# **Bass Connections in Energy - Spring 2016** Electric Vehicle Team Technical Report

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# I. Executive Summary

The Bass Connections Electric Vehicle Team designated three main components for the development of sustainable transportation. The most technical of these goals was to design and build a prototype of an urban concept electric vehicle for Duke Electric Vehicles (DEV). The focus of this report is to address the various challenges and goals of the urban concept vehicle.

The urban concept provides an opportunity for DEV to showcase some of its most progressive and inventive engineering feats. The urban concept will be considered a flagship vehicle for its most novel pursuits. In addition to building upon DEV's previous work with carbon fiber composite materials and monocoque designs, the urban concept is undergoing more extensive finite element analysis (FEA) through SolidWorks. Said FEA was utilized to test the suspension in response to various road and driving conditions. It was also used to determine the strength properties of the body of the urban concept. Last, but not least, experimentation was conducted to document the mechanical behavior of the suspension.

After thorough refinement of the body design via aerodynamics simulations, ergonomics studies, and aesthetic aspirations, the urban concept body was finalized and made into molds for production. As this project continues in the 2016-2017 academic year, the students engaged in it will build upon the work our team has done here. This team is proud to have completed many of the most challenging aspects of the urban concept and to have set up the resources for finishing it.

## **II.** Rear Suspension

## A. Prototype Design

The design for the rear suspension was inspired by a trailing link suspension, often found supporting the rear wheel of mountain bikes (see *fig. 1* below). This type of suspension is ideal for mounting bike wheels, such as the ones being used for the Urban Concept v.1, since it supports the wheel axle on both sides and thus reduces the cantilever effect on the axle as it encounters impact forces.



Figure 1: Trailing link suspension on part of a mountain bike frame. The suspension used in the Urban Concept v.1 was adapted from this design, placing the wheel in front of the shock and separating the prismatic truss into two triangular arms (source: http://www.global-trade.com.tw/images/Product/MTB\_suspensio\_10211S.jpg)

A few notable changes were made to the design of this type of suspension to ensure its compatibility with the Urban Concept, as well as its cost-effectiveness and ease of manufacturability. First, since the lower rear face of the monocoque body was the most sturdy location to secure the wheels, the trailing link suspension was modified into a leading link suspension such that the wheels were located in front of the shocks. This necessitated that the base plate, which would be inlaid into the carbon fiber to mount the suspension, would have to be mounted at an angle to the ground. To minimize the risk of delamination, the shock was positioned such that it would be exactly perpendicular to the plate when the car was in its fully-weighted resting state, ensuring that the majority of the force transferred into the body would be in the form of compression rather than shear. The team had also previously agreed that both the front and rear suspensions should have a maximum vertical travel of 1" to avoid collisions between the front tires and the wheel wells. In order to meet both these design requirements when the geometry of the triangle arms and shocks was laid out, the anchor point of the shock and the location of the upper

pin were iteratively adjusted until the shock sat perpendicular to the base plate when a vertical travel of exactly 1" was obtained from the shock's extended resting position. Careful consideration was given to ensure that the car would sink to just above the ground clearance required by the competition when fully loaded, giving at least 1" of suspension travel both up and down and a perpendicular neutral position for the shock.

Consideration was also given to the manufacturability of the suspension. Since the team had little access to or training on welding equipment, the prismatic aluminum tubing design was modified to a double triangle design, allowing the pieces to be plasma cut directly from a sheet of metal. The primary disadvantage that arose from this design was the long, narrow pin that connected the two triangle arms at the top and supported the shock. Since all of the force from the wheel was transmitted into the shock, the upper pin would have to sustain the greatest amount of force, and with a length-to-diameter ratio so large, ran the risk of permanent deformation under load. In addition to finding a high-strength, easy-to-machine steel to use for the pins, an additional pair of thick aluminum collars were bolted to the triangle arms to support the pin and effectively shorten its length.

Other considerations were made for aspects such as ease of assembly; the entire suspension was made to be quickly and easily put together using only a few snap rings and nuts, and simple bushings were used between rotating parts as only a small degree of movement was required. The base plate was machined out of a single piece of metal that spanned both rear wheels to distribute the load across a greater portion of the back wall and to minimize the risk of delamination. The method chosen for mounting the clevises to the base plate proved to be slightly more challenging; the clevises needed to be firmly attached to the base plate, but any holes drilled through the carbon fiber with bolts and nuts protruding from the body would significantly impact the car's aerodynamic efficiency. Tapping the base plate was thought to be potentially problematic, as the team would have run the risk of sealing the threads with resin during the carbon fiber layup, and the position of the clevises relative to the plate would not have been adjustable if the holes were not aligned properly. After the prototype was machined and assembled, however, it was found that the holes were aligned accurately enough such that additional flexibility in alignment was unnecessary. Should the team choose not to bolt the base plate through the body, it should be relatively simple to lay the plate under the innermost layer of carbon fiber (perhaps with an additional 1/4" thick honeycomb surrounding the 1/4" thick plate, to keep the wall a consistent thickness and further decrease the chances of delamination), then re-drill and tap the holes in the base plate by hand after the resin has cured. This would allow the car to keep its external aerodynamic shape and ensure that the clevises are securely fastened to the base plate.

The resulting design is shown in CAD below, and the exploded assembly view, bill of materials, and machining drawings are provided in Appendix A.



Figure 2: CAD rendering of leading link rear suspension for Urban Concept v.1



Figure 3: Exploded assembly of rear suspension in CAD



*Figure 4: Rear suspension mounted in car body with wheels* 



Figure 5: Machined suspension with wheels mounted



Figure 6: Side view of machined suspension

## **B.** Testing

## 1. Experimental Setup

To assess the mechanical behavior of the suspension, our team setup an experiment which involved securely fastening the suspension, statically deflecting it, and dynamically releasing it (see fig. 7, 8, & 9 below). To begin, the base was secured to a table with clamps (see figure 10 below). Next, a system was assembled to apply a measurable force to deflect the suspension. Straps were tied to the base of the table and looped around the edge in order to create an anchor point. The other end of the strap was fed into a tensioner (see figure 11 below). The tensioner was attached to a force gauge, which itself was attached to a mock axle in the suspension where the wheel would otherwise be (see figure 12 below). Lastly, a potentiometer was attached to the mock axle such that displacement could be measured (see figure 13 below).

With the experimental setup complete, a procedure was followed to document the suspension's static and dynamic characteristics. First an arduino data acquisition program was run to begin collecting data from the potentiometer. Then the tensioner was applied to slowly increase force applied to the suspension. After settling on an appropriate maximum force, the force was recorded manually. Finally, the zip-ties connecting the force gauge to the tensioner were rapidly severed to allow the suspension to dynamically respond to a change in force. All of the displacement data was recorded via the arduino program at 100 Hz. The analysis of this data can be found in the next section of this report.



