I. EXECUTIVE SUMMARY

Each year Society of Automotive Engineers (SAE) challenges engineering students to design and build an offroad vehicle that will survive and compete in their annual competitions. For this senior design project, the objective was to design and build the front suspension for the GW Baja vehicle to compete in the 2018 Maryland SAE International Competition. The customer required an all weather, rugged, single-seat, off-road recreational vehicle, that complies with all of the competition requirements and can withstand the demands of the various dynamic and endurance competitions. The front suspension plays a key role in providing stability of the vehicle and maintaining consistent tire contact with the control surface.

The geometrical design chosen for the front suspension was a dual, unequal length A-arm system. With the upper arm slightly shorter than the bottom, the system provided high stability and a negative camber to lower chances of a vehicle roll over. The upper arm swings through a shorter arc than the lower arm and pulls in the top of the tire as the wheel travels upwards in reaction to the ground surface. This system meets all specifications and parameters outlined by the competition, in addition to providing the driver with ideal road handling control and a preferable ride height. Additionally, the suspension system design made the adjustability of the lengths easy. The stiffness was tunable with the shock absorbers dual chambers, each providing optimal pressures to handle the specific driving terrain and tests. During the actual testing phase, the group conducted both physical test drives and Solidworks analysis of the system. After testing, all requirements and results were satisfied as the suspension had the ability to handle various stresses and loads on all of the critical areas.

II. INTRODUCTION

An independent suspension system is a system of linkages that hold the shock absorbers and connect the frame of the vehicle to the wheels. It compensates for vertical acceleration of the wheels and impacts both the ride and handling of the vehicle. A suspension system should offer control over the camber angle of the wheel, provide minimal roll and sway to provide consistent steering, give firm contact of the wheels to the road, and allow driver comfort/ride quality. Due to the impact the front suspension has on not only the ride quality of the vehicle, the front suspension is one of the most critical subsystems of any vehicle. The scope for this project was to have successfully designed and built an improved front-end suspension system to be used at the SAE Mini Baja Competition from April 17th to April 20th.

The design requirements for the front suspension were not only governed by the customer (SAE Baja), but also the other subsystems on the vehicle. The front suspension had to work in conjunction with the steering system and the braking system, in order to ensure a smooth and effective driving experience. With all customer constraints laid out, and the chassis of the Baja vehicle already built, the design requirements were calculated based on the optimal geometry of the suspension system. The vehicle ride height was set at a minimum of 11 inches, the track width to 56 inches, and the wheelbase to 60 inches. The suspension had to achieve a leverage of 11:1, and the camber needed to be between -1 to -3 degrees.
III. DESIGN DESCRIPTION

The design of the double-wishbone system was constrained by the overall track width, steering knuckles, and wheels. Because the steering knuckles were used by the previous vehicle and the wheels and tires had already been selected, there was not much flexibility allowed in terms of design. An opportunity was noticed in the frame suspension mounts to specifically chose where the arms mounted to the frame. Because the frame was incomplete at the time of design, the location of the upper control arm had not been set. The frame at the front of the car was also too narrow for the required track width, so the control arms would need to be spaced out from the frame.

III.I THE GEOMETRY

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The initial suspension geometry design was performed using a GUI suspension analysis software RacingAspirations. The software allowed for iterative design and analysis of different geometries, while staying within the design restrictions. Varying geometries effect the camber of the wheel through its range of motion. A system was considered effective if the wheel maintained an overall negative camber as well as contact with the surface.

![Figure 1: Geometry Modeled in RacingAspirations (Left/Right Turn)](image)

As such, the control arm mounts were designed with a 2.5” and 1.5” spacing out from the frame for the upper control arm (UCA) and lower control arm (LCA). The distance between the UCA and LCA was set at 4.5”. To achieve the 56” front track width, lengths of 12.875” and 16.25” for the upper and lower control arms respectively were selected. This gives very shallow upper and lower control arm angles at the vehicle’s static position.

