BIAXIAL & TWIST TESTING OF COMPOSITE CARBON-FIBER SANDWICH PANELS FOR AUTOMOTIVE RACING VEHICLES

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Approval Page

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Acknowledgements

This project was made possible only through the generous donation of time and money by Pratt & Miller Engineering. I would like to thank Grant Browning of Pratt & Miller for his invaluable help with the design and manufacture of the fixture and his knowledge of the twist test mechanics. I would also like to thank my advisor, Prof. Blair London for all of his assistance in keeping this project on schedule and his daily advice and insight.

Abstract

Composite sandwich panels were constructed with 4-ply plain weave carbon-fiber/epoxy face sheets in the $0^{\circ}/45^{\circ}/0^{\circ}/45^{\circ}$ orientation and 1/8th inch Nomex honeycomb core. The panels were cut into 5-inch square test plates for mechanical testing. All testing was done on a fixture designed and fabricated by Pratt & Miller Engineering and installed on an Instron testing system at Cal Poly. The twist test was performed by supporting diagonal corners of the plate while simultaneously loading the opposite two corners at a crosshead rate of .06 in/min (ASTM 3044-94R11). Out of 10 panels tested, six were tested longitudinally, with the L direction of the honeycomb parallel to the front of the test plate, and four were tested in the transverse orientation, with the L direction of the honeycomb perpendicular to the front of the plate. The average compliance for the longitudinal loading was 1.303 mil/lb, and the transverse loading was 1.394 mil/lb. The panels failed with a combination core shear failure and face-to-core debonding. The anisotropic nature of the honeycomb core supports the difference in the compliance of the panels; however the complex loading of the twist test results the difference being not statistically significant. The biaxial bending tests involved supporting the composite plate on each corner and loading it in the center. In 5 tested plates, the measured average compliance was .4363 mil/lb with face-to-core debonding being the primary failure mode. The results of these tests will be used to improve existing FEA models for the performance of racing vehicle composite panels.

Key Words: Materials Engineering, Composite, Sandwich, Carbon Fiber, Automotive, Racing, Finite Element Analysis, Twist, Biaxial Bending, Compliance

1. Background

Pratt & Miller Engineering (New Hudson, MI) uses composite sandwich structures in many of their automotive racing vehicles including the Corvette C6-R. As a high performance racing vehicle, the option to save weight anywhere on the structure while maintaining performance is highly attractive. In stiffness-critical components such as the rear spoiler, twisting stiffness and biaxial bending is especially important due to the effect these components have on the handling and speed of the racecar and the mounting configuration of the components. For the design of these components, finite element analysis (FEA) is usually used instead of experimental testing to reduce cost and speed of design to production (Figure 1). The accuracy of the FEA model therefore is important to ensure that the part is properly designed. In order to minimize weight and maintain high performance, designers need an accurate model that has been verified by experimental testing. Performing twist and bi-axial bend tests will provide quantitative data to verify the FEA model for this panel construction, and ensure that the panels are adequately designed for the in-service loads they will experience.

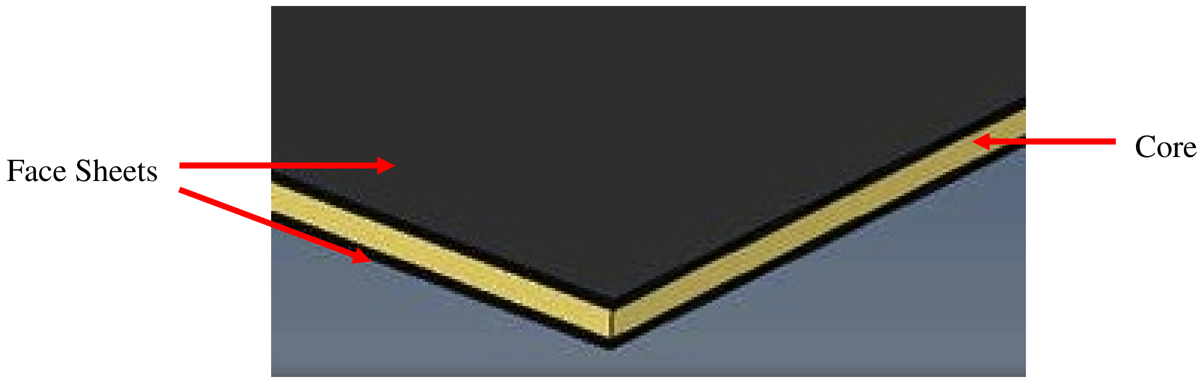

Figure 1: Schematic of the Finite Element Analysis (FEA) model of the test sandwich panel. Face sheets consist of 4 plies of carbon fiber-epoxy on each side with a Nomex honeycomb core.

1.1. Application Background

From the beginning of sport as competition, the push towards finding an advantage over the opponent has led to technical breakthroughs. The use of composite materials in automotive racing is no exception. Composite materials have helped produce the lightest and fastest cars ever made; however, the pressure to reduce weight is continuous. One of the first, and most famous applications of fiber-reinforced composites in automobiles was the fiberglass and epoxy body panels of the Chevrolet Corvette in 1953 (Figure 2).

Figure 2: The 1953 Chevrolet Corvette was one of the first production cars to use composite technology for body panels.[1]

The large scale application of composites in production cars has been limited because of the associated cost with fundamentally changing the way cars are made, and the longer time needed to produce composite parts.[i] However, in auto racing, composite materials are standard. The unique performance, customization and construction time put into each vehicle allows insight into the performance benefits of composite materials in automotive applications.

Corvette Racing, an American Le Mans Series (ALMS) racing team is the most successful team in the history of series. They hold a record 73 class wins and have finished first and second 50 times. Since 1999, Pratt & Miller

Engineering has been in charge of building, maintaining and managing the race team for the ALMS, leading them to eight consecutive manufacturer and team championships in their grand touring (GT1) class. The Corvette Racing team has won 72 times in 102 races, including an overall win in the 2001 Daytona 24-hour race and five GT class titles in the 24 Hours of Le Mans.[ii]

Pratt & Miller designs, constructs and maintains the Corvette C6.R, below, for Corvette Racing. With races lasting up to 24 hours, the vehicle's weight has a great effect on the race performance and therefore the finishing place. The extensive use of composite sandwich panels in the C6.R means that accurate modeling of the loads experienced is essential to the safety and performance of the car. Riding on the "cutting edge" of performance produces faster cars, but never at the cost of safety. The C6.R utilizes composite panels for the front splitter, rear spoiler and most body panel components (Figure 3). The front splitter acts as an air damn, creating an area of high pressure above and low pressure below, creating down force for the front of the car while ensuring that the engine has a constant supply of air for highest performance. The rear spoiler works like an inverted wing, producing an area above of high pressure and low pressure below. In the ALMS, drivers regularly attain speeds of over 150 mph and can average over 100 mph for an entire 12-hour race, creating large, sustained forces on the splitter and spoiler. Body panels protect the driver and also distribute the stresses from the frame around the body. For this component, the strength of the panels is of primary importance, however the extensive use of the panels means that the weight of each panel is compounded numerous times.