Figure 7: Overview of suspension experimental setup



Figure 8: Right-side view of experimental setup



Figure 9: Left-side view of experimental setup



Figure 10: Close-up view of clamped suspension base



Figure 11: Strap tensioner for force application



Figures 12 & 13: Force gauge and potentiometer for data acquisition

#### 2. Test Data

The analysis aspect of the suspensions testing was mainly focused on understanding how the suspensions respond to varying compression forces at a given setting of its inside pressure chambers. The suspension has two chambers: the upper one was varied from 90 psi, 105 psi and 120 psi while the lower one was varied from 50 psi, 53 psi and 58 psi. While these pressure values could be adjusted depending on the load, our experiment focused on finding some optimal values of internal pressures that could be relevant to the use of our urban concept vehicle, taking into consideration the weight of the whole car and maximum weight of the two-seaters and their luggages. Through this analysis, we will focus on finding the least amount of deflection for the greatest amount of compression that the suspensions can possibly take. Ideally, we're trying to keep the suspensions displacement under 1 inch, for both structural support, damping and safety reasons. Although this suspension testing doesn't directly analyze the optimum damping coefficient that's desired for the suspensions, it certainly reveals how much air pressure we should expect to put in the suspensions. Moreover, how much displacement should be expected on the shock-absorbers when the car hits a bump, carries heavy loads etc., which paves our analysis into further understanding and research of the optimum damping coefficient.





Figure 14: A graph of suspension displacement versus time, taken when the suspension was filled with a 90 psi on its upper chamber, and 50 psi on its lower

chamber. When subjected to different compression forces, the results are as shown above. As expected, the force versus displacement response is steepest when the suspension is compressed the most.

On *Figure 14* above, we can see that at the 24.89 lbf of compression applied to the suspensions, the maximum displacement effect due to that was 2.6496 inches. This being about 2.6 times the desired displacement, it certainly means that the suspensions need to be filled with more air pressure for a much higher damping ratio that would result in less displacement. At this applied force, the suspension constant, which is equivalent to the spring constant, is 9.3939 lbf/in. Compared to spring constants of 78.7490 lbf/in, 36.0477 lbf/in and 18.9548 lbf/in for responses when the applied force was much lesser (see *Figure 14*), it's clear that we need a optimize between a higher spring constant at any given load and the displacement response of the suspensions. So far, we definitely need a suspension that can take more than 24.89 lbf without displacing too much.



Figure 15: A graph of suspension displacement versus time, taken when the suspension was filled with a 105 psi on its upper chamber, and 53 psi on its lower chamber. When subjected to different compression forces, the results are as shown

# above. As expected, the force versus displacement response is steepest when the suspension is compressed the most.

On *Figure 15* above, we can see that at the 35.45 lbf of compression applied on the suspension, the maximum displacement effect due to that was 4.5630 in. This being about 4.6 times the maximum allowable displacement of 1 in, it certainly means that the suspensions need to be filled with more air pressure for a much higher damping ratio that would result in less displacement. At this applied force, the suspension constant, which is equivalent to the spring constant, is 7.7690 lbf/in. Compared to spring constants of 37.1413 lbf/in, and 19.1647 lbf/in for other responses when the applied force was much lesser (see *Figure 15*), it's clear that having more air pressure in the suspension chambers has improved our results, but still the displacement responses are too much for a higher load. We still need to optimize our variables for better damping effects.



Figure 16: A graph of suspension displacement versus time, taken when the suspension was filled with a 120 psi on its upper chamber, and 58 psi on its lower chamber. When subjected to different compression forces, the results are as shown above. As expected, the force versus displacement response is steepest when the suspension is compressed the most.

On Figure 16 above, we can see that at the 47.69 lbf of maximum compression applied on the suspension, the maximum displacement effect due to that was 4.7548 in. This being about 4.8 times the maximum allowable displacement of 1 in, it certainly means that the suspensions need to be improved for a much higher damping ratio that would result in less displacement and the maximum applied load possible. At this applied force, the suspension constant is 10.0299 lbf/in. Compared to spring constants of 73.7295 lbf/in, and 24.8628 lbf/in for other responses when the applied force was much lesser (see Figure 16), this implies that having more air pressure in the suspension chambers has definitely improved our results, but displacement responses are still too much for a higher load. At this point it has become clear that if we pressurize the chamber high enough, we will get to a point where our desired loading capacity is met with a lower displacement response. For now, a car suspension that can't take 47.69 lbf without displacing for more than an inch simply needs to be improved. Pressuring the chambers further is definitely still possible at this point, until the maximum allowable pressures in the chambers becomes our limiting factor. In that case, our report would suggest much more powerful suspensions for the urban concept.



Figure 17: The above graph puts all the graphs in Figure 14, 15 and 16 together for comparison purposes. It's apparent that pressurizing the chambers results into higher

spring constant values, which initially makes the suspensions to respond with a lesser displacement for an applied static load. However, the more the load is applied, the more pressure is required to sustain a relatively lower displacement on the suspensions.

In summary, our observations on the suspension testing reveal that certainly pressuring the upper and lower chambers increases the spring constant, as expected. While this improves the damping ability on the suspensions, an optimal balance between air pressure and applied force needs to be reached in order to determine which values of allowable pressure and applied load will keep our displacement under 1 in. For the given experimental method above, more variations of force may be applied on the suspensions to see how much it can take for the most minimal displacement. Given the limitations of our testing facilities in the workshop, more advanced testing equipments are need in order to subject the suspension to more static and dynamic loads safely. This could certainly not be done safely within our workshop.

#### **C. MATLAB Calculations**

All calculations below were written into a script and solved in MATLAB -- see Appendix B for full code and results.

#### 1. Resting Weight & Cornering

The mass of the car was found using the Mass Properties feature in Solidworks of the full car assembly, including the body, suspension, ballast, wheels, one passenger, and seats. An additional 30lbs were added to the number obtained to approximate the weight of steering, seat belts, and other miscellaneous accessories that were not included in the assembly. A simple force balance using a free body diagram of the car was used to determine the amount of weight carried by each wheel: 68lbs for each front wheel and 74lbs for each rear wheel. Cornering was similarly simplified, and the forces going into each wheel were obtained by using the equation for centripetal force,  $F = \frac{mV^2}{r}$ , and assuming each wheel carried 1/4 of the force with no slippage. The magnitude of the force found, assumed to be acting perpendicularly to the front wheel wells and in shear along the back wall, was 72lbs. These values were later used for conducting a finite element analysis on the suspension and the body in Solidworks.

#### 2. Braking

The forces going into the front and rear suspensions during braking were found using two sets of force-balance equations: one on a free body diagram of the entire car, and the other on a free body diagram of one side of the rear suspension. The first set of equations considered the front two wheels grouped together and the rear two wheels grouped together, and a deceleration in the x-direction was included in the force balance under the assumption that braking would occur purely due to pure static friction. The system of equations obtained were as follows (see Appendix C for diagrams and derivations):

$$\frac{mv_0}{t_b} = \sigma_s(F_y + R_y) \tag{1}$$

$$mg = R_y + F_y \tag{2}$$

$$\sigma_{s}y_{c}(R_{y}+F_{y}) = F_{y}x_{c1} - R_{y} + x_{c2}$$
(3)

where *m* was the total mass of the car, obtained from Solidworks,  $x_{cl}$ ,  $x_{c2}$ , and  $y_c$  were the distances from the COG to the front axle, to the rear axle, and to the ground, respectively, also obtained from Solidworks.  $\sigma_s$  was the coefficient of static friction between the tires and the road, assumed to be 1.2, and  $v_0$  was the running speed of the car, assumed to be 25 mph.

