lbs}', 'Specimen 2-\text{BCC-4.8 ft-lbs}', 'Specimen 3-\text{BCC-6.3 ft-lbs}', 'Specimen 4-\text{BCC-10.6 ft-lbs}', 'Specimen 5-\text{BCC-13.2 ft-lbs}', 'Specimen 6-\text{BCCAV-6.3 ft-lbs}', 'Specimen 7-\text{BCCAV-4.8 ft-lbs}', 'Specimen 8-\text{BCCAV-6.3 ft-lbs}', 'Specimen 9-\text{BCCAV-10.6 ft-lbs}', 'Specimen 10-\text{BCCAV-13.2 ft-lbs}', 'Specimen 12-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 13-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 14-\text{Honeycomb-13.2 ft-lbs}')
ylabel('Displacement (in)')
xlabel('Time (sec)')
grid on;

%% Plot Velocities
figure(3)
hold on
h = plot(Test1_time/10^-3, Test1_Disp,'b1','LineWidth',2);
hold on
h = plot(Test2_time/10^-3, Test2_Disp,'g','LineWidth',2);
hold on
h = plot(Test3_time/10^-3, Test3_Disp,'cy','LineWidth',2);
hold on
h = plot(Test4_time/10^-3, Test4_Disp,'r','LineWidth',2);
hold on
h = plot(Test5_time/10^-3, Test5_Disp,'k','LineWidth',2);
hold on
h = plot(Test6_time/10^-3, Test6_Disp,'b1--','LineWidth',2);
hold on
h = plot(Test7_time/10^-3, Test7_Disp,'g--','LineWidth',2);
hold on
h = plot(Test8_time/10^-3, Test8_Disp,'c--','LineWidth',2);
hold on
h = plot(Test9_time/10^-3, Test9_Disp,'r--','LineWidth',2);
hold on
h = plot(Test10_time/10^-3, Test10_Disp,'k--','LineWidth',2);
hold on
h = plot(Test11_time/10^-3, Test11_Disp,'m','LineWidth',2);
hold on
h = plot(Test12_time/10^-3, Test12_Disp,'m:','LineWidth',2);
hold on
h = plot(Test13_time/10^-3, Test13_Disp,'m--','LineWidth',2);
hold on
h = plot(Test14_time/10^-3, Test14_Disp,'m--','LineWidth',2);
legend('Specimen 1-\text{BCC-6.3 ft-lbs}', 'Specimen 2-\text{BCC-4.8 ft-lbs}', 'Specimen 3-\text{BCC-6.3 ft-lbs}', 'Specimen 4-\text{BCC-10.6 ft-lbs}', 'Specimen 5-\text{BCC-13.2 ft-lbs}', 'Specimen 6-\text{BCCAV-6.3 ft-lbs}', 'Specimen 7-\text{BCCAV-4.8 ft-lbs}', 'Specimen 8-\text{BCCAV-6.3 ft-lbs}', 'Specimen 9-\text{BCCAV-10.6 ft-lbs}', 'Specimen 10-\text{BCCAV-13.2 ft-lbs}', 'Specimen 12-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 13-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 14-\text{Honeycomb-13.2 ft-lbs}')
ylabel('Displacement (in)')
xlabel('Time (sec)')
grid on;

