Development of a Weight-Saving Carbon-Fibre-Reinforced Polymer Component for a FSAE Race Car

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Formula SAE is an annual competition that challenges university students to conceive, design, fabricate and compete with a small formula style vehicle. This competition aims to encourage engineering innovation and as such permits the use of a wide range of materials. Carbon-fibre-reinforced polymer (CFRP) composites offer a highly favourable alternative to traditional materials such as steel and aluminium in FSAE design and fabrication, being used extensively by all of the top teams in the competition. The use of CFRP composites can lead to significant reductions in weight, increasing the overall track performance of the vehicle through increasing its acceleration and deceleration characteristics, as well as its traction, control and handling. The limited use of CFRP composites by the Academy Racing Team to date is primarily due to the added complexity in the design, computational modelling and fabrication of such components. As part of this research project a conceptual review of the 2017 FSAE vehicle was undertaken, in which the main member of the rear suspension system was identified as the most suitable candidate for replacement with CFRP composite materials. This report details the engineering process, decisions and justifications behind the development and design of the CFRP composite rear suspension beam for integration into the 2017 FSAE vehicle. All computational design, modelling and structural analysis throughout this project was completed in CATIAV5 CAD software.

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Nomenclature

CFRP = Carbon Fibre Reinforced Polymer
FSAE = Formula Society of Automotive Engineers
ART = Academy Racing Team
CAD = Computer-Aided Design
FEA = Finite Element Analysis
I. Introduction

A. Motivation

Each year the Academy Racing Team at UNSW Canberra takes part in the Formula SAE competition, in which undergraduate teams are encouraged to demonstrate and prove their creativity and engineering skills against universities from around the world. The competition challenges students to “conceive, design, fabricate, develop and compete with small, formula style, vehicles” (SAE International, 2016) which are to be targeted for the nonprofessional, weekend competition market. As such the vehicle needs to have very high performance in terms of acceleration, braking and handling, while being sufficiently durable, reliable, inexpensive and producible in a student workshop. Having a high strength-to-weight ratio carbon-fibre-reinforced polymer (CFRP) materials present a strong, light-weight alternative to the steel used in many of the race car’s current components. Decreasing the weight of selected components of the vehicle assembly can greatly enhance track performance.

B. Aim

The aim of this project is to identify, design and develop a weight-saving carbon-fibre-reinforced polymer composite component for the Formula SAE race car at UNSW Canberra. This will culminate with the submission of engineering design documents for the CFRP composite component, and all additional accessories required for integration with the 2017 vehicle, to the Academy Racing Team. This will include a requirement and load case analysis, finite element analysis (FEA) and experimental testing and results. A wet lay-up process will be used for fabrication for all test pieces and the final product due to the availability of materials and complexity of the geometric design.

II. Background

A. Carbon Fibre Reinforced Polymer Composites

Carbon fibre reinforced polymer (CFRP) composites are a type of fibre reinforced composite material consisting of two key constituents; the carbon fibre reinforcement which acts as the primary structural load bearing component, and the polymer which acts as the matrix that binds these fibres together. The development of CFRP composites began in the 1950s, attaining status as a mature structural material in the 1980s (Chawla, 2012). They have since become a commonly used material in many high performance vehicles due to their high strength, high modulus, low density, and reasonable cost (Chung, 1994).

The carbon fibre used in these composite structures refers to short or continuous fibres which are of at least 92 wt.% carbon in composition (Chung, 1994). The carbon atoms are bonded together in crystalline formations, aligned in long chains parallel to the axis of the fibre. This alignment of the crystals determines the mechanical properties of the individual fibres. Carbon fibre can be purchased in a range of different material arrangements, such as uni-directional, weaves, yarns, fabrics and braids. These raw materials consist of thousands of continuous individual carbon filaments.

B. Epoxy Resin

The epoxy resin provides the stable matrix which binds the carbon fibres together, transferring the loads between the fibres and maintaining their structure. The resin also provides a barrier for the fibres, protecting them against weather, water and corrosive chemicals (Composites Australia, 2016). Epoxy resins typically perform well under compressive and shear loading, and are able to be tailored as required for a range of different applications. This includes altering the length of cure times, temperature resistance, flame retardency and increased fracture toughness (Allred and Associates, 2016). For wet lay-up processes epoxies are supplied as a two part system, consisting of a liquid resin and hardener. When these are mixed together during the fabrication process they undergo a chemical reaction that forms extensive crosslinking between the two chemicals, producing the thermosetting polymer that cures into a solid. The resin and hardener mix is impregnated into the raw carbon fibre material before this curing process is complete (Composites Australia, 2016).

C. Mechanical Properties of CFRP Composites

As carbon fibre is an anisotropic material the mechanical properties of CFRP composites are directionally dependent, with the greatest strength achieved when the carbon fibres are orientated along the loading directions. However, this orientation can be varied in a number of different ways in order to attain specific strain or flexural structural properties (Hossein, 2014). The mechanical properties of CFRP composites are also dependent on the individual properties of its two constituents, the fibre to resin ratio, the fibre form (unidirectional, fabric, chopped, etc) and the stacking sequence of the composite structure (Clearwater Composites, 2016).

One of the greatest advantages of CFRP composites are their high stiffness-to-weight and strength-to-weight ratios. The stiffness (rigidity) of a material is measured as Young’s modulus, which is expressed as a ratio of
stress over strain. However, as discussed above the Young’s modulus varies greatly depending on the direction of the load. CFRP composites have a much higher Young’s modulus when the force is parallel to the orientation of the fibres (Oskar, 2016).

D. Fatigue in CFRP Composites
While CFRP composite structures generally show better fatigue behaviour than most metals, determining their resistance to cyclic stress is a significant challenge due to the complex failure mechanisms under fatigue loading as a result of their anisotropic strength and stiffness characteristics and overall non-linear behaviour. “Fatigue causes extensive damage throughout the specimen volume, leading to failure from general degradation of the material instead of a predominant single crack” (Eric Green Associates, 1999). This makes it extremely difficult to identify the onset of failure due to fatigue, without extensive NDI techniques such as ultrasonics. The four primary failure mechanisms as a result of fatigue are matrix cracking, delamination, fibre breakage and interfacial debonding (Eric Green Associates, 1999). The analysis of the fatigue life of the CFRP composite component designed and fabricated in the laboratory at UNSW Canberra is outside the scope of this project. While a brief investigation into the degradation of the material properties of the composite panel under cyclic loading during the fastener connection tests was conducted, the number of tests conducted was inadequate to draw any reasonable conclusions on the fatigue life of these connections. This is discussed further on, with recommendations of both testing before use by the ART, and inspections throughout its life.

E. FEA in CATIAV5 – Composite Failure Criterion
All computational design and analysis undertaken for this project was conducted with CATIAV5. The CFRP composite models were developed through first building the base geometry in the ‘Part Design’ workbench. After this the ‘Generative Shape Design’ workbench was used to extract the surface geometry and contours of the design, which was then imported into the ‘Composite Design’ workbench as the building block for the composite stacking engineering. A custom material file was created from researched material data, and then further refined using experimental results. The load points, directions, orientations and stacking sequences of each ply were then defined as part of the stacking engineering.