Once  $t_b$ ,  $R_x$ ,  $R_y$ ,  $F_x$ , and  $F_y$  (time to brake, x and y components of the rear wheel forces, and x and y components of the front wheel forces) were obtained in MATLAB, they were plugged into the second set of equations to find the theoretical forces going into the upper and lower pins of the rear suspension:

$$S\cos\theta_s + P_x = R_{x1} \tag{4}$$

$$S\sin\theta_s - P_y = -R_{y1} \tag{5}$$

$$R_{x1}y_r + R_{y1}x_r = S\left(\cos\theta_s y_s - \sin\theta_s x_s\right) \tag{6}$$

where  $\theta_s$  was the angle between the ground and the rear shock (obtained from Solidworks), *S* was the magnitude of the force being transmitted at angle  $\theta_s$  to the upper pin in the suspension,  $P_x$  and  $P_y$  were the x and y components of the force going into the lower pin of the suspension,  $x_{R'}$ ,  $x_{S'}$ ,  $y_{R'}$ , and  $y_S$  were the x and y distances from the lower pin to the axle and the upper pin, respectively (obtained from Solidworks), and  $R_{x1}$  and  $R_{y1}$  were the rear wheel reaction forces found above, divided by two to obtain the effects on one wheel.

Once *S*,  $P_x$ , and  $P_y$  were calculated in MATLAB, the results were plugged into the beam bending stress equation ( $\sigma = \frac{My}{T}$ ) and shear stress equation ( $\tau = \frac{F}{A}$ ) to obtain the maximum stresses in the pins. Since the lower pin experiences less force and is significantly shorter than the upper pin, the upper pin was checked for failure in bending and the lower pin was checked for failure in shear. The results for maximum stress in the entire system, found in the upper pin, and the corresponding FOS for the car in braking were found to be 26688.13 psi and 4.68, respectively.

#### **D. FEA Results**

Once the forces going into the front and rear suspensions were calculated for all three driving conditions -- at rest, braking, and cornering -- their values were plugged into Solidworks FEA to obtain a simulated estimate for the maximum stress in the system. The resulting stress and deformation plots are shown below in *figs. 18-23*, and the maximum stress and FOS found for each driving condition are tabulated below in *Table 1*.

	Maximum Stress (psi)	Factor of Safety	Maximum Deformation (in)
Resting	17016	7.3	0.32
Braking	11402	11.0	8.4e-3

Table 1: Max. stress, deformation, and FOS under 3 driving conditions found using FEA

# 1. Resting Weight



*Figure 18: Stress plot for rear suspension under resting weight. Max stress = 17016 psi* 



Figure 19: Deformation plot for rear suspension under resting weight. Max deformation = 0.32"

# 2. Braking



*Figure 20: Stress plot for rear suspension under braking forces. Max stress = 11402 psi* 



*Figure 21: Deformation plot for rear suspension under braking forces. Max deformation = 8.4e-3 in* 

# 3. Cornering



*Figure 22: Stress plot for rear suspension under cornering forces. Max stress = 40954 psi* 



Figure 23: Deformation plot for rear suspension under cornering forces. Max deflection = 0.13 "

#### III. Carbon Fiber and Honeycomb Testing

In order to obtain accurate results in a finite element analysis of the Urban Concept body, a few properties of the composite material had to be experimentally established and loaded into Solidworks. A mathematical relation was derived with the help of Dr. Knight (see Appendix C) that relates the stresses found in a Solidworks model of the body -- which assumed a completely uniform, homogenous material -- and the theoretical stresses found in the far more complex composite material actually used. This information, along with the yield strength determined during the break testing, provided a factor of safety for the entire car body under various driving conditions.

#### A. Bending test procedure

To evaluate the optimal pairing of honeycomb and carbon fiber for the Urban Concept body, a break test was conducted on eight sandwich panels constructed with different combinations of honeycomb and carbon fiber. Three of the panels had two layers of 1/4" thick honeycomb with two layers of carbon fiber on the outside and two layers in between, another three panels had two layers of 1/4" thick honeycomb with three layers of carbon fiber on the outside and one layer of 1/2" thick honeycomb with two layers of carbon fiber on the outside. Each panel was roughly 6" wide by 12" long, and care was taken to ensure that their resin was fully and properly cured.

For ease of analysis, the test was set up to simulate a two-dimensional, simply-supported beam in bending with a point load in the center. A Tinius Olsen three-point bending system in the Department of Civil Engineering lab was used to conduct the tests, which had a maximum load capacity of 10,000 lbs and a digital output monitor for collecting force vs. displacement data in real time. Because the panels were 6" wide, a thin rectangular piece of steel was used to apply the load in a straight line across the width of the specimen and simulate two dimensional bending conditions. The panels were placed across two simple supports, and the distance between their centerlines was measured and recorded as the span.

As each sample was slowly loaded with increasing force, the force and its corresponding deflection were read off the output monitor and recorded in Excel for plotting and analysis. The first sample was simply observed during the break test to watch its behavior, and as a result, no data was taken. For the first few sample panels after that, data was only collected up until the point of failure, and so the graphs for those panels only include the linear portion of the force vs. deflection curve. The last few panels included the behavior after failure, and the exact point of failure was recorded for all of the tests except one.



Figure 24: Bending test setup before applying load



Figure 25: Fully broken test specimen; failure due to delamination

## **B.** Testing Data

The force and displacement data collected was plotted to find the Young's Modulus for each sample panel, examples of which can be found below in *figs. 26-27* (see Appendix E for all raw data).



*Figure 26: Force vs. deflection plot for two-layer, 1/4" thick honeycomb sample* 



*Figure 27: Force vs. deflection plot for one-layer, 1/2" thick honeycomb sample* 

By taking the slope of the linear portion of the plot and various dimensions of the specimen, the Young's Modulus was calculated using the beam deflection equation:

$$E = \frac{F}{\delta} \frac{L^3}{4bs^3} \tag{7}$$

Additionally, the yield stress was calculated from the yield strength found during the break tests using the normal stress in beam bending equation:

$$\sigma_{cr} = \frac{3}{2} \frac{F_{cr}L}{bs^2} \tag{8}$$

The resulting average values for Young's Modulus and yield stress are summarized in *Table 2* below, and were input as the material properties in Solidworks FEA to obtain stress and deflection data for the car body.

_		-		-
	2x0.25", 1+2-layer CF	2x0.25", 2-layer CF	2x0.25", 3-layer CF	1x0.5", 2-layer CF
Average Young's Modulus (psi)	491212.2	476236.7	429107.9	783449.3
Average yield strength (psi)	3713.5	3393.5	3298.1	6550.0

Table 2: Average Young's Moduli and yield stresses for each type of sandwich panel

## C. FEA results

Using the values for the single-layer, 1/2" thick honeycomb as the input material properties, as well as the estimated forces calculated in Section 2C above, the maximum stress in the uniform material monocoque was found in Solidworks FEA for each of three loading conditions: resting weight, braking, and cornering (see *Table 3* below). *Figs. 28-33* show heat maps for both stress and displacement under each of these conditions.

