Lightweight Racing Suspension

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Judah Ari-Gur

Dr. Kujawski, Dr. Keil

Isaac Burdick, Colin Haynes, Nicholas Patzer

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Lightweight Racing Suspension

Senior Design Project

Group 6
Isaac Burdick
Colin Haynes
Nicholas Patzer

Western Michigan University

Reviewed By: ____________  Approved By: ____________

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Work Completed from January 2019 - December 2019

DISCLAIMER

This project was completed by senior level engineering students at Western Michigan University. This is a summary of the completed work for the scope of the design project. The designs are specifically made for use in Western Michigan’s 2020 Sunseeker Solar Car. No recommendations are made for usage of any components outside of this intended purpose. Western Michigan University, its faculty, and students take no responsibility for the use of any components outside of the intended purpose. People or organisations who choose to use any components outside of the intended purpose do so at their own risk.
Abstract

The purpose of this senior design project is to design and optimize the new suspension system for Western Michigan University’s 2020 solar car. The current solar car was 43% heavier than the average solar car, and was the heaviest single occupant vehicle competing in the American Solar Challenge. The scope of the project is to reduce the weight as much as possible from the old 2016 suspension. The new suspension weighed 34.91 lbs less than the 2016 suspension system while maintaining a minimum factor of safety greater than 1.5, compliant with American Solar Challenge regulations. The design consists of a leading double control arm suspension in the front and a trailing arm suspension in the rear. A bell crank shock is used for both systems. All components were mounted to a three layer M10 carbon fiber chassis board which was selected based on material testing.
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Introduction

Background

Western Michigan’s Sunseeker solar car team is an engineering Registered Student Organization that designs, builds, and races a solar car in the American Solar Challenge (ASC) against other universities from around the world. The ASC is a multi-day competition where teams compete in a cross country rally event, ranging from 1500 to 2000 miles taking place across multiple states. The solar cars undergo a scrutineering process to qualify since the solar cars will race on public roads during the competition. During these races, it is critical that the vehicle has been designed properly to ensure safe operation for drivers and bystanders. The goal of the race is to create the most efficient vehicles that can travel the furthest distance in the shortest amount of time on only solar power.

Objective of Work

The objective of this project is to design a lightweight suspension system for the 2020 Sunseeker solar car. The focus will be on weight reduction without compromising safety, while still meeting ASC requirements. Weight reduction is based on the previous design and the scope involves everything from the mounting points to the upright. Braking systems, steering system, the hub, wheel, and motor are not part of this projects design. Chapters 4 and 5 of Tune to Win acted as a guide for this design process and can be found in the references.
Definition of Problem

The Sunseeker solar car team is constructing a new solar car for the 2020 race that must comply with the current rules and regulations set by ASC, which can be found in the references. In the 2018 ASC competition, Western Michigan’s solar car weighed the most out of any single occupant vehicles at 316 kg, or 876 lbs, fully loaded with battery and driver. The suspension accounts for about 130 lbs, approximately 19.3% of the total weight of the vehicle. This represents a large portion of the vehicle weight and reducing it will improve the overall efficiency of the solar car.

This suspension system will be used on track and public roads, so the safety of the driver must be considered. ASC has a series of dynamic tests required for racing eligibility which includes slalom, brake, U-turn, and figure eight. ASC also designated loads for worse case scenarios that the suspension must be able to handle (10.2.B Ref. 1). The systems in the car will be compliant with ASC codes by performing under the specified loading conditions.

Scope/Limit

While evaluating different solutions to the suspension it is important to look at what aspects are being considered. To keep the project focused on the functionality of the suspension system, the design work is scoped to start at the upright and move up to the mount points on the composite chassis. This eliminates the wheel hub assembly, aspects of the steering arm and aspects of the brake system. The design of the steering arm and brake system will be determined by a separate team and be incorporated into the design at a later date. A list of items that were designed include lower control arms, upper control arms, uprights, shock mounts, and mounting
positions and brackets. The designed components have desirable suspension geometry and articulation while maintaining a minimum safety factor of 1.5 for an estimated 700lbs fully weighted car. The last limitation comes from the packaging of the design. The aeroshell encompases the suspension to reduce the drag on the car, so the design will have to fit within the fairing and be able to articulate and turn fully within this design space.

Specification

Direct specifications for the design come from the regulations specified by ASC to create safe and functional designs for the competition. Critical regulations include the forces and load conditions the suspension must withstand. Regulation 10.2.B states the design must be able to withstand the loads and forces applied by the vehicles mass, speed capability, and braking potential. These are further defined in the regulations under appendix D.1 to be a 1G turn, 2 G bump, and a 1G braking force applied to the tire patch where 1G is defined as the weight of the vehicle on each wheel. Further expanding upon this regulation the braking force will be split amongst the front and rear wheels assuming 25% weight transfer to the front wheels. That makes the design account for a 1.25G brake force on the front wheels and 0.75G brake force on the rear wheels. Regulation 10.9 defines the various dynamic tests the vehicle must be able to perform and be considered during the suspension design. In regulation 9.3, the vehicle must have a minimum of 50 mm of ground clearance while fully loaded. Regulation 10.1.B calls out clearance needed between moving parts of the suspension and wheels to prevent interference during full range of motion. Regulation 10.2.A defines the car must have four wheels that are logically spaced about the vehicles centerline. All of these requirements must be met in order for the solar car to compete in the American Solar Challenge which can be found in Reference 1.
Geometry

Steering Radius Geometry

The placement of the tires for the design were determined by two constraining requirements. The first being to comply with ASC regulation 10.7.C which calls out a minimum turning diameter of 16 meters. The other constraining factor is the aeroshell of the car which will enclose the suspension system through its full range of motion. To allow some compliance the system was designed to make a U-turn in 600 inches or 15.24 meters. Looking at Figure 1, the direction of the car is toward the bottom of the page while turning to the drivers left. The 600 inch diameter turning radius is defined by a center point located along the line of the rear axis while intersecting with the center point of the front right tire or outside wheel. The circle represents the path the tire follows when turning a fixed angle off center. The inner circle seen in Figure 1 represents the path the inner tire follows and is defined by the same center point as the outside wheel path and intersecting the centerpoint of the inside tire. To maintain a uniform steering arc the inside tire must turn further than the outside to follow a smaller arc of rotation. Based on this model the inside tire must turn 17.69 degrees while the outside tire must turn 15.17 degrees. The steering is assumed to be symmetrical meaning each tire will turn 17.69 degrees toward the outside of the car and 15.17 degrees toward the inside of the vehicle. This turning range is shown in Figure 2 as well as the placement of the tires within the aeroshell.
Figure 1: Turning Radius Geometry

Figure 2: Tire Patch Top View
Upright Geometry and Design

After determining the tire patch location within the aeroshell the upright geometry could be determined for design. Figure 3 provides a frontal 2D image of the driver's left front tire. The black colored rectangle represents the outside dimensions of the tire and rim that will be used in the final design, the gray section represents the hub surface, and the blue rectangle represents the axle connecting the hub and tire to the rest of the suspension. The light blue background represents the mounting surface and a 2D representation of the space on each side of the tire as seen from the front. It should be noted that the wheel and rim used in this design is concave allowing the design to sit inside of the rim without any interference issues. The connection points to the control arms were chosen along a straight line, 14.6 degrees from vertical. This setup allows the tire to turn from the center of the tire patch.

To account for other components, the placement of the mount points vertically on the mounting board were desired to stay within the diameter of the tire in the resting position. It was also desired to maximize vertical height between the control arms to allow space for the shock placement and articulation. With this in mind, the lower control arm connects to the upright 1.7 inches off the centerline following the diagonal line placing the joint just inside of the rim below the axle to allow for clearance between the rim and control arm during articulation. To maximize the height between the control arms, the upper joint is determined at the edge of the rim sitting outside at 5.09 inches from the center point of the tire. This can be seen in Figure 3 as the red outline representing the angle of the tire patch and the position of the upper and lower control arm mounts.
Using the determined geometry the upright was 3D modeled to mate to the axle and hub and provide structure for the joints connecting to the upper and lower control arms. Figure 4 shows an isometric representation of the upright design. The design also includes a mount for the steering link to attach to. The steering mount is located above the upper control arm mount.

Figure 3: Upright Geometry Connection Points
The total weight of the upright is 3.57 lbs. Compared to the weight of the 2016 upright, the new upright is 12.1% lighter.

Control Arms

Front Lower Control Arm

The Lower Front Control Arm, pictured in Figure 5, is designed to mount behind the wheel to provide support to the assembly. To fit within the turning regulation 10.7.C set by ASC, the control arm has to allow for a maximum turn inwards of 17.69 degrees from a 21.5 inch diameter tire. The center of the hub to the mounting wall is 16.5in. All suspension designs are created with a 0.5 inch between the end of the part and rod end and 1.25 inch between the
bracket and the mounting board. The part is 14.5 in long from the connection point from the upright to the end of the part. Ideally, the shock mount would be as close to the upright connection to reduce moments created from the loading conditions. Due to the design and clearance needed for the turning radius, the shock mount was moved out away from the upright connection point in such a way so the two-force member coming from the shock is in a straight line to the lower control arm. The applied forces and the shock reaction create large moments on the arm. To combat this, a rib was added to increase the rigidity at the corner near the shock mount. The parts thickness was also increased to resist these large moments and forces.

![Figure 5: Lower Front Control Arm](image)

The total weight of this component is 2.38 lbs. This is 3% heavier than the current suspension on the 2016 solar car. Due to the large force concentration and need for a shock mount, the total weight could not be reduced for this part.
Upper Front Control Arm

The upper front control arm seen in Figure 6 is designed to mount to the upright in the same way as the lower control arm. The upper control arm receives far less forces than the lower control arm as seen by the calculations in Appendix C. For this reason, the control arm is lighter than all of the other control arms. The control arm still has to account for the working area of the tire to prevent collisions between the components. The upper control arm initially saw a large stress concentration when the control arm split to attach to the chassis. To relieve the stress seen, a triangular section was extruded from the main cylinder, allowing the stress to be distributed over a greater distance. Within this new extruded section, some material was removed in order to save weight. The material that was removed saw very little of the stress, allowing for weight savings without compromising the safety of the part. The weight of this component is 0.86 lbs. This is a 48.6% reduction in weight from the 2016 upper control arm.

