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FSAE Impact Attenuator



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Abstract

WPI's FSAE racecar requires an impact attenuator that can protect the driver and frame when mounted on the front bulkhead of the vehicle. This project included the design, modeling, fabrication and testing of two impact attenuator design. The result was the creation of a single crash protection structure which adequately met all of SAE's rules and regulations for competition.

Executive Summary

Each year WPI's FSAE (Formula Society of Automotive Engineers) racecar team builds a car and competes against many different schools. A winning car is based on a number of different judging criteria. The purpose of this report is to focus on a very important aspect of the vehicle, the impact attenuator. The rules of competition explain that each car must be equipped with a crash protection device, and it must absorb enough energy so that the driver can walk away without sustaining serious injury and damage to the frame is minimized. In order for the car to compete in the road race it must pass all regulations.

After completing extensive research on previous designs and physics of a collision, preliminary design concepts were formulated. These designs were then compared and contrasted against each other based on cost, safety, reliability, feasibility and weight. The honeycomb and impact foam concepts were the designs with the overall highest scores. The two designs developed for modeling and testing were the Honeycomb Pyramid and the Foam-Honeycomb-Foam design. These were further analyzed and designed using solid modeling software.

Finite element analysis was conducted to get an idea of how the attenuators would deform under the high stresses of a collision. The complex honeycomb structure proved problematic during the analysis. The program was unable to mesh the pieces of the honeycomb pattern and complete a full test. A solution for this was to model each layer as a solid block but giving it the same properties. Due to the magnitude of force during impact and the non-linear deformation of honeycomb the program we used was unable to produce accurate and reliable results on our models. However the team did get a depiction of how the model would crush based on the test results.

The next step was to manufacture the two designs and prepare them for physical testing. The method of testing chosen was the drop test. A weight of 732 pounds was raised 7.5 feet into

the air and released onto each prototype design. This height and weight were chosen to meet the requirement of the car reaching a velocity of 7 m/s at the point of impact. During the test a high speed camera recorded the test in slow motion so that the deformation patterns of each design could be analyzed. Also in the test two accelerometers were used to measure the deceleration during impact of the weight. This would be equivalent to the forces the driver would be enduring during a crash. The rules state that this deceleration cannot have a peak of more than 40 g's with and an average deceleration of less than 20 g's.

The results showed that both of our design met the criteria for energy absorption during the physical crash testing. The Honeycomb Pyramid design had a peak deceleration of 26.38 g's and an average of 14.73 g's. The Foam-Honeycomb-Foam design had a peak deceleration of 22.08 g's and an average of 11.8 g's. Both designs were well below the maximum deceleration criteria as stated by SAE rules.

The test data shows that the Foam-Honeycomb-Foam design is best in this application. It produced both a lower peak and average deceleration over the period of impact. Also a new clarification of the dimension rule was published after this project was well underway which ultimately rules out our pyramid design. The foam honey comb design meets all FSAE rules and regulations for competition.

Although our pyramid design does meet all specifications, there are a multitude of things that could be looked into further to create an optimal design. During our tests, neither designs crushed all of the way meaning the honeycomb used was too stiff. Exploring the options of different cell sizes and densities could allow for more deformation and energy absorption. In regards to finite element analysis, a different program such as Abaqus should be used or possibly even some kind of impact simulation software that is on the market.

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Introduction

WPI's FSAE race team builds and races vehicles every year for the SAE Competition in the spring. The competition promotes student teamwork and problem solving for automotive engineering students. Furthermore, it encourages innovations and developments to further success in the tournament.

One of the main requirements for the competition is that the team designs, tests, and builds an impact attenuator for the front of the car. The attenuator's purpose is to absorb impact energy, thus minimizing damage to the car frame and the driver. A SAE rule requires the attenuator to be mounted to the front bulkhead and provides an average deceleration of no more than 20 g's during an impact with the peak deceleration not exceeding 40 g's at any point during the impact. The conditions needed to be replicated for impact testing are a vehicle traveling at 7 m/s (23 ft/s) with a total mass of 300 kg (661 lbs.).

Proof of the practical testing must be provided by the design team. Photos of the impact attenuator before and after impact must be provided to FSAE prior to the competition for inspection. The team is also required to show a schematic drawing of the test method. During the technical inspection of the vehicle, the tested impact attenuator must be shown to the official and compared to photographs against the one fitted to the prototype (2010 FSAE Rules, pg 24).

Our team's main goal was to design an impact attenuator that will meet and surpass the deceleration and size requirements set forth by FSAE rules. Our first objective was to design a set of impact attenuators. The second objective was to put these designs into Finite Element Testing on SolidWorks. The attenuators that showed the most promise were used in a drop test to measure the peak and average deceleration during an impact.

Background Information

Prior to forming any preliminary designs, the group conducted extensive research focusing on the FSAE program as well as any previous attenuator designs. This was intended to give the group a concrete background for SAE vehicle impact attenuator design. Also, various testing and analysis procedures were reviewed to find the best method of testing for the designs. The acquisition of this literary material guided the group toward an ample background in designs, equations, and testing necessary for vehicle impact attenuation.

FSAE

FSAE is a student design competition created and organized by the Society of Automotive Engineers created in 1978 (SAE International, 2009). The competition is based on a hypothetical situation in which the students are asked to design a small autocross prototype formula racecar for a manufacturing company. There are



Figure 1: Prototype Design by UAlberta 2009
(<http://www.uofaweb.ualberta.ca/ece/images/FSAE.jpg>)

relatively few restrictions on the students; other than they must compete in the various scoring competitions explained by the FSAE 2010 Rules. This competition set up allows the students to think creatively and use real world problem solving techniques to conquer the challenge (2010 FSAE Rules, pg 5). A typical example of a FSAE prototype is shown in Figure 1.

The completed vehicle should perform at a very high level and be able to complete all competition events. At the end of competition, the students will attempt to sell their design to the

fictional manufacturing corporation. The cars at the competition will be judged against each other and a winner will be chosen that “best meets the FSAE vehicle design goals and which can be profitably marketed” (2010 FSAE Rules, pg 5). The cars are judged on performance both statically and dynamically. The available points for each category are presented below in Table 1.

Table 1: FSAE Judging Categories and Points Possible

Event	Points Possible
Static Events	
Presentation	75
Engineering Design	100
Cost Analysis	100
Dynamic Events	
Acceleration	75
Skid-Pad	50
Autocross	150
Fuel Economy	100
Endurance	300
Total Points	1000

Impact Attenuator

An impact attenuator is a structure used to “decelerate impacting vehicles gradually to a stop”¹. By gradually decelerating the racecar, the frame and driver are protected from significant deformation and injury. The bulk of impact energy is transferred into the deformation of the impact attenuator structure. Attenuators can be placed on vehicles or on road barriers to absorb large impacts to protect frames and people. FSAE specifies that each car in operation must have an attenuator that meets specifications and testing criteria.

Design Tasks/ Specs

There are many specific rules and regulations pertaining to all aspects of the vehicle, but this section provides information pertaining to the Impact Attenuator. The following information is referenced from section B3.20 of the 2010 FSAE Rules.

- The Impact Attenuator must be mounted forward of the Front Bulkhead. The surface of the attenuator must be over 200mm long(fore/aft of the frame), 100mm high, and 200mm wide. This will allow the Impact Attenuator to be a minimum distance of 200mm from the Front Bulkhead. An impact shall not cause the Impact Attenuator to penetrate the Front Bulkhead. It should be “mounted directly to the Front Bulkhead and not be part of non-structural bodywork“(2010 FSAE Rules, pg 24).
- The Impact Attenuator also must have 1.5 mm solid steel or 4.0mm aluminum Anti-Intrusion Plate built into the system. The Anti-Intrusion Plate may be bolted to the Front Bulkhead, but must be the same dimensions of the Front Bulkhead’s outer features. If the plate is bolted, it must use a minimum of four 8mm Grade 8.8 bolts. If the plate is welded

¹ Getting Impact from Your Impact Attenuator, Jeff Smith, Lexis Nexis

onto the Front Bulkhead, it must extend at least to the centerline of the Bulkhead tubing. (2010 FSAE Rules, pg 24).

The Impact Attenuator must adhere to the following data requirements, pertaining to section B3.21 of the 2010 FSAE Rules.

- The competition team is required to submit data displaying that the Impact Attenuator, when mounted on the Front Bulkhead, would give an average vehicle deceleration of less than 20g's while hitting a non-yielding surface. The data requires the vehicle is traveling at 7 m/s during the impact with a total mass of 300 kg. The peak deceleration during the impact must be under 40g's.
- Proof of the practical testing must be provided by the design team. Photos of the Impact Attenuator before and after impact must be provided to FSAE. The team also is obligated to show a schematic drawing of the test method. During the technical inspection of the vehicle, the tested Impact Attenuator must be shown to the official and compared to photographs against the one fitted to the prototype (2010 FSAE Rules, pg 24).

Teams are required to submit testing data, calculations, photos, and other materials. in a PDF file to the address provided by the Action Deadlines. Late submissions will be penalized 10 points per day to a maximum 50 points.

Preliminary Calculations

Initial Conditions:

$$V_{\text{impact}} = \frac{7m}{s}$$

$$V_{\text{Final}} = 0m/s$$

$$G = 9.8m/s^2$$

$$M = 300kg$$

$$A_c = 20 * G = 196m/s^2$$

Kinetic Energy:

$$K_e = \frac{1}{2} * M * (V_{\text{impact}})^2 = 7.35 * 10^3 (kg * m^2/s^2)$$

$$= 7350 \text{ J}$$

By Conservation of Energy, Kinetic Energy is equal to potential energy

$$K_e = P_e$$

Calculating the Desired Drop Height:

$$P_e = m * g * H_d$$

$$H_d = \frac{P_e}{(M * G)} = \frac{7350}{M * G}$$

$$H_d = 2.5m = 8.2 ft$$

Time of Impact:

$$t = V_{\text{impact}} / A_c$$

$$t = .036s$$

Impulse and Force:

$$I_m = M(V_{\text{impact}} - V_{\text{final}})$$

$$I_m = 6.3 * 10^5 (\text{kg})/\text{s}^2$$

$$F = \frac{I_m}{t}$$

$$F = 58,800N$$

Previous Designs

In order to get an accurate idea of what exactly is to be created; research was done to see what cars have used in the past. Many different designs have been used in the past. The goals of this project include utilizing the concepts of ones used before and making improvements on them or formulating an entirely new design.

The 2007 Western FSAE car used a rectangular prism made of aluminum honeycomb for their impact attenuator. The total crush strength of this design was found to be 2.65 MPa. Their design reduced the overall weight of the attenuator by about 476%, which is a dramatic improvement from the year before. An interesting fact that can be taken from this project is their concern with the initial force jump on the attenuator. They resolved this problem by pre-crushing the honeycomb to have a much smoother transfer of energy during the impact. The prism was attached to the frame by a carbon fiber plate, which has a significantly lower mass than steel would. Their method of attachment, Plexus, also attributed to the weight loss of the attenuator. Plexus is adhesive for structural base applications. The team at the University of New Hampshire also came up with a similar design that met the criteria for FSAE impact attenuation. Last year's Worcester Polytechnic Institute FSAE car used aluminum honeycomb as well. Carbon fiber honeycomb was sought as an ideal material for the attenuator but because of cost and time requirements the idea could not be examined further.

The team at San Jose State University used a steel truncated pyramid for their crash protection. This design included a hollow interior within the impact attenuator, effectively reducing the weight. The strength of the steel met the requirements necessary for the 2008 FSAE rules. The design also included strategically placed holes in the steel pyramid. The reduction of the material decreases the overall attenuator mass without sacrificing the strength of it.

Students of the Kanazawa University in Japan aimed to create a lighter and less expensive impact attenuator than used in years before. They came up with two main ideas in which they conducted further research. The first idea was a steel frame structure with different levels that would crush onto itself. This idea was ruled out during calculations because of the length needed to adequately absorb the required energy would have a negative effect on the car's turning ability. This is due to an increase in moment due to the added overall length of the car. The second was a carbon fiber reinforced plastic monocoque structure. A monocoque is a single shell that uses an external "skin" to absorb all of the load rather than using some type of internal frame. It is a type of unibody design, where the body of the vehicle is integrated with the chassis. This stressed skin design was first used in the field of aviation to dramatically decrease the overall weight of aircrafts. This design is most interesting because of the unique idea. Most projects tend to use some type of truss system or support network that is located underneath the nose cone of the racecar. The monocoque essentially is the nose cone of the car, but it's also used as crash protection for the driver. The monocoque reduced the weight of the attenuator by about 22%. The only major downfall that can be seen in the project is the increase in cost that comes with using carbon fiber reinforced with plastic. It is about a 40% increase in cost from prior year's designs.

The Oxford Brookes University received an innovative design award for their impact attenuator, which was made from balsa wood. No exact details could be found regarding their design but it had to have met the standards set forth by the FSAE in order to participate in competition.

Physical Testing Techniques

Testing is an integral part in producing an effective impact attenuator for racing use. Preliminary calculations and finite element analysis give the engineer a good estimate of how a model will act in the real world. This is only an estimate though, as there are many other variables that can come into play. The reason physical testing must be completed is due to these variables. The physical testing creates a realistic situation that each design will encounter. In the following sections we will explore different possibilities of physical testing techniques.

Deceleration

During testing, an accelerometer must be attached to the rigid body decelerating. The average deceleration needs to be nearly 20 g's, with the maximum deceleration under 40 g's. Since only one physical test will be performed for each attenuator design, the accelerometer test must be successful for each sample. A possible solution to this problem is to perform multiple accelerometer tests each time. A LabVIEW program was created to record the data points from the accelerometers.

High Speed Camera

During impact, a high speed camera records the amount of time it takes for the impact to occur. This time value can be paired with the displacement value to calculate the deceleration value. This calculation is important because FSAE sets limits regarding deceleration. The high speed camera also gives a visual documentation of deformation of the sample.

Weighted Drop

A weighted drop is one method of testing the impact attenuator's effectiveness. This type of testing is favorable because the environment and parameters are repeatable. In this test, the

impact attenuator is placed below a rigid body of a certain weight that is raised to a specific height. The rigid body test needs meet the required speed and weight, and can consist of anything from concrete to weights. This body also must have little horizontal motion during the drop period. Previous FSAE teams have dropped the rigid body from a height of around 8 feet to simulate an impact at the necessary speed.