Maximum Stress<br/>(uniform material) (psi)Maximum Deformation (in)Resting59.01.5e-2Braking58.91.65e-3Cornering145.88.4e-3

Table 3: Average Young's Moduli and yield stresses for each type of sandwich panel

# 1. Resting Weight



*Figure 28: Stress plot for car body under resting weight. Max stress = 59.0 psi* 



*Figure 29: Deformation plot for car body under resting weight. Max deformation = 1.5e-2 in* 

# 2. Braking



*Figure 30: Stress plot for car body in braking forces. Max stress = 58.9 psi* 



Figure 31: Deformation plot for car body in braking forces. Max deformation = 1.65e-3 in

# 3. Cornering



*Figure 32: Stress plot for car body in cornering forces. Max stress = 145.8 psi* 



Figure 33: Deformation plot for car body in cornering forces. Max deformation = 8.4e-3 in

#### **D. MATLAB Calculations**

Using the values for maximum stress found in Solidworks' uniform material model, an approximate theoretical value for the maximum stress in a car body made of the composite material was found. The relation between the two was found by combining the beam bending stress equation with the definition of normal stress:

$$\left[\sigma_{SW} = \frac{Mc}{I}, I = \frac{1}{12}bh^3, c = \frac{h}{2}\right] \to \sigma_{SW} = \frac{6M}{bh^2}$$
(9)

$$\sigma_{actual} = \frac{F}{A_c} = \frac{M}{hbt} \tag{10}$$

Dividing (10) by (9) and cancelling yields the relation:

$$\frac{\sigma_a}{\sigma_{SW}} = \frac{h}{6t} \tag{10}$$

where  $\sigma_{SW}$  is the maximum stress as reported by Solidworks FEA for the uniform material,  $\sigma_{actual}$  is the theoretical actual stress in the composite car body, *h* is the thickness of the honeycomb, and *t* is the thickness of the carbon fiber.

The yield stress was calculated using equation (8) and plugging in the experimental yield strength: the maximum force withstood by the 1/2" thick sample before failure occurred. This value, divided by the maximum stresses found using the above relation, resulted in the factors of safety tabulated below in *Table 4* (see Appendix B for MATLAB script).

Table 4: Maximum stress calculated for composite material in car body and related FOS

	Maximum Stress (composite material) (psi)	Factor of Safety
Resting	289.2	22.6
Braking	288.7	22.7
Cornering	714.7	9.16

#### **IV.** Conclusions and Future Modifications

In summary, the technical report of the Bass Connections Electric Vehicle Team encompasses different aspects of building and testing the prototype. Different testing methods were devised in order to evaluate our engineering decisions on the choice of materials we used to build the body as well as the mechanical systems we designed for the car. Our use of the carbon-fiber honeycomb materials to build the monocoque was validated through a series of strength tests that were conducted on selected samples. The choice between single layers of much thicker honeycomb material versus double layers of thinner honeycomb materials was evaluated during strength testings and compared after a thorough data analysis. The  $\frac{1}{2}$ " thick honeycomb was most resistant to failures that would normally result from delamination compared to the 1/4" thick honeycomb samples. There was a slight benefit to the double-layer 1/4" honeycomb; as the force on the samples increased, only one of the two layers would delaminate at a time -- giving the user some audible feedback that failure was occurring before both layers completely delaminated. However, the significantly higher strength of the single-layer honeycomb would greatly outweigh the benefit of gradual failure in the double-layer honeycomb, and in any case neither of those failures were ultimate, as the carbon fiber had still been stretching without tearing when the tests were ended. Several improvements could be implemented in the future, such as conducting more extensive testing on different number of layers of carbon fiber honeycomb as well as experimenting with many layers of just carbon fiber itself. Moreover, the team will have to determine the best way to inlay the metal mounting plates if a single layer of honeycomb is used instead of two, as the plate will protrude from the surface of the honeycomb and create an uneven wall that is more at risk for delamination. Consideration can be given to adding a layer of 1/4" thick honeycomb with the shape of the metal inlay cut out on top of the 1/2" thick honeycomb for that purpose.

Another important part of the technical report is the FEA data and results. After having decided which layer of honeycomb gave us the strongest and most durable structure, we applied the <sup>1</sup>/<sub>2</sub>" thick layer as the material for our prototype car and conducted an FEA on it in SolidWorks under different driving conditions. Factors of safety were determined according to each driving condition: the prototype car at rest would have a factor of safety of 22.6, 22.7 while braking, and 9.16 while cornering. As expected, the car would be subjected to significantly higher stresses when cornering, almost twice as much as compared to the other situations, as the forces are acting normal to the front wheel well walls, which have small radii of curvature and therefore more locations of high stress concentration. Our analysis methods did not allow us to account for delamination of the composite when shear stress was applied and the software was generally limited in its capabilities; using a more sophisticated

FEA package or conducting further tests on the highly complex composite material would be advisable for obtaining more accurate results in the future.

Furthermore, the suspension testing aspect of the project demonstrated that an optimal level of pressure in both suspension chambers may be reached for a specified amount of load going into the suspension. Our analysis has shown that the current suspensions can be pressurized to much higher values to fit our needs. However, one of the main challenges we experienced during the experiment was that achieving such levels may require the use of a compressor instead of the hand pump used during experimentation. Subjecting such a highly pressurized suspension to static and dynamic loads is extremely dangerous and therefore requires much more sophisticated testing equipment and facilities. Another area of improvement could be testing the suspension on a much more rigid jig frame than the one built for this experiment. The front wheel suspensions would also have to be mounted on this frame with steering and brakes to test rolling conditions in a more realistic environment. Among others, the tests could also include obtain damping coefficient data when the prototype is rolled over a bump, or the actual forces going into the suspension could be taken under various driving conditions to make the FEA simulation more accurate.

Overall, the prototype comes with much promise for success and we look forward to seeing its completion in the coming years. We hope the team will build upon our designs to make the car lighter and more aerodynamic through refinement of the design and material choices in order to directly improve our performance and efficiency -- as this was the team's first Urban Concept design, many parts may have been over-engineered and several changes can be made to cut down on weight. With our distinguished advisors and Duke Electric Vehicle's steady hands, our hopes to paint the town while riding in this electric car are higher than ever.