Figure 6: Upper Front Control Arm.
A full assembly of the front suspension is shown in Figure 7. This full assembly also includes the shock design and mounting brackets which are discussed later in the report.

![Front Suspension Assembly](image)

**Figure 7: Front Suspension Assembly**

Rear Control Arm

The rear suspension system is different than the front since it does not need to turn. For this reason, the upright design was removed in favor of a rear trailing arm with shock design. This design greatly improves weight savings as it does not include two other components. The rear control arm mounts directly to the hub or motor. The rear control arm is shown below in Figure 8.
The rear control arm sees similar loading conditions to the front suspension. However, this component has the loads distributed through the six bolt holes that connect to the motor housing. This is the bolt pattern called out by the motor manufacturer that the solar car team purchased for this new car. The rear control arm weighs 3.85 lbs. The rear suspension, compared to the 2016 solar car, includes the upright, lower and upper A-arms. For this reason, the total weight for the rear suspension is reduced by 52%.

Shock Design

This design uses a bell crank system that keeps the shock away from the wheel assembly and translates the motion of the control arms back to where the shock rests near the wall. The design is more complex and entails more parts than traditional systems, but solves all the issues that occur with a leading-trailing suspension. The shock was already selected for this system, so
the design consists mainly of the bell crank analysis and motion studies of the entire system. Front and rear assemblies are similar in concept, but their geometries are slightly different, as the rear mounting board is at a 61.8° angle from horizontal. The assembly of the front and back designs are pictured in Figure 9 and Figure 10 respectively.

The bell crank mechanism transmits the vertical articulation of the wheel into a two-force member. The force from the two-force member transmits the force into the bell crank, which
transmits the force into the shock itself. The geometry of this design distributes the forces across the whole system and allowing for a smaller shock to be chosen. The two-force member and bellcrank of this system are made of steel as aluminum parts would have been too bulky to fit in the given area between the control arms.

The weight of each component for the bell crank was measured for weight comparison. The shock was taken out of weight measurements as the same shock is used in both systems. The previous design weighed .72 lbs for all modeled parts. The new front design weighs 1.39 lbs, and the rear weighs 1.31 lbs. This slight increase in weight is the trade-off for needing a bell crank system as opposed to a traditional system.

Mounting

Requirements

The suspension systems are mounted on the forward and backward walls of the composite chassis. As seen in Figure 11, the front suspension mounts to the red board and the rear suspension mounts to the slanted blue board. The suspension system mounts to these boards through the control arms and shocks with brackets. The material selection for these boards can be seen in Composite Testing Results, and the forces applied to these boards at each bracket can be seen in Appendix C. Bringing these two components together, it is assumed all the forces calculated at the brackets are applied at the center of the board. This gave a worse case scenario for deflection, and is the method of calculating the maximum deflection of the boards.
Brackets

The two brackets shown below cover all 10 mounting points on the front and back. The worst load cases are taken for each bracket type and shown in Figure 12 and Figure 13. These designs are changed from previous iterations to make all bolts easily accessible, which was one of the design criteria for this project. Currently these are made of 6061-T6 aluminum with a total weight gain of .09 lbs. This is a relatively small weight increase that was needed to be able to service the vehicle.
Hardware

To design a fully functional suspension system, hardware had to be chosen to allow the correct range of motion and also withstand the loading conditions. To focus on the suspension components, 3/8 inch, grade 5 bolts are used for all the connection points of the systems. To prevent bolts from loosening and meet ASC requirements, steel lock nuts are used for fastening the bolts. Spherical bearings are used for the connection points between the upright and the
control arms in the front suspension. High misalignment bearings from Aurora Bearing Company were selected, model number HAB-6T, due to having a $\frac{3}{8}$ in inner diameter for bolts. For connection points from the front control arms to the chassis, 3/8 inch spherical rod ends are used. The model chosen for design was Aurora Bearing Company’s high performance rod ends, PRM-6T. The rear control arms connect to the chassis using 1/2 inch rod ends to accommodate the higher loading conditions. The model is PRM-8T from Aurora Bearing Company’s high performance rod end options. Calculations for rod end shear loads are calculated in Appendix C.

FEA Testing

Constraints

Testing for the upright, mounts, and control arms were done through ANSYS Static Structural program. Analyzing the suspension components in this way was the most cost and time effective way to determine the effectiveness of the parts. To create an accurate simulation, the parts are constrained to resemble the actual motion desired by the suspension.

To accurately model the three control arms, spherical joints are used in the rod ends at the mounting points. These joints allowed full rotation around all axes and no translation in any coordinate direction. This constrains the inner faces of the rod end to act in the same way the bolted connection would see on the car, providing accurate results from the FEA software. The joint can be seen in Figure 14 below.
The upright is modeled with the lower control arm mount as a spherical connection allowing rotation in all axes directions, but not movement in the coordinate directions. The upper control arm mount was modeled as a spherical connection allowing all rotations as well as translation in the Y direction, no movement in the X or Z. The steering mount was treated as rotational joint only, allowing the part to rotate about the Y axis, and resisting motion in all other directions.

Other constraints for the control arm include adding a fixed connection between the rod ends and the control arms. This accurately reflects how the rod ends will be threaded into the control arms allowing for no rotation or translation between the two parts. Additionally, the bolts used to connect the shock to the rear control arm and front lower control arm are treated as fixed-pin connections this allows the shock to rotate around the bolt but not translate or rotate in another direction. The shock brackets are constrained using two pin connections at the bolt holes.
for the bracket to restrict the rotation and simulate a semi-static position. This configuration allows the part to develop stresses in the control arm and shock mounts.

Shock

The front lower control arm and rear control arm both are designed to accommodate a shock dampener system. For this reason, a shock was modeled within ANSYS. A connection is created between the shock mounts and ground as seen in Figure 15. Constraining the shock in this way allows the force to be applied at a midpoint between the two plates, accurately depicting how the shock mount would receive the forces.

![Figure 15: Shock Mount Constraint](image)

The ground connection of the shock is placed at the angle the two force members would be acting in. The spring constants are calculated in Appendix C. The springs will allow the control arms to move the wheel up to an inch vertically, which is desired by the solar car team’s requirements for the design. The shock helps create a more realistic analysis of the control arms,
and allows the suspension to be modeled with ball joints instead of being modeled as a cantilever beam.

Material Assignment

The upright and all of the control arms are made out of aluminum 6061-T6. The brackets that mount to the chassis are also made out of the same grade aluminum. The rod ends, bolts and shock mounts are all modeled as high strength steel. The materials are chosen based on their relative strengths and weight.

For the rear control arm, the rod ends are modeled as an indestructible material. The reaction forces on the rod ends are calculated in Appendix C to determine the appropriate sizing. This calculation determined that a 0.5 inch rod end is needed to withstand the forces with a factor of safety of 1.7. In ANSYS software, accurate representations of the rod ends were difficult to acquire, so the decision was made to adjust the material properties of the rod ends to create a material that does not yield. This allows the simulation to calculate the maximum stresses and factor of safety focusing on the designed rear control arm.

Mesh

For all parts, the standard mesh size was set to 0.197 in. The refinement feature was used for critical stress areas to make a finer mesh because facing was small or more precise measurements were needed.
Applied forces

Forces are applied to the tire patch and then distributed throughout the suspension system at the connection points. The force seen while braking will be in the X direction, the force seen from the vertical bump will be in the Y direction, and an inward force from turning is along the Z direction. Moments about each axis follow the previous coordinate system. For the upright, forces and moments are applied through the axle connection showing how the rigid connection would allow forces to be distributed by the material. The force at the tire patch created both forces and moments for this member. In the simulation, the upright sees 350 lbs of force in the X and Y direction, and 175 lbs in the Z direction. The upright also experiences 1608 lbf-in, -273 lbf-in, and 3762.5 lbf-in moments in the X, Y, and Z directions respectively. The front control arms have a force applied in the bolt hole that join the control arms to the upright. Since a spherical bearing is used between the two components, no moment is allowed to develop, leaving just the resultant forces along the X, Y, and Z axis. The front upper control arm only sees 41.2 lbs and 175.35 lbs in the X and Z direction respectively. There is no force in the Y direction because it is assumed that the shock counteracts all of the vertical forces acting on the system. The lower control arm sees 217 lbs, 350lbs, and 459.7 lbs in the X, Y, and Z directions respectively. The force calculations for each control can be seen in Appendix C.
As seen above in Figure 16, the forces for the rear control arm were applied through 6 bolt holes. The motor requires this bolt array for mounting. For the rear control arm, a remote force is used to simulate the forces and moments seen by the rear control arm. The force is set on the tire patch, -10.75 and 3 inches in the Y and Z directions respectively from the middle of the remote force origin. The applied force is 131.25lbs, 350lbs and 175lbs in the X, Y, and Z directions respectively.

Composite Testing

Purpose

Ideally, no deflection will occur in the board during loading conditions. Having deflections in the board can cause the toe angles to shift, lower efficiency, and in extreme cases,
can cause problems in the geometry of the suspension. Based on previous car design limits, maximum deflection for optimal performance of the car is 0.25 inches. Using the three point bend testing data, the stress strain curves of different potential materials were obtained. These graphs can be seen in the Testing Results section. By testing these materials, the best composite board can be selected that meets Sunseeker’s parameters as well as ASC regulations.

Creating Samples

Samples were created and tested in accordance with ASTM standard D7264/D7264M. The sample boards were sized to 9.6 inches in length, 2.75 inches in width, and 0.5 inches in thickness. This follows the procedures ratio of 16:1 and fits in the test fixture, pictured in Figure 17. Table 1 describes the differences between the test samples and Figure 18 shows them.

