Carbon Fiber Monocoque Chassis Platform for Formula SAE and Formula SAE Electric Race Cars

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# Table of Contents

- **Introduction** 19
- **Background Information** 21
- **Design Requirements** 24
- **Scheduling** 28
- **Design Concepts** 30
  - Lead Design Concept 31
- **General Car Layout** 33
- **Concept Development** 34
  - Geometry 35
  - Working Concept 36
- **Final Design** 40
  - Nosecone Fairing and Impact Attenuator 40
  - Roll Hoops 41
  - Shoulder Harness Mounts 42
  - Firewall, Seatback and Headrests 42
- **System Integration** 44
  - Platform Overview 44
  - Compromises 44
  - Suspension Subsystem Integration 45
  - Aerodynamic Subsystem Integration 50
  - Driver Ergonomics 51
- **E-Car Specific Subsystem Integration** 54
- **C-Car Specific Subsystem Integration** 59
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Mounting</td>
<td>61</td>
</tr>
<tr>
<td>Monocoque Heat Study</td>
<td>69</td>
</tr>
<tr>
<td><strong>Stiffness</strong></td>
<td>74</td>
</tr>
<tr>
<td>2016 Analysis Synopsis</td>
<td>75</td>
</tr>
<tr>
<td>Cross Section Emphasis</td>
<td>76</td>
</tr>
<tr>
<td>Torsion Tube Analysis</td>
<td>78</td>
</tr>
<tr>
<td>Final Stiffness Driven Geometry</td>
<td>84</td>
</tr>
<tr>
<td>Suspension Contribution</td>
<td>85</td>
</tr>
<tr>
<td>Full Chassis FEM Analysis</td>
<td>85</td>
</tr>
<tr>
<td>Results</td>
<td>88</td>
</tr>
<tr>
<td>Torsional Stiffness Testing</td>
<td>88</td>
</tr>
<tr>
<td>Suspension Stiffness Test Fixture Design</td>
<td>93</td>
</tr>
<tr>
<td><strong>Hardpoints and Loads</strong></td>
<td>95</td>
</tr>
<tr>
<td>Prior Failure Analysis</td>
<td>95</td>
</tr>
<tr>
<td>Load Categorizing</td>
<td>96</td>
</tr>
<tr>
<td>Design Considerations</td>
<td>98</td>
</tr>
<tr>
<td>Out of Plane Loads</td>
<td>98</td>
</tr>
<tr>
<td>Potting Sizing Trends</td>
<td>101</td>
</tr>
<tr>
<td>Bolt Patterns and Attachments</td>
<td>103</td>
</tr>
<tr>
<td>Insert Testing</td>
<td>105</td>
</tr>
<tr>
<td>Finalized Insert Design</td>
<td>108</td>
</tr>
<tr>
<td>Full Tab Testing</td>
<td>109</td>
</tr>
<tr>
<td>Failures</td>
<td>111</td>
</tr>
<tr>
<td><strong>Impact Attenuator</strong></td>
<td>113</td>
</tr>
<tr>
<td>Section</td>
<td>Page</td>
</tr>
<tr>
<td>------------------------</td>
<td>------</td>
</tr>
<tr>
<td>Test Fixture</td>
<td>116</td>
</tr>
<tr>
<td><strong>Mold Design</strong></td>
<td>119</td>
</tr>
<tr>
<td>Preliminary Testing</td>
<td>119</td>
</tr>
<tr>
<td>Locating Features</td>
<td>121</td>
</tr>
<tr>
<td>Rocker Boss</td>
<td>121</td>
</tr>
<tr>
<td><strong>Laminate Design</strong></td>
<td>123</td>
</tr>
<tr>
<td>Core Selection</td>
<td>127</td>
</tr>
<tr>
<td>Staged-cure vs Single-cure Layup</td>
<td>127</td>
</tr>
<tr>
<td><strong>SES Development</strong></td>
<td>129</td>
</tr>
<tr>
<td>3-Point Bend Test</td>
<td>129</td>
</tr>
<tr>
<td>Perimeter Shear Test</td>
<td>130</td>
</tr>
<tr>
<td>Side Impact Structure</td>
<td>134</td>
</tr>
<tr>
<td>Front Bulkhead</td>
<td>135</td>
</tr>
<tr>
<td>Harness Attachements</td>
<td>146</td>
</tr>
<tr>
<td>Roll Hoop Attachments</td>
<td>141</td>
</tr>
<tr>
<td>SES Quirks and Limitations</td>
<td>142</td>
</tr>
<tr>
<td><strong>Manufacturing</strong></td>
<td>144</td>
</tr>
<tr>
<td>Plugs</td>
<td>144</td>
</tr>
<tr>
<td>Molds</td>
<td>153</td>
</tr>
<tr>
<td>Mold Repair</td>
<td>158</td>
</tr>
<tr>
<td>Rocker Boss</td>
<td>160</td>
</tr>
<tr>
<td>Test Layup</td>
<td>162</td>
</tr>
<tr>
<td>Layup</td>
<td>164</td>
</tr>
<tr>
<td>Core Forming Lessons Learned</td>
<td>171</td>
</tr>
<tr>
<td>Section</td>
<td>Page</td>
</tr>
<tr>
<td>---------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>Debulks</td>
<td>172</td>
</tr>
<tr>
<td>Cure</td>
<td>174</td>
</tr>
<tr>
<td>Post-Processing and Assembly</td>
<td>175</td>
</tr>
<tr>
<td>Drilling</td>
<td>185</td>
</tr>
<tr>
<td>Inserts</td>
<td>186</td>
</tr>
<tr>
<td>Temporary Aesthetics</td>
<td>188</td>
</tr>
<tr>
<td>Additional Touches</td>
<td>191</td>
</tr>
<tr>
<td>Fixes</td>
<td>195</td>
</tr>
<tr>
<td>Roll Hoops</td>
<td>198</td>
</tr>
<tr>
<td>Nosecone</td>
<td>199</td>
</tr>
<tr>
<td>AI/IA Plate</td>
<td>200</td>
</tr>
<tr>
<td>Seatback Mounting</td>
<td>202</td>
</tr>
<tr>
<td>Seatback/Firewall</td>
<td>203</td>
</tr>
<tr>
<td>Harness Mounts (Shoulder)</td>
<td>203</td>
</tr>
<tr>
<td><strong>Vehicle Testing</strong></td>
<td>205</td>
</tr>
<tr>
<td><strong>Competition Results</strong></td>
<td>207</td>
</tr>
<tr>
<td>Mass Props</td>
<td>210</td>
</tr>
<tr>
<td>Aesthetics</td>
<td>211</td>
</tr>
<tr>
<td>Recommendations</td>
<td>216</td>
</tr>
<tr>
<td>Conclusion</td>
<td>219</td>
</tr>
<tr>
<td>References</td>
<td>220</td>
</tr>
<tr>
<td>Extra Photos</td>
<td>221</td>
</tr>
<tr>
<td><strong>Appendix</strong></td>
<td>238</td>
</tr>
</tbody>
</table>
List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 1</td>
<td>The 2016 e-car undergoing testing.</td>
</tr>
<tr>
<td>Figure 2</td>
<td>The team lining up a tub and a subframe for joining.</td>
</tr>
<tr>
<td>Figure 3</td>
<td>The 2013 tub, Betty, in action.</td>
</tr>
<tr>
<td>Figure 4</td>
<td>Catie enjoying the sun at the 2015 Lincoln competition</td>
</tr>
<tr>
<td>Figure 5</td>
<td>Ji-lo during a testing session.</td>
</tr>
<tr>
<td>Figure 6</td>
<td>Daisy in action as the e-car at FSAE Electric 2016</td>
</tr>
<tr>
<td>Figure 7</td>
<td>Overall team schedule with chassis dates included.</td>
</tr>
<tr>
<td>Figure 8</td>
<td>Chassis manufacturing flowchart. The red lines indicate what will be done in parallel, once the first monocoque is laid up.</td>
</tr>
<tr>
<td>Figure 9</td>
<td>Man-hour study conducted on monocoque manufacturing.</td>
</tr>
<tr>
<td>Figure 10</td>
<td>Concept 1 for the monocoque. A basic nosecone shape is also included.</td>
</tr>
<tr>
<td>Figure 11</td>
<td>Concept 2, the rough packaging layout. Although very simple, it was extraordinarily helpful in determining the final dimensions of the monocoque.</td>
</tr>
<tr>
<td>Figure 12</td>
<td>Final monocoque concept presented at senior project CDR. Sharp corners are present in the model due to the inability to properly fillet the compound edges.</td>
</tr>
<tr>
<td>Figure 13</td>
<td>Cutaway view showing driver layout sketch and integration with the battery box and e-drivetrain.</td>
</tr>
<tr>
<td>Figure 14</td>
<td>Chassis assembly SolidWorks model.</td>
</tr>
<tr>
<td>Figure 15</td>
<td>Cockpit swoopy, front roll hoop, and steering wheel.</td>
</tr>
<tr>
<td>Figure 16</td>
<td>Pictures of rear rocker mounting and front boss</td>
</tr>
<tr>
<td>Figure 17</td>
<td>Exploded view of nosecone/IA assembly.</td>
</tr>
<tr>
<td>Figure 18</td>
<td>Fastener setup for AI plate.</td>
</tr>
<tr>
<td>Figure 19</td>
<td>Front and rear roll hoops</td>
</tr>
<tr>
<td>Figure 20</td>
<td>Roll hoop attachment assembly and bracket flat pattern</td>
</tr>
<tr>
<td>Figure 21</td>
<td>Seatback/firewall panel (green) supported by the bonded seatback flange (carbon fiber texture).</td>
</tr>
<tr>
<td>Figure 22</td>
<td>A side by side comparison between the combustion and electric vehicle platforms.</td>
</tr>
<tr>
<td>Figure 23</td>
<td>Camber and toe deflection plotted as a function of changing pickup location on the chassis. Kinematics and loads from the 2016 c-car are used.</td>
</tr>
<tr>
<td>Figure 24</td>
<td>Render of the front suspension geometry</td>
</tr>
<tr>
<td>Figure 25</td>
<td>Render of the rear suspension geometry</td>
</tr>
<tr>
<td>Figure 26</td>
<td>MDF fixture to locate chassis holes</td>
</tr>
<tr>
<td>Figure 27</td>
<td>Aluminum fixture for control arm geometry</td>
</tr>
<tr>
<td>Figure 28</td>
<td>2016 Nosecone and front wing. Air is forced down past the wing, separating the flow on the underside. The separation is shown as the dark blue area on the underside of the wing.</td>
</tr>
<tr>
<td>Figure 29</td>
<td>The ergonomics rig in action.</td>
</tr>
</tbody>
</table>
Figure 30  Driver layout sketch used as a reference for the monocoque model.

Figure 31  Finalized electric car packaging scheme. Image courtesy of the 2017 FSAE team.

Figure 32  A view of both the external of the accumulator (left) and its insides (right).

Figure 33  Renders of the E-Car with all subsystems packaged in the chassis.

Figure 34  A view of the current drivetrain configuration.

Figure 35  A cutout view of the e car packaging.

Figure 36  A preliminary graphic to show how much access is available through the seatback to work on components of the engine during a test day.

Figure 37  A cutaway of the c-car model gives a clear look at both the electronics packaging and the accessibility of the engine with the seatback removed. The electronics are the 4 boxes along the bottom of the chassis. From left to right they are the battery, the PDM, the ECU and the ACL.

Figure 38  A sectional view of the car from the side also shows the location of the electronics in the c-car relative to the seatback and fuel tank.

Figure 39  Roughly sketched ideas for mounting options for the engine in the rear of the monocoque.

Figure 40  Dynamic engine loads due to eccentric rotating engine components at 10k rpm. The graph can be read counter-clockwise around each approximately closed loop as a full engine rotation beginning at piston TDC at the top of the graph.

Figure 41  A plot of transmissibility ratio for different damping ratios and frequency ratio. For any damping ratio, a frequency ratio above 1.41 is necessary to have a transmissibility ratio that is less than 1.

Figure 42  Visual representation along with specifications for the vibrations isolators used for mounting the engine in the formula car.

Figure 43  Views of the engine upright and baseplates as the are positioned in the chassis. Bottom left: The rear base plate and uprights with the differential carriers included as well.

Figure 44  Photos from Go-Pro footage of the car during testing. The top row shows the engine bucking forwards (front mounts in compression and rear mounts in tension), and the bottom row shows the engine bucking backwards (rear mounts in compression and front mounts in tension). The middle row shows the engine in its nominal, no load position.

Figure 45  Side profile of the engine with the red line representing the location of the third engine mount from the top of the engine to the exhaust mounting bolt.

Figure 46  Cut-out section of the rear most engine mount insert after competition.

Figure 47  Sandwich panels cut to fit around the subframe and exhaust components.

Figure 48  Panels enclosing the subframe.

Figure 49  McMaster 595K16 temp-indicating sticker placed on a sandwich panel. Each white square will turn black once the corresponding temperature is reached.

Figure 50  Approximate locations of temp-sticker placement on the car.

Figure 51  Sticker near the oil pan showing exposure to 160-170°F.
Figure 52  Maxed-out stickers near the exhaust piping.
Figure 53  Unaffected stickers that were protected by aluminum tape.
Figure 54  Lateral load transfer distribution vs roll stiffness distribution for varying chassis stiffnesses. A chassis stiffness of 1500 ft-lb/degree achieves 88% of a rigid chassis.
Figure 55  The radius decrease at the front of the cockpit can be easily seen as the chassis floor in the front of the vehicle is significantly higher than the chassis floor behind the front wheel.
Figure 56  J-Lo (2015 e-car) chassis design
Figure 57  A picture of the geometry and constraint method for the torsion tube analysis. The ends were allowed to deform radially.
Figure 58  Convergence Graph of the torsion tube mesh.
Figure 59  Picture of chassis with low (left) and high (right) cockpit sidewalls.
Figure 60  FEA model and study results for the cockpit side profile torsional stiffness study.
Figure 61  The above image shows the chassis of the 2015 electric vehicle, noted for the high specific stiffness. The high cockpit wall are characterized by a series of stressed bulkheads throughout the section.
Figure 62  A view of the closeout curls on the cockpit opening on the torsion tube model.
Figure 63  FEA model and study results for the top view cockpit profile torsional stiffness study. The red portions represent the approximate locations of max deflection.
Figure 64  A graph showing stiffness as a function of uniform core thickness.
Figure 65  Finalized monocoque geometry.
Figure 66  Chassis and suspension geometry imported into Ansys as a surface and wireframe.
Figure 67  Contour plot of vertical deflection of the chassis model, with deformation scaled up to be clearly visible. The maximum deflection occurs at the front right upright, as expected.
Figure 68  Proposed concept for new chassis fixturing table/torsion test rig for Cal Poly Racing.
Figure 69  Chassis stiffness testing setup.
Figure 70  Rear hubs constrained for stiffness testing
Figure 71  Torsional Stiffness of E-car
Figure 72  Torsional stiffness of C-car
Figure 73  Two Force Member Test Fixture in Instron #1331
Figure 74  Three/Four Force Member Test Fixture in Instron #1331
Figure 75  Failure of a lower control arm mount on the 2016 tub. Note how the outboard edge (left edge in the left picture) has sheared through the skin, due to lack of reinforcement.
Figure 76  An example of core pan downs being used on competitor vehicles.
Figure 77  Comparison of specific compressive strength of aluminum honeycomb vs balsa.
Figure 78  Comparison of specific shear strength (1z) of aluminum honeycomb vs balsa.
Figure 79  Comparison of spindle insert (left) and flange insert (right).
Figure 80  The graph displays the increase in strength of an insert when increasing the insert and potting diameter. As is shown, in plane properties are very sensitive to potting radius, while out of plane strength are primarily driven by core replacements.

Figure 81  Example of placing the flange in line or on top of the faceskin. Each methods has advantages and drawbacks.

Figure 82  The above image shows the front suspension. Notice the three bolt patterns on the lower control arms.

Figure 83  The figure above is an example of the reduction factor on inserts based on spacing. As is shown, it is a linear relationship based on spacing and potting diameter. As the spacing to potting ratio reaches 10, there is no knockdown factor.

Figure 84  Core shear stress distribution in the vicinity of a potted insert.

Figure 85  The test fixture employed for shear testing. This was used to determine a value for in plane shear strength of the insert.

Figure 86  Typical failure mode of the small flanged insert tests. The primary mode is a bearing failure on the thin, unreinforced skins, as well as debonding of the potting resin from the insert.

Figure 87  Failure mode of the large flanged insert. You can see the dimpling in the core a distance away from the potting portion.

Figure 88  Load-displacement data from the round of insert testing.

Figure 89  The general insert geometry.

Figure 90  Failure mode of a suspension tab sitting on top of the inserts.

Figure 91  Failure mode of a suspension tab sitting directly on a padded up region of the laminate.

Figure 92  Overlay of strength parameters of the comparison pull test.

Figure 93  Rocker mount failure from the inside of the chassis.

Figure 94  Exterior image of rocker mount failure.

Figure 95  Cross sectional image of tub failure due to engine.

Figure 96  The 2016 impact attenuator undergoing crush testing.

Figure 97  A standard impact attenuator fitted to the 2015 e-car.

Figure 98  Specific crush strength characteristics for 5052 honeycomb of varying cell size.

Figure 99  The most efficient rules-compliant honeycomb, plugged into the spreadsheet.

Figure 100  Two of the purchased Plascore attenuators, after delivery. The pre-crushed surface is apparent on top.

Figure 101  Cutting the anti-intrusion plate testing fixture base on the Hangar plasma cutter. Even on the highest power and slowest speed settings cutting all the way through the 3/4 inch plate was a challenge. Slag still held the edges on after cutting all around and the circuit breaker on the plasma cutter tripped multiple times while cutting the plate.

Figure 102  The top of the anti-intrusion plate testing fixture after being tack welded in a couple places. The tubing was clamped to the frame table in the Hangar for the first set of tack welds to help prevent warping.
Figure 103  The fully welded anti-intrusion (AI) plate testing fixture. The tig welds can be seen at the joints, and were ground down on the top of the fixture to make sure the AI plate had a flat surface to sit on.

Figure 104  The anti-intrusion (AI) plate quickly setup on the AI plate testing fixture to get a quick visual confirmation that it will work for testing the AI plate.

Figure 105  The Baja CVT cover mold (left) and 2013 nosecone mold (right) used as plugs for the test molds.

Figure 106  Test mold surfaces fresh off the plugs. The carbon mold exhibited some pinholes due to the age and worn surface of the CVT cover mold.

Figure 107  Finalized plug geometry. The rocker boss is removed, holes have been added, and the cockpit opening has been filled in.

Figure 108  Short beam shear test setup.

Figure 109  Aftermath of an ASTM D3909 tensile test conducted on an M46J specimen

Figure 110  Staged-vs-single cure comparison by 3 point bending. The panels showed almost identical behavior.

Figure 111  FSAE Rules schematic of 3-point bend test setup. Dimensions in mm.

Figure 112  Perimeter shear test example force-displacement curve from the SES.

Figure 113  3 Point bend test of the 1010 low carbon steel

Figure 114  3 point bend test underway.

Figure 115  3-point bend test data for the [ 45c / 0 / 0c / core]s laminate with 0.5” core, with the calculated properties from the SES.

Figure 116  Perimeter shear test data for the [ 45c / 0 / 0c / core]s laminate with 0.5” core.

Figure 117  Perimeter shear test on a sandwich panel.

Figure 118  3-point bend data for the side impact structure laminate, with the calculated properties from the SES.

Figure 119  Perimeter shear data for the side impact laminate.

Figure 120  Our test fixture for the lab and anti-sub belt mounts. To be representative of the belts in the car, our rigid fixture clamps on the edge of a flat carbon sandwich panel.

Figure 121  Load test of the lap and anti-sub belt mounts.

Figure 122  Failure mode of lap and anti-sub belt mounts.

Figure 123  Shoulder harness bracket with mounting hardware.

Figure 124  First Shoulder harness mount test, after failure. The test shown here is representative of the original shoulder harness mount location on the chassis with respect to the distance of the mount from the edge of the panel. This test was not valid as we needed to test in a direction normal to the panel.

Figure 125  To test the shoulder harness mounts, we used old harness belts clamped in the instron jaws to pull on the mounts.

Figure 126  Revised shoulder harness mount test setup.
Figure 127 Pad-up for the right upper attachment of the main roll hoop during the layup. These local reinforcements were implemented in a similar way to those used for suspension points.

Figure 128 Comparison of two Side Impact Structure Panels with identical layups of [45c / 0 / 0 / 0c / Core]s and core with identical thickness and different compressive strengths.

Figure 129 The chassis plugs before sanding, covered in blue guide coat.

Figure 130 Sandling the flat surfaces with rubber sanding block. This surface is nearly complete. The edge of the flat surface (full white area) has been oversanded slightly, since it’s easy to put too much pressure on an edge by accident.

Figure 131 Flats almost done on the lower plug.

Figure 132 Curves done on the top half. The darker grey areas are where the plug has been sanded all the way through its primer and down to the foam, which is bad. The top half was separated from its base, which made this much harder to avoid.

Figure 133 Covering the bottom half with filler. Because we’re inexperienced, most of this has too much filler applied. The white filler should be barely visible. The earplugs are plugging the locating holes so they don’t get clogged with filler.

Figure 134 Fixing up an oversanded area on the parting line. A liberal amount of filler was applied and then carefully sanded down.

Figure 135 The nosecone before and during surfacing. The small area relative to the chassis meant that it went quickly.

Figure 136 The step between the plugs. The other side is identical. The step is most severe towards the front, and becomes barely noticeable at the rear of the chassis.

Figure 137 The plugs arrive from Zodiac in the back of Kevin’s swaggin waggon!

Figure 138 The nose cone plug in the paint booth after arriving from Zodiac.

Figure 139 Top half plug in the paint booth after arriving from Zodiac, and before being glued to its base board.

Figure 140 The bottom half plug just prior to being glued to its base board. Sometimes glue will make interesting patterns as it’s being applied.

Figure 141 Spreading the surface coat over the top plug.

Figure 142 1’ x 1’ squares of bulk cloth for the chassis molds after being wetted out and before being applied to the plug.

Figure 143 The bottom plug with all 4 layers of carbon and the core. The tiny white dots on the mold are the non-stick teflon pins extending through the drill bushings.

Figure 144 Vacuum bagging of the top half mold/plug while the resin cures. Multiple lines of tacky tape can be seen along the seams of the two vacuum bags used.

Figure 145 Top left: Chunking out sections of the stuck plug from the top half mold. Top right: Residue left on the inside of the top half mold after the plug was dug out of it. Bottom: The top half of the mold after being removed from the plug.

Figure 146 High-temp filler applied to defects/pinholes in the mold, before being sanded.

Figure 147 Machined and prepped plug. Alignment holes in the part were intended to align with chassis mold holes, with alignment pins cured inside the boss.
Figure 148  Bottom view of rocker boss. Note: only one alignment pin achieved its target location.

Figure 149  Top view of the rocker boss.

Figure 150  Fiberglass ply laid for the test layup.

Figure 151  Layup by region - c indicates woven fabric, otherwise unidirectional fiber is used.

Figure 152  Core thicknesses and types, by region. Local core reinforcements not shown.

Figure 153  Chassis team members laying up a 0 cloth ply for the outside skin on the top half mold of the c-car, making sure that the section was placed correctly and pressed tight against the previous ply to prevent air bubbles between the plies.

Figure 154  Chassis team members laying up a 0 cloth ply for the outside skin on the top half mold of the c-car. The pad ups for the roll hoop bracings can be seen at the bottom of the picture.. They are the raised circular sections in the carbon.

Figure 155  The bottom half of the c-car tub after a 0 ply of cloth has been applied throughout the entire half. Notice how the large flat porting of the mold lend to nice, smooth, and well packed down carbon while some fillets look like they could potentially have air bubbles between the plies.

Figure 156  Cut sections of core to match the patterns for relatively flat sections on the chassis for the c-car.

Figure 157  Core placement in the top half of the mold for the c-car. The white lines between sections of core is the foaming core splice that is extremely sticky and helps to fill gaps while bonding adjacent pieces of core together.

Figure 158  Core being formed onto the top half of the mold for the c-car. Trimmings of the core can be seen around the mold as team members modified the pre-cut core sections to fit exactly in the layup.

Figure 159  x left of the photo the red section of core is the denser fiberglass core used under the s: at the core with ahead of time.

Figure 160  Core being formed on the bottom half of the e-car tub. You can see the dense, red fiberglass core in two rectangles where the lap and anti-sub belts will be bolted to. Also, the blue film that is clearly seen on the side of and bottom of the tub is the film adhesive that we used to ensure a good bond between the core and the skins.

Figure 161  Core forming in the bottom half of the e-car chassis. You can see the section of fiberglass core down by the bottom of the picture that will support the brake pedal.

Figure 162  The method used for core forming and planning on the electric chassis. After difficulties forming core on the combustion core templates, core templates and partition locations were planned and simplified prior to the layup on the physical part. This significantly reduced layup time and increase part quality.

Figure 163  FEP placed on the bottom half of the c-car tub to prep for a debulk after the outer skin has been laid up on the mold.

Figure 164  Nathan leads chassis team members in placing padups on the bottom half of the c-car tub in between the 0 uni and the 0 cloth plies of the base laminate.
Figure 165 Chassis team members laying up a 0 ply of uni for the top of the c-car. As can be seen, multiple team members can be working on each half of the tub at the same time. So for about the first 15 people working on the layup productivity is directly proportional to the number of people in the composites lab.

Figure 166 In the Hangar, team member Phillip Coleman shows off the first part to be pulled from the molds: The top half of the c-car.

Figure 167 From the left: Kevin, Nathan, and Ford throw down a “we have the two chassis halves for the c-car ready for post processing and bonding” dab in the Hangar.

Figure 168 Kevin shows off the comfort, seating, and other ergonomics of the two freshly cured c-car chassis halves on the floor of the Hangar.

Figure 169 Cutting the cockpit and rear openings. The sharp edge of the paint is the cut line.

Figure 170 Kevin being extremely excited to start bonding the two trimmed halves of the c-car together in the Hangar paint booth. Meanwhile in the background Nathan checks the fitment of a bend main roll hoop.

Figure 171 The two trimmed halves of the c-car chassis jigged together and ready to be bonded together. The MDF and aluminum jigs are used to get the correct spacing between the two tub halves after the required trimming of the halves caused the parts to be about 1/8th of an inch shorter (each) than necessary to match the desired overall chassis geometry.

Figure 172 Shims are being used to get the correct spacing between the chassis halves before bolting the front jig on. Because the majority of the front bulkhead gets cut out, there were no locating holes on it to mount the aluminum jig to easily get the right spacing like the MDF jigs had on the sides.

Figure 173 With the chassis halves jigged together, and before bonding them, both the front and main roll hoops get mocked up to check their fit.

Figure 174 A rear view of the chassis with the engine being mocked up in it for the first time. This is after the halves were bonded together but before closeouts or the lap joint were added.

Figure 175 A top view of the engine being placed in the chassis for the first time. The seatback flange had not been bonded into the tub yet, making it easier to put the engine in than it was latter in the season.

Figure 176 A front view of the engine in the engine bay for the first time. The joint between the two halves is clearly visible as the white line of resin and microballoons around the chassis. But look at all that space!

Figure 177 A glamour tunnel shot of the engine in the chassis. To note though, you can clearly see the difference in trim and therefore gap height between the two sides of the chassis.

Figure 178 The engine with the differential being removed from the chassis through the cockpit. Without anything else in the chassis this was possible, but was too tight longitudinally for us to remove both the engine and the differential together once other components were mounted in the chassis (specifically the seatback flange and the steering column).

Figure 179 Front view of the engine being removed from the chassis through the cockpit. Side-to-side the engine and differential had more than enough room to be removed from the chassis together even once other components were mounted to the cockpit.
Figure 180  d shade 5 welding safety glasses, because you know... Safety 3rd.

Figure 181  Mocking up a rear rocker mount on the drilled insert holes while the seatback flange gets bonded in the c-car. The c-clamp and 8020 jig in the background is fixturing the seatback flange in place while the resin used for bonding it to the chassis dries.

Figure 182  Fully potted and cured inserts for the rear shock and rocker mounts on the left side of the c-car.

Figure 183  Both the e-car and c-car chassis sitting next to one another on the frame table in the Hangar. The c-car chassis has had resin and microballoons applied to cover some dry spots that were noticed in the chassis, making the bare carbon look much worse on the c-car than the e-car.

Figure 184  Top view of the two chassis next to each other on the Hangar frame table. The c-car chassis is farther along than the e-car chassis since it was the first one to be laid up and manufactured.

Figure 185  Vehicle progress two days before wheels on ground. The vehicles are nearly at the same completion point. Chassis lap joint quality difference can be easily seen.

Figure 186  Both cars in the Hangar after getting wheels on the ground. The quality in the carbon work between the lap joints of the two chassis is very obvious in this photo and is highlighted by the white resin and microballoons that were used on the c-car to smooth out the edges of the lap joint.

Figure 187  Both cars are displayed in ready to run trim at the unveiling event. Gaffers tape was used to cover up the poor aesthetics of the white resin and microballoons on the c-car chassis.

Figure 188  Aluminum tape was used on the side of the chassis for heat shielding along the section that was in close proximity to the exhaust. The wire leading under the aluminum tape connects to a thermocouple that was placed there to keep an eye on chassis temperatures. The beautifully blued stainless steel exhaust pipe and aesthetic gaffers tape can also be seen.

Figure 189  The second revision of the heat shielding aluminum tape on the car is shown here. The main difference between revision one is the aesthetics of it for photos. Revision one did it’s job just fine and so did revision two.

Figure 190  The inside of the cockpit of the c-car facing forwards. Gaffers tape was used along the lap joint edge to protect the driver from the rough carbon edge. On the floor of the chassis, around the driver’s legs, gaffers tape is also used to hold foam on top of the bolt heads and extra threads from suspension mounts to protect the driver’s legs and prevent the driver suit from getting caught during an egress.

Figure 191  The fully painted c-car chassis prior to having all the rest of the car’s components bolted back onto it. This photo is taken outside of the Bonderson high bay just before the Cal Poly Racing end of the year banquet.

Figure 192  Printer paper was used to mask off the freshly painted chassis to apply a ceramic paint coat over the area of the chassis that is close to the exhaust to help protect the chassis and paint.

Figure 193  The completed protective ceramic paint on the rear of the chassis.
Figure 194 Due to interference with the wheel rims, the front lower aft control arm mounts had to be moved forward on the chassis. New inserts for this were done on the c-car after the car was painted and before it was reassembled. To help keep spirits high on the team with all the long hours we were putting in, we had all the team members in the Hangar that night sign the bottom of the chassis.

Figure 195 Re-bolting up the c-car post painting and post potting in the second set of FLACA mount inserts. The front lower control arms were mounted on the old set of inserts to give the resin on the new inserts time to completely cure before taking suspension loads.

Figure 196 View of the welding position from inside the tub.

Figure 197 Tacking the main roll hoop braces while using the chassis as a jig to make sure that the roll hoop will fit correctly.

Figure 198 Off chassis welding setup to full weld the brackets onto the main roll hoop.

Figure 199 The main roll hoop braces during a fit check on the c-car in between grinding operations. The closer of the two braces still has the paper template taped on it to guide the grinding process.

Figure 200 The nosecone mold after being pulled from its plug.

Figure 201 A completed Impact attenuator/anti-intrusion plate assembly.

Figure 202 Bonded-on IA mounting bolt+plate assembly, viewed from inside the chassis.

Figure 203 Seatback flange pieces rough trimmed and on the scale before bonding into the chassis.

Figure 204 The seatback flange jig in action/ The 80-20 member extending forward is touching up against the front of the cockpit opening.

Figure 205 Finished shoulder harness mount on the scale.

Figure 206 Marvelous example of the negative impact cg has on performance.

Figure 207 Render of car paint and graphics.

Figure 208 Electric car being primed and painted.

Figure 209 Combustion car being painted.

Figure 210 Electric car paint being finished.

Figure 211 Sponsor stickers being applied.

Figure 212 Painted c-car on the left and e-car on the right.

Figure 213 The e-car displayed with aero, suspension, and drivetrain on the lawn by Spanos theater for Cal Poly open house. The rain was worrisome on the unpainted steel roll hoops and suspension links which needed to be thoroughly dried to prevent rust.

Figure 214 “Formula Bobsled” as demonstrated by Nathan and Carl (the team’s brakes lead) in the e-car. To get a quick check of static load and setup of the suspension, Carl sat in the battery bay and Nathan sat in the cockpit. We also contemplated pioneering the new collegiate challenge of Formula Bobsled...

Figure 215 The e-car out on the apron of the Hangar after the first Go-No Go for the car. The nosecones were still being manufactured, which lets you clearly see the AI plate and the impact attenuator.
Figure 216  Packaging and space on top of the engine bay. The main roll hoop braces (and the main roll hoop itself) have started to rust since they haven’t been painted yet.

Figure 217  Packaging and accessibility of the electronics, fuel tank, catch cans, and engine with the seatback removed. Although it isn’t the cleanest and prettiest to look at, everything is visible and accessible.

Figure 218  Steensma, the team’s assistant engine lead, working on the fuel tank with the c-car up on sawhorses. The large opening that was accessible with removing the seat back was incredibly valuable for all engine and engine related work.

Figure 219  The c-car after being completely rebuilt post painting. Ready for more testing!

Figure 220  “Ellie” as the c-car was known among the team with her hood stickers on. Shown front and center is the decal “Save the Manatee” which is an organization that a team member donated to in order to support the mascot for both the cars: the humble yet magnificent manatee. This mascot was adopted for the cars after our advisor, Dr. Fabijanic, made a comment that one of our preliminary chassis designs looked like a manatee.

Figure 221  The c-car after the full livery, including partner logos, had been put on. The right side of the car had different partners displayed than the left side. We made a big effort to make sure that the cars were aesthetically pleasing to help promote the overall team and improve our image to the public, Cal Poly, students, and our partners.

Figure 222  The left side of the c-car with full livery applied. Time and care were put into making sure that all the partner logos were fit well and be able to be seen while the car was driving. We also made sure that the partner logos looked good in their layout and were parallel to the main lines of the chassis when they were applied.

Figure 223  “Epiphany” as the e-car was known amongst the team. The names for both cars follow the traditional naming convention that the team has followed since Annie, the 2012 car. The naming convention follows that every new chassis year takes a name starting with the next letter in the alphabet and ending with an “ee” sound. We made sure to leave just enough space between the locating lip for the nosecone and the Zodiac partner logo to place the technical inspection stickers at competition. It was tight, but we did double check to make sure that they would fit before going to competition.

Figure 224  Both the c-car (on the left) and the e-car (on the right) are being shown off at the senior project expo. Both are in full livery (minus the final nosecone painting) and ready for final testing before competition.

Figure 225  Positioning both cars for a team photoshoot. If you look good and feel good, you’re going to perform good too.

Figure 226  Both cars with the subsystem leads of the team. The Fast and the Formula team members in the picture are Kevin (second from left), Nathan (tenth from left), and Ford (second from right).

Figure 227  Both cars with (essentially) the entire team. The Fast and the Formula team members in the picture are Kevin (sitting in the e-car), Nathan (standing row, thirteenth from the right), Mike K. (standing row, tenth from the right), and Ford (sitting in the c-car).
Figure 228  Ellie, the c-car, with all her technical inspection stickers at competition. With all four of the safety inspections passed, she's ready for the practice track and dynamic events! You can also see her weigh-in sticker verifying the front/rear and left/right weight distributions (48.2% front, and 50.2% left) along with overall weight (428 lbs).

Figure 229  Checking tire pressures on the c-car before sending it out to the practice track with our hot-shoe driver to try and workout some last minute aero issues before endurance.

Figure 230  The competition team on the last day of competition with our 9th place overall trophy, our 2nd place fuel efficiency trophy, the last few rays of sunlight, and a smiling Fabio.

Figure 231  Team lead, Adam, and c-car technical director (and The Fast and the Formula team member), Ford, receiving the trophy for placing 9th overall at competition.

Figure 232  Team lead, Adam, and c-car technical director (and The Fast and the Formula team member), Ford, receiving the trophy for placing 2nd in fuel efficiency at competition.
List of Tables

Table 1  Point sensitivities for the combustion car.
Table 2  Point sensitivities for the electric car.
Table 3  Combustion car point targets and metrics per event compared to the points achieved at competition in 2016.
Table 4  Electric car point targets and metrics per event compared to the points achieved at competition in 2016 as percentages of total points possible in an event.
Table 5  Results of the competition study.
Table 6  Decision matrix used to determine the style of chassis to design.
Table 7  Car parameters for both the c-car and e-car as provided by the Formula team.
Table 8  Chassis shape design guidelines
Table 9  Measured hub-to-hub torsional stiffness vs. desired torsional stiffness of historic chassis.
Table 10  2016 chassis failures.
Table 11  Cornering and braking loads into the different suspension links.
Table 12  Anticipated maximum vibrational engine loads at the mounting locations on the front and rear of the engine.
Table 13  Overview of available prepregs considered for use.
Table 14  Elastic properties obtained from tensile testing. Blank cells indicate untrustworthy or nonexistent data.
Table 15  Percent difference from tested to datasheet values
Table 16  2017 combustion car results and targets per event compared to the points achieved at competition in 2016.
Table 17  2017 Electric car results and targets per event compared to the points achieved at competition in 2016 as percentages of total points possible in an event.
Table 18  Combustion Car Chassis Mass Properties.
Introduction

Currently, the Cal Poly Racing team builds two complete cars each year to compete in the Formula SAE series. The Cal Poly Formula SAE Racing team is split into two integrated sub-teams with each sub-team specializing in each of the two powertrains of the two cars. One of the cars is powered by an electric motor and a battery pack (E-car) while the other car is powered by a traditional gasoline combustion engine (C-car). Each of these two powertrains offer a variety of challenges when it comes to the design of a chassis including motor/engine mounting, heat management, and fuel/battery storage.

The team has implemented a hybrid chassis design in the C-car for the previous four years. This hybrid chassis consists of a carbon fiber front tub mounted with a rear steel tube sub-frame. The E-car was recently adapted into this hybrid chassis design for the 2016 season. Although the current chassis design has been successful in accommodating both the electric and combustion powertrains, the Cal Poly Racing Team has identified the chassis as a system in need of improvement. Improvements to the manufacturability and increased emphasis on component integration were sought after. The Cal Poly Racing Team has tasked The Fast and the Formula senior project team with designing a new chassis platform that will accommodate both powertrains and serve as an adaptable foundation for future years. The goal is to have little to no differences between the C-car chassis and the E-car chassis.