Another important handling characteristic is the vehicle’s roll center. The roll center was determined by drawing lines through the upper and lower control arms of a double-wishbone suspension system. The intersection of these lines is located, and another line is drawn from this intersection point to the center of the contact patch with the ground on the same side of the vehicle.
The roll center lies where this final line intersects the midpoint of the front track width. Figure 1 shows the location of the roll center through the use of this instant center method. The geometric relationship between the roll center and the center of gravity is what determines a vehicle’s dynamic responses.

![Figure 2: Roll Center Explained](image1.png)

It is important to note that roll center changes as the vehicle’s suspension system travels through its range of motion. Because the roll center changes, a more appropriate name would be an instantaneous roll center. and its motion in relation to the vehicle’s center of gravity is just as important. As the vehicle’s suspension system compresses, the roll center typically moves upward. This reduces body roll the more the suspension compresses. Off-road handling will be significantly improved through a lower static roll center that moves upwards with suspension compression.

![Figure 3: Model of Front Suspension Fully Assembled](image2.png)

As seen in Figure 3 above, the suspension was designed with two unequal length arms. The frame side of the arms are attached with heim joints and the wheel side of the arms are attached with ball joint mounts. The upper A arm is the shorter length arm, whereas the lower A arm is the longer of the two. Without varying the lengths of the arms and just having a dual A-arm suspension there is a lack of camber gain as no negative camber can be generated when the wheel moves into a bump on the course. The result of having no negative camber is that as the car rolls, the wheel gains positive camber and then would lose traction very quickly. Camber is defined as the angle between the vertical axis of the wheels used for steering and the vertical axis of the vehicle when viewed from the front. Now by using an upper control arm that is shorter than the lower one, the
suspension has the ability to hold a slight negative camber at rest and then gain an even greater negative camber with any upward wheel travel while performing on the course. This happens as the upper arm swings through a shorter arc than the lower arm does and then pulls in the top of the tire as the wheel travels upwards.

Heim joints were chosen to join the suspension arms with the frame. They provide minor adjustability and help account for any small dimension errors from the manufacturing stage. Threaded rod ends for the heim joints would be welded into the end of the chromoly tube.

1” Chromoly steel with a wall thickness of .0625” was selected as the main material. Chromoly steel was the ultimate fit as it provides high tensile strength (yield of 460 MPa) and malleability, can be easily welded, and has corrosion resistance. Though it can be costly in comparison to other materials, the benefits of Chromoly steel outweigh its costs. A careful comparison of steel alloys and their properties was needed to effectively determine the best steel for the intended purposes. Both the heim joints and the rod ends are made of high-strength steel.

Once a working Solidworks model of the suspension system was created, the next step was to ensure that the material and design would hold up under the most extreme conditions of competition. Finite Element Analysis (FEA) was imperative to confirm the working design was able to withstand the elements on the competition course. From reading previous group’s competition reports, the importance of considering failure scenarios became evident. The design may hold up fine while driving and turning with nothing impeding the course and with miniscule obstacles such as rocks and smaller jumps in the way. The real challenge is durability and ensuring the suspension can combat the most significant challenges presented by the course.

III.II FABRICATION

The process of fabricating the arms and cross beams started with cutting rods of 4130 Chromoly steel to the required length. After marking off the start and end points of the arm length, as well as the bending limits, the group used a pneumatic bender with a 4.5 inch die to achieve the correct angle on each bend. Using a hand-held adjustable protractor, the correct angle for each bend was achieved on all upper and lower control arms. Precise measurements that were double and triple checked were employed during this process to confirm the bends were not overdone. The angle measuring process was double-checked outside the bender by using a magnetic angle finder that would adjust to being placed on an angle from a horizontal surface.

Ensuring the correct angles was imperative, due to the symmetry needed on each upper and lower control arm of the suspension system. If either arm was bent too far, the whole geometry of the system would be thrown off and the suspension would not function effectively.

The cross beams were cut using an angled Bridgeport mill with a 1 inch diameter drill bit. This process entailed adjusting the machine to the correct angle on both sides of the cross beams, as well as centering each beam on the machine to ensure symmetry and a correct fit. Each beam was marked to the correct length, and the orientation was double checked on each crossbeam to confidently drill without making two of the same angled cuts on the same beam. This process was tediously carried out due to symmetry and test fit concerns on each cross beam. If each side of the beam was cut with the same angle, it would have to be thrown away and re-fabricated.