Figure 17: Three Point Bend Fixture
<table>
<thead>
<tr>
<th>Sample</th>
<th>Manufacturer</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>M10 ½” 2 Layer</td>
<td>Prepreg, Hand Assembled</td>
<td>Nomex Core with 2 Layers of M10 Prepreg, one at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>Double Resin M10 ½” 2 Layer</td>
<td>Prepreg, Hand Assembled</td>
<td>Nomex Core with 2 Layers of M10 Prepreg - extra resin, one at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>M10 ½” 3 Layer</td>
<td>Prepreg, Hand Assembled</td>
<td>Nomex Core with 3 Layers of M10 Prepreg, two at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>M10 Expanded Nomex Core ½”</td>
<td>Prepreg, Hand Assembled</td>
<td>Nomex Expanded Core with 2 Layers of M10 Prepreg, one at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>Plascore Nomex Core ½”</td>
<td>Plascore</td>
<td>Nomex Core with 2 Layers of M10 Prepreg, one at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>Plascore Plastic Core ½”</td>
<td>Plascore</td>
<td>Plastic Core with fiberglass covers</td>
</tr>
<tr>
<td>Plascore Aluminum Core ½”</td>
<td>Plascore</td>
<td>Aluminum Core with fiberglass covers</td>
</tr>
<tr>
<td>M10 ¼” 2 Layer</td>
<td>Prepreg, Hand Assembled</td>
<td>Nomex Core with 2 Layers of M10 Prepreg, one at 0° angle, and one at 45° angle</td>
</tr>
<tr>
<td>Plascore Nomex Core ¼”</td>
<td>Plascore</td>
<td>Nomex Core with fiberglass covers</td>
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</table>
The procedure provides stress and strain equations. This gives comparable parameters to determine the best material to be used, regardless of core thickness.

Test Procedure

With the apparatus used, Procedure A in the D7264/D7264M is the test procedure used. Sample materials were centered on the lower fixture, which raised and caused the upper fixture to put force on the board until it failed. Figure 19 shows a composite board after the test was complete. Time, force, and vertical displacement were all measured during the test. The speed of the testing was performed at 1mm/sec with the exception of M10 2 layer which was performed at 0.1mm/second. This difference was noted and was determined to not have an affect on the testing results.
Applying the equations from Appendix C, the maximum stress being applied to the board, and corresponding strain data was calculated. These two parameters are used to form a stress strain curve, then three random points before yielding are chosen and used to calculate the slope which gives us E (young’s modulus). Using this E, the amount of deflection was calculated for a worst case scenario. Maximum loading conditions were applied in the center of the board, a similar situation to the tests. Materials were eliminated due to not being able to withstand the force applied, having a deflection larger than the maximum deflection. The weights of those that remained to determine the best option for mounting.
Results

FEA Results

Upright

The upright seen in Figure 20 has its applied forces and moments at the hub connection. The steering mount allows for rotation around the Y axis only. The mount to the upper and lower control arms both allow for free rotation on all axis, but the upper mount is free to travel in the Y direction while the lower is not.

Figure 20: Stress Analysis of Upright
Figure 21: Safety Factor of Upright

As seen by Figure 21, the lowest factor of safety for this part is 1.96. The parts that have lower factors of safety are the same regions of high stress concentration.

Upper Control Arm

The forces for the upper control arm are applied to the connection point to the upright. The rod ends are constrained to allow rotation around all axes but have no translations.
In Figure 23, the lowest factor of safety for the part is 4.83 and occurs where control arm splits. This area also has a higher stress than the rest of the control arm as seen in Figure 22.
Lower Front Control Arm

The lower front control arm seen in Figure 24 is constrained in the same way as the upper control arm. The shock mount is constrained by pin connections in the bolt holes. This prevents the shock mount from moving relative to the rest of the control arm. The shock is applied between the shock mount and is positioned at the angle the shock will be acting.

Figure 24: Stress Analysis of Lower Control Arm
Figure 25: Safety Factor of Lower Control Arm

Figure 25 shows the lowest factor of safety for the lower control arm is 1.52 and occurs at the rib connection. This part has an even distribution of stress throughout with an average factor of 2.0.

Rear Control Arm

The rear control arm is constrained by the rod ends where the connection can rotate freely in any direction but not translate. The bracket for the shock is bolted into both sides of the control arm. The forces for the rear control arm are applied to the six bolt holes used to connect the motor to the control arm. Figure 26 shows the stress distribution throughout the part.
In figure 27, the lowest factor of safety is 1.67 and this occurs in the inner mount point for the shock mount. Due to the large forces and moments, there is a large stress concentration throughout the neck of the rear control arm.
Shock Assembly

Figures 28 and 29 are the stress and safety factor plots respectively for the crank of the shock assembly. The part is made of plain carbon steel and all connections are cylindrical supports, which are only able to rotate in the Y direction.

Figure 28: Stress Analysis of Triangle

Figure 29: Safety Factor of Triangle
As can be seen from Figure 28, the stresses are concentrated between two back connections, and in Figure 29, the lowest factor of safety is 4.75. This part is designed to be small in packaging to be able to operate between the control arms of the system.

Brackets

The brackets are able to perform well above the maximum loading condition. Since the factor of safety of the brackets is large compared to the safety factors for other components, this ensures a failure will not occur at these connections. This keeps the driver safe, as the mount points are near the driver compartment. The brackets designed are slightly heavier than what was designed in the past to prevent bolt collision during maintenance. This slight increase in weight compared to safety factor achieved lends itself to a different material options, for future designs.

The worst case scenario brackets are shown in Figures 30, 31, and 32. Figure 30 shows the stress distribution. It is concentrated at the base of the two mounting heads. Figure 31 and 32 show the factor of safety on both sides of the bracket. A factor of safety of 5.13 was achieved under given load conditions, which can be found in Appendix C.
Figure 30: Stress Analysis of Bracket (Worst Case)

Figure 31: Stress Analysis of Bracket (Worst Case)
Composite Testing Results

Of the 10 samples tested, three composite boards passed the strength requirements to be considered. Double resin M10 sample, the three layer M10 sample, and the metal core sample all were able to take the maximum forces without breaking or yielding. Graph 1 shows the stress strain curve for these materials. Looking at the data comparison between them, the metal core is more malleable, deflecting more and absorbing more energy. In comparison, the double resin and 3 layer are stiff deflecting minimally.
The desired deflection of the board is under a 0.25 inches at max loading to ensure the suspension operates normally under loading conditions. Calculating the maximum deflection for the worst case scenario, 218lbf of loading was applied across the middle of a 21”x 22” of each material. As seen in Table 2, the two lowest deflection materials are double resin and 3 layer composite boards.
Comparing these two, the double resin is over double the weight of the 3 layer, making 3 layer the best option for this design. Double Resin and 3 layer both had under the required amount of deflection in the worst case which is shown in Table 2. Comparing weights of the two boards, 3 layer would be the best option for deflection and weight optimization which the table shows below.

<table>
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<th>Points</th>
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<th>Max Deflection</th>
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<td>Stress (psi)</td>
<td>Strain (in/in)</td>
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<tr>
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<td>0.00091</td>
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<td></td>
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<td></td>
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<td>2344</td>
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<td></td>
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<td></td>
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<td></td>
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<td>0.01131</td>
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<td>1336</td>
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Weight Reduction

One of the key criteria for the design on the 2020 suspension is to reduce the weight of the suspension while maintaining a minimum factor of safety of 1.5. The values seen below are only the components that were designed for the project. For example, the table does not include the weight of the wheel or hub. As seen in Table 4, the total weight of the suspension is reduced by 44%. Most of the weight savings came from the rear suspension. The rear suspension only consists of the trailing arm, reducing the weight of the rear assembly by 52%. The trailing arm design can only be used in the rear since it does not need to turn.

<table>
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<th>Material</th>
<th>$\sigma_{\text{max}}$ (psi)</th>
<th>$\sigma_{\text{max}}$ (ksi)</th>
<th>Material weight (lbf)</th>
<th>Ratio of $\sigma_{\text{max}}$, ksi to Material Weight, lbs</th>
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<td>0.077</td>
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<td>4.83</td>
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<td>Double resin 10</td>
<td>10055</td>
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<td>3 Layers M10</td>
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Table 4: Weight Comparison Table

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<tr>
<td>Upright</td>
<td>4.06</td>
<td>3.57</td>
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<tr>
<td>Upper Control Arm</td>
<td>1.66</td>
<td>0.86</td>
<td>48%</td>
</tr>
<tr>
<td>Lower Control arm</td>
<td>2.31</td>
<td>2.38</td>
<td>-3%</td>
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<tr>
<td>Rear Suspension Components</td>
<td>8.04</td>
<td>3.85</td>
<td>52%</td>
</tr>
<tr>
<td>Shock system Front</td>
<td>1.34</td>
<td>1.39</td>
<td>-4%</td>
</tr>
<tr>
<td>Shock system Rear</td>
<td>1.34</td>
<td>1.31</td>
<td>2%</td>
</tr>
<tr>
<td>Bracket</td>
<td>0.11</td>
<td>0.2</td>
<td>-88%</td>
</tr>
<tr>
<td>All Bolts</td>
<td>1.70</td>
<td>1.70</td>
<td>0%</td>
</tr>
<tr>
<td>Total weight in Front</td>
<td>19.75</td>
<td>14.95</td>
<td>24%</td>
</tr>
<tr>
<td>Total weight in Rear</td>
<td>19.75</td>
<td>7.09</td>
<td>64%</td>
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<tr>
<td>Total weight to car</td>
<td>79.00</td>
<td>44.09</td>
<td>44%</td>
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</table>

Despite large reductions in weight, it is hard to draw conclusions based on how different the two designs are. However, the total weight of the suspension can be compared to the total weight of the car. The 2016 suspension was 9% of the weight of the old 877 lb car while the new suspension is only 6% of the total weight of the 700 lb new car.

Conclusion

Overall, the new suspension design reduces the total weight of the suspension by 34.95 lbs. The new suspension system is 44% lighter than the suspension for the previous generation solar car. The lowest factor of safety for any member is located on the rib of the lower front control arm with a value of 1.52. Since the design goal is to achieve a factor of safety of 1.5 or greater and reduce the weight of the total system, the team satisfied all requirements given for the suspension system. Since the factor of safety for the upright, upper control arm, and rear
control arm were all above the minimum safety factor of 1.5, some additional weight savings
could be achieved in this area.
Acknowledgements

We would like to thank the following groups and individuals for aiding in the completion of this suspension. The success of this design was made possible with their guidance, input, and expertise.

Dr. Kujawski

Dr. Keil

Suraj Nikram

WMU Solar Car Team
   Kyle Lyman: President
   Cortney York: Mechanical Team Lead
   Solar Car Alumni

PolyTech Montreal, Esteban Solar Car Team
References


Appendix A: Materials Testing Data

Stress Strain Curves for all Three Point Bend Testing

This is a summary of all of the stress strain data from the material testing.
Prepreg Composite Stress Strain Curves

This graph shows the stress strain data for all prepreg carbon fiber samples.