The positioning of the body will affect the angle and orientation at impact and therefore affect the test results. It is important to make sure the body remains in a vertical position and moves in one axis.

Hydraulic Testing

Hydraulic testing provides force and displacement values to help us understand the strength of the designs being used. These tests are used to see how the material responds to forces and will not be used as the final impact test. The sample to be tested is attached to a surface so that any movement is prevented. A slow moving surface contacts the impact attenuator, which is sandwiched between another surface and collapses while the displacement data is recorded. Peaks in the data generally show where the attenuator begins to collapse and where it completes the collapse.

Rolling Cart Test

The rolling cart test is a realistic scenario because it would best represent the racecar impacting a surface. The sample would be mounted to the front of a cart, and the cart would be accelerated to a given speed and crashed into a rigid surface. This test method would involve the design of the cart. The cart needs to also be accelerated whether by a pushing/pulling force on a horizontal track or by gravity down an incline. A challenge of this type of experiment is getting a repeatable impact speed and angle. To get the same angle, a track or means for keeping the

cart straight needs to be used. Gravity is a good way to get the same speed for multiple tests, but a horizontal track with a push/pull force will enable the group to explore multiple venues.

Swinging Pendulum Test

A swinging pendulum test has traditionally been used to determine the amount of energy absorbed during the fracture of a test specimen. It consists of a rigid body attached to an arm swinging from a fixed pivot point. This testing has been often used to determine resistance to impact for plastic materials. A downside to this test is that test conditions and geometry of the material play a large role in the success of the experiment. It may not be feasible for the group to obtain a pendulum large enough for a proper test.

Design Analysis

Design Selection

With an understanding of the 2010 FSAE rules and the requirements of the impact attenuator, a brainstorming session produced 5 basic concepts that could meet these needs. Each of these design concepts was put into a design matrix (Table 2) in alphabetical order so that they could be evaluated. The factors that these were measured on were cost, weight, reliability, safety, and feasibility. Weight is an important component of the entire car design as the overall design is meant to keep the car lightweight to not take away from its speed. Cost was evaluated because as with any engineering project, budget constraints cannot be over looked. Reliability of the design concept was another important factor in the design selection as this is essential to how the design will perform. Safety also needed to be assessed as the attenuator will be taking high impact forces and should pose no threat to the driver. Feasibility was weighted heavily as the team needed to evaluate how likely the concept could be implemented into the attenuator while keeping FSAE restrictions in mind.

Table 2: Design Concept Idea Chart

Design Concept	Cost	Weight	Reliability	Safety	Feasibility	Rank
Weighting factor	0.35	0.25	0.20	0.15	0.30	
Airbag	1.0	9.0	8.0	6.0	3.0	6.0
Crimped Metal Lattice	6.0	6.0	5.0	5.0	6.0	7.2
Foam	7.0	8.0	7.0	8.0	8.0	9.5
Honeycomb	8.0	8.0	9.0	8.0	8.0	10.2
Rubber Bumper	7.0	7.0	6.0	7.0	7.0	8.6

Airbag Design

The first design in the decision matrix is a design that utilizes an airbag held within the nose of the racecar that will activate on impact. This concept scored well in weight, reliability, and safety, but was ranked the lowest in terms of cost. Although utilizing an airbag would be lightweight and reliable, the cost could not be overlooked. It would cost a good deal of money to obtain the necessary parts while part replacement would be expensive. This idea was rejected after doing further detailed analysis of the pressure requirements of the impact and comparing these to the limitations found in common airbags.

Crimped Metal Lattice Design

Next, a crimped metal lattice design was investigated. This concept involved separating several rows of metal plates with crimped metal that would absorb much of the force by crushing upon impact. The lattice design scored fairly well in almost every category without performing extraordinarily well or poorly in any one area. A main concern of the design was the ability to be rebuilt after impact due to the intensive time involvement in fabrication.

High Impact Foam Design

Impact absorbing foam was then evaluated within the design matrix. This scored the second highest of the designs by performing well in each category. Compared to other materials, foam provides a lightweight and cost friendly material for impact absorption. The foam was evaluated as being one of the two safest concepts and the ability to easily manipulate it in any size or shape necessary made the use of the foam very feasible. As the use of foam scored highly it was to be looked into in much more detail later in the design process.

Honeycomb Design

Furthermore, the use of a honeycomb structure was evaluated within the matrix. This scored the highest of all of the designs. Its light weight and low price make the material appealing in that it has the ability to be created relatively cheap and also easily replaced. Honeycomb also scored high in safety and reliability as its ability to reduce impact will not vary a great deal due to design and construction variance. This design was looked into in much more detail further into the design process.

Rubber Bumper Design

A rubber bumper was then evaluated as a possible design solution. Rubber scored well in terms of weight, cost, and feasibility as it is a common material that can be made to fit into the proper dimensions. Problems were foreseen with this material because its elasticity is small compared to the other designs. Thus, it would transfer too great an amount of force to the body of the car rather than absorbing it on its own.

Final Selections

After careful consideration of all previously mentioned concept designs, the group decided to look further into two of the preliminary models. The two models being the honeycomb and foam ideas. Both of the selections scored the highest overall on our decision matrix chart. Each has great energy absorbing properties while being cost effective and lightweight. These concepts could be manipulated into many shapes and combinations in order to fit the requirements designated by FSAE regulations. The geometry of the attenuator is also very important in the energy analysis of each design. With these concepts, the group can tweak and adjust the attenuator until it meets the design goals. Before final designs could be created,

materials properties were researched to provide a better understanding on how the attenuator should be modeled.

Materials

While researching the many types of honeycomb products available, the group discovered a company called Plascore. They are a global manufacturer of honeycomb core products. Plascore provides a range of products for many uses and their honeycomb products are ideal for energy absorption. The group was fortunate that Plascore chose to sponsor our MQP and provided us with PCGA-XR1 3003 Aluminum Honeycomb. The team felt this type of honeycomb was the best choice for our application. It is constructed from 3003 aluminum alloy foil which makes it very lightweight for its size and strength. In racing, the lighter the material is, the faster the car becomes. Using a light material could ultimately shave seconds off a lap time. The shape of the honeycomb cells is what gives this material its strength. The material, 3003 aluminum foil, is not very strong in itself. A flat sheet of it can be easily bent and twisted by the average person. When arranged in a honeycomb lattice, the material's strength properties vastly increase due to the support of the hexagonal cells. This strength provides the stopping power of the honeycomb structure. The lattice resists a good amount of the impact forces that are applied to it. The honeycomb possesses ideal energy absorption because it is made from foil. Once the yield strength is breached, the ductility of the foil comes into play in absorbing the rest of the impact energy. The honeycomb itself comes in multiple forms but the team chose the expanded sheets with cell sizes of $\frac{1}{4}$ "². We chose to have a 2" pre-crush for one of our models to further reduce spikes at impact. The mechanical properties of PCGA-XR1 3003 Aluminum Honeycomb are located below in Table 3

² "PCGA-XR1 3003 Aluminum Honeycomb" Plasacore Incorporated, <http://www.plascore.com/aluminum-honeycomb-pcga-xr-3003.htm>, 2/18/1

Table 3: Honeycomb Mechanical Properties³

Core Type	Cell Size	Density	Strength PSI	Modulus KSI	Strength PSI		Modulus KSI	
					L	W	L	W
PCGA-XR1	1/4"	5.2	620	148	345	215	63	31

The next material we chose was closed cell polyethylene impact foam. This material is known for its shock absorption and minimizing vibrations. It can be commonly seen in packing applications to prevent damages while in transit. It is also found in a number of flotation devices. Closed cell means that all spaces in-between are occupied by the material, making it firm and resilient to impacts. Due to this structure it has a higher compressive strength and in turn a greater density. Open cell foam leaves a space for air bubbles which makes the material softer, susceptible to liquids, but is ideal for small forces. In the case of our attenuator, the closed cell foam is best suited for our application. The manufactures' properties of the foam can be found in Table 4.

Table 3: Polyethylene Foam Properties⁴

Property	Cell Count(cells/inch)	Density (lbs/ft ³)	Strength PSI	Compression Creep (%)
2.2 LB	24	2.2	14	148

³ "PCGA-XR1 3003 Aluminum Honeycomb" Plascore Incorporated, 2/1/08, http://www.plascore.com/pdf/Plascore_3003.pdf, 2/18/10

⁴ "Product General and Technical Information" The Foam Factory, <http://www.closedcellfoams.com/tech/PolyethyleneTech.html> 2/15/10

FSAE regulations indicated that there must be a solid plate at the base of the attenuator to prevent any penetration into the driver's compartment. This plate provides a last resort safety barrier for the operator of the car. In the event that the attenuator fails, the force will transfer to the frame of the vehicle instead of into the driver's legs. To meet these requirements, our group chose 4130 annealed steel with properties seen in Table 5.

Table 4: 4130 Annealed Steel Properties

Property	Value	Units
Elastic Modulus	205,000	N/mm ²
Poisson's Ratio	.285	
Shear Modulus	80,000	N/mm ²
Density	.00785	g/mm ³
Tensile Strength	745	N/mm ²
Yield Strength	470	N/mm ²

Solid Modeling

Based on the strength of the materials, dimensional models were developed into various combinations of materials and size. The amount of materials we needed was determined from the solid models. The team created solid models of the two main designs, the Honeycomb Pyramid and Honeycomb-Foam design. Each model was then tested using the FEA within SolidWorks.

Honeycomb Pyramid Solid Model

The Honeycomb Pyramid was first modeled by creating a solid steel back plate. According to the 2010 FSAE rules, the minimum thickness of the piece is 1.5mm or .060 inches of steel. The group chose to use a .063 inch thick piece of 4130 annealed steel. The piece was first modeled at 300mm x 330mm x 1.6mm. 5/16th inch, or approx 8 mm, bolt holes were placed according to the framing at the front of the racecar, so the impact attenuator could be attached properly. Each hole is positioned 35mm from the 300mm side and 32.5mm from the 330mm side. The bolt holes are specified according to the 2010 FSAE Rules. The structure is seen below

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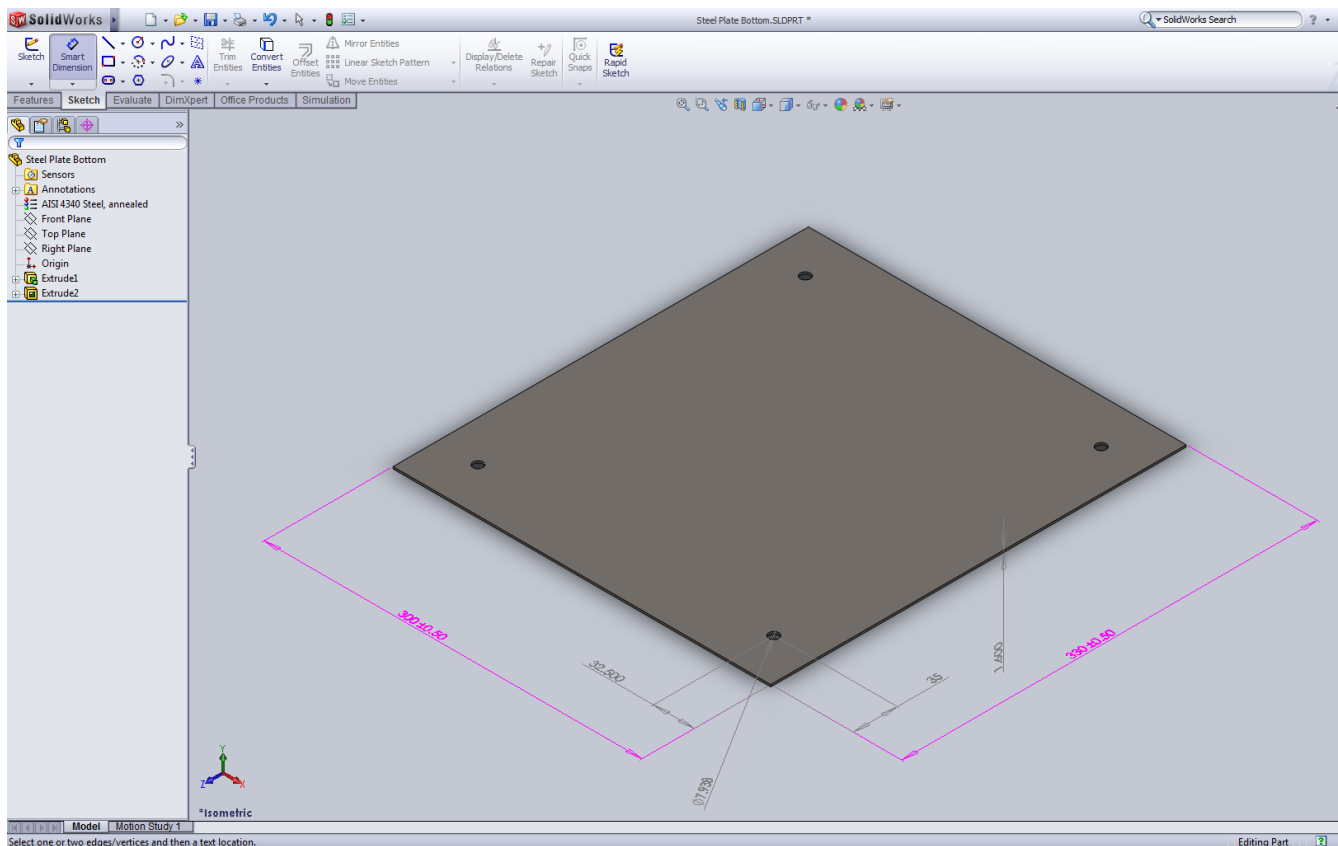


Figure 2: Solid Model of the Steel Back plate

The next step in the creation of the model was the construction of the bottom honeycomb layer. This model was created by first extruding a solid block that was sized to 9 in x 9 in x 3 in, which is 228.6 mm x 228.6 mm x 76.2 mm. A linear pattern was then created on the top of the block in the form of hexagons. The pattern was extruded cut into the block to create a honeycomb pattern. The material used is 3003 Aluminum. A similar method was used for the middle and top honeycomb pieces. The middle piece is 6 in x 6 in x 3 in, while the top piece measures 3 in x 3 in x 3 in. The honeycomb piece used for the bottom layer is shown in Figure 3.