Unfortunately, the 2016 racing team could only manage to manufacture a single carbon fiber front tub and a single rear steel tube subframe for the 2016 season due to complications that arose during manufacturing. To field two running cars during the testing season, parts of the 2015 c-car chassis were used. The new chassis proved to be a struggle, and took about 3 times longer to manufacture than had been scheduled. One of the most notable issues was the difficulty in releasing the tub from the mold. It was thought that low to zero draft angles were the primary cause but thermal expansion differences between the mold and the part had been suggested. Additionally, accurately trimming the two front tub halves and joining them together with precision proved to be difficult. The primary cause of this was the left-right mold-split two piece construction. The chassis lacked datums and flats which contributed to the difficulty during precise joining, while also making locating and mounting components a difficult, inaccurate and time-consuming task. Joining the tub and the subframe was also a long and imprecise operation, due to the same issues mentioned above. The width of the main
hoops on the two chassis differed by 0.25” making it impossible to interchange components as had been originally planned. These differences prove that the manufacturing process needs improvement in accuracy. Subsystems dependent on accurate placement suffered difficulties because of the inaccuracies present in the chassis.

Figure 2. The team lining up a tub and a subframe for joining.

Some other improvements to be made include the weight and stiffness of the car. The 2015 C-Car had a torsional stiffness of 1160 ft-lb/deg, just 68% of the 1700 ft-lb/deg design target. A better understanding of how to introduce loads into the chassis was needed in order to fully utilize the characteristics of carbon fiber composites to achieve a sufficiently stiff chassis with minimal weight. This goal will be reached by reducing the amount of structure required whenever possible and adding structure in high stressed locations as needed. The focus will be on the execution and production of the two new chassis platforms with the ideal chassis type being selected via our decision process.

The Cal Poly Racing team plans to build two complete new cars for the upcoming 2017 season. It is vital that the chassis is very manufacturable with more attention paid to precision, to ensure consistency between each chassis produced. The chassis must also accommodate both the electric and combustion powertrains efficiently with minimal compromises for either car.
Background Information

Prior to the 2016 year, the electric and combustion cars were built by completely separate teams, even though both cars were governed by essentially the same ruleset and share many components between them. The decision was made in 2016 to merge the two teams into a single entity, the idea being that most components could be shared between the cars, reducing the time and resources required to bring two competitive cars to competition. The combustion car’s chassis design was chosen to create a common "platform" between the two vehicles. The cars shared a carbon fiber front tub, with different steel tube rear subframes to accommodate the differing packaging, particularly the bulky accumulator for the e-car. Performance was sacrificed on the e-car in favor of simpler integration, and the e-car wheelbase was lengthened 4 inches to meet packaging requirements and weight distribution goals. The e-car subframe gained 8 lb over the c-car. These sacrifices were deemed acceptable for the first year of platforming, and the less competitive nature of FSAE Electric (only one team finished the Endurance event in 2015), with the expectation that the issues would be properly addressed the following year.

Cal Poly Racing has built up a wealth of chassis design and manufacturing knowledge since 2013, through three senior projects, a master's thesis, and four years of competition. These resources were invaluable for the 2016 season, and were primary references for our project.

Formula Chassis Works, the project that established the current design in 2013, set a solid foundation for the chassis and the team as a whole [1]. Even though technical issues hindered the performance of the 2013 car, Betty, the project produced a high-quality chassis, reintroduced composites to the team, and set a standard for extremely thorough documentation of the design and manufacturing process. Their design process, materials testing, and manufacturing processes are laid out in great detail, along with their suggestions for improvement. Additionally, the report contains records of past Cal Poly Racing teams from 2002 onwards.

![Figure 3. The 2013 tub, Betty, in action.](image)

Formula Monocoque Development set out in 2015 to improve upon the work of the FCW team, with the goal using their suggestions to drastically reduce weight and increase stiffness of the design [2]. They originally attempted to build to the Alternative Frame Rules in an effort to reduce weight, but found that the design wasn’t suited to the loading cases, and that it was difficult to get timely feedback from the judges. They focused on using higher-performing materials and removing weight wherever possible, most notably using aluminum honeycomb core and omitting film adhesive in the layup.
They also attempted to increase the temperature resistance of the tub by post-curing it. In the end, manufacturing was not fully considered, and the new tub, Catie, came out of the oven dry and with many bridging spots that required patching, eating into their planned weight savings. Additionally, core failures at the front rocker mounts occurred, resulting in two more repair patches. Nevertheless, despite many issues, the chassis was slightly lighter and stiffer than Betty, and served the team well for one competition year, as well as another year as a testing platform. Their report provides more useful documentation of the team's experience with chassis design and manufacturing.

Frame Engineering Associates designed the 2015 e-car chassis, reconfiguring from a steel space frame to an innovative “cut’n’fold” monocoque design [3]. The monocoque was a claimed 65 lb, much lighter than the e-car chassis, and split in half for removal of the accumulator. “J-lo” was designed under the Alternative Frame (AF) Rules, extensively using FEA to prove structural equivalency, instead of the time-consuming and costly material testing required by the standard Structural Equivalency Spreadsheet (SES) ruleset. The process of creating the very detailed FEA model required for AF was documented, and is a valuable resource for any FEA work that we do. Additionally, the report provides insight to how an exclusively electric car might be laid out.
Element Analysis” [4]. He developed a dynamic model in Abaqus that simulated the quasi-static crush test of the IA, and was able to correlate the model to within 5% for the physical test results. His thesis provides a detailed background on composite failure analysis using FEA, and suggestions to improve the design and reduce the weight of the Impact Attenuator.

An attempt was made to find student project reports of similar quality and depth from other universities, in order to get a broader perspective on possible methods to use, but any publicly available documents found were lacking in detail and polish.

Additionally, experience and knowledge gained from the 2016 competition year was used to the benefit of this project. The 2016 tub, Daisy, was a continuation of the progress made since 2013, with a large focus on manufacturing/layup improvements and increasing torsional stiffness. Through the use of core splice material and preformed core sections, there were no major manufacturing defects. The layup was completely redesigned to use mostly high-modulus uni, with only a single ply of cloth, resulting in a 21% increase in torsional stiffness, up to 1305 ft-lb/degree.

![Figure 6. Daisy in action as the e-car at FSAE Electric 2016](image-url)
Design Requirements

The driving motive behind a new chassis design for the FSAE team was to increase the points scored by the car, and therefore team at competition. The chassis was selected as an area to improve on over previous vehicles because it contributes a significant amount to the overall car weight, which has been historically high for the Cal Poly team relative to other top teams. To come up with parameters for our senior project to design with, the team first went about quantifying the effects on competition performance of certain vehicle parameters, such as weight and weight distribution, that could be affected by the chassis design. Then the performance gains were calculated for each event, static and dynamic, to determine a net approximate effect on vehicle score. This analysis allowed for a creation of sensitivities that showed how much a certain parameter affected total score. In turn this then allowed values to be chosen for different car parameters that could then be designed around. These ranges are assumed linear for simplicity, thus are only valid around the ranges set up for technical direction. Below are the sensitivities for the vehicles:

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<th>Accel</th>
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<th>AutoX</th>
<th>Endurance</th>
<th>Total</th>
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Table 1. Point sensitivities for the combustion car.

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Table 2. Point sensitivities for the electric car.

The team’s goal was to at least achieve minimum performance targets at competition for 10th place and 3rd place (for the combustion and electric cars respectively). In order to achieve these goals the
combustion car needed to get at least 690 points and the electric car needed to get 460 points (based on the points historically required to get 10th place and 3rd place at the car’s respective Lincoln competitions). This would mean a 90 point increase in the combustion car score and a 240 point increase in the electric car score from 2016. The technical direction then determined parameters that achieved the end result while staying within limited team resources. These parameters included car performance requirements, based on car characteristics, as well as methods for performing at a more rigorous standard in the static events (design report, cost report, business presentation).

<table>
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<th>Points Target</th>
<th>Times/Metric Required</th>
<th>Average Historical Lincoln 10th Place</th>
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Table 3. Combustion car point targets and metrics per event compared to the points achieved at competition in 2016.
Table 4. Electric car point targets and metrics per event compared to the points achieved at competition in 2016 as percentages of total points possible in an event.

<table>
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</tbody>
</table>

With these goals, it was determined that the chassis subsystem needed to be able to contribute 30 points to the overall vehicle points gain. This was expected to be achieved through weight reduction of the chassis by 25 lbs for the c-car and 30 lbs for the e-car, a cheaper cost for the chassis in cost report, better aesthetics for the business presentation, and a more thorough design for the design report. In addition the manufacturing time needed to be reduced by 50% per chassis due to the requirement to create one chassis per vehicle per year. The feasibility of all this and creating two chassis all had to be done in one year. Lastly the expected lifetime cycle of the tooling was required to be 5 years (or 10 parts pulled from the tooling).

With the team having determined what was required from the chassis to make the overall car appropriately competitive, they proceeded to give us our design requirements for the chassis senior project. These included that the chassis needed to accommodate both a combustion and an electric powertrain (for the combustion and electric cars that the team sends to competition) while at the same time serving as a base platform that could be adapted to fit future powertrain setups. Also, since the Cal Poly FSAE team primarily does in-house manufacturing, it is necessary that the chassis be quick to manufacture so that two chassis could be manufactured every year without being a major bottleneck for the production of the cars. The team also provided requirements to ensure that the chassis created would not neglect important features that would lead to a less suited chassis in future years. Requirements included easy accessibility to the powertrains and supporting systems so that maintenance and repairs could be carried out quickly and with minimal effort. Another requirement was that the chassis should be stiff enough to allow suspension tuning to be effective, and that the chassis must be aerodynamic at a minimum that it is not detrimental to any aero package on the car. Finally, we were given a $3,000 dollar budget (excluding sponsored materials/tooling/labor) to work with. From this we then added our own specifications that we would aim for our chassis to meet. We
took all of these requirements and specifications and determined which ones we would focus most intently on (and how we would address them) through the use of a QFD, presented in Appendix A. A weight goal of 70 lb was set for the main chassis assembly (frame/tub, roll hoops, Impact Attenuator and AI plate, firewall), a 30% reduction in weight from the 2016 c-car chassis, which weighed in at 97 lb. Along with this weight goal, a torsional stiffness goal of 1500 lb-ft/deg (hub to hub) was set, about a 16% increase from the 2016 c-car tested torsional stiffness of 1305 lb-ft/deg. This torsional stiffness goal was further broken down into a chassis torsional stiffness of 3300 lb-ft/deg. Other chassis goals included having a comfortably packaged driver, simplifying manufacturing, integrating both c-car and e-car components, and meeting FSAE driver template rules. Additionally, due to a very tight overall budget for the FSAE team for the 2017 season, saving money to the team was a top priority. It was decided that no part, material, or service would be purchased without first attempting to secure a sponsorship or donation for it.

We then changed our focus to our competition, the other FSAE teams. Our aim was to see if there was an indisputably superior chassis type that stood above others. Initial research focused on comparing successful FSAE teams who run both an electric and a combustion powertrain on a similar chassis platform (such as GFR, Munich, and Kansas). In an effort to understand possible driving parameters behind a winning car, a trade study was conducted between top teams at the 2015 FSAE Michigan, FSAE Lincoln, and Formula Student Germany competitions. General trends of successful cars were observed along with engine size (combustion) and chassis type. It was observed that a majority of top teams in the Michigan combustion event ran a four cylinder engine, with no particular preference toward chassis types. Only one top ten team ran with a monocoque-steel subframe hybrid. These trends held in the Lincoln combustion competition, although with an increased number of high scoring spaceframes. For the electric vehicle, German competition teams were the primary source of comparison due to the limited field of competition at the Lincoln event. The majority of teams at the European competition favored a monocoque structure, and most of the teams ran with aero devices. Additionally, the importance of the endurance event was observed to be a large point advantage for the winning teams. The percentage scored of all possible points at competition was found for each chassis type, and the results are presented below.

<table>
<thead>
<tr>
<th>Chassis Type</th>
<th>% of total points scored</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spaceframe</td>
<td>74.6</td>
</tr>
<tr>
<td>Hybrid</td>
<td>70.9</td>
</tr>
<tr>
<td>Monocoque</td>
<td>76.0 (combustion) 68.3</td>
</tr>
</tbody>
</table>

Table 5. Results of the competition study.

Although the monocoque held a slight lead over the other types, it was decided that the results were too close to draw any real conclusions. There are so many factors involved in making a high-scoring car, that it would be impossible to isolate just the chassis. It was also noted that teams with the resources to run a monocoque would likely have a well-developed car overall, and score higher anyway. It was also interesting that the hybrid design fared the worst out of the three.
Scheduling

The manufacturing schedule of the two chassis’ required by the team is a driving factor in the overall car build schedule, as many tasks, such as locating drivetrain and suspension components, cannot take place until the monocoque is complete. We worked closely with the management of the team to create a feasible manufacturing schedule that met the requirement of having both cars fully operational by March of 2017.

From here, we broke down the chassis schedule into more detail, focusing on how to combine as many operations as possible. It became apparent that once the first monocoque was laid up and cured, the trimming, closing out, and post-processing work could continue in parallel with the layup of the second monocoque. This was critical in allowing us to meet the team’s schedule.

We also conducted a study on how many man-hours each operation would require, presented below. It should be noted that our manufacturing team did not only consist of our senior project group, but the approximately 10-15 members of the Chassis and Composites subsystems of the team. We will almost always be able to field a workforce that is as large as is possible to crowd around a chassis. We
found that 3-4 people were able to work on one plug at a time when prepping them, and 7-8 people were able to work on each half of the mold for both the mold layups and the actual part layups.

Figure 9. Man-hour study conducted on monocoque manufacturing.

Throughout the season though our schedule changed a significant amount. This was due in part to many things, including receiving the molds later than expected from Zodiac Aerospace, adjustments in team set dates, lack of resources (specifically manpower) around midterms and finals, and other subsystems not being fully ready to bolt up when the chassis was completed. Because of this we moved to a more fluid, week to week type of schedule during the season while attempting to stick to our previous schedule but not deeming that absolutely critical. Even with this loose system of scheduling we were still able to complete of both chassis in time to test both the cars and bring them to competition.
Design Concepts

A number of design concepts had been suggested during early brainstorming with some being disregarded as being unreasonable or infeasible. Ultimately, the design team was able to narrow the options down to six different chassis types. The chassis types considered include: hybrid, carbon fiber monocoque (Mold), carbon fiber monocoque (Cut’n’Fold), SES steel space frame, AF steel space frame, and an aluminum spaceframe.

A weighted decision matrix was implemented in order to quantify how each chassis type met each of the design considerations. The design considerations included items such as: weight, stiffness, manufacturing time, ergonomics. Each of these design considerations was weighted and ranked with regard to priority. With the AF steel space frame as our control, we compared how each of the different chassis types met the design requirements relative to the control chassis type.

<table>
<thead>
<tr>
<th>Decision Matrix</th>
<th>AF Steel Spaceframe (Control)</th>
<th>SES Steel Spaceframe</th>
<th>Aluminum Spaceframe</th>
<th>Hybrid-Partial Monocoque</th>
<th>Full Monocoque Mold</th>
<th>Full Monocoque Cut and Fold</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Weight</td>
<td>Stiffness</td>
<td>Manufacturing Time</td>
<td>C-car Accessibility/Packing</td>
<td>E-car Accessibility/Packing</td>
<td>Ergonomics/Driver Comfort</td>
</tr>
</tbody>
</table>
|                 | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | 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Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unweighted | Weighted | Unw
Lead Design Concept

A full carbon fiber monocoque was selected as our lead concept as it was the most desirable option based on our weighted design matrix. Coincidentally, the lowest ranking chassis type was the hybrid chassis type which has been implemented in the C-car for the past four seasons. It was concluded that the main drawback to the hybrid design was a need to construct two very different halves of the chassis which requires knowledge and execution in the manufacturing needed for both of the chassis half types. An understanding of carbon fiber monocoque design is needed in addition to the steel space frame design in order to properly manufacture a hybrid chassis. These two separate halves had notable issues with joining, due to inaccuracies created during manufacturing. To prevent these issues, a one piece chassis can be implemented.

A full carbon fiber monocoque is a one-piece chassis eliminating the need for a separate rear steel subframe. The carbon fiber Cut’n’Fold chassis, which ranked second in our decision matrix, is also a one piece chassis design differing from the traditional molded monocoques by having only flat faces of carbon sandwich paneling, making any curved surfaces impossible. This issue along with the Cut’n’Fold’s higher cost (panel production would most likely need to be outsourced) made the molded monocoque the better option despite the molded monocoque requiring more time for manufacturing. The aluminum spaceframe was shown to be second from the worst chassis type considered with the drawbacks of increased manufacturing time and increased cost heavily outweighing the possible benefits of weight savings. The SES steel spaceframe performed slightly worse than the AF steel spaceframe (control) due to its weight drawbacks despite an advantage in rule adherence.

The chosen full monocoque design should understandably be approached with some caution. The team had previously attempted to design a monocoque in 2013, but not all issues, namely the rear section of the car, could be addressed in time. The engine was the elephant in the room, with problems regarding mounting, accessibility, and heat management. Too much time had to be spent laying the groundwork for a composite chassis. The result was the hybrid layout that Cal Poly Racing has used for 4 years, for which some problems have become apparent.

The joint between the two halves of the chassis was heavy and difficult to manufacture, and neither the tub nor the subframe had a good provision for joining the pieces of the chassis together. In 2016, joining the chassis halves took a full week, and they were still not properly located. The main issue, however, is that the chassis is built using two different construction techniques, and therefore requires expertise in both, and almost twice the manufacturing work. A chassis using a single manufacturing method would be quicker and easier to make.

So why not a steel spaceframe? In addition to the factors shown in the decision matrix, the team’s experience has shown that composite structures are lighter than the equivalent steel structure. The 2016 tub had an average weight per length of 0.59 lb/in, and the e-car subframe was 0.80 lb/in, excluding the main roll hoop (since a monocoque would have one anyway), 30% heavier per length. If the e-car subframe was “converted” to a composite, approximately 7 lb would be lost. While this isn’t quite an apples-to-apples comparison (structural requirements vary along the chassis) it shows the trend of a light composite structure. Additionally, there were concerns with welding a large portion of the chassis when building two cars with many other welded parts. Uprights, suspension arms, bracketing and mounting, intake, exhaust, accumulator container, and the pedal box are all welded parts, and it was foreseen that too much chassis welding would put these parts behind
schedule, thereby negating any manufacturing advantage held by a spaceframe. Composites manufacturing, although time-consuming, could run in parallel with welding operations.

After 4 years and 3 well-documented tubs, the team decided that the groundwork had been laid to pursue the full monocoque design. Composites sponsors and a steady material supply had been established. The biggest problem was the question of how to manage the engine and its accessibility, but this was a design problem and was to be treated as such.
General Car Layout

In order to have a competitive car, several key dimensions have to be met including weight, wheelbase length, track width, center of gravity height, and weight distribution. These values were determined by the team and are included in table 3. A shorter wheelbase has been verified to be quicker using the Formula team’s lap time simulation tool due to the car’s ability to turn in quicker and provide quicker direction changes. The majority of an autocross track is quick direction changes and tight turns where a car with a shorter wheelbase will have an advantage. Because the type of racing our car does, it does not need to be very stable and won’t benefit from a longer wheelbase like some high speed racecars would. Determining our ideal track width is a tradeoff between weight transfer and maneuverability. A car with a narrower track width will be easier to maneuver, especially through the slalom, but it will also increase weight transfer which will reduce ultimate cornering grip. This may be beneficial in that faster weight transfer makes the car react quicker. A lower center of gravity also benefits the car by reducing lateral weight transfer which allows the car to use more of the tires’ cornering grip. Although, a higher center of gravity does help in acceleration events. With all this being said, from a chassis perspective, meeting these overall car parameters is primarily a function of packaging. We need to be able to efficiently package components to allow for the small wheelbase length of 61 inches without raising the cg to high, moving the weight distribution too far forward or rearward, while decreasing the weight of the chassis and overall car as well.

<table>
<thead>
<tr>
<th></th>
<th>C-Car Target</th>
<th>C-Car Actual</th>
<th>E_Car Target</th>
<th>E-Car Actual</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Weight (Car)</strong></td>
<td>395 lbs</td>
<td>428 lbs</td>
<td>470 lbs</td>
<td>487 lbs</td>
</tr>
<tr>
<td><strong>Weight (Chassis)</strong></td>
<td>75 lbs</td>
<td>76.5 lbs</td>
<td>80 lbs</td>
<td>79.3 lbs</td>
</tr>
<tr>
<td><strong>Wheelbase</strong></td>
<td>61 in</td>
<td>61 in</td>
<td>61 in</td>
<td>61 in</td>
</tr>
<tr>
<td><strong>Track width</strong></td>
<td>47 in (front), 46 in (rear)</td>
<td>47 in (front), 46 in (rear)</td>
<td>47 in (front), 46 in (rear)</td>
<td>47 in (front), 46 in (rear)</td>
</tr>
<tr>
<td><strong>CG Height</strong></td>
<td>11 in</td>
<td>12 in</td>
<td>10.5 in</td>
<td>10.5 in</td>
</tr>
<tr>
<td><strong>Weight Distribution</strong></td>
<td>47-48% front</td>
<td>48% front</td>
<td>46% front</td>
<td>46% front</td>
</tr>
</tbody>
</table>

Table 7. Car parameters for both the c-car and e-car as provided by the Formula team.
Concept Development

Geometry

It was critical to end up with a chassis geometry that would meet all of our specifications for both cars, in order to have a successful platform. In particular, the most important requirement was that the chassis would provide an effective platform for both the combustion and electric cars. Weight was a close second, as the weight of the chassis is directly proportional to the surface area of the monocoque. Additionally, the shape has a large influence on the manufacturability of the monocoque. Complex curvature especially is very difficult to properly lay up. Finally, our goal of creating a platform for future years necessitated a geometry that would be easily adaptable to changes in suspension geometry and the like.

From these requirements and the template requirements laid out by the FSAE Rules [5], we formulated design guidelines for the chassis geometry, and began to create concept models. We went through 3 major iterations of varying complexity and completeness before arriving at the final design

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Design Guideline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effective Platform</td>
<td>Fits electric and combustion powertrains</td>
</tr>
<tr>
<td></td>
<td>Meets EV-specific rules</td>
</tr>
<tr>
<td>Weight</td>
<td>Minimize surface area</td>
</tr>
<tr>
<td></td>
<td>Minimize length</td>
</tr>
<tr>
<td>Manufacturability</td>
<td>Reduce complex curvature</td>
</tr>
<tr>
<td></td>
<td>Horizontal parting line to locate all suspension on one half</td>
</tr>
<tr>
<td>Adaptability</td>
<td>Large flat spots near suspension pickups</td>
</tr>
<tr>
<td>Ergonomics</td>
<td>Accommodate driver layout from team ergo study</td>
</tr>
<tr>
<td>FSAE Rules</td>
<td>Fit cockpit and driver templates</td>
</tr>
<tr>
<td></td>
<td>Size requirements for Side Impact Structure, etc.</td>
</tr>
<tr>
<td></td>
<td>Firewall rules</td>
</tr>
</tbody>
</table>

Table 8. Chassis shape design guidelines

The first shape developed was as much modeling practice as anything else. It was constructed in Solidworks using multiple lofts through the length of the chassis. The shape was primarily defined by
the cockpit templates, the Hagan driver template modeled from 2013, and the previous year’s front suspension geometry and battery box. Guide curves in horizontal and vertical planes were used to constrain the loft, and an extrude cut provided the cockpit opening area. Fillets were then added.

![Figure 10. Concept 1 for the monocoque. A basic nosecone shape is also included.](image1)

This model educated us on the difficulties of surface modeling and gave some insight on how best to approach the modeling process. We had included too much detail in the base loft itself, to where the loft was complicated and cumbersome. Additionally, it was far too swoopy to be of practical use.

Our second concept was created as a packaging study for the team PDR at the end of summer 2016. The goal was a simple, effective model that could be distributed and used by every member of the team to adjust chassis dimensions almost at will, as the overall car systems integration came. In light of this goal, it was as simple as possible, with one loft and many extrudes, so as to not be confusing for other team members. Most of the work here focused on the rear half of the chassis, as this was the area of most concern for packaging. First, the driver layout was modeled as laid out by an ergonomics study conducted by the team, with a seatback angle of 35 degrees and a nearly vertical steering wheel. From there, planes were defined corresponding to the front and rear axle lines, front bulkhead, etc. As the battery box, electronics, engine, drivetrain, and other major systems came into the model, dimensions were changed until all components comfortably fit.

![Figure 11. Concept 2, the rough packaging layout. Although very simple, it was extraordinarily helpful in determining the final dimensions of the monocoque.](image2)

This was much more helpful than the first model in moving towards a final monocoque geometry and was arguably the approach that should have been taken first. However, the first concept had provided valuable modeling practice, so we felt that the time spent on it was justified. Our final geometry would simply have to combine the practical packaging study and the cleaner lines of the modeling concept, as well as meeting the yet-to-be-developed suspension geometry.
**Working Concept**

This model was developed with the intent of being a fully functional “top-down” model, as dictated by the team’s modeling practices. The car master assemblies contain guide sketches and planes that define high-level car parameters, such as wheelbase, seatback angle, and ride height. A chassis master assembly was created, that contained matching planes, and the planes were mated between assemblies. In this way, the chassis assembly can talk to other system assemblies by way of the master assembly, while keeping individual parts and assemblies completely separate. In addition, all major car components are mated origin-to-origin in the master assembly, so that even when separated, all components are located at the correct coordinates. Other chassis-relevant planes were added, such as front and rear bulkhead planes. Then, the monocoque part model was created in the chassis assembly, with its own set of reference planes.

![Figure 12. Final monocoque concept presented at senior project CDR. Sharp corners are present in the model due to the inability to properly fillet the compound edges.](image)

The model primarily consisted of 8 individually-lofted sections, with extruded cuts to define the top surfaces of the front and rear of the car. Individually lofting the sections made it easy to make flat spots, and allowed us to tailor the transition between each section. However this method produced a monocoque that was somewhat ungainly and uneven. Even worse, the disjointed edges made it impossible for Solidworks to fillet the edges to produce a manufacturable shape.

Cutouts were made in the model for the cockpit opening, rear accessibility, and front accessibility. In addition to a large opening in the front bulkhead, a small opening was incorporated on the upper surface of the monocoque to allow servicing of the pedals and steering without removal of the impact attenuator.

It was decided that the main roll hoop would be placed on the outside of the monocoque, rather than the inside. This would get it out of the way of battery box removal through the cockpit, and allow it to be used to mount exterior components such as the radiator. Additionally, the hoop itself would be easier to assemble/disassemble, as it could simply be lifted off the car.
Figure 13. Cutaway view showing driver layout sketch and integration with the battery box and e-drivetrain.

In creation of the monocoque geometry, we were able to meet all of our design requirements by creating a short, compact shape with a large opening in the rear to accommodate drivetrain components and accessibility. It was also noted that the monocoque resembled the graceful and elegant shape of a manatee. Future work was dedicated to refining this attribute.
Final Design

Figure 14. Chassis assembly SolidWorks model.

From the working concept presented at senior project CDR, the geometry was refined to a final shape that was more aesthetically pleasing, and modeled more robustly to fix the errors encountered previously. Additionally, many interfaces with other subsystems on the cars were not mature at the time of senior project CDR, so geometry was adjusted to suit the completed car design.

Rather than using many lofted sections, just two lofts were used to define the basic shape, split roughly at the main roll hoop. This resulted in a smoother look and fixed the fillet issues. As before, extrude cuts were used to modify the shape. Variable fillets were used along the top edges of the monocoque to round out the shape. The cutout on the top surface of the monocoque was removed, as we were concerned about impacts on meeting SES requirements in that area and we expected to be able to quickly remove the nosecone and impact attenuator to adjust pedal position anyway.
Since the rules dictate that the steering wheel must be completely below the top of the front roll hoop, and we didn’t want to compromise visibility by making the monocoque excessively tall, we implemented the “cockpit swoopy”, a gently curved raised section in the center of the front of the cockpit that allowed the front roll hoop to rise above the steering wheel. Since generally drivers will be looking to either side of the car while turning rather than straight ahead, the raised center portion had little impact on visibility compared to raising the entire monocoque.

![Figure 15. Cockpit swoopy, front roll hoop, and steering wheel.](image)

The control arms and steering rack were easy to locate to the flat sides and bottom of the monocoque, but the shocks and rockers proved more difficult to integrate, especially in the front. The rear suspension’s pushrod actuation required that a face be created at the proper angle and position for the rocker mounts, which was accomplished without any major change of geometry or interference with other subsystems, as the face was essentially a large chamfer on the top edges of the rear of the car. The front suspension was pullrod-actuated, with the shocks placed as low as possible on the car. To obtain the desired motion ratio, the position of the pullrod and rocker needed to be such that the rocker and rocker mount interfered with the monocoque. The solution was to indent the monocoque locally to provide the mounting face for the rocker, thus the “rocker boss” was born. The cockpit template provided enough free space at the corners to implement the boss without having to increase the overall dimensions of the monocoque.

![Figure 16. Pictures of rear rocker mounting and front boss](image)
The rocker boss was designed to be as large as possible without interfering with the lower control arm mounts, with large radii to attempt to make manufacturing easier. However, this did not prove to be the case as discussed later in the manufacturing section.

**Nosecone Fairing and Impact Attenuator**

![Figure 17. Exploded view of nosecone/IA assembly.](image)

Once the design of the Impact Attenuator was chosen as a block of honeycomb, design began on the rules-mandated Anti-intrusion Plate (AI Plate) and the nosecone fairing. Although it is not a structural part, the nosecone is required to meet bodywork radius rules, as well as protect the impact attenuator and improve the aesthetics of the car.

The shape of the nosecone was a natural extension of the loft used to create the front portion of the monocoque, made to slope gently down and slightly in on the sides. We wanted a smooth transition between the nosecone and the monocoque, and a small reduction in size on the monocoque at the front allowed the nosecone to slide on and perfectly match the contour of the chassis. To fasten the nosecone to the monocoque, we used 4 ball-lock quick release pins, allowing the nosecone to be removed in seconds.

The anti-intrusion plate was also designed with easy removability in mind. The rules require 8 5/16” bolts to attach the IA/AI plate assembly to the monocoque, with steel backing plates sized based on laminate shear strength, similar to the roll hoop attachments. To meet these fastening requirements and keep the AI plate easy to remove, we welded the bolts to the backing plates, and bonded the backing plate to the inside surface of the front bulkhead, basically creating an integrated stud that met the letter of the rules. The bolts and backing plates still provide the clamping force and shear strength to properly fasten the AI plate, and the bonding holds them in place so that no wrenching is required on the bolt head. Additionally we used “keyholes” in the AI plate to allow the plate to slide off over the nuts. Instead of fully unthreading 8 locking nuts, it’s only required to loosen them a few turns, then lift and slide the AI plate off of the car.
To attach the Impact Attenuator to the AI plate, we planned to use a structural adhesive to bond the two together.

Roll Hoops

SAE rules specify that the car must have a main roll hoop directly behind the driver, and a front roll hoop in front of the driver. The main hoop must be made of 1”x.095” round steel tubing, while the front hoop may be made of the same, or an equivalent aluminum tube of any closed-section shape. Both hoops must be securely fastened to the chassis (governed by the SES), and the the option is given of laminating the front hoop into the monocoque instead. Lamination of the front hoop into the monocoque was initially considered as a way to eliminate 2 lb of brackets and potentially increase stiffness, but required more research and testing than was feasible in the scope of this year. The front hoop will be attached with traditional bracketing, but the monocoque will be designed to accommodate an integrated front hoop in the future. Additionally, a laminated hoop would mean a drastic change in how the steering wheel and dashboard would be integrated into the car; a welded dash attachment and steering column supports would be impossible. It is recommended that testing and validation of a fully laminated aluminum front roll hoop be conducted in future years.
The size and position of the main roll hoop is mainly driven by driver positioning and the “broomstick test” - A broomstick or tube resting on the tops of both roll hoops must be at least 2” away from the driver’s helmet. The main hoop braces must also maintain this 2” spacing. Placing the main hoop outside of the monocoque provided more space for removing the battery box, simplified installation and removal, and allowed us to weld mounts for certain components like the radiator instead of installing more inserts. The front roll hoop was similarly driven by driver positioning, as well as steering wheel position.

We wanted to make sure that the roll hoops would be easy to make and install accurately, so both hoops were bent only in one plane to simplify the bending process. All bend radii were chosen to be 3” to match the TubeShark’s bending die for 1” tubing.

The roll hoop attachments to the monocoque were sized based on the SES requirement of withstanding 20kN of load in any direction. Rules specify 0.080” steel for brackets and backing plates, and the use of two 5/16” bolts per attachment. The size of the bracket and backing plate is based on the shear strength of the laminate, as the bracket must have the perimeter required to withstand 20kN of shear. The hoop bracing attachments only use single 3/8 bolts, which is allowable per rule T3.39.5 and simplified the mounting.

Figure 20. Roll hoop attachment assembly and bracket flat pattern

Shoulder Harness Mounts

The harness mounts were designed to meet the FSAE T5.4 rule section while remaining as light as possible. Because of this we went with an aluminum bracket that was bolted to the top of the monocoque, behind the driver’s shoulders to achieve the required belt angles. The details of the shoulder harness mount design and iterations are covered in detail in the SES section of the report.

Firewall, Seatback and Headrest

Since the monocoque is essentially hollow, a structure was required to both protect the driver and support their back. FSAE rules specify that the firewall must protect the driver from all 3 forms of heat transfer - conduction, convection, and radiation. A sandwich panel does this well, as the low density core is a good insulator, as well as being lightweight. We designed the firewall and seatback to be one dual purpose sandwich panel, as there was no real reason to keep the two separate. This
gave us a lightweight firewall while still meeting rules. The seatback/firewall must be attached to the monocoque by some method that can safely bear the loads from the driver. Previously, the seatback was integrated into the front tub of the c-cars, and J-lo, the late 2015 e-car, used L-brackets to attach it to the walls of its monocoque. Our solution was to bond on a flange inside the monocoque, that the seatback/firewall would rest on. This has a couple distinct advantages over the L-bracket design. First, the seatback itself can be made significantly narrower than the actual monocoque, making it smaller which speeds up removal and installation. Second, the flange is an easy interface on which to place some sort of weatherproofing/gasketing material to further seal the rear bay of the car from the driver.

![Figure 21. Seatback/firewall panel (green) supported by the bonded seatback flange (carbon fiber texture).](image)

The headrest must have energy absorbing foam, and withstand up to 200 lbs horizontal load per the rules. Again, a sandwich panel is a great choice as a support for the foam, since any load fill be distributed over a large area by the headrest foam. Simple struts welded to the main roll hoop provide a secure mounting for the firewall assembly.
System Integration

Platform Overview

One of the more difficult parts of developing the platform was integrating subsystems into the chassis without significant compromise between the two vehicles. Because the primary difference between the two cars is the powertrain, the rear of the monocoque (the engine and battery bay) was the main area of focus in platforming. However, in order to hit each vehicles target dynamic parameters, such as wheelbase and weight distribution, we first needed to explore positions of different components to determine if a common chassis platform would even be feasible.

In a brief summary, after iterations with spreadsheets that included the CG location of substantially heavy components (including the driver) and basic block models with volumes of components, it was determined that our target vehicle parameters could be achieved on a common platform. This did result in some compromises as expected, but none of them were significant enough to prevent us from developing this common platform. Due to the nature of the competitions, with the combustion competition being more competitive than the electric competition, we decided that compromises were necessary that would benefit the combustion car over the electric car. As we went through the design phase, no substantial conflicts between the vehicles remaining in the same chassis were observed.

Figure 22. A side by side comparison between the combustion and electric vehicle platforms.

In terms of system integration of the vehicles, the primary advantage is that all common subsystems could be designed once, and then manufactured twice. With the use of common tooling, manufacturing processes, and a streamlined design phase; we believe we can produce two higher quality vehicles than if we developed the vehicles individually. By keeping compromises to a minimum, this platform will allow subsystems to focus on one design instead of two.

Compromises

Before detailing how each subsystem fits into the chassis, we wanted to address the compromises required to fit the vehicles on one platform. While major compromises were able to be avoided through effort being put into the vehicle architecture early in the design phase, that does not mean that the platform was completely without any conflicts. While a majority of these conflicts were very small and likely will have no bearing on how the vehicles perform, we still believe it is important to address these issues to understand the platforms limitation.
A primary packaging issue was how the vehicles managed to hit their weight distribution goals. The combination of component locations that was found to work best between the vehicles was placing the driver significantly in front of the center of gravity (5 inches on the combustion car and 9 inches on the electric car) in order to counteract the heavy rearward bias of the electric car and the comparatively light powertrain components of the combustion car. Fortunately, it was determined through lap simulators that the electric vehicle favored a more rearward center of gravity, which made the distribution much more achievable. The combustion car ended up having to move the engine and exhaust as rearward as possible to meet its weight distribution goals. The electric car had to move the batteries and motor as far forward in the rear bay as possible. Even with these packaging tradeoffs, both vehicles were still able to achieve their performance goals without deal breaking compromises. While we did not compromise in weight distribution between the vehicles, the moment of inertia between the vehicles will be affected by this decision. As of the moment this decision was made, we had no way to quantify how this parameter affects performance. Additionally, there was significant conflict among advisors to how much vehicle inertia actually matters, so we swept the discussion under the rug in favor of meeting static weight distribution goals.

Another topic of interest in offsetting the driver from the center of gravity is how the drivers weight can affect the vehicles properties. Inherent in moving the driver significantly off the center of gravity, the weight distribution of the vehicles are very sensitive to the driver’s weight, with this property significantly more pronounced in the electric vehicle. Our team reasoned that this could likely be beneficial as the sensitivities showed that vehicle properties can be contradictory based on event. For example, placing a smaller driver in an upright position for the acceleration event would increase both the rearward and upward positioning of the center of gravity, allowing for more weight transfer to the rear wheels and increasing that events scoring. A heavier driver in endurance and autocross, in which driver selection tends to focus on skill over weight or height, will results in a more balanced vehicle. Driver weights and desired center of mass locations per event were kept in mind throughout the vehicles design phase.