Purchased Composite Stress Strain Curves

This graph shows the stress strain data for all purchased materials.
Quarter Inch Sample Stress Strain Curves

This graph shows the stress strain relationship of the two quarter inch samples tested.
Appendix B: FEA Data

This section is a summary of all Constraints, Forces, and Results from ANSYS not used in the body of the report.

Upright

<table>
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<td>Detature Size</td>
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<td>Behavior</td>
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</table>

Mesh Data for Upright

Sizing for mesh applied to the upright during analysis.

Displacement Analysis for Upright

Deformation of upright under loading conditions.
Turning Mount Revolute Joint
Revolute joint added to steering arm allowing rotation around global Y axis.

Upper Mount Constraint
Revolute joint added to upper mount allowing rotation around all axis with movement in the Global Y direction.
**Lower Mount Constraint**

Revolute joint added to upper mount allowing rotation around all axis.

**Applied Force on Upright**

Location of force applied to the upright.
Upright Force Data
Data on the amount of force applied along each global axis.

Applied Moment on Upright
Location of moment applied to the upright.
Applied Moment Data
Data on the amount of moment applied along each global axis.

Upper Front Control Arm

Upper Control Arm Displacement Analysis
Deformation of upper control arm under loading conditions.
Aluminum material assignment for Upper Control Arm

High strength low alloy steel assignment for rod ends.
Outer Rod End Fixed Connection
Fixed connection between the rod end and control arm.

Inner Rod End Fixed Connection
Fixed connection between the rod end and control arm.
Inner Rod End Spherical Joint
Spherical joint created at inner face of rod end allowing for rotation about all axis but no translation.

Outer Rod End Spherical Joint
Spherical joint created at inner face of rod end allowing for rotation about all axis but no translation.
Upper Control Arm Mesh

Total area meshed for FEA.

### Details of "Body Sizing" - Sizing

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<td>Element Size</td>
<td>Element Size</td>
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</table>

### Advanced

- Defeature Size: Default
- Behavior: Soft

Upper Control Arm Mesh Data

Sizing for mesh applied to the upper control arm assembly during analysis.
Upper Control Arm Applied Force

Location of force applied to the Upper Control Arm.

**Details of "Force"**

- **Scope**
  - Scoping Method: Geometry Selection
  - Geometry: 2 Faces

- **Definition**
  - Type: Force
  - Define By: Components
  - Coordinate System: Global Coordinate System
  - X Component: 42.1 lbf (ramped)
  - Y Component: 0. lbf (ramped)
  - Z Component: 175.35 lbf (ramped)
  - Suppressed: No

Upper Control Arm Applied Force Data

Data on the amount of force applied along each global axis.
Lower Front Control Arm

Deformation of lower control arm under loading conditions.

Refined Mesh Area for Lower Control Arm
Scope of refined mesh area on structural rib for lower control arm.
Applied Force for Lower Front Control Arm

Location of force applied to the Upper Control Arm.

Applied Force Data for Lower Front Control Arm

Data on the amount of force applied along each global axis.
Inner Shock Bolt Constraint for Lower Front Control Arm
Bolt constraint between control arm and shock.

Outer Shock Bolt Constraint for Lower Front Control Arm
Bolt constraint between control arm and shock.
Fixed Joint Between Inner Rod End for Lower Front Control Arm
Fixed connection between the rod end and control arm.

Fixed Joint Between Outer Rod End for Lower Front Control Arm
Fixed connection between the rod end and control arm.
Spherical Joint Constraining Inner Rod End to Ground
Spherical joint created at inner face of rod end allowing for rotation about all axis but no translation

Spherical Joint Constraining Outer Rod End to Ground
Spherical joint created at the outer face of rod end allowing for rotation about all axis but no translation
Shock Constraints for Lower Front Control Arm
Shock force constrained to shock mount between bolt holes.

Rear Control Arm

Aluminum Material Assignment for Rear Control Arm
Aluminum material assignment for ANSYS testing.
High Strength Steel Material Assignment for Rear Control Arm
High strength low alloy material assignment for ANSYS testing.

Proven Hardware Material Assignment for Rear Control Arm
Proven hardware material assignment for ANSYS testing.
Spherical Joint for Inner Rod End
Spherical joint created at inner face of rod end allowing for rotation about all axis but no translation

Spherical Joint for Outer Rod End
Spherical joint created at inner face of rod end allowing for rotation about all axis but no translation.

Outer Shock Mount Far Bolt Connection
Constraint between inner shock mount and bolt.

Outer Shock Mount Close Bolt Connection
Constraint between inner shock mount and bolt.
Inner Shock Mount Far Bolt Connection
Constraint between outer shock mount and bolt.

Inner Shock Mount Close Bolt Connection
Constraint between outer shock mount and bolt.
Fixed Outer Rod end to Rear Control Arm
Fixed connection between outer rod end and control arm.

Fixed Inner Rod end to Rear Control Arm
Fixed connection between inner rod end and control arm.
Nut-Bolt Constraint 1
Fixed connection between bolt and nut.

Nut-Bolt Constraint 2
Fixed connection between bolt and nut.
Bolt to Rear Control Arm 1

Fixed connection between bolt and control arm.

Bolt to Rear Control Arm 2

Fixed connection between bolt and control arm.
Shock Constraints Spherical Joint Shock Mount

Connection between shock and shock mount.

| Details of "Longitudinal - Ground To controlarmschock bracket" |
|----------------------|------------------|
| **Graphics Properties** |
| **Definition** |
| Material | None |
| Type | Longitudinal |
| Spring Behavior | Both |
| Longitudinal Stiffness | 1161 lbf/in |
| Longitudinal Damping | 6.866 lbf/in/s |
| Preload Load | Load |
| Suppressed | No |
| Spring Length | 6.6328 in |
| Scope | Body Ground |
| **Reference** |
| Coordinate System | spring |
| Reference X Coordinate | 0. in |
| Reference Y Coordinate | 4.2 in |
| Reference Z Coordinate | 5. in |
| Reference Location | Click to Change |
| **Mobile** |
| Stopping Method | Geometry Selection |
| Applied By | Remote Attachment |
| Scope | 2 Faces |
| Body | controlarmshock bracker |
| Coordinate System | spring |
| Mobile X Coordinate | -5.4654x-008 in |
| Mobile Y Coordinate | -0.15824 in |
| Mobile Z Coordinate | -4.9977x-009 in |
| Mobile Location | Click to Change |
| Behavior | Rigid |
| Pinball Region | All |

Shock Data for Rear Control Arm

Front mount shock properties data.
Mesh Scope for Rear Control Arm
Scope of meshed surfaces for FEA analysis.

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<td>Behavior: Soft</td>
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Mesh Data
Sizing for mesh applied to the upright during analysis.
Refined Mesh area for Rear Control Arm
Scope of refined mesh area on structural rib for lower control arm.

Refined Mesh Data
Details of mesh refinement from ANSYS.
Applied Force for Rear Control Arm

Rear control arm force application area.

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Applied Force Data

Rear control arm force data.
Appendix C: Equations

Three Point Bend Mathematical Representation

\[ \sigma = \frac{3PL}{2bh^2} \]

\[ \varepsilon = \frac{\delta h}{L^2} \]

Where:

- P is applied force
- L is specimen length
- b is specimen width
- h is specimen thickness
- \( \sigma \) is stress
- \( \varepsilon \) is strain
- \( \delta \) is deflection at the center

These formulas were pulled from ASTM Standard listed as Reference 3
Appendix D: Material Data Sheets

Appendix E: ABET Accreditation

Appendix F: Resumes
Upright Force Calculations

Loading conditions at pile patch:

\[ P_x = -219 \text{ lb} \]
\[ P_y = 350 \text{ lb} \]
\[ P_z = -175 \]

\[ \Sigma F_x = 0 = P_x + R_{x_1} + R_{x_2} \]
\[ \Sigma F_y = 0 = P_y + R_{y_1} + R_{y_2} \]
\[ \Sigma F_z = 0 = P_z + R_{z_1} + R_{z_2} \]

\[ 0 = -219 + R_{x_1} + R_{x_2} \]
\[ 0 = 350 + R_{y_1} + 0 \]
\[ 0 = -175 + R_{z_1} + R_{z_2} \]

\[ R_{x_1} = 219 - R_{x_2} \]
\[ R_{y_1} = 350 \text{ lb} \]
\[ R_{z_1} = 175 - R_{z_2} \]

\[ \Sigma M_{about point 1} = 0 \]

\[ \Sigma M_{x_1} = 0 = P_y (1.70) - P_z (6.53) + R_{z_2} (19.53 - 6.53) \]
\[ \Sigma M_{x_1} = -350 (1.70) - (-175)(6.53) + R_{z_2} (13) \]
\[ R_{z_2} = 42.13 \text{ lb} \]

\[ \Sigma M_{y_1} = 0 = P_x (1.70) - R_{x_1} (5.09 - 1.25) \]
\[ \Sigma M_{z_1} = 0 = P_z (6.53) - P_x (19.53 - 6.53) \]

Upper Mounts:

\[ R_{x_2} = ? \quad M_{x_2} = 0 \]
\[ R_{y_2} = 0 \quad M_{y_2} = 0 \]
\[ R_{z_2} = ? \quad M_{z_2} = 0 \]

Lower Mounts:

\[ R_{x_1} = ? \quad M_{x_1} = 0 \]
\[ R_{y_1} = ? \quad M_{y_1} = 0 \]
\[ R_{z_1} = ? \quad M_{z_1} = 0 \]

\[ R_{z_1} = 132.87 \text{ lbs} \]
Front Upper Control Arm Force Calculations

**Reaction Forces at point 1:**
- $P_x = -109.82 \text{ lb}$
- $P_z = -42.13 \text{ lb}

**Reaction Forces at Point 2**

\[ \sum F_x = 0 = P_x + R_{1x} + R_{2x} \]
\[ 0 = -109.82 + R_{1x} + R_{2x} \]
\[ R_{2x} = 109.82 - R_x \]
\[ R_{1x} = 209.76 \text{ lb} \]

\[ \sum F_z = 0 = P_z + R_{1z} + R_{2z} \]
\[ 0 = -42.13 + R_{1z} + R_{2z} \]
\[ R_{1z} = R_{2z} = 21.06 \text{ lb} \]

\[ \sum M_{2y} = 0 = P_x (6.45) - P_z (15.30) - R_{1x} (6.45) \]
\[ 0 = 109.82 \times 6.45 + 42.13 \times 15.30 = R_{1x} (6.45) \]