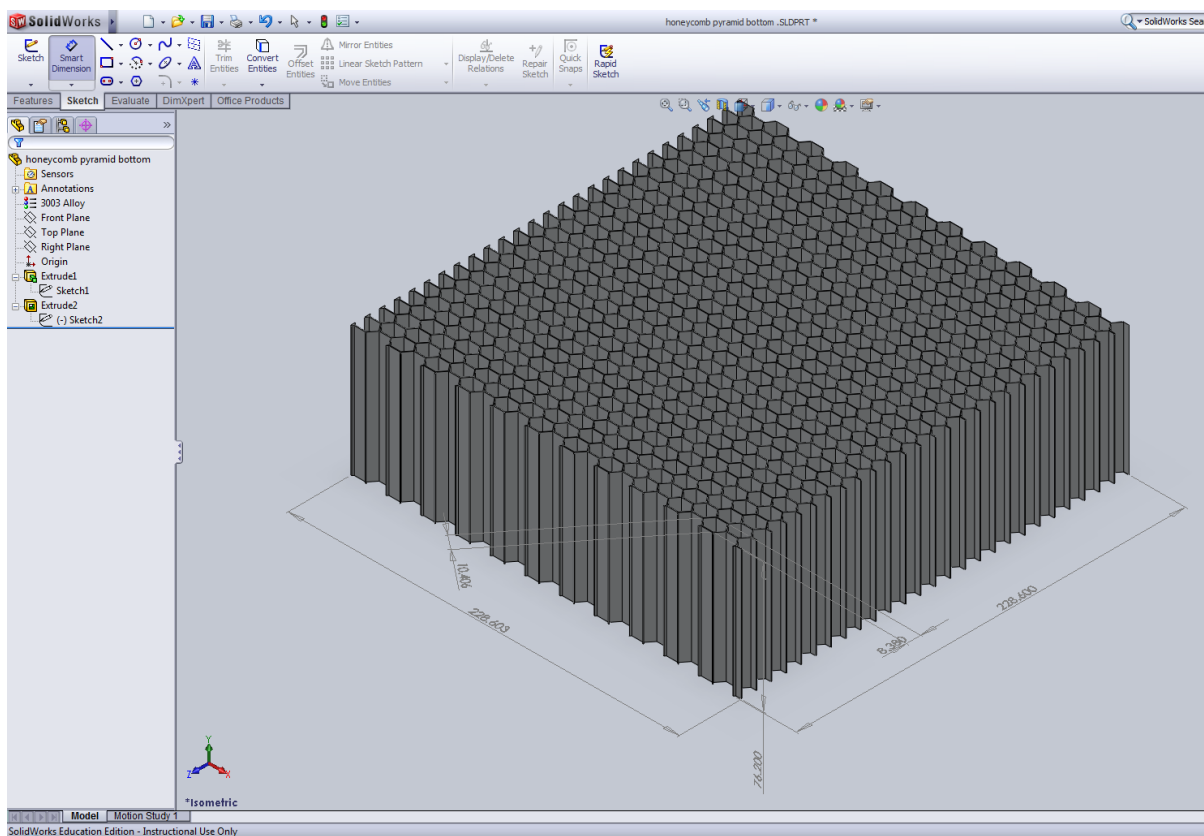


Figure 3: Honeycomb Pyramid Bottom Honeycomb Piece

Lastly, the in-between aluminum plates were modeled. The plates were placed between each of the honeycomb layers to evenly distribute the force to the next level. Each plate was 1/32nd of an inch thick. The plate placed on the bottom honeycomb was 8 in x 8 in (203.2 mm x 203.2 mm) while the plate placed on top of the second layer measured 6 in x 6 in (152.4mm x

152.4 mm). Every plate was made of 3003 Aluminum. The in-between plate used for the bottom level is modeled below in Figure 4.

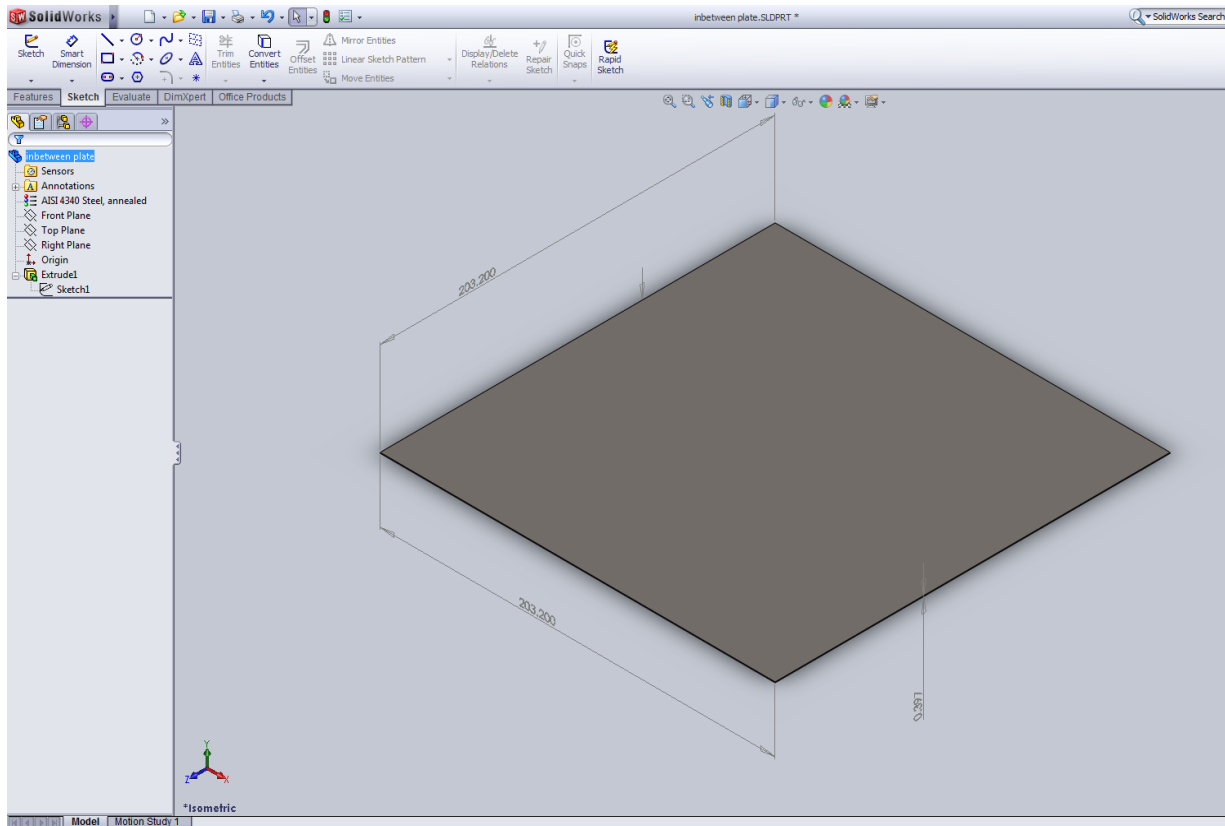


Figure 4: In-Between Aluminum Plate Model Honeycomb Pyramid

An assembly was then created using the individual components that were constructed above. The first step was to insert the bottom steel plate and the honeycomb bottom components into the assembly screen. The bottom surface of the honeycomb was lock mated and centered to the top surface of the steel plate. The bottom face of the 8 in x 8 in x 1/32nd inch aluminum plate was then mated and centered to the top surface of the honeycomb. The middle layer of honeycomb was then inserted into the drawing and had its bottom surface lock mated to the top surface of the in-between plate. The middle layer in-between plate was then inserted into the assembly and coincidentally mated to two sides of the middle honeycomb so that it was perfectly covering all parts of the middle honeycomb structure. Finally, the top honeycomb piece was

inserted into the drawing. The bottom surface of the top layer honeycomb was lock mated to the top surface of the middle in-between plate. The honeycomb was centered on the plate, and the assembly completed. A screenshot of the assembly and the mates is show below in Figure 5.

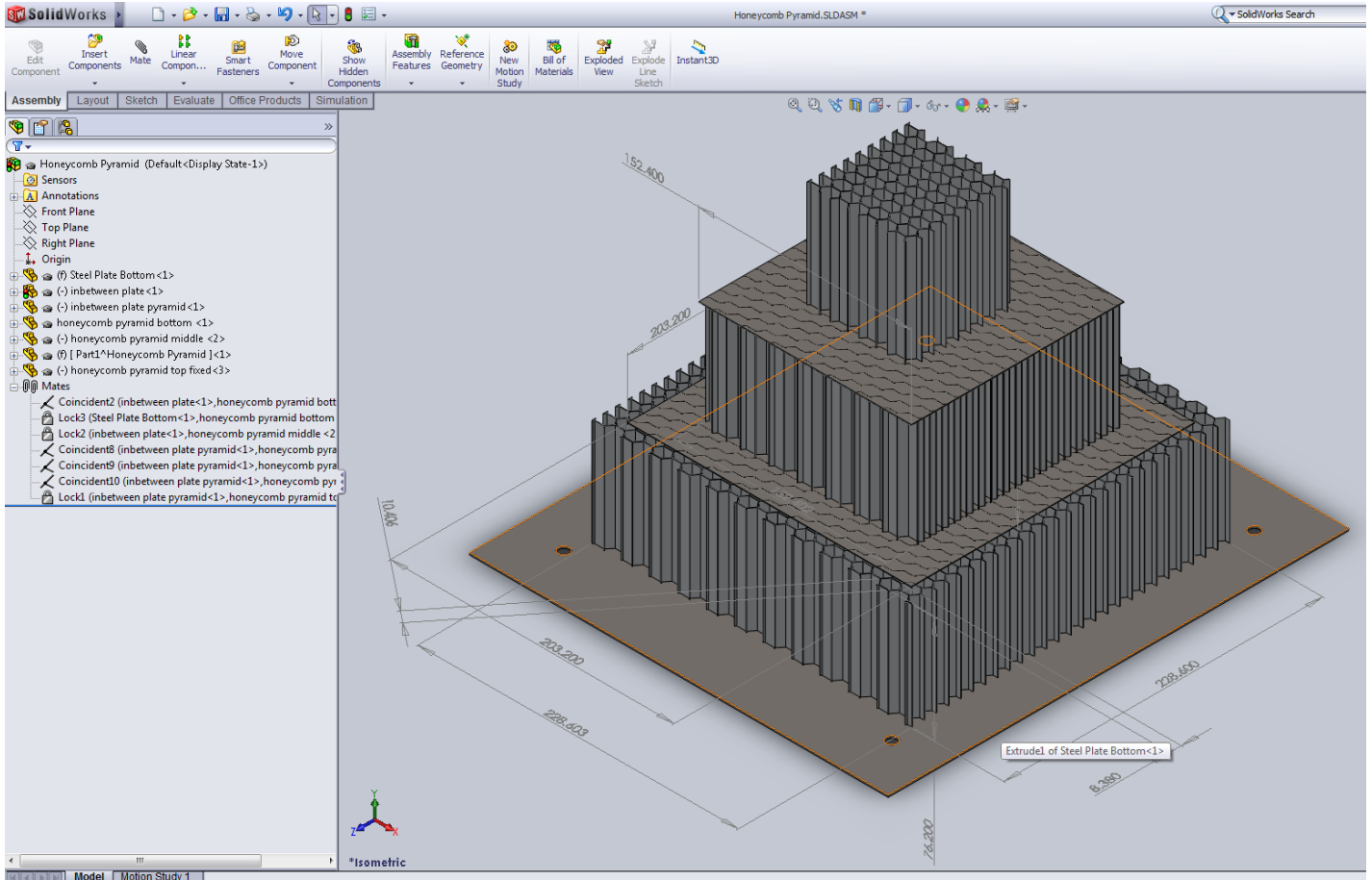


Figure 5: Complete Honeycomb Pyramid Model

Honeycomb-Foam Solid Model

The Honeycomb-Foam design was modeled using a similar back plate to that of the Honeycomb Pyramid. The 4130 annealed steel back plate model used is identical with the exception of four additional bolt holes. Each bolt hole is placed in the middle of the center line between the two existing frame bolt holes (130mm from the center of the corner holes on the 330mm side and 117.5mm from the center of the corner holes on the 300mm side). The new

holes will allow an aluminum strapping to be bolted down, which will support the impact attenuator structure (strapping not shown in the solid models, as it is not important in the design function during finite element testing). A screenshot of the steel back plate used is shown in

Figure 6.