%% Plot Velocities
figure(3)
hold on
h = plot(Test1_time, Test1_Vel,'b1','LineWidth',2);
hold on
h = plot(Test2_time, Test2_Vel,'g','LineWidth',2);
hold on
h = plot(Test3_time, Test3_Vel,'cy','LineWidth',2);
hold on
h = plot(Test4_time, Test4_Vel,'r','LineWidth',2);
hold on
h = plot(Test5_time, Test5_Vel,'k','LineWidth',2);
hold on
h = plot(Test6_time, Test6_Vel,'b1--','LineWidth',2);
hold on
h = plot(Test7_time, Test7_Vel,'g--','LineWidth',2);
hold on
h = plot(Test8_time, Test8_Vel,'c--','LineWidth',2);
hold on
h = plot(Test9_time, Test9_Vel,'r--','LineWidth',2);
hold on
h = plot(Test10_time, Test10_Vel,'k--','LineWidth',2);
hold on
h = plot(Test11_time, Test11_Vel,'m','LineWidth',2);
hold on
h = plot(Test12_time, Test12_Vel,'m:','LineWidth',2);
hold on
h = plot(Test13_time, Test13_Vel,'m--','LineWidth',2);
hold on
h = plot(Test14_time, Test14_Vel,'m--','LineWidth',2);
legend('Specimen 1-\text{BCC-6.3 ft-lbs}', 'Specimen 2-\text{BCC-4.8 ft-lbs}', 'Specimen 3-\text{BCC-6.3 ft-lbs}', 'Specimen 4-\text{BCC-10.6 ft-lbs}', 'Specimen 5-\text{BCC-13.2 ft-lbs}', 'Specimen 6-\text{BCCAV-6.3 ft-lbs}', 'Specimen 7-\text{BCCAV-4.8 ft-lbs}', 'Specimen 8-\text{BCCAV-6.3 ft-lbs}', 'Specimen 9-\text{BCCAV-10.6 ft-lbs}', 'Specimen 10-\text{BCCAV-13.2 ft-lbs}', 'Specimen 12-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 13-\text{Honeycomb-13.2 ft-lbs}', 'Specimen 14-\text{Honeycomb-13.2 ft-lbs}')
ylabel('Velocity (ft/s)')
xlabel('Time (sec)');
grid on;

%% Plot Loads
figure(4)
set(gca, 'FontSize', 12)
hold on
h = plot(Test1_time/10^3, Test1_Load, 'b', 'LineWidth', 2);
% h = plot(Test2_time/10^3, Test2_Load, 'g', 'LineWidth', 2);
% h = plot(Test3_time/10^3, Test3_Load, 'cy', 'LineWidth', 2);
% h = plot(Test4_time/10^3, Test4_Load, 'r', 'LineWidth', 2);
h = plot(Test5_time/10^3, Test5_Load, 'k', 'LineWidth', 2);
% h = plot(Test6_time/10^3, Test6_Load, 'b--', 'LineWidth', 2);
% h = plot(Test7_time/10^3, Test7_Load, 'g--', 'LineWidth', 2);
% h = plot(Test8_time/10^3, Test8_Load, 'c--', 'LineWidth', 2);
% h = plot(Test9_time/10^3, Test9_Load, 'r--', 'LineWidth', 2);
h = plot(Test10_time/10^3, Test10_Load, 'k', 'LineWidth', 2);
% h = plot(Test11_time/10^3, Test11_Load, 'm', 'LineWidth', 2);
% h = plot(Test12_time/10^3, Test12_Load, 'm:', 'LineWidth', 2);
% h = plot(Test13_time/10^3, Test13_Load, 'm--', 'LineWidth', 2);
h = plot(Test14_time/10^3, Test14_Load, 'm--', 'LineWidth', 2);

ylabel('Force (lbf)', 'fontsize', 14)
xlabel('Time (msec)', 'fontsize', 14);
grid on;

%% Plot Accelerations
figure(5)
hold on
h = plot(Test1_time, Test1_Acc, 'b', 'LineWidth', 2);
% h = plot(Test2_time, Test2_Acc, 'g', 'LineWidth', 2);
% h = plot(Test3_time, Test3_Acc, 'cy', 'LineWidth', 2);
% h = plot(Test4_time, Test4_Acc, 'r', 'LineWidth', 2);
h = plot(Test5_time, Test5_Acc, 'k', 'LineWidth', 2);
% h = plot(Test6_time, Test6_Acc, 'b--', 'LineWidth', 2);
% h = plot(Test7_time, Test7_Acc, 'g--', 'LineWidth', 2);
% h = plot(Test8_time, Test8_Acc, 'c--', 'LineWidth', 2);
% h = plot(Test9_time, Test9_Acc, 'r--', 'LineWidth', 2);
h = plot(Test10_time, Test10_Acc, 'k', 'LineWidth', 2);
legend('Specimen 1', 'Specimen 2', 'Specimen 3', 'Specimen 4', 'Specimen 5', 'Specimen 6', 'Specimen 7', 'Specimen 8', 'Specimen 9', 'Specimen 10')