Figure 3: The Corvette C6-R, designed and built by Pratt & Miller utilizes sandwich construction composite panels for the front splitter, rear spoiler and body panels.[2]

1.2. Broader Impacts[iii]

Accurate modeling of the twist and biaxial bend responses of Pratt & Miller's carbon fiber sandwich panels will produce greater weight savings and improved safety in the Corvette C6.R. The application of lighter panels could be applied to many different transportation sectors. In the aircraft industry, reducing the weight of composite sandwich panels used could reduce emissions and improve gas mileage while maintaining high levels of safety. The technology transfer from high performance automobiles to average passenger automobiles also occurs. This is evident in the crossover between the Corvette C6-R, and the Corvette Z06 on which it is based. Because Corvette Racing participated in the Grand Touring class in the ALMS, the cars are more closely based on production models than other racing classes. For high-performance cars, accurate design models mean higher performance and more success on the track; however, for the majority of consumers, this equates to greater safety through high-strength structural materials and higher gas mileage and less greenhouse emissions though lightweight constructions.

1.3. Fiber-Reinforced Composites

A fiber-reinforced composite is defined as having two macroscopic phases that maintain a distinct interface between them. Most commonly this takes the form of strong and stiff fibers that carry nearly the entire imposed load and are embedded into a matrix that provides form, protection and transfers the load to the fibers.[iv]

1.3.1. Carbon Fibers

Known for their exceptional stiffness to weight and strength to weight ratios, carbon fibers have become common knowledge as a result of increased use in high-performance sporting goods. The mechanical properties of these fibers are a direct result of the atomic structure of the fiber itself. Oriented along the axis of the fiber, strong carbon-carbon covalent bonds present in the basal planes of the graphite structure give carbon fibers stiffness and strength (Figure 4). The orientation of the planes allow the weak Van der Waal's forces between planes to be avoided.

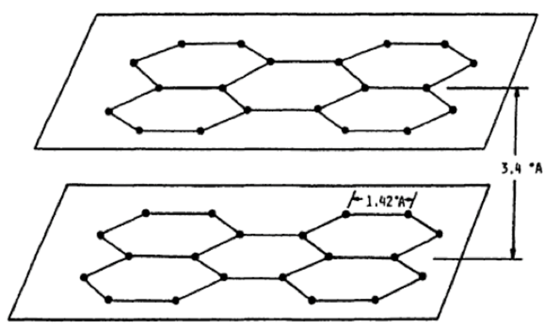

Figure 4: The structure of graphite and the orientation of the basal planes give carbon fiber their strength and high stiffness along the fiber axis.[5]

The most common carbon fiber precursor is polyacrylonitrile (PAN). The PAN molecule contains polar CN side groups that are randomly oriented due to the folding of the polymer chain. The aligning of these highly polar CN groups is important to the conversion to carbon fiber. To produce carbon fibers, PAN is first wet spun into filaments and then stretched to orient the polymer's molecular along the longitudinal axis of the filament and ensure that most of

the CN groups are on the same side of the chain. After being stretched, the filament is heated for several hours in air between 200-300°C which allows the CN side groups to combine with neighboring chains and produce a stable ladder molecular structure (Figure 5). Next, the filaments are carbonized in an inert atmosphere between 1000-2000°C to eliminate the oxygen and nitrogen atoms. Finally, the filaments are graphitized above 2000°C to allow the structure to become more ordered and allow the proper orientation of the graphite planes with the longitudinal axis of the fibers as was seen previously.[v]

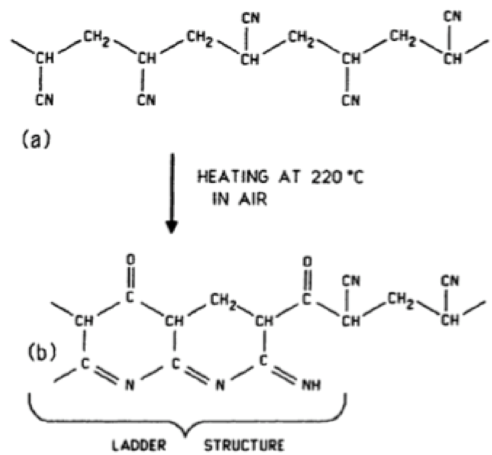

Figure 5: The heating of PAN fiber in air transforms the molecule to a strong ladder structure.[5]

The carbon fiber used at Pratt & Miller has been woven from filament form into a carbon fiber fabric. Plain weave is the most common, with fibers running perpendicular to each other alternating over and under. This gives the final composite component better performance to general stresses than unidirectionally oriented fibers with each direction of fiber taking the loads applied to the weak transverse direction of the other. These plain weave layers are then stacked onto one another creating a laminate structure that is stronger in more directions and able to deal with transverse stresses more easily. These laminates are then stacked into a laminated facesheet for the sandwich composite.

1.3.2. Epoxy Matrix

In order for a composite structure to take full advantage of the exceptional mechanical properties of carbon fiber, the matrix must keep the fibers aligned and effectively transmit the majority of the load to the fibers. This means that an effective matrix must have a strong bond with the fibers, protect the fibers from damage, and keep them aligned in their intended direction. Epoxy resins are two part polymer systems: the resin, which contains epoxide groups, and a reactive curing agent. The two are mixed together before combining with the fibers while the epoxy is in a liquid state. As the hardening reaction of the epoxy progresses, crosslinks between molecular chains are formed, creating a three-dimensional network structure. The matrix is constructed on a molecular scale around the fibers as the reaction progresses, giving epoxy resins excellent adhesion to the fibers. In addition, epoxy matrices have low shrinkage during cure, important in keeping the fibers aligned as intended.[vi]

1.3.3. Lightweight Cores

The unique geometry of the honeycomb (Figure 6) makes it the one of the easiest ways to add thickness to a sandwich structure while adding as little weight as possible. The honeycomb has been observed as a natural engineering marvel for thousands of years. Greek mathematician Euclid found that the hexagon shape was the most efficient use of building space and materials. The hexagonal cell shape minimizes surface area while maximizing volume, giving the largest volume with the smallest amount of material.

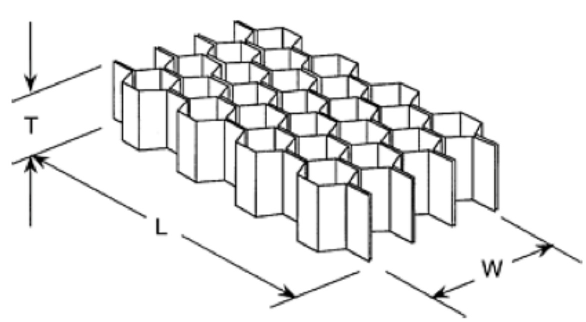

Figure 6: The conventional dimensions of the honeycomb structure.[9]

However, developing an efficient technique to manufacture honeycombs for structural applications was slow in being developed. It was not until 1948 when the honeycomb was adapted for use with fiber-reinforced composites. Current Nomex honeycomb cores, developed by DuPont in the 1960's, give sandwich structures excellent fire-retardant capabilities for use in aircraft and automotive applications.[vii] Nomex is a meta-aramid material, a polymer related to Nylon but with an aramid polymer molecular backbone. For use in the honeycomb core, the polymer is coated in phenolic resin to fireproof the core.