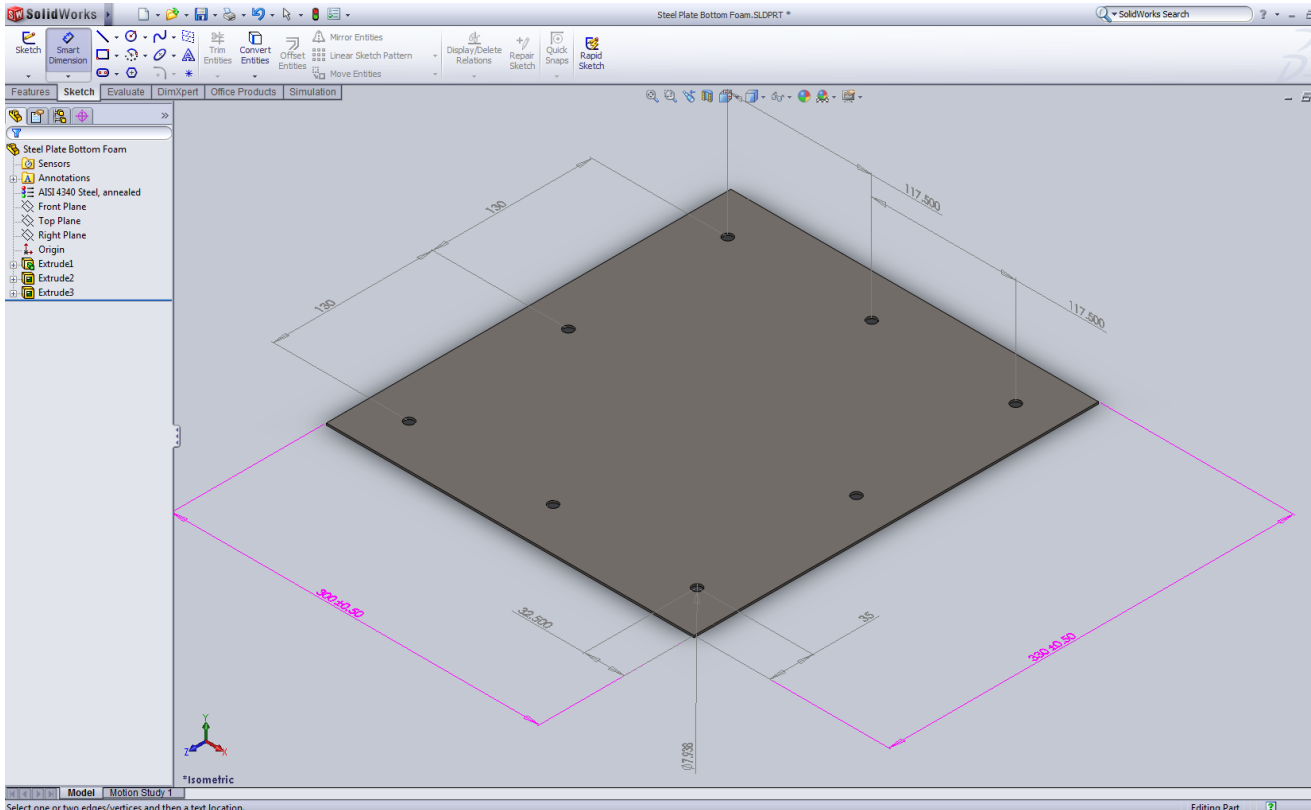


Figure 6: Honeycomb-Foam Steel Back Plate Solid Model

The second piece modeled for the Honeycomb-Foam design was a single layer of foam, this foam component is used twice in the solid model assembly. The component measures 8 in x 8 in x 1 ½ in (203.2mm x 203.2mm x 38.1mm) and was modeled by simply extruding a rectangular box. The properties of the high density polyethylene are available in SolidWorks, and were utilized. A screenshot of the component is seen in Figure 7.

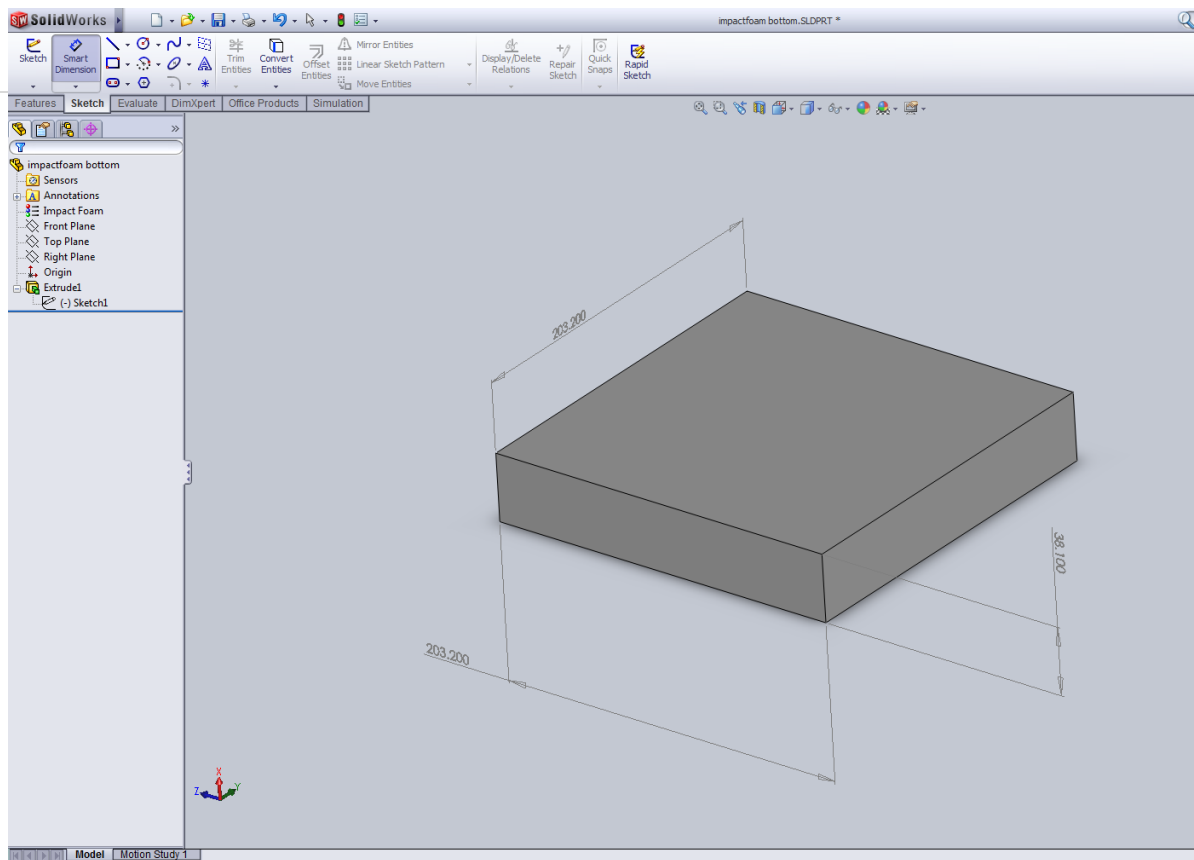


Figure 7: Honeycomb-Foam Foam Solid Model

An in-between 3003 Aluminum plate was then modeled similarly to that used in the Honeycomb Pyramid. The plate is placed between the foam and honeycomb structure on both the top and bottom to evenly distribute the impact force onto the honeycomb. The plate measures 8in. x 8in. x 1/32nd in., or 203.2mm x 203.2mm x .397mm. The solid model is displayed below in Figure 8.

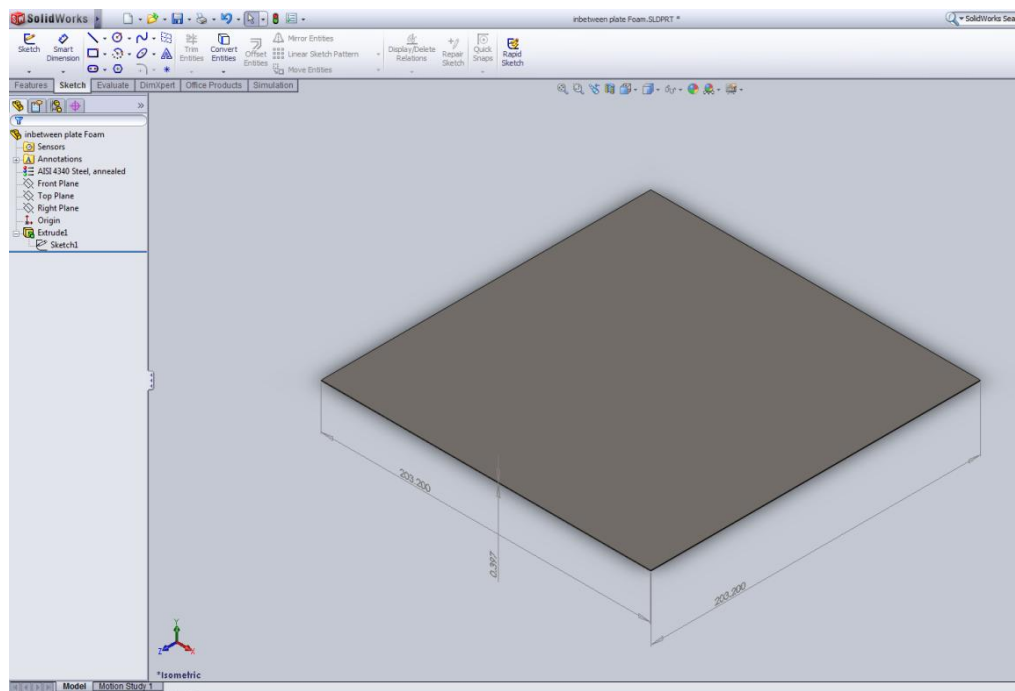


Figure 8: Honeycomb-Foam In-Between Steel Plate Solid Model

The creation of the large honeycomb structure then preceded the in-between plate solid model. The honeycomb structure was formed from a solid block measuring 8in. x 8 in. x 7in. (203.2mm x 203.2mm x 177.78mm). A linear pattern was created on the surface of the block in a hexagonal shape and then extruded cut into model, creating the honeycomb pattern. The material used in the solid model is 3003 Aluminum. The solid model is shown below in Figure 9.

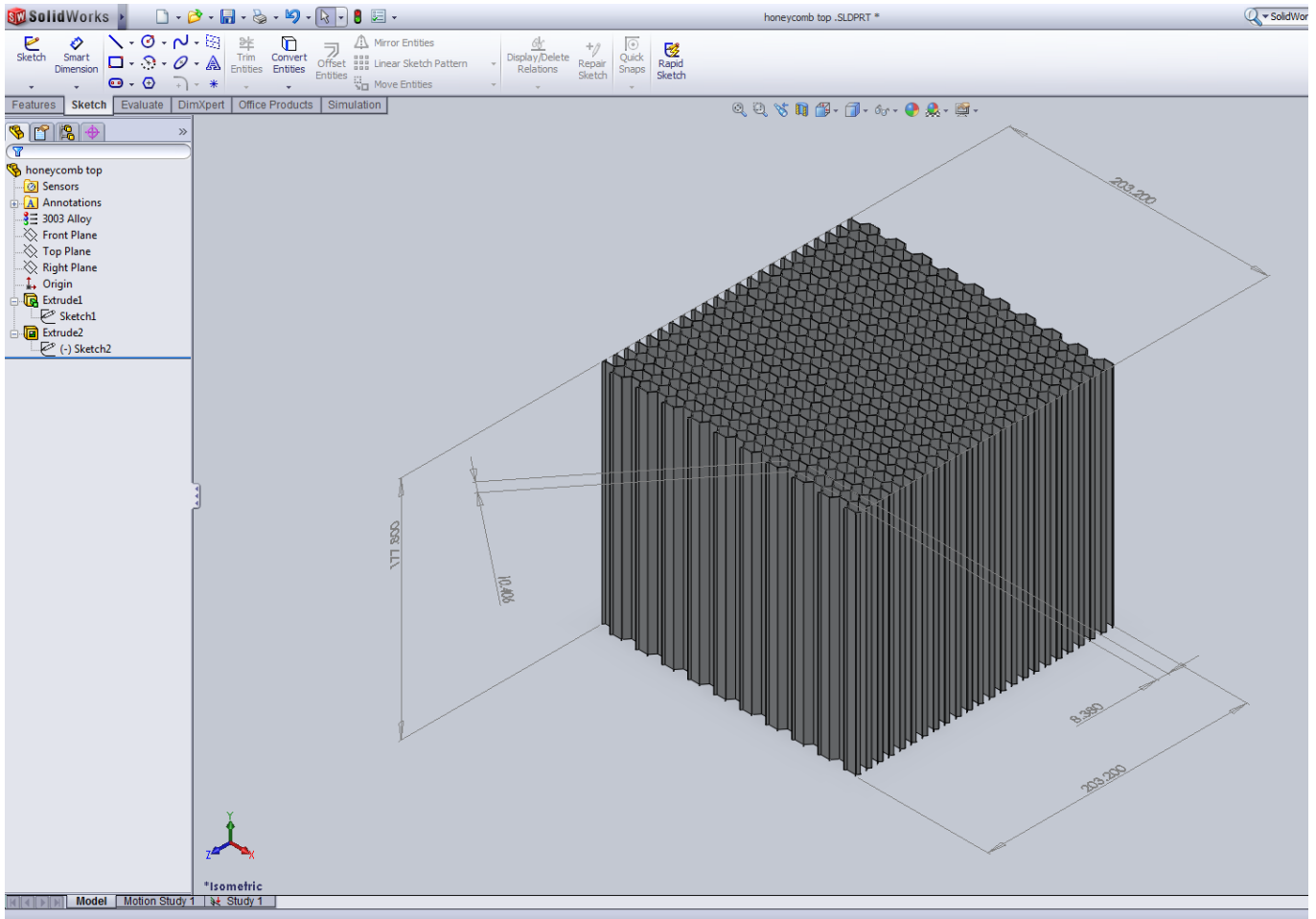


Figure 9: Honeycomb-Foam Honeycomb Solid Model

The solid model of the Honeycomb-Foam design was then created using the components shown above. The first step was to insert the steel plate bottom and a single foam layer into the assembly screen. The bottom face of the foam was then lock mated to the top surface of the steel plate and centered between the bolt holes. The top surface of the foam was then lock mated to

the bottom face of the in-between aluminum plate. Two bottom outside edges of the aluminum plate were then coincidentally mated to two top outside edges of the foam, respectively. This was done so the plate covered all areas of the foam and was perfectly centered on the component. The honeycomb model was then inserted into assembly and its bottom face was lock mated to the top face of the in-between aluminum plate. Two edges on the bottom of the honeycomb were also coincidentally mated to two top sides of the steel plate. Another in-between aluminum plate was inserted and had its bottom surface lock mated to the top surface of the honeycomb structure. Once again, two edges of each component were coincidentally mated together to correctly line up the parts. To end the assembly, a second layer of foam was inserted and had its bottom surface lock mated to the top surface of the in-between aluminum plate. Two edges were also mated coincidentally from each piece to finish the assembly. The finished solid model of the Honeycomb-Foam design is displayed in Figure 10.