Looking back after the design phase, the only tradeoff between the vehicles that could not be rectified was the chassis width and height behind the driver. Because of the necessity to fit two different powertrains while maintaining performance parameters, the floor of the chassis was slightly larger than desired for the combustion car in order to fit the batteries of the electric vehicle. Even though the rear geometry will result in a very slight weight penalty for the combustion car, it quickly became apparent throughout the year that more space was well worth the extra fraction of a pound in added surface area. Without the added accessibility that comes with the larger rear bay, servicing engine and drivetrain components would have been significantly more time consuming. Additionally, the wider chassis also justifies the potential use of a larger engine in future years without necessitating a chassis redesign and remanufacturing of tooling, achieving a more modular chassis setup that could be used for longer without major compromises.

**Suspension Subsystem Integration**

The largest shared subsystem between the two vehicles is the suspension subsystem. Not only did this subsystem have a large effect on performance, but the quantity of components within it made expedited design and implementation a high priority for quality vehicles. Chassis’s job in helping suspension is to make the integration for its many connection points as seamless as possible, while balancing suspensions requirements with other subsystems.
The interface between suspension and chassis was identified as an area that needed the most improvement in the vehicle. On several occasions the carbon tub failed where the control arm mounts meet with the chassis, which caused unnecessary downtime as well as an increase in weight in that area after it was repaired. Many of the reinforcements of last year’s chassis were incorrectly placed, and the suspension mounting points on the chassis were curved instead of being flat, making it difficult to interface with. While composite strength and bolt patterns are addressed in detail in a later section in this report, the chassis geometry in the attachment regions stands to make major improvements. The primary improvement we identified was the chassis geometry near the attachment regions. Through allowing broad flat regions near the front and rear suspension links, this allowed greater flexibility and easier implementation of suspension attachments. These flat regions also simplified hard point design, and made it easier to design and implement solutions.

One major change we are implementing in the vehicle is a trailing axle design where the differential is mounted such that it hangs out the back of the tub, thereby allowing the chassis to be shorter. A shorter chassis is a lighter chassis and by hanging the diff out the back of the monocoque we saved 3.3 lbs of chassis weight. Additionally, opening up the rear of the vehicle gives an incredible amount of accessibility to powertrain related subsystems, which is a key issue in moving to a monocoque. A trailing axle also allows sprocket placement to not be limited vertically by the chassis floor, which can be used as a valuable method to reduce vehicle vertical center of gravity. Despite its benefits, one of the worries we had with running a trailing axle was that the suspension loads may increase to the point the benefits would be outweighed by heavier suspension links and mounting points. However, after running the numbers, the suspension only had a gain of 1.2 pounds, partially from the fact that the suspension stiffness goal was higher this year than previous years, so a weight gain for stiffness was necessary anyway. Also, a quick compliance study was done to quantify changes in toe and camber when the suspension points on the chassis were moved forward. As can be seen in the graphs below using the same outer diameter and wall thickness for the a-arms, compliance increased exponentially the further the pick up points were moved forward, especially in the more important toe direction. By only moving the the suspension points forward 3 inches relative to the upright, we were able to avoid major compliance increases.
Figure 23. Camber and toe deflection plotted as a function of changing pickup location on the chassis. Kinematics and loads from the 2016 c-car are used.

Integration with the front suspension links was determined to be critical to vehicle performance and required the most effort in the chassis design to accommodate. The lower control arms are mounted to the chassis floor, as far towards the vehicle centerline as possible. This not only is favorable from a suspension kinematic standpoint, but it also converts the highly loaded joints into in-plane loads in the laminate. The upper control arms had little choice but to be mounted directly to the side of the chassis, as little design freedom was allowed there. The front shocks are mounted underneath the car to reduce the center of gravity of the suspension system by .1 inches. While this did increase link loads, it was determined to be manageable. This pullrod configuration required a groove in the chassis to be made to allow the rocker to sit underneath the chassis. The rocker attaches to the upper a-arm via a pull rod configuration. An image of the configuration is shown in figure 24.
The rear suspension integration was a compromise between kinematics and packaging components in the rear bay. From a kinematics standpoint, just like the front, the closer the control arm mounting points to the centerline the better. Unfortunately, the rear outer surface of the chassis was driven by packaged components accessibility. Fortunately, the suspension team was able to come up with a solution that works well with the given constraints. The solution was to mount the rear shocks high on the sides of the chassis. The profile of the upper portion of the chassis was designed to match the angle the rocker needs to be at to prevent out of plane loading in the rocker and chassis, as well as enable the desired motion ratio. To reduce loads and reduce the number of mounts, the aft part of the rear a-arm and the rear toe link are mounted to the same bracket at the back bulkhead of the chassis. A render of the rear suspension is shown in figure 25.
Another issue seen in prior vehicles was a lack of tolerance control on the suspension to chassis interface. To locate the suspension mounts, several holes were designed into the chassis mold to serve as locating features. In order to locate the remaining holes properly the fixtures made out of MDF were created. Steel inserts were pressed into the plate to provide accurate drilling locations. These fixture plates were then bolted to the side of chassis, utilizing the prelocated points in the mold, and all critical holes on a plane were drilled out at one time. This proved a quick and easy way to accurately locate a majority of components. Similar methods were utilized on locations such as the rocker boss.

Figure 25. Render of the rear suspension geometry
To insure the control arms lined up nicely to the holes in the chassis, the suspension team created a fixture out of CNC’d aluminum.

**Aerodynamic Subsystem Integration**

Early on the design process, the decision was made to design the chassis around a full aerodynamics package, including front and rear wings, an undertray, and possibly some form of sidepods. For two years in a row the team has attempted to run an aero package, but the package never made it to competition because of questions of how it affects the performance of the car in testing. With the benefits of aero on the vehicles finally being more thoroughly understood, a vehicle without an aero package did not seem likely.
Our goal was to make it as easy as possible to create a competitive aero package and properly integrate it with the rest of the car. To do this, we had the aerodynamics team identify several areas of improvement based upon the 2016 car that would improve the aerodynamics of the vehicle. Most notable was the interaction of the nosecone with the front wing. The shape of the nose, combined with its proximity to the wing, resulted in the nose redirecting airflow from the top of the wing downward, causing flow separation on the lower trailing edge, reducing the effectiveness of the wing. We used this feedback to design the nosecone in tandem with the front wing to reduce the flow separation on the underside of the wing from wing-nosecone interaction.

**Figure 28.** 2016 Nosecone and front wing. Air is forced down past the wing, separating the flow on the underside. The separation is shown as the dark blue area on the underside of the wing.

Another issue seen in 2016 was the difficulties encountered in mounting the aero package. Elaborate mounting designs were conceptualized that added weight, were time-consuming to manufacture, and were slow to take on and off of the car. The new chassis sought to improve front wing mounting by keeping the front sides of the chassis longitudinally parallel. While doing this was beneficial, the biggest issue that the aero package ended up facing with regards to its interaction with the chassis was the draft angle on the vertical walls of the tub. While a more elaborate solution probably could have been engineered, the aero team ended up using spacers between their inner endplate mounts and the tub which proved to be difficult to work with when mounting and unmounting the front wing. Unfortunately this happened almost every time that the car was transported to and from testing. Plenty of work still be done with the aero-chassis integration for future years to improve the aerodynamics of the overall car as well as improving the mounting of the aero to be easier and quicker to attach or remove. This could potentially save hours of testing time which is invaluable to the team.

**Driver Ergonomics**

Ergonomics are a key area where the new chassis has to excel. A car with well-designed ergonomics allows the driver to communicate exactly what he wants the car to do, without any unnecessary thoughts, movements or barriers to make it more difficult. Of course making the car as small as possible to minimize weight is crucial for a competitive car, but if the driver is not comfortable, his/her focus and overall ability to make quick decisions will be decreased causing slower lap times. To ensure the driver is comfortable we have to design the chassis around the driver. We did this by first outsourcing the production of an adjustable ergo rig to the FSAE team. In last year’s car, although it had a lot of room, there were still a few areas that need to be improved, such as providing more elbow room for the driver to enable a full range of motion lock to lock. The limited elbow room
as well as a steering wheel angle far from vertical contributed to giving the driver difficulty communicating exactly what he/she wanted the car to do.

Figure 29. The ergonomics rig in action.

Another area we have to look into is pedal placement. Last year’s pedal placement was catered to taller drivers making it difficult for shorter drivers to feel comfortable while driving the car. Designing a pedal box that suits a wide range of drivers and provides good positioning of the feet is crucial. The rig allowed us to play with various seatback angles, pedal distance and angles, and steering placements to see how low we can place the driver and determine the location of controls while still keeping the driver comfortable enough. What we ended up getting out of our ergo rig was that a seat back angle of 35 deg was both comfortable to drivers as well as it helped lower the driver’s CG and reduced driver frontal area to help with aero. Also by working with the driver controls team we were able to achieve a more comfortable steering wheel angle of 15 deg off vertical which allowed for more comfortable steering and required less elbow room for the driver to feel comfortable.

Figure 30. Driver layout sketch used as a reference for the monocoque model.

One thing that was difficult to test using the ergonomics rig but is crucial is the seat design. A proper seat design allows our driver to use his/her arms and legs to control the car, instead of bracing themselves during high acceleration forces. We were able to leave enough room in the cockpit to
allow the driver controls team space to put in a bead seat that is formed around the driver to give appropriate bracing between the driver’s torso and the cockpit walls.

Once the car was manufactured and we began testing, we were able to get valuable feedback on the ergonomics from our drivers. The biggest issue that our drivers experienced was that the pedal position was a little too close for taller drivers which meant that their legs were bent and their knees would hit the steering wheel in turns. This was also in part due to the somewhat wide, oval shape of the steering wheel which could have been trimmed down to avoid this issue as well as the farthest pedal position could have been extended slightly too. The other ergonomics complaint that our drivers had was that the driver’s upper torso was poorly constrained laterally. This was primarily a result of the bed seat not adequately filling the space between the tub and the driver, and with more seat material could probably be easily solved. Other than these two issues the drivers were happy with the ergonomics over previous cars. There were no complaints about elbow room and the steering wheel angle was good.
E-Car Specific Subsystem integration

The electric car’s jump to a monocoque presents a formidable packaging issue considering the quantity and criticality of components needing placement. The quantity of items in the electric car’s specific subsystems, and the level of detail required to maintain accessible packaging that meets vehicle dynamic weight distribution requirements, posed a considerable task. A major advantage compared to last year’s platformed vehicle is that this chassis platform was designed with the electric components in mind, allowing greater flexibility in terms of packaging and meeting vehicle dynamics standards.

In terms of mass distribution target, the electric vehicle posed challenges to meet it’s goals that required some creativity and tight squeezes to achieve. The past electric cars were notorious for long wheelbases and rearward biased centers of gravity that were required to fit all components. A large effort was made in terms of component packaging to reduce the wheelbase and better control the center of gravity. For the first time, we were able to meet all electric vehicle’s mass distribution and wheelbase lengths without compromising packaging. To achieve that, the driver had to be moved off the center of gravity of the vehicle, the battery pack was integrated in a more vertical manner than prior years, and the motor was moved above the differential and partially into the accumulator container.

The critical components needing access in the vehicle can be shown labelled in the above image. A top cutout (the cutout directly behind the cockpit) allows access for a user outside the vehicle to the HVD (High Voltage Disconnect) and the Low Voltage Box. The placement of the HVD in this cutout is driven by FSAE accessibility restrictions on height and visibility of the part. The placement of the Low Voltage Box allows basic troubleshooting of the contents when the Low Voltage Box is inside the vehicle, but restricts more invasive access until the system is removed from the vehicle. While the Low Voltage Box placement was done predominantly as a safety concern with accessibility highly considered, in hindsight the restricted access to the internals of the Low Voltage Box while the system

Figure 31. Finalized electric car packaging scheme. Image courtesy of the 2017 FSAE team.
was in the vehicle caused numerous delays and large amounts of time lost during time sensitive troubleshooting. The TSMPS is placed directly to the side of the Low Voltage Box on the top cutout. This item’s placement was also driven by FSAE restrictions, and did not have flexibility to efficiently move anywhere else in the vehicle. The removable seatback allows access to the accumulator container (battery box), the front of the motor controller, and the main interfaces with the vehicle’s harness. The large rear cutout allows access to the rear of the motor controller, the drivetrain, and the liquid cooling loop. In hindsight, high voltage wire routing from the rear of the motor controller to the motor was considered only for a certain connector, which we later found did not meet FSAE specifications and regulations. The subsequent late in the year solution of directly connecting the wires to the motor (instead of utilizing the easy access connector) compromised the speed in removing the high voltage system, and made access considerably more difficult. More attention should have been paid to clearances in that region, with further backup plans considered if the initial connection methods did not work.

![Figure 32](image-url). A view of both the external of the accumulator (left) and its insides (right).

The high voltage system was packaged primarily inside the accumulator container, with the high voltage disconnect (HVD) requiring a separate container outside of the accumulator. While this division is far from ideal, it was required to pass competition restrictions by keeping the HVD accessible within the chassis and visible to those around the vehicle. The accumulator is removed from the cockpit of the vehicle, which expedited the process compared to prior years. Additionally, the accumulator was attached to the monocoque through bolts only through the floor of the vehicle, which we decided was the easiest and lightest way to pass strict competition structural requirements. The motor controller sits directly above the accumulator, and provides a convenient and mass effective location to route high voltage wires. Unfortunately due to how the motor controller was mounted directly to the accumulator, a high voltage line break was required between the motor controller and motor, which made the competition requirement of removing the accumulator for charging difficult to comply with in a quick manner. Additionally, the cooling system had to be rebled during accumulator removal at competition, although this did not pose a significant time constraint.
In order to meet the thermal demands of the vehicle’s high voltage system and powertrain, two separate cooling loops were implemented into the vehicle. To cool the batteries, air channels ran through the accumulator longitudinally with the ability to add up to 8 fans on either end to power the system. Analysis and testing showed that only 4 fans were needed and used, and the ability to add an additional 4 fans was considered redundant in the case that overheating was observed during testing. The air into the accumulator was channeled from behind the driver’s head through the LV system cover, into the accumulator, and out the back of the vehicle. The addition of scoops on the sides of the vehicle was explored, although deemed unnecessary upon airflow calculations of the system.

For the powertrain, a liquid cooling loop was utilized. The radiator was placed behind the LV cover on the rear of the vehicle, as CFD simulations showed that the amount of air running through that region would be more than satisfactory in cooling the low heat generating system. Additionally, the radiator placement “backpacked” at the rear of the vehicle proved a superior packaging option over alternatives. This placement is in contrast to prior years in which the radiator was placed on the side of the vehicle. The backpacking configuration, while far from optimal, easily dissipated the systems thermal load along with providing short and conveniently placed water tube routing accessible from the rear of the vehicle. The ideal placement would have been for the radiator to be situated inside the monocoque and below the motor, although we were not able to acquire funding to purchase a smaller radiator.

The monocoque decision eventually proved to be quite favorable, albeit initially challenging, for the packaging of drivetrain components. The drivetrain would be mounted in a chained configuration from the bottom to the top of the chassis, with the motor connected directly to the differential. The lower steel mounts (connecting the differential to the chassis) are mated to slots on the differential.
carrier, absorbing the manufacturing tolerances present by locating between the non-tooled chassis surfaces. This alleviates a majority of prior tolerance concerns when locating the motor to the differential as they will be aligned with respect to each other regardless of the rest of the drive system placement on the monocoque. Additionally, this significantly reduces structural chassis weight in that region as a majority of the chain loads are self reacted through the mounts rather than into the chassis. This was considered a significant instance of chassis weight savings, as the prior electric car subframe required beefy tubes in the region of the drivetrain to adequately handle these loads.

The sprocket and differential were placed as close to the ground and rearward as possible in order to enable packaging clearances between the motor and the accumulator container. Since motor position is a function of distance away from the differential, moving the differential rearward allowed the motor to clear the rear portion of the accumulator container. An added benefit of this placement was a slight reduction in vehicle center of gravity, a prominent concern that is associated with having a motor mounted above the differential. Additionally, access to the vehicles drivetrain proved significantly easier using a trailing differential compared to an enclosed system. However, this packaging method did have drawbacks associated with additional complexity in the system. Clearances were a key concern when locating the motor and differential, as distances from both the motor to the rear of the chassis and halfshafts to the rear of the chassis were tight compared to the rest of the vehicle. This is due to the fact that the monocoque chassis must fully enclose the motor because of FSAE regulations, and the halfshafts must have clearance with the rear vertical face of the chassis. Minimum clearances between the halfshafts and rear vertical face of the chassis were designed to be .03” (minimum clearance being an unshimmed differential at maximum tire bump height), which proved an adequate tolerance for the chassis jigging methods to accommodate.

Another issue that has been present in the electric car is passing the grounding requirement for the chassis components. Prior years the team has relied on a copper mesh within the laminate, which has not proven effective. This year, the chassis laminates have been tested utilizing the three point ground resistance method to determine the minimum distance between contact and grounding wire in the laminate that pass grounding rules. Grounding wires were then strategically placed throughout the chassis to assure that the system met those distance requirement. All grounding locations were able to be integrated in the washer of already bolted components. The accumulator was grounded through a post bonding wet layup of a thin aluminum sheet onto the outside of the carbon laminate. These grounding methods proved robust and easy to use, letting our team speedily move through this portion
of technical inspection. The ability to rapidly insert a grounding wire from a common ground to a component not meeting the requirement during the inspection proved to be invaluable. The only hangup was the seatback, whose bonding method made a clean grounding wire difficult to pass through.

With water intrusion during the rain event at competition proving itself a high risk scenario at previous competitions, the rear compartment of the chassis was also designed to be waterproofed in addition to each individual component. The geometry and mating covers of the access holes from the top cutout and through the seatback were designed to be easily sealed, which would prevent water from coming into contact with the high and low voltage enclosures. In the event water does break the seals, strategically placed drain holes along with waterproofing each individual box with gasketing (and tape if needed) allows some confidence that waterproofing will not be an issue with this chassis.
C-Car Specific Subsystem Integration

The C-Car specific subsystems that need to be integrated with the chassis are the engine (and its supporting systems: exhaust, intake, fueling, and cooling) and the combustion electronics. Two of the biggest issues we ran into with using a full carbon fiber monocoque for the C-Car were engine heat from enclosing the engine, and engine vibrations through the engine mounting. However, the engine is significantly smaller than the battery pack used in the electric car and therefore the components had more than enough space to fit in the monocoque as long as adequate mounting and heat shielding was developed. Both of these are addressed in detail in the subsequent sections titled engine mounting, and monocoque heat study.

Also, accessibility is very important for the engine. Engine work and maintenance must be able to be done on the engine without removing the engine from the car since this would be a waste of time to do basic engine maintenance or work. In order to have access to the engine, while keeping a good shape for stiffness and platform, two options were considered. Either the back end of the chassis could be U shaped with two “arms” sticking out the back of the chassis, or the back end of the chassis could be enclosed with a removable seat and access holes. Since we opted for the enclosed rear chassis design with a removable seat, we looked at what would be accessible on the engine with the seat removed (the quick maintenance situation), and what would need to be accessed on the engine for a relatively quick testing day fix/maintenance situation. We determined that the necessary engine testing day work that would need to be done at the track would include oil check/change, coolant check/change, injector replacement, spark plug replacement, coil replacement, clutch cable adjustment, shifting cylinder adjustment, and throttle cable adjustment. With the seat being removable and the rear of the chassis being open for diff mount/chain pass through, there was enough space to access the necessary components to perform our required testing day engine maintenance. Any significant engine work, such as anything requiring opening up the case, would require the engine to be removed from the car. These instances of significant engine work though could not feasibly be done at the track during the span of a testing day, so we decided that it would not be required to do this extensive engine work with the engine in the car.

![Figure 36. A preliminary graphic to show how much access is available through the seatback to work on components of the engine during a test day.](image-url)
The other main system that isn’t shared between the two cars is the combustion electronics, which mainly includes the ECU (engine control unit) and the PDM (power distribution module). Similarly to the engine, these electronics tend to require regular maintenance and so they must be readily accessible and easy to work on. We initially didn’t want to place the electronics in the engine bay due to the potential heat in there. But once we completed our engine heat testing (which is detailed in the Monocoque Heat Study section) we decided that it would be ok to place the electronics in the engine bay. The heat testing showed us that any components in the engine bay that were more than 6 inches away from the exhaust (when the exhaust was exposed) wouldn’t see temperatures above 160 F. We also found in the heat testing that a majority of the heat transfer from the exhaust was through radiative heat transfer, and when the exhaust was covered by a simple piece of aluminum as a heat shield (with an air gap between the two), that locations in the engine bay even within an inch from the exhaust didn’t get above 160 F. So with an aluminum heat shield around the section of the exhaust facing the seatback/fuel tank/electronics, and the electronics being over 6 inches from the exhaust, we assumed that they would be in an area with acceptable operating temperature. According to the electronics team, the temperature of the electronic components shouldn’t exceed 140 F, so we made sure to keep an eye on the ECU temperature during all preliminary testing using the internal thermocouple in the ECU. What we found from this testing was that the temperature of the ECU never got higher than 120 F. So regardless of electronics placement in future years, if it is in the engine bay, as long as there is a heat shield around the exhaust and the electronics aren’t super close to the engine, then from a temperature perspective they’ll be fine. Also, since the engine bay is sized to house the battery box in the E-Car, there was more than enough space behind the seatback to place the electronics while still fitting the fuel tank between the engine and seat back as well. This also meant that with the seat removed, the electronics were in an extremely accessible location. The three other locations that we considered were under the seat, up by the pedal box, or in a sidepod. We didn’t end up using any of these other locations because of packaging and fitment issues.
**Figure 37.** A cutaway of the c-car model gives a clear look at both the electronics packaging and the accessibility of the engine with the seatback removed. The electronics are the 4 boxes along the bottom of the chassis. From left to right they are the battery, the PDM, the ECU and the ACL.

**Figure 38.** A sectional view of the car from the side also shows the location of the electronics in the c-car relative to the seatback and fuel tank.

For the drivetrain (engine/motor and differential) we initially wanted to mount the differential the same way in both cars. But because we decided to soft mount the engine, the differential in the C-Car needed to be essentially rigidly mounted to the engine to prevent chain misalignment due to engine mount deflection. The differential in both cars though is still not enclosed by the chassis. This allows for less material/weight being used because there is no diff box, but requires a trailing suspension arm design. This is detailed more in the suspension section of the report.

**Engine Mounting**

When we started looking at mounting the engine, the first problem that we encountered was figuring out where on the monocoque the engine should be attached to. With the engine enclosed in the back of the monocoque, the options available to us were mounting the engine to the top, sides, or bottom of the monocoque. Due to the shape of the engine, using a mounting system that attached to the top of the monocoque seemed like it would need to be overly complex and cumbersome just to mount to the engine and the somewhat narrow area surrounding the cutout in the top of the monocoque. Mounting the engine to the side of the monocoque initially seemed to be a viable option. However since the engine bay was 5 inches wider than the front and 8.5 inches wider than the rear on each side of the engine mounting points, the majority of the load from the engine (which we determined would be primarily vertical and longitudinal) would put the mounts in bending because of these relatively long moment arms. Therefore the inserts into the monocoque would also be placed in bending, which they handle very poorly (specifics on the inserts are covered in the insert design section). The one way to counter this would be to use a very large footprint for the insert pattern on the monocoque, but this would be heavy and also take up a lot of space in the engine bay for mounting other components and for accessing the engine. Our final option that we looked at was mounting to the bottom of the monocoque. This would allow us to mount to a large flat plate of the monocoque as well as giving us a mounting point where only the longitudinal portion of the anticipated engine loads would put the inserts in bending. After making some quick sketches of potential mounting options, we decided to mount the engine to the bottom of the monocoque.
Figure 39. Roughly sketched ideas for mounting options for the engine in the rear of the monocoque.

Once we had decided roughly where on the monocoque we were going to have the engine mount to, we needed to get a better idea of what loads the mounting of the engine would have to be capable of withstanding. Based on some information that we found that discussed where the primary loads on engine mounts originated from (combustion event vs rotating components) [13], we decided to look primarily at the dynamic loads created by the eccentrically weighted, rotating, internal components of the engine. We looked at this because the other large source of load is the combustion event which at low rpm (0-2000 rpm) should be taken into account, but at higher rpm (2000+ rpm) can be considered negligible. The formula car when running operates above 4000 rpm, and historically has had an idle that at the low end is 2000 rpm, so we assumed that we could follow what we read and neglect the combustion event in our loads on the engine mounts. To get an idea for what the dynamic loads from the engine were, we deconstructed one of the team’s old WR450 engines to get dimensions of the internal components (length, width, height, weight, and C.G. location) that would be rotating with eccentric weights or moving linearly. The components that we included were: the piston, connecting rod, crankshaft (with counterbalance), and balance shaft. Once we had the necessary dimensions of these components we used matlab to determine the accelerations of these components at different engine rpm and therefore forces at the engine mounts (using component mass and component location relative to the engine mounts). This matlab script is included in appendix D and goes through the position, velocity, and acceleration of the different components and then uses the mass of each component to turn the accelerations into forces and then figures out the necessary reactions at the engine mounts. From this we were able to get a representation of the load at the engine mounts for specified rpm.
Figure 40. Dynamic engine loads due to eccentric rotating engine components at 10k rpm. The graph can be read counter-clockwise around each approximately closed loop as a full engine rotation beginning at piston TDC at the top of the graph.

With these loads from the engine being vibrational loads that are decently large (~2500 lb vertically and ~1800 lb longitudinally) at fairly high rpm (10k rpm) at their maximum values, we were worried about how the large flat sheet of carbon sandwich panel at the bottom of the engine bay would handle it. Primarily we were worried about the vibrations causing delamination between the carbon plies and therefore damaging the integrity of the monocoque (specifically in the bottom of the engine bay). Because of this we decided to go ahead and soft-mount the engine in an effort to decrease the load transferred from the engine mounts into the monocoque. For soft-mounting the engine we were primarily interested in using rubber mounts to achieve a transmissibility ratio between the engine and chassis of less than 1 by having the ratio of the engine frequency to the natural frequency of the mounts be greater than 1.5. This is based on equation 1 which deals with the transmissibility ratio as a function of damping, and frequency ratio (input frequency/system natural frequency).

Equation 1. Transmissibility ratio

\[ T = \frac{F_T}{F_o} = \frac{1 + \frac{\omega}{\omega_n}}{\sqrt{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(\frac{2\delta\omega}{\omega_n}\right)^2}} \]
Figure 41. A plot of transmissibility ratio for different damping ratios and frequency ratio. For any damping ratio, a frequency ratio above 1.41 is necessary to have a transmissibility ratio that is less than 1.

As can be seen, where damping and frequency ratio are swept, the frequency ratio needs to be greater than 1.414 for the transmissibility ratio to be less than 1 regardless of the system damping. Because the frequency ratio is based on the input frequency of the engine, which is the engine rpm, we also needed to pick an rpm “cutoff” where we would hit the minimum frequency ratio to have a transmissibility ratio that was less than 1. The advantage of running a lower cutoff rpm, such as the idle rpm, is that the entire rpm range of the engine would exist at frequencies where the transmissibility ratio would be less than 1 and the load into the monocoque would be lessened more at all rpm than with a higher cutoff rpm. The downside though is that to achieve this, the rubber mounts need to be softer which means that the engine will deflect more when it vibrates. The disadvantage of a higher rpm cutoff however is that there will be engine rpms, such as the idle rpm, where the transmissibility is greater than 1, and potentially a lot greater than 1 if the engine operates around the mounts’ natural frequency. We decided to start off by choosing an engine speed of 4000 rpm (slightly higher than the engine idle speed of 2500 rpm) as our cutoff rpm while also selecting stiffer rubber mounts to set the cutoff rpm at 15000 rpm. We used the 60166-1 isolators as the softer mounts and the 60166-5 isolators as the stiffer mounts. For the isolators we also considered size, weight, and packaging, because larger and heavier isolators were capable of carrying larger loads and being stiffer, but were also heavier and more difficult to package.
Figure 42. Visual representation along with specifications for the vibrations isolators used for mounting the engine in the formula car.

Once we had the rubber isolators chosen, we also had to design the structure from the engine mounts to the isolators. This included working with the drivetrain team because the team runs a chain drive, which means that the differential needs to be rigidly mounted relative to the engine to prevent relative movement between the two sprockets which would cause massive issues with the chain. We were primarily concerned with weight and manufacturability for the engine and diff mounting, which led to us considering boxed steel, machined aluminum, and waterjet aluminum structures to connect the engine mounts to the isolators which would then bolt to the chassis. After multiple revisions, we decided to go with a bolted assembly that was composed of waterjet aluminum uprights, machined aluminum diff mounts, and steel base plates to distribute the loads to rubber isolators. The whole assembly could and should be analyzed further to find unnecessary/low stressed material to remove to weight advantages. In total the assembly weighed 8.51 lb, with a good portion of the low stressed material removed during the revisions. This weight could and should be cut down to below 5 lbs to make the engine mounting in a monocoque worthwhile from a weight perspective.
During testing of the car, we started out by running the softer rubber isolator mounts. What we noticed that the engine mounts had significantly more deflection than anticipated when the car was both engine braking, and when it was accelerating. Primarily this seemed to happen when shifting, or when the driver was getting on and off the throttle. The amount of deflection could be seen in the gopro footage that we took of the engine during a testing day (with the softer rubber isolators and no third engine mount). We lifted stills from that video at maximum deflection in each direction along with static positioning of the engine to give a visual of what was seen during testing days and in the gopro footage.
Because of this deflection that we saw during testing, we switched to the stiffer rubber isolators as well as adding a third engine mount that mounted to the top mount on the engine and to the right muffler mounting bolt. Adding this third mount to the engine gained two things. First, it provided another point with which to react the loads from the engine. Secondly, and most importantly, it provided a constraint on the engine that prevented the engine from rotating about the y-axis or “bucking” during engine braking and acceleration which removed the worryingly large deflections we were seeing during testing. Even with this setup, there was enough movement of the engine such that the exhaust header (due to only having a slip joint in the lateral and not also in the longitudinal direction) was stressed and cracked multiple times during the testing season. During testing, the third engine mount also did shear the AN-4 bolt that was mounting it to the chassis. Due to the vibrational loads resolved by the third mount (effectively a cyclical loading case) it was assumed that the bolt was sheared due to fatigue and not a single impact load. Because of this, we recommend that the third engine mount be used, and that any mounting hardware for it be swapped out for new hardware after testing and prior to competition. However, overall the engine mounts and rubber isolators did hold up in this configuration through testing and competition without catastrophic failures.

Figure 44. Photos from Go-Pro footage of the car during testing. The top row shows the engine bucking forwards (front mounts in compression and rear mounts in tension), and the bottom row shows the engine bucking backwards (rear mounts in compression and front mounts in tension). The middle row shows the engine in its nominal, no load position.
After competition, the section of the tub under the engine mounts was inspected and a section around one of the chassis inserts that the isolators and baseplates were mounted to was removed. It appears that both the honeycomb and the insert were crushed and the carbon seems to have delaminated locally around the bottom side of the insert. The way that the core is buckled to the right of the insert leads us to believe that the core failed first due to the insert reacting a counterclockwise moment and an upward load. Once the core failed, the insert would no longer be able to handle as high of compressive loads, causing it to be crushed. This crushing of the insert would then lead to the local delamination around the insert that we also see in the photo. Because of this (and because we didn’t check the carbon after every testing day to give us a progression or a definite occurrence of this crushing of the insert and core) our guess is that this failure occurred with the softer mounts (and without the third engine mount) and was caused by the engine bucking during engine braking which would have resulted in a high impact load being applied in an upward direction and a moment being applied in the counterclockwise direction to the rear engine mounts.
Looking back on the decision and execution of the engine mounting, there is a lot of room for improvement that can and should be carried out by future teams. For starters, there were two things that our senior project more or less took educated guesses on and then took a conservative approach to deal with: engine loads, and a carbon sandwich panel under vibration. We made assumptions that the engine loads were primarily coming from internal eccentrically rotating, and we assumed that a large, flat carbon sandwich panel would potentially delaminate when continuously exposed to the vibrational load of the engine. As we noticed in testing, the loads placed on the engine and differential during acceleration and braking were significant, and need to be looked at for future designs (they are either originating from the tire’s interaction with the track, or from the impact load caused by the load in the chain changing direction, or a combination of both). Along with including these loads in the design, the engine loads should also be characterized better than the way we did it through our matlab script. Potentially this could be done on the dyno with accelerometers mounted on different points of the engine or strain gauges and a mounting system of two force members. The big advantage of better characterising these engine loads is that the entire engine mounting system could be lightened. If the loading is better understood, then a way to react those loads can be more efficiently designed for material use. Also, if the team can test a carbon structure that is representative of the bottom of the tub under vibration similar to what the engine outputs, then the team might be able to find that using vibration isolators between the engine and the tub is not necessary. This could be done by making a dyno test stand out of a carbon sandwich panel and running the engine on it with the engine hard mounted to the sandwich panel. A final thing to note for future teams is that the third engine mount should be included in the initial design, because it is an extremely valuable tool for decreasing engine movement and helping to react engine loads.

Monocoque Heat Study

One of the major concerns brought to the table was the effect of the engine heat on the composite structure of the engine bay. Epoxy resins have a glass transition temperature \( T_g \) at which the polymer loses much of its stiffness and strength. All of the team’s prepregs currently have \( T_g \)’s of at least 265 °F, but the weakest link in our composite materials is our Newport NB102 film adhesive, which has a glass transition temperature of 230 °F. Epoxies will also gradually lose stiffness as they near the \( T_g \), due to the fact that the \( T_g \) is a smooth range rather than a discrete value. Accordingly, we decided that our maximum “safe driving temperature” for any monocoque structure would be 200°F. Although this was a somewhat arbitrary decision, we believed that it gave a safe margin, while remaining practically attainable, given that on a hot day, skin temperatures have been consistently measured up to 150°F just from ambient heat.

The YFZ450r engine’s operating coolant temperature is 190-200°F, so we assumed that the engine surface temperature would be at least 200°F. The exhaust temperature was estimated to be 1600°F at the pipe surface. We considered conducting some sort of heat transfer analysis to determine steady state temperatures, similar to the analysis conducted by the Formula Chassis Works team [1], but we decided that we would not be able to achieve an accurate result, and that physical testing was the best way to obtain temperature data. We needed to quantify how both the engine and exhaust affected monocoque temperatures, and if insulation was needed, what type and how much should be used in order to be effective.

To obtain relevant temperature data, we needed a way to simulate the engine running inside a monocoque, and a way to record the maximum temperature reached in critical areas. Originally, we
had hoped that we could use our accumulation of old test panels to enclose an engine on the planned-to-be-restored FSAE engine dyno. However, early in the year the dyno refurbishment was scrapped, so we changed plans, and decided to fit carbon panels around the c-car subframe to enclose the engine, and then run a mock endurance (approximately 20 laps around our typical track at H1).

Old test panels and firewall panels were cut using a tile saw to fit around the subframe, and holes were drilled to accommodate zip-ties for mounting.

![Figure 47. Sandwich panels cut to fit around the subframe and exhaust components.](image1)

The panels were zip-tied to the subframe, and any gaps between the tub, subframe, and panels were sealed with aluminum HVAC tape, which is safe to use up to 260°F. Sections of the top and rear of the car were left open, to mimic the expected layout of the new chassis.

![Figure 48. Panels enclosing the subframe.](image2)

To record temperature, we ordered Thermax temperature-indicating stickers from McMaster-Carr (Part #595K16). These stickers record the maximum temperature reached through a color-changing chemical reaction. Ours read from 160°F to 230°F in 10 degree increments, which captured the range of temperatures that we were concerned with.
Figure 49. McMaster 595K16 temp-indicating sticker placed on a sandwich panel. Each white square will turn black once the corresponding temperature is reached.

10 stickers were ordered for use on the car. One was sacrificed to verify that they were functioning properly. The other 9 were placed at varying distances from the engine and exhaust, with the closest stickers within an inch of the oil pan and the exhaust by the seatback.

Figure 50. Approximate locations of temp-sticker placement on the car.

After two weekends of not being able to run the car long enough to get data, we were able to consistently run the car for 20 laps, with the engine at an operating temp of 190°F. After the test day concluded, we peeled off the panels to observe the results. The sticker within an inch of the exhaust was maxed out at 230°F and looked wrinkled and distorted from heating, and the sticker within an inch of the oil pan showed that 160-170°F had been reached. All other stickers had no reading at all, including a sticker 6” from the exhaust pipe. From this first round of testing, we concluded that we were safe from the engine heat, but exhaust heat needed more data.
We took the unused stickers and arrayed them around 2-4” from the exhaust piping, on the seatback. The original plan had been to place them at regular 1” intervals, but there was no space to properly place them, and they were stuck wherever we could get our hands past the exhaust. The car drove with these stickers for the IAB Drive Day in October. Even with a low-moderate amount of driving, all stickers were thoroughly toasted.

It was now clear that we needed to use some sort of insulation for any composite within 6” of the exhaust. Five more temp stickers were purchased and third round of testing was conducted, with stickers placed at more varying distances from the exhaust, and covered in the same aluminum HVAC tape used to seal the panels. After a day of extended driving, the tape was pulled off, and no stickers showed any heating whatsoever. From this, our final conclusion was our monocoque would be safe from engine and exhaust heat, as long as aluminum tape was used as insulation within 6” of exposed exhaust piping.
Figure 53. Unaffected stickers that were protected by aluminum tape.
Stiffness

In assessing the stiffness requirements for our chassis, the only loading scenario we considered is the vehicles hub to hub torsional stiffness. A low hub to hub torsional stiffness not only impacts vehicle lateral load transfer in a negative way, but it also makes tuning a vehicle more difficult as the chassis itself is an additional spring that reduces linearity/predictability of tuning parameters. Prior Cal Poly FSAE vehicles have tried numerous methods to develop and achieve concrete stiffness goals, although to limited success.

<table>
<thead>
<tr>
<th>Year</th>
<th>Actual Torsional Stiffness</th>
<th>Torsional Stiffness Goal</th>
</tr>
</thead>
<tbody>
<tr>
<td>2016</td>
<td>1305</td>
<td>1500</td>
</tr>
<tr>
<td>2015</td>
<td>1224</td>
<td>1500</td>
</tr>
<tr>
<td>2014/2013</td>
<td>1050</td>
<td>1700</td>
</tr>
</tbody>
</table>

Table 9. Measured hub-to-hub torsional stiffness vs. desired torsional stiffness of historic chassis.

The subject of what is a realistic torsional stiffness target, along with how much mass a given stiffness is worth, has been a subject of debate for several years. While our team has never actually met our torsional stiffness goal, we have also never had any handling concerns that we can directly contribute to low stiffness. The fact that we never have had a problem that we could outrightly contribute to low torsional stiffness has made it difficult to justify devoting a significant amount of time of several senior members to solving the issue instead of fixing more pressing matters.