1.3.4. Sandwich Panels

The structure of a sandwich plate consists of three primary components: two face sheets and a core. The top and bottom face sheets are strong and stiff and give the panel its bending stiffness. A lightweight (low density) core material made of honeycomb or a foam gives the panel sufficient thickness for the application (Figure 7).[viii]

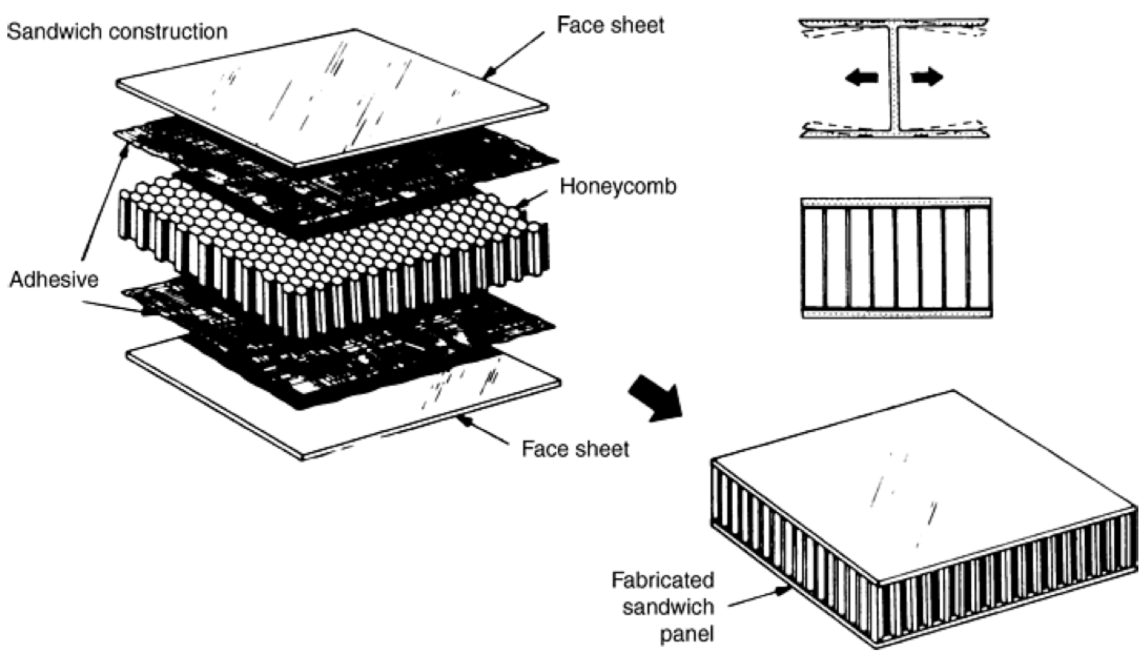

Figure 7: The basic components of a sandwich panel. High modulus face sheets and a light, compliant core material are combined to form a stiff, light structure.[8]

Thin, woven composite laminas are ideally suited for the face sheets because of their high stiffness and tensile strength. For the core material, honeycomb structures allow the addition of significant thicknesses while remaining lightweight. While each of these components alone are relatively weak in bending, when combined they form a rigid structure at low weight. The face sheets take the maximum tensile and compressive stresses involved in bending because they are furthest from the center of the panel's neutral axis. The honeycomb maintains the rigidity of the structure so the face sheets can be loaded higher than the buckling point of thin laminas. Because it transmits the tensile and compressive stresses between the face sheets, the core experiences mostly uniform shear loading with minimal tensile or compressive stresses. Correspondingly, the maximum shear strength in hexagonal honeycomb structures is in the L and W directions (Figure 6). The combination of these three components results in a light, stiff and strong structural member, capable of extremely high strength-to-weight and modulus-to-weight ratios. Table I shows the effect on stiffness of adding a honeycomb core of two thicknesses (B and C) compared with a 0.81 mm thick aluminum sheet (A). The aluminum sheet was halved and used as the face sheet for the two sandwich panels, and the thickest panel is over 37 times stiffer than the aluminum sheet, at only a 6% weight increase.[ix]

Table I: Effect of sandwich panel thickness on the relative stiffness and strength of two aluminum-honeycomb sandwich panels.

Property	A	B	C
Relative stiffness (*D*)	100	700	3700
Relative strength	100	350	925
Relative weight	100	103	106

Sandwich panels are made at Pratt & Miller using wet lay-up in a vacuum bag molding process. This manufacturing technique provides a high degree of bonding between the face sheets and the core; however, it is inherently a low volume production method. For the production of panels for a few racecars this is not a problem, and the high level of control over the properties available

in this construction method is beneficial. The Pratt & Miller traditional panel is made of two [0/+45/0/+45] laminate face sheets of T-300 plain weave carbon fiber with an epoxy matrix. T-300 is a PAN carbon fiber with moderate strength and modulus values for carbon fiber. The Nomex honeycomb core gives the panel excellent fire retardant capabilities at low density, an ideal combination for a racing application.

1.4. Mechanics

1.4.1. Background

The mechanics of sandwich panels is built on the mechanics of laminated composites. Lamination theory includes several assumptions:

1. The laminate is thin and wide (width >> thickness)
2. A perfect interlaminar bond exists between various laminas
3. Strain distribution in the z-direction is linear
4. All laminas are macroscopically homogeneous and exhibits linear elastic behavior

The laminate contains the xy axes and the z-axis defines the thickness direction. The thickness of the laminate is h, and the thickness of each lamina is represented by t_1, t_2, … t_n. Following from the assumption that strain distribution is linear in the z direction, the resultant forces [N] are given by,

$$\begin{bmatrix} N_{xx} \\ N_{yy} \\ N_{xy} \end{bmatrix} = [A] \begin{bmatrix} \varepsilon^{\circ}_{xx} \\ \varepsilon^{\circ}_{yy} \\ \gamma^{\circ}_{xy} \end{bmatrix} + [B] \begin{bmatrix} k_{xx} \\ k_{yy} \\ k_{xy} \end{bmatrix} \qquad (1)$$

Where [A] is the extensional stiffness matrix and [B] is the coupling stiffness matrix and the resultant moments are,

$$\begin{bmatrix} M_{xx} \\ M_{yy} \\ M_{xy} \end{bmatrix} = [B] \begin{bmatrix} \varepsilon^{\circ}_{xx} \\ \varepsilon^{\circ}_{yy} \\ \gamma^{\circ}_{xy} \end{bmatrix} + [D] \begin{bmatrix} k_{xx} \\ k_{yy} \\ k_{xy} \end{bmatrix} \qquad (2)$$

Where [D] is the bending stiffness matrix.[x]