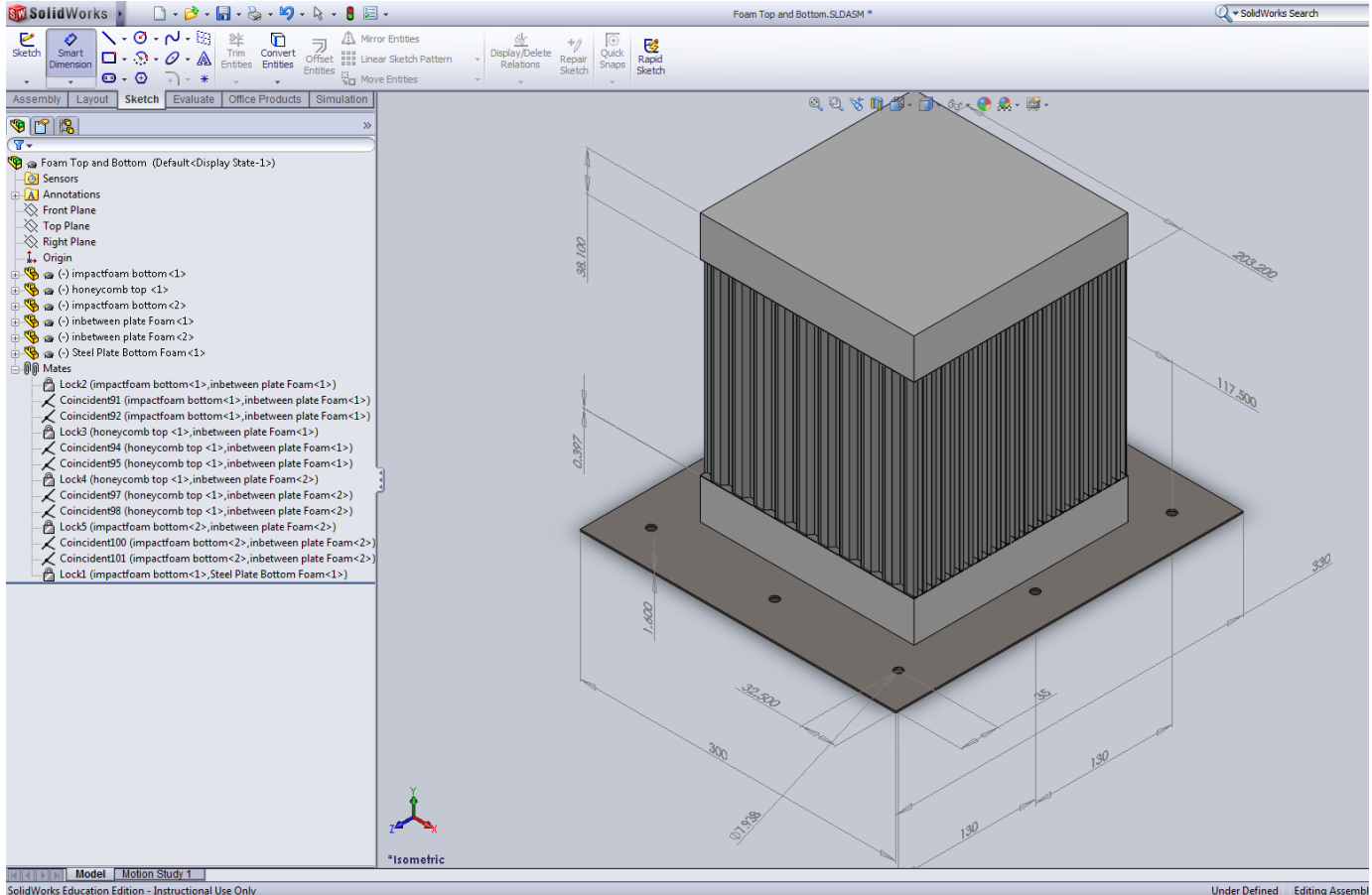


Figure 10: Honeycomb-Foam Solid Model

Finite Element Testing

Initial testing on the models often failed or was unable to finish due to the complexity of the honeycomb structure. The computer often stalled out or crashed when it came to the honeycomb piece of the assembly, while testing times often exceeded 5 hours. Thus, the team created a replacement part for the honeycomb that could be used in the finite element testing. The component had the mechanical properties of the honeycomb, yet was modeled as a solid block. The solid block replaced the honeycomb in the testing assemblies of both the Honeycomb-Foam design and Honeycomb Pyramid. The replacement part vastly improved testing time and allowed the group to try different variations of the designs with little consequence to testing time. The properties of the honeycomb replacement used are shown below in Table 5. A screenshot of the replacement part is shown below in **Error! Reference source not found.**; the size of the block was altered for the specific designs but is very similar.

Table 5: Honeycomb Replacement Component Mechanical Properties

Property	Value	Units
Elastic Modulus	71,000	N/mm ²
Poisson's Ratio	.3	
Shear Modulus	440	N/mm ²
Density	8.32×10^{-5}	g/mm ³
Tensile Strength	290	N/mm ²
Compressive Strength	4.6	N/mm ²
Yield Strength	268	N/mm ²

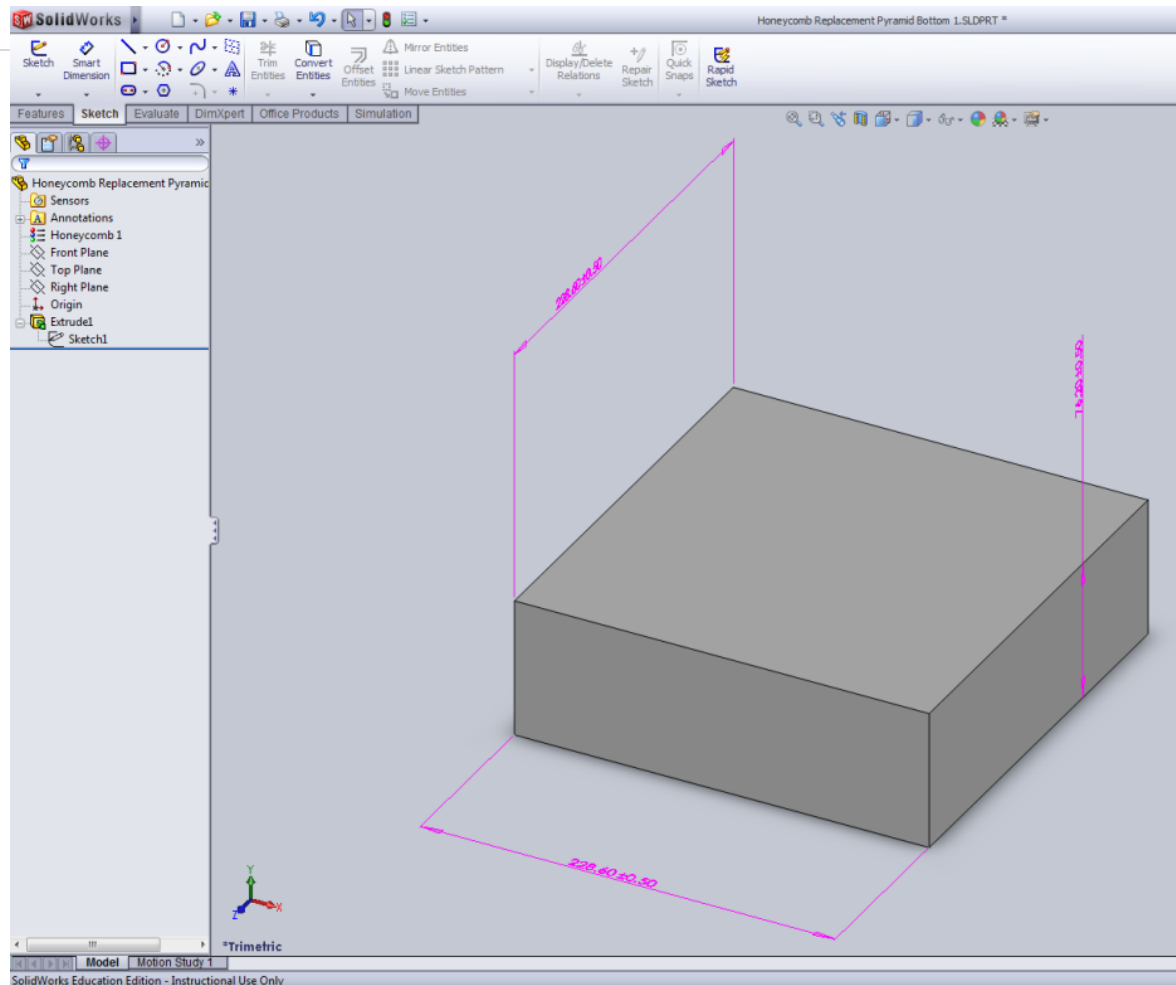


Figure 11: Honeycomb Replacement (Testing)

Test Set Up

Each model was tested using the SolidWorks Simulation tool taken from the Add-Ins section of the Tools drop down menu. The meshing for both test models was generally coarse to allow the testing to run more smoothly, giving a general idea of the deformation. The a/b ratio varied depending on how much impact the part was going to observe. The Static Test Simulation was chosen for each model and was set up in the following manner.

The Honeycomb Pyramid had its bottom steel plate contact set established as no-penetration with the underside of the largest honeycomb step. The contact set between the remaining models component surfaces was set to Bonded. A fixed geometry was applied to the underneath of the steel plate and around the bolt holes. An even force was applied to the top surface of the highest honeycomb block. Mesh controls were then applied to the model, and the

bottom steel plate was set as mostly coarse with an a/b ratio of 1.4. The bottom in-between plate was set to all coarse and had an a/b ratio of 1.4. The top in-between plate had a mesh control

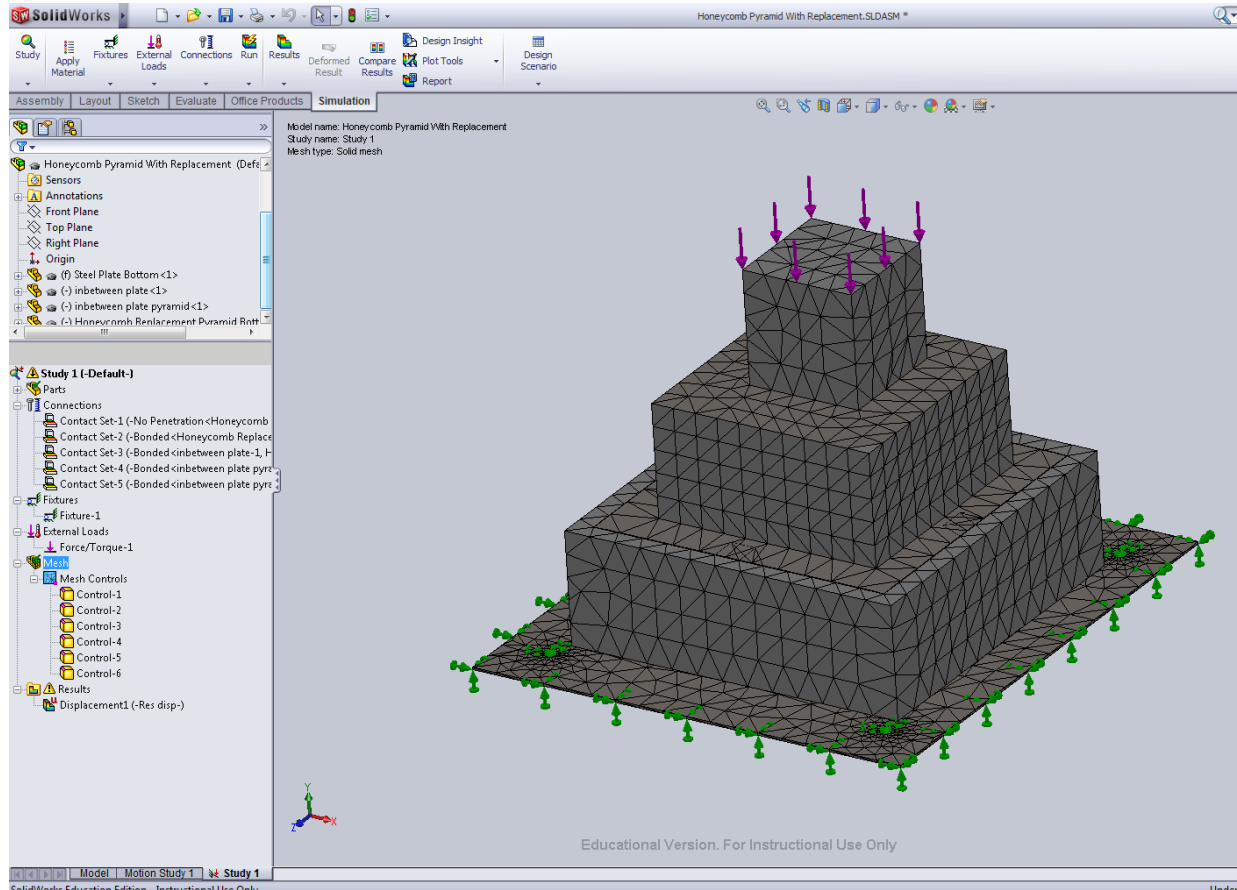


Figure 12: Honeycomb Pyramid Meshed Model

implied that set it to fairly coarse with an a/b of 1.4. The bottom honeycomb layer had a mesh control that held it to practically all coarse with an a/b of 1.4. The middle honeycomb step was set to fairly coarse with an a/b of 1.4. While the top step was also fairly coarse with an a/b of 1.4. The overall mesh was done by using a curvature based mesh that included: the minimum number of elements in a circle set to 12, the element size growth ratio set to 1.3, and the slider set to fairly fine. The meshed model is shown in The contact set between the surfaces of all the components in this model was set to Bonded. A fixed geometry was applied to the bottom of the steel plate along with each bolt hole. An evenly distributed force is then placed on the surface of

the top foam layer. Mesh controls were then set for each component as follows. The bottom steel plate was set to fairly coarse with an a/b of 1.3, while the lowest layer of foam was also established as fairly coarse with an a/b of 1.4. The bottom in-between plate had its mesh set to moderately coarse with an a/b of 1.3. The honeycomb itself was set to an intermediately coarse level with an a/b of 1.3. The top in-between plate was set to fairly coarse with an a/b of 1.3, while the uppermost layer of foam was set to mostly coarse with an a/b of 1.4. The meshing was done by using a simple mesh density of moderately coarse. The meshed model is shown in Figure 13.

Test Results

The finite element testing results found by the team were not appropriate for the given forces applied. The solid modeler used by the team interpreted the forces differently and produced unexpected results. The calculated force of 58800N, created during a crash for the given parameters, resulted in very small deformations for the solid models (less than a cm). This could be explained in a multitude of ways.