	8 7 6	5	4	3	2	1
F	Appendix A: Rear Suspension CAD		1		9 (17) (8)	F
_						
E						
D						D
	ITEM NO. PART NUMBER	QTY.	(14)			
	1 Base plate	1				
	2 Triangle arm	4	C	(6)		
	3 Upper pin v2	2		$\bigcirc$	4	~
C	4 Lower pin	2				C
	5 Shock pin	2				
	6 Arm clevis	4				
Н.	7 Shock clevis	2				_
	8 Upper pin collar	4				
	9 Shock absorber	2				
B	11 5/16-18 hex nut	4				R
	12 M12-1.75 hex nut	4				U
	13 6-32 x 1-1/4" socket head screw, fully threaded	20	UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES	D:	DEBURR AND BREAK SHARP DO NOT SCALE D	RAWING REVISION
	14 10-32 x 1/2" socket head screw, fully threaded	24	TOLERANCES: LINEAR: 0.001"		EDGES	
$\vdash$	15 5/16" external snap ring	8	NAME	SIGNATURE DATE	TITLE:	
	16 5/16" flanged sleeve bearing	4	DRAWN		DEV Re	ar Suspension
	17 5/16" nylon sleeve bearing	4	APPV'D		A	nny Ning
A	18 3/8" nylon thrust bearing	4	MFG Q.A	MATERIAL:	DWG NO.	A
				WEIGHT:	Assemb	SHEET I OF 10
	8 7 6	5	4	3	2	1



















#### **Appendix B: MATLAB Script for Force Loads**

% Anny Ning

```
% DEV senior design project
% Standard braking forces
clear; clc
%% Initialize variables
% Assume:
q = 32.2; % ft/s^2; acceleration due to gravity
sigma = 1.2; % coefficient of static friction b/w tires and ground
v0 = 25 * 5280/3600; % mph to fps; running speed of car
% Taken from SolidWorks:
m = (252.77+30)/g; % lbm to slugs, with 30 added lbs; mass of car + driver
xc1 = 41.8/12; % ft; dist from COG to center of front wheels
xc2 = 38.2/12; % ft; dist from COG to center of rear wheels
yc = 15.3/12; %ft; dist from COG to ground
%% System of equations to find braking forces
syms tb Ry Fy
eqn1 = sigma*(Fy+Ry) == m*v0/tb; % force balance in x dir; accel from braking
eqn2 = m*g == Ry + Fy; % force balance in y direction
eqn3 = sigma*yc*(Ry+Fy) == Fy*xc1 - Ry*xc2; % moment balance about COG
solution = solve([eqn1, eqn2, eqn3], [tb, Ry, Fy]);
tbSol = vpa(solution.tb, 6) % seconds; time required to brake
FySol = vpa(solution.Fy, 6); % lbf; force in y dir on both front wheels
RySol = vpa(solution.Ry, 6); % lbf; force in y dir on both rear wheels
FxSol = vpa(sigma*FySol, 6); % lbf; force in x dir on both front wheels
RxSol = vpa(sigma*RySol, 6); % lbf; force in x dir on both rear wheels
%% Braking forces, per wheel
Fx1 = vpa(FxSol/2, 5) % lbf; force in x dir on one front wheel
Fy1 = vpa(FySol/2, 5) % lbf; force in y dir on one front wheel
Rx1 = vpa(RxSol/2, 5) % lbf; force in x dir on one rear wheel
Ry1 = vpa(RySol/2, 5) % lbf; force in y dir on one rear wheel
%% Initialize variables
% Taken from SolidWorks:
thetaS = 47.2; % degrees; angle b/w ground and rear shock
xR = 14.7/12; % in to ft; distance from lower pin to axle in x direction
yR = 1.88/12; % in to ft; distance from lower pin to axle in y direction
xS = 2.67/12; % in to ft; distance from upper pin to axle in x direction
yS = 7.63/12; % in to ft; distance from upper pin to axle in y direction
```

```
%% System of equations to find forces in rear suspension
syms Px Py S
eqn4 = S*cosd(thetaS) + Px == Rx1; % force balance in x direction
eqn5 = S*sind(thetaS) - Py == -Ry1; % force balance in y direction
eqn6 = Rx1*yR + Ry1*xR == S*(cosd(thetaS)*yS-sind(thetaS)*xS); % moment balance
solution2 = solve([eqn4, eqn5, eqn6], [Px, Py, S]);
Px1 = vpa(solution2.Px, 6) % lbf; forces on lower pin in x direction
Py1 = vpa(solution2.Py, 6) % lbf; forces on lower pin in y direction
S1 = vpa(solution2.S, 6) % lbf; force on upper pin at angle theta from ground
%% Max bending stress in upper pin
% Assume:
yieldstrength pin = 125000; % psi; yield strength of 1144 steel from McMaster
% Taken from SolidWorks:
L = 3.937; % in; length of shock pin between support collars
11 = 0.498; % in; length of pin between collar and shock
12 = 0.945; % in; width of shock
D = 0.3125; % in; diameter of pin
I = pi/64*D^4; % moment of inertia, circular cross-section
r = D/2; % radius of pin
M max = (1/2)*S1*(11+12/4); % maximum bending moment in upper pin
max stress upperpin = vpa(M max*r/I, 7) % psi; maximum bending stress in upper
pin
FOS upperpin = vpa(yieldstrength pin/max stress upperpin, 3) % factor of safety
for upper pin
%% Max shear stress in lower pin
% Assume:
shearyieldstrength pin = yieldstrength pin*0.58; % psi; approx. shear yield
strength of 1144 steel
max shearstress lowerpin = vpa(sqrt(Px1^2+Py1^2)/(pi*r^2),6) % psi; maximum
shear stress in lower pin
FOS lowerpin = vpa(shearyieldstrength pin/max shearstress lowerpin,3) % factor
of safety for lower pin
%% Max stress in car body at rest
% Assume:
```

```
s = 0.5; % in; thickness of honeycomb, 1 layer of 0.5" thick
t = 0.017; % in; thickness of carbon fiber fabric
% Taken from SolidWorks FEA, for uniform material:
max stress SWcar resting = 59; % psi; max stress in body at rest
max stress SWcar braking = 58.9; % psi; max stress in body during braking
max stress SWcar cornering = 145.8; % psi; max stress in body during cornering
% Taken from sample testing:
yieldstrength car = 6550; % psi; critical stress for 1/2" honeycomb, from
testing data
max stress car resting = vpa(max stress SWcar resting*s/(6*t),6) % psi;
theoretical stress at rest
FOS car resting = vpa(yieldstrength car/max stress car resting,3) % factor of
safety at rest
max stress car braking = vpa(max stress SWcar braking*s/(6*t),6) % psi;
theoretical stress while braking
FOS car braking = vpa(yieldstrength car/max stress car braking,3) % factor of
safety while braking
max stress car cornering = vpa(max stress SWcar cornering*s/(6*t),6) % psi;
theoretical stress while cornering
FOS car cornering = vpa(yieldstrength car/max stress car cornering,3) % factor
of safety while cornering
```

# MATLAB output:

tbSol =	FOS_upperpin =
0.94893	4.68
Fx1 =	<pre>max_shearstress_lowerpin =</pre>
119.95	2919.77
Fyl =	FOS_lowerpin =
99.959	24.8
Rx1 =	<pre>max_stress_car_resting =</pre>
49.711	289.216
Ry1 =	FOS_car_resting =
41.426	22.6
D-1	
PXI =	max_stress_car_braking =
-98.2701	288.725
Pv1 =	FOS car braking =
201.231	22.1
S1 =	<pre>max_stress_car_cornering =</pre>
217.798	714.706
<pre>max_stress_upperpin =</pre>	<pre>FOS_car_cornering =</pre>
26688.13	9.16