1.4.2. Twist Test Mechanics

For the twist test performed in this project, the resultant moments are of primary importance, specifically the twisting stiffness, D_{66}. Experimental determination of D_{66} using the twist test for composite panels has been modeled using classical laminated plate theory (CLPT), finite modeling analysis (FEA), and experimentally using a derivation of the ASTM twist test for plywood by F. Avilés et al. in Experimental Mechanics.[xi,xii] Additional assumptions for the twist test are that the sandwich is symmetric in nature and that the face sheets and core are either isotropic or orthotropic. For symmetric matrices, the coupling stiffness matrix, [B] = 0. This allows Equation 1 to be simplified to:

$$\begin{bmatrix} M_{xx} \\ M_{yy} \\ M_{xy} \end{bmatrix} = [D] \begin{bmatrix} k_{xx} \\ k_{yy} \\ k_{xy} \end{bmatrix} \quad (3)$$

Where [D] is the bending stiffness matrix for an orthotropic material:

$$\begin{bmatrix} D_{11} & D_{12} & 0 \\ D_{12} & D_{22} & 0 \\ 0 & 0 & D_{66} \end{bmatrix} \quad (4)$$

It then follows from laminated plate theory that the twisting stiffnesses, D_{66}, is

$$D_{66} = (G_{12})_f \left(\frac{2h_f^2}{3} + \frac{h_c^2}{2} + h_f h_c\right) + \frac{(G_{12})_c h_c^3}{12} \quad (5)$$

Where G_{12} is the shear modulus of the facesheets, h_f is the thickness of the face sheet, and h_c is the core thickness. The second term in this equation can usually be ignored in cases where $(G_{12})_c << (G_{12})_f$ which is the case for most sandwich panels from their mechanics. This approach however fails to account for transverse shear deformation, limiting the accuracy for the predicted plate compliance when compared with experimental data. F. Avilés revisited this problem with a subsequent paper in which he detailed the nature of the error. In panels with core material having a high transverse shear modulus, the twist test will provide the in-plane shear modulus of the face sheets, however some low-density core materials are very deformable in shear, which can lead to errors. The use of a Nomex honeycomb core should provide adequate shear

modulus to model the in-plane shear modulus by correcting for transverse shear deformation. The core shear modulus of the face sheets in the case of a low shear modulus core material simplifies to:

$$(G_{13})_c = \frac{c_1}{h_c(C_{\exp} - \frac{ab}{16D_{66}})} \tag{6}$$

Where c_1 is an FEA derived non-dimensional constant to correct for the shear deformation and C_{exp} is the experimentally measured compliance. The constant has been found to vary slightly around 1.00 for a wide range of panel sizes, as seen in Figure 8. For this reason, the core shear will only be estimated, and c_1 will be assumed to have a value of 1.00.[xiii]

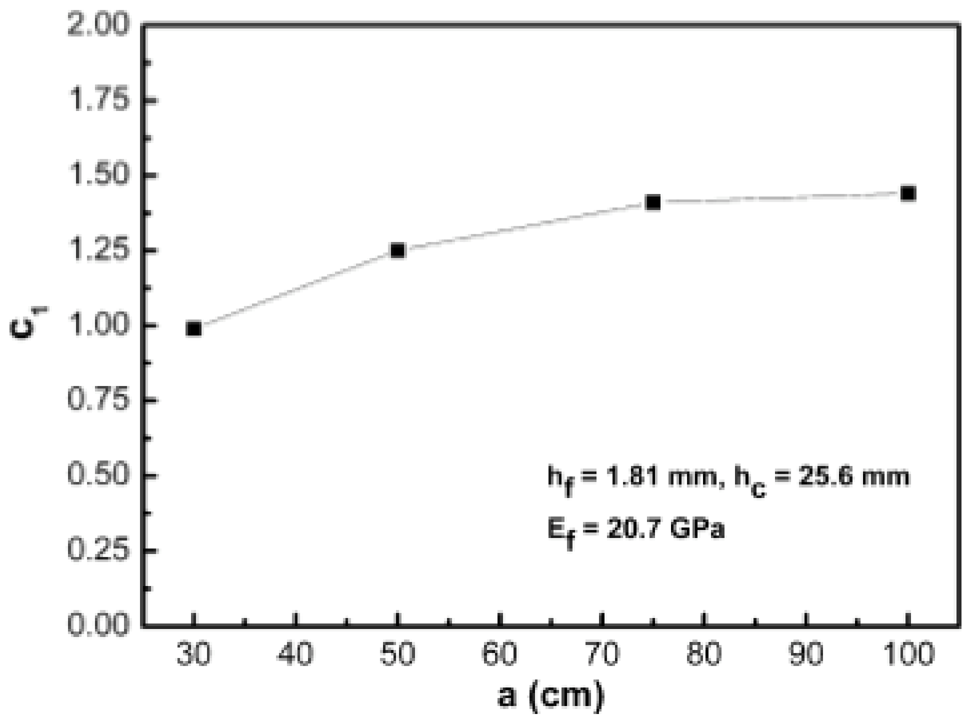

Figure 8: Value of c_1 as a function of panel edge length for a sample panel.*

2. Materials and Methods

2.1. Pratt & Miller's Composite Sandwich Panel

The samples tested were produced and cut by Pratt & Miller Engineering into 5-inch test panels. The panels consisted of 4 carbon fiber/epoxy face sheets and a honeycomb Nomex core with an average thickness of .33 in. The face sheets of the sandwich panel consisted of 4 plies of carbon fiber/epoxy laminates. The fiber reinforcement was plain weave T-300 carbon fabric with 3,000-count warp and fill produced by BGF Industries. The matrix is a high temperature RTM resin produced by De-Comp Composites. The laminas are oriented in a $0^{\circ}/45^{\circ}/0^{\circ}/45^{\circ}$ manner on each side of the panel. The sandwich panel core is Nomex honeycomb produced by Euro-Composites with a cell size of .125 in (3.2 mm). A close-up of a representative test panel can be seen in Figure 9, showing the 3 basic layers of an engineering sandwich panel.

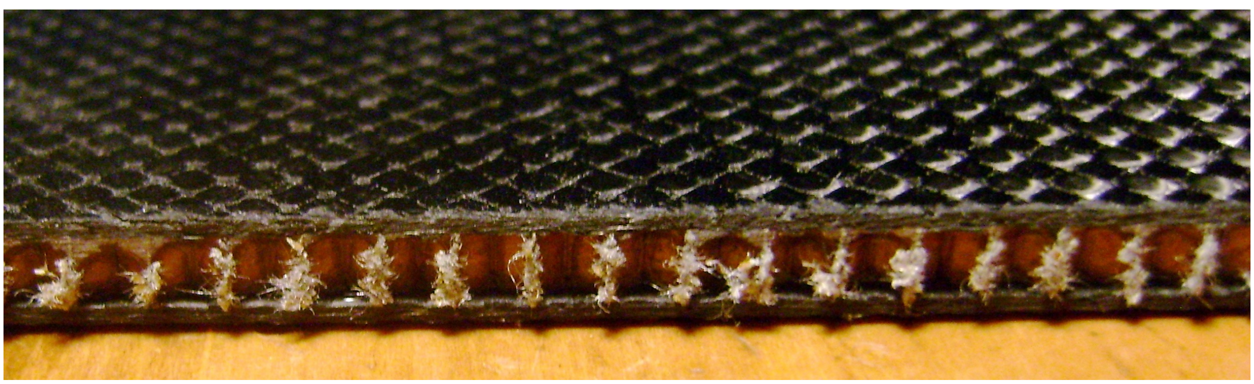

Figure 9: Pratt & Miller's composite sandwich panel. The plain weave carbon epoxy face sheets can be seen along with the 1/8 inch Nomex honeycomb core.