In setting a target for torsional stiffness, the suspension system plays a key role in assigning the target lateral load transfer. As a rule of thumb, a lateral load transfer distribution of 90% of the “ideal” case of a rigid chassis will provide acceptable load transfer characteristics, while staying at a reasonable weight [8]. Using script developed by the team in 2015, we input the car’s planned roll stiffness of 664 lb-ft/degree, and investigated the effect of differing chassis stiffness on lateral load transfer distribution. The output is shown below.
Figure 54. Lateral load transfer distribution vs roll stiffness distribution for varying chassis stiffnesses. A chassis stiffness of 1500 ft-lb/degree achieves 88% of a rigid chassis.

From looking at the trends in this graph, the difference between a hub to hub torsional stiffness of 1500 to 2000 ft-lb/degree was fairly insignificant. Based on prior FSAE vehicle experience, achieving above a 2000 ft-lb/degree stiffness value was unrealistic at this time. Thus, a system torsional stiffness of 1500 ft-lb/degree, which is 2.25 times the total roll stiffness, was selected as our goal. This is within the range considered to be acceptable in maintaining handling balance and tunability [9]. After lengthy collaboration with the team, it was determined that the most mass effective method to achieve this goal was to aim for a 3300 ft-lb/degree chassis frame stiffness with a hub to hub stiffness of 1500 ft-lb/degree.

2016 Analysis Synopsis

With a fairly well defined target for the 2017 vehicles, analysis was completed on the 2016 vehicle to attempt to validate the finite element model to show where stiffness loss in the system was occurring, and how to best fix it. By spending considerable amount of time on the prior vehicle, it was hoped to determine a modeling method that could accurately represent the system and give confidence to future model predictions. Additionally, a trustworthy model validated by vehicle testing would allow us to place weight where it could best affect the stiffness of the structure, instead of allocating weight and resources in stiffening up an already stiff part of the assembly.

The lead contributions to modeling the 2016 chassis was through Manuel Herman’s thesis [12], in which he compares torsional test data with a chassis finite element model. The finite element model was correlated to within 5% of test data. The results of this model showed that a majority of compliance was due to the vehicles subframe, and the chassis and suspension had near equal contributions to stiffness. While this report does not quantify the significance of parameters such as mount stiffness and suspension geometry, it is the first piece of evidence that shows our team that hub to hub torsional stiffness is affected by suspension as much as chassis. Additionally, the conclusion
that the subframe was likely a large cause of torsional stiffness loss helped solidify our decision to stay with a monocoque design for 2017 over alternative chassis types.

**Cross Section Emphasis**

In order to accomplish the goal of improving the stiffness of the chassis, it was determined the most time effective method to solve this issue was to compare certain chassis cross sections at key locations, and pick the section that best met all requirements. This partial analysis would allow us to define chassis features utilizing simpler (and quicker to make) models. Once the general geometry is finalized through these partial sections, more detailed stiffness tuning will utilize a full chassis model.

In selecting the areas to focus on for partial models, the obvious target for potential improvement was the cockpit region. Not only is the cockpit region known to be the weak spring in the chassis stiffness, but it also is the area in which the most design freedom is allowed in the chassis geometry. The 2013 FCW project team estimated that compared to a fully closed-section tube, the cockpit opening was 89% less stiff, based on FEA modeling of a simplified cockpit section [1]. This dramatic stiffness loss is likely due to the transition of a stiffer closed tube cross section to a much less stiff open section. Additionally, the front of the chassis decreases in radius towards the front in order to better accommodate packaged subsystems and for ergonomics reasons. This radius decrease affects the front of the cockpit opening, which would suffer a dramatic stiffness loss through both an open cross section and a smaller cross sectional radius.

![Figure 55](image_url)

**Figure 55.** The radius decrease at the front of the cockpit can be easily seen as the chassis floor in the front of the vehicle is significantly higher than the chassis floor behind the front wheel.

One simple method attempted previously to stiffen the cockpit region was to add plies around the vertical edges of the cockpit. Through adding additional plies and beefing up the laminate, you should be able to replicate a stiffness similar to the rest of the chassis, but with more plies. The FMD senior project team analyzed the effect of adding plies to the side impact structure to stiffen the cockpit [2]. It was found that adding 4 plies overall, stiffness was improved by 7%, but weight increased by 8%, making this method ineffective. No further solutions were pursued in 2015. This information shows that adding laminate in the region, which is by far the easiest solution to the issue, would not yield positive results without a significant weight penalty. A substantial portion of the stiffness loss in the open cross section could be due to the structure “parallelograming” under the torsional moment, and thus increasing the bending stiffness in that direction could yield positive benefits. Additionally, a well designed closeout could yield similar benefits win increasing that sections warping stiffness without most of the weight gain. This ended up being a focus of our analysis.
Another prior method to increase cockpit stiffness was to avoid a cross sectional radius decrease in the chassis cockpit altogether through implementing a flat monocoque floor and a carefully planned radius change. The 2014-2015 electric chassis [3], is a great example of how this concept would work. Designed by a self developed evolutionary algorithm, this chassis was known for its specific stiffness. Utilizing a thin baseline laminate throughout its entire structure and relying on geometric stiffness instead of increasing ply count allowed it to achieve target stiffness without major weight penalties. Additionally, the thin laminate was required to be high strength fabric to pass structural requirements, so little additional stiffness was achieved through the laminate. While the need to integrate stiffness through solely geometry changes was likely due to their manufacturing restrictions of a single thin baseline laminate, our project team favored their geometry instead of plys based ideology. This geometry emphasis coupled with higher performance and more tailored composites laminates was the style we gravitated towards in the design phase. Unfortunately, in terms of the flat bottomed chassis, we did not pursue this idea because it would require significant compromises in major subsystems that relied on front chassis clearance, and that those compromises were not entirely justified.

![Figure 56. J-Lo (2015 e-car) chassis design](image)

Another way we explored to combat the torsional stiffness decrease of the front cockpit was to place the “swoopy”© above the radius decrease. The “swoopy” could then be designed to maintain an equivalent cockpit cross section torsional area to negate the effect of the radius decrease and torsional stiffness loss. Unfortunately, the vehicle could not ergonomically handle the placement of a swoopy in that location, as the drivers steering ability and egress times would be significantly compromised. It was determined driver comfort in this scenario was more important than a slight increase in torsional stiffness, so the idea was abandoned.
Torsion Tube Analysis

Now that previous projects have made it near certain the weak spring in the system torsional stiffness is coming from the cockpit, and the FMD senior project team proving that stacking laminate on the sides has little effect, a study of the cockpit geometry was in order. It is believed that this section accounts for so much stiffness loss in the chassis, that a stiffness increase in another region would have little to no effect without first addressing the cockpit. Additionally, this region represented the area of the chassis with the most design freedom. Unfortunately as stated prior, the front of the cockpit was off limits due to looming ergonomic issues, so a uniform cockpit geometry was studied in order to best increase the specific stiffness of the section.

The stiffness of different cockpit geometries were compared for a sample cockpit in a tube representing the chassis. After researching the mechanics of an open section in torsion, FEM model iterations over a closed form solution were pursued due to the complexities and difficulty required to build a trustworthy open cross section torsional model. The samples used for comparison were a shell model of a filleted square torsion tube. The dimensions of the fillets and cross section geometry were chosen to be roughly representative of the chassis at the time of the analysis. The shell was comprised of core separating isotropic laminates whose dimensions and properties were similar to the properties of potential laminates and cores. The laminates were kept isotropic through the comparisons in an attempt to isolate only the geometric effect on stiffness and deflection.

![Figure 57. A picture of the geometry and constraint method for the torsion tube analysis. The ends were allowed to deform radially.](image)

The initial models fixed the ends of the torsion tubes from any warpage, although the models were updated and reran to allow radial deformation following concern expressed during the team’s design review. The concern was that by restricting movement in the radial direction, the structure would not reflect an accurate stiffness trend for the chassis. By allowing the models to deform radially at the
ends, the analyzed deflection change of the structure amounted to less than a 3% change from the original models. Updated values are shown in the graphs. Radial warpage of the section was achieved through applying a cylindrical coordinate system boundary condition with free DOF in the radial direction at the ends of the tube. The end with the applied moment was also able to freely rotate in the theta direction in addition to the radial direction. The torsional moment applied to one end of the tube was accomplished by utilizing a structural coupling that only transmitted torsional moment from the central to the perimeter nodes.

With issues of nonconvergence mentioned previously in the FCW senior project report, a mesh convergence study was ran on the base models to determine the validity of the results, and the model converged appropriately for the reduced lengths. This is contrary to what was seen in the models presented in the FCW senior project report detailing nonconvergence without a long sample section. As the nonconvergence issues were not seen or were not properly identified with these newer torsion tubes, an elongated section was not pursued.

![Torsion Tube Mesh Convergence](image)

**Figure 58.** Convergence graph of the torsion tube mesh.

The first geometry comparisons we conducted were the sidewall heights of the chassis. It is known that by decreasing sidewall height you will greatly increase ergonomic comfort and decrease section area (and thus weight), but there are repercussions in terms of stiffness loss and manufacturing time increase. Although, without numbers, it is difficult to determine if these sacrifices would be worth it. This sort of cockpit sidewall reduction is commonplace in both prior Cal Poly and competitor’s chassis’s.
Studies were conducted using the finite element method detailed above. These studies showed the cockpit was highly sensitive to the reduced height, increasing deflection by upwards of 113% for a height reduction of 6 inches in our sample cockpit. An explanation for the graphs nonlinearity is that the 0 inch fillet radius also incorporates a small edge closeout flange (by nature of the model with filleted corners), which could increase the stiffness and reducing linearity. Through limiting the cockpit cutout in one dimension, the specific stiffness of the section (assuming a uniform laminate throughout the side) increased by 50% over the side filleted section. This result suggests that utilizing the sidewall height reduction as a mass saving method is not an effective way to reduce mass while maintaining adequate stiffness. An explanation for why no sidewall height reduction is desirable is that an opening in only one dimension partially mimics the properties of a closed cross section by reducing the ability of the structure to “parallelogram”. This is achieved through increasing the height of the torsional center (when compared to an open cross section), which allows the laminate to react to the moment in plane instead of bending. This could also be viewed as essentially an implementation of a high moment of inertia closeout incorporated into the structure, which greatly aids in the cockpit stiffness.

One example of a high cockpit side was the previously mentioned 2014-2015 electric chassis. In addition to its front cockpit geometry, the cockpit has the largest cross sectional diameter of any other portion of the chassis. The cockpit geometry, is an example of how high side walls could be used to stiffen the cockpit region to better match the stiffness of the rest of the chassis. Although the torsional stiffness was ultimately unvalidated through physical testing, the high cockpit walls and stressed bulkheads likely played a large role in the high finite element model stiffness.
Figure 61. The above image shows the chassis of the 2015 electric vehicle, noted for the high specific stiffness. The high cockpit wall are characterized by a series of stressed bulkheads throughout the section.

After seeing specific stiffness gains through raising the cockpit sidewalls, we next pursued the option of adding additional closeouts to the full sidewall height models in order to see if we could gain additional stiffness. The downside of the addition of large closeouts would be reduced ergonomics, more difficulty in meeting egress requirements, and an addition of complexity into the manufacturing process. This would also likely require a wider cockpit opening to account for the downfalls. The geometry of closeout pursued was essentially a curl and flange that would run parallel to the cockpit sidewall. This was selected because of its relative simplicity in manufacturing, and the fact that other teams have successfully implemented this design into their chassis. In the analysis models, the thickness and length of the cutout were varied in the torsion tube. Unfortunately, the closeout had little effect in increasing the sections torsional stiffness, with a maximum increase of 2% seen, and would definitely not justify the challenges it would pose to packaged subsystems. A possible reasoning for why a fancy closeout had minimal effect on stiffness is that the full sidewall model already incorporated a sizeable closeout by nature of the chassis geometry. The chassis flange analysis results suggest that there could be diminishing returns with regards to closeouts and chassis stiffness. What was not pursued was how this type of closeout could affect a chassis cockpit with a reduced sidewall, which could see benefits that were not apparent in this analysis.
The next geometry tested was the top view cockpit cutout. This model utilized the chassis section without any sidewall reduction. Traditionally, our team has favored small radius to allow for large amounts of clearance for the driver and mounted components, although the loss in system stiffness is unknown. After running the comparison models, the opening was reasonably sensitive to larger fillets, and there was a 26% decrease in deflection by going from no fillet to a 6 inch radius fillet. Thus, our chassis geometry emphasized the largest top view fillets our packaged components could handle.

Interestingly enough, this study showed that significant stiffness loss occurs immediately after the fillets instead of at the beginning of the cockpit opening (the red portion in the below figure). At the point where the fillet is finished necking down into the cockpit opening is also where a sharp increase in angular deflection is seen.

Additional models investigated whether constraining the end of the fillets in the cockpit would result in a system stiffness increase. By constraining the two fillets to each other with a plate equivalent to a single chassis face skin (meant to represent a stressed seat), the section stiffness increased by 20%. The single faceskin plate was chosen over a metal tube since it was considered much more realistic to stress a carbon seat than the roll hoop at the time of the analysis. A stressed roll hoop would likely yield better results, but require significantly more time and design effort to implement. The results of the analysis lead us to believe that a large amount of torsional stiffness lost around the cockpit opening can be recuperated through stressing bulkheads at the beginning and ends of the cockpit.
stressed seat and a stressed front roll hoop was considered, but unfortunately proved to be out of the scope for this senior project.

The last important parameter we looked at in the torsion tube analysis was how a laminate increases or decreases in certain regions of the cockpit would affect the sections stiffness. We utilized the section model without any sidewall height decrease in these comparisons as this geometry would likely be most representative of our completed chassis.

The first part of the laminate we changed was the core size. An increase in core size throughout the section does increase torsional stiffness, by about 35% if you increase core thickness from .25” to .75”. Looking at local core thickness sensitivities, a local core reduction (from .75” to .25”) on the floor of the chassis resulted in a stiffness loss of only 6%. This could make a core reduction on the bottom of the chassis in order to lower center of gravity very feasible. Similar values were found for reducing laminate thickness on the floor. This lends itself to the idea that the stiffness of the floor does not play a large role in the sections torsional stiffness. A potential reason as to why the chassis floor was not incredibly important to the cockpit’s stiffness was that a majority of the floors contribution to stiffness was due to it just being there rathering than it being stiff. The torsional center of the structure matters little if the floor is beefed up.

![Figure 64. A graph showing stiffness as a function of uniform core thickness.](image)

In summarizing what we learned from the torsion tube analysis of the cockpit region, it appears the stiffness of the region can be viewed as both a function of warpage of the cross section (“parallelograming”) and a pure torsional reaction (such as a reaction to pure torsion in a closed tube). The structure’s warpage appears to add additional compliance on top of a structures normal torsional stiffness and drastically reduces section specific stiffness. Additionally, the results from the FEM models seem to suggest that the most effective method to increasing the sections specific stiffness is to target the systems warpage. After some research into the contributing factors of warpage, it appears the largest contributor to warpage is the torsional center of a structure, and how geometry and cutouts affect it. An open section drives the torsional center to the closed section (the floor in case of the cockpit), with the resulting FBD showing that the bodies reaction to torsion results in a large bending force that attempts to warp and “parallelogram” the structure. This bending of the sidewalls would add considerable compliance to the system. While this is likely an oversimplified view of the subject, the idea behind it appears to remain consistent throughout the FEM models. The models with a torsional center closest to the middle of the structure (the case of pure torsion in a tube) tended to have significantly higher specific stiffness values. The addition of structure parallel to the floor (such as the
closeouts inherent on the high sidewall section models) move the torsional center closer to the middle, and thus reduce the warpage stiffness loss. High sidewalls aid in increasing the systems pure torsional stiffness by increasing cross section diameter. Additionally, moving this torsional center appears to be significantly more effective in terms of weight than combating warpage through increasing bending stiffness of the sidewalls. This would also explain how the closeouts parallel to the sidewall had little effect, as they would only contribute to sidewall stiffness and not moving the torsional center. This would also seem to explain how the bulk addition of 45 plies in prior chassis had little effect on the stiffness of the section, as it would increase the stiffness in a pure torsion case, but would do nothing to effect the torsional center nor sidewall bending, which appear to be the dominant factor in section stiffness.

**Final Stiffness Driven Geometry**

The final chassis design as it relates to stiffness is largely a compromise between adequate packaging of components and implementing what we learned through the torsion tube models. As is apparent through prior years of FSAE work, improperly packaged components were likely to affect the vehicle’s performance significantly more than an incredibly stiff chassis, letting us be strongly partial towards component packaging. Additionally, as this was the first vehicle our team has ever packaged a combustion powertrain in a monocoque (and one year removed from an electric monocoque), accessibility took priority over performance parameters. With that being said, the sections directly behind and in front of the cockpit would not contribute as much to system stiffness if cockpit stiffness is the driving factor. Geometry in regions away from the cockpit were not driven by stiffness concerns.

![Figure 65. Finalized monocoque geometry.](image)

Taking inspiration from the 2014-2015 electric chassis and the torsion tube models, the cockpit for this vehicle featured high sidewalls with the chassis’s largest diameter throughout as much of the cockpit as possible. Additionally, large top-view radius were added in the cockpit opening in order to increase section stiffness. Unfortunately, we were not able to match the “swoopy” with the front cockpit radius decrease due to packaging issues. This will likely be the spot of the most stiffness loss in the structure. Any stressed member to reinforce the cockpit were not pursued for this year, although the chassis geometry was designed to allow future teams to consider the possibility.
The rear cutouts in the chassis were allowed only for required packaging and access locations. All cutouts were kept in a single plane to reduce any warpage concerns, although component accessibility will take priority over chassis stiffness when a tradeoff is needed. The bulkhead for the rear suspension mounts was initially intended to be highly reinforced to both transmit suspension loads and to serve as a bulkhead guarding against open end effects at the rear of the chassis. This was initially going to be similar in reinforcement to how the front bulkhead is with required SES regulations. Observing how little the radial deformation mattered in the torsion tube models discouraged the idea, and a lighter laminate was kept in the rear section to favor a lighter structure.

In terms of the stiffness driven laminate design, a majority of the laminate is kept as simple as possible for ease of manufacturing and passing structural requirements. The benefit of emphasizing geometric stiffness in the chassis is that a more uniform laminate can be utilized throughout the structure to achieve a similar stiffness when compared to a complicated laminate with compromises in geometry. This is in contrast to prior vehicles that relied on stiffness driven and intricate laminates to achieve satisfactory stiffness. Since the floor was an unregulated region for strength, and did not affect the structure's stiffness significantly, thinner core (.5” instead of the typical .7”) was utilized on the bottom of the chassis at a small tradeoff for stiffness. A potential weight savings technique not pursued this year was decreasing the laminate and core size in the top half of the cockpit above the SES regulated region. This could mimic the weight savings close to that of a short sidewall cockpit, but the stiffness benefits from a full side walled cockpit.

**Suspension Contribution**

In terms of how the suspension affects the torsional stiffness, a majority of the work was left to the suspension subsystem and accompanying projects/thesis. Unfortunately, due to the other resource requirements of this project, a detailed suspension stiffness analysis was not conducted. Additionally, a detailed understanding of stiffness loss from tire to loading the chassis was viewed at the time as likely taking multiple years of team development and vehicle iterations to narrow down. Thus, this analysis was determined to be outside the scope of our senior project.

For reference, the method chosen by our team to meet the suspension stiffness target requirements was to increase link and mount stiffness. As the pull rod was the contributing factor for stiffening the vertical DOF of the front upright, this cross sectional area was increased in size. The other links on the front were also increased in size to account for toe and camber compliance. While weight was added to the pull rod in the front to achieve a higher torsional stiffness, our comparisons showed it was justified through a reduction in the assemblies center of gravity. As for the tabs, all suspension attachments were designed from aluminum that would increase the attachments stiffness over the steel mounts. Unfortunately, there was a mishap with the 3d printed aluminum mounts, so thin steel mounts were resorted to late in the manufacturing process. How suspension geometry affected hub to hub torsional stiffness was not considered in the design phase.

**Full Chassis FEM Analysis**

A finite element model of the car was constructed to evaluate the stiffness of the vehicle as built, using Ansys 17.1 with the ACP (Ansys Composite PrePost) plugin to help with constructing the laminate in the finite element model. ACP has two components: a preprocessor, which is used to define materials and laminates over desired geometry, and a postprocessor which manipulates and displays solver outputs to evaluate just about any result or quantity, i.e. stresses and strains in each
ply, interlaminar stresses, etc. We were only concerned with deflection in our case, so the ACP postprocessor was not used, and our deflection results were taken from standard Ansys mechanical. Although previous analyses had used Abaqus, which is the program taught in the FEA class at Cal Poly, Ansys was used because of ACP, and because we considered Ansys to be more user-friendly than Abaqus. The hope was that this might result in more team members using the chassis model and/or becoming familiar with FEA, even if they had not yet taken the class.

One of the first steps in creating any FEM is deciding how the model will be created. We chose to model the monocoque as a 2-dimensional surface and the suspension links as 1-dimensional beam elements. In both cases, this is a good method since composite layups are well suited to surface modeling (length and width are much greater than thickness of the laminate) and suspension links are well suited to being modeled as beams or trusses (length is much greater than either width dimension). This type of modeling is also great for studies or iteration, since things like number of plys or tube thicknesses can be changed without having to update any geometry. Previous years analyses had included some sort of engine model, but since our engine is soft-mounted and would have no stiffening effect, it was left out for the sake of simplicity. In the interest of time, several other simplifications were made. Neither roll hoop brackets nor suspension brackets were modeled, and smaller cutouts such as the steering rack, exhaust, and electronics holes were not modeled. In the future, taking the time to model brackets, or at least approximate them with a compliance inducing feature like a bushing element would be a big improvement.

To get the chassis and suspension geometry into Ansys, first we had to make the surface and beam models. The chassis surface was extracted using the Midsurface command in Solidworks, and was converted to a paratool (.xt) file to import into Ansys. The suspension links were created by making a 3d sketch, and copying over points from the suspension guide sketch that defines the position of the links. Uprights and rockers were sketched in with lines, as they would be represented by rigid links. Once the sketch was complete, it was saved as a STEP file wireframe and imported into Ansys. We made sure that the part origins of the suspension and chassis parts were in the same location, so that as imported into Ansys, everything would be in the right position and ready to go. The roll hoops were imported in the same fashion as the suspension.

Figure 66. Chassis and suspension geometry imported into Ansys as a surface and wireframe.
Each suspension member and roll hoop was assigned its respective tube section and steel as the material, and the chassis laminate was applied to the monocoque surface, making sure to have each section of the laminate oriented in the right direction. ACP made it quick and easy to define a certain laminate, and then apply it over selected regions. The cutout behind the driver had not been modeled in the imported geometry, so the region that would have been cut out was assigned a very very low elastic modulus, almost zero. The uprights and rockers were modeled as rigid in this analysis, and the rockers were allowed to rotate by use a revolute constraint.

The suspension members were attached to the chassis with spherical connections in Ansys. This connection fixes translation and allows rotation in any axis, like the rod ends on the end of each link. The pinball regions of each connection were adjusted to roughly match the surface area of each bracket attaching to the chassis. A “pinball region” in Ansys is defined as a sort of sphere of influence of the connection - how much of the attaching surface is involved in the connection. When the surface is meshed, this translates into which elements are part of the connection. The surface in a pinball region can either be made rigid or deformable. Initially we had wanted to keep the pinball regions deformable as this was a bit more representative. However, we ran into difficulties getting the model to run, and switching to rigid pinball regions fixed this. The roll hoops were attached using a fixed connection, which better represented how the welded roll hoop brackets were attached to the chassis.

The boundary conditions were selected to mirror that of the torsion test fixture. The rear uprights were fully fixed in translation, but were allowed to rotate about the y-axis. The left upright was only fixed in translation in the z-direction to mimic the vertical support of the test, and the right upright, where load was applied, was left free. A 1 pound load was applied in the +z direction (down) on the front left upright. Since this is a linear elastic model, the actual magnitude of the load does not matter. The calculated torsional stiffness value is just the ratio of the moment applied to the chassis (therefore the load, since the moment arm is always the same) to the vertical deflection of the FR upright/angular deflection of the chassis. This ratio is the same regardless of load in a linear model. If nonlinear contacts or material behavior were used, then the load would have to match the load applied in a test.

**Results**

A static structural analysis was run, and a deformation probe was set up to output the vertical deformation of the FR upright. For a 1 lbf load, the deflection was $2.34 \times 10^{-3}$ in, corresponding to a torsional stiffness value of 1665 ft-lb/degree. This value was 11% higher than our target value which was reassuring, but given our simplifications to the model (especially not modeling suspension brackets) we weren’t entirely convinced that our tested stiffness values would meet our goal.
Figure 67. Contour plot of vertical deflection of the chassis model, with deformation scaled up to be clearly visible. The maximum deflection occurs at the front right upright, as expected.

The model results also gave visual insight into what regions of the chassis were the most compliant. The figure above shows how the cockpit opening warps as the chassis twists, while the front and rear sections do not deform as much. The compliance at the cockpit opening, especially the corners, still has a large influence on overall chassis stiffness.

We had far less time than we would have liked to perform detailed stiffness analysis for this project, and more time would have allowed us to build a more detailed and probably more accurate model of the car. If future teams continue to use our monocoque geometry, they should take advantage of the time not spent in Solidworks to improve the state of the torsional stiffness model. As well as making it more detailed, more iteration may allow a more varied and lighter laminate.

Torsional Stiffness Testing

We initially considered rebuilding the chassis torsional test rig to address usability and accuracy concerns, although ran into budgetary and manpower issues. The possibility of combining the two apparatus to reduce the total cost makes the new equipment and resources more attainable given our budget. With fixturing being addressed in a cheap and non-resource intensive manner in our vehicle design, we could no longer justify the creation of an extensive fixturing and test table within the scope of this project.
Figure 68. Proposed concept for new chassis fixturing table/torsion test rig for Cal Poly Racing.

We tested the overall torsional stiffnesses of both chassis using the torsional stiffness test fixtures utilized in prior years. This was done by constraining the two rear corners at the hubs, constraining one front corner only in the vertical direction to allow horizontal movement, and leaving the fourth corner unconstrained where displacement was measured. The setup is shown in figures 69 and 70.
Figure 69. Chassis stiffness testing setup.
The resulting torsional stiffness for the combustion car was quite a bit softer than the electric car, at 701 ft-lb/deg compared to 936 ft-lb/deg. The data from both tests is plotted below:
The large difference in stiffness between the two cars is an issue because it shows how two mostly identical cars can be affected by variability in the manufacturing process and/or defects. If this should happen to future cars, it should be more thoroughly investigated and a definite cause should be found if at all possible so that the team can close the loop on the issue. One theory is that the plastic inserts and possibly the bearings in the suspension became more compliant over time - noticeable slop was seen on the right front rocker of the c-car during test. Another possibility is that although the c-car was torsion tested before its rocker mount failure occurred, some damage may have already accumulated, reducing the stiffness of the connection. Although the core was placed a different way on each car, this is not expected to be the cause as the thickness of the core was identical on each car.
Suspension Stiffness Test Fixture Design

Another long standing question we hoped to address was the subscale testing of the suspension links, and if they produced expected stiffness and strength results. This is critical because all forces must be resolved through these links, and an accurate understanding of how they function is needed. Testing fixtures were developed in order to be able to test these links to failure on the Instron machine. The suspension components to be tested can be placed into three general categories: two-force members, three-force members, and four-force members. Two-force members include all pull rods, push rods, tie rods, and any other component that is only loaded at two points. These members can be assumed to be in pure tension/compression with no bending or torsion.

Three-force members include all control arms that are loaded at three points. This includes the rear upper control arm, and front lower control arm. These components being constrained with two points on the chassis, allow the free point to travel in fixed axis rotation. These components can be loaded in compression/tension toward/away from the rotation axis, or loaded in bending in the direction of the rotation axis. These loads will be seen under forward acceleration and braking/deceleration. Currently, we are choosing to focus our attention on purely the compression/tension compliance. As time permits, we will further analyze compliance in these members with braking/acceleration forces introduced.
Four-force members include all components that are loaded at four points. These components include the front upper control arm and the rear lower control arm. These control arms see an additional point force from the rear pushrod and front pull-rod near the outer upright point. Because these additional loading points are so close to the outer point, we will resolve the forces to the upright and will only analyze the tension and compression compliance of these members.

Each of these members has several components within them. There are machined pieces for the spherical bearings to press into, tubes welded to that, threaded bungs welded to the tubes, and rod ends threaded into those bungs with jam nuts. Each of these components stiffness will be a combination of these sub-component systems. Strain gauges will be placed on the tube members and the machined pieces to measure compliance in these components and determine how much of the compliance is from the rod ends, and how much is from the machined pieces and tube. We were not able to test the stiffnesses of these suspension members in time for completion within this project.
**Hardpoints and Loads**

One of the largest concerns with building a full monocoque is proving that we could produce a lightweight structure that is not prone to failure at highly loaded attachments. Localized reinforcements have been chosen as an emphasis primarily because of reliability issues in past seasons. On average, prior vehicles experienced 2 failures per year that have hampered testing days. Each failure eliminates a weekend of testing, thus each failure having the potential to eliminate approximately 10% of a vehicle's testing time. With two cars now moving to a full monocoque, the estimated attachments through composites are expected to triple per vehicle, with the potential for failure eliminating much needed testing time.

Using measured masses from the 2015 and 2016 vehicles, the average vehicle composite attachment weighed 0.2 pounds in backing plate alone per joint (not including laminate weight increases). These backing plates were used to spread the load throughout a composite surface to increase joint strength to acceptable levels. With the potential for approximately 40 attachments per vehicle, the weight of the backing plates alone has potential to offset the weight benefit of a monocoque. Thus, a lighter solution was pursued.

**Prior Failure Analysis**

The team’s typical method of local reinforcements have been cylindrical inserts in a balsa core replacement, with the balsa contributing to out of plane loading and the bonded cylindrical insert to increase bearing strength in the composite panels in plane loading. The machined inserts also presented a potential manufacturing bottleneck when considering scaling up production for all attachments to both vehicles. With the prior reinforcement methods, the below table summarizes last years failures and likely causes.

<table>
<thead>
<tr>
<th>Location</th>
<th>Description</th>
<th>Likely Cause</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lower A-arms</td>
<td>Core failure through laminate skin at edge of tab</td>
<td>Missing out of plane reinforcements</td>
</tr>
<tr>
<td>Rocker Mount</td>
<td>Bracket puncturing through skin</td>
<td>Poor tab-chassis interface design</td>
</tr>
</tbody>
</table>

*Table 10. 2016 chassis failures.*

Front suspension tub failures has been concentrated at the rocker mounts and lower control arm mounts. No complete failures have been observed in the upper control arm attachments. As we have only ran a composite tub in the front of the vehicle, we are not certain if any of the plethora of rear suspension or engine/drivetrain attachments will be prone to failure.
Figure 75. Failure of a lower control arm mount on the 2016 tub. Note how the outboard edge (left edge in the left picture) has sheared through the skin, due to lack of reinforcement.

Load Categorizing

For the next portion of the study, significant loads the vehicle was expected to see were categorized. All loads are considered alternating, exhibiting both tension and compression into their joints. Stiffness values for each joint will vary based on whether it is a tension or compressive load, but strength properties will be assumed to be constant.

After looking at the different loading scenarios, the significant dynamic loads appeared to be coming only from the suspension and engine subsystem. The electric vehicle drivetrain is not projected to dump as much load into the chassis as prior designs due to the self reacting mounting system. The battery box, while competition mandated load requirements were strict, was determined to not need additional reinforcement to pass competition and dynamic loading requirements. Loads were categorized for additional systems such as the brake pedal assembly, steering rack assembly, and aero mounting, but those systems were not projected to insert enough load into the chassis to cause a failure.
<table>
<thead>
<tr>
<th></th>
<th>Front Link Loads (Magnitude) lbf</th>
<th>Rear Link Loads (Magnitude) lbf</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cornering (1.8g)</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fore Lower A Arm</td>
<td>770.7</td>
<td>Fore Lower A Arm</td>
</tr>
<tr>
<td>Aft Lower A Arm</td>
<td>88.7</td>
<td>Aft Lower A Arm</td>
</tr>
<tr>
<td>Fore Upper A Arm</td>
<td>49.8</td>
<td>Fore Upper A Arm</td>
</tr>
<tr>
<td>Aft Upper A Arm</td>
<td>539.0</td>
<td>Aft Upper A Arm</td>
</tr>
<tr>
<td>Tie Rod</td>
<td>284.1</td>
<td>Tie Rod</td>
</tr>
<tr>
<td>Push/Pull Rod</td>
<td>818.9</td>
<td>Push/Pull Rod</td>
</tr>
<tr>
<td><strong>Braking (1.5g)</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fore Lower A Arm</td>
<td>343.3</td>
<td>Fore Lower A Arm</td>
</tr>
<tr>
<td>Aft Lower A Arm</td>
<td>659.3</td>
<td>Aft Lower A Arm</td>
</tr>
<tr>
<td>Fore Upper A Arm</td>
<td>49.4</td>
<td>Fore Upper A Arm</td>
</tr>
<tr>
<td>Aft Upper A Arm</td>
<td>906.5</td>
<td>Aft Upper A Arm</td>
</tr>
<tr>
<td>Tie Rod</td>
<td>115.2</td>
<td>Tie Rod</td>
</tr>
<tr>
<td>Push/Pull Rod</td>
<td>861.0</td>
<td>Push/Pull Rod</td>
</tr>
</tbody>
</table>

**Table 11.** Cornering and braking loads into the different suspension links.

<table>
<thead>
<tr>
<th></th>
<th>Front Engine Mount [lbf]</th>
<th>Rear Engine Mount [lbf]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Vertical Load (z-axis)</strong></td>
<td>-2250 min, 1125 max</td>
<td>-1690 min, 1125 max</td>
</tr>
<tr>
<td><strong>Longitudinal Load (x-axis)</strong></td>
<td>-470 min, 470 max</td>
<td>-470 min, 470 max</td>
</tr>
</tbody>
</table>

**Table 12.** Anticipated maximum vibrational engine loads at the mounting locations on the front and rear of the engine.

A desired minimum factor of safety of 2.0 was the goal for joint capability. With use of uniform reinforcements, a majority of joints will greatly exceed this safety factor. As detailed later, all joints exceed these margins.
Design Considerations

To define the scope of this study, the attachments were separated into two categories: composite reinforcement and tab design. The chassis senior project team took lead on the responsibility of the composite design portion. This included the design and selection of the composite laminate, inserts, and bolt spacings. The tab design was assigned to the appropriate subsystem, with the senior project team creating guidelines and providing advice for certain systems based on loading conditions.

Based on the vehicle's emphasis on reliability, the primary design constraints of the joints were specific strength and manufacturing time. While stiffness did play a role in the selection process, we determined that reducing manufacturing time and potential failures on the vehicle were significantly higher priority to getting a high placing vehicle this year.

The first step we took prior to designing certain solutions was to research the driving factors into composite joint design. For developing the understanding of what influences the design of a hardpoint, technical documents [10] were referenced to develop the driving trends behind the strength capabilities supporting the local load introductions. From this research, the most effective approach to these joints appeared to be centered around the selection of inserts and core replacement. In most cases, the core replacement handled a majority of the load out of plane of the laminate, and the insert and potting bore the majority of in plane loads. Core replacements can be utilized to help us through increasing the out of plane shear and compressive strength of the core property, each vital to reducing failures in the core. Additionally, special care must be taken at discontinuities of the core (such as the splicing of two core types), as the interface can be prone to failure. Inserts and potting help in distributing in plane loads into the skin, while also providing a barrier to protect against the bolt from chewing up fibers. These viewpoint contradict prior methods our team had of designing joints, in which backing plates and ply thickness were the primary tools used to increase joint capability.

Out of Plane Loads

Focusing exclusively on the out of plane and core replacement portion, the primary downside of core replacements over other methods was a slightly more complex chassis layup. It was quickly determined that the additional time needed for core replacements was not considerable because of the small quantity of joints requiring the reinforcements. The different types of core replacements considered were balsa, garolite (fiberglass with hardened resin), high density core, and pan downs (carbon puck). A quick comparison study was conducted to downselect options.

Initially, the leading contender for the core replacements were pan downs with a carbon puck reinforcement, which were featured on various competitors vehicles. Our competitors ran this reinforcement method on suspension attachments and other high in plane loading locations. The advantage of pan downs was an elimination of a soft core, which often proves to be the “weak link” for out of plane loading, and bringing the two faceskins together to increase the bearing strength for in plane loading. Despite the advantages, pan downs were ultimately eliminated from consideration due to a variety of concerns. A primary concern was the reliability of pan downs when you have the bolted connection near the core ramp down. Pan downs near attachments have well known problems with reliably sizing the area where the faceskins meet, as the variations in strength are largely based on difficult to avoid manufacturing defects. Additional reinforcements and beefed up laminates are often needed in pan down region in order to guard against this, which dramatically increases both chassis manufacturing time and attachment weight. Additionally, without prior team experience in machining
core ramps, learning this techniques would require a large time commitment by the team along with also dramatically increase core forming and layup time. In terms of chassis weight, pan downs prove to have little advantage in terms of weight saving, and will likely result in a heavier joint than the other proposed methods. While pan downs are eliminating a small section of core weight, you are adding a “puck” of laminate to replace it. Based on area weights, relentless stalking competition studies, and conversations with other teams, it was determined that the top teams who use pan downs utilize a carbon puck of approximately 0.25” thickness, required to overcome the prior mentioned disadvantages. This would equate to an approximately .05 lb weight increase per hardpoint over alternatives, or approximately 2-2.5 lb gain over the whole monocoque over other proposed methods. Additionally, an effect not quantified was the loss in stiffness as a result of this joint. By eliminating core in the local regions, albeit by essentially replacing it with solid laminate, the effective bending stiffness is significantly decreased. This stiffness decrease was not quantified as the carbon pan downs were removed from selection due to the weight and reliability concerns.

![Figure 76. An example of core pan downs being used on competitor vehicles.](image)

The reason to not use a garolite reinforcement is very similar to the reasoning with the carbon puck. While garolite would likely prove superior in direct out of plane shear strength compared to the carbon puck due to it’s resin, the inability to cure the puck with the faceskin laminates is a considerable downside that promotes failure in the interface region. For the size of garolite needed for our loading scenarios, all of the concerns of panning down the core and reliably bonding and integrating garolite with honeycomb and laminate become an amplified concern. The weight penalty for the garolite reinforcement was determined to be, at minimum, equal to the carbon puck.