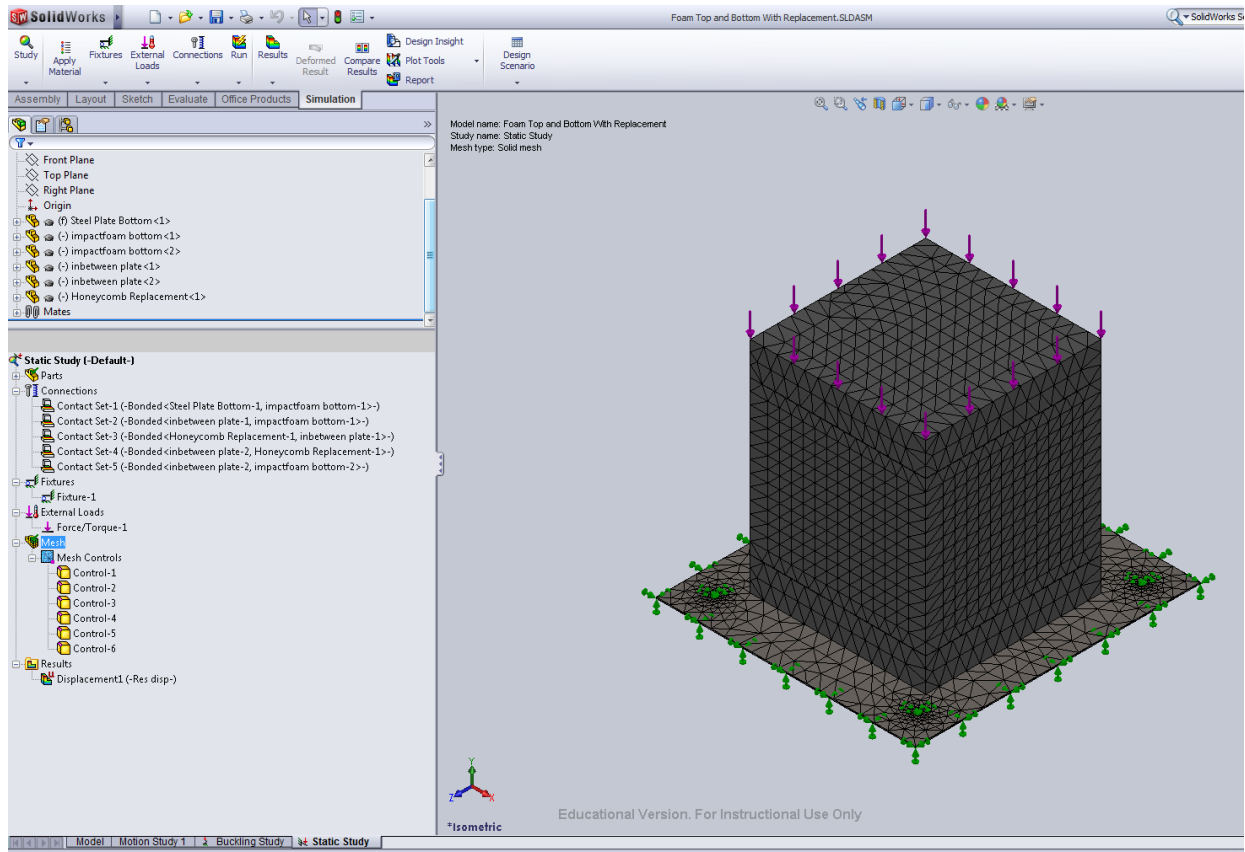
The solid modeler itself does not take into account that there is an impact, rather than a static force applied to the model. The drop test simulation was applied and tried many times to the models but did not create a successful test; there were constant failures due to computing restraints. Therefore the static testing was used, which does not accurately represent a crash. A highly dynamic testing simulator would create better results such as Abaqus; SolidWorks does not provide the necessary tools for an accurate test in this situation.

Consequently, the forces applied to each model were increased by many factors to produce a desired deformation. For the Honeycomb Pyramid a force of 2.7×10^8 N was applied to the model, it produced a deformation that looked congruent to the results produced during the actual test. The Foam-Honeycomb-Foam design used a force of 2×10^7 N to yield a collapse that is similar to a real situation. The resulting deformations are shown in Figure 14 and Figure 15.

The Foam-Honeycomb-Foam model was also prepared for its finite element testing in the same method. The contact set between the surfaces of all the components in this model was set to Bonded. A fixed geometry was applied to the bottom of the steel plate along with each bolt hole. An evenly distributed force is then placed on the surface of the top foam layer. Mesh controls were then set for each component as follows. The bottom steel plate was set to fairly coarse with an a/b of 1.3, while the lowest layer of foam was also established as fairly coarse with an a/b of 1.4. The bottom in-between plate had its mesh set to moderately coarse with an a/b of 1.3. The

honeycomb itself was set to an intermediately coarse level with an a/b of 1.3. The top in-between plate was set to fairly coarse with an a/b of 1.3, while the uppermost layer of foam was set to mostly coarse with an a/b of 1.4. The meshing was done by using a simple mesh density of moderately coarse. The meshed model is shown in Figure 13.

Figure 13: Meshed Honeycomb Foam Model



Test Results

The finite element testing results found by the team were not appropriate for the given forces applied. The solid modeler used by the team interpreted the forces differently and

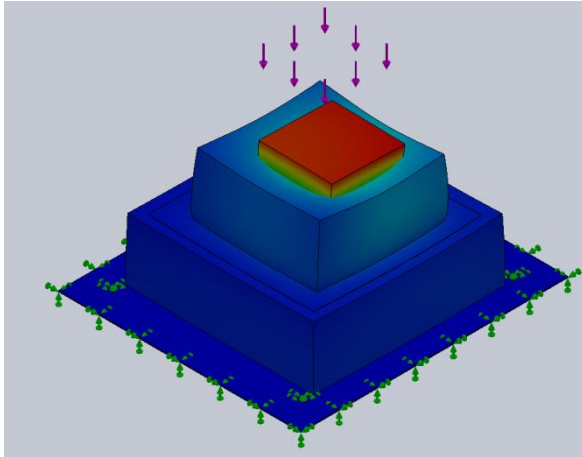


Figure 14: Honeycomb Pyramid Test Results

produced unexpected results. The calculated force of 58800N, created during a crash for the given parameters, resulted in very small deformations for the solid models (less than a cm). This could be explained in a multitude of ways.

The solid modeler itself does not take into account that there is an impact, rather than a static force applied to the model. The drop test

simulation was applied and tried many times to the models but did not create a successful test; there were constant failures due to computing restraints. Therefore the static testing was used, which does not accurately represent a crash. A highly dynamic testing simulator would create better results such as Abaqus; SolidWorks does not provide the necessary tools for an accurate test in this situation.

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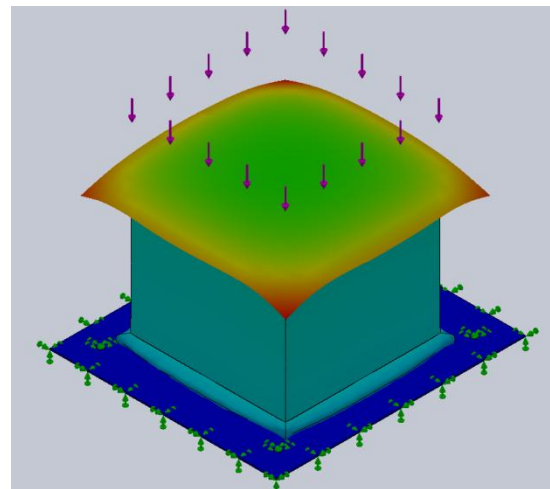


Figure 15: Honeycomb Foam Test Results

produced a deformation that looked congruent to the results produced during the actual test. The Foam-Honeycomb-Foam design used a force of 2×10^7 N to yield a collapse that is similar to a

real situation. The resulting deformations are shown in Figure 14 and Figure 15. Although these results are not practical, they provided information on how each model would collapse. This allowed the team to move onto the fabrication stage.

Additionally, SolidWorks could not account for the non-linearity of a deformation during a force that true honeycomb would produce. The solid blocks would create a similar crush pattern but not produce a realistic result.

Fabrication

Once finished with finite element analysis testing, the group was ready to move onto the next step. The first step was to order the materials. Going through WPI's Mechanical Engineering Department with the help of Barbra Furhman, the team ordered the necessary components to assemble the impact attenuators. All hardware was purchased from a local home improvement store as there was no need to have those items shipped.

As the materials arrived, the group began the process of constructing the attenuators. First and foremost was the sizing and cutting of the steel base plates of each attenuator. The group got help from Torbjorn Bergstrom in instructing us on how to accurately cut this strong material. We were directed to use Washburn Lab's power shear cutting machine. The machine presses a moving blade onto a stationary one. These two forces amount to the yield strength of the material, cutting it to our desired dimensions.

To make cuts on the aluminum honeycomb, a much less powerful tool was needed. The group used a band saw to slice the blocks into the dimensions of each attenuator. One problem encountered was the gaps between materials on the honeycomb. While pushing it through the saw there tended to be jumps when the blade met an open space in the material. Some cuts came

out uneven and with frayed edges. This was solved using a traditional file to straighten out the edges and produce a smoother final cut

Cutting of the foam was the simplest part of fabrication. A sharp pair of scissors easily made it through the porous material with ease. Some cuts did come out fairly rough but the use of sand paper created smoother surfaces after the cuts.

Holes for the both the attachment of strapping and mounting purposes was done using a drill press. Each hole was sized up for ease of integration to the existing FSAE frame. The corresponding drill bit was inserted into the chuck of the press and secured tightly. The steel plate was securely mounted using a standard size C-clamp. During cutting, the feed rate was set to 90 RPM due to the strength of the material. All holes were made with no major complications.

What turned out to be the most problematic piece of fabrication was our adjustment made to the testing apparatus. The drop tester is usually used by the civil engineering department for testing steel framing for durability at their joints. This means they want a large force concentrated in a small area. For our test, the group wanted a flat, even spread of force across the top of our proposed designs. We took apart the drop tester weight assembly and acquired one of the large weight plates for our point of contact.

The weight of the drop tester is held up by four metal rods that have a bolt at each end. In order to obtain a level striking surface, we needed the bolts to be below or flush with the bottom surface of the drop tester. We decided it would be best to counter-bore into the steel plate from the test apparatus. A one inch counter-bore was chosen to sufficiently fit the nut and a ratchet for tightening. A depth of one half of an inch was also agreed upon for the cut. The group sought the help of Michael Scanlon and Adam Sears. They directed us to find the proper

bit and bore to meet or correct width and depth requirements. They also aided us during the cutting process. The large plate was fastened to the drilling table by the means of two large C-clamps. A feed rate of ninety rpm was again used during this drill. While cutting the second hole, the bit itself caught the edge of the plate and sheared itself into pieces. This was a setback for our target testing date. A new custom bore had to be fabricated to fit our dimensions and complete the fabrications. Once completed, all prototypes were assembled and ready for impact testing.

Sandwiching together the components of the attenuators was done using an industrial grade adhesive. Only the adhesive was used for the test models because the only force being applied was vertically down on each design. On the final designs, aluminum strapping was used to secure the attenuator to the base plate.

Testing Procedure

In order to ensure the functionality of the designs created, the group decided to conduct a drop test. The purpose of this test is to subject the attenuators to forces required by FSAE regulations. In the rule book is a list of regulations that each car in competition must abide by. Section B3.21.1 of the rulebook states that:

“The team must submit test data to show that their Impact Attenuator, when mounted on the front of a vehicle with a total mass of 300 kgs (661 lbs) and run into a solid, non-yielding impact barrier with a velocity of impact of 7.0 meters/second (23.0 ft/sec), would give an average deceleration of the vehicle not to exceed 20 g’s, with a peak deceleration less than or equal to 40 g’s.”⁵

⁵ 2010 Formula SAE Rules pg. 25 © 2009 SAE International. All Rights Reserved.

This testing criteria dictates the creation of the drop test. In order to evaluate which designs will meet these criteria, a drop testing mechanism located on the campus of Worcester Polytechnic Institute was utilized. It was designed for controlled impact tests on structural assemblies for the civil engineering department. The group got the chance to use and adapt the equipment for their impact testing needs.

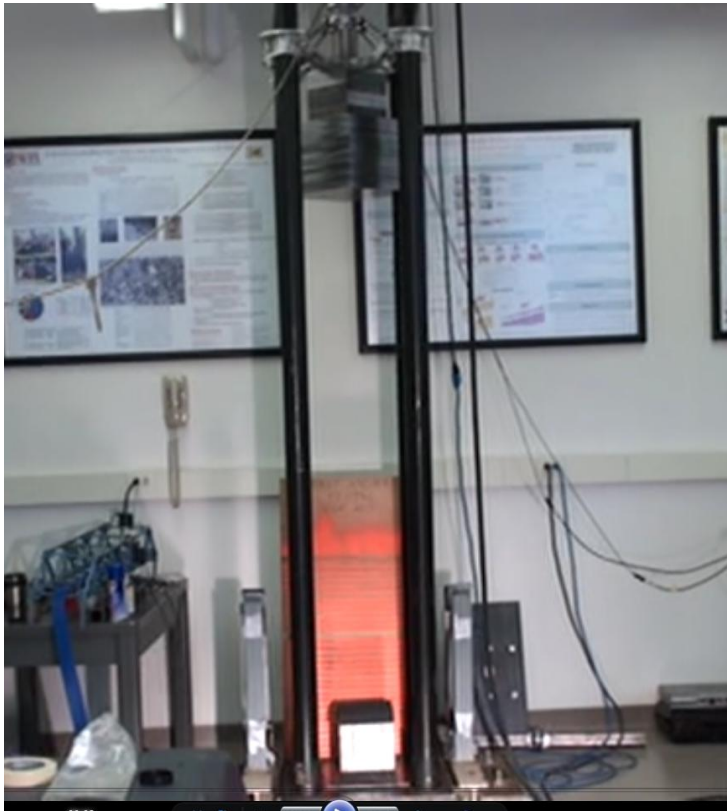


Figure 16: Test Apparatus Moments before Impact

Figure 12 is a picture of the drop testing apparatus prior to impact with the honeycomb and foam design. It has four main supports which hold the weight needed for each specific test. The supports also act as a track system to keep the load on a straight path to its intended target. Although the device itself is 10 feet tall, once weights are attached the center of mass drops to about 7.5 feet.