2.2. Twist Testing

In the twist test, the square panel was supported on two diagonal corners and the load was imposed on the two opposite corners (Figure 10). The tests were performed using a fixture designed and fabricated at Pratt & Miller Engineering to replicate the loading described above with the ability to adjust for different sized panels. The steel base plate was attached directly to the base of the load frame, Instron 5584, and contained holes along the diagonal to accommodate various composite panel dimensions. The load was imposed by a steel crossbar attached directly to the load cell and moving crosshead of the

frame. The crossbar had a length of 18 inches and cross-section of 1x1.5 inches with holes drilled along its axis corresponding to the holes drilled on the base plate.

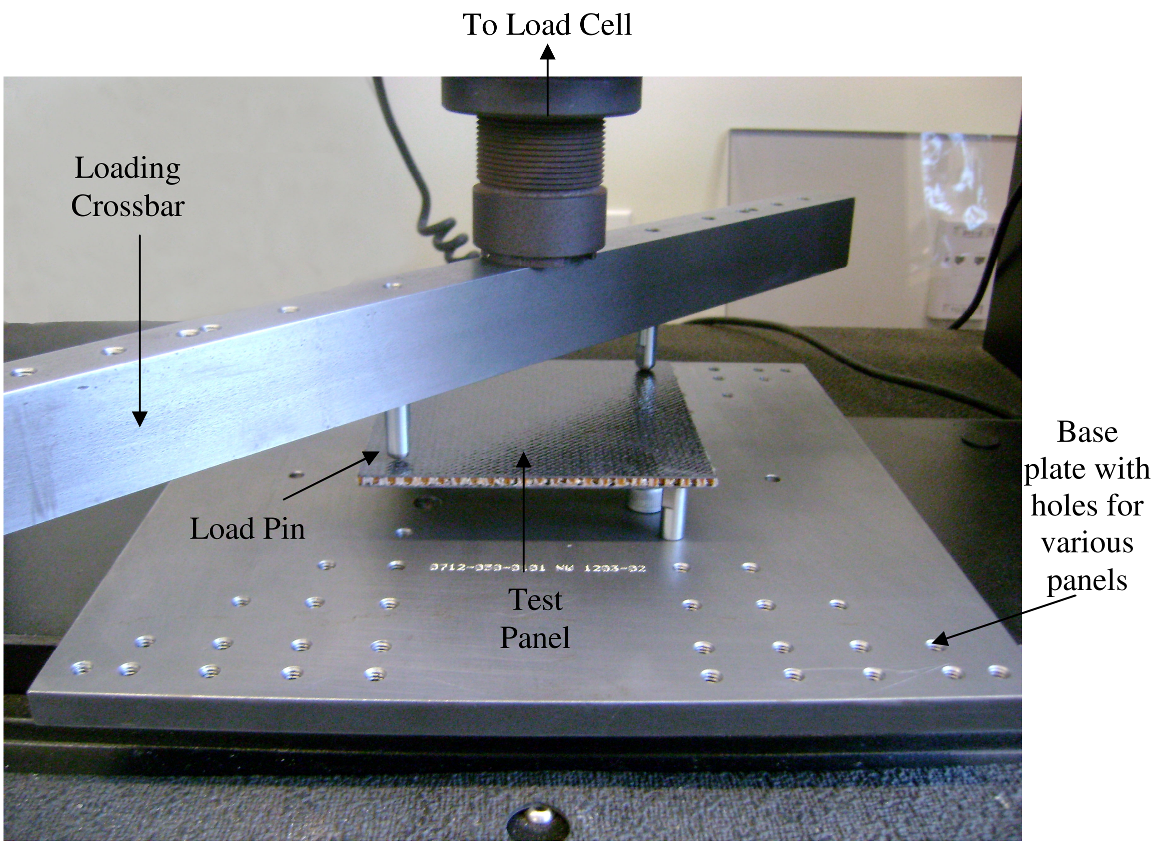

Figure 10: Twist test fixture designed and fabricated by Pratt & Miller. The fixture has the option of testing multiple different panel sizes and geometries. It also can be used for biaxial bend testing.

Loading and support pins of 0.37 inch diameter were used with hemispherical ends. Prior to twist testing, the compliance of the fixture was measured by recording the slope of the displacement vs. load curve of the crossbar against the base plate at a crosshead displacement rate of 0.01 in/min. The test was repeated 5 times for an average compliance value for the fixture.

2.2.1. Sample Preparation

For the twist test, composite panels were oriented tool-side up. Each sample was labeled with its number and honeycomb orientation with a paint marker for easy identification. Using the nominal dimensions of the test panels

and support pins, it was determined that to center each panel, the loading pins must contact the panels at a point 0.75 inches from the corner along the diagonal. Each of these points was also marked using a silver paint marker to accurately load each panel and reduce variation from panel to panel due to loading.

2.2.2. Test Procedure

The test method was developed based on a standard Bluehill software (Instron Corp.) compression method, with a crosshead rate of 0.6 in/min, as described in ASTM 3044, the standard for determining the shear strength of a plywood panel.[xiv] Ten panels were tested in two test runs, split with five specimens tested during each run. Four panels were tested with the L direction of the honeycomb core, also known as the ribbon direction, oriented perpendicular to the front edge of the panel while six panels were tested with the ribbon direction oriented parallel to the front edge. The differences in orientation are shown in Figure 11 with side on views of two representative test panels.

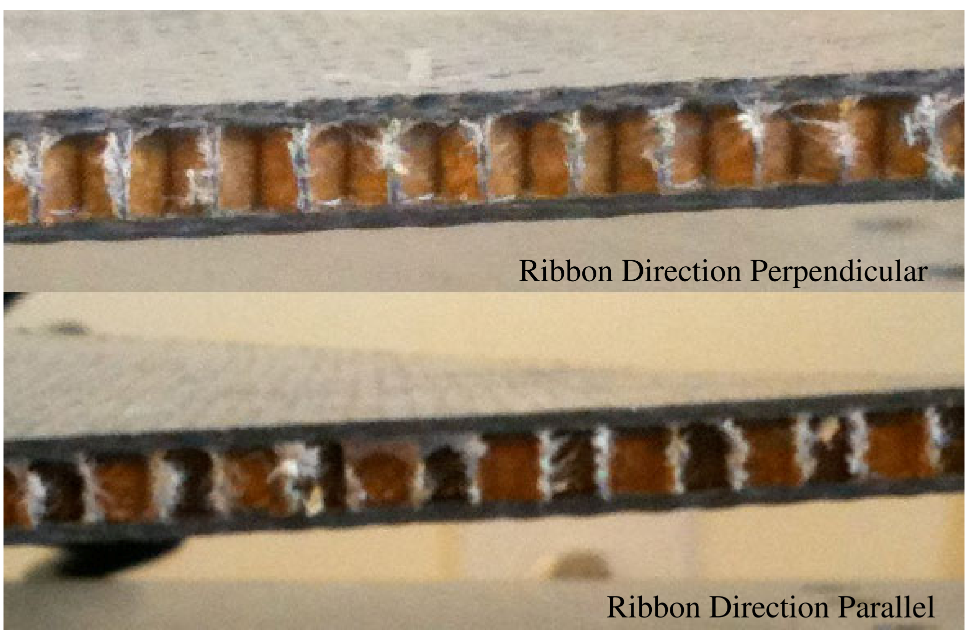

Figure 11: Seen from the declared front of the test panel, the difference in honeycomb orientation can be seen. In the top sample, the ribbon direction can be seen coming out of the page, towards the reader. In the bottom sample, the ribbon direction travels parallel to the front of the plate and left to right on the page.