**Applied Loads (P):**
- $P_x = -109.82 \text{ lb}$
- $P_z = -42.13 \text{ lb}$
Simple ride frequencies

Sports Car - 70 - 90 cpm
or 1.17 - 1.50 Hz

Quarter Car model.

\[ f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \]

Rear

\[ 1.50 = \frac{1}{2\pi} \sqrt{\frac{k}{175(52.2)}} \]

Solve for \( k \)

\[ k = 482.15 \text{ lb-ft-in} \]

Front

\[ 1.17 = \frac{1}{2\pi} \sqrt{\frac{k}{175(32.2)}} \]

Solve for \( k \)

\[ k = 239.707 \]

\[ Ku = k_s \left( \frac{a}{b} \right)^2 \cos^2(\theta) \]

Solve for \( k_s \)

\[ k_s = 1166 \text{ lb-ft-in} \]

\[ 482.75 = k_s \left( \frac{15.3 - 2.63}{15.3} \right) \cos^2(45) \]

\[ 239.707 = k_s \left( 1 \right) \cos^2(60) \]

Solve for \( k_s \)

\[ k_s = 1174.83 \text{ lb-ft-in} \]

This equation is used to calculate the spring constant for the front and rear suspension for ANSYS testing.
Rear Control Arm

Guess

\[
\begin{align*}
F_{sx} &:= 1 \text{lbf} & F_{sy} &:= 1 \text{lbf} \\
F_{1x} &:= 1 \text{lbf} & F_{2x} &:= 1 \text{lbf} \\
F_{1y} &:= 1 \text{lbf} & F_{2y} &:= 1 \text{lbf} \\
F_{1z} &:= 1 \text{lbf} & F_{2z} &:= 1 \text{lbf}
\end{align*}
\]

Known

\[
\begin{align*}
F_{ax} &:= -525 \text{lbf} & F_{ay} &:= 350 \text{lbf} & F_{az} &:= -217 \text{lbf} \\
F_{sx} &:= 1 \text{lbf} & F_{sy} &:= 1 \text{lbf}
\end{align*}
\]

Given

\[
\begin{align*}
&F_{sx} + F_{1x} + F_{2x} + F_{ax} = 0 \text{lbf} \\
&F_{sy} + F_{1y} + F_{2y} + F_{ay} = 0 \text{lbf} \\
&F_{1z} + F_{2z} + F_{az} = 0 \text{lbf}
\end{align*}
\]

\[
\begin{align*}
x \text{ mom} & \quad F_{az} d_{say} - F_{ay} d_{saz} + F_{1z} d_{s1y} - F_{1y} d_{s1z} + F_{2z} d_{s2y} - F_{2y} d_{s2z} = 0 \text{lbf} \cdot \text{in} \\
y \text{ mom} & \quad F_{ax} d_{sax} - F_{az} d_{saz} + F_{1x} d_{s1z} - F_{1y} d_{s1x} + F_{2x} d_{s2z} - F_{2y} d_{s2x} = 0 \text{lbf} \cdot \text{in} \\
z \text{ mom} & \quad F_{ax} d_{sax} - F_{ay} d_{sax} + F_{1x} d_{s1y} - F_{1y} d_{s1x} + F_{2x} d_{s2y} - F_{2y} d_{s2x} = 0 \text{lbf} \cdot \text{in}
\end{align*}
\]

\[
\begin{pmatrix}
F_{sx} \\
F_{sy} \\
F_{ax} \\
F_{ay} \\
F_{1x} \\
F_{1y} \\
F_{1z} \\
F_{2x} \\
F_{2y} \\
F_{2z}
\end{pmatrix}
= \text{Find}(F_{sx}, F_{sy}, F_{1x}, F_{1y}, F_{1z}, F_{2x}, F_{2y}, F_{2z}) = \begin{pmatrix}
266.306 \\
-363.697 \\
-770.711 \\
221.084 \\
94.79 \\
1.029 \times 10^3 \\
-207.387 \\
122.21
\end{pmatrix} \text{lbf}
\]

\[
F_{\text{steeltube}} := \left( F_{sx}^2 + F_{sy}^2 \right)^{1/2} = 450.77 \text{lbf}
\]
Front Straight Rod End

\[
\begin{align*}
F_{1x} &= -770.711 \text{-lbf} \\
F_{1y} &= 221.084 \text{-lbf} \\
F_{1z} &= 94.79 \text{-lbf}
\end{align*}
\]

\[
F_{\text{shear}} := \left( F_{1y}^2 + F_{1z}^2 \right)^{\frac{1}{2}}
\]

\[
F_{\text{axial}} := |F_{1x}|
\]

Company Data

\[
\sigma_{\text{yield}} := 710 \text{MPa}
\]

From Table 14-1

\[
d_{\text{major}} := .4375 \text{in}
\]

\[
d_{\text{minor}} := .3447 \text{in}
\]

\[
n_{\text{tpi}} := 14 \text{ in}
\]

Assumption

\[
d_{\text{rod}} := .75 \text{in}
\]


PRM-7T Rod End 7/16

\[
\Lambda_{\text{thread}} := \frac{\pi}{4} \left( d_{\text{major}} - 0.938194 \cdot \frac{1}{n_{\text{tpi}}} \right)^2 = 0.108 \text{ in}^2
\]

\[
M := F_{\text{shear}} \cdot d_{\text{rod}}
\]

\[
\sigma_{\text{stud}} := \frac{F_{\text{axial}}}{\Lambda_{\text{thread}}} + \frac{32 \cdot M}{\pi \cdot d_{\text{minor}}^3} = 358.647 \text{MPa}
\]

\[
\text{SF} := \frac{\sigma_{\text{yield}}}{\sigma_{\text{stud}}} = 1.98
\]

\[
F_{\text{shear}} = 240.548 \text{-lbf}
\]
Front Angled Rod End

From Table 14-1

$$F_{2x} = 1.029 \times 10^3 \text{lbf}$$

$$F_{2y} = -207.387 \text{lbf}$$

$$F_{2z} = 122.21 \text{lbf}$$

$$\theta_x := 180^\circ - 41.54^\circ = 138.46^\circ$$

$$\theta_z := \theta_x - 90^\circ = 48.46^\circ$$

$$F_{\text{axial}} := \left( F_{2x} \cdot \sin(\theta_x) + F_{2z} \cdot \sin(\theta_z) \right)^2 + F_{2y}^2 \right]^\frac{1}{2}$$

$$F_{\text{axial}} := F_{2x} \cdot \cos(\theta_x) + F_{2z} \cdot \cos(\theta_z) = -689.46 \text{ lbf}$$

Assumption

$$d := 0.5 \text{ in}$$

Company Data

$$\sigma_{\text{yield}} := 710 \text{ MPa}$$


HXAB-6T Rod End 7/16

$$A_{\text{thread}} := \frac{\pi}{4} \left( d_{\text{major}} - 0.938194 - \frac{1}{n_{\text{tpi}}} \right)^2 = 0.144 \text{ in}^2$$

$$M = F_{\text{shear}} \cdot d_{\text{rod}}$$

$$\sigma_{\text{axial}} := \frac{F_{\text{axial}}}{A_{\text{thread}}} + \frac{32 \cdot M}{\pi \cdot d_{\text{minor}}^3} = 406.313 \text{ MPa}$$

$$SF := \frac{\sigma_{\text{yield}}}{\sigma_{\text{stud}}} = 1.747$$

$$F_{\text{shear}} = 801.414 \text{ lbf}$$
Front Shock Geometric Analysis

\[ L_{\text{steeltube}} := 15\text{in} \]
\[ L_{\text{triangle\_wt}} := 4\text{in} \]
\[ L_{\text{triangle\_st}} := 1.5\text{in} \]
\[ L_{\text{triangle\_ws}} := 3.5\text{in} \]

Position := 0in

Guess
\[ \theta_{\text{triangle\_wt}} := 100^\circ \]
\[ \theta_{\text{steeltube}} := 50^\circ \]

Given
\[ L_{\text{triangle\_wt}} \cos(\theta_{\text{triangle\_wt}}) + L_{\text{steeltube}} \cos(\theta_{\text{steeltube}}) + 1\text{in} = 16.5\text{in} \]
\[ L_{\text{triangle\_wt}} \sin(\theta_{\text{triangle\_wt}}) + L_{\text{steeltube}} \sin(\theta_{\text{steeltube}}) = -9.5\text{in} + \text{Position} \]

\[ \begin{bmatrix} \theta_{\text{triangle\_wt}} \\ \theta_{\text{steeltube}} \end{bmatrix} := \text{Find}(\theta_{\text{triangle\_wt}}, \theta_{\text{steeltube}}) = \begin{bmatrix} 361.837 \\ -39.932 \end{bmatrix}^\circ \]

\[ \theta_{\text{triangle\_ws}} := \theta_{\text{triangle\_wt}} - 21.7868^\circ = 340.05^\circ \]

\[ L_{\text{shock\_x}} := L_{\text{triangle\_ws}} \cos(\theta_{\text{triangle\_ws}}) = 3.29\text{in} \]
\[ L_{\text{shock\_y}} := L_{\text{triangle\_ws}} \sin(\theta_{\text{triangle\_ws}}) + 8.75\text{in} \]
\[ L_{\text{shock}} := \left( L_{\text{shock\_x}}^2 + L_{\text{shock\_y}}^2 \right)^{\frac{1}{2}} = 8.241\text{in} \]
\[ \theta_{\text{shock}} := \text{atan} \left( \frac{L_{\text{shock\_y}}}{L_{\text{shock\_x}}} \right) = 66.471^\circ \]
Front Triangle Forces

\[ \text{F}_{\text{steel tube}} = 450.77 \text{ lbf} \]

\[ F_{\text{st}_x} := -F_{\text{steel tube}} \cdot \cos(\theta_{\text{steel tube}}) = -345.652 \text{ lbf} \]

\[ F_{\text{st}_y} := -F_{\text{steel tube}} \cdot \sin(\theta_{\text{steel tube}}) = 289.341 \text{ lbf} \]

\[ r_{\text{st}_x} := L_{\text{triangle wt}} \cdot \cos(\theta_{\text{triangle wt}}) = 3.998 \text{ in} \]

\[ r_{\text{st}_y} := L_{\text{triangle wt}} \cdot \sin(\theta_{\text{triangle wt}}) = 0.128 \text{ in} \]

\[ M_{\text{applied}} := (r_{\text{st}_x} \cdot F_{\text{st}_y} - r_{\text{st}_y} \cdot F_{\text{st}_x}) = 1.201 \times 10^3 \text{ lbf \cdot in} \]

\[ r_{\text{sh}_x} := L_{\text{triangle ws}} \cdot \cos(\theta_{\text{triangle ws}}) = 3.29 \text{ in} \]

\[ r_{\text{sh}_y} := L_{\text{triangle ws}} \cdot \sin(\theta_{\text{triangle ws}}) = -1.194 \text{ in} \]