In consideration of balsa reinforcements, which has been a team favorite in the past, the primary advantage of balsa reinforcements would be the ease of manufacturing and availability associated with them. By utilizing a solid cell material, it has been suggested that an insert could be avoided, with just drilling through the area and bolting it through. The lack of any insert was not pursued due to the concern of dramatic bearing strength and stiffness decrease with bearing on exposed face skins. Balsa reinforcements did exhibit considerable out of plane strength and less potential reliability issues when compared to laminated core replacements.

When comparing balsa reinforcements with higher density honeycomb, it quickly become apparent that high density honeycomb had a strong advantage. The specific compressive strength of the honeycomb per unit weight exceeds that of balsa. Balsa did show mechanical properties comparable to mid density honeycomb, although they were at a disadvantage to higher density core. There are
numerous types of honeycomb better suited to compressive strengths than aluminum, although aluminum was used as a conservative baseline.

![Figure 77. Comparison of specific compressive strength of aluminum honeycomb vs balsa.](image)

Additionally, we suspect honeycomb is much more effective in interfacing with other core types, as the open cell core seems like it would allow better splicing and potting adhesion than the closed cell balsa. We determined honeycomb would be easier to interface an insert to than a closed cell core during the vehicles manufacturing process. The primary downside to high density honeycomb was cost and availability, which is largely sponsor based. Additionally, the out of plane shear strength of balsa (assuming isotropic allowables for balsa due to lack of comprehensive material data) does appear pose disadvantages towards honeycomb. Determining how this affects joints should be left to future years, as we did not use shear properties to downselect core options.

![Figure 78. Comparison of specific shear strength (1z) of aluminum honeycomb vs balsa.](image)
With the comparison of different options, the decision was made to utilize high density honeycomb if available. Balsa could prove an acceptable, low cost solution to the honeycomb if availability becomes an issue. To finalize the decision, a large quantity of high density fiberglass reinforced core was acquired, which appears to be more effective than aluminum honeycomb in compression. This supply provides us with a more than adequate supply for the foreseeable future.

**Potting Sizing Trends**

After the decision was made on core replacement methods, attention then shifted to the insert design and in plane properties. As stated before, the purpose of inserts is to transmit shear loads into the facesheet, and in the case of potting, provided additional localized strength for out of plane loading. Additionally, potting compounds are widely used to provide an glued interface between the insert and core material along with adding stability in the case of a moment.

The different types of inserts under consideration were the spindle double shear joints, spindle single shear joints, and unflanged straight shank inserts. The straight shank inserts used in prior years were not selected for consideration as they did not provide the surface area increase that flanged inserts possess, and would be more difficult to reliably pot and adhere to the faceskin among other things. While straight, unflanged inserts were considerably easier to manufacture, this additional manufacturing restriction was not considered more important than the strength increase of a flanged insert.

![Figure 79. Comparison of a spindle insert (left) and a type of flange insert (right).](image)

The first consideration between the flanged and the two types of spindle inserts was installation ease. Spindle inserts add significantly more complexity in the manufacturing and installation process over the alternative. Two separate diameter holes (or a much more complex insert geometry) is needed to allow the flange of the insert to sit on the laminate. Additionally, manufacturing spindle inserts (if we choose to manufacture them in house) was significantly more time consuming over a flanged insert, with more complicated geometry and the potential need to manufacture twice as many components (one for each side of the panel).

The next consideration was strength. Looking at our projected loads and bolt patterns, the strength increase of a spindle insert over a flanged insert was not considered absolutely necessary to pass loading requirements. While spindle inserts did provide superior mechanical properties compared to a
more simpler flanged insert, we did not believe this strength increase warranted selection over a simpler to manufacture and install flanged insert. Additionally, even though the flanged insert has to take a strength penalty, it was still more than sufficient to meet our loading demands. Thus, we decided to utilize a flanged insert. Additionally, a fender washer the same OD as the flange could be utilized to mimic moment allowables to the spindle insert.

In order to get a relative scale of how much potting was required for a certain strength on the flanged insert, a model of the insert was developed from ESA and NASA manuals. Figure 40 shows how insert potting affects strength properties for an insert and laminate combination likely close to the combination we would use on the vehicle. This model also showed that the most effective way to handle in plane loads is to increase the potting diameter, the thickness of the faceskin, and the diameter of an adhered insert flange. Inserts with a certain potting diameter offer the potential ability to eliminate the need for core replacements in a majority of joints.

![Insert Load Capability](image)

**Figure 80.** The graph displays the increase in strength of an insert when increasing the potting diameter. As is shown, in plane properties are very sensitive to potting radius, while out of plane strength are primarily driven by core replacements.

The next obstacle was deciding how the insert would interface with the skin. The two primary considerations were allowing the flange to sit on the faceskin or embedding the flange into the faceskin. Through embedding the insert within the skin either through careful post processing or a cocure, the insert should see increased strength capacity through increasing bond strength. Although, flanges on the skin would be significantly easier to manufacture. We decided upon the method of sitting the flange on the skin, and to just add an additional step to bond the insert flange to the skin prior to potting.
Bolt Patterns and Attachments

With insert design squared away, we turned our attention to the vehicles standard mounting tab. Historically, our team has utilized two bolt tabs which were favored because of their ease of assembling and manufacturing a tab, along with any bolts seeming unnecessary. After completing background research on the inserts, it became apparent how vulnerable composite attachments were at failing due to a moment load. Two bolt tabs must resolve any force off of their line pattern with a moment, which would make these connections especially vulnerable. We decided that switching to a minimum three bolt pattern at every highly loaded location was needed to prevent any potential moment failure. In terms of the suspension connections, in order to offset the weight gain of the addition of an extra bolt, the bolt size was decreased to achieve a comparable weight penalty.

Figure 82. The above image shows the front suspension. Notice the three bolt patterns on the lower control arms.
After determining the bolt pattern, the bolt spacing was the next topic to focus on. It was researched that a reduction factor for closely spaced inserts must be implemented based on potting dimensions. Spacing becomes a concern because the combined stress state creates a point of maximum shear stress a distance away from the inserts. This reduction factor for closely spaced inserts is not exclusive to composite attachments, as metals and any other close bolted connection have similar requirements. This sort of loading often contributes to core failure at a certain radius between inserts. The shear stress at a distance from the insert decreases nearly linearly, so it makes reasonable sense to incorporate a reduction curve. Spacing patterns were sized based on a compromise between potting radius, reduction factor, and tab design. A minimum spacing which resulted in a 30% reduction was chosen as a compromise between all of the aforementioned factors.

![Insert Reduction Factor](image)

**Figure 83.** The figure above is an example of the reduction factor on inserts based on spacing. As is shown, it is a linear relationship based on spacing and potting diameter. As the spacing to potting ratio reaches 10, there is no knockdown factor.

![Core shear stress distribution](image)

**Figure 84.** Core shear stress distribution in the vicinity of a potted insert. Two inserts loaded near each other compound stress states, resulting in the reduction factor.
Another topic that was debated was whether the insert flange should sit directly below the tab, or on the other side of the laminate. There was compelling argument for both reasons. On one hand, the in-plane strength is not affected by which side the flange head sits and the flange could eliminate any need for a backing plate. Although, we decided to place the tabs on the insert head as it results in the most reliable and controlled joint load introduction over the alternative, which mesh well with the joint reliability goal. While there could be a strength decrease due to a small moment induced and reduced out of plane surface area, we believe the added benefits outweigh the risk. Not having the tab in direct contact with the skin allows for loads to be directed into the laminate in a controlled fashion, which would result in easier to predict joint allowables. Additionally, by physically offsetting the tab from the faceskin, you force all loads to react through the potting compound instead of smearing it over unreinforced area. Placing the tab directly on the skin was the likely cause of at least one failure last year. Additionally, the preload required for bolts will reduce, as a carbon to metal interface has significantly less friction than a metal to metal contact. The downsides to mounting the tabs offset from the surface is that tab design becomes crucial as it sees higher stresses under certain loading conditions, and tolerancing could be an issue if not kept in mind during the design phase.

**Insert Testing**

Testing was conducted to determine the validity of the calculations for the insert geometry and to assess the feasibility in our vehicle. Additionally, the insert testing was used to determine how close we were to triggering separate failure modes based on the method load was introduced into the skin. As is usually the status quo with composite strength, you can only beef up one aspect of the structure so much until a new failure mode takes prominence. With limited time and resources, we only decided to test the in-plane properties of the insert after looking at the highly loaded joints in our vehicle, specifically the front lower control arms, and how they were previously susceptible to failure. The reason we were not specifically interested in the out of plane allowables of the insert was that the highest loaded out of plane joint is seen on the front upper control arms (since engine spreads out its load to a large bolt pattern), and prior years ran similar joints on those connections without an issue. With our team pressed for time, we decided that in-plane allowables were significantly more critical to obtain for a reliable vehicle this year.

The shear tests performed looked at the difference between 0.625” sized flange inserts and 1 larger 1” flange. These were chosen to determine if insert flange diameter (and the associated bearing allowable increase) would be prominent at this level of potting radius. The potting radius was kept consistent between the two sizes at a .5-.6” radius (large variation based on assumed manufacturing tolerances). The laminate thickness in the test panels was kept conservatively thin in an attempt to artificially trigger failure modes associated with the core, potting, or insert assembly itself. Additionally, test results with the thin laminate could be used as a worst case scenario for how the inserts would perform in the vehicle.
The test results did not show a conclusive failure mode as two different modes were observed during post test inspection. Failure was observed at debonding of the adhesive to laminate interface, which allowed the joint to gradually fail and eat away at the skin due to exceeding the bearing strength of the skin. Additionally, the potting sheared in half. It is unclear in this test if the potting or adhesion to the faceskin failed first, or if it was simultaneous. The calculations suggest that by increasing the flange diameter, more adhesive strength can be obtained. Additionally, “pad-ups”, or localized skin thickness increases, would dramatically increase the bearing strength of the laminate.

In the test for the increased flange diameter, a core shear failure mode was observed as the primary cause of failure. Additional failure modes similar to the previous smaller diameter flange tests were observed. The increased adhesive area did little to increase the total strength of the system.
In the test data, one test resulted in a significantly lower value than the others. This was quickly ruled as a result of fixture and test specimen misalignment rather than inherent spread in the data due to the potting method. Allowables were based on the lowest failed insert of the other three specimens.

Conclusions from the insert testing is that the potting diameter is the limiting factor for shear strength of an insert using our particular geometry and method, and an increased flange diameter does little to help the system strength for in plane loading as it would just trigger another failure mode. This is likely due to the result that the potting compound is the weak link in the chain, and will fail prior to the benefits seen by a larger flange. Additionally, a larger flange can result in bringing out another failure mode of core shear. In plane strength results of the inserts showed that the inserts exceeded all in plane requirements of the vehicles joints to pass margins, although took less load than predicted. Failure was predicted in the 1200-1400 pound range, but was tested to failure at around 900 pounds. This partially justified the use of a high factor of safety of 2.0 in the joint design. As an added measure, we also decided to utilize pad ups around highly loaded shear joints.

Investigating a concern brought up during team CDR that loads are difficult to introduce rigidly into a composite structure, the stiffness of these tests were determined to approach 40,000 lb/in (pounds applied to produce an inch of deflection) per insert. For reference, the effective stiffness of a suspension link calculated as part of the 2016 suspension subsystem approached 100,000 lb/in per
link. Granted, this test was in no way designed to accurately characterize stiffness, rather it serves as a rough ballpark estimate. With stiffness values of an insert approaching that of a link, insert stiffness is worth further investigation and would require improvement in testing methods for future years.

**Finalized Insert Design**

With basic proof testing proving that a flanged insert will meet all structural and manufacturing requirements, the decision was made to pursue this option. Purchased flanged inserts were searched for, although none were found that met our teams specification. This led to the decision to manufacture the inserts in house. When considering the different material and manufacturing options present, we decided to outsource manufacturing to Stratasys to print the inserts out of Ultem. Not only did our FSAE team have a strong partnership with Stratasys, but their printed Ultem material provided superior mechanical properties compared to alternative solutions. When tested against aluminum inserts of identical geometry, there was no notable strength decrease.

Last year, standard thickness backing plates were used for all composite attachments. These attachments proved heavy, as attachment backing plates weighed on average twice as much as the tab. The backing plates that will be utilized this year were a fender washer of the same diameter to the insert flange. Any washer size above or below that value should be unnecessary. Additionally, relatively thin washers can be formed to complex curvature, allowing bolts closer to radius to have sufficient backing. A flange diameter of 1” and a potting radius of .625” with a .16” diameter bolt was chosen as the best combination of weight and strength.

The installation procedure we found that produced the most consistent results requires drilling a slight clearance hole for the shank of the insert through the laminate. Then a high tech assembly consisting of a custom ground allen wrench (rest in peace all smallish FSAE allen wrenches) attached to a hand drill was used to clear out core to a certain radius. The insert was then inserted and the face bonded to the skin. Once the faceskin to flange adhesive hardened, the flange holes were drilled through the top skin face and filled with potting. Finally, the remaining insert shank was ground away with an air tool, which allowed for one standard shank length to be used on a variety of thicknesses.

Each assembly was measured as .01 pounds per insert. The new style of insert would eliminate the need for a majority of core reinforcements and is projected to weigh approximately 1 pound throughout the vehicle. This was measured as 1.5 pounds in the actual vehicles. This could be a result of a leaning towards a conservatively large potting diameter when clearing out core for vehicle inserts.
Full Tab Testing

With the insert geometry finalized, we decided to conduct one more test utilizing standard inserts with a standard suspension tab. As this took place after the team’s design review and printing mishaps, the tab design had already switched from aluminum to steel. A standard three bolt suspension tab was utilized with standard insert and potting geometry.
The failure mode showed that the tab was the first part to fail instead of the composite. The strength of the tab showed that the joint could withstand nearly 2,000 pounds before failure. This result solidified a minimum factor of safety through our vehicle of 2.2, which met the target factor of safety goal of 2.0.

After the results of these tests, we were curious what the effect of the system would be if you eliminated the tab failure mode. To achieve these results, the flanged insert was replaced with a carbon plate that covered the entirety of the area below the tab. The potting radius and bolted connection was kept the same. This test also resulted in a tab failure at a similar force.

![Figure 91. Failure mode of a suspension tab sitting directly on a padded up region of the laminate.](image)

The results of these tests showed that both methods failed at similar strength values due to the tab being the weak point in each system. Eliminating the tab failure mode was not achieved. The stiffness values between the two tests showed that the tab attached directly to the laminate was a fair amount stiffer. It appeared conclusive that this stiffness reduction was a result of the flanged inserts unfavorable tab boundary conditions coupled with a poor tab design. It is inconclusive whether the elevated tab resulted in higher strength allowables for the laminate, which was its intended goal. As this test was completed too late in the year to address the attachment’s stiffness issue in the 2017 vehicle, future teams should make it a priority to both redesign the vehicle’s suspension tabs and reassess whether placing the tab offset from the composite surface was the right decision.
Failures

The only major failure during testing was the failure of the front rocker mount. This suspension-chassis interface sees the highest loads, and the cause of the failure was determined to be that during installation, the insert was not potted. It was also discovered that the hole pattern for the rocker mounts were below the acceptable spacing limit set by the insert reduction factor due to fitment constraints. While this was not the contributing factor to failure, it does serve as a warning for further attention in the design phase.

Figure 93. Rocker mount failure from the inside of the chassis.
One other partial failure observed was the dimpling of the core on the bottom of the car chassis. This was observed after one of the first testing days, and was deemed a non-issue. We ran with the dimpling for the entirety of the season without incident. During the 2017-2018 season, the team members cut apart the dimpled section to look at the failure mode of the structure. The primary driving failure mode appears to be a core failure seen to the right of the potting compound assembly in the picture, although it was difficult to pick out due to the joint use after the failure. The current suspect for failure was an excessive moment load that was originally intended to be resolved due to the mount design. The theory for the driving failure mode in this situation is when the engine displaced on soft mounts, the added shear and moment loading in the base plate overloaded and put excess moment at the composite interface. We believe this can be rectified in the future most easily by an engine mount redesign.
Impact Attenuator

SAE rules require that the car be fitted with an impact attenuator device on the front of the vehicle, to protect the driver from a head-on collision. The device must absorb at least 7350 J of energy, with a maximum average deceleration of 20 g and max peak deceleration of 40 g, and has a minimum footprint of 200 x 200 x 100 mm. If the team designs their own attenuator, it must be tested, and the data submitted to SAE. Three concepts were developed for the impact attenuator design: The structural carbon skin used in previous years, the standard FSAE impact attenuator, and a block of energy-absorbing aluminum honeycomb. Additionally, an Anti-intrusion (AI) plate must be fitted between the attenuator and the front bulkhead to prevent any foreign objects from penetrating into the footwell area.

Since 2013, the team has designed a roughly pyramid-shaped prepreg skin that absorbs energy by fiber failure and delamination. The design is theoretically very lightweight, but in the past has proven difficult to manufacture properly, due to the tight, complex curvature and many plies required.

![The 2016 impact attenuator undergoing crush testing.](image)

The next option considered was the standard impact attenuator specified by SAE rules, made of blue impact-absorbing foam. The standard attenuator does not require any physical testing, and so would reduce time spent on manufacturing and testing. However, its fixed dimensions limit the geometry of the front section of the chassis, and it is expensive to buy, with prices ranging from $180-250. Given the availability of donated materials, it’s likely that the standard impact attenuator would be the most expensive option.
Many teams use a large block of aluminum honeycomb bonded to the Al plate to absorb energy. Honeycomb is inexpensive compared to the standard impact attenuator, and its energy absorption properties are linear and easy to modify. The length, width, and height are simply increased until the requirement is met. Depending on the size of the block required, there would be more freedom to shape the nose fairing to meet the aerodynamics needs. However, there was concern that the assembly may add excessive weight. The simplicity, ease of understanding, and low cost of this design seemed attractive to use so we took a closer look.

Using equations provided in the Hexcel Honeycomb Energy Absorption Brochure [6] and datasheets for 5052 aluminum honeycomb [7], a spreadsheet was developed to calculate energy absorption and deceleration based on IA dimensions, core density, cell size, precrush, and foil thickness. The goal of the spreadsheet was to determine the optimal type of honeycomb to be used. The spreadsheet was validated by crushing a sample of 0.7" thick 3.1 pcf (pounds per cubic foot), 3/16" cell size core on the Instron, to 14% difference between the test and the calculated energy absorption. A calculation was also added to include the force required to fail the front wing attachments, a key player in the maximum deceleration requirement for the impact attenuator.

Using datasheet values for all density/cell size permutations of the 5052 honeycomb, we plotted specific crush strength against density to determine the most efficient honeycomb characteristics.

Cell size had no effect, and specific crush strength was proportional to density. 8.4 pcf core with a 5/32 cell size would theoretically be the most efficient core for an Impact Attenuator. However, when it was run through the spreadsheet, it was discovered that it was too good - when the rules-specified minimum IA dimensions (4"x 8"x 8") were put into play, the calculated energy absorption was 15.4 kJ, twice as much was necessary, and both average and peak decelerations were too high to meet rules. Additionally, it was quite beefy, at 1.2 lb.
We reduced core density until we came to the most efficient size that just barely met the energy requirement - 5.7 pcf with 3/16 cell size.

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**Figure 99.** The most efficient rules-compliant honeycomb, plugged into the spreadsheet.

Weighing in at a much more reasonable 0.84 lb and meeting deceleration requirements, we inquired to Plascore and Hexcel about availability of this core for donation or purchase. It turned out that Plascore sells a standard FSAE honeycomb impact attenuator, certified pass the IA test, for $25 per unit. The attenuator weighs 0.85 lb and is made from 5.7 pcf, 3/16” cell honeycomb - Exactly the same as our calculated custom attenuator would have been! Satisfied with this result, we decided to purchase the standard Plascore IA. 4 units were purchased; One for each car, one for the IA test, and one spare.

**Figure 100.** Two of the purchased Plascore attenuators, after delivery. The pre-crushed surface is apparent on top.

The honeycomb attenuator requires a nosecone fairing. It was decided to make this a thin carbon skin, with a weight estimate of approximately 1 lb for a 2-ply ‘cone. This brings the total IA/nosecone assembly to approximately 1.9 lb, still 0.2 lb lighter than the lightest structural ‘cone ever produced by the team, and much quicker to manufacture.
Test Fixture

For the testing of the impact attenuator and AI plate we designed and manufactured a test fixture to replicate the geometry of the front bulkhead of the chassis while being extremely stiff. The manufacturing process for the test fixture is included below.

**Figure 101.** Cutting the anti-intrusion plate testing fixture base on the Hangar plasma cutter. Even on the highest power and slowest speed settings cutting all the way through the 3/4 inch plate was a challenge. Slag still held the edges on after cutting all around and the circuit breaker on the plasma cutter tripped multiple times while cutting the plate.
**Figure 102.** The top of the anti-intrusion plate testing fixture after being tack welded in a couple places. The tubing was clamped to the frame table in the Hangar for the first set of tack welds to help prevent warping.

This year because we moved away from a nose cone that doubled as an impact attenuator and to a more standard aluminum honeycomb style, we wanted to create a new, very rigid test fixture for our impact attenuator and anti-intrusion plate combo. This fixture consisted of a piece of inch thick steel plate with a square steel tube frame resembling the front bulkhead of the chassis welded on to it. The base plate of the fixture was cut using the Hangar plasma cutter with the optical follower used to follow the shape from a printed template. The plasma cutter was set to its highest output power and lowest travel speed, and still had difficulty cutting through the metal. The profile had to be broken up into multiple cuts because the plasma cutter overheated a couple times and had to cool down before continuing the cut. The rest of the fixture (the square steel tube frame) was tig welded together while clamped to the frame table to ensure flatness of the section that would interface with the AI plate. The supports and the base plate were also tig welded to the square steel tube frame.

**Figure 103.** The fully welded anti-intrusion (AI) plate testing fixture. The tig welds can be seen at the joints, and were ground down on the top of the fixture to make sure the AI plate had a flat surface to sit on.
Figure 104. The anti-intrusion (AI) plate quickly setup on the AI plate testing fixture to get a quick visual confirmation that it will work for testing the AI plate.
Mold Design

For a mold, a plug representative of the part is needed to create the net shape. Due to the volume of material needed and the amount of machining to accurately represent a chassis, plugs are often made with high-density tooling foam, which is durable and easy to machine and sand. If the plug is required to withstand the high temperature of a prepreg cure cycle, this greatly limits material availability and the temperature changes produce a possible point of error with plug degradation. Another factor or interest would be the Coefficient of Thermal Expansion (CTE). During the cooldown cycle, since carbon has a negative CTE and most other materials have a positive CTE, the carbon part would be “press fit” into the mold. Utilizing a carbon mold could make it so the parts pop out of the mold after the cure cycle. While CTE could be overcome with brute force, matching CTE’s would decrease the effort to remove the part from mold (saving member time and effort) and would help in extending the mold life. Additionally, CTE mismatch will result in residual laminate stresses decreasing the intended strength of the part. Thus, the ideal mold would be made from carbon cloth cured with a tooling resin. Generally, such molds are made using prepreg, resin infusion, or wet layup processes. We ended up deciding on a wet-layup tool, because of our unfamiliarity with resin infusion and the more durable temperature-resistant plug materials needed for a prepreg mold.

The chassis will be laid up in two separate halves, so that the layup is easy to perform and multiple people can lay up at once. When the halves must be joined, it is important that they are joined accurately, and the chassis remains dimensionally sound. Our leading concept is using the mold to join the halves. The mold halves will be located with two pins, and clamped together with bolts, locating the monocoque halves together for bonding. This approach will decrease post-processing time and result in a higher-quality part.

As one of the biggest decisions made this year was manufacturing new molds, a system was needed in order to test and compare different options. More than just calculations will be needed to prove the effect of surface finish and thermal expansion of the mold on part quality, and if the mold material would be worth the cost to the team.

Preliminary Testing

We used two existing molds as plugs - the 2013 plaster nosecone mold, which had a proven history as a good test piece - and an old foam CVT cover mold generously given to us by the Baja team. Both molds have tight radii and complex curvature.
Were were able to make two test molds with materials that we had on hand - 12K fiberglass and carbon fiber fabric, and a PTM&W PT2876 high-temp tooling resin. We made a fiberglass mold from the nosecone, and a carbon mold from the CVT cover. Originally, we had planned to lay 8 plies in a quasi-isotropic fashion for each mold, but we ran out of our limited amount of resin on hand and had to only use 3 plies. Additionally, the thick fabric soaked up 1.5-2 times as much resin as was calculated before it was properly wet out, which produced a resin-rich part.

Once the molds were room-temp cured, we released them from the plugs. They came out looking good, with no visible defects except some slight roughness, due to insufficient mold preparation. Even though only 3 plies were used, the shape of the molds made them, especially the carbon mold, nearly rigid. 8 plies with core in large flat areas would be sufficient for a monocoque-sized mold.

Unfortunately, there was not time to post-cure the molds so that test prepreg parts could be made, but the practice gave us a good estimate of the time and manpower required to make a wet-layup composite mold, and we learned how much resin we would really have to use to properly wet out the fabric.

Our final decision was to use the same mold technique. Through generous support from Cal Poly MESFAC and Aero SFC, we obtained 50 yards of 12K carbon fabric, and 12 gallons of PTM&W
PT2520A/B1, a lower-temperature version of the PT2876 used in the test mold, with a service temperature of up to 350 degrees F.

Zodiac Aerospace in Santa Maria generously provided tool design, material and machining services for the male foam plugs used to make the molds. The plugs will be machined on a 5-axis router, then sanded and prepped for the mold layup.

**Locating Features**

In past years, locating suspension points and other components was often problematic and time-consuming due to the lack of good datums or locating features on the tub. We endeavored to fix this by designing locating features into the mold, in the form of drill bushings bonded in during the layup. Once the monocoque is cured, the bushings will be drilled through before removing the part from the mold. This way the positional tolerance of suspension points would be controlled by the accuracy of the CNC router used to machine the plugs, rather than a complex jig using one or more faces of the chassis as a datum. Additionally, locating and drilling all of the required holes would be quicker and easier. The drill bushings would be located in the mold by having corresponding holes in the plug; pins would be inserted into the holes, and the bushings slid onto the pins before the mold layup.

Initially we wanted to place a bushing for every single hole on the monocoques, however by the time we needed to send our model to Zodiac, some suspension points were still not fully mature. Additionally, the engineers at Zodiac advised that drilling a large number of holes into the plug significantly increased machining time, because of the way that they needed to operate the 5-axis router to drill an accurate hole. The compromise was to include holes for some suspension points, with enough to accurately locate simple drill jigs on each flat face of the monocoque. For example, the front floor area requires 20 separate holes to mount the lower control arms, steering rack, and shocks. We had just 4 holes drilled into the plug, which doubled as locating holes for a drill jig to quickly drill the rest. This way we could speed up the machining of the plugs without too much compromise in speed or accuracy of drilling suspension holes. The drill jigs and other suspension fixtures are discussed in the suspension integration section.

**Rocker Boss**

A couple of hours Well before we had to send the completed monocoque geometry off to Zodiac, the decision was made to leave the rocker boss out of the plug and mold. We had already encountered issues with packaging the lower control arm mounts so close to the boss, and we expected that if the boss were a permanent feature in the mold, it would make it very difficult for future teams to adjust suspension points. We designed a mold insert fit into the bottom half of the mold that let us manufacture the boss, while being removable for future years. The insert was manufactured out of the same carbon fiber and tooling epoxy as the molds, and the foam plug that would mold the rocker boss insert was to be machined on the ShopBot router in the Hangar. Pins would be molded into the insert, and these would locate the insert to holes in the mold.

The primary disadvantage of this method is that only one side of the rocker boss insert is controlled - the other side will have the uncontrolled bag-side surface of the composite. We chose the controlled side to be the surface that mates to the mold, so that we could have proper fit-up. Consequently, the side that actually molds the part is uncontrolled. Bumps and imperfections can be sanded flat but the position of the molded surface is now controlled by the laminate thickness of the rocker boss insert.
Ideally both surfaces would be machined to the exact geometry but we ran out of time to design a solution to this. The solution we used was to offset the rocker mount geometry in CAD by the expected mold thickness, and then shim the mount to obtain desired suspension properties.

**Figure 107.** Finalized plug geometry. The rocker boss is removed, holes have been added, and the cockpit opening has been filled in.
Laminate Design

The first step we chose for laminate selection was laminate level testing to affirm datasheet material properties. For context, our team primarily receives prepreg composites through aerospace and prepreg companies who instead of throwing out a batch of expired or defective prepreg, they donate it to our team. A majority of these materials are unsuitable for aerospace use, but are adequate for our application. With data from testing from prior years showing that properties of some of the prepregs were significantly compromised, and complete datasheets from manufacturers difficult to find, we decided that some quick and small scale testing of the laminate could help to narrow down possible materials to use. This would also serve as a method to identify and avoid laminates with significantly compromised properties, which would help to expedite and streamline chassis structural testing. The primary beneficiary from these tensile tests are the stiffness driven models, such as the chassis torsional stiffness, whose values are sensitive to mechanical properties. It is within reason that the large gap between previous chassis stiffness models and tests could be partially due to poor material data.

For the properties we looked to validate, we determined that the in plane mechanical properties of the laminate and the resin integrity were critical to determining if a prepreg was suitable or not. The quickest and easiest methods we chose to characterize these values were the 0 degree tensile test (ASTM D3909), the off axis tensile test (ASTM D3518), and the short beam test standard (ASTM D2344). There are more accurate and detailed methods to determine these material property values, but the chosen tests were selected for their time and resource effectiveness. The tensile tests would be able to accurately characterize all in plane stiffness values, along with giving a good estimate of strength parameters. The 0 degree test is used for all tensile stiffnesses, while the off axis test can be utilized for shear stiffness. Coupling this with a formula to determine out of plane shear stiffness values, these tensile tests would allow you to experimentally determine all laminate properties required for structural modeling.

![Figure 108. Short beam shear test setup.](image)

The short beam method was added as a way to gauge resin integrity in isolating a delamination failure mode that would become critical in thicker laminates. Additionally, we tested hybrid laminates, with
both TC250 and TC275-1 resin systems, so that we could see if mixing these epoxies that cure at different temperatures would have undesirable results. All tests were conducted at room temperature for simplicity.

Next, the prepreg types were narrowed down to availability to the team. The team has build up a considerable inventory of composite materials in the past few years, but is still limited to a few materials stocked in sufficient quantity to build two monocoques. In general, the team usually has a high strength woven cloth fabric and a high-modulus unidirectional fiber available in sufficient quantities to produce a chassis, and this year is no exception. Table 5 details the materials that are available to use.

<table>
<thead>
<tr>
<th>Material</th>
<th>Type</th>
<th>Resin Content [%]</th>
<th>Measured Area Weight [gsm]</th>
<th>Cured Ply Thickness [in]</th>
<th>Description</th>
<th>2016-17 Stock</th>
</tr>
</thead>
<tbody>
<tr>
<td>AS4C 3K 8HS/TC250</td>
<td>Thick high-strength satin-weave cloth</td>
<td>40</td>
<td>644</td>
<td>0.017</td>
<td>Used on 15’ and 16’ tubs</td>
<td>½ Roll (Possibility of getting more from Tencate)</td>
</tr>
<tr>
<td>M46J 12K/TC250</td>
<td>Ultra-high modulus unidirectional</td>
<td>38</td>
<td>334</td>
<td>0.012</td>
<td>Used with success on 16’ tub</td>
<td>2 Rolls</td>
</tr>
<tr>
<td>HTS40 3K 2x2 Twill/TC275-1</td>
<td>Thin high-strength cloth, dual-curable toughened resin</td>
<td>42</td>
<td>346</td>
<td>0.010</td>
<td>Resin specifically designed for out-of-autoclave, thick sandwich structures</td>
<td>6 Rolls</td>
</tr>
</tbody>
</table>

Table 13. Overview of available prepregs considered for use.

We chose M46J and HTS40 as our two prepregs to use in the chassis largely due to inventory quantity available. M46J is the only 250-cure high-modulus fiber we had in quantity and HTS40 was selected due to our massive stock and its superior mechanical properties. Even though we will only be using two of these materials, we determined that we needed to test all three of these prepregs in order to better correlate prior models (prior chassis’s heavily utilized AS4) and for present vehicle models.
Fortunately, we were able to outsource a majority of M46J/TC250 testing to Mello’s composites class, who were characterizing that material as part of their lab course. The FSAE team was able to supply ample resources for the layup, strain gauge application, and testing manpower to characterize the specimens. Strain gauges were sourced from VPG Micrometreach in exchange for a case study on the material testing process. Production and testing of these specimens proved to be a valuable low skill process in which younger FSAE team members could learn the basics of composites early in the year. To gain a decent sample size, six specimens were prepared and tested for each fiber orientation. Ideally, higher quantity of tests would be conducted in the same environment to establish material basis values, although that posed to be too resource intensive to pursue (specifically due to the quantity of strain gauges required for the testing). Further team-specific procedures developed for the manufacturing and testing of the specimens are available on the team’s Google Drive.

There were difficulties when producing the HTS40 specimens primarily due to problems with correct sizing of the fiberglass grip tabs for 0 degree specimens. Additional problems were encountered with output range for the strain gages on the shear samples, where the high toughness of the TC275-1 resin allowed the specimens to strain far past the output range before failure. This could have been a function of the expired prepreg resin, although conclusive results were not produced. The out-of-plane shear modulus, $G_{23}$, was not tested, but datasheet and formula values were obtained for the laminate. The tested mechanical properties are shown in Table 14, and the percent difference from datasheet values are shown in Table 15.
Table 14. Elastic properties obtained from tensile testing. Cells with dashes indicate untrustworthy or nonexistent data.

<table>
<thead>
<tr>
<th></th>
<th>M46J/TC250</th>
<th>AS4C/TC250</th>
<th>HTS40/TC275</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1$ [msi]</td>
<td>28.77</td>
<td>8.406</td>
<td>6.7</td>
</tr>
<tr>
<td>$E_2$ [msi]</td>
<td>0.78</td>
<td>8.155</td>
<td>6.7</td>
</tr>
<tr>
<td>$G_{12}$ [msi]</td>
<td>0.47</td>
<td>0.54</td>
<td>-</td>
</tr>
<tr>
<td>$G_{23}$ [msi]</td>
<td>0.21</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$v_{12}$</td>
<td>0.22</td>
<td>0.003</td>
<td>-</td>
</tr>
<tr>
<td>$v_{23}$</td>
<td>0.87</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 15. Percent difference from tested to datasheet values

<table>
<thead>
<tr>
<th></th>
<th>M46J/TC250</th>
<th>AS4C/TC250</th>
<th>HTS40/TC275</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1$ [msi]</td>
<td>25.3</td>
<td>10.5</td>
<td>-</td>
</tr>
<tr>
<td>$E_2$ [msi]</td>
<td>69.5</td>
<td>13.2</td>
<td>-</td>
</tr>
<tr>
<td>$G_{12}$ [msi]</td>
<td>15.8</td>
<td>29.9</td>
<td>-</td>
</tr>
<tr>
<td>$v_{12}$</td>
<td>1.1</td>
<td>40</td>
<td>-</td>
</tr>
</tbody>
</table>

It is readily apparent that the tested values vary fairly significantly from the datasheet values. While a valid argument could be made that our test procedure and methods may have been flawed, especially with sensitive processes like adhering strain gauges, it still does not explain property variations in directly measured properties that did not require sensitive equipment calibration. Especially considering stiffness should be fairly robust and easy to replicate properties, and that the ASTM standards were followed in detail, we determined that the difference in property values must be attributed to a flaw or expiration in the material. Looking at what properties deviate the most from the datasheet values, the likely culprit seems to be a compromised resin system.

Our short beam shear testing was inconclusive, as we found we had cut all of the specimens too short. Every single one failed in bearing with the load-applying pin, rather than in transverse shear. This wasn’t ideal, but it did show us that using two different resin systems didn’t appear to have a huge effect on strength - at least not large enough to change the failure mode from bearing to shear.

As detailed in Manuel’s thesis [12], these results appeared to give prior chassis stiffness models an accuracy within 5% with test data. Unfortunately, due to difficulty in finding alternative material, we were forced to utilize the M46J prepreg in our chassis. In order to avoid this in the future, we recommend testing expired materials at periodic intervals and when first receiving the prepreg to better understand the limitations. Additionally, the material testing proved to be a low manpower task and could be utilized as a valuable tool for younger team members.
Core Selection

Selection of an appropriate core material for the sandwich structure is based on specific stiffness and strength, availability to the team, and manufacturability. Aluminum honeycomb core is a commonly available, inexpensive core with high specific strength and stiffness. Prior analysis and experience has shown that aluminum honeycomb core proves superior in performance characteristic, but presents forming difficulties compared to nomex honeycomb. Pre-forming sections of aluminum core can overcome this difficulty, but is time-consuming. Additionally, nomex is typically much more expensive than aluminum honeycomb. Another option was aluminum Flexcore, a special type of core with sombrero-shaped cells, which is extremely formable and retains its strength even in tight curvature. Traditionally, this core has not been available to the team, as it is expensive and we had no points of contact for Hexcel, the company that manufactures it.

We were lucky enough to get a contact at Hexcel this year, and he generously provided us with 15 slices of aluminum core in 0.5” and 0.7” thicknesses - 9 4’ x 8’ slices of 3.1 pcf 3/16” cell 5056 honeycomb, and 6 3’ x 8’ slices of 4.3 pcf 5052 Flexcore. The large donation of flexcore provided us with a reliable way of manufacturing tight curvatures, without spending hours pre-forming sections of honeycomb. Additionally, Plascore provided a donation of 6 sheets of aluminum honeycomb in 0.5”, 0.375”, and 0.25” thicknesses. Also, 8.0 pcf fiberglass core was generously donated through an unknown contributor, which was an ideal material to use for local reinforcements.

Staged-cure vs Single-cure Layup

It was considered to conduct a staged-cure layup process. The layup itself would be conducted in stages, contrary to previous years where the full chassis was laid up and cured in one step. The outer skin could be laid up and cured first, then the core, then the inner skin. The staged cure would allow direct application of the vacuum force onto the skin, whereas with a single-step cure, the core might deform and block the transfer of the force to the skin. The end result is better compaction of the skin, which leads to higher strength and a better surface finish. The main concern initially with curing in stages was the integrity of the bond between the core and the separately cured skins. If the skin-to-core bond is significantly weaker, takes longer to manufacture, or is less stiff staged than in a single-step cure, we would want to use a single-step cure.