Each panel was photographed for documentation after loading and at specific loads during testing. Pictures were taken parallel to the plate at 115, 170, 225 lbs, every 25 lbs after as the test progressed, and at the point of failure. The test was stopped after the load had decreased enough to ensure that the panel had failed and another picture was taken after the test was stopped. The final documentation was recorded after the panel was removed from the fixture.

2.3. Biaxial Bend Testing

2.3.1. Sample Preparation

The biaxial bend test used the same fixture as the twist test, but instead of loading and supporting opposite corners, the 5x5 in square composite panel is supported in all four corners. The panels are oriented with the tool side up as in the twist test. The crossbar has a centrally located hole for loading the test panel. To ensure all the panels were centered on the support pins and loaded in the center, the front right corner was marked similarly to the twist test panels (0.75 inches from the corner along the diagonal) however this time on the bottom of the plate (bag side). The center of the plate was then located and marked on the top of the plate (tool side). These two points were then used to align each panel, ensuring a higher degree of accuracy than by visually centering them. The panels were numbered to keep track of each specimen.

2.3.2. Test Procedure

The biaxial bend test used the same method and crosshead displacement rate as the twist test. For the first five specimens, the force from the loading pin was not distributed in any way. This led to local face sheet failure with the pins puncturing through to the core, which invalidated the test results. To distribute the force, a steel washer was placed under the loading pin. The washer was replaced after noticeable deformation occurred after the third test, and one washer per test was used for the final two test specimens.

3. Results and Statistical Analysis

3.1. Twist Test Results

Twist testing occurred in two separately run tests of five specimens each. The honeycomb orientation of the panels was not checked until after the testing was completed so the results displayed in Figure 12 and Figure 13 below do not represent the original test runs, but were sorted after testing. Figure 12 shows the specimens that were tested with the "L" direction of the honeycomb core or the ribbon direction parallel to the declared front of the test specimen. Figure 13 contains the twist test data from the other four specimens which were tested with the "L" direction of the core tested perpendicular to the front of the test specimen.

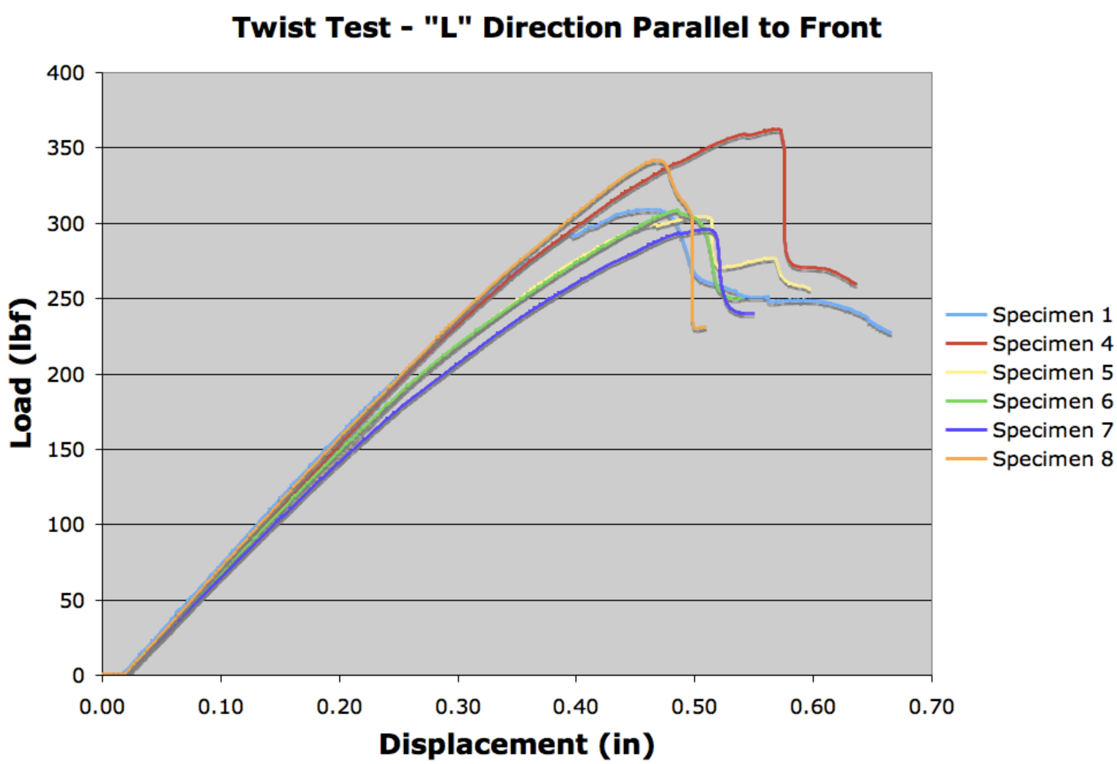

Figure 12: Twist test results for the 6 samples oriented with the "L" direction of the honeycomb core parallel to the front of the plate.