Guess

\[ F_{\text{shock}} := 1 \text{ lbf} \]

Given

\[ r_{\text{sh}_x}(F_{\text{shock}} \cdot \sin(\theta_{\text{shock}})) - r_{\text{sh}_y}(F_{\text{shock}} \cdot \cos(\theta_{\text{shock}})) = -M_{\text{applied}} \]

\[ F_{\text{shock}} := \text{Find}(F_{\text{shock}}) = -343.84 \text{ lbf} \]

\[ F_{\text{sh}_x} := F_{\text{shock}} \cdot \cos(\theta_{\text{shock}}) = -137.268 \text{ lbf} \]

\[ F_{\text{sh}_y} := F_{\text{shock}} \cdot \sin(\theta_{\text{shock}}) = -315.251 \text{ lbf} \]

Spring

\[ K_s := 1168 \frac{\text{lbf}}{\text{in}} \]

\[ F_{\text{pre}} := 175 \text{ lbf} \]

\[ F_{\text{shock}} := (8.5 \text{ in} - L_{\text{shock}}) \cdot K_s = 302.503 \text{ lbf} \]
Front Shock Brackets

\[ F_{\text{bracket\_bot\_x}} := -F_{\text{sh\_x}} = 137.268\text{-lbf} \]
\[ F_{\text{bracket\_bot\_y}} := -F_{\text{sh\_y}} = 315.251\text{-lbf} \]

\[ F_{\text{bracket\_top\_x}} := -F_{\text{sh\_x}} - F_{\text{st\_x}} = 482.92\text{-lbf} \]
\[ F_{\text{bracket\_top\_y}} := -F_{\text{sh\_y}} - F_{\text{st\_y}} = 25.91\text{-lbf} \]
Rear Control Arm

Guess

\[
\begin{align*}
F_{sx} & := 1\text{lbf} & F_{sy} & := 1\text{lbf} \\
F_{1x} & := 1\text{lbf} & F_{2x} & := 1\text{lbf} \\
F_{1y} & := 1\text{lbf} & F_{2y} & := 1\text{lbf} \\
F_{1z} & := 1\text{lbf} & F_{2z} & := 1\text{lbf} \\
\end{align*}
\]

\[
\begin{align*}
d_{s2x} & := 12.77\text{in} & d_{s2y} & := -1.76\text{in} & d_{s1z} & := -6.5\text{in} \\
d_{s1x} & := 12.77\text{in} & d_{s1y} & := -1.76\text{in} \\
\end{align*}
\]

\[
\begin{align*}
r_{\text{patch}} & := \begin{pmatrix} 2.62\text{in} \\ 10.75\text{in} + 2.12\text{in} \\ 3\text{in} \end{pmatrix} & F_{\text{patch}} & := \begin{pmatrix} 132.25\text{lb} \\ 350\text{lb} \\ 175\text{lb} \end{pmatrix} \\
\end{align*}
\]

Given

\[
\begin{align*}
F_{sx} + F_{1x} + F_{2x} &= -132.25\text{lb} \\
F_{sy} + F_{1y} + F_{2y} &= -350\text{ lb} \\
F_{1z} + F_{2z} &= -175\text{lb} \\
\end{align*}
\]

\[
\begin{align*}
x \text{ mom} & = -F_{1y}d_{s1z} + F_{1z}d_{s1y} + F_{2z}d_{s2y} = -3302\text{lb} \cdot \text{in} \\
y \text{ mom} & = F_{1x}d_{s1z} - F_{1z}d_{s1x} - F_{2z}d_{s2x} = 855.25\text{lb} \cdot \text{in} \\
z \text{ mom} & = F_{1x}d_{s1y} - F_{1y}d_{s1x} + F_{2x}d_{s2y} - F_{2y}d_{s2x} = -785.058\text{lb} \cdot \text{in} \\
F_{1z} & = F_{2z} \\
\end{align*}
\]

\[
\begin{pmatrix}
F_{sx} \\
F_{sy} \\
F_{1x} \\
F_{1y} \\
F_{1z} \\
F_{2x} \\
F_{2y} \\
F_{2z}
\end{pmatrix} := \text{Find}(F_{sx}, F_{sy}, F_{1x}, F_{1y}, F_{1z}, F_{2x}, F_{2y}, F_{2z}) = \begin{pmatrix} -248.064 \\ -395.515 \\ 212.231 \\ -555.385 \\ -87.5 \\ -96.417 \\ 600.899 \\ -87.5 \end{pmatrix} \text{lb}
\]

\[
F_{\text{steeltube}} := \left(\frac{F_{sx}^2 + F_{sy}^2}{2}\right)^{\frac{1}{2}} = 466.87\text{ lb}
\]
Out of Line Rear Rod End

From Table 14-1

\[ F_{1x} = 212.231 \text{lbf} \]
\[ F_{1y} = -555.385 \text{lbf} \]
\[ F_{1z} = -87.5 \text{lbf} \]

\[ \sigma_{\text{yield}} := 710\text{MPa} \]

\[ \frac{1}{2} \left( F_{1y}^2 + F_{1z}^2 \right)^2 \]

\[ F_{\text{shear}} := F_{1y} + F_{1z} \]

\[ F_{\text{axial}} := F_{1x} \]

Company Data

\[ d_{\text{major}} := 0.5\text{in} \]
\[ d_{\text{minor}} := 0.4001\text{in} \]
\[ n_{\text{tpi}} := \frac{13}{\text{in}} \]

Assumption

\[ d_{\text{rod}} := 0.5\text{in} \]


PRM-8T Rod End 1/2

\[ \Lambda_{\text{thread}} := \frac{\pi}{4} \left( d_{\text{major}} - 0.938194 \cdot \frac{1}{n_{\text{tpi}}} \right)^2 = 0.144\text{in}^2 \]

\[ M := F_{\text{shear}} \cdot d_{\text{rod}} \]

\[ \sigma_{\text{stud}} := \frac{F_{\text{axial}}}{\Lambda_{\text{thread}}} + \frac{32 \cdot M}{\pi \cdot d_{\text{minor}}^3} = 318.428\text{MPa} \]

\[ \frac{\sigma_{\text{yield}}}{\sigma_{\text{stud}}} = 2.23 \]

\[ F_{\text{shear}} = 562.235\text{lb} \]
Rear Rod Ends

From Table 14-1

\[ F_{2x} = -96.417 \text{-lb} \]
\[ F_{2y} = 600.899 \text{-lb} \]
\[ F_{2z} = -87.5 \text{-lb} \]

\[ d_{major} := 0.5 \text{in} \]
\[ d_{minor} := 0.4001 \text{in} \]
\[ n_{tpi} := 13 \text{ in} \]

Company Data

\[ \sigma_{yield} := 710 \text{ MPa} \]
\[ d_{rod} := 0.5 \text{in} \]

Assumption


PRM-8T Rod End 1/2


\[ \Lambda_{thread} := \frac{\pi}{4} \left( d_{major} - 0.938194 \cdot \frac{1}{n_{tpi}} \right)^2 = 0.144 \text{-in}^2 \]

\[ M := F_{\text{shear}} \cdot d_{rod} \]

\[ \sigma_{axial} := \frac{F_{\text{axial}}}{\Lambda_{thread}} + \frac{32 \cdot M}{\pi \cdot d_{minor}^3} = 328.297 \text{ MPa} \]

\[ SF := \frac{\sigma_{yield}}{\sigma_{stud}} = 2.163 \]

\[ F_{\text{shear}} = 607.237 \text{-lb} \]
Front Shock Geometric Analysis

\[ L_{\text{steeltube}} := 11\text{in} \]
\[ \theta_{\text{board}} := 61.8^\circ \]
\[ \theta_{\text{bracket}} := \theta_{\text{board}} - 90^\circ \]
\[ L_{\text{triangle\_wt}} := 4\text{in} \]
\[ L_{\text{triangle\_st}} := 1.5\text{in} \]
\[ L_{\text{triangle\_ws}} := 3.5\text{in} \]

Position := \(-0\text{in}\)

Guess
\[ \theta_{\text{triangle\_wt}} := 0^\circ \]
\[ \theta_{\text{steeltube}} := -40^\circ \]