To test the differing strengths of a staged-cure and single-cure laminate, two identical flat panels were laid up, using a [ (0)_3 / core], layup, with 0.7” thick aluminum core. The staged panel was a two-stage cure; the first skin was cured separately, then the core and second skin were cured onto the first skin. Both panels ran identical cure cycles and were prepared concurrently to minimize variations. The panels were tested using the SAE-specified 3-point-bend test, and the test data were compared. The staged-cure panel showed no signs of delamination, and both panels exhibited the same failure mode; core crushing followed by fiber buckling.
Figure 110. Staged-vs-single cure comparison by 3 point bending. The panels showed almost identical behavior.

As seen in the plot above, the single-cure panel was actually slightly stronger, but the strength difference is quite small and most likely within statistical variation. It displaced more as well, absorbing more energy. The staged-cure panel was slightly stiffer, most likely due to the fact that the cured skin was not pressed into the core, giving an ever-so-slightly greater panel thickness, and therefore moment of inertia.

The next step was to try and determine the quality of layup achieved with a staged cure. A test was conceptualized using the 2013 nosecone mold as a test piece; Its very high curvature would be good a good test of layup quality. The two layup methods would be compared in terms of dryness, bridging, and difficulty of layup. The single-cure nosecone section was laid up, but there was not time to produce a staged-cure cone. This test highlighted the main difficulty of staged curing; the time required to produce parts more than doubled.

In the end, we determined that we would stick with the single cure. The team has experience with this method, part strength is adequate, and most importantly, it allows a much more feasible manufacturing schedule. We did not feel that we could meet our schedule and produce two monocoques, with twice as many 10+ hour cure cycles. Additionally, a timely donation of several sheets of aluminum Flexcore from Hexcel put to rest fears of defects in the layup around tight corners. The flexcore will allow us to use a single cure with confidence, as it is specifically designed to be used in tight curvature.
SES Development

The most important factor in the final weight of the chassis is the Structural Equivalency Spreadsheet (SES), and the laminate testing that it requires. SES is required documentation that teams must submit to show that their chassis design meets the safety requirements of the FSAE rules. For a steel tube chassis, filling out the SES is as simple as confirming that the required tubes are the correct size and in the correct positions. For a monocoque chassis, the idea is that equivalency to a steel tube design must be proven. That is, each section of the monocoque design must be equal or greater than the equivalent steel tube section, in terms of EI (elastic modulus*moment of inertia of the section), ultimate failure strength in tension and shear, and energy absorption. To keep it simple for teams and for judges reviewing the SES, the monocoque calculations are based on a flat panel in 3-point bending, as well as a perimeter shear test. Each unique laminate used in the chassis must be physically tested.

3-Point Bend Test

The 3-point bend test is used to derive material properties of the panel skin; Ultimate tensile strength, Elastic Modulus, and energy absorption. A large cylindrical load applicator is driven into a simply supported panel until failure, and the force-displacement data is recorded.

![3-point bend test](image)

**Figure 111.** FSAE Rules schematic of 3-point bend test setup. Dimensions in mm.

The slope of the linear region of the force-displacement graph is used as the stiffness of the panel, and along with the thickness of both the skins and the panel is used to calculate an elastic modulus of the skin. The maximum load from the test is used to calculate the ultimate tensile strength. Energy absorption is obtained by integrating the tested force with respect to displacement - the area under the curve. The SES compares these calculated properties to the equivalent steel tubes, and the laminate passes if the properties are greater than those of the steel tubes. A bend test of two steel tubes is
conducted as well to establish a real baseline for laminate properties, and to eliminate the effect of the compliance of the test fixture on results.

**Perimeter Shear Test**

The perimeter shear test is used to prove the puncture resistance of the laminate. This requirement is particularly hard to meet for the Side Impact Structure and the Front Bulkhead regions. The data is also used for calculating the strength of the roll hoop attachments to the monocoque. A 1” diameter punch is forced through a sample of the laminate into a 1.25” hole. The force-displacement curve should have two distinct peaks - one for each skin shearing.

![Perimeter Shear Test](image)

**Figure 112.** Perimeter shear test example force-displacement curve from the SES.

The first peak is used to calculate the shear strength of the skin, but the highest overall peak may be used for the shear requirements of the Side Impact Structure and the Front Bulkhead Support. In general, the perimeter shear requirement is easier to meet than the 3-point bend requirements, and a panel that passes 3-point bend will probably pass perimeter shear as well, with some exceptions.

We initially conducted the baseline tube test for T3.30 with our new load-balancing top end of the 3 point bend fixture and unfortunately the wrong tubing. From the high load endured and massive amount of energy (58 J) absorbed, it was immediately clear that the tubing was incorrect. The tubing we had used was A513 tubing, not the 1010 low-carbon steel spec’d by the rules. A513 has a yield strength of ~70 ksi, whereas 1010 is a much lower 44 ksi… you get the idea. If we were to use this data the energy-absorption requirement for side-impact would be almost impossible to meet. We went back and tested with the correct 1010 low-carbon steel to get the correct baseline for SES. An image is shown below:
Now that we had compliance data, we plugged in panel test data from our single-vs-staged cure test (discussed in the material testing section) to see where we stood. This panel was a [(0.0)\_3 / core]„ with HTS40/TC275-1 fabric and 0.7” thick 3.1-3/16 5052 aluminum honeycomb. The failure mode of this panel was the typical core crushing followed by fiber buckling. We would’ve never used this layup on the car since it provides very little torsional stiffness, but the skin thickness was reasonable for the base layup of the ‘coque and it would give a pretty good idea of where we stood, particularly in our strength-stiffness balance.

The laminate was inserted into every laminate tab, including the side impact structure, and panel heights were assigned from rough cad measurements of the monocoque dimensions. In all sections, the panel needed to be stiffer. Strength factors of safety were 1.5-6 for all panels, and stiffnesses were 0.6-1.05, except for the Front Hoop Bracing, which passed with flying colors. Interestingly, the laminate met Side impact strength and gradient requirements in the laminate test tab, but was nowhere close to stiff enough in the SIS geometry-based tab.
It’s clear that we needed stiffer skins, especially if we wanted to use 0.5 core anywhere. So we laid up two panels of \([45_c / 0 / 0_c / \text{core}]_c\), one with 0.5 and one with 0.7 core. These use the same HTS40/TC275-1 cloth fabric, and M46J/TC250 uni to attempt to stiffen it up. The idea was to have the uni do most of the work bending, while inner and outer cloth plies would help “cuddle” it and stabilize it to prevent buckling. Because M46J uni is so much stiffer than the HTS40 cloth it takes most of the bending loads in the panel. So by placing the uni in between the plies of cloth the effect of external stress concentrations on the uni was reduced. The top ply reduced the effect of the load applicator, and the bottom ply interfaced with the core to get good core bonding. The different core thicknesses let us investigate whether SES will calculate more favorable material properties with a thinner core. This laminate is very thin (skins are 60% lighter than 2016), so we expected some problems with passing strength for most regions. Fortunately for us though, this laminate did pass SES. There are a couple reasons why we believe that we were able to get this laminate to pass SES. First off, extreme care was taken in the manufacturing of the panels. The plies were carefully cut, and then squeegeed together while making sure that the fiber orientation was perfect. After each skin was made it was debulked for about 36 hours, which we believe was the primary benefit to the quality of the laminate. Only once the skins had been debulked were they attached to the core and cured.
This base laminate of $[45_c/0/0_c/\text{core}]_s$ did work well to pass SES, and gave us a base laminate that was 20% lighter than the 2015 base laminate. The 0.5” core panel was the stronger of the two, although 0.7” core had to be used in the actual chassis laminate to meet stiffness requirements in most regions. We considered testing to remove the inner 0 cloth ply, but we were running out of time to
develop the side impact structure and the rest of the SES, and were worried that such a thin laminate would not pass the perimeter shear test.

**Figure 117.** Perimeter shear test on a sandwich panel.

**Side Impact Structure**

The requirements for the side impact structure are more difficult than other regions of the chassis, and the panel tested must be exactly what is used on the car. To create a passing side impact laminate, we started with the base laminate and added another ply of uni to stiffen and strengthen the laminate, and tested both 0.5” and 0.7” cores. Our initial test was not encouraging - the 0.5” core didn’t stand a chance, and the 0.7” core panel met strength and stiffness requirements, but not energy absorption. Adding two more plies would increase the weight of the chassis by over a pound, which we did not want to do. We were also not sure that the core we were using would support such a thick panel, we might just be increasing weight without much increase in strength or energy absorption.

Our next step was to try a stronger core with the same laminate. Rather than using our 3.1 lb/ft$^3$ honeycomb core, we made a panel with the 4.3 lb/ft$^3$ flexcore that we had on hand. The flexcore has a compressive strength of 542 psi compared to the 335 psi of the 3.1 honeycomb, which would help prevent the core crushing/fiber buckling that we had seen, and allow the panel to develop a greater load before failure. Luckily the panel barely met the requirements for energy absorption, with a safety factor of 1.07. Perimeter shear also barely scraped by, with a safety factor of 1.08.
Figure 118. 3-point bend data for the side impact structure laminate, with the calculated properties from the SES.

Figure 119. Perimeter shear data for the side impact laminate.

Front Bulkhead

The front bulkhead is somewhat of an outlier in SES, because it has a low stiffness requirement and a massive shear requirement. Rather than test a unique panel, which would require a lot of time and material, we used the based laminate and increased the thickness until it passed. We had hoped to get away without any core at all, since this would effectively double our shear strength (two skins come together to form one), but 0.25” core had to be included in the laminate to meet the stiffness requirement. In the end, the front bulkhead laminate was a massive [(45c / 0 / 0e)5 / core]s with 0.25” aluminum honeycomb, with a safety factor of 1.05 for shear and 1.16 for stiffness.
Harness Attachments

Another important region that we had to develop for SES was for the harness mounts. Since the competition is very heavily focused on the safety of all students competing, harnesses and their attachment points are very important to get right. Per the FSAE rules, the shoulder harness mounts and lap belt mounts must hold at least 3000 lb each before failure. The anti-submarine belts must hold at least 1500 lb each before failure. If the lap and anti-sub belts are combined into a single attachment to the monocoque, they must sustain at least 4500 lb each before failure. The keep the cockpit from getting too cluttered and uncomfortable, we aimed to combine the lap and anti-sub belt mounts in this way.

For the lap and anti-sub belt mounts, our test laminate was \([(45c / 0 / 0 / 0c)2 / \text{core}]s\) - a doubled-up side impact laminate. The core used was a very dense and strong 8.0 lb/ft\(^3\) fiberglass honeycomb core. We went to the denser fiberglass core because for highload impact structures (like how the harness mounts are treated), SES requires the laminate to be thick enough that our standard 3.1 aluminum honeycomb core becomes the weak link in the panel. This will almost always cause the panel to fail through core crushing at the load applicator which then causes the fibers to buckle. A denser, stronger core like the fiberglass core can sustain more load before crushing and therefore supports the skins to a higher load at a smaller weight increase than compared to adding enough plies to meet the requirement.

The mounts themselves were large 7/16” eyebolts designed for the purpose and purchased with our harnesses. Additional 1/4” bolts were used with a .080” steel plate on each side to distribute the load over a larger area of the laminate.

With this setup for the harness mounts though we came within 290 lb of the failure load that would have allowed us to combine the lap and anti-sub belt mounts. Unfortunately, we were running out of time as usual, and couldn’t prepare another test sample. We went with separate mounts. Future teams should definitely attempt to pass that combined mount load since it will provide a decent weight savings in the mounts and clean up the cockpit floor a bit.
Figure 120. Our test fixture for the lab and anti-sub belt mounts. To be representative of the belts in the car, our rigid fixture clamps on the edge of a flat carbon sandwich panel.

Figure 121. Load test of the lap and anti-sub belt mounts.
The shoulder harness mounts were a little more troublesome. Not only did we have to test an attachment, we had to fill out the relevant Shoulder Harness Bar tab in the SES to prove that the small strip of laminate that the mounts were attached to could take the load. The result was a harness bar laminate of \((45c / 0 / 0 / 0c)2 / \text{core}\) - a tripled-up side impact laminate. The mounts themselves were simple aluminum brackets bolted through the laminate with two \(\frac{1}{4}\)" bolts each.

Figure 122. Failure mode of lap and anti-sub belt mounts.

Figure 123. Shoulder harness bracket with mounting hardware.
There were two big problems with our original design of these harness attachments. First, we had tested the brackets with the load in plane with the shoulder harness bar. This seemed sensible as this would be the direction that the mounts would be loaded in a head-on collision, but we failed to notice a note in the rules that the harness mounts had to be tested *normal* to the panel.

**Figure 124.** First Shoulder harness mount test, after failure. The test shown here is representative of the original shoulder harness mount location on the chassis with respect to the distance of the mount from the edge of the panel. This test was not valid as we needed to test in a direction normal to the panel.
Figure 125. To test the shoulder harness mounts, we used old harness belts clamped in the instron jaws to pull on the mounts.

This setup passed the required load, and was included in our first submission of the SES. When it was rejected, we had to redesign the test.

The second problem was how close the harness mount bracket was to the edge of the panel. Originally the bracket was within an inch of the panel edge. We had not seen anything in the rules suggesting this was problematic, and this setup had passed the harness test. When our first SES was rejected, the ruling was that the bracket needed to be placed in the middle of the ~4” harness bar, which required moving it back about 1.5”. This made us have to move the headrest back as well.

With our new requirements, we made a new test fixture and test panel for redoing the test. The shoulder harness bar was included at the correct width, plus a part of the seatback flange bonded underneath to simulate the rigidity that it would add on the car.
This setup passed the approval of the judges, but we recommend a redesign or at least a fresh look on how to do this well and perhaps more elegantly (it would be cool to forgo metal brackets entirely and just cut holes in the harness bar area to wrap the harnesses around).

**Roll Hoop Attachments**

The SES required that each roll hoop mount must withstand 30 kN (about 6750 lb), although does not require this to be tested. The main failure modes considered for this are perimeter shear (pullout) and in-plane shear/tearout. The key players in this calculation are the skin shear strength established from the perimeter shear test, the skin thickness at the attachment, and the edge distance of each attachment for tear-out. The base laminate and side impact laminates were too thin this year to support a small, light attachment, so unlike previous years where a large attachment was used, we locally reinforced the laminate around the attachments, doubling or tripling the skin thickness where necessary. The judges accepted this without issues, however in future years it would be a good idea to submit a rules clarification about it beforehand. The thick skin allowed us to minimize the amount of heavy .080” steel used for the brackets, as well as make more aesthetically pleasing brackets than previous years.
Figure 127. Pad-up for the right upper attachment of the main roll hoop during the layup. These local reinforcements were implemented in a similar way to those used for suspension points.

SES Quirks and Limitations

Since the SES uses a simplified procedure, there are some assumptions and calculations which don’t necessarily reflect reality. The result is some annoyances as well as some “loopholes” that may or may not be helpful.

The 3-point bend test is the biggest offender. The test supposedly measures the tensile modulus and strength of the skins, with no difference between tension and compression. However, composites are well-known to be significantly weaker in compression than in tension. Additionally, the load applicator puts a large stress concentration on the center of the panel so that at the point where the bending moment is largest, there is also a large compressive stress on the core. The result is that the panel will always fail in compression in some way, usually brought on by local crushing of the core which will make the fibers buckle. For thick-skinned panels like the Side Impact Structure or the Front Bulkhead, the core may undergo transverse shear failure well before the skin would have failed. The result is that it's hard to make a panel that will fail in the “right way” that the SES calculations are based on. The take-away from this is that the core is a big player in how a panel will fail in 3-point bending and the SES does not take this into account.
Figure 128. Comparison of two Side Impact Structure Panels with identical layups of [45c / 0 / 0 / 0c / Core]s and core with identical thickness and different compressive strengths.

Because the core is so important in the test and because the SES doesn’t take any core-induced effects into account, the solution is the minimize the influence of the core as much as possible. Luckily, for every laminate region except the Side Impact Structure, teams must test only the unique skin laminate for each section. The choice of core thickness seems to be irrelevant. So the best thing to do is to use thinner core in the test - the shear loads that the core must take will be less, the panel will behave more like it’s in pure bending, like the SES is based on.

Additionally, a tested laminate may be multiplied as many times as necessary to pass. For example, a [0 / 90 / core]s laminate could be tested to obtain the properties, and the actual laminate could be [0 / 90 / 0 / 90 / core]s if necessary, and so on. What this means is that a thinner and therefore weaker skin can be tested, making it more likely that the skin will be the weak point rather than the core.

Therefore the best type of panel to test is a panel with thin core and thin skins. The SES calculations will determine the tensile strength, and the core thickness and/or skin thickness are increased to the necessary size for each regulated section.

This approach is not allowed for the Side Impact Structure - the panel tested must exactly represent the laminate used on the car, which makes it more difficult to pass. The above figure shows that the preferred solution was to use stronger, slightly denser core that could better support the skins.

The perimeter shear test can be done similarly, where the core and skin thickness used don’t necessarily have to be the actual values used on the car. The effect was not as drastic as with the 3-point bend test, but we seemed to get better shear strengths using thin skins.
Manufacturing

Plugs

Once the Zodiac engineer completed his tweaks to the model we sent, they began machining the plugs on their 5-axis routers. The plugs were constructed from 4” thick sheets of 15 lb/ft^3 density foam glued together with a foam-bonding adhesive. The bottom plug was machined fine, but the top plug was machined slightly incorrectly and had too much material taken off of the front section of one side. An error in the CNC code or potentially carelessness during the machining operating could have caused it. Regardless of the cause though, this resulted in a visible step of up to 100 thousandths of an inch at the mating line of the two plugs (and therefore the same step existed with the molds and parts as well). On top of this, the plugs were delayed by Zodiac by 5 weeks which prevented us from being able to start the mold process during build week as we had planned, so we lost a week of full work days without needless distractions like classwork. Instead we had to wait until January 11th to start the process, one month later than we would have liked. Not ideal.

The first round of work was conducted at Zodiac’s facility in Santa Maria, so that their technicians could teach us and guide us on the best way to prepare the plugs. This was incredibly helpful and we learned a lot about how to sand and surface without distorting the machined shape of the plugs. The major downside was that we had to drive 40 minutes to Santa Maria to work, and we had to work during the week which made it hard to organize around classes to get a good group of people to stick around for hours.

![Figure 129. The chassis plugs before sanding, covered in blue guide coat.](image)

When we arrived, the plugs were set up in Zodiac’s sanding booth, sprayed with a white primer and then a thin layer of blue spray paint called a guide coat. The first step was to gently sand and smooth the plugs, as they were still relatively rough from machining. The purpose of the guide coat is to let you know when you’ve sanded too much - If the blue is gone, stop sanding that area. If it’s sanded
perfectly, there should be a light speckle of blue, denoting the low spots/pits in the foam. The flat areas were sanded first, as they’re the easiest. Large rubber sanding blocks were used to make sure the surfaces were sanded evenly and stayed flat.

Figure 130. Sanding the flat surfaces with rubber sanding block. This surface is nearly complete. The edge of the flat surface (full white area) has been oversanded slightly, since it’s easy to put too much pressure on an edge by accident.

Sanding is a methodical process, and it’s easy to rush it, but this will create mistakes which may need to be fixed later. Slow and steady wins the race.

Figure 131. Flats almost done on the lower plug.
Once the flats were complete, it was time to move on to the corners. The proper way to sand a corner is to use a curved sanding block, and apply only slight pressure. Move the sandpaper diagonally across the curve so that pressure is not concentrated in any one area.

Figure 132. Curves done on the top half. The darker grey areas are where the plug has been sanded all the way through its primer and down to the foam, which is bad. The top half was separated from its base, which made this much harder to avoid.

The next step was to fill in the low spots left over from the first round of sanding with a filler, similar to Bondo. The material used at Zodiac was Axson Technologies APF 4 White, which is a quick-setting filler with low shrinkage - Thin layers were dry enough to sand in 10-15 minutes. Thin, almost invisible coats of filler were applied to the plugs in small sections. The thinner the better, as any excess would have to be sanded off anyway. When dry, the filler was sanded just enough to smooth it. First, coarse 120-180 grit sandpaper would be used sparingly to remove large spots of excess/burrs/ridges from lifting the paint scrapers used to spread it. Then we moved up to 320-400 grit to smooth it out.
Figure 133. Covering the bottom half with filler. Because we’re inexperienced, most of this has too much filler applied. The white filler should be barely visible. The earplugs are plugging the locating holes so they don’t get clogged with filler.

This process was repeated a couple times until the entirety of the plugs was smooth and glossy with no defects. A lot of time was spent touching up small defects, especially around the parting lines of the plugs. The bottom plug had a base which made it hard to sand near the parting line, and the top plug did not, which made it very easy to oversand the edge. If we had to pick one, leaving the base on the plug was the better option.
Once we were satisfied with the finish, we lined up the halves to check for any major issues. We could clearly see the step caused by the mismachined top half, but we decided to send it, since we didn’t have the time to attempt to fix it. Zodiac’s delay with machining had eaten up all extra margin in our schedule.

While we were prepping the chassis plugs, we also worked on the nosecone plug with the same method. The sanding was more difficult since most of the nosecone is curved, but there was less pressure to do it perfectly, since the nosecone didn’t have any important surfaces that needed to be accurate.
Figure 135. The nosecone before and during surfacing. The small area relative to the chassis meant that it went quickly.
Figure 136. The step between the plugs. The other side is identical. The step is most severe towards the front, and becomes barely noticeable at the rear of the chassis.

We declared the surfacing done, and Zodiac techs sprayed the plugs with a green sealer, REN RP 802, which dried overnight. The following evening, we came back to Zodiac to pick up the finished plugs.
Figure 137. The plugs arrive from Zodiac in the back of Kevin’s swaggin waggon!

Figure 138. The nose cone plug in the paint booth after arriving from Zodiac.
When we got the plugs back from Zodiac, the last bit of work that they needed to have done on them before they were ready to have the layup for the molds on them was to mount them to a base. We hadn’t pre-specified any material or requirements for the base for the plugs to sit on, so we ended up using a 1 inch thick piece of foam board glued to the base of each plug. This was a bad, hurried solution for the base of the plugs since it was intended to help create the mold flange, but ended up being too weak structurally and broke apart when we tried to remove the molds. As is detailed in the mold section, the plugs for the molds no longer exist because they ended up getting stuck to the molds and had to be cut out of the molds in chunks. This is one of the many little things that was not properly thought through and had major consequences. Attention to detail would’ve gone a long way here. A proper solution would be to use a much thicker piece of dense foam.
Figure 140. The bottom half plug just prior to being glued to its base board. Sometimes glue will make interesting patterns as it’s being applied.

Molds

The next step was making the molds using the newly finished plugs. To do this, we used the following procedure. First off, we went about prepping the surfaces of the plugs to have the tooling carbon to be laid up on them. We thoroughly cleaned the plug with water and a small amount of acetone (the acetone started to dissolve the sealer, something less volatile should’ve been used), to make sure that we would have a nice surface to apply mold release. First, a Frekote sealer (organic solvent-based) was applied to seal any porosity that may have been left. 2 coats were applied. Then we used Fibre Glast Fibrelease (water-based) as a mold release, applying 5 or so coats. This was a huge mistake, as we learned when the molds refused to release from the plugs, as discussed below. Our leading theory is that the sealer and mold release were not compatible, and the organic solvent-based sealer repelled the water-based mold release, preventing it from forming the coating it should have.

Once we had a nice even application of the mold release, we inserted teflon locating pins into the plug and placed steel drill bushings on them. The teflon locating pins were simply pieces of teflon rod purchased in the correct diameter from McMaster. The bushings were cut and drilled from small steel tube stock, since real bond-in drill bushings were out of our budget and attempts to get a partnership/donations fell through. After all the pins and bushings were in place on both plugs we then applied the surface coat, the final prep step prior to laying up the tooling carbon on the plugs. Surface coat is a thick, durable epoxy coating has a high hardness and can be polished to a very smooth surface that will form the actual surface of the mold. For the surface coat we used Premium Resin Tech PCR-1902 High-temp surface coat, which is stable up to 350 degrees with no post-cure needed. This is a very thick black two part epoxy coating that was hard to mix, so we mixed it with the “double-cup” procedure. To “double-cup” place the correct proportions of the surface coat and hardener into the first cup and mix with a wide popsicle stick thoroughly. Then pour the mixed surface coat into a new cup and continue mixing. This helps prevent any unmixed epoxy stuck to the
walls of the cup from being applied to the plug. When the surface coat had been fully mixed, we then brushed it onto the plug in a thin, even layer. We let the surface coat sit on the plug for about an hour to allow it to gel slightly before laying any of the tooling carbon onto the plug. If we had laid the carbon down immediately after applying surface coat, it’s likely that the carbon would have been forced through the surface coat during the first debulk.

With the plugs completely prepared, we then moved on to preparing the carbon to be laid up on the plugs. Because of the somewhat complex geometry of the plugs and molds we decided to cut the tooling carbon into 1’x1’ squares at the recommendation of PTM&W. This made forming the plies of tooling carbon to the plugs and just handling the carbon easier. On some of the squares the tows started getting pulled out of the weave, meaning that the edges of the squares were essentially fraying. We found however that cutting a 45 degree angle off the corners of the squares helped to mitigate this issue. Large pieces of plastic sheeting from Home Depot were placed on the frame tables (both Baja and Formula’s) in the Hangar such that 12 of the carbon squares could be placed on each sheet of plastic and then have the plastic folded back on them. For each set of plastic sheeting and carbon squares, the carbon squares were placed close together in a grid pattern on half of the sheet of plastic with the other half of the plastic hanging off the table. Next the PTM&W PT2520 epoxy and B1 hardener were weighed out to the correct proportions (100 parts resin to 19 parts hardener by weight) and mixed using the “double-cup” procedure as mentioned above to prevent any unmixed resin/hardener on the walls of the first cup from getting into the layup. The mixed resin was then poured out across the carbon squares at a ratio of between 1.25-1.5 parts resin to 1 part carbon by weight (77-92.5g of resin per 1 ft square of carbon). The second half of the sheet of plastic was then folded over to sandwich the carbon squares and resin between the two halves of plastic. We then used rolling pins and squeegees to press the resin into the fabric.
For best results we found that pouring the resin so that it initially covered most of the square and then using a squeegee to press it into the rest of the square was the quickest and easiest way to wet out the carbon. The process also went faster when using the higher ratio of resin to carbon and using a squeegee to push any excess resin out of the carbon square when wetting out the carbon. We also found during the wetting process that it was very important to check both sides of the carbon square to make sure that the entire square is covered in resin because the tooling carbon is so thick that even if the top side of the square is completely wetted out with resin the bottom side won’t necessarily be completely wetted out.

After all the carbon squares in a sheet were wetted out the entire sheet was moved over to the plugs and another sheet and squares were set up and wetted out on the frame table. At the plugs, the sheet of plastic was opened up, the squares were removed and placed on the plug smoothly, making sure that there are no air pockets and a minimum of wrinkles when pressed firmly to the plug. Around the drill bushings, the teflon pins worked fine to poke holes in the weave of any squares that were being laid over them, and then the square could be worked down to fit around the drill bushing quite nicely.
The layup schedule we used initially involved a 45 surfacing ply of thinner cloth, because it was believed that this would result in a better surface. Unfortunately, we ran out of thin cloth on the top half mold, so the bottom half didn’t have any. No difference in surface quality was observed between the two because of it. Then [(0<sub>c</sub> / 45<sub>c</sub>)/<sub>2</sub> / core], of the thicker bulk cloth. Using the squares, each ply was laid up as a patchwork with minimal overlap of squares within the same ply. Along the base of the plug, where it was attached to the base board, we made sure to layup at least 3 inches of carbon to give us a flange on the mold. For the core in the layup we used ¾” Nomex core along the flat sections of the mold. We tried to use it everywhere on the mold, however we had difficulty with the core forming around corners and holding in place. In order to help all of the plies compact to each other well and to suck out excess resin (too much resin could change the CTE of the mold for the worse) we used a 20 minute debulk on the mold after every 2 plies. For the debulks we used perforated teflon release cloth, peel ply, breather, and a large vac bag that was adhered to the plug flange with tacky tape to create a seal around the mold.

**Figure 143.** The bottom plug with all 4 layers of carbon and the core. The tiny white dots on the mold are the non-stick teflon pins extending through the drill bushings.
After completing the layup of the molds, which took about 16 hours for each half with a full team of 4-10 people, they were given two days to room-temperature cure before we attempted to pull them off the plugs. The thin foam board base that the plugs were mounted on in combination with the sealer and mold release we used caused the molds to get stuck on the plugs when we tried to pull them off. We believe that the reason why the sealer and mold release caused the molds to get stuck on the plugs is that the sealer used was incompatible with the mold release used. The sealer was organic solvent based (like Frekote) and the mold release used was Fibrelease which is water based. The organic (oily) coating just caused the Fibrelease to bead up and not wet the surface which prevented the entire plug from being covered in mold release, and as a result the molds got stuck on the plugs. Because of this we had to spend about half a week chipping out the plugs from the molds before we could even start to reprep the molds for making parts.
When the last pieces of the plugs were finally removed from the molds, the molds were given a cleaning and then brought to the composites lab for post-cure. They were post-cured for about 14 hours using PTM&W’s recommended post-cure cycle. It took about 3 weeks to repair and resurface the molds to a usable state.

**Mold Repair**

Once the molds were cleaned and residue from the stuck plugs had been removed, the main problem was that there were many pinholes or rough spots in the surface coat, which needed to be filled. Additionally, there were some rough areas around some of the drill bushings, and the parting line. We used a filler and method similar to how the plugs were done at zodiac, applying thin layers of filler, letting them dry, and then sanding smooth.
Figure 146. High-temp filler applied to defects/pinholes in the mold, before being sanded.

It was a simple process, but took a long time as there was a lot of area to cover and new pinholes could appear as the mold was sanded more and more. Generally, we would apply a round of filler, and sand it to about 400 grit. We’d look for remaining defects, and rinse and repeat. When the molds were finally repaired, they were sanded to 1500 grit in preparation for layups.

Although it took longer than we would have liked to go from getting plugs to having useable molds, the molds did turn out well and we gain a lot of valuable knowledge and lessons from the process. Looking at the entire mold manufacturing process in retrospect, there were many good and bad things that happened from making these carbon tooling molds, and these things are included in the bulleted lists below.

The bad.

- Easily the biggest issue: the top half mold was not properly debulked, as suffered massive porosity issues because of that. There were pin holes all over the surface and every time a part was pulled, a few more would appear.
- Placing the core really needed more planning than cutting and placing it as we went and the molds may have came out better if we had done so.
- Drill bushings were not thought out as much as they should have been. Some were so close to the flange/parting line that it was impossible to drill through them. Others were at strange angles, because the nonstick Teflon pins used to locate the bushings to the holes in the plugs were soft and deformed under vacuum bag pressure.
- Not enough thought was put into the transition at the parting line between the mold surface and the flange. Initially it had been envisioned as a sharp corner that would have enabled us to bolt or position the two mold halves together to possibly make a
one-piece monocoque, but making that sharp corner need planning and probably some testing. As it turned out, the surface coat made the sharp corner, but the carbon did not and formed a radius. The result was a very weak layer of surface coat, a big bubble, and then carbon. The surface coat broke away almost immediately, leaving a large radius where there shouldn’t have been one. This caused delays and extra weight added in the strap joint further down the road. In retrospect, the one-piece idea should have been abandoned and excess space should have been made on the surface to make trimming the halves easier.

The good!

- Making the molds out of carbon to match the coefficient of thermal expansion (CTE) of the part was fantastic. Even with the less than perfect molds it took less than an hour to pull both halves, versus days with the old molds.
- Low thermal mass. The part temperature was verified to be within a few degrees of the air temperature in the oven even at a fast ramp rate. This was a huge improvement from the previous plaster molds, in which the large thermal mass meant that part temperatures could lag air temperature by almost 100 degrees, which made it difficult to control the cure cycle.
- The drill bushings did what we wanted them to. At least using a tape measure (and calipers where possible), the holes in the parts were dead-on. Down the road they were immensely helpful in getting suspension, engine, and drivetrain holes drilled accurately. Most of the jigs could be simple 2-dimensional parts.
- The molds were much lighter and infinitely more practical. Each half could be easily moved by two people, and easily strapped to a car for easy transport to the lab. This was unexpectedly awesome as we were able to do more with the molds, such as post-cure, practice layup, e-car top cover plate, and of course both chassis layups. This wouldn’t have been possible if we had to use 6 people and a truck to transport molds like in previous years.
- When it came time to send the part geometry to Zodiac, suspension was not ready with all of their points to be turned into mold drill bushings. We got enough to make most things work, but the ones that were left out (rockers and rear shocks) were some of the most difficult to locate. This should have gone one of two ways: Either we should have had every suspension point on the molds, or we should have just planned to have holes exclusively used to locate jigs. What we ended up with was somewhere in between.

Rocker Boss

Manufacturing for the rocker boss was done in a manner similar to the chassis mold and plug. First, a plug was machined out of high density foam utilizing the swiftness and precision of the glorious ShopBot. Alignment holes were machined into the part to allow pins to be laminated into the boss. This will provide locating features to align the part within the chassis mold. Unfortunately, three of the holes in the plug were misaligned due to machinist error. The rocker boss plug was prepped in a similar manner to the chassis plug.
Figure 147. Machined and prepped plug. Alignment holes in the part were intended to align with chassis mold holes, with alignment pins cured inside the boss.

A carbon mold was then pulled from this plug utilizing the same method as the chassis mold. Alignment pins were embedded into the laminate, although all but one had to be removed when test fitted into the chassis mold.

Figure 148. Bottom view of rocker boss. Note: only one alignment pin achieved its target location.
The rocker boss was then integrated into the chassis mold through alignment pin locating. As the plug was designed in a way to allow the part to sit absolutely flush with the mold surface, the one alignment pin was more than enough to locate the part.

Test Layup

Due to all the problems that we had making the molds, we wanted to test them before we trusted them with a chassis’ worth of time and material. We also wanted to validate the rocker boss plug would behave nicely. We decided to do a test layup with a single ply of fiberglass prepreg and some core test pieces thrown in as well. This way, the fragile fiberglass ply could be ripped out easily if it stuck in the molds. The test also gave us an opportunity to see how the molds would behave in the oven, specifically if the thermal mass was low enough that the part temperature would closely follow the specified cure cycle.
Thermocouples were inserted between the fiberglass ply and the surface of the mold, so that we could see how the temperatures were doing in real time. The part temperature was checked every few minutes during the ramp, and the temperatures were in step with the cure. This meant that we could avoid having to constantly monitor the part temperature during the cure and adjust the cure cycle.

The test came out of the oven looking good, and was easy to remove from the molds. Unfortunately, some new pinholes had formed in the mold, so we had to perform some minor repairs before we could lay up a chassis.
Layup

Figure 151. Layup by region - c indicates woven fabric, otherwise unidirectional fiber is used.
- **BLUE** – [45c / 0 / 0c / 0c / core]s
- **RED** – [(45c / 0 / 0c)2 / core]s
- **GREEN** – [45c / 0 / 0c / core]s
- **MAGENTA** – [(45c / 0 / 0c)s / core]s

Figure 152. Core thicknesses and types, by region. Local core reinforcements not shown.
- **BLUE** - 0.5”hex
- **RED** - 0.5” HRP
- **GREEN** - 0.7” hex
- Grey/White - 0.375 hex
- **YELLOW** - 0.7” flex
- **MAGENTA** - 0.25” hex
The first step in the layup process was to thoroughly clean the molds and then apply mold release. Cleaning was accomplished by wiping over and over again with acetone-soaked rags until no residue was visible on the wipes. Next, small pieces of flash tape were placed over each bushing hole to ensure no resin could leak into them. The mold release used was Fibre Glast Fibrelease; 8 coats were applied in approximately 20-minute intervals.

![Figure 153. Chassis team members laying up a 0 cloth ply for the outside skin on the top half mold of the e-car, making sure that the section was placed correctly and pressed tight against the previous ply to prevent air bubbles between the plies.](image)

The entire layup process for both chassis was done in the composites lab, where there was plenty of clean space to cut the carbon, lay it up, debulk, and then use the oven to cure them. To cut the many different pieces of carbon for the tub, we used templates created in solidworks. To make the templates, we took the section in the solidworks model that we wanted a template for, used the surface flatten command to make it two dimensional and then added an inch around the perimeter for overlap on the actual layup. With these templates printed out we were then able to mass produce the carbon shapes needed for the layup by tracing them onto carbon rolls, making sure that the fibers are oriented correctly, and then cutting them out with X-Acto knives.

The first ply was the most difficult to lay because the mold surface with release agent applied is extremely slippery, so plies wouldn’t stay in place easily when laid down. The best way to get around this was to use several people to hold pieces in place until two or more were laid. Then the edges of each piece were pressed together so that they’d stick to each other and help hold each other in place. We also made sure that the edges of each piece were very cleanly cut, as any shoddy work here would be easily visible on the outside of the chassis, especially if we decided to incorporate areas of bare carbon into our paint scheme.
At the same time that the carbon shapes were being cut out, we started laying them up in the molds, following our layup schedule. Using these template based carbon shapes was a great benefit because it allowed for the carbon to be worked into the shape of the mold with minimal wrinkles and overlap. And wrinkles or bubbles that did appear were taken care of with an X-Acto knife. By slicing parallel to the carbon fibers in the uni, we were able to get the air out and flatten any bubbled areas (usually occurring in tight corners) without breaking the continuity of any fibers. Working from both the front and back of the tub moving towards the middle of the tub worked best as it allowed for the most number of team members to work on a mold at a time. Pad ups were also placed at the appropriate intervals throughout the layup, and were held in place well enough by the tackiness of the resin as other plies were placed on top of them. We also made sure that in placing the plies that we allowed enough of the pre allocated extra carbon in the templates to fold up onto the flange of the mold to give us a secondary flange to help separate the part from the mold.
Figure 155. The bottom half of the c-car tub after a 0 ply of cloth has been applied throughout the entire half. Notice how the large flat portions of the mold lend to nice, smooth, and well packed down carbon while some fillets look like they could potentially have air bubbles between the plies.

Figure 156. Cut sections of core to match the patterns for relatively flat sections on the chassis for the c-car.
Figure 157. Core placement in the top half of the mold for the c-car. The white lines between sections of core is the foaming core splice that is extremely sticky and helps to fill gaps while bonding adjacent pieces of core together.

Figure 158. Core being formed onto the top half of the mold for the c-car. Trimmings of the core can be seen around the mold as team members modified the pre-cut core sections to fit exactly in the layup.