In determining the kinetic energy required for the impact test earlier in the report, it was found that 7350 Joules of energy needed to be involved in the impact. Using the weight of the car as 661 pounds, as specified in the rules, required a drop height of 8.2 feet. In order to meet the requirements the team needed to complete a kinetic versus potential energy balance at the 7.5 ft. that was available from the drop tester. In order to ensure proper results, this new height was

converted to meters (2.286 m) and used to find the mass that could provide the same kinetic energy.

$$P_e = m * g * h = K_e = 7350 J$$

$$m = \frac{7350}{g * h} = \frac{7350}{9.81 * 2.286} = 327 kg$$

$$327kg = 721 pounds$$

This yielded a result of 327 kg which is 721 pounds. The weight system is managed by individual steel plates of different sizes so that the desired load can be accurately manipulated. There were 4 plates weighing 59.6 pounds each, 8 plates at 44.7 pounds, and 7 plates that weighed 18 pounds, for a total of 722 pounds. The guidance system for the dropping mechanism added ten pounds to this weight, yielding a total weight of 732 pounds. This additional ten pounds of the guidance mechanism provides a balance for the friction factor that decreases the velocity at the moment of impact. The weight is raised and lowered by an electronic motor pulley system which is remotely operated. There is a quick release mechanism where the pulley attaches to the load. This is operated by a ripcord that has adequate length for a safe initiation of the drop test. The attenuator was centered at the bottom of the apparatus for the testing.

Data collection for the tests was done in two ways. The first being accelerometers rated at 50 and 500 g that were mounted to the impacting load. These accelerometers were integrated into a LabVIEW monitoring system through a Data Acquisition Board which allowed for the data to be recorded. The 50 g accelerometer was measured using a charge of 100 mV/g and the 500 g accelerometer was measured by using 10 mV/g. Data from the accelerometers was set to

be recorded at 4500 readings per second which ensured that there would be plenty of data over the short period of time the impact was expected for. All of the recorded data was transferred into Microsoft Excel for further analysis. This information indicated whether the particular



Figure 17: High Speed Camera

attenuator passed or failed the FSAE regulations of 20g's of deceleration with a peak less than 40g's.

A high speed camera was also utilized to record the actual impacts, displayed in Figure 13. This provided a good picture of

how each design deformed at the moment of impact and over the duration of the crash. Pictures of the attenuators were taken before and after impact as required in the rulebook.

Results

In order to review the results from our impact testing, we relied on Professor Don Pellegrino and Mario Mongiardini from the Civil Engineering Department. Professor Pellegrino helped our group to edit the high-speed camera videos. We cut out the portions of the videos which did not show the impact or deformation. The remaining video portions were saved as Windows Media files and placed on flash drives for easy reference.

To review the accelerometer data, we asked for assistance from Mario Mongiardini from the CE Department. Mario was instrumental in taking data from LabVIEW and displaying it in a manner we could utilize. The time vs. acceleration values were placed into Matlab and plotted. After that, Mario gave us the data in an excel file so that our group could reproduce the plots and

have the numerical data on hand. In this way, we were able to get a visual of the peak acceleration values while also taking data from excel to calculate velocity and average acceleration to calculate whether or not the designs were up to FSAE specifications.

Test One- Honeycomb/Foam Attenuator

The first design tested was the Foam-Honeycomb-Foam design. This design was expected to dampen the impact and excessive vibrations because the foam would keep a crushed honeycomb from creating these vibrations by impacting the solid back structure. Also, the honeycomb block was expected to absorb nearly all of the impact. The attenuator design directly before impact is shown in Figure 14.



Figure 18: Honeycomb-Foam Prior to Testing

In observing the visual data and the plots of deceleration, our group determined that the polyethylene foam helped to decelerate the drop tester without causing a large peak. The top and bottom foam pads were compressed completely before the honeycomb began to deform. The plot shows a linear deceleration increase before the much larger deceleration peak from the honeycomb emerges. This is also confirmed in the data where linear deceleration can be seen directly before each peak of much higher deceleration magnitude. Since including the foam acceleration would greatly reduce the average deceleration with many smaller values, we decided to take the average of the honeycomb peak only to maximize the average.

The 50 g (Table 5) and 500 g (Table 4) accelerometers were precise in that both produced a similar peak. In looking at the 500 G accelerometer data, the peak value for the Foam-Honeycomb-Foam design was about 22.08 G's. Also, the drop tester bounced a few times on of the attenuator, giving several peaks of smaller sizes in deceleration. The first bounce gave a peak of 10.39 g's, the second was 6.47g's, and the others were smaller. However, these secondary peaks were not used in calculating average deceleration as nothing would be forcing the car to continuously accelerate forward like gravity in the drop test. The Excel Data showed the average acceleration of the initial peak to be 11.18 g's over a time period of .06 seconds ($t=15.26$ to $t=15.32$) before the drop tester bounced upward. The impact resulted in the deformed structure displayed in Figure 15.

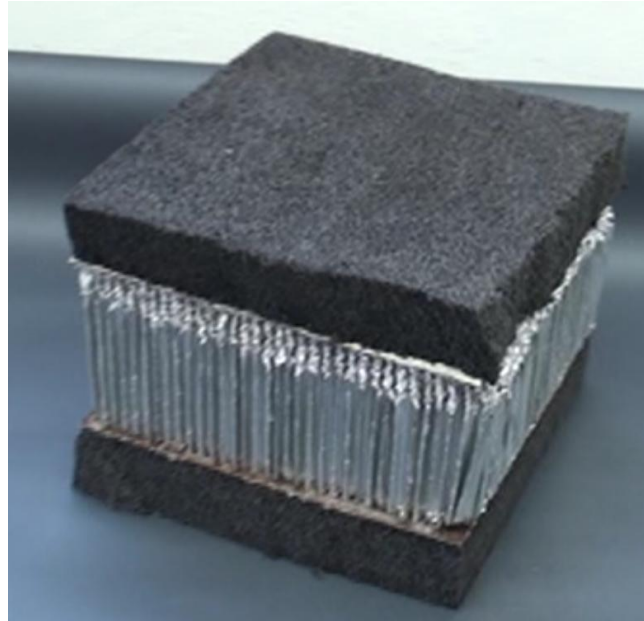


Figure 19: Honeycomb Foam Test Deformation

The 50 g accelerometer data shows the peak acceleration value to be about 22.998 g's. It also showed the drop tester bouncing on the attenuator, with small peak values of 12.01 g's and 7.54 g's. The Excel Data from the 50 g accelerometer showed the average deceleration of the initial peak to be 12.48 g's over the same .06 seconds. Our group will use this value since the 50 g accelerometer is more accurate and yields higher values than the much larger rated 500 g accelerometer.

Table 6: 500G Honeycomb/Foam Data

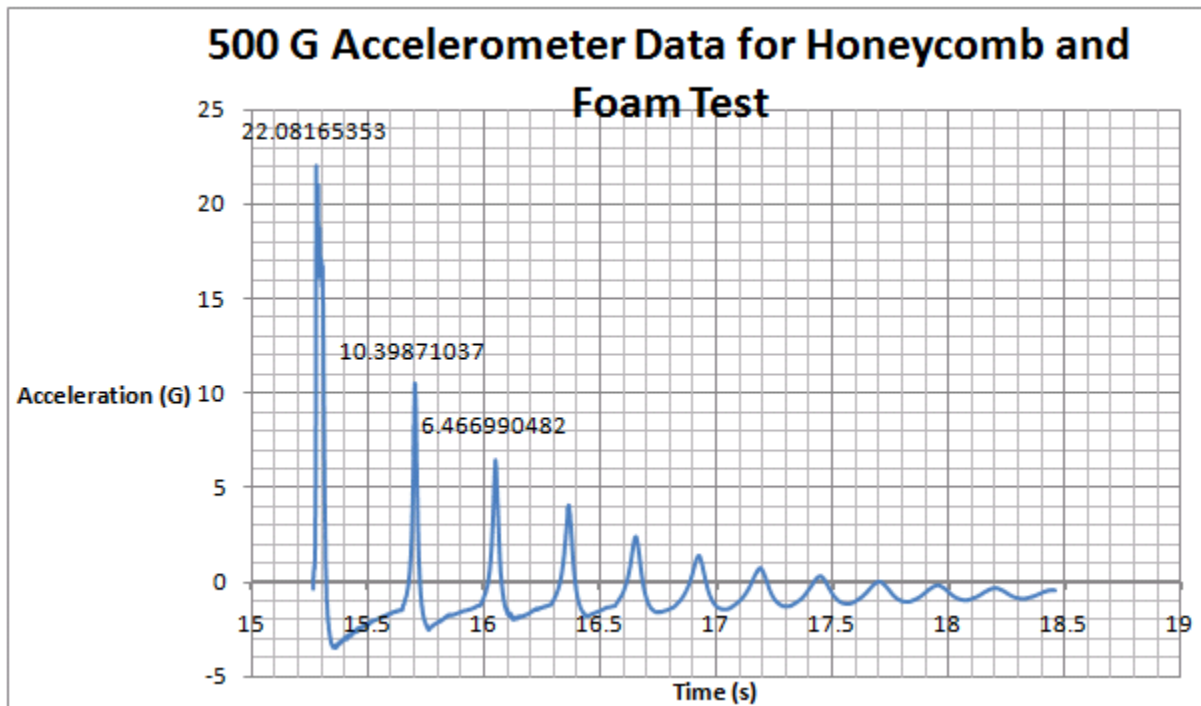
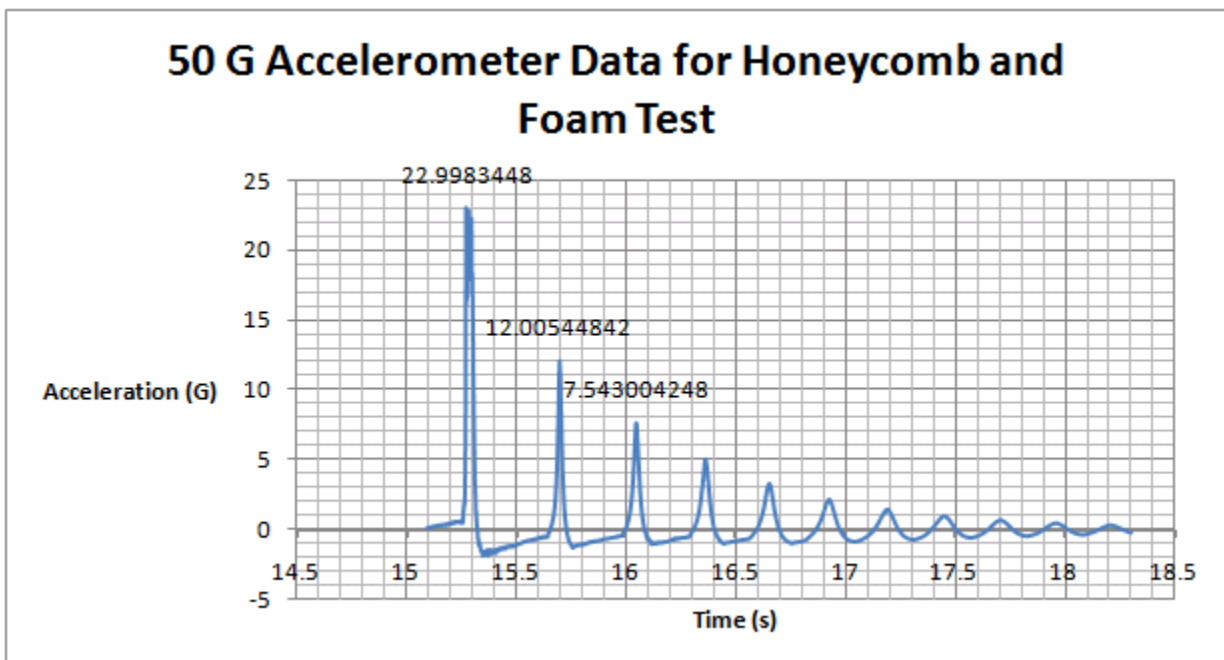


Table 7: 50G Honeycomb/Foam Data



Discussion

There were small differences between the 500 and 50 g accelerometer data, but our group used the 50 g data for analysis because it is more reliable with smaller values experienced in the drop test. It also showed higher values for the peaks and averages. The initial peak of 22.998 g's is much lower than the specified 40 g maximum. This large peak was caused when the polyethylene foam was completely compressed and the honeycomb began to absorb the impact.

The polyethylene foam helped decrease the spike of acceleration. If the honeycomb block were alone, the impact may have completely compressed the honeycomb and caused a collision of hard materials, with the steel drop tester hitting the completely crushed aluminum honeycomb with only a steel plate underneath. Foam helps to dampen any of the vibrations caused by these hard collisions. At the top of the highest peaks are a series of miniscule peaks. These represent the polyethylene absorbing some of the large vibrations from the impact.

The foam also retained its size and shape through the impact testing. When compressed, the top and bottom polyethylene layers acted like springs to push the drop tester off of the attenuator. In a car crash, this would translate to the attenuator pushing the car away from a non-yielding barrier. Also, small impacts that cause little or no deformation of the honeycomb would allow the attenuator to be used again.

The aluminum honeycomb did not crush completely. Therefore, our group could reduce the size of the block in order to optimize the weight. However, much larger impacts could be devastating to an attenuator of less impact deformation yield.

Our group chose to put sheets in between honeycomb and foam because we felt the foam might penetrate the gaps during an impact. In this way, the sheets protected the foam to allow it to serve its purpose.

Test Two: Honeycomb Pyramid

The two accelerometers were also precise during the second impact test. In looking at the 500 g accelerometer data (Table 7), the peak value for the Honeycomb Pyramid design was 24.40 g's. Once again, there were peaks of 6.41 g's and 2.45 g's that followed when the drop tester bounced upward.

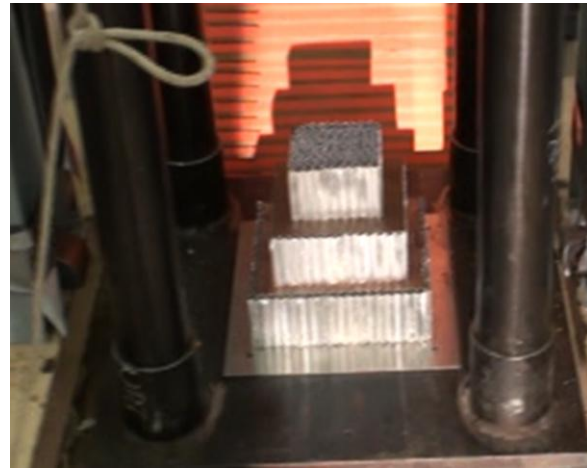


Figure 20: Honeycomb Pyramid Before Impact

These peaks were not used in the group's analysis. The Excel Data for the 500 g accelerometer also showed an average acceleration value of 13.49 g's over the initial peak time period of .04 seconds ($t=3.88$ to $t=3.92$). An image of the Honeycomb Pyramid prior to testing is shown in Figure 16.