The ball joint mount side of each control arm had to be carried out using a different process, due to the lack of a 1 ⅜ inch drill bit in the machine shop. A rig was constructed to
accurately angle each arm on a horizontal plane to the correct specifications, and a drill was centered on the same horizontal surface to cut directly in the middle of the arm, producing an angled fishmouth cut. This process required much more time and effort, due to chatter on the drill bit and the need for stabilization.

Once all the individual parts were manufactured, the test-fitment process began. Due to the high degree of symmetry in the suspension arm designs, the completed parts must match the required arm dimensions when assembled. To do this, the individual parts were laid out on a flat surface. Powerful magnets were used to hold the components in the correct location while the arm dimensions were checked. Measurements were taken between the points where the arm would meet the frame, as well as from the frame mount to ball joint.

The fishmouths must also be checked for fitment at each joint. Each must allow for clean weldment, so no large gaps could be left. The material must also be prepared for welding by grinding and polishing each joint. Once an arm was deemed toleranced and complete, it went on to the initial tack welding phase.

A professional TIG welder was scheduled to perform the final weldment of the suspension arms. Before this, the arms needed to be lightly held together once appropriately toleranced. A MIG welder was used to perform these because of its forgiveness and ease of use.

The arms were placed on a welding table, appropriately dimensioned, and then magnetized in place. Two small welds were then made at each joint, ideally placed on opposing faces of the joint. Once the entire arm was secured, these tack welds were cleaned and ground down. It is important to make these tack welds as unobtrusive as possible. This ensured the professional TIG welder could make clean, strong welds at each joint.

Prior to the competition, the team must submit a weld sample from the same professional welder. The sample consisted of a small section of welded material that had undergone a tensile strength test. At the end of the test, the material itself must yield, not the weld itself. Therefore the strength and quality of the welds was a high priority.

IV. EVALUATION & TESTING

The testing and evaluation of the front suspension system was broken into three different phases. The first phase was done through SolidWorks Software Simulation and the second phase was two different days of test drives at an off road location in Virginia. After the testing the third phase included tuning the shock absorbers with data collected, knowledge of the competition events and using the shock manual as a valuable guide.

IV.I SOLIDWORKS

SolidWorks Software Simulation:

The suspension arms were each individually tested in SolidWorks for various loadings and displacements. While it is useful to test random loadings and visualize where potential failure points may be located, it is much more constructive to think of real scenarios in which the suspension system may fail/break. The two scenarios that were taken into consideration were a fall from 5 feet and a collision with a tree. For both scenarios, a “worst case” scenario is imagined, where the estimated loadings and/or other related values exceeded possible capacity.
First scenario: 5 foot drop and landing on all 4 wheels

In this scenario, the force of the falling car will disperse through all four tires, meaning that one fourth of the force will be put on each tire. The first thing that was needed in this scenario was the mass of the car. With the body weighing in at 467 lbs, and an estimated rider weight of 170 lbs, the total weight of the car/driver is 637 lbs. This value is a force, and has to be converted to mass. That calculation was done as follows:

\[ F = ma \]
\[ 637 \text{ lbf} = (m)(32.2 \text{ ft/s}) \]
\[ m = 19.78 \text{ slugs} \]

Before the mass was used in any calculations, the scenario called for a drop of the car, meaning that the velocity of the car upon impact also had to be calculated. Using kinematic equations and reducing due to the lack of an initial velocity for this scenario, the impact velocity calculation looked like this:

\[ V = \sqrt{2gh} \]
\[ V = \sqrt{2 \times 32.2 \text{ ft/s} \times 5 \text{ ft}} \]
\[ V = 17.94 \text{ ft/s} \]

At this point, all the components to calculate the car’s Kinetic Energy are available.