The most difficult and time consuming aspect of the layup process (other than debulks and curing the carbon) was placing the core in the layup. Just prior to placing the core into the molds, we added a ply of film adhesive to help the core stick to the laminate and also to help prevent the resin in the prepreg
from all flowing into the pockets of the core. On the first chassis, the c-car, we didn’t have a great handle on how to form the core or how best to cut it so that the core application process went smoothly. As a result (and as you can see from the photos) we ended up using many small pieces of core that didn’t fit up to each other very well and required many more even small pieces of core to span the gaps between the larger pieces. It became very time consuming to have to try and expand and compress the edges of these many pieces of core to get them to fit together well, and the smaller pieces were more likely to get stuck to the residual core splice on gloves and tools which made getting the pieces to sit firmly against the laminate extremely difficult and frustrating. The root problem was that we had cut the core based on the templates used to cut the carbon, which all had edges in corners of the chassis. This is great for laying carbon since it reduces issues with bridging, but bad for core as it means that the core splices are in curved sections of the chassis. A good core splice needs the core pieces nicely butted up against each other, and this was almost impossible to do on a curved surface.

We learned from this on the second chassis, and abandoned using the carbon templates to cut core pieces. We were able to start off with larger pieces of core and form them around curves in the chassis in a freeform manner. This helped us to use less foaming core splice, which is heavy and messy. The core sat against the mold better because the larger pieces were easier to deal with, and we managed to avoid the splicing woes from the c-car. We also started placing strips of core splice on any difficult curvature of the mold to help stick the core down in those locations.

![Figure 159. Core being formed into both the top and bottom halves of the e-car chassis. As you can see in the bottom left of the photo the red section of core is the denser fiberglass core used under the shoulder harness mounts. Also if you compare this to the photos of the c-car core forming it is clearly a lot fewer distinct pieces of core due to using the chassis as a template to pre cut and layout the core with ahead of time.](image-url)
**Figure 160.** Core being formed on the bottom half of the e-car tub. You can see the dense, red fiberglass core in two rectangles where the lap and anti-sub belts will be bolted to. The blue film that is clearly seen on the side of and bottom of the tub is the film adhesive that we used to ensure a good bond between the core and the skins.

**Figure 161.** Core forming in the bottom half of the e-car chassis. You can see the section of fiberglass core down by the bottom of the picture that will support the brake pedal.
Core Forming Lessons Learned

Laying down core will always be a difficult process, but we think that improving the methods the team uses to do it is the best way to improve the quality of the teams monocoques in the future. In the first monocoque, we had huge issues based on poorly thought through core templates, too many changes in core as a result of an overuse of flexcore, and difficulty in clean core splices. Fortunately, we had two chassis to try different techniques.

Using the hard lessons we learned in the first chassis core placing, we changed the templates considerably for the second. Instead of relying on computer templates and models, we created templates directly from the mold and the first (combustion) chassis. This allowed us to quickly iterate problem areas while allowing us to get a visual of where problem joints may occur. By laying out each core division in tape, we were able to simplify and plan better for troublesome contours. A great example of this is in the rear of the chassis on the top mold. Nearly a dozen individual core pieces on the combustion chassis were replaced with just two on the electric. Similar issues were resolved in the top half main roll hoop support, top half upper side impact, and bottom half rear portion. Additionally, too many joints often lead to separations in the core pieces during the cure, which greatly reduces part quality and creates large gaps on the inside. Flexcore was used on every fillet in the combustion chassis out of fear of gapping and bridging that normal core can bring in tight radius. It is noted that eliminating nearly all flexcore at fillets in the electric chassis did not bring any part quality defects, nor was it any more difficult to form. In fact, it seems to have reduced defects considerably compared to the combustion monocoque. While flexcore is hugely beneficial, it is not necessarily worth the added joints and core splices to be used in every single radius.
Figure 162. The method used for core forming and planning for the electric chassis. After difficulties forming core on the combustion monocoques core templates, core templates and partition locations were planned and simplified prior to the layup on the second chassis. This significantly reduced layup time and increase part quality.

Debulks

To help make sure that the plies were properly compacted and because our testing had shown that regular debulking of a part helped increase the overall quality, we debulked the chassis after every 2 plies laid. On the debulks, to save time we created large, reusable vacuum bags that could be slipped over the entire mold and have enough extra to form to the cavity of the mold and press the part into the mold. We left one end of the bag open so that we could use tacky tape to shut the bag after enclosing the mold, and enough extra material in the length of the bag that the tacky tape on the “resealable” end could simply be cut off to remove the mold from the bag and there would still be enough material to use tacky tape to shut it again for the next debulk. We were able to get away with using just a single large bag per chassis for all debulk and cure steps. We were able to reuse breather and FEP as well. To prevent the core at the edges of the mold from being crushed by the bag under vacuum, wooden blocks were used as edge dams placed along the flange between the breather and vacuum bag.
Figure 163. FEP placed on the bottom half of the c-car tub to prep for a debulk after the outer skin has been laid up on the mold.

Figure 164. Nathan leads chassis team members in placing padups on the bottom half of the c-car tub in between the 0 uni and the 0 cloth plies of the base laminate.
Figure 165. Chassis team members laying up a 0 ply of uni for the top of the c-car. As can be seen, multiple team members can be working on each half of the tub at the same time. So for about the first 15 people working on the layup productivity is directly proportional to the number of people in the composites lab.

Because of the low technical knowledge required to actually layup the carbon, we were able to get many members of the team to help with the layup under Nathan and Kevin’s direction. Kevin and Nathan were able to direct the team members with which templates go where, where the pad-ups go, when to debulk, and which direction each layer should be oriented in. With a brief overview of technique to the team members helping, a good chunk of the manual labor that goes into constructing a lay up could be handled by many hands, making the work faster and easier than if left to the handful of technically knowledgeable members. The space in the composites lab along with the size of the chassis molds made it relatively easy for up to 7 team members to work on each chassis half at a time. This added in knowledge transfer, helped team members feel involved, and helped move the process along quickly. Team member involvement, especially in this aspect of the chassis manufacturing was invaluable.

It is to be noted for future years, that from beginning final mold prep of the first chassis to removing the second finished chassis from the mold took 10 days in the middle of a quarter. This comes out to 5 days per chassis. Granted, the postwork and joining took 2 weeks of comparatively relaxed work per chassis after demolding. This is in comparison to taking several weeks of vacation time to layup and demold a single chassis in prior years.

Cure

The chassis’ were cured in the Cal Poly ME Composites lab oven under vacuum pressure. This large oven can easily fit both chassis halves at once. Since the oven has 4 vacuum hoses, 2 hoses were used on each half, on in the front and one in the back. We used Tencate’s recommended 275°F cure cycle for the TC275-1 epoxy used in most of the carbon [11]. The uni used TC250 epoxy, which has a shorter and cooler cure cycle, but our previous material testing and SES testing had showed that this was not an issue in terms of strength.

As our practice layup had shown that the part and mold temperatures closely followed air temperature in the oven, we avoided the complication of tracking part temperature with thermocouples during the cure. We checked vacuum pressure about every half hour of the cure to be sure that no leaks had developed in the bags.

Once the cure was completed, the chassis halves were moved up to the hangar for unbagging, hole drilling, and releasing.
Post-Processing and Assembly

Before we released the halves from their respective molds, we needed to use the drill bushings in the mold to drill our suspension locating holes. We found a long enough drill bit in the required 3/16 size, and set to drilling. We made sure to only drill through the outer skin of the chassis in this step, because not all drill bushings were adequately perpendicular to the part surface. Once the parts were out of the molds, we would use a drill guide to finish the holes. Most of the holes worked out fine, but the holes within a few inches of the parting line were not accessible with the drill, due to the flange being in the way. This wasn’t the end of the world, but wasn’t ideal as now we needed to make more jigs to locate these holes.

Thanks to the low CTE of our carbon molds, the parts popped out of the molds quickly and without much effort. The best way to release them was to hammer wedges in under the flange until they developed enough force to pop the part out of the mold. This momentous occasion of popping the first parts out of the molds obviously called for a mini celebration and photoshoot to help as a morale booster and just to keep the excitement of the team going.

Figure 166. In the Hangar, team member Phillip Coleman shows off the first part to be pulled from the molds: The top half of the c-car.
Figure 167. From the left: Kevin, Nathan, and Ford throw down a “we have the two chassis halves for the c-car ready for post processing and bonding” dab in the Hangar.

Figure 168. Kevin shows off the comfort, seating, and other ergonomics of the two freshly cured c-car chassis halves on the floor of the Hangar.

With the parts out of the molds, we needed to keep marching on with the manufacturing of the chassis. So after our quick new parts photo time, we got right to trimming up the parts so that they could be bonded together. The 5 main locations that needed to be trimmed were the cockpit opening, the two flanges and interface of the parts, the front and rear bulkheads on both parts, and the top of the
engine/battery bay. The majority of the trimming was done in the hangar paint booth to prevent any carbon dust from getting spread throughout the shop, and the carbon specific vacuum (which has a special HEPA filter on it for catching very small particles) was used to catch as much of the carbon dust as possible. Just to be cautious though, we made whoever was trimming the parts wear long sleeves, safety goggles, and a dust mask (or respirator if they were trained on it). We found that the best trimming tool was a dremel with a diamond cut off wheel. Although expensive, one of these wheels would last many times longer and cut faster than the usual Dremel reinforced cutting wheels. When using a dremel to cut carbon though it is very important that a carbon vacuum is carefully used to capture all the carbon dust possible otherwise the dust will get into the dremel motor and short circuit it and destroy bearings, rendering the dremel useless. We also used them at a very high duty cycle, usually until they were almost too hot to hold comfortably. These issues caused us to go through 2 dremels, which was a costly mistake. For carbon trimming it would probably be best to buy the cheapest knockoff dremels/rotary tools available, since they suffer so much abuse anyway quantity is more important than quality.

First up for trimming were the parting lines of each half. This was the longest trimming job, since we had to first cut off the leftover flange, then nicely trim the edge along the whole parting line for each half. Because of our problems with the mold, we knew that there would be a gap between the halves that would have to be filled, so we focused on trimming away and protruding material left from the flange.

Next it was time to cut out the cockpit openings, as well as the rear openings for each car. To do this accurately, we used one of the ME plotters to plot out a full-scale top view drawing of the chassis, and cut out the opening areas from this template. The template was positioned on the top half of the chassis and the position was verified using measurements from CAD. A paint pen was used to cover the edges of the opening with paint, covering both the exposed chassis and the template. The template was removed and the sharp line on the edge of the paint was used as the cut line.

![Image of cockpit and rear openings trimming](image)

**Figure 169.** Cutting the cockpit and rear openings. The sharp edge of the paint is the cut line.

We cut as close to the line as we were comfortable with the dremel, and finished and smoothed the edge using a large rotary sanding bit on an air tool.

After trimming the flange and parts interface, cockpit opening, and engine/battery bay top, but before trimming the front and rear bulkheads we bonded the two halves of the chassis together. We did this
because leaving in the bulkheads gave us a nice flat surface with which to help jig the two halves for bonding them together. For jiggging the halves we machined three jig blocks out of MDF and one aluminum one. The MDF jig blocks were machined to match the curvature of the top half of the chassis while bolting into the existing suspension mounting/locating holes in bottom half of the chassis. This allowed us to shim a gap between the two halves until the jig blocks sat flush to both halves of the chassis and then using the aluminum jig block in the front and one of the MDF blocks in the rear we bolted the two halves together, locking in their relative positioning (so that it matched the chassis CAD) and making them ready to have their bonding seam of resin and microballoons applied.

**Figure 170.** Kevin was extremely excited to start bonding the two trimmed halves of the c-car together in the Hangar paint booth. Meanwhile in the background Nathan checks the fitment of the main roll hoop.

**Figure 171.** The two trimmed halves of the c-car chassis jigged together and ready to be bonded together. The MDF and aluminum jigs are used to get the correct spacing between the two tub halves.
after the required trimming of the halves caused the parts to be about 1/8th of an inch shorter (each) than necessary to match the desired overall chassis geometry.

**Figure 172.** Shims are being used to get the correct spacing between the chassis halves before bolting the front jig on. Because the majority of the front bulkhead gets cut out, there were no locating holes on it to mount the aluminum jig to easily get the right spacing like the MDF jigs had on the sides.

Because of how we had designed the plugs, we ended up with molds that created parts without enough excess material to allow us to trim the flanges and part interfaces to the exact right dimensions required for a flush fitment of the halves. This left us with a sizable gap to fill in between the two halves. The distance of the gap ranged all the way from about ⅛” to around ½” in some places. Because of this sizable gap we decided to bond the two halves together using a thick mixture of resin and microballoons that we held in place using tape on the outside of the chassis while filling the gap from the inside of the chassis to prevent the bonding resin from flowing into the inside of the chassis. Larges syringes and popsicle sticks were used to push the mixture into the gap and smooth it as much as possible. The joint was left to cure and then was subjected to a final sanding after cure for more smoothing.
Figure 173. With the chassis halves jigged together, and before bonding them, both the front and main roll hoops get mocked up to check their fit.

Figure 174. A rear view of the chassis with the engine being mocked up in it for the first time. This is after the halves were bonded together but before closeouts or the lap joint were added.

When the resin bonding joint of the chassis was cured, we were then able to go about trimming out the front and rear bulkheads. Using the same method of trimming as on the rest of the chassis, we were able to cut out and trim the two bulkheads to match printed out templates that we had taped on
the chassis. This gave us openings with geometry that we trusted to fit all of the necessary components through and around. At this time, we also filled the exposed honeycomb at the edges of cutouts with a resin/microballoons mixture.

With all of the major trimming of the chassis done, we were able to start mocking up and test fitting major components into the chassis. This was very valuable to subsystem leads and team members as they not only saw the car coming together (which is always a morale boost), but they also got more confidence in their designs and abilities by seeing what they had made work (at least from a packaging perspective). The other valuable aspect of being able to mock major components in the chassis prior to being completely done with the chassis was that it allowed us the opportunity to find any manufacturing issues or design oversights with regards to the finer fitment of parts in the car. Luckily we didn’t run into any major issues with fitment, as the engine went in and out of the cockpit and engine bay nicely, and so did the battery pack. The half shafts didn’t contact the tub, the pedal box fit nicely at the front of the tub and didn’t require major shimming of any sort, and the roll hoops all fit snugly.

**Figure 175.** A top view of the engine being placed in the chassis for the first time. The seatback flange had not been bonded into the tub yet, making it easier to put the engine in than it was later in the season.
Figure 176. A front view of the engine in the engine bay for the first time. The joint between the two halves is clearly visible as the white line of resin and microballoons around the chassis. But look at all that space!

Figure 177. A glamour tunnel shot of the engine in the chassis. To note though, you can clearly see the difference in trim and therefore gap height between the two sides of the chassis.
Figure 178. The engine with the differential being removed from the chassis through the cockpit. Without anything else in the chassis this was possible, but was too tight longitudinally for us to remove both the engine and the differential together once other components were mounted in the chassis (specifically the seatback flange and the steering column).

Figure 179. Front view of the engine being removed from the chassis through the cockpit. Side-to-side the engine and differential had more than enough room to be removed from the chassis together even once other components were mounted to the cockpit.
The biggest benefit of mocking up components in the chassis though was that we were able to confirm that what we had said would fit, did fit. This allowed us to push forward with finishing up trimming the openings, potting inserts, finishing off the lap joint between the halves, and closeouts knowing that we wouldn’t have to go back and spend precious time on redoing one of those aspects of the chassis.

Figure 180. Our main man, Whackielassie, with his license to drill insert holes for the steering column mounts. And shade 5 welding safety glasses, because you know... Safety 3rd.

To finish bonding the halves of the chassis together after checking fitment of the major components of the car required a carbon strap joint on both the inside and outside of the chassis to be used over the resin seam. For the strap joint we used two plies of cloth on each side, the layup being [45c / 0c]. This strap transfers load between the two halves. We used a wet layup technique so that the chassis would not have to be taken back to the composites lab for another oven cure as the resin used for bonding the two halves was not rated to the necessary temperature. We had originally wanted to do the strap joint with prepreg and an oven cure, but we did not have time to deal with the complications that that would require. On the first chassis we were low on manpower and resources when we applied the strap joint. Because of this we decided just to cover the strap joint area with vacuum bag and use tacky tape to adhere the bag to the chassis around the strap joint. This unfortunately didn’t work very well, as the chassis was not an airtight surface, resulting in poor compaction of the strap joint while it cured. The strap joint then ended up with poor adhesion to the chassis and lots of sharp fibers sticking up off the chassis along the edge of the strap joint. We took care of this by sanding
down these rough edges of the strap joint. On the second chassis however, we learned from our previous mistake and made a sort of vacuum bag doughnut that encased the entire chassis and gave compaction on every surface. We did this by making a long cylinder out of vacuum bag, using tacky tape to connect the ends of the flat sheet into a cylinder, sliding the cylinder through the chassis, and then rolling the cylinder back on itself and the chassis to enclose the chassis with vacuum bag. This gave us significantly better results on the strap joint than using tacky tape to seal the vacuum bag directly to the chassis. Subsequently, we used this doughnut bag method for doing all the closeouts along the trimmed edges as well.

**Drilling**

Having finished the bonding of the two halves, we began to drill all of the necessary holes for mounting components into the tub. Because we already had locating holes on most of the flat surfaces of the chassis, we machined MDF boards into simple jigs that just had holes drilled in the correct locations on them. Steel bushings were pressed into the boards to serve as bushings. These jigs were then secured to the chassis using the preexisting locating holes from the drill bushings in the mold and used as a drill guide to drill the rest of the holes on the respective chassis face. There were a few holes that could not be drilled this way, as there were no locating holes on the same face as the holes that needed to be drilled. These included the front rocker mounting holes, the rear rocker mounting holes, and the rear shock mounting holes. To drill the front rocker mounting holes, an MDF jig was made to mimic the contours of the rocker boss from the outside of the tub with long holes machined into it so that when it was fit up against the contour of the rocker boss, a handheld drill with an 8 inch long drill bit would be capable of drilling through the outside skin of the tub. Then using a drill block, a squared block of aluminum with a hole slightly larger than the bit to be used, the rest of the hole was finished so that the hole still ended up being perpendicular to the chassis face that it was on. The rear rocker and shock mounting holes were done in a similar fashion with a machined jig out of MDF that fit the contour of the chassis and could be located using the rear suspension mounts on the side of the chassis. With the jigs located and bolted in place, the mounting holes could then be drilled with a handheld drill through the machined drill holes in the jigs. Although it took 6 flat jigs and the 2 specialty jigs (front and rear shocks and rockers) to accurately located all components, we were able to quickly knock out all of the holes in the chassis with minimal setup and lots of ease while having confidence in the location of the holes being where we wanted them. While no exact tolerance was determined due to not having an adequate measuring technique, all components lined up as expected.
Figure 181. Mocking up a rear rocker mount on the drilled insert holes while the seatback flange gets bonded in the c-car. The c-clamp and 8020 jig in the background is fixturing the seatback flange in place while the resin used for bonding it to the chassis dries.

Inserts

Figure 182. Fully potted and cured inserts for the rear shock and rocker mounts on the left side of the c-car.

Potting inserts came next once all of the holes on the chassis were double checked and resized for fitting inserts if necessary. To prepare the holes for potting an insert into them, first an insert was test
fitted into the hole to double check that it had been sized properly. Then the insert was removed and a special 1/16 inch allen wrench with its short arm cut down to 1/8 of an inch in length was used in a handheld drill to remove a radius of core from between the carbon around the insert hole. This allowed for space for the potting resin to be injected around the insert to solidly bond it to the chassis. So after removing the core around the insert holes we cleaned the surface of the carbon around the holes with acetone to make sure that nothing would contaminate the flange bonding surface. Then we mixed up cups of the bonding epoxy, using our “double-cup” method, and then applied it to the flanges of the inserts and inserted them into the chassis, making sure to firmly press them against the chassis face. The epoxy used was Loctite 9309 NA Aero, with glass microspheres mixed in for bondline thickness control, however any structural adhesive that flows well enough for potting and is temperature-resistant will work. Once all of the inserts on a side were bonded to the tub in this way, and before the flange bond was cured, we moved on to drilling through the small side holes on the insert flanges. To drill these holes we used a small 1/16 inch drill bit and drilled through only the first skin of the chassis. In this process, only the first skin of the chassis should be drilled through because the idea is to create a pocket of epoxy around the insert and if both of the skins are drilled through, then the epoxy will just leak out the other side of the chassis. As we moved from insert to insert drilling these holes into the cavities of removed core, we then filled the cavities with the same epoxy that we used on the flanges of the inserts. To fill the cavities around the inserts though, we mixed the epoxy and then poured it into a plastic syringe that had about a 1/16 inch wide nozzle. The syringe was then used to squirt the epoxy through one of the small holes into the cavity until it came oozing out of the second small hole. As soon as epoxy came oozing out of the second hole, we stopped injecting it and wiped off any excess epoxy from the chassis. Initially we didn’t have a method in place for marking which inserts had already been filled, and we just based it on the memory of the person who was injecting the epoxy around the inserts. However this quickly proved to be an unreliable method and as can be seen in the front rocker mounting failure section this was a costly method as well. We did transfer over to a paint pen marking system where as soon as epoxy had been injected around an insert, then a paint pen would be used to place a clearly visible checkmark next to the insert, indication that it had indeed been fully potted. This method, or a similar method that ensures and confirms that no inserts get only their flange bonded (and not fully potted) is strongly recommended for any future teams using inserts that require a bonding style like the ones used in this senior project. The cost for not potting an insert is too costly down the line, especially during testing, to not use a checking method to make sure every insert gets potted fully.

Once all of the inserts on a side were potted though we gave them overnight to cure enough so that we could then flip over the chassis and pott the inserts on the other side without the first side of inserts falling out. After all of the inserts had been potted and fully cured, most of them still stuck out past the chassis on their flange side because it was a lot easier to just get bulk inserts than to have to worry about specific length of inserts for every location on the chassis. To get the non-flange sides of the inserts flush with the chassis we used a rotary air tool (90 or 180 degree work, it’s more dependant on what the operator feels comfortable with) with either a flat sanding disk, or a sanding cylinder attachment. This worked very well to quickly trim the inserts until they sat flush with the chassis. However we did run into a couple issues when this task was given to other team members. The first issue came about when a team member tried sanding inserts they had just potted, ended up pushing them instead of sanding them. Luckily it was noticed that the flanges of the inserts were no longer sitting flush against the chassis before the epoxy cured too much and it was not much of an issue. It did help bring to light the fact that you can’t rush the curing process and need to give adequate time for the epoxy to cure before doing more work on or around the inserts. The second issue that we had involved a team member who was sanding down some inserts and wasn’t focusing enough on what
the entire sanding bit that he was using was doing. In an effort to sand down some inserts, part of the chassis was also sanded down all the way to the core, purely due to lack of attention to what all the sander was actually sanding. This was quite a significant error that ended up being fixed with some epoxy and a carbon patch over it, but could have been easily avoided during the sanding process and ended up costing the team unnecessary time and resources.

Figure 183. Both the e-car and c-car chassis sitting next to one another on the frame table in the Hangar. The c-car chassis has had resin and microballoons applied to cover some dry spots that were noticed in the chassis, making the bare carbon look much worse on the c-car than the e-car.

Temporary Aesthetics

Aesthetic prep on the car began after all of the inserts we potted on the car. One of the team’s goals for the year was professionalism, and that included the cars looking professional as well. So the chassis had to be smoothed and painted to achieve this look. The c-car because of more dry spots in the outermost ply of the skin and because of the issues with compaction of the strap joint, needed lots of filler to smooth out the outer bodywork. For filler we used microballoons mixed with a little resin because they are fairly light weight and sanded off relatively easily to smooth out bodywork. The aesthetics of the white microballoon and resin mix on the black carbon though was quite appalling. And because the e-car chassis looked so much better, while we presented the cars prior to painting them, we used black gaffers tape to cover up the white spots on the c-car chassis to make it more visually appealing. This worked quite well to show off and run the cars while we waited for an appropriate time to paint them since the gaffers tape looks similar to black carbon from a distance and didn’t leave any harmful residue when removed.
Figure 184. Top view of the two chassis next to each other on the Hangar frame table. The c-car chassis is farther along than the e-car chassis since it was the first one to be laid up and manufactured.

Figure 185. Vehicle progress two days before wheels on ground. The vehicles are nearly at the same completion point. Chassis lap joint quality difference can be easily seen.
Figure 186. Both cars in the Hangar after getting wheels on the ground. The quality in the carbon work between the lap joints of the two chassis is very obvious in this photo and is highlighted by the white resin and microballoons that were used on the c-car to smooth out the edges of the lap joint.

Figure 187. Both cars are displayed in ready to run trim at the unveiling event. Gaffers tape was used to cover up the poor aesthetics of the white resin and microballoons on the c-car chassis.
Additional Touches

**Figure 188.** Aluminum tape was used on the side of the chassis for heat shielding along the section that was in close proximity to the exhaust. The wire leading under the aluminum tape connects to a thermocouple that was placed there to keep an eye on chassis temperatures. The beautifully blued stainless steel exhaust pipe and aesthetic gaffers tape can also be seen.
Figure 189. The second revision of the heat shielding aluminum tape on the car is shown here. The main difference between revision one is the aesthetics of it for photos. Revision one did it’s job just fine and so did revision two.

In the attempt to make the car look professional we also needed to be attentive to safety precautions. Because we needed to protect the carbon that was in close proximity to the exhaust pipe on the e-car, we went through a number of revisions on the heat shield to make it more aesthetically pleasing. The first revision was purely functional, involving just straight pieces of aluminum tape to prevent the radiative heat transfer from the exhaust to the chassis. Aesthetically this was poorly done, and because the aluminum tape didn’t have any insulator behind it to reduce conductive heat transfer from the aluminum tape to the carbon we weren’t completely okay with the protection of the carbon. So the second revision involved a more aesthetic cut and placement of some actual aluminum heat shielding material that was basically just aluminum tape with an insulator between the aluminum and adhesive. The application of this was very easy with the exhaust off the car after approximate coverage lines were marked on the chassis. The sheet of the aluminum heat shield was easily cut with scissors and then simply stuck into place.
Figure 190. The inside of the cockpit of the c-car facing forwards. Gaffers tape was used along the lap joint edge to protect the driver from the rough carbon edge. On the floor of the chassis, around the driver’s legs, gaffers tape is also used to hold foam on top of the bolt heads and extra threads from suspension mounts to protect the driver’s legs and prevent the driver suit from getting caught during an egress.

One of the last things that had to be done to the chassis so that it was ready for testing was to pad the driver’s cell. A lot of the suspension mounts required the head of the bolts to be on the same side as the mount for the bolts to actually fit. Which meant that the excess bolt threads were sticking into the driver’s cell. This posed a problem for us as the drivers found that the excess bolt threads made the driver’s cell painful and uncomfortable when cornering or going over any bumps. It was also a safety issue in the event of a fire or any other reason that a driver might need to egress from the car quickly, as their driver’s suit or shoes could easily get caught on the bolts slowing down or even stopping their egress. To remedy this issue we covered the excess bolt threads with rags and foam and then held this down with gaffers tape. Had we had more time to design and manufacture (or have manufactured for us) we would have ideally made plastic bolt covers that could be threaded or snapped on to cover the excess bolt threads and provide a smooth protective covering between them and our drivers. This is something that future teams should definitely look into designing and implementing on the car, design the suspension components so that the heads of the bolts can all be inside the chassis.
Figure 191. The fully painted c-car chassis prior to having all the rest of the car’s components bolted back onto it. This photo is taken outside of the Bonderson high bay just before the Cal Poly Racing end of the year banquet.

Figure 192. Printer paper was used to mask off the freshly painted chassis to apply a ceramic paint coat over the area of the chassis that is close to the exhaust to help protect the chassis and paint.

Figure 193. The completed protective ceramic paint on the rear of the chassis.

Aesthetics won’t be harped on much here, because as far as performance is concerned, it is unimportant. The actual paint, design, and painting process is mentioned later on in the aesthetics portion of the report. In this paragraph though we would just like to quickly talk about the exhaust heat protection that was incorporated into the aesthetics of the car. Once the car was painted we used a ceramic engine enamel paint in an aerosol can to paint the section around where the exhaust ran along
the chassis. Using masking tape and regular printer paper, we masked off the area and applied 3 coats of the paint. The idea behind using this paint in the area was that it would just add an extra layer of insulation (poor conducting material) between the exhaust and the chassis. The aluminum heat shielding material was still applied over this ceramic engine enamel paint. This was just an extra precaution that we included.

Fixes

Figure 194. Due to interference with the wheel rims, the front lower aft control arm mounts had to be moved forward on the chassis. New inserts for this were done on the c-car after the car was painted and before it was reassembled. To help keep spirits high on the team with all the long hours we were putting in, we had all the team members in the Hangar that night sign the bottom of the chassis.

During the testing of the car it was found that the front lower control arms (FLCAs) needed to be modified to give the car enough steering angle to maneuver the competition course effectively. The
The design fix for this issue was to move the mounting point for the front lower aft control arm (FLACA) forward on the chassis, thereby narrowing the angle between the two FLCAAs and increasing the maximum steering angle possible for the car. We did this fix immediately after painting the whole chassis because that way all of the other systems were off the chassis and it could be flipped upside down for easy access to working on the bottom of the car. With the car flipped upside down, we stuck two long pieces of 8020 aluminum extrusion through the chassis and then held both the ends of the 8020 up on sawhorses to support the chassis up off the ground and at a decent working height. Then followed the same process as with all the other inserts. We cleaned the area around where the inserts would be bonded in with acetone, drilled the new holes using machined drill jigs, removed the core around the insert holes using the modified allen wrench and drill combo, bonded the flanges of the inserts, and then potted them as normal.

While all of this was happening, and the rest of the team was waiting to start the rebolting process, we had all of the team members sign the bottom of the chassis as a morale boosting event and to give every team member a feeling of collective ownership over the car that they had worked so hard on throughout the year.

With the inserts potted, we couldn’t wait with the chassis hanging upside down on two pieces of 8020 for the epoxy to cure, so we used bolts and large washers to secure the inserts in place while the epoxy cured. This allowed us to bolt the rest of the car back together and proceed with testing on the old FLCAAs over the weekend while the epoxy on the inserts finished curing. Then during the week we swapped to the new FLCA setup and were able to continue on with testing without loosing a full weekend while waiting for the new inserts to be ready to handle suspension loads.

![Figure 195](image.png) Re-bolting up the e-car post painting and post potting in the second set of FLACA mount inserts. The front lower control arms were mounted on the old set of inserts to give the resin on the new inserts time to completely cure before taking suspension loads.
Roll Hoops

The roll hoops were started by using the pneumatic “tube shark” tube bender in the hangar. We made sure to order enough material so that a couple practice hoops could be made prior to bending the final versions. We used a digital angle finder to track the bend angles. All of the bends were pretty simple, except the bend in the top of the main roll hoop, which is massive 127 degree bend with a 3 inch radius - up against the limit of what a .095” tube can practically be bent to. With the first couple practice rear hoops we noticed that when we backed off the cylinder to advance the tube shark to its next “cog”, the tube would spring back and unseat itself from the bending die. When the cylinder was engaged, the improper seating would cause the tube wall to buckle. Our solution was to eliminate any springback by jamming a big allen key into one of the holes that can be used for holding dies, before we released tension on the cylinder. This way the tube was not allowed to spring back at all and remained fully seated in the die, eliminating the buckling problems. By the final hoops we had figured out the correct amount of lube to use with each bend, and how to progressively complete a bend to make sure that the spring back of the steel was accounted for to get the correct angle on the bend. A progressive bend refers to making small, incremental bends until the desired bend angle is achieved. The benefit of this method is that it is much less likely to cause the hoop to be bent more than is desired. After all 4 hoops were bent, the mounting brackets were water jet cut and bent, and the mounting holes in the chassis were drilled, we mocked up the roll hoops in each chassis so that the mounting brackets could be tack welded onto the roll hoops. Tack welding the brackets onto the main roll hoop was a relatively standard tacking job, but the high cockpit walls made tacking the brackets onto the front roll hoop a challenge. In order to get a good angle and visibility to weld the tacks it was found that laying down inside the tub was the best option (especially since not only did it give a good angle and visibility for tack welding on the front hoop brackets, but it was pretty comfortable as well). Once the brackets were tacked on, we removed the roll hoops from the car so that they could be fully welded. The reason for doing it this way is so that everything can be positioned correctly on the car, but then you don’t burn the carbon or resin by full welding any joint while it’s contacting carbon. The last bits to be manufactured for the roll hoops were the braces for the main hoop. For these the mounts were made first by simply welding the water jet cut tabs to the vertical tubes. Then 2-D cutouts of the tube profile were made, printed, cut, and taped onto tube stock. From this we hand notched the tubes with bench grinders until they matched the profiles taped to them. This gave us a pretty close to finished brace which we then were able to fit up with the main hoop, brace mount, and tub to get final fitment and do finishing touches with the grinder until both braces on each car fit snug and they both measured equidistant from the main hoop centerline and equidistant from the top of the main hoop. These fit and finish grinding operations need to be carried out because the templates from solidworks assume that the ends of the tube will approach zero thickness, which you almost never achieve with a grinding wheel. Similarly to how the roll hoop brackets were welded, the main hoop braces were also tacked into place on the tub and then removed to be fully welded. For all the welding on the roll hoops there was never any jig used besides the chassis during the tack welding.
Figure 196. View of the welding position from inside the tub.

Figure 197. Tacking the main roll hoop braces while using the chassis as a jig to make sure that the roll hoop will fit correctly.
Figure 198. Off chassis welding setup to full weld the brackets onto the main roll hoop.

Figure 199. The main roll hoop braces during a fit check on the c-car in between grinding operations. The closer of the two braces still has the paper template taped on it to guide the grinding process.

Nosecone

The nosecone was made in a manner very similar to the chassis, with a carbon mold using leftover materials from the chassis molds. Since the nosecone is a smaller part and has complex curvature which helps stiffen it, no core was used in the molds, and the layup was made thinner, using only 4 plies total.
Figure 200. The nosecone mold after being pulled from its plug.

The actual layup of the nosecone was much simpler than the layup of the chassis, because the nosecone had no structural needs and was only really a fairing around the impact attenuator, so it only needed to be a couple plies of carbon. This also meant that we could make multiple nosecones in a relatively short period of time to determine how few of plies were necessary in the nosecone. The first nosecones we made consisted of two plies of HTS40 cloth, and carbon templates were made in a similar way to the chassis templates. This first ‘cone proved very difficult to pull out of the mold, and it was damaged in the process, with multiple cracks where it had been pulled at with vise grips. We resurfaced the nosecone mold and used more mold release for future nosecones, which helped alleviate this problem. Although it held its shape, this first nosecone was somewhat flimsy. Our next iteration used two rips of ¼” core around the circumference of the ‘cone to stiffen it up, and this worked well.

To mount the nosecones to the chassis, we used ball-locking pins from McMaster for a fastening method that would be easy to remove quickly. The holes in the chassis were drilled first, and aluminum tube inserts were bonded into the holes for the balls in the pins to bear against. The holes were match drilled into the nosecone. This setup worked well more a while, but after a few test days, the holes in the thin nosecone had begun to tear. To fix this, we bonded on plastic washers to either side of each hole to reinforce the skin, and this worked well. It would be easier in future years to just make the nosecone thicker where the holes will be.

AI/IA Plate

The anti-intrusion (AI) plate and honeycomb impact attenuator were probably the easiest components to make for the chassis. The AI plate was a fairly simple aluminum waterjet cut part and the impact attenuator was a purchased section of pre-sized aluminum honeycomb core. All the manufacturing that we had to do was sand and clean the surface of the AI plate, mark the position of the impact attenuator, and bond it on.
The eight holes to mount the AI plate to the chassis were drilled using a spare plate as a jig to mark hole locations. Next up was to weld the mounting bolts to their plates, and bond them onto the front bulkhead. This was a straightforward process, and the plates were clamped to the chassis by tightening nuts and washers onto the outside of the bulkhead.

Unfortunately, not all of the mounting bolts went in straight, and the keyholes in the A plate has to be filed slightly to accommodate. In the future a better jig should be used to position these bolts and keep the straight and parallel.

One comment that we received during the technical inspection for the combustion car at competition was that we should have bonded the pre crushed side of the honeycomb to the AI plate instead of the uncrushed side like we did. Doing so would provide a better bond between the two components because of the increased surface area on the honeycomb from the pre crushed core. However, we were still able to get a good enough bond to hold the impact attenuator onto the AI plate when bonding the uncrushed side of the honeycomb.
Seatback Mounting

The flange that would be used to mount the seatback/firewall was constructed in two pieces so that we could easily fit it into the cockpit opening. It was made of a carbon sandwich panel with $\frac{3}{8}$" aluminum honeycomb core, and the pieces were cut to a rough shape using paper templates and a dremel. We knew that the inside surface of each chassis would be different, especially around the strap joint area, so the final trimming was done manually by checking the fit of the pieces inside the chassis.

![Figure 203. Seatback flange pieces rough trimmed and on the scale before bonding into the chassis.](image)

The jig used to get the flange pieces in the right spot for bonding was made of several pieces of 80-20 extrusion, and was before being clamped to the chassis was set to the proper seatback angle. It was then inserted into the chassis and clamped down, locating off of the front of the cockpit opening and the top of the shoulder harness bar. The pieces were trimmed bit by bit until they fit well, the angle was checked with a digital angle finder, and a resin/microballoon mixture was applied to the edges of the flange pieces using a syringe.
The flanges were bonded in nicely, although the epoxy bond line was very visible and not very aesthetically pleasing. We didn’t have time to cover it with a wet layup or paint it, but this is recommended for future teams.

**Seatback/Firewall**

The seatback, ah yes the proverbial seatback. Made from the sandwich of carbon, forged in the ovens of the composites lab. The seatback was made out of a standard carbon sandwich panel that used prepreg carbon and 1/2 inch aluminum honeycomb core cured on a large metal tool plate. After coming out of the oven the seatback was trimmed using a printed template and a dremel with a diamond cut off wheel. What wasn’t included in the printed template when the seatback was initially trimmed was the pass through for the wiring of the electronics system. This was added in as a cut out radius based on the approximate diameter of the wiring bundle. Mounting holes were also drilled into seatbacks using the printed template and a hand drill with a drill block. Mounting on the chassis side of the seatback interface was accomplished by bonding bolts into the seatback flange in the same manner as the nosecone, so that it could be removed easily.

The e-car seatback/firewall needed additional material layers to pass the EV firewall rules; a conductive layer on the side opposite the driver, and a fireproof insulating layer on the driver’s side. The conductive material used was a sheet of .015” aluminum, and the insulating material was a .020” sheet of FR4 garolite purchased form McMaster. Both were prepped and bonded to the carbon sandwich panel to complete the firewall.