The 50 g accelerometer (Table 6) showed a peak of 26.38 g's. The secondary peaks reached values of 6.91 g's and 3.17g's. These values are very similar to those from the 500 g data. Using excel data of time vs. acceleration, our group found the average acceleration of the impact test was 14.75 g's over the .04 seconds. Since this is higher than the 500 g data and is more reliable, our group used the 50 g data in our analysis.

Table 8: 50G Pyramid Data

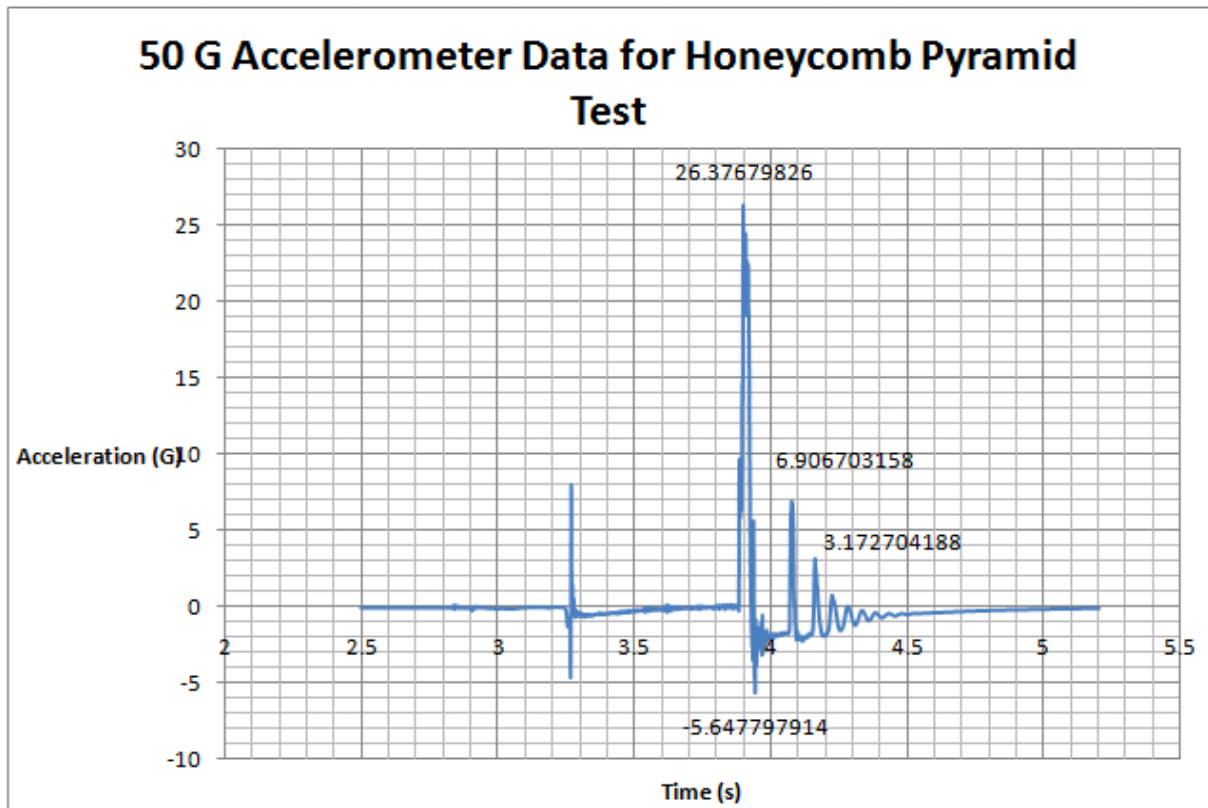
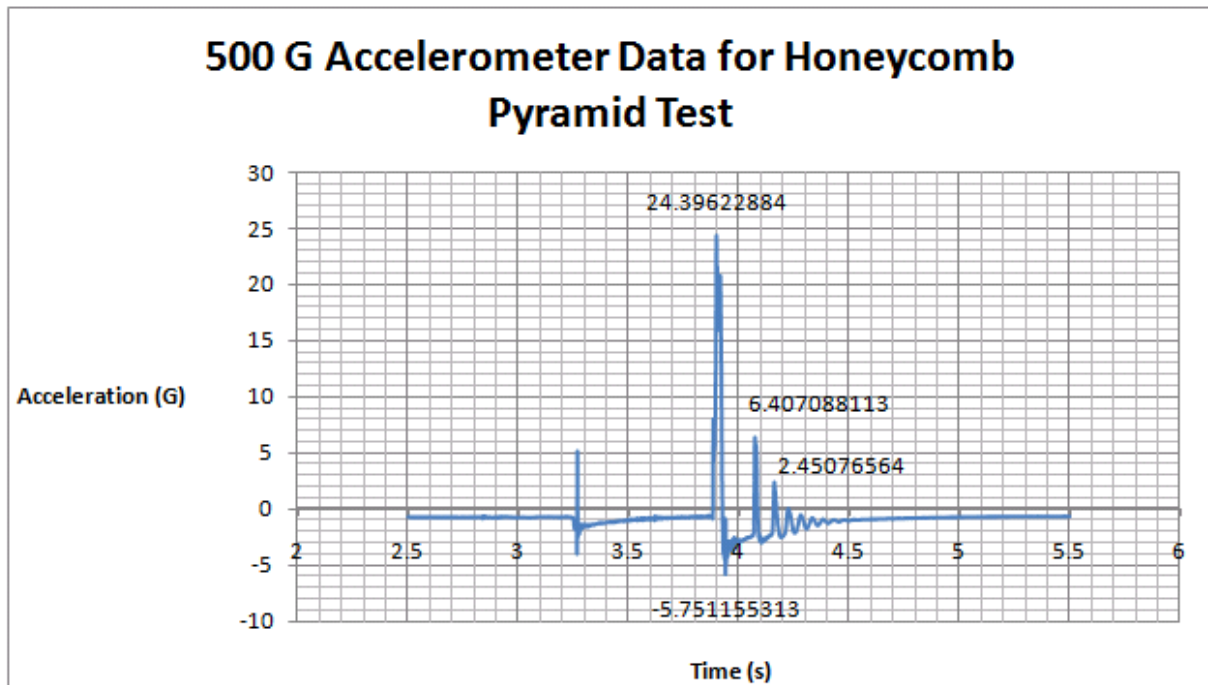


Table 9: 500G Pyramid Data



Discussion

There were somewhat larger differences in the values of peaks and averages between the two accelerometers. Once again, we used the 50 g accelerometer data because it provided higher values and was more reliable for this impact than the 500 g data. The initial peak of 26.38 g's was again lower than the specified 40 g maximum deceleration value. Also, the average of 14.75 g's was lower than the maximum average of 20 g's in the rulebook.

The top aluminum honeycomb block absorbed the initial energy as the plot began to show the peak. When this block of honeycomb was completely crushed, the second block began to deform. It absorbed more of the impact. In watching the video and looking at the data, the impact lost most of the energy before this second block was completely crushed. The final block of honeycomb was barely deformed. The deformed result of the test is show in Figure 17. The

crushed attenuator and the video signify that the momentum was not enough to overcome the initial impact needed to deform the final block. A better design would have a deformed and smaller 3rd block or a reduction in the sizes of aluminum honeycomb blocks altogether.

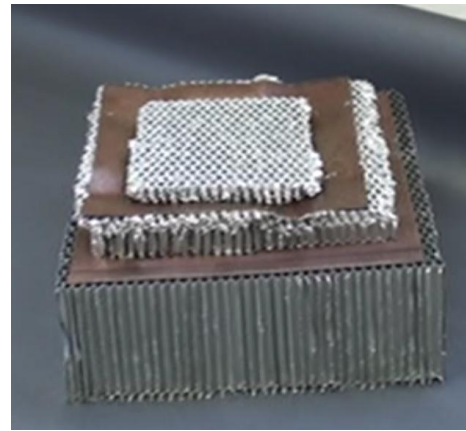


Figure 21: Honeycomb Pyramid Test Deformation

The sheets in between the layers once again served their purpose. None of the honeycomb was able to penetrate the adjacent layer. Since the sheets were the size of the larger adjacent blocks, the smaller blocks fully deformed before the larger ones absorbed the impact.

After the testing and analysis, a different interpretation of the minimum dimension requirements was found to be in effect. This rule stated that the minimum size of 3.9 inches high and 7.8 inches both high and long. Our team believed this to state that the impact attenuator had to have these minimum dimensions at only the base and the length had to be 7.8 inches. The interpretation that was published stated that an approved impact attenuator had to be large enough to fit a volume of those dimensions.

Although this discovery meant that the pyramid design failed one of the team's main objectives of being fully in conformance with FSAE rules and regulations, many good findings came from this design as can be shown in the next section "Recommendations".

Recommendations

After the analysis of our test data, the team has developed a set of recommendations to improve upon materials, design and testing. The group has found imperfections in these areas that could be improved upon. These suggestions could help to create an optimal design for use in an FSAE racecar. They will be split up into sections and discussed in detail below.

Materials

The materials we used for our design were aluminum honeycomb, polyethylene foam and aluminum sheets. In regards to the honeycomb there are many aspects that could be adjusted to produce the best possible design. A recommendation for the pyramid design would be to explore the various depth of pre crushing. Pre crushing the middle and bottom layers would further assist in dampening the peak deceleration spike. It would also allow more energy to be transferred smoothly into the structure. If all layers were pre crushed then there could have greater deformation in the pyramid. In our test the bottom layer had no deformation and this technique could have transferred the force through the entire attenuator. Another suggestion would be to explore the effects of different depths of pre crushing. A larger pre crush could prove to further increase the energy absorbed by the structure. It could also lead to a much smaller spike in deceleration. This depth of pre crushing should be explored to produce the optimal balance of impact yield versus the effect on the initial spike during a collision.

In the both designs, aluminum sheets were placed in between the different layers of materials. The group feels as if this is the best way to properly displace the force into the subsequent layer. This could be used in a multitude of different design where layered materials are used. This design idea is also successful in preventing the penetration of one layer into another. Penetration would not allow the separate layers to crush correctly which would affect the materials impact properties.

The foam that was used for our design was high density closed cell polyethylene. This material created a spring effect during the test producing subsequent peaks after the initial impact. These peaks would create more force on the driver and also act as reaction that would push the vehicle away from the barrier. The results illustrate that foam chosen had a density that was too large for this application. Slightly less dense foam could be explored to try and dampen the peaks after impact. The less dense foam would be more of a cushion than a spring during the dynamic process. The foam should hold its deformation rather than trying to resist against it. This would produce an overall greater transfer of energy from the impact.

In holding the structure together the team used a commercial strength adhesive called Gorilla Glue. This can be found in any local home improvement store. This bonded to all materials sufficiently and held them together during the test. The group realizes that these designs will be mounted horizontally on the car and endure many different forces when participating in a race. The solution for this is to attach malleable strapping to provide further restraint for the attenuator. These would be firmly bolted down to the steel backing and tightened to ensure the designs will not slide. Due to their weak nature, these straps would have minimal affect on the force absorption properties of the device.

Design

In regards to our final design, there are multiple areas that could be looked into further to create the best possible attenuator. Honeycomb was used in both designs. The team decided to stick with one cell size for both prototypes. The sizing of the cells is a feature that could be adjusted to optimize performance; a larger cell size would allow more deformation. A larger cell size would also absorb the same impact energy with a larger surface area to fill the volume requirement. Manufacturers offer honeycomb in a wide variety. These different sizes could be

explored to find the optimal cell sizing for energy absorption. How much impact the material can absorb could be directly related to cell size. There are also many different kinds of honeycomb out on the market. The team chose to use aluminum. Different metals/materials could be investigated to see which can produce the best results. Although aluminum seemed like the best for this application, new materials are being developed that could be used for impact attenuation in the honeycomb type arrangement.

Finite Element Testing

Finite element testing is a useful tool in determining the effects of forces on materials. In the case of an impact attenuator, it was difficult to replicate an impact on SolidWorks. It took a great deal of trial and error to get the correct variables and mesh to get test results. When we were failing to get results, our group was not sure if our design was incorrect or the program was not sufficient. To simulate an impact, a future team might find it beneficial to use other programs. The most important components of a program would be the ability to mesh complex models, simulate impacts, and provide detailed results. One program we would suggest is Abaqus because it has nonlinear finite element analysis, which is perfect for honeycomb.

Impact Testing

Our group used a drop test to simulate an impact. The drop tester in the Kaven Hall basement was perfect for simulating a straight on impact. However, this test was not a perfect success because of the friction forces. The poles that guided the drop tester provided friction, which reduced the acceleration. Our group lubricated these poles, but the degree of acceleration reduction was not recorded. We were not absolutely certain the drop tester reached the correct speed. We are convinced the attenuators would still pass peak limits at higher speeds. This method of testing requires an accurate weight and speed to replicate real-life conditions, and we

calculated the weight based on the height and vice versa. If future drop testing takes place, the drop tester should still be guided but with very little friction. Also, future groups should run tests to calibrate the force of impact before testing on attenuators.

If a future team decides against drop testing, a horizontal impact test would be a more realistic simulation. Our group chose to perform a drop test because it was the best way to get high speed camera footage, accelerometer data, and a reliably head-on impact. A rolling vehicle or tracked impact test would create a perfect simulation to a frontal car impact. It would test the strength of the adhesive bonds and better show how a car would react after the impact. For example, our test results for the Foam-Honeycomb-Foam design showed bouncing after impact. A vehicle would react as a car would in a real life situation, either bouncing or absorbing excess energy with frame deformation.

There are a few issues with this method. The first obstacle is finding a vehicle that will get up to speed and stay intact especially with insufficient attenuators. Another problem is getting a track, whether it is an inclined hill with a barrier at the bottom or an automatic track that guides to the correct speed and location. The attenuators are a one-use material, which means that the testing must satisfy all simulation requirements, produce quantitative results for the project group to use, and maintain safety for all people and objects in the surroundings.

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