### **Appendix C: MATLAB Script for Suspension Tests**

```
%% Abraham, Anny, Charlie
%% Import data from text file.
% Script for importing data from the following text file:
2
8
      /Users/abrahamnghwani/Documents/MATLAB/Suspension data.csv
2
% To extend the code to different selected data or a different text file,
% generate a function instead of a script.
% Auto-generated by MATLAB on 2016/05/03 23:13:03
%% Initialize variables.
filename = '/Users/abrahamnghwani/Documents/MATLAB/Suspension data.csv';
delimiter = ',';
%% Read columns of data as strings:
% For more information, see the TEXTSCAN documentation.
%% Open the text file.
fileID = fopen(filename, 'r');
%% Read columns of data according to format string.
% This call is based on the structure of the file used to generate this
% code. If an error occurs for a different file, try regenerating the code
% from the Import Tool.
dataArray = textscan(fileID, formatSpec, 'Delimiter', delimiter,
'ReturnOnError', false);
%% Close the text file.
fclose(fileID);
%% Convert the contents of columns containing numeric strings to numbers.
% Replace non-numeric strings with NaN.
raw = repmat({''},length(dataArray{1}),length(dataArray)-1);
for col=1:length(dataArray)-1
     raw(1:length(dataArray{col}), col) = dataArray{col};
end
numericData = NaN(size(dataArray{1},1), size(dataArray,2));
for col=[1,2,3,4,5,6,7,8,9,10]
      % Converts strings in the input cell array to numbers. Replaced
non-numeric
      % strings with NaN.
```

```
rawData = dataArray{col};
      for row=1:size(rawData, 1);
      % Create a regular expression to detect and remove non-numeric prefixes
and
      % suffixes.
      regexstr =
'(?<prefix>.*?)(?<numbers>([-]*(\d+[\,]*)+[\.]{0,1}\d*[eEdD]{0,1}[-+]*\d*[i]{0,
1}) | ([-]*(\d+[\,]*)*[\.]{1,1}\d+[eEdD]{0,1}[-+]*\d*[i]{0,1})) (?<suffix>.*)';
      try
            result = regexp(rawData{row}, regexstr, 'names');
            numbers = result.numbers;
            % Detected commas in non-thousand locations.
            invalidThousandsSeparator = false;
            if any(numbers==',');
            thousandsRegExp = '^{d+?}(, d{3}) * . \{0, 1\} d*$';
            if isempty(regexp(thousandsRegExp, ',', 'once'));
                  numbers = NaN;
                  invalidThousandsSeparator = true;
            end
            end
            % Convert numeric strings to numbers.
            if ~invalidThousandsSeparator;
            numbers = textscan(strrep(numbers, ',', ''), '%f');
            numericData(row, col) = numbers{1};
            raw{row, col} = numbers{1};
            end
      catch me
      end
      end
end
%% Replace non-numeric cells with 0.0
R = cellfun(@(x) (~isnumeric(x) && ~islogical(x)) || isnan(x),raw); % Find
non-numeric cells
raw(R) = {0.0}; % Replace non-numeric cells
%% Create output variable
Suspensiondata = cell2mat(raw);
%% Clear temporary variables
clearvars filename delimiter formatSpec fileID dataArray ans raw col
numericData rawData row regexstr result numbers invalidThousandsSeparator
thousandsRegExp me R;
%% Creating the Variables
Suspensiondata([1 2 3],:)=[];
```

```
y1 = 0.007722.*Suspensiondata(:,1);
y2 = 0.007722.*Suspensiondata(:,2);
y3 = 0.007722.*Suspensiondata(:,3);
y4 = 0.007722.*Suspensiondata(:,4);
y5 = 0.007722.*Suspensiondata(:,5);
y6 = 0.007722.*Suspensiondata(:,6);
y7 = 0.007722.*Suspensiondata(:,7);
y8 = 0.007722.*Suspensiondata(:,8);
y9 = 0.007722.*Suspensiondata(:,9);
y10 = 0.007722.*Suspensiondata(:,10);
t1 = linspace(0,58.98,5898);
% Converting the applied force into a direct force applied at right angle
force = [25.4 36.7 41.1 43.1
                                         41.7 51.5 61.4 61.3 71.4
82.6];
dir force = force.*tan(13.09);
% Creating the data plots for each trial
figure(1);clf
plot(t1,y1,'k')
hold on
plot(t1, y2, 'k-.')
plot(t1,y3,'k-*')
plot(t1,y4,'k-o')
plot(t1,y5,'b-')
plot(t1,y6,'b-.')
plot(t1,y7,'b-o')
plot(t1,y8,'g-')
plot(t1, y9, 'g-.')
plot(t1,y10,'g-*')
title ('A comparative plot of Displacement versus Time for all trials')
ylabel('Displacement (in)')
xlabel('Time (s)')
legend('Displacement at 90/50 psi with 14.67 lbf', 'Displacement at 90/50 psi
with 21.19 lbf', 'Displacement at 90/50 psi with 23.73 lbf', 'Displacement at
90/50 psi with 24.89 lbf', 'Displacement at 105/53 psi with 24.08
lbf', 'Displacement at 105/53 psi with 29.74 lbf', 'Displacement at 105/53 psi
with 35.45 lbf', 'Displacement at 120/58 psi with 35.39 lbf', 'Displacement at
120/58 psi with 41.23 lbf', 'Displacement at 120/58 psi with 47.69
lbf','Location','Southeast')
```

```
plot(t1,y1,'k')
```

figure(2);clf

hold on plot(t1, y2, 'b-.') plot(t1, y3, 'q-\*') plot(t1,y4,'c-o') title('Displacement Vs Time for a 90/50 psi suspension, compressed by 14.67 lbf, 21.19 lbf, 23.73 lbf, and 24.89 lbf') ylabel('Displacement (in)') xlabel('Time (s)') legend('Displacement at 90/50 psi with 14.67 lbf lbf','Displacement at 90/50 psi with 21.19 lbf lbf', 'Displacement at 90/50 psi with 23.73 lbf', 'Displacement at 90/50 psi with 24.89 lbf', 'Location', 'Southwest') figure(3);clf plot(t1, y5, 'r') hold on plot(t1,y6,'b-.') plot(t1,y7,'g-\*') title('Displacement Vs Time for a 105/53 psi suspension, compressed by a 24.08 lbf, 29.74 lbf, 35.45 lbf') ylabel('Displacement (in)') xlabel('Time (s)') legend('Displacement at 105/53 psi with 24.08 lbf', 'Displacement at 105/53 psi with 29.74 lbf', 'Displacement at 105/53 psi with 35.45 lbf','Location','Southwest') figure(4);clf plot(t1, y8, 'm') hold on plot(t1, y9, 'r-.') plot(t1,y10,'g-\*') title('Displacement Vs Time for a 120/58 psi suspension, compressed by a 35.39 lbf,41.23 lbf,47.69 lbf') ylabel('Displacement (in)') xlabel('Time (s)') legend('Displacement at 120/58 psi with 35.39 lbf', 'Displacement at 120/58 psi with 41.23 lbf', 'Displacement at 120/58 psi with 47.69 lbf','Location','Southwest')