**Harness Mounts (Shoulder)**

The shoulder harness mounts for the chassis were made out of individual pieces of 6061 aluminum using a waterjet and a manual mill. The pieces started off by being cut on a waterjet into their overall oval shape. Once these blanks were received from the waterjet, we mounted them one at a time on a manual mill using parallel bars so that they would sit high enough out of the vice to have all
operations done on a part in two setups. After being fixtured on the mill we drilled the bolt holes in
the mounts. Since the most important aspect on the holes was their location relative to each other we
didn’t bother with machining the sides of the mounts so that they were perfectly square in the vice
(the holes would still have an acceptable positional tolerance relative to each other regardless). Next
we used a 1/2 inch end mill to remove the material in the channel in between the bolt holes for the
harness to be wrapped around. After this we the used a 1/2 inch chamfer end mill on the edges of the
channel to provide a nice non sharp corner for the harness to wrap around. Finally we flipped the part
over and used the same chamfer end mill to put a chamfer on the outside of the channel. Lastly we
deburred the parts, bead-blasted them and they were ready for action.

Figure 205. Finished shoulder harness mount on the scale.
Vehicle Testing

Most of the testing that we experienced with the car and monocoque is explained in detail in the section describing the specific aspect of the tub where something of note happened, but in this section we have briefly outlined some of those noteworthy events. Specifically we have picked out items that related to the overall car as well as the chassis, or where the chassis played a non-primary role in the event of note.

One major issue was CG height on the c-car that along with higher than anticipated lateral acceleration allowed for unloading of the inside two wheels and the possibility of flipping. The main reasons for the high cg are high engine placement and driver position. Engine height was higher than expected due to poor engine CAD, which could have been adjusted during testing by remanufacturing the engine mounting components, but wasn’t. A lower, more reclined driver position would also have a significant effect at reducing CG, but would require an increase in the length of the chassis to accommodate taller drivers.

![Figure 206. Marvelous example of the negative impact cg has on performance.](image)

Apart from the negative impact of the seating position on CG, the ergonomics turned out pretty good. There was plenty of room for the driver’s elbows because of the spacious cockpit opening. The pedal position was able to accommodate a wide range of drivers, however placement was a little too close for taller drivers. It caused the driver’s knees to hit the steering wheel during tight turns due to the steering wheels rectangular shape. Another complaint was the lateral support of the torso portion of the seat. The inadequate support caused greater fatigue in the driver’s arms.

Another problem that caused many delays in testing was the chain tensioner solution in the c-car. Instead of using shims like the team historically has, or using an eccentric diff carrier like some other teams do, our car used a chain roller which failed on numerous occasions. The soft mounting of the
engine, as mentioned in the engine mounting section, did not cause any failures that stopped testing, but it did cause worrying amounts of deflection that cause us to add a third engine mount and to increase the stiffness of the engine mounts. Also, as seen from the failed section of core in the engine mount section, the engine loads did damage the monocoque, but never to a degree that prevented car running or that completely ruined the structural integrity of the monocoque.

During testing many changes were made to ensure the car handled as well as possible. To ensure this, the balance of the car was fine tuned to allow the maximum amount of lateral acceleration front and rear and to allow the car to do what the driver asks of it. Changing spring rates was the primary method to fine tuning the balance of the car, however if the chassis is not stiff enough in torsion, these spring changes would have no effect on load transfer distribution. Therefore we needed to design and manufacturer a chassis with an adequate torsional stiffness. As mentioned previously, our chassis ended up being softer than designed, but fortunately this did not inhibit the changes made in roll stiffness distribution. Changing the spring rates on one side of the car (front/rear) made a noticeable change in balance to the car, both in driver feel and timed skidpad laps. It is difficult to say if an even stiffer chassis would be more beneficial however.
**Competition Results**

The first part of competition was passing the tech inspection. Front roll hoop braces needed to be added because it was needed in the tech sheet. This was something that should’ve been checked before we arrived at competition. Mounting the wing to the rear roll hoop was a concern but the rules allowed it so it is important to bring the rulebook to address concerns such as this. Another issue was regarding the Impact Attenuator. The pre-crushed side should be the side that gets bonded. Also the calculations for proving joint equivalency were incorrect because ultimate strength, not proof strength should be used.

Feedback from design judging was very positive for both chassis, although there were details that design judges wanted more focus on. Ergonomics-wise they liked the wide opening in the cockpit, the seat position and comfort, and the pedal adjustability. However they thought access to the bolts should be better to allow for quicker position changes. They also found the same ergonomics issue we did, where the steering wheel hits the drivers thighs. The clutch lever position was something that did not bother our drivers but a judge said it was inadequate.

One judge in particular was concerned about the 3d printed plastic inserts and believed that aluminum would be a better choice and could be bonded successfully. The judges liked that we were thinking about loading points rather than just global properties of the monocoque. However, they wanted to see more stiffness analysis of hardpoints. They also mentioned concerns about rear rocker mount compliance.

The judges were very impressed with all the testing done, including hard points, brackets and aluminum core. They liked that everything was tested to failure and that the failure modes were explained on the physical samples that were brought. We received positive feedback on the suspension bracket test as well, especially for explaining how deficiencies in the bracket changed the failure mode in the composite test piece. The judges liked our explanation of the rocker mount failure, including why it happened and how it was repaired. We said a switch to a pushrod configuration would be beneficial in reducing the loads, therefore reducing the likelihood of a failure like this, which the judges agreed with.

Other things they liked included the documentation of the manufacturing techniques, the explanation of core choice for side impact, and the torsional stiffness study of the cockpit. However they wanted our FEA model to include brackets. They also liked the development and rationale behind the soft engine mounts, including the design process as well as the dynamic loading calculation.

One of the concerns they had was why the physical model was significantly less stiff than what was designed for, especially since we did not have good reasons for this. They stated that the teams that go to design finals have a well-developed torsional stiffness model to which the physical model agrees within 15%. They also did not like that our chassis used uniform laminate all over, and had concerns over weight. The build quality was rated good overall but fairly average compared to other team’s monocoques. The outside was nice but the inside could have better surface finish without exposed honeycomb and glue joints. They also thought the roll hoops should be inside the car for aerodynamic as well as aesthetic purposes. Even though the chassis was not perfect and had some issues, they liked all the explanations and understanding of everything. The design judges specializing in composites said our chassis was one of the best and lightest they saw.
With the combustion car we achieved our goal of a top 10 finish and were just shy of our goal of 690 points. The presentation and acceleration scores were not very high but our autocross, endurance, and efficiency score helped raise the overall score enough to place us in 9th place.

While the electric car did not meet expectations at competitions, there were significant improvements in the vehicle from prior years. The static event scores went from worst among competitors in 2016 to being among the top teams in 2017. This is highlighted in being a close runner up for design finals. Unfortunately, the electric vehicle was not able to pass EV technical inspection, and the relative inexperience of the electrical subsystem made running a huge challenge.

<table>
<thead>
<tr>
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<th>2017 Points</th>
<th>Average Historical Lincoln 10th Place</th>
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<td>688.3</td>
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Table 16. 2017 combustion car results and targets per event compared to the points achieved at competition in 2016.
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<th>2017 Points</th>
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<td>Efficiency</td>
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<tr>
<td><strong>Total</strong></td>
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<td><strong>82.0%</strong></td>
<td><strong>24.5%</strong></td>
<td><strong>45.6%</strong></td>
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Table 17. 2017 Electric car results and targets per event compared to the points achieved at competition in 2016 as percentages of total points possible in an event.
Mass Properties

When it was all said and done, our chassis’s were able to meet their weight targets with surprising detail. The combustion chassis was one pound heavier than expected, and the electric chassis was one pound lighter than the goals we set out to hit. We believe that the one pound difference is well within an acceptable error range.

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Table 18. Combustion Car Chassis Mass Properties.
Aesthetics

Although aesthetics were low on importance since it has no impact on performance, it was decided that appearance helps the professionalism and static events of the team. Pictures of the cars will be seen by thousands of people on various websites and many others at competition and other events. A nice and professional appearance ensures an initial impression that attracts attention and represents the devotion to detail that the team strives for. It shows the cars were not put together last minute and will almost definitely improve the design score.

Figure 207. Render of car paint and graphics.
Figure 208. Electric car being primed and painted.
Figure 209. Combustion car being painted.

Figure 210. Electric car paint being finished.
After rendering various graphics both chassis were primed and painted a gloss gray with a clear coat finish. Yellow stripes were added for the c-car and green stripes for the e-car. White sponsor stickers that are clearly visible were applied afterward. The painting process took about a week per tub to do, including all the sanding and prepping, priming, and then final painting.

All of the prep work was done in the Hangar paint booth, and so was most of the painting. However, because we were on such a tight schedule with regards to testing that we did do some painting off campus in the garage of a team member’s friend so that we wouldn’t not have to stop painting at midnight like the shop requires. To prep the chassis for paint, we used bondo (as minimally as possible) to smooth out any dry spots in the carbon or steps caused by the strap joint. Then we went through and sanded the chassis starting with 80 grit, all the way to 400 grit in increments of 80. Once the prep was done we got to painting, using one layer of primer, then 3 layers of paint (plus the layer of accent color), and then one layer of clear coat. All the painting was done using a paint gun and was given the proper amount of time to dry inbetween coats as specified for the paint.

Small rock chips in the paint were noticed throughout testing and comp, and in the rear of the tub at the opening of the c-car a silver dollar sized chunk of paint flaked off. Other than those blemishes, there was never any major damage caused to the paint, which left the cars looking good even after comp. Occasionally the rubber and dust from the track had to be wiped down though to make the cars ready for a photo shoot. The black nosecones shown in the figure below were replaced with color matching nosecones for competition (similar to the initial rendering). This slight change added greatly to the aesthetic value of the overall car.
Figure 212. Painted e-car on the left and e-car on the right.
Recommendations

Although we are happy about the way the chassis turned out, and we believe they are a substantial improvement over the previous chassis, there are several areas that could be enhanced to make this design more successful.

The ergonomics of this chassis were a markable improvement over the previous chassis, with a large cockpit opening and comfortable driving position, however increasing the seatback angle and therefore lowering the driver’s CG would substantially improve the vehicle’s overall cg. Another minor change that could be made is to slightly raise the steering wheel or place the pedals slightly more forward so that the driver’s thighs do not hit the steering wheel at full lock. It should be noted that this was only an issue for taller drivers.

Another area where the chassis could be improved is stiffness. We started the year with what we believed was a validated model of the 2016 chassis, although it became apparent throughout the year that it was likely not representative of our vehicles testing data. We likely simplified too many parameters within this model, and arriving at a validated stiffness was probably luck. We believe the best way to change this situation is to spend time completing subscale stiffness tests of everything from suspension links and rod ends to inserts and tabs to build up a detailed model. Without that detail, even a validated model is untrustworthy. Key culprits to where we think stiffness was lost are the tab design (nongussetted steel), insert design (offsetting tabs from the laminate), steep pullrod angles in the front suspension, out of plane bending in rockers, and a laminate made with weight instead of stiffness in mind.

In terms of stiffening the chassis structure itself, there are numerous improvements that could have been made in which we did not have the time to complete. Bulkheads throughout the chassis, such as stressed roll hoops and seats, were shown to provide large stiffness increases around the cockpit, although difficult to implement. The realization that warping is likely a contributing factor to cockpit stiffness loss lends itself to further studies needed of longeron or ribbing reinforcements along the cockpit sidewalls. Additionally, ply orientations that focus on bending rather than pure torsional stiffness should be looked at. Implementing core reduction or removal in the floor was found to be beneficial to reduce CG without a significant stiffness decrease, although was not fully implemented.

In addition to a needed increase in stiffness, we have identified several key areas in the chassis where we believe we can save significant weight. One way in which both weight and manufacturing time could be reduced is to eliminate the postbonding of the chassis. Epoxy used just to bond the two halves was estimated to weigh around 3 pounds, while the strap joint and closeouts were upwards of 2 pounds. Through connecting the chassis halves together prior to the cure, there is potential to reduce manufacturing time by a week and most of that weight.

Another issue that needs reopening is removing the AI plate and integrating it’s capability into the front bulkhead. While there would be tradeoffs required with removing the plate, increasing bulkhead capability, and creating a hole above the pedal box for access, it will likely net a few pounds of weight savings.

For manufacturing, the process could be shortened by making all suspension/locating jigs in parallel with the laying up and curing of the chassis. This would prevent down time once the tub halves are
cured and bonded together. Also, having jigs for every single insert/ hole (EVERY SINGLE ONE) that needs to go in the tub so that all the holes can be drilled at once, and so that every insert can be potted at the same time prior to putting components into the tub would be extremely beneficial to manufacturing. As a note on the potting of the inserts, it is very crucial that every insert be potted and that they are potted correctly. Using check marks to denote which inserts have been potted (or a similar method) is necessary to confirm that all inserts have been potted to prevent insert flanges from being bonded to the surface without the area around the insert actually being potted. Similarly when doing any work with carbon (or really any manufacturing at all) it is necessary to make sure that every procedure is followed properly and that short cuts aren’t taken because they can drastically impact the integrity of the part being made. When doing repairs or any post work on the carbon with wet layups it is strongly recommended to take the time to make sure that there is a good vacuum seal around that layup, even if it means bagging the entire monocoque to do so.

Another suggestion for manufacturing of the tub for future teams is to incorporate the lap joint as a pre-preg layup in the initial layup of the monocoque. To do this the team should layup the top and bottom halves of the tub the same way as if they were going to cure both parts separately. Only the faceskin on the tooling should be laid up to the parting line, with core a sizeable distance away. Once both halves are in their molds the two molds can be fixtured together to make effectively one mold that houses the part inside of it. Once everything is secured together the lap joint plies and core can be added to the inside of the tub through the cockpit opening. Curing the tub as one part can eliminate about a week’s worth of post processing. When doing this, the team should be careful to make sure that the halves are lined up correctly in space to prevent creating a tub that is missshapen or too tall/short. Overall though this can save about a week per tub in cutting and shaping the carbon, which is labor intensive and widely regarded as the worst job an FSAE member can have.

The quality of parts should never be sacrificed to save time or trying to manufacture a higher quantity. There were occasions in which seven or more test panels were laid up, but even with a few people there was not enough time or equipment to debulk them properly, and some pieces were not debulked at all before being cured. These parts ended up with many voids and were inadequate for testing. Therefore, it is crucial to spend extra time to produce quality parts, following the procedure and making sure no details are overlooked, even when the team is behind schedule. Two panels were made painstakingly perfect, taking several hours to cut and lay up skins for only two panels, debulking for over 36 hours, and then assembling the sandwich and debulking again. Because of this attention to detail, the panels were 30% lighter than anything we have ever used.

As was mentioned in the engine mounting section, we have some recommendations for the engine mounting in the monocoque. Mainly our suggestion is that future teams should do some tests to better understand two things: engine loads, and a carbon sandwich panel under vibration. We made assumptions that the engine loads were primarily coming from internal eccentrically rotating, and we assumed that a large, flat carbon sandwich panel would potentially delaminate when continuously exposed to the vibrational load of the engine. However, during testing we noticed that the loads placed on the engine and differential during acceleration and braking were significant, and need to be looked at for future designs (they are either originating from the tire’s interaction with the track, or from the impact load caused by the load in the chain changing direction, or a combination of both). Along with including these loads in the design, the engine loads should also be characterized better than how we did it through our matlab script. Potentially this could be done on the dyno with accelerometers mounted on different points of the engine or strain gauges and a mounting system of two force members. Also, if the team can test a carbon structure that is representative of the bottom of
the tub under vibration similar to what the engine outputs, then the team might be able to find that using vibration isolators between the engine and the tub is not necessary. This could be done by making a dyno test stand out of a carbon sandwich panel and running the engine on it with the engine hard mounted to the sandwich panel. In addition to understanding the engine loads and a carbon sandwich panel in vibration more, future teams should really look at decreasing the weight of the engine mounts, making sure to utilize the third mount, and hard-mounting the engine to the monocoque.

While plenty of progress was made in the hardpoint design in the 2017 vehicles, there is still a considerable amount of work needing completion or revisiting for future years. The most prominent issues were the tab and the flange offset from the surface. While it was initially intended to offset the tab from the surface to increase strength, the current tab design did not make it possible to show that the offset had an advantage. Additionally, we now have conclusive test data showing that the tab design is the limiting factor for the suspension joints in terms of strength, and likely in stiffness too. Additionally, further attention is needed into how the local reinforcements affect the system hub to hub stiffness. While this test was primarily focused on strength, as that is what we believed was most crucial to our current vehicle, a study of how parameters such as insert choice, potting radius, and padup thickness affect vehicle stiffness would be beneficial. Out of plane coupon testing of the insert is also needed for future vehicles to complete joint characterization, as that parameter was skipped in the interest of time for this project. Moment allowables could also be tested, although with the current philosophy of avoiding any moment in a bolted connection, it may not be worthwhile.

SES is one portion of the FSAE competition that can be difficult to understand and comply with. At first glance, it seems that the goal of SES is to use testing to obtain material properties for use in proving equivalency, and this is essentially what it does. All the calculations are beam bending calc's using an isotropic facesheet with properties derived from 3 point bend testing. However, the 3 point bend test doesn’t come close to giving you an accurate strength property of the laminate. The panel will never fail with a fiber failure in tension (the failure mode you would need to obtain a good strength number), but will always fail on the compression side, usually with some sort of fiber buckling or core failure due to the nature of the test. Therefore, the data obtained is extremely conservative, and depends on many other factors other than just the tensile strength of the laminate. Of course, this means if you do it right you can exploit it. The 2013 chassis senior project report covers this in more detail.

A laminate can be tested with any core thickness, and the thickness can be varied across the chassis to pass. Using a thinner core is advantageous because the aspect ratio becomes lower and more closely approximates bending, reducing the shear stresses in the weak core and increasing the strength of your material in SES. The overall failure load will be lower, however the tensile strength in SES will be higher. The laminate used as our base laminate barely passed perimeter shear, which means that it will be very difficult to make a thinner laminate. Denser, stronger core greatly improved the performance of the side impact structure. This is because the side impact laminate has to be thick enough that our standard 3.1 aluminum honeycomb is the weak link. The panels will almost always fail by core crushing at the load applicator, which will cause the fibers to buckle. A stronger core can sustain more load before crushing, thus supporting the skins to a higher load at a smaller weight increase as compared to adding more plies. Experimenting with different core densities would also be very beneficial to decrease weight. There is always a balance in strengths between the core and skin. Adding extra plies will not help if the core has a low strength. Another aspect that should be looked at is making the front bulkhead lighter, or at least thinner to make more space inside of the chassis.
Conclusion

At the beginning of this project we set out to develop a chassis platform that could be run for both the combustion and electric Formula SAE cars of the Cal Poly SLO FSAE team. We decided to develop a full carbon fiber monocoque chassis for its superior weight, stiffness, and manufacturing times over other frame types. We were able to overcome the differences between the two powertrains with small compromises to the engine and battery bay to allow both cars to run on the same chassis geometry, but with different layups where necessary. We conducted detailed material testing to characterize our composite materials, which allowed us to select the lightest laminate that passed our requirements. In the combustion car, the issues of engine vibrations and heating were able to be taken care of through rubber soft mounts and aluminum tape heat shielding, and the battery box and powertrain of the electric car was able to be repackaged to fit in the shorter chassis. With the use of lightweight reinforcements and inserts, we were able to efficiently mount components to the monocoque while also managing the loads into the chassis. While we did not achieve our stiffness goal, we identified several key areas in which our team has overlooked in prior years and can improve in coming years. We introduced a simpler, slightly lighter nose and impact attenuator design that saved time and effort. The tooling is the teams first chassis mold to prominently feature removable plugs for features to increase modularity for future years. With the use of a carbon fiber laminated mold, we were able to reduce demolding time from one week to under half an hour. In total, we were able to reduce the manufacturing time of chassis production by well over 50%, highlighted by laying up and pulling two monocoques from the mold in 10 school days. Overall, the two chassis are the lightest that Cal Poly Racing has produced since the introduction of the SES ruleset. The chassis even came out as being comfortable for drivers to sit in, with ample elbow room for everyone. We were able to construct the chassis and cars overall to a higher standard of quality and with more attention to detail compared to the previous couple years. All of this along with the team’s hard work ended up netting a 9th place finish for the combustion car, developed a reliable mechanical platform to further the electric vehicle program, and laid the groundwork for many more successful years to come on this chassis platform.
References


Extra Photos

**Figure 213.** The e-car displayed with aero, suspension, and drivetrain on the lawn by Spanos theater for Cal Poly open house. The rain was worrisome on the unpainted steel roll hoops and suspension links which needed to be thoroughly dried to prevent rust.

**Figure 214.** “Formula Bobsled” as demonstrated by Nathan and Carl (the team’s brakes lead) in the e-car. To get a quick check of static load and setup of the suspension, Carl sat in the battery bay and Nathan sat in the cockpit. We also contemplated pioneering the new collegiate challenge of Formula Bobsled...
Figure 215. The e-car out on the apron of the Hangar after the first Go-No Go for the car. The nosecones were still being manufactured, which lets you clearly see the AI plate and the impact attenuator.
Figure 216. Packaging and space on top of the engine bay. The main roll hoop braces (and the main roll hoop itself) have started to rust since they haven’t been painted yet.
Figure 217. Packaging and accessibility of the electronics, fuel tank, catch cans, and engine with the seatback removed. Although it isn't the cleanest and prettiest to look at, everything is visible and accessible.
Figure 218. Steensma, the team’s assistant engine lead, working on the fuel tank with the c-car up on sawhorses. The large opening that was accessible with removing the seat back was incredibly valuable for all engine and engine related work.
Figure 219. The c-car after being completely rebuilt post painting. Ready for more testing!
Figure 220. “Ellie” as the c-car was known among the team with her hood stickers on. Shown front and center is the decal “Save the Manatee” which is an organization that a team member donated to in order to support the mascot for both the cars: the humble yet magnificent manatee. This mascot was adopted for the cars after our advisor, Dr. Fabijanic, made a comment that one of our preliminary chassis designs looked like a manatee.
Figure 221. The c-car after the full livery, including partner logos, had been put on. The right side of the car had different partners displayed than the left side. We made a big effort to make sure that the cars were aesthetically pleasing to help promote the overall team and improve our image to the public, Cal Poly, students, and our partners.
Figure 222. The left side of the c-car with full livery applied. Time and care were put into making sure that all the partner logos were fit well and be able to be seen while the car was driving. We also made sure that the partner logos looked good in their layout and were parallel to the main lines of the chassis when they were applied.
“Epiphany” as the e-car was known amongst the team. The names for both cars follow the traditional naming convention that the team has followed since Annie, the 2012 car. The naming convention follows that every new chassis year takes a name starting with the next letter in the alphabet and ending with an “ee” sound. We made sure to leave just enough space between the locating lip for the nosecone and the Zodiac partner logo to place the technical inspection stickers at competition. It was tight, but we did double check to make sure that they would fit before going to competition.
Figure 224. Both the c-car (on the left) and the e-car (on the right) are being shown off at the senior project expo. Both are in full livery (minus the final nosecone painting) and ready for final testing before competition.
Figure 225. Positioning both cars for a team photoshoot. If you look good and feel good, you’re going to perform good too.

Figure 226. Both cars with the subsystem leads of the team. The Fast and the Formula team members in the picture are Kevin (second from left), Nathan (tenth from left), and Ford (second from right).
Figure 227. Both cars with (essentially) the entire team. The Fast and the Formula team members in the picture are Kevin (sitting in the e-car), Nathan (standing row, thirteenth from the right), Mike K. (standing row, tenth from the right), and Ford (sitting in the e-car).
Figure 228. Ellie, the c-car, with all her technical inspection stickers at competition. With all four of the safety inspections passed, she’s ready for the practice track and dynamic events! You can also see her weigh-in sticker verifying the front/rear and left/right weight distributions (48.2% front, and 50.2% left) along with overall weight (428 lbs).
Figure 229. Checking tire pressures on the c-car before sending it out to the practice track with our hot-shoe driver to try and workout some last minute aero issues before endurance.
Figure 230. The competition team on the last day of competition with our 9th place overall trophy, our 2nd place fuel efficiency trophy, the last few rays of sunlight, and a smiling Fabio.

Figure 231. Team lead, Adam, and e-car technical director (and The Fast and the Formula team member), Ford, receiving the trophy for placing 9th overall at competition.
Figure 232. Team lead, Adam, and c-car technical director (and The Fast and the Formula team member), Ford, receiving the trophy for placing 2nd in fuel efficiency at competition.
Appendix A - QFD

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2015 C-car

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Appendix B - Engineering Drawings
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2-point compliance test fixture
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<td>5/8&quot; ID X 1 1/2&quot; OD X 0.065&quot; WALL</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>4110 ALLOY STEEL TUBE, 5/8&quot; LENGTH 1 1/2&quot; OD X 0.065&quot; WALL</td>
<td>Main Roll Hoops</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>4110 ALLOY STEEL TUBE, 5/8&quot; LENGTH 1 1/2&quot; OD X 0.065&quot; WALL</td>
<td>Front Roll Hoops</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>4110 ALLOY STEEL TUBE, 5/8&quot; LENGTH 1 1/2&quot; OD X 0.065&quot; WALL</td>
<td>Hoop Bindings</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>4110 ALLOY STEEL SHEET, 0.041&quot; THICK</td>
<td>Machining Plates</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>ALUMINUM BARE SHEET, 0.041&quot; x 1 1/2&quot; x 36&quot;</td>
<td>1 piece</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Online Metals</td>
<td></td>
</tr>
<tr>
<td>1-Time Use Multi-Point Temperature-Indicating Labels, Pack of 5, 3550K15</td>
<td>1 piece</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>McMaster-Carr</td>
<td></td>
</tr>
<tr>
<td>HT7416/7275-1 3/8&quot; x 2 1/2&quot; Twill</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>SpacesX</td>
<td></td>
</tr>
<tr>
<td>NBR7030 12X</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>SpacesX</td>
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</tr>
<tr>
<td>NBR7030 12X</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>SpacesX</td>
<td></td>
</tr>
<tr>
<td>CR1.1/1-5056</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Honeywell</td>
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</tr>
<tr>
<td>CR1.1/1-5056</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Honeywell</td>
<td></td>
</tr>
<tr>
<td>CR1.1/1-5056</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Honeywell</td>
<td></td>
</tr>
<tr>
<td>CR1.1/1-5056</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Honeywell</td>
<td></td>
</tr>
<tr>
<td>RAM715-1.1/1-5056</td>
<td>1 roll</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>Honeywell</td>
<td></td>
</tr>
<tr>
<td>FSAD Impact Attenuator</td>
<td>Honeycomb impact attenuator</td>
<td>1 piece</td>
<td>4.00</td>
<td>10.00</td>
<td>Pacifico</td>
<td></td>
</tr>
<tr>
<td>3M DP420</td>
<td>Structural Epoxy adhesive</td>
<td>1 tube</td>
<td>1.00</td>
<td>1.00</td>
<td>Amazon</td>
<td></td>
</tr>
<tr>
<td>1/2 x 0.25&quot; Steel Bar</td>
<td>Cold Rolled 1020</td>
<td>1 ft</td>
<td>1.00</td>
<td>1.00</td>
<td>BSB</td>
<td></td>
</tr>
<tr>
<td>1/2 x 0.25&quot; Steel Bar</td>
<td>Cold Rolled 1020</td>
<td>1 ft</td>
<td>1.00</td>
<td>1.00</td>
<td>BSB</td>
<td></td>
</tr>
<tr>
<td>Time</td>
<td>Man-hours</td>
<td>1400</td>
<td>0.00</td>
<td>0.00</td>
<td>Boeing</td>
<td></td>
</tr>
</tbody>
</table>

**TOTAL** 3615.17
Appendix D - Engine Vibration Code

clear all
clc
format short g

EngineAngle = 0*pi()/180; %[rad] - Angle below horizontal of line from rear to front engine mount

Crank = 254; %[mm] - Linear distance between the rear engine mount and the center of the crank shaft
CrankAngle = -20*pi()/180; %[rad] - Angle below EngineAngle of line from rear engine mount to center of the crank shaft
CrankY = Crank*cos(EngineAngle+CrankAngle); %[mm] - Vertical distance between the rear engine mount and the center of the crank shaft
CrankX = Crank*cos(EngineAngle+CrankAngle); %[mm] - Horizontal distance between the rear engine mount and the center of the crank shaft

Balance = 349;%[mm] - Linear distance between the rear engine mount and the center of the balancing shaft
BalanceAngle = -15*pi()/180; %[rad] - Angle below EngineAngle of line from rear engine mount to center of the balancing shaft
BalanceY = Balance*sin(EngineAngle+BalanceAngle); %[mm] - Vertical distance between the rear engine mount and the center of the balancing shaft
BalanceX = Balance*cos(EngineAngle+BalanceAngle); %[mm] - Horizontal distance between the rear engine mount and the center of the balancing shaft

FrontMount = 394.61; %[mm] - Linear distance between the rear and front engine mounts
FrontMountY = FrontMount*sin(EngineAngle); %[mm] - Vertical distance between the rear engine mount and the front engine mount
FrontMountX = FrontMount*cos(EngineAngle); %[mm] - Horizontal distance between the rear engine mount and the front engine mount

crankoffset = -1.38; %[mm]
balanceoffset = -17.26; %[mm]
stroke = 31.7; %[mm]
l = 90; %[mm]
w = (2500/60)*pi()*2; %[rad/s]
a = 0; %[rad/s^2]
PistonMass = .260; %[kg]
ConRodMass = .318; %[kg]
CrankMass = 2.1; %[kg]
BalanceMass = .2; %[kg]

for i = 1:360
    theta(i) = i;
    rad = i*pi()/180;
    PistonSy(i) = stroke*cos(rad)+(l^2-stroke^2*sin(rad)^2)^(1/2);
\[
\text{PistonVy}(i) = -\text{stroke} \cdot \sin(\text{rad}) \cdot w + (-\text{stroke}^2 + 2 \cdot \sin(\text{rad}) \cdot \cos(\text{rad}) \cdot w)/2 \times (l^2 - \text{stroke}^2 \cdot \sin(\text{rad})^2)^{(1/2)}; \\
\text{PistonAy}(i) = -\text{stroke} \cdot \cos(\text{rad}) \cdot w^2 - \text{stroke} \cdot \sin(\text{rad}) \cdot a + ((2 \times \text{stroke}^2 \cdot \sin(\text{rad})^2))^2 \times (\text{stroke}^2 \cdot \sin(\text{rad})^2 \cdot w^2 + 2 \times \text{stroke}^2 \cdot \sin(\text{rad}) \cdot \cos(\text{rad}) \cdot w^2 - 2 \times \text{stroke}^2 \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot a) + 2 \times \text{stroke}^2 \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot w^2 -(2 \times \text{stroke}^2 \cdot \cos(\text{rad}) \cdot \sin(\text{rad}) \cdot w)/(l^2 - \text{stroke}^2 \cdot \sin(\text{rad})^2)^{(1/2)}) \\
\text{PistonF}(i) = \text{PistonAy}(i) \times \text{PistonMass}; \\
\text{PistonFy}(i) = \text{PistonF}(i) \cdot \sin(\text{EngineAngle} + (\pi/2)); \\
\text{PistonFx}(i) = \text{PistonF}(i) \cdot \cos(\text{EngineAngle} + (\pi/2)); \\
\]

\[
\text{ConRodSy}(i) = \text{stroke} \cdot \cos(\text{rad}) + (1/3) \times (l^2 - \text{stroke}^2 \cdot \sin(\text{rad})^2)^{(1/2)}; \\
\text{ConRodSx}(i) = (2/3) \times \text{stroke} \cdot \sin(\text{rad}); \\
\text{ConRodVy}(i) = -\text{stroke} \cdot \sin(\text{rad}) \cdot w + (1/3) \times (-\text{stroke}^2 + 2 \times \sin(\text{rad}) \cdot \cos(\text{rad}) \cdot w)/2 \times (l^2 - \text{stroke}^2 \cdot \sin(\text{rad})^2)^{(1/2)}; \\
\text{ConRodVx}(i) = (2/3) \times \text{stroke} \cdot \cos(\text{rad}) \cdot w; \\
\text{ConRodAy}(i) = -\text{stroke} \cdot \cos(\text{rad}) \cdot w^2 - 2 \times \text{stroke} \cdot \sin(\text{rad}) \cdot a + (1/3) \times (2 \times \text{stroke}^2 \cdot \sin(\text{rad})^2)^2 \times (-\text{stroke}^2 \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot w^2 + 2 \times \text{stroke}^2 \cdot \sin(\text{rad})^2 \cdot \cos(\text{rad}) \cdot w^2 - 2 \times \text{stroke}^2 \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot a) + 2 \times \text{stroke}^2 \cdot \sin(\text{rad})^2 \cdot \cos(\text{rad}) \cdot \cos(\text{rad}) \cdot w^2 -(2 \times \text{stroke}^2 \cdot \cos(\text{rad}) \cdot \sin(\text{rad}) \cdot w)/(l^2 - \text{stroke}^2 \cdot \sin(\text{rad})^2)^{(1/2)}; \\
\text{ConRodAx}(i) = -(2/3) \times \text{stroke} \cdot \sin(\text{rad}) \cdot w^2 + (2/3) \times \text{stroke} \cdot \cos(\text{rad}) \cdot a; \\
\text{ConRodF}(i) = \sqrt{\text{ConRodAy}(i)^2 + \text{ConRodAx}(i)^2} \times \text{ConRodMass}; \\
\text{if ConRodAx}(i) \leq 0 \\
\text{ConRodTheta}(i) = \text{atan(ConRodAy}(i)/\text{ConRodAx}(i))+\pi(); \\
\text{else} \\
\text{ConRodTheta}(i) = \text{atan(ConRodAy}(i)/\text{ConRodAx}(i)); \\
\text{end} \\
\text{ConRodFy}(i) = \text{ConRodF}(i) \cdot \sin(\text{ConRodTheta}(i) + \text{EngineAngle}); \\
\text{ConRodFx}(i) = \text{ConRodF}(i) \cdot \cos(\text{ConRodTheta}(i) + \text{EngineAngle}); \\
\]

\[
\text{CrankSy}(i) = \text{crankoffset} \cdot \cos(\text{rad}); \\
\text{CrankSx}(i) = \text{crankoffset} \cdot \sin(\text{rad}); \\
\text{CrankVy}(i) = -\text{crankoffset} \cdot \sin(\text{rad}) \cdot w; \\
\text{CrankVx}(i) = \text{crankoffset} \cdot \cos(\text{rad}) \cdot w; \\
\text{CrankAy}(i) = -\text{crankoffset} \cdot \cos(\text{rad}) \cdot w^2 - \text{crankoffset} \cdot \sin(\text{rad}) \cdot a; \\
\text{CrankAx}(i) = -\text{crankoffset} \cdot \sin(\text{rad}) \cdot w^2 + \text{crankoffset} \cdot \cos(\text{rad}) \cdot a; \\
\text{CrankF}(i) = \sqrt{\text{CrankAy}(i)^2 + \text{CrankAx}(i)^2} \times \text{CrankMass}; \\
\text{if CrankAx}(i) \leq 0 \\
\text{CrankTheta}(i) = \text{atan(CrankAy}(i)/\text{CrankAx}(i))+\pi(); \\
\text{else} \\
\text{CrankTheta}(i) = \text{atan(CrankAy}(i)/\text{CrankAx}(i)); \\
\text{end} \\
\text{CrankFy}(i) = \text{CrankF}(i) \cdot \sin(\text{CrankTheta}(i) + \text{EngineAngle}); \\
\text{CrankFx}(i) = \text{CrankF}(i) \cdot \cos(\text{CrankTheta}(i) + \text{EngineAngle}); \\
\]

\[
\text{BalanceSy}(i) = \text{balanceoffset} \cdot \cos(\text{rad}); \\
\text{BalanceSx}(i) = \text{balanceoffset} \cdot \sin(\text{rad}); \\
\text{BalanceVy}(i) = -\text{balanceoffset} \cdot \sin(\text{rad}) \cdot w; \\
\text{BalanceVx}(i) = \text{balanceoffset} \cdot \cos(\text{rad}) \cdot w; \\
\]
BalanceAy(i) = -balanceoffset*cos(rad)*w^2-balanceoffset*sin(rad)*a;
BalanceAx(i) = -balanceoffset*sin(rad)*w+balanceoffset*cos(rad)*a;
BalanceF(i) = sqrt(BalanceAy(i)^2+BalanceAx(i)^2)*BalanceMass;
if BalanceAx(i)<=0
    BalanceTheta(i) = atan(BalanceAy(i)/BalanceAx(i))+pi();
else
    BalanceTheta(i) = atan(BalanceAy(i)/BalanceAx(i));
end
BalanceFy(i) = BalanceF(i)*sin(BalanceTheta(i)+EngineAngle);
BalanceFx(i) = BalanceF(i)*cos(BalanceTheta(i)+EngineAngle);

if FrontMountX==0
    FrontEngineMountForcey(i) = (-PistonFy(i)+ConRodFy(i)+CrankFy(i)+BalanceFy(i))/2)/1000;
else
    FrontEngineMountForcey(i) = (-
        (CrankX*PistonFy(i)+CrankX*ConRodFy(i)+CrankX*CrankFy(i)+BalanceX*BalanceFy(i))/FrontMountX)/1000;
end
if FrontMountY==0
    FrontEngineMountForcex(i) = (-PistonFx(i)+ConRodFx(i)+CrankFx(i)+BalanceFx(i))/2)/1000;
else
    FrontEngineMountForcex(i) = (-
        (CrankY*PistonFx(i)+CrankY*ConRodFx(i)+CrankY*CrankFx(i)+BalanceY*BalanceFx(i))/FrontMountY)/1000;
end

RearEngineMountForcey(i) = (-PistonFy(i)+ConRodFy(i)+CrankFy(i)+BalanceFy(i))/1000-FrontEngineMountForcey(i);
RearEngineMountForcex(i) = (-PistonFx(i)+ConRodFx(i)+CrankFx(i)+BalanceFx(i))/1000-FrontEngineMountForcex(i);
end
figure
plot(FrontEngineMountForcex,FrontEngineMountForcey,'k',RearEngineMountForcex,RearEngineMountForcey,':');
title('Engine Mount Forces for One Crankshaft Revolution')
xlabel('Force in the X Direction [N]')
ylabel('Force in the -Z Direction [N]')
legend('Front Engine Mount Force','Rear Engine Mount Force')