Given
\[ L_{\text{triangle\_wt}} \cdot \cos(\theta_{\text{triangle\_wt}}) + L_{\text{steeltube}} \cdot \cos(\theta_{\text{steeltube}}) + \frac{6.5}{\tan(\theta_{\text{board}})} \text{in} + 1 \cdot \cos(\theta_{\text{bracket}}) \text{in} = \]
\[ L_{\text{triangle\_wt}} \cdot \sin(\theta_{\text{triangle\_wt}}) + L_{\text{steeltube}} \cdot \sin(\theta_{\text{steeltube}}) + 1 \cdot \sin(\theta_{\text{bracket}}) \text{in} = -8 \text{in} + \text{Position} \]

\[
\begin{pmatrix}
\theta_{\text{triangle\_wt}}
\
\theta_{\text{steeltube}}
\end{pmatrix} := \text{Find}(\theta_{\text{triangle\_wt}}, \theta_{\text{steeltube}}) = \begin{pmatrix}
-1.771
-42.305
\end{pmatrix}^\circ.
\]

\[ \theta_{\text{triangle\_ws}} := \theta_{\text{triangle\_wt}} - 21.7868^\circ = -23.558^\circ \]

\[ L_{\text{shock\_x}} := L_{\text{triangle\_ws}} \cdot \cos(\theta_{\text{triangle\_ws}}) + \frac{6.5}{\tan(\theta_{\text{board}})} \text{in} = 6.694 \text{in} \]

\[ L_{\text{shock\_y}} := L_{\text{triangle\_ws}} \cdot \sin(\theta_{\text{triangle\_ws}}) + 6.5 \text{in} \]

\[ L_{\text{shock}} := \left( L_{\text{shock\_x}}^2 + L_{\text{shock\_y}}^2 \right)^{1/2} = 8.416 \text{in} \]

\[ \theta_{\text{shock}} := \tan^{-1}\left( \frac{L_{\text{shock\_y}}}{L_{\text{shock\_x}}} \right) = 37.311^\circ \]
Front Triangle Forces

\[ F_{\text{steeltube}} = 466.87 \text{ lbf} \]

\[ F_{\text{st}_x} := -F_{\text{steeltube}} \cos(\theta_{\text{steeltube}}) = -345.286 \text{ lbf} \]

\[ F_{\text{st}_y} := -F_{\text{steeltube}} \sin(\theta_{\text{steeltube}}) = 314.238 \text{ lbf} \]

\[ r_{\text{st}_x} := L_{\text{triangle wt}} \cos(\theta_{\text{triangle wt}}) = 3.998 \text{ in} \]

\[ r_{\text{st}_y} := L_{\text{triangle wt}} \sin(\theta_{\text{triangle wt}}) = -0.124 \text{ in} \]

\[ M_{\text{applied}} := (r_{\text{st}_x} \cdot F_{\text{st}_y} - r_{\text{st}_y} \cdot F_{\text{st}_x}) = 1.214 \times 10^3 \text{ lbf \cdot in} \]

\[ r_{\text{sh}_x} := L_{\text{triangle ws}} \cos(\theta_{\text{triangle ws}}) = 3.208 \text{ in} \]

\[ r_{\text{sh}_y} := L_{\text{triangle ws}} \sin(\theta_{\text{triangle ws}}) = -1.399 \text{ in} \]

Guess

\[ F_{\text{shock}} := 1 \text{ lbf} \]

Given

\[ r_{\text{sh}_x} \left(F_{\text{shock}} \sin(\theta_{\text{shock}})\right) - r_{\text{sh}_y} \left(F_{\text{shock}} \cos(\theta_{\text{shock}})\right) = -M_{\text{applied}} \]

\[ F_{\text{shock}} := \text{Find}(F_{\text{shock}}) = -396.972 \text{ lbf} \]

\[ F_{\text{sh}_x} := F_{\text{shock}} \cos(\theta_{\text{shock}}) = -315.736 \text{ lbf} \]

\[ F_{\text{sh}_y} := F_{\text{shock}} \sin(\theta_{\text{shock}}) = -240.62 \text{ lbf} \]
Front Shock Brackets

\[ F_{\text{bracket\_bot\_x}} := -F_{\text{sh\_x}} = 315.736 \text{ lbf} \]
\[ F_{\text{bracket\_bot\_y}} := -F_{\text{sh\_y}} = 240.62 \text{ lbf} \]

\[ F_{\text{bracket\_top\_x}} := -F_{\text{sh\_x}} - F_{\text{st\_x}} = 661.022 \text{ lbf} \]
\[ F_{\text{bracket\_top\_y}} := -F_{\text{sh\_y}} - F_{\text{st\_y}} = -73.618 \text{ lbf} \]
Designed to add value in OEM Product manufacturing, Plascore Honeycomb Panels are available cut to a custom size or shape, or in a standard sheet size of 48” x 96” in varying thicknesses.

Note that all of these are open edge.

### Specifications

<table>
<thead>
<tr>
<th>PANEL ID</th>
<th>SKIN</th>
<th>CORE</th>
<th>ADHESIVE</th>
<th>FLATWISE TENSILE STRENGTH¹</th>
<th>CLIMBING DRUM² (PER UNIT WIDTH)</th>
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<td></td>
<td></td>
<td></td>
<td>lb/in²</td>
<td>N/mm²</td>
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<td>.020” Aluminum with Epoxy Primer</td>
<td>3/8” Aluminum 3.6#pcf</td>
<td>Commercial Grade Toughened</td>
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<td>4.44</td>
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<td>AA5.2-95</td>
<td>.015” Fiberglass Prepreg</td>
<td>1/4” Aluminum 5.2#pcf</td>
<td>Epoxy</td>
<td>998</td>
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<td>.394” Polypropylene 4.0#pcf</td>
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Appendix D: Material Data Sheets
### Characteristics

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<tr>
<th>PANEL ID</th>
<th>AA3.6-80</th>
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* Detailed performance available upon request.

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<tr>
<th>STABILIZED COMPRESSIVE STRENGTH3</th>
<th>THICKNESS</th>
<th>WEIGHT</th>
<th>FLEXURAL RIGIDITY4 (PER UNIT WIDTH)</th>
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<tbody>
<tr>
<td>lb/in² N/mm²</td>
<td>in mm</td>
<td>lb/ft² kg/m²</td>
<td>lb-in² N·mm²</td>
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<td>----------------</td>
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<td>0.75 19</td>
<td>2.34 11.4</td>
<td>55506 15.07</td>
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</tbody>
</table>

---

1 Data obtained through testing in accordance with ASTM C 297
2 Data obtained through testing in accordance with ASTM D 1781 with specimen widths of 3"

3 Data obtained through testing in accordance with ASTM C 365
4 Data obtained through testing in accordance with ASTM C393/C 393M-06 at a Four-Point, Quarter-Point Loading configuration and a space of 20", width 3", except 0.25" @ 1.5" width
PK2 Para-Aramid Fiber Honeycomb

Description:
PK2 para-aramid fiber honeycomb is an extremely lightweight, high strength, non-metallic honeycomb manufactured with para-aramid fiber paper impregnated with a heat resistant phenolic resin. This core material exhibits improved performance characteristics over Meta-Aramid in the areas of weight, strength, stiffness and fatigue.

Applications:
PK2 honeycomb is a high performance non-metallic core which can replace fiberglass and Meta-Aramid honeycomb core materials to achieve significant weight reductions without sacrificing performance in most applications. PK2 honeycomb uses include boat decks, aircraft galleys, flooring, partitions, aircraft leading and trailing edges, radomes, flaps, access panels and doors.

Features:
- Up to 40% higher properties than comparable density Nomex® honeycomb
- Extremely high strength to weight ratio
- Excellent thermal and moisture stability
- Improved shear strength and modulus
- Conforms to stringent smoke, toxicity and flammability standards
- High toughness
- Long shelf life. The mechanical properties referenced are maintained for 10 years minimum if not exposed to moisture, weather or any normal hazard.

Availability:
PK2 honeycomb is available in sheets, blocks or cut to size pieces in regular hexagonal cell configurations. Selected densities available in high shear (HS) configuration for higher stiffness.

Cell Sizes: 1/8" - 3/16"
Densities: 2.0 pcf - 6.0 pcf
Sheet “Ribbon” (L): 48" typical
Sheet “Transverse” (W): 96" typical
Tolerances:
Length: + 3", - 0"
Width: + 6", - 0"
Thickness: ± .006" (under 2" thick)
Density: ± 10%
Cell Size: ± 10%

NOTE: Special dimensions, sizes, tolerances and specifications can be provided upon request.
PK2 Para-Aramid honeycomb is specified as follows:

Material - Cell Size - Density - Cell Configuration

Designates aerospace grade Para-Aramid
The nominal density in pounds per cubic foot
Cell size in inches
Higher shear property configuration

Example: \( PK2 \ - \ 3/16 \ - \ 3.0 \ - \ HS \)

<table>
<thead>
<tr>
<th>PK2 Para-Aramid Mechanical Properties</th>
</tr>
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<tbody>
<tr>
<td>Cell Size</td>
</tr>
<tr>
<td>Typical</td>
</tr>
<tr>
<td>---------</td>
</tr>
<tr>
<td>in</td>
</tr>
<tr>
<td>1/8</td>
</tr>
<tr>
<td>1/8*</td>
</tr>
<tr>
<td>1/8</td>
</tr>
<tr>
<td>1/8</td>
</tr>
<tr>
<td>1/8*</td>
</tr>
<tr>
<td>1/8</td>
</tr>
<tr>
<td>1/8*</td>
</tr>
<tr>
<td>5/32*</td>
</tr>
<tr>
<td>5.32*</td>
</tr>
<tr>
<td>3/16*</td>
</tr>
<tr>
<td>3/16*</td>
</tr>
<tr>
<td>3/16 0V*</td>
</tr>
</tbody>
</table>

Tested at 0.500”T per AMS STD 401 at room temperature.

The above data is based on various sample sizes and is for reference only.

Additional densities and configurations available upon request.

* Limited Testing or predicted values.

Plascore, Inc., employs a quality management system that is Nadcap, AS9100, ISO 9001 and ISO 14001 certified.

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Description
HexPly® M10E is an epoxy resin system preimpregnated into aramid, glass or graphite multiaxial fabrics and unidirectional fibers. M10E prepregs are recommended for the production of low cost, large thick-sectioned industrial structures.