For the structural analysis of composite structures CATIAV5 offers three different failure criterion; Tsai-Hill criterion, Tsai-Wu criterion and Hoffman criterion. For this project the Tsai-Wu failure criterion was used. This material failure theory is widely used for anisotropic composite materials which have different strengths in tension and compression and considers the total strain energy for predicting failure. The difference between this criterion and the Tsai-Hill criterion is that it is more general, distinguishing between compressive and tensile failure strengths (Dassault Systemes, 2016). It predicts failure when the failure index in a laminate reaches 1, however it does not predict different failure modes, such as fibre failure, matrix failure and fibre-matrix interface failure. The base Tsai-Wu failure criterion used by CatiaV5 for each lamina is:

\[
\frac{\sigma_x^2}{S_{1y}S_{2z}} + \frac{\sigma_y^2}{S_{1x}S_{2z}} + \frac{\tau_{xy}^2}{S_{1x}S_{2y}} + \frac{\tau_{yz}^2}{S_{1y}S_{2z}} + \frac{\tau_{xz}^2}{S_{1x}S_{2z}} + \left(\frac{1}{S_{1x}} - \frac{1}{S_{2z}}\right)\sigma_x + \left(\frac{1}{S_{1y}} - \frac{1}{S_{2z}}\right)\sigma_y - \frac{\sigma_x\sigma_y}{(S_{1x}S_{1y}S_{2z})^2} \leq 1
\]

- \(S_{1x}\) and \(S_{1y}\) = Longitudinal Tensile and Compressive Stresses.
- \(S_{2x}\) and \(S_{2y}\) = Transverse Tensile and compressive Stresses.
- \(S_{1y}\), \(S_{2x}\) and \(S_{1z}\) = Shear Stress Limit in the XYZ, YZ and XZ planes.
- \(\sigma\) = Ultimate tensile and compressive strength.
- \(\tau\) = Ultimate shear strength.

F. Integration of CFRP Composites with Vehicle Assemblies
In high-performance motorsport there is a strong precedence for the use of metal attachment pieces with both mechanical bolt fastener connections or adhesive bonds (often both) to CFRP composite components. The effective design of these attachment points is extremely important as they are responsible for transmitting all of the vehicle loading into the composite structure, and are often locations of high stress and the point of failure. The two primary design considerations for the development of attachment pieces are; the method of fastening it to the composite structure (as the machining of a composite component risks significant damage to its structure and degradation of its strength), and the conformity of the two different materials under loading, which can result in significant stress raisers. Figures 1 and 2 illustrate common design concepts for attachment pieces for CFRP composite components used in high-performance motorsport.

Figure 1: Aluminium attachment pieces fastened to a CFRP composite chassis.

Figure 1: Aluminium attachment pieces fastened to CFRP wheel rims.

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III. Candidate Selection

A conceptual review of the 2017 vehicle was conducted in order to identify the candidate that would offer the greatest potential for increasing track performance through fabrication out of CFRP composite materials. Key factors considered during this review were the un-sprung and rotating weight of each component, as well as the manufacturability of components, availability of materials and tooling, complexity of design and vehicle assembly. Following this review, the component identified as the best candidate for replacement with CFRP composite materials was the rear suspension beam. This conceptual review of the 2017 vehicle and details on the current design of the rear suspension beam can be viewed in annex A. Alongside this an assessment of the 2017-18 Formula SAE Rules (SAE International, 2016) was completed, identifying limitations and minimal structural/safety requirements, summarised in annex B.

IV. Preliminary Design Process

A. Rear Suspension Beam Requirement Analysis

In conjunction with the Academy Racing Team, a set of requirements for the new design of the rear suspension beam was established. These were divided into four main categories, being Integration with the 2017 FSAE Race Car, geometric location of connection points, structural and aerodynamics. Each requirement was then classified as Mandatory (M), Desirable (D) or Optional (O). The leading priority when establishing and classifying these requirements was ensuring that the new rear suspension beam design was going to be compatible with the current systems and assembly of the 2017 race car. As shown in Figure 3, there are a number of different components that connect to the rear suspension beam, and are located around the rear of the vehicle that need to be taken into consideration when redesigning its geometry. See annex C for the full table of requirements and their classification.

![Figure 3: CatiaV5 model illustrating the integration of the current rear beam design with the vehicle assembly.](image)

B. Load Case Requirements

Two maximum load cases were considered for the rear suspension beam. The lateral load case was a combination of a lateral force of 6009N and a vertical bump force of 12017N, acting on one rear tire. Meanwhile, the longitudinal load case was a combination of a longitudinal force of 2163N and vertical bump force of 4206N acting on both rear tires. These loads are applied offset to the beam at the contact point between the tires and the track surface, and thus generate significant torsional forces and bending moments on the structure. The assumptions, calculations and factors of safety used to develop these two load cases are detailed in annex D.

C. Finite Element Analysis of Current Design

In order to visualise the distribution of loads and stresses throughout the rear suspension beam, a FEA of the current design under each load case was conducted in CATIAV5. Figures illustrating the results of the FEA are located in annex E. It is evident from this analysis that in both load cases the highest concentrations of stress were at the attachment points of the trailing arms and shock absorbers. Under the lateral load case the beam experienced a large bending moment on one side, with the large cylindrical shaped end being more prone to deformation than the 50mm square sectioned beam. As expected, there were also higher stress concentrations at the changes in direction of the beam. Meanwhile, in the longitudinal load case the beam experienced both a bending moment from the bump force, and torsion along the lateral axis. The cylindrical ends handled these loads well, with the greatest stresses being at the connection point of the square-sectioned beam to the cylindrical ends, and again at the changes in direction of the beam.

D. Preliminary Rear Beam Design Concepts

A range of different preliminary design concepts for the base geometry of the new beam were conceived and modelled in CatiaV5, exploring different ways of meeting the geometric requirements of the vehicle’s current suspension system, while making room for a diffuser and avoiding interference with the drive train system...
components. The preliminary designs were then presented to the FSAE team in order to discuss issues such as vehicle assembly, part accessibility and manufacturability. This was in order to identify weaknesses and strengths of each of the different concepts, before progressing with the design process. Images of each of these preliminary designs and their structural analysis are located in annex F.

IV. Experimental Testing

A. Tensile Tests – Determination of Young’s Modulus

In order to determine the value of Young’s modulus for the CFRP composite material fabricated in the UNSW Canberra composite laboratory, nine dumb-bell shaped test pieces of 4, 6 and 8 plies were tested in the Shimadzu universal testing machine (Shimadzu Corporation, 2016). These test pieces were fabricated out of 200g/m² carbon fibre 0°/90° fabric using a wet lay-up procedure, with a 50% fibre-resin ratio and cured at room temperature under 20psi of pressure for 24 hours. West System 105-C epoxy resin and 206-C slow hardener were used with a ratio of 5:1. These test pieces were then machined in accordance with the International Standardisation Organisation, ISO 527-1:2012, Plastics – Determination of tensile properties, using a diamond cutter and milling machine in the student workshop. Table 1 below displays the values of Young’s modulus determined from the results of each tensile test.