**Appendix D: Derivation of Equations** 

& Brailing Force Fx+ Rx., Fr: 6sty, Rx · Ker 8s (Fy+Ry) = 12 Fy = 0 = Ky+ Fy - mg > mg = Ky+ Fy +) EMase=0 = - Ry Xez - Rx. ye - Frye + => (Rx + Fx) ye = Fy Xc1 - Ry Xc2 Solve For: ty, Ry Fy => bige (Ry+Fy) = FyXe, -· Assuming . 65= 1.2 , Vo = 25 mph = From Solid worlds. M= 25277 12m (add 30 125 For extra chill ye = 15.3. in. , Xe1 = 41.8in, Xe2 = . 38.2 in . 1 MATLAD: Per wheel, [Fx1 = 120.0 16F, Fy1 = 99.96 166, R.x1 = 49.7 & Pear suspension Forces . 95 YA (i) =  $2F_{x=0} = S_{x+}P_{x} - R_{x} \Rightarrow S_{cos}O_{s} + P_{x} = R_{x}$ (i) +1  $2F_{y=0} = R_{y_{1}} + S_{y} - R_{y} \Rightarrow S_{sin}O_{s} - R_{y} = -R_{y_{1}}$ (ii) (1)  $2M_{y=0} = R_{x_{1}}y_{a} + R_{y_{1}}x_{a} - S_{cos}O_{s}y_{s} + S_{sin}O_{s}x_{s} \Rightarrow R_{x_{1}}y_{a} + R_{y_{1}}x_{a}$ · Solve for : · Assuming:  $R_{K1}, R_{y1}$  as above  $x_p = (4.7 \text{ in}, y_p = 1.88 \text{ in}, x_s = 2.67 \text{ in}, y_s = 0$   $\Theta_s = 47.2^\circ$  (assuming in neutral position) S. P.x, Ry. ·MATLANS: Per while, TPx1 = -982716F, Py1=2012 16F, S1=217.8 16F

$M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{12}{7} = \frac{1}{7} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7}$ $M_{max} = \frac{R_{y}A_{1} + \frac{1}{2}(A_{2}, \frac{1}{2})}{R_{max}} = \frac{1}{7}$		
$M_{max} = \frac{S_{ij}}{A_{ij}} + \frac{1}{2} \frac{S_{ij}}{A_{ij}} + \frac{1}{2} \frac{S_{ij}}{A_{ij}} = 0$ $M_{max} = \frac{1}{A_{ij}} + \frac{1}{A_{ij}} + \frac{1}{A_{ij}} + \frac{1}{A_{ij}} + \frac{1}{A_{ij}} = \frac{1}{A_{ij}} = \frac{1}{2} \frac{S_{ij}}{A_{ij}} = S_{i$	Upper shock pin stresses	
$M_{max} = Ay_{11} + \frac{1}{2} A_{12}$ $M_{max} = A_{12}$ $A_{13} = A_{13}$ $A_{13} =$	A State A ANT AT AT AT ANT ANT ANT ANT ANT ANT	$y = sl_2 = S_{1,1}$
$\frac{1}{11 + 1 - 1} = \frac{1}{11 + 1} = $	L. M. G. M. M. M. Aig = Arg	; =A;
*Solve for: Emax in beam $Assuming: L = 3.437 in, l. = 1.448 in, l_2 = 0.445 in.$ S = 5/2L D > 0.3125 in $M_{max} = Ayl_1 + \frac{1}{2}(A_3, \frac{1}{2}L_4)$ $M_{max} = Ayl_1 + \frac{1}{2}(A_3, \frac{1}{2}L_4)$ $= Ay(L_1 + \frac{1}{2}L_2)$ $T = \frac{T}{24}D^4$ $T = \frac{1}{2}D$ $A_1 + \frac{1}{2}S_1(L_1 + \frac{1}{2}L_2)$ $T = \frac{T}{2}D$	Any Right Any Right => 2Ay	.* <u>S</u>
Solve For: Simer in been Assiming: $L = 3.437 \text{ in}$ , $l = 1.498 \text{ in}$ , $l_2 = 0.445 \text{ in}$ $S = 5.7 l_2$ D = 0.3125  in $M_{max} = Ayl_1 + \frac{1}{2}(A_2, \frac{1}{2})l_2)$ $M_{max} = Ayl_1 + \frac{1}{2}(A_2, \frac{1}{2})l_2)$ $M_{max} = Ay(l_1 + \frac{1}{2})l_2$ $T = \frac{M_{max}}{T}$ $T = \frac{T_1}{T}$ $T = \frac{T_1}$		 
$\begin{array}{c} Assuming : L = 3.437 \text{ in}, \ l = 1.448 \text{ in}, \ l_{2} = 0.445 \text{ in} \\ S = 5.7 \text{ le} \\ \Omega = 0.3126 \text{ in} \\ \\ H \\ H \\ H \\ M \\ M \\ M \\ M \\ M \\ M \\$	· Solve for: Emar in beam	 
$S = S/g_{L}$ $D = 0.3125 \text{ in}$ $= (1^{-}) - NA$ $f_{y} = M_{aux} = A_{y}l_{1} + \frac{1}{2}(A_{y}, \frac{1}{2}J_{z})$ $S_{max} = \frac{M_{aux}}{T}$ $= A_{y}(J_{1} + \frac{1}{2}J_{z})$ $T = \frac{T}{24}D^{4}$ $T = \frac{1}{2}O$ $M_{Amax} = \frac{M_{y}l_{1} + \frac{1}{2}J_{z}}{T^{2}}$ $T = \frac{1}{2}O$ $M_{Amax} = \frac{M_{y}l_{1} + \frac{1}{2}J_{z}}{T^{2}}$	· Assiming: L= 3,937 in , l= 1.498 in l= 0.945 in	
$= \underbrace{\int \frac{1}{2} - \frac{1}{2} $	D= 0.3125 in	
$= \underbrace{M_{\text{max}}}_{\text{M}} = \underbrace{A_y l_1 + \frac{1}{2} (A_y, \frac{1}{2} J_z)}_{\text{M} \text{max}} = \underbrace{M_y l_1 + \frac{1}{2} (A_y, \frac{1}{2} J_z)}_{\text{M} \text{max}} = \underbrace{M_y (J_1 + \frac{1}{2} J_z)}_{\text{M} \text{max}} = \underbrace{M_y (J_1 + \frac{1}{2} J_z)}_{\text{M} \text{max}} = \underbrace{T_1}_{\text{M} \text{M} \text{max}} = \underbrace{T_2}_{\text{M} \text{M} \text{M} \text{max}} = \underbrace{T_2}_{\text{M} \text{M} \text{M} \text{M} \text{max}} = \underbrace{T_2}_{\text{M} \text{M} \text{M} \text{M} \text{M} \text{M} \text{M} \text{M}$		
$M_{\text{max}} = Ayl_1 + \frac{1}{2}(A_2, \frac{1}{2}J_n) \qquad \qquad$		
$M_{max} = Ayl_1 + \frac{1}{2}(A_2, \frac{1}{2}J_a)$ $= Ay(l_1 + \frac{1}{2}J_a)$ $T = \frac{T_1}{64}D^4$	ty Moure C.	
$M_{\text{max}} = \frac{1}{2} \sum_{i=1}^{n} \left( l_{i} + \frac{1}{2} l_{a} \right) = \frac{1}{2} \sum_{i=1}^{n} \left( l_{i} + \frac{1}{2} l_{a} \right) = \frac{1}{2} O$ $= M_{\text{M}} = \frac{1}{2} O$ $= M_{\text{M}} = \frac{1}{2} O$	$M_{max} = Ayl_1 + z(Ay, z) = 0$ Mox - 7.	
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# Appendix E: Sample Testing Data