Features
- Flexible Cure Cycle (185° F to 300° F)
- Vacuum Bag or Autoclave Cure
- High Flow, Low Viscosity System
- Medium to High Tack
- Can be Stored at Room Temperature for 60 Days
- Optically Clear Resin

Properties
- Glass transition temperature: Dry, 241° F, Wet, 194° F
- Resin density 1.19
- Barcol Hardness 33

Availability

<table>
<thead>
<tr>
<th>Form</th>
<th>Hexcel Designation</th>
<th>Fiber</th>
<th>Fiber Areal Wt. G/m²</th>
<th>Weave</th>
<th>Count Warp x Fill</th>
<th>Available Widths Standard Widths in (cm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fabric PW</td>
<td>282</td>
<td>3K</td>
<td>197</td>
<td>Plain weave</td>
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<td>50&quot;</td>
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<tr>
<td>Graphite UD</td>
<td>AS3C-S 150-300 GSM</td>
<td>AS3C-S</td>
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<td>12”, 24&quot;</td>
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Physical Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Carbon Tapes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile and compressive</td>
<td>M10E 37% AS3C-S, 300 GSM</td>
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</tbody>
</table>
Mechanical Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Temperature</th>
<th>Condition</th>
<th>Carbon Tapes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile Strength</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>293 ksi</td>
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<tr>
<td>Tensile Modulus</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>17.6 Msi</td>
</tr>
<tr>
<td>Tensile Strain</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>1.54%</td>
</tr>
<tr>
<td>Compression Strength</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>154 ksi</td>
</tr>
<tr>
<td>Compression Modulus</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>17.0 Msi</td>
</tr>
<tr>
<td>Compression Strain (calculated)</td>
<td>Room Temp.</td>
<td>Dry</td>
<td>0.91%</td>
</tr>
</tbody>
</table>

Cure Cycle
- 185° F, 14 hours
- 200° F, 120 minutes
- 250° F, 30 minutes
- 300° F, 8 minutes

Cure Procedure
Vacuum bag or autoclave cure using ramp rate of 2-5° F per minute. Recommend hold at 250° F for 30 minutes.

Storage
- At 0° F, 18 months
- At 41° F, 6 months
- At 72° F, 2 months

For more information
Hexcel is a leading worldwide supplier of composite materials to aerospace and industrial markets. Our comprehensive range includes:

- HexTow® carbon fibers
- HexForce® reinforcements
- HexPly® prepregs
- HexMC® molding compounds
- HexFlow® RTM resins
- Redux® adhesives
- HexTool® tooling materials
- HexWeb® honeycombs
- Acousti-Cap® sound attenuating honeycomb
- Engineered core
- Engineered products

For US quotes, orders and product information call toll-free 1-800-688-7734. For other worldwide sales office telephone numbers and a full address list, please go to:

http://www.hexcel.com/contact/salesoffice
Appendix E: ABET Accreditation

**ABET student outcome 4:**

An ability to recognize ethical and professional responsibilities in engineering situations and make informed judgments, which must consider the impact of engineering solutions in global, economic, environmental and societal contexts.

Performance Indicator #2: student is able to make informed judgments based on the impact of engineering solutions in global, economic, environmental and societal context.

**ME4800 Assessment of PI #2**

(to be completed by students and included in the ABET Appendix of the final report)

Did you adapt your project to make it useful in many countries? Y / N / NA If yes, explain:

   NA

Did you consider standards and regulations, either U.S. or international? Y / N / NA If yes, explain how they affected your project:

   Y: America Solar Challenge puts forth regulations that defined the whole design.

Did you consider the effects of manufacturing in various locations? Y / N / NA If yes, where in the report did you address this issue?

   Y: The parts would eventually have to be manufactured. Parts were designed to be easily machinable.

Did you have to balance effects of costs and performance? Y / N / NA If yes, explain and refer to the report as appropriate.

   N: This design was not considering cost. Most materials and labor are donated.

Did you consider effects of maintenance, failure and repair on cost, safety, etc.? Y / N / NA If yes, where in the report did you address them?

   Y: serviceability was one of our goals for the project, specifically on the brackets. Safety was another main proponent of the project throughout the design.

What were your considerations (e.g., cost, weight, manufacturing, availability, safety, recycling, etc.) in the selection of materials? List, explain and refer to the text of the report as appropriate.

   Weight: we weighted each component for weight savings
   Manufacturing: we made parts simpler to be easier to manufacture
Safety: All the components were tested for their worst-case scenarios and have them exceed in these conditions.

Does your project impact air quality, water quality, noise levels, and other environmental aspects? Y / N / NA If yes, explain how and show what were your actions.

NA

Does your project impact human health during manufacturing or normal use? Y / N / NA If yes, explain what you did to alleviate the risks.

Y: If enough forces are put into the system the members can punch through the board and injure the driver. Also, if the suspension fails, loss of control of the care could occur. This could cause the car and driver crash. We alleviated risks by testing the components, and proving components work in their required conditions.

Are there any other safety issues typical to your project? Y / N / NA If yes, explain your decisions and actions. Refer to the report as appropriate.

General danger that could occur during machining and working in a workshop environment. In addition, the dangers that go with driving on any public road.
Enthusiastic Mechanical Engineer student who is motivated to apply education and work experience in a full-time position. Possesses good communication and critical thinking skills needed to address immediate and long-term issues. I would be a value add to Epic Systems for addressing issues in support of improved operations, efficiencies, and company success.

Bachelor of Science, Mechanical Engineering
Minor in Mathematics | Western Michigan University, GPA 3.0
SEPTEMBER, 2015 – DECEMBER, 2019

Aided in Probabilistic Risk Assessment compliance and improved error detection in monitoring systems.

Established procedure to evaluate critical components and aided in thermal performance improvement.

Created Access Database to monitor recurring maintenance and streamline work load planning.

Organized and compiled student data into Banner software while maintaining high level of confidentiality.

Lightweight Racing Suspension | Senior Design Project
Designed a lightweight suspension with projected weight savings of 15%. Performed material testing and did cost benefit analysis to choose optimal material. Made cost effective decisions and adhering to timelines set by team leads.

Microsoft Office
ANSYS Fluent
NX Modeling
Solid Works
LTSpice
Catia
LabView
AutoCAD

Student Organization Treasurer/Advisor - Swing Dancing, September 2018 – April 2019
Objective

Motivated, enthusiastic, and hands-on individual seeking a full-time position after graduation in December 2019. Offers a technical mindset, critical thinking and interpersonal communication skills in order to solve both short and long-term problems.

Education

Bachelor of Science in Engineering
Western Michigan University, Lee Honors College Kalamazoo, MI
Expected Graduation: Dec. 2019
Major: Mechanical Engineering Minor: Mathematics GPA: 3.51

Work Experience

WKW Extrusion- Erb sloeh Aluminum Solutions, Inc.
Mechanical Engineering Intern January 2018 - Present Portage, MI

• Created Extrusion die drawings in CAD to ensure quality of production
• Adjusted factory floor plans to increase workflow efficiency for new processes
• Fabricated base for BMW roof rail for Rockwell hardness tests to provide consistent mounting for tests
• Collaborated with fabrication floor managers to create concise and accurate standard work instructions for various saw, CNC, and punch press operations

Official Finders
Baseball Umpire May 2014 - July 2018 West Chicago, IL

• Officiated baseball games for kids between ages of 8-17
• Regulated games alone or with a partner
• Applied rules of the game fairly, efficiently, and effectively to keep games consistent
• Resolved conflicts with coaches and spectators through clear communication
• Effectively communicated with co-workers through signals and meetings between innings

Relevant Experience

Senior Design Project: Lightweight Racing Suspension for 2020 Solar Car March 2019 - Present

• Engineering suspension components to reduce weight by estimated 15%
• Run material strength testing on composite samples to determine most cost-effective material
• Adhered to strict timeline to adhere to in order to complete project

Sunseeker Solar Car September 2015 - Present Kalamazoo, MI

• Designed battery box for 2016 car
• Fabricated mold to hold carbon fiber for heat treatment
• Addressed other sub-teams to ensure cohesion between designs
• Optimized air flow to cool battery by 17%

Familiar Software

• AutoCAD, Inventor, Solid Works, MATLAB, ANSYS Fluent, LTspice, LabVIEW, Microsoft Office, BDV, SAP, Verse, MathCAD.
Objective:
Highly motivated, hands-on engineer looking for full time employment in January of 2020 to apply my knowledge and passion of engineering. Offering leadership skills, hands on experience, and problem solving abilities to develop and improve current projects and future designs.

Education:
- Western Michigan University
  - Bachelor of Science in Mechanical Engineering
  - Graduation: December 2019
  - GPA: 3.05

Technical Experience:

Project Engineer Intern: Summit Polymers Inc.  
- Jan 2019-Present
  - Created corporate trainings for safe operation of machinery: mill, drill press, band saw, and belt sander
  - Mock up changes on existing parts to simulate future changes that address and fix design issues
  - Determine cause of issues with current products and determine design change to fix the problem
  - Test and inspect products to meet customer specifications: effort forces, dimensional accuracy, fitment etc.
  - Assist Project engineers with kaizen’s to document part issues and solutions
  - Lead the organization of benchmarked parts room, cataloged inventory and developed organizational layout of parts for easy reference and identification

Mechanical Engineering Intern: FEMA Corp.  
- Jan. 2017-Aug. 2018
  - Quality Intern working with the Quality department and Inspection team
    - Assist engineers by inspecting parts to determine features affecting line performance and not in conformance with sales drawing specifications
    - Worked with IQS to keep track of reoccurring issues with parts and suppliers to provide feedback data for the department
  - Field Service Intern working with the Field Service team
    - Investigate returned valves to through electrical and hydraulic testing to determine the cause of failure and construct reports of findings for customers
    - Worked with engineers to prevent future failures and create a reaction plans with manufacturing
    - Worked with assembly line teams to improve production changeovers using standardized work cards

Western Michigan University Sunseeker Solar Car Team Member  
- Since 2015
  - Mechanical Team Lead 2018-2019
    - Organize the mechanical team through the design phase of the next generation car: Assign projects, provide technical background for assigned projects, guide members through projects
    - Teach members important design concepts and how to work in a teamwork environment by creating cross team communication
    - Teach and enhance members’ technical abilities with composite and metal fabrication to assist in design
  - Project Manager 2017-2018
    - Oversee and manage sub team leads, projects, design choices, event organization, race logistics and planning, and member engagement and retention
    - Determining priority projects and delegating tasks to members
    - Lithium Ion battery pack design and fabrication for Solar Car
  - Light Weight Suspension: Senior Design 2019-Current
    - Design mechanical components in solid works and perform FEA analysis to simulate loading conditions
    - Design for tight space constraints by working with multiple groups to ensure no interference of assemblies

Skills and Strengths:
- Hard working and hands on learner with drive and a background in quality
- Solid works, Inventor, AutoCAD, Microsoft Office, MATLAB, LabVIEW, and Java

Honors:
- Eagle Scout – leadership roles: Junior Assistant Scoutmaster, Senior Patrol leader