Table 1: Calculated values of Young’s modulus

<table>
<thead>
<tr>
<th>Test No.</th>
<th>No. of Plies</th>
<th>Young’s Modulus (GPa)</th>
<th>Percentage Error (%)</th>
<th>Error (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4</td>
<td>43.65</td>
<td>±12.93</td>
<td>±5.64</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>40.17</td>
<td>±12.14</td>
<td>±4.89</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>40.65</td>
<td>±12.51</td>
<td>±5.08</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>48.67</td>
<td>±9.72</td>
<td>±4.73</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>45.66</td>
<td>±9.58</td>
<td>±4.37</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>45.55</td>
<td>±9.98</td>
<td>±4.54</td>
</tr>
<tr>
<td>7</td>
<td>8</td>
<td>45.39</td>
<td>±7.72</td>
<td>±3.50</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
<td>34.51</td>
<td>±7.79</td>
<td>±2.69</td>
</tr>
<tr>
<td>9</td>
<td>8</td>
<td>44.05</td>
<td>±7.52</td>
<td>±3.31</td>
</tr>
</tbody>
</table>

The error analysis conducted for these experimental results can be found in annex G. The sources of error in these results arose from; the accuracy of the Shimadzu machine in measuring both the force and position throughout the tensile tests (Shimadzu Corporation, 2013), the accuracy of the digital vernier callipers used to measure the width and thickness of each test piece, and the variation in both width and thickness across the test section of each piece due to the accuracy of the fabrication process. It is evident from the results in Table 1 above that accuracy of the test pieces was better for samples with more plies (i.e. greater thickness). This is because the variation in thickness of the test pieces due to the fabrication process (i.e. excess resin), was the greatest source of error. This is a result of the fact that for test samples of lower thickness, the variation in thickness was a larger fraction of the total thickness resulting in a higher percentage error, and a decrease in total accuracy.

The ultimate tensile strength of the dumbbell shaped samples was not tested, with the force and strain of each piece only being recorded for the linear region of the stress-strain curve. This decision was made as a result of observations made during the tensile testing to failure of the three practice pieces, which resulted in all three of the samples failing at the machined radius, as illustrated in Figure 5. As they did not fail in the region of uniform cross-sectional area, the results obtained from these three pieces were inconclusive.

![Figure 4: CFRP composite dumbbell test pieces.](image)

![Figure 5: Failure of test pieces during ultimate tensile strength testing.](image)
B. Cantilever Beam Test

For further experimental testing, a CFRP composite beam was fabricated in the composite laboratory and subjected to cantilever load testing. The purpose of this experiment was twofold. The first was to test the suitability of the intended method of fabrication, which was a wet lay-up over a foam mould that was then dissolved using acetone to produce a hollow cross-sectioned beam. The second purpose was to assess the validity of the computational results from CFRP composite CATIAV5 material model.

The testing was carried out through using the steel attachments to clamp the beam at one end with a vice and a g-clamp, and attach a vertical loading arm to the opposite end. A dial gauge was then positioned and levelled on top of the loaded end of the beam in order to measure the deflection as it was loaded, as seen in Figure 6. Two tests were then conducted, loading and unloading the beam three times from both 0-20kg and 0-65kg. Graphs 2 and 3 below illustrate the results of these tests alongside the computed values from FEA in CATIAV5 (Figure 7). The values from CATIAV5 are graphed for both the model with the initial material file (created from the mechanical properties of CFRP composite materials listed in relevant literature), and the model with the updated material file following the determination of Young’s modulus.

Firstly, it is evident from the results in Test 1 that the beam had a degree of cured stresses from the fabrication process, causing outlying data in the first cycle. Furthermore, as the beam did not return to its initial position after being completely unloaded this demonstrates that something in the system did not behave elastically. This could be because either a component in the system has undergone plastic deformation (i.e. the beam itself or the epoxy/filler bond), or that the boundary conditions at the clamped end were not entirely rigid. However, this issue appears to have only significantly affected the first cycle, with the beam returning to its initial position in subsequent loading cycles.

The graphs also show that there was a good correlation between the behaviour of the CATIAV5 model using the updated material file and the CFRP composite component fabricated in the laboratory.

C. Three-Point Bend Test

The CFRP composite beam was also subjected to three-point bending in the Shimadzu universal tester (as pictured in Figure 8). The purpose of this additional experiment was to further validate the behaviour of the modelled component in CATIAV5 (Figure 9). Three tests were completed, recording the displacement of the beam as the loading force was steadily increased. Graph 4 below illustrates the displacement at the centre of the beam against the applied force for all three tests, as well as the computational results from CATIAV5. The maximum force applied was increased for each test.

As seen in Graph 4 below, for the three-point bending tests the recorded displacement values from the Shimadzu were significantly greater than that of the
computational model. A factor that may be the source of this disparity is that during the cantilever load testing up to 65kg, there were audible indications of failure within the composite layers during the final test. Upon visual inspection no failure across the surface of the beam was visible; however non-visible failure within the plies could have contributed to the poor agreement of the results in the three-point bend testing.

**Graph 4: Force vs Displacement**

- **Test 1**
- **Test 2**
- **Test 3**
- CATIAV5 Results

**Graph 5: Force vs Displacement to Failure**

In second cyclic test, the panel was loaded and unloaded for 500 cycles at approximately 65% of its maximum load. As seen in Graph 7, over these 500 cycles the maximum and minimum displacement of each cycle continually grew and showed no evidence of approaching a steady state as seen in the 40% loading case. In order to determine if this would occur further testing would be required for a greater number of cycles, however this was not within the scope of this project.

**E. Fastener Test – Metal to Composite Attachment**

One of the most common means of integrating CFRP composite components with vehicle assemblies in high performance motorsport is through specially designed metal fastened attachment pieces. In order to explore this design concept, 300x300mm CFRP composite panels of fourteen 0/90° fabric plies were fabricated and tested in the Shimadzu universal tester. The fastener connection test setup consisted of two 8mm bolts, fastened to each panel with an 80x40x3mm mild steel backing plate.

The first two fastener tests conducted were static tests to failure at loadings applied at 90° and 45° to the plane of the panel, as seen in Figure 10. The results displayed in Graph 5 below illustrate the resisting force of the fastened panel as the displacement was increased at 2mm/min. It is evident from these results that the panel loaded at 90° withstood a slightly greater loading before failure than the one at 45°, reaching a maximum of 18.5kN and 18.2kN respectively. Furthermore, in the 90° load case the failure was rapid (at point A), with a large crack propagating along the centre line of the panel from the backing plate.

In the 45° load case multiple smaller cracks began to propagate from point B, up until point C where one of these rapidly formed into a large crack. From point D onwards the panel’s cracks continued to grow. The second two tests conducted were cyclic loading at an orientation of 90°. The first of these was for 100 cycles at approximately 40% of the maximum load of the panel (set to cycle between 100-7200N). As seen in Graph 6, while the maximum and minimum displacement of the panel initial grew within first 20 cycles, after this it began to approach what appears to be a steady state.

**Figure 9: Three-point bending displacement analysis of CFRP beam in CATIAV5.**

**Figure 10: Fastener test rig and setup while loading at 45°.**

Further observations made during the fastener testing were that the mild steel backing plate underwent plastic deformation before any failure in the CFRP composite panel. A consequence of this was that once the backing plate had plastically deformed the epoxy adhesive bond between the plate and the panel would fail as it was unloaded, as the strength of the bond was not strong enough to uniformly deform the backing plate back to its initial state.
form with the panel. This can be seen in the additional testing images located in annex H. This highlights the need for the backing plate to be designed so that its displacement under loading conforms to that of the CFRP composite component, as failure of the adhesive significantly degrades the structural integrity of the fastener connection. This issue could be avoided through using a higher grade or metal, a thicker backing plate or increasing its length and width.