ylabel('Acceleration (g-force)')
xlabel('Time (sec)');
grid on;
%% Plot Load vs. Displacements
figure(6)
set(gca,'FontSize',12)
hold on
h = plot(Test1_Disp, Test1_Load,'b-','LineWidth',2);
h = plot(Test2_Disp, Test2_Load,'g-','LineWidth',2);
h = plot(Test3_Disp, Test3_Load,'c-','LineWidth',2);
h = plot(Test4_Disp, Test4_Load,'r-','LineWidth',2);
h = plot(Test5_Disp, Test5_Load,'k-','LineWidth',2);
h = plot(Test6_Disp, Test6_Load,'b--','LineWidth',2);
h = plot(Test7_Disp, Test7_Load,'g--','LineWidth',2);
h = plot(Test8_Disp, Test8_Load,'c--','LineWidth',2);
h = plot(Test9_Disp, Test9_Load,'r--','LineWidth',2);
h = plot(Test10_Disp, Test10_Load,'k--','LineWidth',2);
h = plot(Test11_Disp, Test11_Load,'m-','LineWidth',2);
h = plot(Test12_Disp, Test12_Load,'m-','LineWidth',2);
h = plot(Test13_Disp, Test13_Load,'m-','LineWidth',2);
h = plot(Test14_Disp, Test14_Load,'m-','LineWidth',2);


ylabel('Force (lbf)','fontsize',14);
xlabel('Displacement (in)','fontsize',14);
grid on;

%% Plot Absorption Energies
figure(7)
set(gca,'FontSize',12)
hold on

h = plot(Test1_time/10^3, Test1_AE,'b-','LineWidth',2);
h = plot(Test2_time/10^3, Test2_AE,'g-','LineWidth',2);
h = plot(Test3_time/10^3, Test3_AE,'c-','LineWidth',2);
h = plot(Test4_time/10^3, Test4_AE,'r-','LineWidth',2);
h = plot(Test5_time/10^3, Test5_AE,'k-','LineWidth',2);
h = plot(Test6_time/10^3, Test6_AE,'b--','LineWidth',2);
h = plot(Test7_time/10^3, Test7_AE,'g--','LineWidth',2);
h = plot(Test8_time/10^3, Test8_AE,'c--','LineWidth',2);
h = plot(Test9_time/10^3, Test9_AE,'r--','LineWidth',2);
h = plot(Test10_time/10^3, Test10_AE,'k--','LineWidth',2);
h = plot(Test11_time/10^3, Test11_AE,'m-','LineWidth',2);
h = plot(Test12_time/10^3, Test12_AE,'m-','LineWidth',2);
h = plot(Test13_time/10^3, Test13_AE,'m-','LineWidth',2);
h = plot(Test14_time/10^3, Test14_AE,'m-','LineWidth',2);

legend('Specimen 1-BCC-6.3 ft-lbs','Specimen 2-BCC-4.8 ft-lbs','Specimen 3-BCC-6.3 ft-lbs','Specimen 4-BCC-10.6 ft-lbs','Specimen 5-BCC-13.2 ft-lbs','Specimen 6-BCCAV-6.3 ft-lbs','Specimen 7-BCCAV-4.8 ft-lbs','Specimen 8-BCCAV-6.3 ft-lbs','Specimen 9-BCCAV-10.6 ft-lbs','Specimen 10-BCCAV-13.2 ft-lbs')
ylabel('Absorption Energy (ft-lbs)', 'fontsize', 14)
xlabel('Time (msec)', 'fontsize', 14);
grid on;

max_A_E
FINISH: RAW

MATERIAL: LOW CARBON STEEL

TOTAL QTY: 4 EA.

TOTAL QTY: 4 EA.

TOGGLE CLAMP RISER

4X Ø 0.325 THRU

4X 0.25

4X 0.25

1.37

1.00

1.50