\[ KE = \frac{1}{2}mv^2 \]
\[ KE = \frac{1}{2}(19.78 \text{ slugs})(17.94 \text{ ft/s})^2 \]
\[ KE = 3183 \text{ ft*lbf} \]

The final goal here is to solve for the impact force on each tire, and the final parameter to take into consideration is the shock travel. Based on initial field testing, the length the tires can travel vertically before the shocks bottom out is 7 inches.

\[ d = 7 \text{ in} \times (1\text{ft}/12 \text{ in}) \]
\[ d = 0.5833 \text{ ft} \]

With all the necessary values available, the total impact force can be calculated by dividing the Kinetic Energy of the falling car by the distance the wheels will travel based on impact.

\[ F_{avg} = KE/d \]
\[ F_{avg} = (3183 \text{ lbf*ft})/(0.5833 \text{ ft}) \]
\[ F_{avg} = 5456.62 \text{ lbf} \]
\[ 5456.62 \text{ lbf}/4 = 1364.155 \text{ lbf/wheel} \]

The final line of calculations here is what really matters, because that is what can be tested in SolidWorks. Applying the force at the ball joint mount, the stress and displacement are shown.
in Figure 4 and Figure 5. The maximum areas of stress are where parts are welded together, including the tube adapters.

Figure 4: Bottom Arm Stress from a Drop of 5 feet

Figure 5: Bottom Arm Displacement from a Drop of 5 feet

Second scenario: Hitting a rigid obstacle at 25 mph

The second scenario was more variable in the parameters that could be maximized. Some of the values here were estimated based on The Motor Insurance Repair Research Centre that related impact G-force to impact velocity. Based on initial trial runs, a safe estimate of the car’s velocity is 25 mph, or 36.7 ft/s. Taking information from the Repair Research Centre and estimating based on the mass of the car in use, the maximum G-force value can be taken as 6. The calculation is a simple Newton’s Second Law problem that will give the impact force on the suspension system.

\[ F = ma \]
\[ F = (19.78 \text{ slugs}) \times (6 \text{ G}) \times (32.2 \text{ ft/s}) \]
\[ F = 3821.5 \text{ lbf} \]

This value is what is estimated to be the absolute maximum force exerted by a rigid object on the suspension system. As shown in Figure 6, the displacement on the arm is at the point of
impact and propagates out into lower levels of displacement. The maximum stress value experienced by the arm is around 50 ksi, with one or two small outliers at the tube adapter connection points, which is to be expected since the connections are the weakest links of the suspension arms, which is shown in Figure 7.

![Figure 6: Bottom Arm Displacement from hitting a tree at 25mph](image)

![Figure 7: Bottom Arm Stress from hitting a tree at 25mph](image)

**IV.II TEST DRIVES**

On two different occasions at a remote Maryland location there were various tests run on the car and suspension system in order to best simulate how the car will react to events that will occur at the competition.

The first test drive was completed in March after full assembly of the vehicle. The goal of the test drive was to have a smooth run with no immediate failures as well as seek the adjustments that would be needed. The shocks in this test drive were initially tuned to higher stiffness to accommodate the acceleration test and overall endurance of the vehicle. They were left at the same pressure throughout the day as the movement and durability of the suspension was the primary concern at this stage. The vehicle was first run on pavement to achieve acceleration and maximum speed data. Secondly, the vehicle was introduced to a slightly rougher terrain, which included mud, sticks, rocks, and small bumps. The goal was to run the vehicle in the marked track until failure of a system. When there was no failure, the vehicle tried a slightly rougher unmarked terrain. The test drive ended when the vehicle ran into a large fallen tree directly onward of the left side front and rear suspensions. The front suspension withstood the impact and had no deformation, deeming a successful first test drive for the team.

During the second test drive the final suspension system was tested. The completed Baja car can be seen in Figure 8. Testing for the suspension system was conducted by qualitative trials.
that included jumping on the front tow hitch and driving the car over variable terrain in field. The
terrain included hills, rocks, drops, and brush. The goal of the tests were to understand how to
tune the shocks to optimal settings for various scenarios that would be experienced in the
competition. Unfortunately, another part of the car underwent too much stress and eventually failed
during a test drive. This failure occurred early in the day, so only minimal data could be extracted.