Additionally, it was also noted that the failure of the adhesive bond was with the surface of the metal and not the face of the CFRP composite panel. Better surface preparation and more suitable adhesives could also reduce the risk of this failure.

**IV. Detailed Design Process**

**A. Finite Element Analysis of the Composite Rear Suspension Beam**

After the selection of the geometry of the rear suspension beam from the preliminary design phase the CFRP composite beam was modelled in CATIAV5 for further analysis, pictured in Figure 11. This initial model consisted of eight plies (sequence: 0°/90°, +45°/-45° repeated). FEA was then run for both the lateral and longitudinal load cases. Post processing permitted the analysis of the Tsai-Wu failure criterion throughout each individual ply, as well as the von Mises stress in the metal attachment pieces and the deflection across the entire assembly.

**B. Ply Stacking Sequence Analysis**

The stacking sequence of a CFRP composite component (i.e. the fibre orientation of each ply) is one of the key determining factors of its mechanical properties, and as such should be tailored to the intended application of the component. In order to determine the stacking sequence for the CFRP composite rear suspension beam a study was done through which FEA was run on the above model at five different stacking sequences for both the lateral and longitudinal load cases. The Tsai-Wu failure criterion across each individual ply, as well as the maximum deflection were then analysed and recorded. Graph 8 below illustrates the maximum value of the Tsai-Wu Failure Criteria within each ply for both load cases.

**Figure 11: Structural analysis and fibre orientation representation of the CFRP composite design.**

**Graph 8: Longitudinal Tsai-Wu Failure Criteria for Different Stacking Sequences**
As illustrated above the stacking sequence that resulted in the lowest failure in majority of the plies was ‘all +45°/-45°’. This is to be expected, as the greatest loading through the beam is in the longitudinal load case. In this load case the beam experiences primarily biaxial torsional forces, and bi-directional carbon fibre at 45° provides the greatest torsional strength and resistance (increasing shear stiffness) (Justin Eitel, 2016). The above graph also shows that while the all 90° stacking performs poorly in the longitudinal load case, as expected it performs significantly better in the lateral load case, where there is a combination of both bending moment and torsional forces.

Furthermore, it is important to note that the maximum failure criteria is not the only design consideration, with the size and the number of the failure areas also being of high importance. This is because this determines the amount/size of additional reinforcement plies required, and thus weight of the final beam.

*Graph 9* illustrates the maximum deflection across the CFRP composite beam for each stacking sequence. It is evident from these results that the all 90° stacking sequence had a significantly greater maximum deflection than the other sequences.

### B. Design of Attachment Points: Upper Trailing Arms and Hub Attachment Pieces

In order to integrate the CFRP composite rear suspension beam with the rest of the vehicle assembly, the beam must have attachment points for each of the trailing arms, shock absorbers and wheel hubs. A number of preliminary design concepts for different techniques of achieving this were conceived and discussed with the ART. Images of these concepts modelled in CATIAV5 are located in annex I. From these designs three key concepts (*Figure 12*) were investigated further;

**Concept 1:** The hub attachment pieces are sleeve fitted to each end of the CFRP composite beam, bonded to the inner walls with adhesive. The chromoly upper attachment piece is bonded with adhesive to the top and side faces of the beam, and fastened with 8mm bolts through to two backing plates on the inner walls. The main benefits of this design are its manufacturability and use of the side faces of the upper attachment piece to mechanically transmit longitudinal loads into the composite beam. The main disadvantage is that both the inner and outer surfaces of the beam must be dimensionally consistent in order to achieve a strong connection.

**Concept 2:** This design combines the hub and upper attachment piece into a single part, with an external sleeve fit over each end of the composite beam. It is then again fastened on either side of the beam with 8mm bolts. The primary benefit of this design is that it relaxes the need for accurate dimensions of the inner surface of the composite beam, however it is more complex to manufacture than concept 1 and has a greater weight. Furthermore, as the assembly is more rigid and restrictive of the deflection of the beam it causes larger stress raisers along its boundaries.

**Concept 3:** In this design the hub attachment piece is sleeve fitted against the inner walls of the composite beam and serves a secondary purpose, being the backing plate for the fastening of the upper attachment piece. The upper attachment piece is manufactured out of aluminium, and is only bonded to the top face of the beam. It is fastened to the beam and the hub attachment piece with three 8mm bolts through the top face. The main benefit of this design is that it is the most lightweight of the three, and requires minimal machining of the CFRP composite beam post fabrication. This is because unlike the previous two concepts, the upper attachment piece does not conform around the rectangular geometry. This also reduces the risk of stress raisers due to non-conformity between the two different materials. The primary disadvantage is the complexity and greater cost in manufacturing the aluminium upper attachment piece.

*Figure 12*: Three primary attachment concepts investigated (1 though to 3 from left to right).
FEA of each concept with the full rear beam assembly was conducted for both the lateral and longitudinal load cases. An iterative process was then undertaken through which the design of each concept was further refined, distributing the stress through each piece as evenly as possible and thus reducing their weight. Table 2 below details the weight of each of these designs and its components after this iteration process.

**Table 2: Weight comparison of Attachment Pieces**

<table>
<thead>
<tr>
<th></th>
<th>Shock Absorber/Upper Trailing Arm Attachments (x2)</th>
<th>Wheel Hub Attachment (x2)</th>
<th>Backing plates (x4)</th>
<th>Lower Arm Attachment (x2)</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concept 1</td>
<td>1552g</td>
<td>707g</td>
<td>120g</td>
<td>438g</td>
<td>2817g</td>
</tr>
<tr>
<td>Concept 2</td>
<td>2435g</td>
<td>N/A</td>
<td>120g</td>
<td>438g</td>
<td>2993g</td>
</tr>
<tr>
<td>Concept 3</td>
<td>505g</td>
<td>1446g</td>
<td>N/A</td>
<td>438g</td>
<td>2389g</td>
</tr>
</tbody>
</table>

Following the above analysis, concept 3 was identified as the most suitable solution. Figure 13 illustrates the distribution of stresses in the hub and upper attachment pieces of the rear suspension beam for the longitudinal load case, while Figure 14 illustrates the Tsai-Wu failure criterion within the first ply of the CFRP composite beam for this configuration.

**C. Design of Lower Trailing Arms Attachment Pieces**

Due to the increase in width of the rear suspension beam at the lower arm connection points of the new design, in order to maintain the same suspension geometry the lower arm attachment point must be internal. Having an internal attachment piece also prevents peeling at the adhesive bond, transmitting the loads mechanically into the beam. The design conceived for the lower attachment piece is illustrated in Figure 15. This design is relatively simplistic making it easily manufacturable.

**D. Manufacturability of Attachment Pieces**

In order to ensure that the above design concepts are manufacturable at UNSW Canberra, a meeting was organised with the workshop to discuss the designs. For both the hub and lower arm attachment pieces only minor modifications were needed, being primarily to do with the cutting tools available (i.e. the radius of certain curves). However, for the aluminium upper attachment piece a significant number of changes needed to be made due to its complex shape, and the fact that it is to be machined out of a single block of aluminium. The main design changes were the removal of the chamfers (using steps instead) as well as the angling of the sloped edges to ensure they could be machined with minimal setups (as cost effectively as possible). Figure 16 displays the top attachment piece after these design changes.