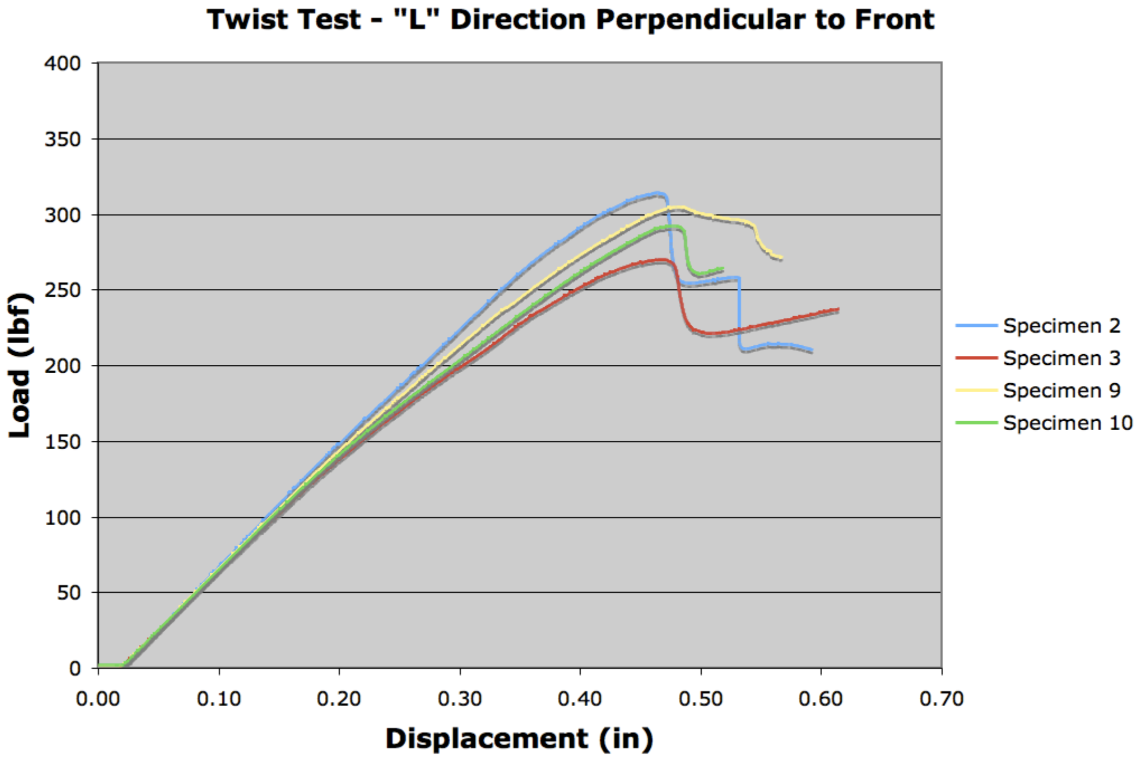

Figure 13: Twist test results for the 4 samples oriented with the "L" direction of the honeycomb core perpendicular to the front of the plate.

It can be seen that the general data trend seems to be similar throughout testing. Generally the strength of the panels dropped off sharply after failure occurred. During testing failure was usually marked by a single sharp popping sound or series of pops. Based on the panel response during testing and at failure, the failure mode initially looked to be core shear failure. The specimens with "L" parallel also have a higher strength at failure in general, with an average maximum load of 320.7 lb compared with 295.6 lb for the perpendicular orientation. This could be due to the orientation of the honeycomb, however more investigation is required to make that assessment.

From the slope of the graph, the experimental compliance of the panels can be calculated (Table II). Using the linear portion of each plot, a line of best fit was created, with the slope being equal to the load per unit displacement, or stiffness. Upon inversion, this value is the compliance of the plate, with units of in/lb. To more easily visualize this value, it has been converted to

thousandths of an inch per lb, or mil/lb. To accurately obtain this value, the linear fit was calculated with only the linear portion of the data. Each line of best fit was accurate up to an R^2 value of over 0.995 to ensure the validity of the slope.

The location of failure was primarily in the loaded corners of the plates. As seen in Figure 14, the maximum deflection occurred most often in the back right corner of the plate. The green boxes represent the location of the failure for the panels loaded in the parallel orientation and yellow boxes represent the panels loaded perpendicularly. The number of the test specimen is listed in the quadrant where failure occurred.

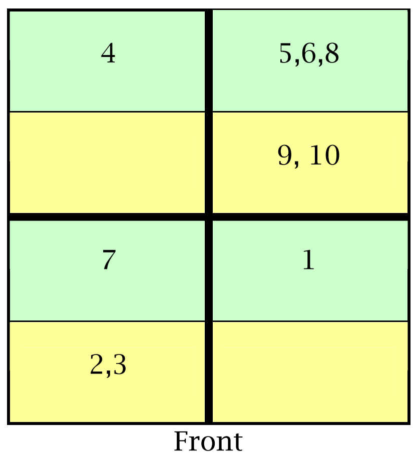

Figure 14: Location of greatest deflection during each twist test. The green boxes represent parallel orientation and the yellow boxes represent perpendicular orientation. The most common location regardless of orientation was in the loaded corners, not within the support corners.

3.1.1. Statistic Analysis

Once the compliance values were calculated (Table II), a two-sample t-test was performed to compare the compliance and maximum loads of the two different orientations. The t-test was performed with a 95% confidence interval and an alternative hypothesis of the sample population not being equal. The t-test revealed a p-value of 0.176, above the alpha value of 0.05, supporting the null hypothesis that the core orientation of the does not have a statistically significant effect on the compliance of the panel. Figure 15 shows a box plot of

the compliance data calculated from the slope of each test along with calculated mean, median, upper and lower quartile and the range of the dataset.

Table II: Compliance Values Sorted by Honeycomb Orientation and Statictical Analyis

	Specimen	C (mil/in)	
Parallel	1	1.257	**Average**
	4	1.235	1.303
	5	1.331	**StDev**
	6	1.349	0.081
	7	1.427	
	8	1.216	
Perpendicula r	2	1.272	**Average**
	3	1.492	1.394
	9	1.368	**StDev**
	10	1.446	0.096

Two Sample T-Test p-value = 0.176

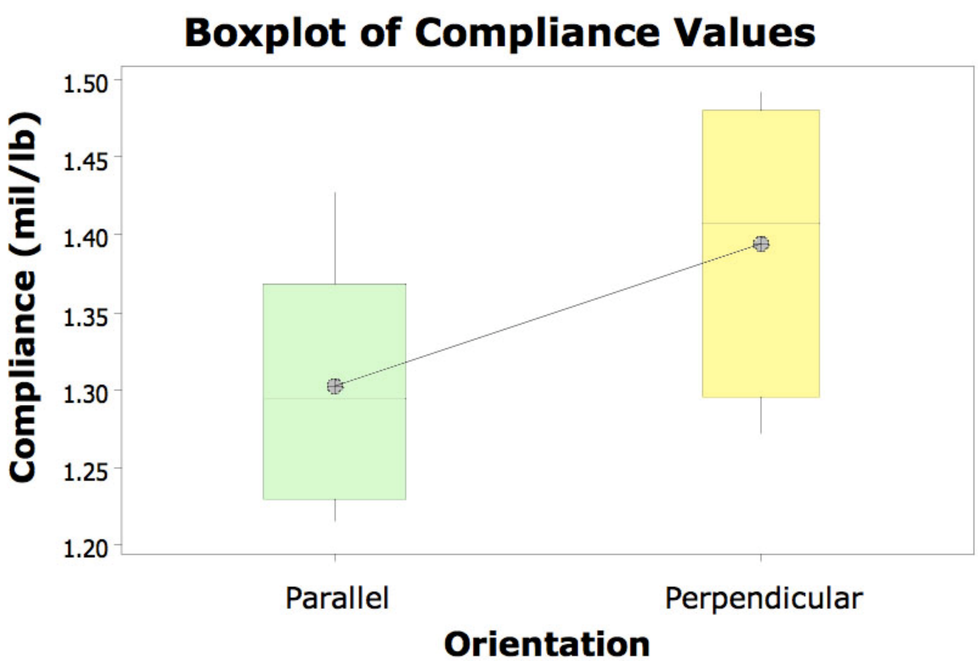

Figure 15: Box plot developed for the twist test using Minitab statistical software package and a two-sample t-test. The boxes represent the upper and lower quartile values and the median while the dot represents the mean of the data set. The data sets cannot be statistically differentiated.