			Honeycomb												
	# Outer	Width	thickness &	Thickness	Recorded	Span	Recorded	Deflection	Force	Breaking strength	Yield strength	Force/ Deflection			
Sample	CF layers	(b, in)	# layers	(s, in)	span (cm)	(L, in)	deflection (mm)	(δ, in)	(F, lb)	(F_cr, lb)	(psi)	Slope (F/δ)	L^3/(4bs^3)	E = (FL^3)/(4δbs^3)	Notes
1	3	6	2x0.25in	0.5	23.2	9.13	5.7	0.224	230	283.6	2590.4	1024.9	254.0	260332.7	
2	2	6	2x0.25in	0.5	24.0	9.45	0.2	0.008	10	282.5	2669.3	1536.8	281.2	432144.9	
							0.7	0.028	40						
							1.6	0.063	100						
							2.1	0.083	130						Only took data up to
							2.7	0.106	160						breaking strength
							3.3	0.130	200						
							3.8	0.150	230						
_	-						4.5	0.177	260						
3	3	6	2x0.25in	0.5	24.0	9.45	0.5	0.020	20	337.2	3186.1	2077.6	281.2	584216.7	
							1.1	0.043	60						
							1.6	0.063	100						
							2.2	0.087	150						Only took data up to
							2.8	0.110	200						breaking strength
							2.5	0.150	240						
							3.0	0.150	200						
4	3	6	2x0 25in	0.5	24.0	9.45	4.5	0.105	20	435.8	4117.8	1574.6	281.2	442774 2	
-	5	Ū	2x0.25	0.5	24.0	5.45	1.2	0.047	50	400.0	4117.0	1574.0	201.2	112//112	
							2.1	0.083	100						
							2.7	0.106	140						
							3.3	0.130	180						
							3.8	0.150	230						
							5.1	0.201	167						
							5.7	0.224	160						
							6.4	0.252	150						
							7.0	0.276	140						
							7.8	0.307	124						
							8.7	0.343	120						
							9.4	0.370	116						
							10.0	0.394	108						
							11.0	0.431	105						
							11.5	0.453	100						
							12.6	0.496	94						
							13.4	0.528	85						
							15.0	0.591	88						
							15.9	0.626	85						
							17.0	0.669	80						
							18.0	0.709	80						
							19.0	0.748	83						
							20.0	0.787	00						
							20.9	0.825	90						
							21.3	0.898	91						
							22.0	0.937	90						
							25.5	1.004	88						
							26.6	1.047	90						
							28.5	1.122	90						
5	2	6	2x0.25in	0.5	24.0	9.45	0.5	0.020	10	435.8	4117.8	1850.4	281.2	520328.5	
							1.2	0.047	40						

							1.7 2.2 2.7 3.1 3.6 4.3 4.7 5.2 5.8 6.2 6.8	0.067 0.087 0.106 0.122 0.142 0.169 0.185 0.205 0.228 0.224 0.268	80 110 150 220 270 300 330 380 400 435						Only took data up to breaking strength
6	2 and 1	6	2x0.25in	0.5	25.2	9.92	0.5 1.1 1.8 2.4 2.9 3.4 4.1 4.7 5.4 5.9 6.5 7.0 10.3 11.0 11.8 12.3 12.9 13.8 14.7 15.3 16.1 17.0 18.0 19.0	0.020 0.043 0.071 0.094 0.114 0.134 0.161 0.185 0.213 0.256 0.276 0.276 0.406 0.433 0.465 0.484 0.508 0.543 0.579 0.602 0.634 0.669 0.709 0.748	20 40 80 101 140 170 220 250 290 320 350 360 147 149 149 146 147 147 148 100 97 97 97	374.3	3713.5	1509.0	325.5	491212.2	Improperly laid outer CF
7	2	6	1x0.5in	0.5	25.2	9.92	0.5 0.9 1.8 2.3 2.7 3.2 3.6 4.1 4.5 5.0 5.4 5.9 6.6 7.1 7.7 8.2 8.8 9.2	0.020 0.035 0.071 0.091 0.106 0.126 0.142 0.161 0.177 0.213 0.260 0.280 0.280 0.303 0.323 0.346 0.362	15 25 50 80 110 150 230 270 310 350 350 350 450 450 490 505 530 540 550	didn't collect max value so check sample 8 results		2214.7	325.5	720932.9	kink in side

							-				-				
							9.9	0.390	290						
							10.8	0.425	290						
							11.4	0.449	290						
							12.6	0.496	290						
							13.3	0.524	280						
							14.0	0.551	281						
							14.8	0.583	282						
							15.6	0.614	278						
							16.7	0.657	276						
							17.6	0.693	263						
							18.6	0.732	250						
							19.6	0.772	251						
							20.6	0.811	248						
							21.4	0.843	245						
							22.5	0.886	242						
							22.5	0.925	240						
8	2	6	2x0.5in	0.5	25.2	9.92	0.3	0.012	20	660.2	6550.0	2598.8	325.5	845965 7	
-	_	-					0.8	0.031	50					0.00000	
							1.2	0.047	90						
							1.6	0.063	130						
							2.1	0.083	180						
							2.5	0.098	220						
							2.9	0.114	270						
							3.5	0.138	320						
							3.8	0.150	370						
							4.6	0.181	430						
							5.1	0.201	490						
							5.5	0.217	530						
							6.6	0.260	630						
							73	0.287	190						
							7.9	0.311	220						
							85	0 335	228						
							9.4	0.370	225						
							10.1	0.398	228						
							11.0	0.433	234						
							11.0	0.457	237						
							12.6	0.496	227						
							13.5	0.531	227						
							14.5	0.571	224						
							15.2	0.598	216						
							16.0	0.630	216						
							17.2	0.677	210						
							18.3	0.720	202						
							19.0	0.748	200						
							19.8	0.780	200						
							20.7	0.815	200						
							21.6	0.850	201						
							22.6	0.890	199						
							23.8	0.937	197						
							24.8	0.976	195						
							25.8	1.016	190						
							26.8	1.055	190						





	2x0.25",	2x0.25",	2x0.25",	1x0.5",
	1+2-layer CF	2-layer CF	3-layer CF	2-layer CF
Voungle	491212.2	432144.9	260332.7	720932.9
Moduli (pci)		520328.5	584216.7	845965.7
woduli (psi)			442774.2	
Average:	491212.2	476236.7	429107.9	783449.3
Vield	3713.5	2669.3	2590.4	6550.0
strongths (psi)		4117.8	3186.1	
strengtris (psi)			4117.8	
Average:	3713.5	3393.5	3298.1	6550.0