**V. Final Design**

The final design of the CFRP composite beam is illustrated in Figure 17 below. The base layer of the beam consists of twelve plies, all with an orientation of $[+45^\circ/-45^\circ]$. The base layer of the CFRP composite beam is estimated to have a weight of 2688g, with an additional 300g of reinforcement plies. The total weight of
the attachment pieces is 2389g. This gives a total assembly weight of 5377g, which is a 27% reduction in weight from the original design.

The beam is to be fabricated using a wet lay-up process. This will be conducted over a foam core for the lower section, with male moulds being used for the upper sections at each end. This will provide the accurate internal dimensions required for the fitting of the hub and lower attachment pieces. Both the hub and lower attachment pieces are to be manufactured out of 4130-chromoly, with the upper attachment piece being manufactured out of 7075-aluminium.

![Figure 17: CFRP composite rear suspension beam assembly integrated with the Academy Racing Team’s 2017 vehicle.](image)

**VI. Conclusion**

While the replacement of metal components with CFRP composite structures on the FSAE race car shows significant potential in weight reductions of the overall vehicle assembly, the integration of such components with other sub-systems on vehicle requires specially designed and manufactured metal attachment pieces. If designed inefficiently this hybrid assembly can negate the weight savings offered by the high strength-to-weight ratio of CFRP composite materials.

The use of CFRP composite materials in the fabrication of a new rear suspension beam is predicted to result in a 27% reduction in component weight. Being unsprung mass this has the potential to result in significant benefits in the traction and wheel control characteristics of the vehicle around corners and uneven surfaces through reducing the workload on the suspension system.

While the use of CFRP composite materials can be greatly beneficial, their mechanical properties and failure mechanisms are highly complex. As the CFRP composite rear suspension beam is a critical load bearing structure, before it can be integrated with the 2017 FSAE vehicle it will need to undergo extensive static and cyclic load testing in order to further validate the design and account for any defects that may have arisen during the fabrication process. Additionally, if the FSAE car is thought to have come close to either of the two maximum load cases discussed in this report during operation, the ART should carry out a close inspection of the CFRP structure for any signs of local failure.

**VII. Recommendations**

**Testing and Inspection:** Extensive static and cyclic load testing of the CFRP structure should be conducted by the ART before integration into the 2017 vehicle. Once in operation, the CFRP composite structure should be inspected at the end of each drive day for damage.

**Galvanic Corrosion:** As carbon fibre is a good electrical conductor, when coupled with steels and aluminium alloys they produce a large galvanic potential. As such the ART should inspect the fastened connection of the attachment pieces with the composite structure routinely throughout its lifespan, and consider using a sealant to prevent this galvanic coupling if these connections exhibit any signs of corrosion. Due to the rules-restricted life time of FSAE designs this is expected to be of minimal impact or concern.

**Programs used for FEA:** If the FSAE team conducts similar research into the design and development of composite structures it is recommended that they investigate the potential use of ABAQUS instead of CATIA for FEA. While the ART has traditionally used CATIA for its computational design and development, computational times, functionality and customer support of CATIA for such structures can be limiting.

**Future Projects:** A number of other candidates for replacement with CFRP composite structures were identified during the conceptual review of the FSAE vehicle including the; chassis, wheel rims and centres, turbo muffler, aerodynamic package and drive shaft. These present good concepts for future final year projects with the ART.

**Acknowledgements**

I would like to acknowledge and thank Steven Pender for his consistent guidance, support and time throughout this project. His genuine interest in my work and always open door was greatly appreciated and valuable resources. Furthermore I would like to thank all of those on the composites panel for their direction and feedback on my work, as well as the many other staff members at UNSW Canberra in both the engineering faculty and workshop that helped me along the way. Finally, I would like to thank Matthew Walker for his time and assistance with the Shimadzu universal testing machine and David Hamilton for his support with CATIAV5.
References

Academy Racing Team, *CatiaV5 working model*, University of New South Wales at Canberra, Australian Defence Force Academy, Formula SAE, 2016


Bagherpour, S., *Fibre Reinforced Polyester Composites*, Mechanical relationships in FRP composites, Islamic Azad University, Department of Materials Science and Engineering, Najafabad-Branch, Iran, 2012.


Dassault Systemes, “Tsai-Wu Failure Criterion”, Composite Failure Criteria, URL: http://help.solidworks.com/2013/English/SolidWorks/cworks/c_Tsai_Wu_Failure_Criterion.htm?id=a9681c84a1f474ab3eb858bf9c408e3#Pg0&ProductType=&ProductNames= [cited 24 June 2016].


**Annexes**


*Annex B*: Overview of the Formula SAE Rules on the Use of Composite Materials

*Annex C*: Requirements Analysis

*Annex D*: Load Case Calculations

*Annex E*: Finite Element Analysis of Current Rear Beam Design

*Annex F*: Preliminary Design Concepts and Analysis for the New Geometry of the Rear Suspension Beam

*Annex G*: Error Analysis of Young’s Modulus Calculations

*Annex H*: Fastener Testing – Additional Images

*Annex I*: Upper Trailing Arms and Hub Attachment Designs and Analysis
Annex A: Conceptual Review of the 2017 Vehicle

A. UNSW at ADFA FSAE Race Car

The Academy racing team competes in the Formula SAE Australasian competition each year. With the competition taking place in December, the CFRP composite component will be designed for implementation in the Academy racing team’s (ART) vehicle for the 2017 competition. The ART’s vehicle weighs approximately 182kg (without driver), and is broken down into four main sub-systems, being Primary Structures, Dynamics, Electronics and the Power Train. The heaviest components of each of these sub-systems will be investigated in order to identify the best candidate for replacement, offering the greatest potential increase in performance. Figure A.1 illustrates one of the Academy Racing Team’s previous FSAE race cars.

B. Paetro Principle

The Paetro Principle is a relation that describes causality and results, claiming that for many events roughly 80% of effects are a direct result of about 20% of the causes (Kiremire, 2011). This principle can be applied to a range of different fields, such as economics and business, in order to aid efficiency by focusing one’s effort. Similarly, the Paetro Principle can be used as an effective tool in identifying how to reduce the weight of a race car, with the heaviest 20% of components contributing to the vast majority of the vehicle’s total weight. Hence, in order to narrow down the investigation only the top 20% of heaviest components will be considered. These are listed in the following weight manifest.