3.2. Biaxial Bend Test Results

The biaxial bend tests have a higher strength and less displacement than the twist tests. Supporting the panel in all four corners increases the resistance to flexure and loading the center of the panel effectively halves the support span. For the first five specimens, it was mentioned previously that the loading pin caused local face sheet failure, nullifying the tests. The second set of panels was tested using a steel washer, which allowed the force to be distributed over a larger area and reduced local compressive stress on the face sheets. After placing the washer under the loading pin, the remaining five specimens failed similarly to the twist tests, and with little to no permanent local indentation. The data from the biaxial bend tests can be seen in Figure 16. The compliance of the panels when loaded in biaxial bending was found in the same manner as it was for the twist test specimens. The steeper slope results in a lower compliance and a stiffer response to loading as mentioned before. Table III shows the compliance values for each panel tested, and is generally 1/3 the value of the compliance of the twist test values.

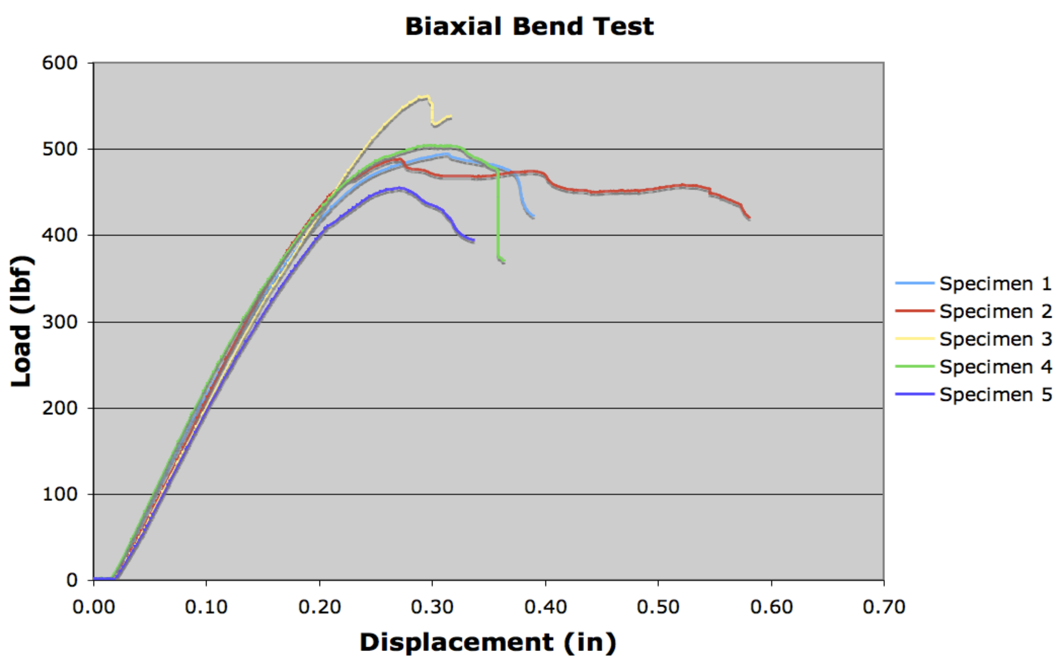

Figure 16: Biaxial bend test results. The failure mode was similar to twist testing after the use of a steel washer to distribute the load.

Table III: Compliance Values from Biaxial Bend Testing.

Specimen	C (mil/lb)
1	0.4460
2	0.4161
3	0.4315
4	0.4350
5	0.4535
Average	0.4364
Stdev	0.0144

4. Discussion

After finding the compliance of the plates using the twist test, the experimental value can be compared with the predicted compliance based on FEA modeling. The C_{FEA} value was provided by Pratt & Miller after FEA modeling was 1.426 mil/lb. This is within one standard deviation (0.948) of the experimental data collected and well within the 95% confidence interval for our experimental data, assuming normal distribution. Therefore the FEA model used by Pratt & Miller agrees well with the experimental data and is accurately modeling the core shear phenomenon.

Using the lamina properties of the face sheets and Equation (5), the twist stiffness of the plate can be calculated as well. The D_{66} value was determined using the material properties and sandwich panel properties, however this equation is based on the basic CLPT equations and may contain some error due to shear. The value of D_{66} is reported in Table IV.

According to Aviles et al, the core shear modulus can be estimated using Equation (6). Two assumptions must be made to find the core shear modulus using this method. One is that the core can be treated as an isotropic material under twist test loading conditions. As shown above, the compliance of the panels are not significantly different between the two orientations, and therefore under the complex loading of the twist test, the assumption of the honeycomb as an isotropic material holds. The second is the value of the non-dimensional constant c_1. F. Aviles uses this constant as a free variable to more accurately fit the FEA model for the twist test. In the literature, for a large variance of core moduli and panel sizes, the value never varies significantly

from 1.00.[xv] While using a value of 1.00 is not likely to be completely accurate, it should give a reasonable value that can be compared with the manufacture's value. The calculated value, reported in Table IV, is approximately half of the manufacturer's value. While varying the c_1 has only a slight effect on the value calculated, variation of the D_{66} value has a great effect. This effect most likely due to the D_{66} value being based on CLPT relationships which are not completely accurate for sandwich panels. Another issue is the use of the c_1 value. This value was found using a different sandwich panel structure than used in this test and that could lead to error in assuming that value is 1.00.

Table IV: Calculated Values and Experimentally Determined Values for the Twist Tests.

Compliance (mil/lb)		Twist Stiffness	Core Shear Modulus	
C_{FEA}	$C_{EXPERIMENTAL}$	D_{66} (ft-lb)	Euro Composites	Experimental $(G_{12})_C$ (psi)
1.43	1.34	339.23	4351.13[xvi]	2450.87

Based on the panel response to testing, the test panels failed primarily due to the transverse shear applied to the panels. This is a common failure mechanism for sandwich panel composites which can manifest as either core shear failure or as face-to-core debonding.[xvii] Examining the load response curves for the twist tests, the sharp drop in the load carried my nearly all of the samples indicates a rapid failure of the top face sheet to transfer the load to the bottom face sheet. This rapid loss of panel integrity can be attributed to core shear failure. However in the biaxial bend testing, the load response curves do not generally exhibit the same rapid drop in supported load. The reduction of the load seen is more gradual and seems to fit more with progressive face-to-core debonding. In order to achieve core shear failure using the biaxial bend test, a larger panel size is recommended. This would result in a greater support span for the plate, which would increase the forces passed through the core resulting in core shear failure instead of the progressive delamination seen.

5. Conclusions

1. The experimental twist test results verify the accuracy of the FEA model being used at Pratt & Miller. The FEA model compliance (1.43 mil/lb) is within 1σ of the experimental data average value (1.34 mil/lb).

2. The orientation of the core is not a factor in the compliance of the panel in the twist test. A two-sample T-test showed a P-value of .176, which means that the differences in the two data sets collected are not statistically significant.

3. Failure of the panels in the twist test occurred primarily through core shear failure. Failure of the panels in the biaxial bend test occurred through face-core debonding as a result of the decreased support span length.

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