Figure 8: Complete Car at Test Drive

IV.III SHOCK ABSORBER TUNING

The shock absorbers are a fundamental part to the front suspension. They control the rate
of the motion between the chassis and the wheels while dampening the shock effects from the
ground. This helps to improve the ride quality and overall vehicle handling. The predetermined
shock absorbers were FOX FLOAT 3 EVOL FACTORY SERIES air shocks. With this selection,
it is the use of high pressure air that enables the system to dampen the encountered impacts. Built
with 6061-T6 aluminum, the shock's strength, damping consistency, and lightweight construction
prove to be a great selection with the proper pressure chamber adjustments.

There are two parts, the main air chamber and the EVOL air chamber. These chambers are
independently tuned to support the rider and vehicle's weight on a specific travel terrain. The main
chamber is primarily responsible for the vehicle's ride height and stiffness of the suspension, or
the rate at which the shock absorbers will react to the ride's ground forces. The additional EVOL
chamber supports the last third of the shock travel. This beneficial feature can control the vehicle's
ability to bottom-out or corner roll. The figure below shows the impact and adjustability the two
chambers allow versus the standard coil shock selection.
When tuning the shocks, it is important to consider the different conditions the suspension system will be put through. At the competition, the vehicle will experience large jumps, long straightaways, and a combination of the two in a long endurance race. It is permitted to change the shock absorber pressure chambers between events; therefore, different pressures for the EVOL Air Chamber must be effectively selected. Table 1 below gives a rough estimate in determining the proper settings for each type of terrain while Figure 10 shows the effects of altering each of the chambers; both were used when determining the appropriate settings.

<table>
<thead>
<tr>
<th>Reference Air Pressures</th>
<th>EVOL Air Chamber</th>
<th>Main Air Chamber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mountain/Backcountry</td>
<td>120-150</td>
<td>35-65</td>
</tr>
<tr>
<td>Performance/Trail</td>
<td>160-180</td>
<td>35-95</td>
</tr>
<tr>
<td>Sno-Cross/X-Country</td>
<td>200-250</td>
<td>100-120</td>
</tr>
</tbody>
</table>
**Figure 10: Effects of Altering the EVOL vs. Main Chamber**

Table 2 below holds the optimal settings for the three aspects of the competition. The higher pressures were selected for the speed and acceleration portions to give the vehicle a stiffer ride. The endurance and jump competitions require more flexibility and were given less pressure.

<table>
<thead>
<tr>
<th>Competition</th>
<th>EVOL Air Chamber (psi)</th>
<th>MAIN Air Chamber (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed/Acceleration</td>
<td>250</td>
<td>120</td>
</tr>
<tr>
<td>Endurance</td>
<td>180</td>
<td>120</td>
</tr>
<tr>
<td>Jumps/Rough Terrain</td>
<td>150</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 2: Air Chamber Pressure Selections

When mounting the shock absorbers, the location on the bottom arm was determined by the optimal suspension leverage. For this system, a ratio of 1.6:1 was selected based on suspension geometry and clearance issues. This system yields an overall wheel travel of 7.2” from the 4.5” of the shock absorbers, see calculation below:

\[
\text{Suspension Leverage} \times \text{Shock Absorber Travel} = \text{Wheel Travel} \\
1.6 \times 4.5” \times 0.3 = 2.16”
\]

The shock absorbers main chamber has control over the vehicle's sag height, or the height at which the motionless vehicle sits with a driver on board in relation to the vehicle's full extension. The sag is dependent on the vehicle weight, the rider weight, and the specific usage of the vehicle. The recommended sag height is between 25% - 35% of the free length wheel travel. The calculations to obtain an optimal sag using 30% are below:

\[
\text{Suspension Leverage} \times \text{Shock Absorber Travel} \times 0.3 = 30\% \text{ of Wheel Travel} \\
1.6 \times 4.5” \times 0.3 = 2.16”
\]
The true sag height calculation of the vehicle is shown below with max pressure in the main chamber. All the determined pressures fit the optimal front sag height mentioned previously.