C. Weight Manifest (Heaviest 20% of Components)

Table A.1: Heaviest 20% of Components

<table>
<thead>
<tr>
<th>Assembly</th>
<th>Component</th>
<th>Quantity</th>
<th>Weight (g)</th>
<th>Rotating</th>
<th>Un-sprung</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chassis</td>
<td>Chassis</td>
<td>1</td>
<td>35200</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Powerpack</td>
<td>Engine</td>
<td>1</td>
<td>32600</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wheels and Tires</td>
<td>Tyres</td>
<td>4</td>
<td>15463.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wheels and Tires</td>
<td>Rims &amp; Centres</td>
<td>4</td>
<td>8536.8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Suspension</td>
<td>Rear Beam</td>
<td>1</td>
<td>7349</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Outboards</td>
<td>Rear Outboards</td>
<td>2</td>
<td>5720.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Drivetrain</td>
<td>Diff Assembly</td>
<td>1</td>
<td>5049</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Outboards</td>
<td>Front Outboards</td>
<td>2</td>
<td>3988.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HMI</td>
<td>Pedal Box</td>
<td>1</td>
<td>3794.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear Bulkhead/Baseplate</td>
<td>Engine Baseplate</td>
<td>1</td>
<td>3712.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbo</td>
<td>Muffler</td>
<td>1</td>
<td>3055.8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brakes</td>
<td>Brake disc</td>
<td>4</td>
<td>2555.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HMI</td>
<td>Seat</td>
<td>1</td>
<td>2446.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rear Bulkhead/Baseplate</td>
<td>Rear Bulkhead</td>
<td>1</td>
<td>2421.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aero</td>
<td>Front Panel</td>
<td>1</td>
<td>2238</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fuel</td>
<td>Fuel Tank</td>
<td>1</td>
<td>2084.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aero</td>
<td>Side pods</td>
<td>2</td>
<td>1694</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HMI</td>
<td>Seatbelts</td>
<td>1</td>
<td>1632.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brakes</td>
<td>Brake &amp; clutch lines</td>
<td>1</td>
<td>1291.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chassis</td>
<td>Firewall</td>
<td>1</td>
<td>1284</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aero</td>
<td>RHS Cockpit Panel</td>
<td>1</td>
<td>1280.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Drivetrain</td>
<td>drive shaft (Right)</td>
<td>1</td>
<td>1254.4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>brakes</td>
<td>Rear Brake Callipers</td>
<td>2</td>
<td>1208.4</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure A.1: Previous model of the Academy Racing Team’s race car (Academy Racing Team, 2016).
When identifying the best candidate for replacement with weight-saving composite material, it is pertinent to note that not all weight can be considered “equal”. This is because the impact of a component’s weight on the vehicle’s performance is more complex than purely its physical mass, with both rotating and un-sprung weight having noteworthy effects upon the vehicle’s behaviour on the track.

D. Rotating Weight
Rotating mass has a significant impact upon acceleration and deceleration of a vehicle due to its rotational inertia. The mass moment of inertia, denoted by \( I \), is the extent to which an object resists rotational acceleration about a particular axis. In other words, the rotational inertia determines the torque required to achieve a desired acceleration of a rigid body about a rotational axis. Hence, the greater the rotating mass on a vehicle such as the FSAE race car, the greater the torque required to accelerate the rotating components to the required angular velocity, and the lower the overall forward acceleration produced by the power pack. Similarly the greater the rotating mass of a vehicle, the greater the force required to decelerate the rotating components, resulting in greater loading on the brakes. Consequently, decreasing weight from rotating components such as the wheels and tires has a more effective impact on performance than removing non-rotational weight, i.e. from the main chassis.

D. Unsprung Weight
Sprung weight is a term used for all components of a vehicle which are supported by the front and rear suspension, such as the chassis, engine, driver and power train. Meanwhile, un-sprung weight defines all components, such as the wheels, tires and rear axle assembly which are not isolated from the track surface. Unlike rotating weight, the greatest improvements from reducing un-sprung weight are observed in the handling of the vehicle. The lower the un-sprung weight, the less work the shocks and springs have to do to maintain the tires’ contact with the road over bumpy surfaces and around corners (Seas, 2016). This is because it is the inertia of the un-sprung mass which determines the workload on the suspension system, required to accelerate and decelerate the shocks and springs and maintain a consistent tire load. Less un-sprung mass will mean that the un-sprung components will move more readily in response to an imperfect track, providing a more constant grip with the track surface. Hence, reductions in un-sprung weight result in both greater traction and wheel control around corners and over uneven surfaces (Hillier, 2004).

F. Candidate Reviews
The above weight manifest was narrowed down to seven key components which each showed a reasonable potential for replacement with a composite structure, eliminating components such as the engine that while heavy, were not viable due to their complexity, cost or limitations within the rules. These seven components were the chassis, rims and centres of the wheels, rear suspension beam, turbo muffler, aerodynamic package (consisting of the front panel, side pods and RHS cockpit panel), fuel tank and drive shaft. These seven components were then individually reviewed and explored to identify the best candidate.

G. Candidate Selected
Main member of the Rear Suspension System. Following the conceptual review of the vehicle, the component identified as the best candidate for replacement with CFRP composite materials was the rear suspension beam. This is the main structural members of the rear suspension system connecting the rear wheels together laterally at the wheel hubs system and is the fifth heaviest component on the vehicle at 7349g. The beam has six independent connection points to the steel space frame chassis, with the location and triangulation of these connections being a key component of the vehicle’s suspension system. The main disadvantage of the current design is its weight, being over-engineered and made out of 1.6 - 5.0mm thick 4130 chromoly steel. Additional drawbacks are its disruption of the air flow under the vehicle and its interference with the design of a rear diffuser. Figure 8 shows an isometric view of the current rear beam design.

Figure A.2: Isometric view of the current beam design.
Annex B: Overview of the Formula SAE Rules on the Use of Composite Materials

The 2017-18 Formula SAE Rules is an extensive document which details a range of restrictions on materials to be used, manufacturing processes and test requirements throughout the design and fabrication of the Formula SAE race car. While the rules aim to give teams maximum design flexibility, they minimise the safety risk to those involved, and control the competition fairness across different university teams.

One of the most pertinent sections of the document is the Minimum Material Requirements (T3.4.1). This section details the baseline material requirements for a range of safety critical components of the car, using steel as a material standard. It is noted within this section that while the steel listed in T3.4.1 is the standard, alternative materials such as CFRP composites may be used throughout the vehicle, provided a Structural Equivalency Spreadsheet is submitted by the team where required. This entails calculations for the material chosen, demonstrating “equivalency to the minimum requirements for yield and ultimate strengths, buckling and tension, for buckling modulus and for energy dissipation” (T3.5.3). However, the use of composite materials within certain areas of the vehicle requires further testing and documentation.

“For use of any composite materials in the Driver’s Cell the team must:

a) Present documentation of material type, e.g. purchase receipt, shipping document or letter of donation, and of the material properties
b) Submit details of the composite lay-up technique as well as the structural material used (cloth type, weight, resin type, number of layers, core material and skin material if metal).

Submit a Structural Equivalency Spreadsheet as mentioned above.” (SAE International, 2016).

Similar restrictions apply for the fabrication of a composite space frame, requiring approval from their organising body and a range of test data proving either structural equivalency or obtaining a structural requirements certification, as per section T3.26. Furthermore, the use of composite materials in a monocoque design are specifically detailed in section T3.27, requiring similar documentation to the Driver’s Cell, as well as both 3-point bend and shear test data. Further information required for some sections include fibre orientation and lay-up technique.

Finally, for the Main Roll Hoop and Main Roll Hoop Bracing the use of any materials other than steel is strictly prohibited (T3.5.1). A list of all sections of the Formula SAE rules relating to the use of CFRP composites in the vehicle’s design is detailed below.

<table>
<thead>
<tr>
<th>Chapter</th>
<th>Paragraph</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>T3.4</td>
<td>T3.4.1</td>
<td>Minimum Material Requirements</td>
</tr>
<tr>
<td>T3.5</td>
<td>T3.5.1, T3.5.3</td>
<td>Alternative Tubing, Tubing Geometry and Materials - General Notes for all Applications</td>
</tr>
<tr>
<td>T3.8</td>
<td>T3.8.1, T3.8.2</td>
<td>Composite Materials</td>
</tr>
<tr>
<td>T3.26</td>
<td></td>
<td>Composite Space Frames</td>
</tr>
<tr>
<td>T3.27</td>
<td>T3.27.1, T3.27.2, T3.27.3, T3.27.4</td>
<td>Monocoque General Requirements</td>
</tr>
<tr>
<td>APPENDIX S-1-7</td>
<td>7.1, 7.2, 7.2.1, 7.2.2, 7.2.3, 7.3, 7.4, 7.5,</td>
<td>Cost Model and Cost Methodology - Composites</td>
</tr>
</tbody>
</table>

Key Deductions for the Development of the Rear Suspension Beam: As the rear suspension beam is not a component of the driver’s cell, cockpit, chassis or driver’s equipment, no structural equivalency spreadsheet (SES) as stated above is required. This means that no specific testing in order to determine the materials strengths and moduli are required. However, records of the fabrication process and materials used, including the lay-up, number of plies, curing operations, finishes and tooling must be submitted for justification in the Cost Report. Finally, all suspension mounting points must be visible at Technical Inspection, either by direct view or by removing covers.
### Annex C: Requirements Analysis

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Over-arching Requirement</th>
<th>Sub-requirement</th>
<th>Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>Integration with the 2017 FSAE Race Car</td>
<td>1.1 Suspension system. The new rear beam is to be compatible with the current design of the suspension system, providing connection points for the shock absorbers and trailing arms as per the current geometry, allowing for the triangulation and suspension characteristics to remain the same.</td>
<td>M</td>
</tr>
<tr>
<td></td>
<td>1.2 Wheel hub/beam interface. The new rear beam design is to employ a flat flange connection interface between the beam and the wheel hub. This is in order to avoid the re-designing and manufacturing of new wheel hubs.</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.3 Vehicle assembly. The rear beam design is not to interfere with other working components of the vehicle, particularly the drive train. The beam must be designed around the drivetrain components, with the driveshaft running through the centre of each rear wheel.</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.4 Adjustability. In order to properly tune the geometry of the suspension system, the connections of the trailing arms need to have a degree of adjustability. While the arms themselves can be adjusted along their longitudinal axes, the connection needs to have lateral adjustability.</td>
<td>D</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.5 Real estate allowance. The approximate spatial area available is a rectangular volume of 12000x400x200mm. This is provided that the beam is designed around the drive train system.</td>
<td>O (The real estate allowance does not all need to be used).</td>
<td></td>
</tr>
<tr>
<td>2.0</td>
<td>Geometric location of connection points*</td>
<td>2.1 Beam length (laterally from RHS wheel hub to LHS wheel hub): Y = 1200mm</td>
<td>M</td>
</tr>
<tr>
<td></td>
<td>2.2 Right rear shock absorber. X = 4.645mm, Y = 416.437mm, Z = -300mm</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.3 Left rear shock absorber. X = 4.645mm, Y = -416.437mm, Z = -300mm</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.4 Low-right trailing arm. X = -35.4mm, Y = 144.347mm, Z = 68.944mm</td>
<td>D (May be adjusted along the lateral (Y) axis, closer towards the centre of the vehicle.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.5 Low left trailing arm. X = -35.4mm, Y = -144.347mm, Z = -68.944mm</td>
<td>D (May be adjusted along the lateral (Y) axis, closer towards the centre of the vehicle.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.6 High right trailing arm. X = 0mm, Y = 450mm, Z = -300mm</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.7 High left trailing arm.</td>
<td>M</td>
<td></td>
</tr>
<tr>
<td>Section</td>
<td>Description</td>
<td></td>
<td></td>
</tr>
<tr>
<td>---------</td>
<td>-------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>X = 0mm, Y = -450mm, Z = -300mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.0</td>
<td><strong>Structural</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.1</td>
<td><strong>Lateral load case.</strong> A lateral force of 6008N and 'bump' force of 12017N, acting at the interface between the track surface and tire of one rear wheel.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.2</td>
<td><strong>Longitudinal load case.</strong> A longitudinal force of 3004N and 'bump' force of 6008N, acting at the interface between the track surface and the tire at both rear wheels.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.3</td>
<td><strong>Deflection.</strong> To optimize the performance of the suspension system and control of the vehicle, the rear beam is to be designed as a rigid structure, having minimal deflection. The beam should have no more than 20mm of deflection across the length of the beam, under the above load cases.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.0</td>
<td><strong>Aerodynamics</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.1</td>
<td><strong>Diffuser Integration.</strong> The geometry of the current beam design interferes with the development of a rear diffuser as part of the aerodynamics package. With the re-designing of the rear beam, one of the Academy Racing Team's requirements is to accommodate for the future development of a rear diffuser.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.2</td>
<td><strong>Aerodynamic drag of the beam.</strong> The beams interruption of the airflow through the rear of the vehicle should be limited, decreasing aerodynamic drag and increasing performance.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.0</td>
<td><strong>Weight</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.1</td>
<td><strong>Weight target.</strong> The weight of the current rear suspension beam design is 7349g. The target is a 40% weight reduction, with a total weight of 4400g for the new beam design.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Each of the coordinates provided as part of the geometric requirements refers to the reference system of the model in CatiaV5.  **The justification behind each load case requirement is discussed in greater detail in Chapter IV, paragraph C of the main report.
Annex D: Load Case Calculations

The lateral and longitudinal load cases for the analysis of the rear suspension beam were determined through using the following approximations, safety factors and scenarios.

For these load cases the weight of the vehicle was approximated as 300kg (with driver), with a safety factor of 7/6 added to this value to account for the uneven distribution of the mass across the four wheels and the assumption of no weight transfer as the vehicle undergoes acceleration (bringing the value to 350kg). The maximum coefficient of friction between the wheels and the track surface is 1.75 ($\mu_{\text{frict}}$). As such, any forces exceeding this value will result in the slipping of the wheels (rather than the transmission of the forces to the rear beam). To simplify this analysis the conservative assumption was made that the shock absorbers are rigid bodies, when in fact their dynamic response would reduce the loading on the beam. A “bump” force was added to each load case to account for the worst case scenario of the vehicle hitting a gutter during either scenario. This “bump” force is taken as the instantaneous vertical acceleration of the vehicle. Finally, a factor of safety (FoS) of 2 was added to the lateral, longitudinal and bump forces within both load cases.