\[
\text{Full Extension - Ride Height} = \text{Sag} \\
11.25'' - 9.0'' = 2.25''
\]

The angle at which the shock absorbers were mounted also has to be accounted for when completing spring calculations. The shock absorbers are advantageous when in a vertical position, or perpendicular to the ground. Due to clearance issues with the frame of the upper suspension arm, the absorbers were attached at a 20° angle from the vertical. The angle needed to be accounted for when calculating the spring rate, or the amount of weight that is needed to compress the shock by one inch. The calculations of the spring rate with the max stiffness and the highest pressures in both chambers is shown below:

\[
ACF = \cos (A^\circ) \\
0.94 = \cos (20^\circ)
\]

\[
\text{Vertical Mounted Shock: Spring Rate} = \frac{\text{Weight with Driver}}{\text{Sag}} \\
= \frac{607 \text{ lbs.}}{2.25''} = 269.78 \text{ lbs./in}
\]

\[
\text{Shock Mounted at 20}^\circ = \frac{269.78 \text{ lbs./in}}{0.94} = 287 \text{ lbs./in}
\]

The shock absorbers are a crucial element in the design of the front suspension; thus, the shocks were tuned optimally for their event and the driver.

V. SUMMARY & RECOMMENDATIONS

Through the design requirements and goals set forth by both the SAE competition rules and the GW Baja team, the senior design group has designed and manufactured a front suspension able to work with the vehicle and perform in all dynamic events. As shown in Table 3 below, we were able to meet all of the design requirements.

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Actual</th>
</tr>
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<tbody>
<tr>
<td>Ride Height (before SAG)</td>
<td>At least 11&quot;</td>
</tr>
<tr>
<td>Track Width</td>
<td>56&quot;</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>60&quot;</td>
</tr>
<tr>
<td>Suspension Leverage</td>
<td>11:1</td>
</tr>
<tr>
<td>Camber</td>
<td>-1 to -3 degrees</td>
</tr>
</tbody>
</table>

Table 3: Design Requirement Comparison to Actual Performance
During and after the competition, several observations were made about the effectiveness of the design as well as possible alterations. Principally, the arm was made to be very strong. This was proven to be true as the arms not only survived the competition, unlike their predecessor, but only had slight deformation due to a direct impact on a sharp rock. Even with the impact the deformation was only slight and did not impact the effectiveness of the system. The downside of the strength is that the arms were incredibly heavy. While this did not have an immediate impact on the effectiveness of the system, the goal of the overall car is to be light as possible so minimizing weight in any location is optimal. To make the arms lighter, less material must be used so a further design would be arms that are straight without bends and have smaller dimensions. The arms could be shorter and thinner on both the top and bottom and a crossbeam was not necessary on the top arm at all. These modifications would also improve a large inadequacy with the car that was its steering angle. The car in general was very large and bulky compared to other schools. This is due to the fact that the GW had not gone to competition in several years and the previous iterations had not been competitive in the races at all. By decreasing the track-width and also making the arms narrower the steering angle and turning radius would be increased greatly. Another improvement would be making the arms asymmetric with the leading edge being at a closer angle to perpendicular to the frame than the trailing edge. This asymmetry would allow for a shorter front edge of the car while still keeping the tie-rods protected by the leading edge. Finally, having a suspension system where the axis of motion is off of vertical. This means that when the tire hits something the suspension system not only travels in the vertical axis but also the horizontal axis. The benefit of this would be that less shear stress is on the joints because any impact on the arms can translate into motion in two axis instead of just one. The easiest way to accomplish this would be having the mounting members on an angle themselves so that the arms are still square with their mounting points. Furthermore, eliminating mounting braces connecting the arms to the members would also minimize weight and track-width.

Like all designs, improvements can and should be made. In future competitions, the improvements listed would greatly improve the performance of the car. It would have been optimal for the designers to have previous experience with competitions or with Baja as many of the improvements were envisioned at competition. Despite these possible improvements, the suspension system fulfilled all of the desired goals and the team is very content with the design performance.