Lateral load case. The maximum lateral load case occurs when the vehicle is cornering at high speeds, with all of the forces being transmitted through the wheels on one side of the vehicle (with the others having zero traction with the track surface). With all of the mass being distributed to two tires, each tire is assumed to take a load of 175kg. The maximum bump force is approximated to accelerate the wheels at 3.5g. Therefore the Lateral and bump forces can be calculated as follows, where $F_N$ is the normal force on the tires from the track surface.

\[
F_{\text{lateral}} = F_N \cdot \mu_{\text{frict}} \cdot \text{FoS} = mg \cdot \mu_{\text{frict}} \cdot \text{FoS} = 175 \cdot 9.81 \cdot 1.75 \cdot 2 = 6008.63 \text{ N}
\]

\[
F_{\text{B}} = ma \cdot \text{FoS} = 175 \cdot 9.81 \cdot 3.5 \cdot 2 = 12017.25 \text{ N}
\]

Longitudinal load case: The maximum longitudinal force on the beam occurs when the driver accidentally shifts down a gear, and both wheels grip the track surface at the same time. This force is backwards, with both tires under braking. In this scenario, the weight of the vehicle is approximated to be evenly distributed through all four tires, giving a load of 87.5kg on each wheel. Again, as 1.75 is the maximum coefficient of friction between the tires and the track surface, the maximum longitudinal load can be calculated as follows.

\[
F_{\text{long}} = F_N \cdot \mu_{\text{frict}} \cdot \text{FoS} = mg \cdot \mu_{\text{frict}} \cdot \text{FoS} = 87.5 \cdot 9.81 \cdot 1.75 \cdot 2 = 3004.32 \text{ N}
\]

\[
F_{\text{B}} = ma \cdot \text{FoS} = 87.5 \cdot 9.81 \cdot 3.5 \cdot 2 = 6008.63 \text{ N}
\]

A structural analysis of the current beam was then conducted using the above load cases. For the lateral load case the maximum stress within the beam was 115MPa (not exceeding the yield strength of chromoly of 460MPa), with a displacement of 0.26mm at the tire/track surface interface. However, for the longitudinal load case the maximum stress exceeded the yield strength of chromoly by a significant amount, reaching 637MPa with a displacement of 3.49mm at the tire/track surface interface.

While the longitudinal load case exceeded the yield strength of the beam, the FSAE team was confident that the current beam was over-engineered, and that this was due to having too large of a factor of safety on the original load cases. As such, multiple analyses were run at a range of different factors to determine the maximum longitudinal load case the rear suspension beam could withstand. This was found to be at a factor of safety of 1.4. As the FSAE team was confident that the current beam is structurally adequate, having been and tested and used, a factor of safety of 1.4 for the longitudinal load case was utilised for the new design. This gives the following loads:

\[
F_{\text{long}} = F_N \cdot \mu_{\text{frict}} \cdot \text{FoS} = mg \cdot \mu_{\text{frict}} \cdot \text{FoS} = 87.5 \cdot 9.81 \cdot 1.75 \cdot 1.4 = 2163.11 \text{ N}
\]

\[
F_{\text{B}} = ma \cdot \text{FoS} = 87.5 \cdot 9.81 \cdot 3.5 \cdot 1.4 = 4206.04 \text{ N}
\]
Annex E: Finite Element Analysis of Current Rear Beam Design

Figure 1 illustrates the setup of the structural analysis of the current rear beam design, replicating the triangulation geometry of the suspension systems trailing arms and shock absorbers, with the forces acting at the contact interface between the tire and the track surface. Figures 2-5 display the stress distribution and displacement of the rear beam under both the lateral and longitudinal load cases.

Figure 1: Setup of the structural analysis of the current rear beam design, modelled in CatiaV5.
Figure 2: Von mises stress analysis of the current rear beam design under the lateral load case.

Figure 3: Locations of high stress concentrations in the lateral load case.

Figure 4: Displacement analysis of the current rear beam design under the lateral load case.
Figure 5: Von mises stress analysis of the current rear beam design under the longitudinal load case.

Figure 6: Locations of high stress concentrations in the longitudinal load case.

Figure 7: Displacement analysis of the current rear beam design under the longitudinal load case.

Summary Report 2016, z3457299 Matthew Bell, UNSW Canberra at ADFA
Annex F: Preliminary Design Concepts and Analysis for the New Geometry of the Rear Suspension Beam
1. Rectangle beam with Cylindrical Heads:
   1.1 Longitudinal Load Case

   ![Image of rectangle beam with cylindrical heads, longitu

   1.2 Lateral Load Case

   ![Image of rectangle beam with cylindrical heads, lateral

2. Simple Rectangle beam
   2.1 Longitudinal Load Case

   ![Image of simple rectangle beam, longitudinal load cas

   2.2 Lateral Load Case

   ![Image of simple rectangle beam, lateral load case]

3.0 Simple cylindrical Beam
   3.1 Longitudinal Load Case

   ![Image of simple cylindrical beam, longitudinal load cas

   3.2 Lateral Load Case

   ![Image of simple cylindrical beam, lateral load case]
Annex G: Error Analysis of Young’s Modulus Calculations

The following uncertainty values were used for determining the overall uncertainty of the calculations of Young’s modulus:

Uncertainty in the force measured by the Shimadzu: ±0.5%
Uncertainty in Venier callipers: ±0.01mm
Uncertainty in displacement recorded by the Shimadzu: ±0.1% or 0.01mm (whichever one is larger)
Observed variation in ply thickness: ± 0.1mm
Observed variation in ply width: ± 0.3mm

Using these uncertainties, the total uncertainty for each value of Young’s modulus was calculated in Excel. An example of the uncertainty calculation is shown below:

Example Calculation: Test 1

Width = 14.51mm  Thickness = 1.09mm

Variation in ply thickness relative uncertainty = \( \frac{0.10}{1.09} = 0.0917 = 9.17\% \)

Thickness measurement relative uncertainty = \( \frac{0.01}{1.09} = 0.00917 = 0.917\% \)

Variation in ply width relative uncertainty = \( \frac{0.30}{14.51} = 0.0207 = 2.07\% \)

Width measurement relative uncertainty = \( \frac{0.01}{14.51} = 0.000689 = 0.0689\% \)

Total thickness uncertainty = 9.17% + .917% = 10.087%

Total width uncertainty = 2.07% + 0.0689% = 2.1389%

Area = width x thickness

Area uncertainty = width relative uncertainty + thickness relative uncertainty = 10.087% + 2.139% = 12.226%

Stress = Force / Area

Stress uncertainty = Force percentage uncertainty + Area uncertainty = 0.5% + 12.226% = 12.726%

Strain = \( \Delta \text{length} / \text{gauge length} \)

Strain uncertainty = 0.1% + 0.1% = 0.2%

Young’s modulus = Stress / Strain

Young’s Modulus uncertainty = stress uncertainty + strain uncertainty = 0.2% + 12.716% = 12.926%
Annex H: Fastener Testing – Additional Images

The following images are of the testing of the fastener attachment design, and the failure of the test panels. The loading on the CFRP composite panel and fastener connection pieces was also modelled in CATIA V5 at both 45° and 90°. The images below are as follows:

A. Testing setup for loading at 90°.
B. Testing setup for loading at 45°.
C. Deformation of the steel backing plate and washers at failure (18kN at 90°).
D. Deflection of the panel before failure at 45° loading.
E. Failure on the top of the panel during 45° static test.
F. Finite Element Analysis of a loading of 7.2kN at 45° in CATIA V5.
G. Finite Element Analysis of a loading of 18kN at 90° in CATIA V5.
Annex I: Upper Trailing Arms and Hub Attachment Designs and Analysis

The figures below illustrate a number of different design concepts and iterations undertaken during the development of the attachment pieces for the CFRP composite beam, as discussed in the Summary Report.