2013-2014 WPI SAE Baja Vehicle

A Major Qualifying Project
Submitted to the faculty of
Worcester Polytechnic Institute
in partial fulfillment of the requirements for the
Degree of Bachelor of Science

Submitted By:

______________________
Bertan Atamer

______________________
Julian Enjamio

______________________
Stephen Oliveira

______________________
Travis Van Dale

______________________
Jeff Wong

Approved By:

______________________
David Planchard, Advisor

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Abstract

Worcester Polytechnic Institute (WPI) was developing a racecar for the Baja Society of Automotive Engineers (BSAE) competition. Due to changes in the competition’s rules, specifically a change in the roll cage’s minimum metal tube wall thickness, the WPI racecar became ineligible to compete. This Major Qualifying Project (MQP) tested the BSAE racecar, identified deficiencies inherent to the frame’s design, then developed and fabricated a completely updated and legal BSAE racecar frame eligible for future competitions. Additionally, the team undertook an engineering outreach initiative by collaborating with Assabet Valley Regional Technical High School. In exchange for fabricating the frame, the team educated the Assabet Valley shop students about the engineering principals behind the new racecar’s design. Finally, the team validated the new frame by utilizing finite element analysis software, preforming a design review in accordance with SAE guidelines, and concluding with a comparison of the final deliverable to the previous BSAE frame design.
Acknowledgement

We would like to acknowledge the following individuals and organizations for their contributions and advice to our project:

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- Worcester Polytechnic Institute, for providing us the opportunity and resources to work on this project
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Chapter 1: Introduction

The Baja Society of Automotive Engineers (BSAE) competition is an intercollegiate, engineering design competition intended to challenge students to create and manufacture a competitive, off-road sporting vehicle. The competition simulates real-world engineering environments, where students must design, fabricate, and test their own vehicle, all while complying with the Baja SAE competition rules, and working within the constraints of their budgets. Additionally, the competition fosters working relationships with sponsors, professionals, and peers.

Each year, SAE Baja competitions are held at locations both across the United States, as well as internationally in countries including Korea, Brazil, India, and more. The competition consists of various dynamic events to measure vehicle performance, and static events to judge the engineering and economics of the vehicle. Dynamic events include acceleration events, hill climbs, maneuverability courses, rock crawls, and endurance races. The static events include written reports, presentations, and design evaluations presented by teams to SAE officials.

Vehicles participating in the Baja SAE competition must use an unmodified Briggs & Stratton - Intek 20 single-cylinder engine, with a power output of roughly 10 horsepower. [1]

Problem Statement

Worcester Polytechnic Institute (WPI) has dedicated previous Major Qualifying Projects (MQPs) to designing a vehicle for the Baja SAE competition. However, due to changes in the rules for the upcoming 2014 Baja SAE competition, the previous Baja vehicle is not eligible for competition. Specifically, the thickness of the roll cage members are not large enough to satisfy the new rules. Moreover, the previous vehicle also has room for improvement in certain areas, most notably the very large turning radius. Also, the tie-rods interfere with the previous vehicle’s frame when the wheel is turned in one direction, and collide with the shock when turned the other way, both factors contributing to limiting how far the driver can turn the steering wheel.

Due to the multitude of issues identified with the previous WPI Baja vehicle design, a new competition-eligible vehicle must be engineered and fabricated that improves upon the
previous WPI Baja vehicle. Suggestions will also be provided for improving the new vehicle for future MQPs.

**Design Goals**
The following are our design goals for the 2013-2014 WPI Baja SAE Vehicle:

1. Design and build a competition-eligible vehicle for the 2014 Baja SAE competition that satisfies all necessary Baja SAE competition rules. This includes redesigning and reassessing all major areas of the previous WPI Baja SAE vehicle: the frame, body, suspension, steering, and drivetrain systems.

2. Improve upon the previous WPI Baja SAE vehicle by reducing the overall weight and reducing the turning radius of the vehicle.

3. Incorporate a reverse gear into the drivetrain system for better maneuverability during dynamic events.

**1.2.1 Design Specifications**

1. Specified by page 10 of the Baja rules [1], the vehicle must accommodate up to the 95th percentile male with the following dimensions: 186.7cm (6ft 1 ½in) tall, 102kg (225lbs), and has an erect sitting height of 97cm (38 ¼in), while also complying with the SAE Baja competition rules for clearances.
Chapter 2: Background Research

2.1 Roll Cage

2.1.1 Material Selection

Rule B8.3.12 of the SAE Baja 2014 rules specifies that the material for the primary roll cage members must be steel with a carbon content of at least .18%. [1] Modulus of elasticity, tensile strength, yield strength, and density are important properties for materials utilized in the roll cage. Materials with high modulus of elasticity increase the bending stiffness of the frame which is desirable. High yield strength is necessary so the steel does not yield if impacted which is vital in securing driver safety. High tensile strength is also necessary so that the steel does not break if impacted, which could be hazardous to the driver. Low density steels will decrease the overall vehicle weight, which is one of the most important factors impacting the performance of the vehicle. Ultimately, however, safety is the primary consideration in the design, including the material selection. Table 1 lists common steels with over .18% carbon weight and their relevant properties [2].

<table>
<thead>
<tr>
<th>Material</th>
<th>Young's Modulus (Gpa)</th>
<th>Yield Strength (Mpa)</th>
<th>Tensile Strength (Mpa)</th>
<th>Density (g/cc)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI 1018</td>
<td>205</td>
<td>370</td>
<td>440</td>
<td>7.87</td>
</tr>
<tr>
<td>AISI 1020</td>
<td>200</td>
<td>295</td>
<td>395</td>
<td>7.87</td>
</tr>
<tr>
<td>AISI 4130</td>
<td>210</td>
<td>460</td>
<td>560</td>
<td>7.85</td>
</tr>
</tbody>
</table>

**TABLE 1 - MATERIAL PROPERTY COMPARISON**

AISI 4130 has more desirable material properties than the other steels across the board. The higher strengths and elastic modulus will allow us to construct a strong frame using less material, which is also less dense, reducing overall weight of the frame.

2.1.2 Roll Cage Structure

The structure of the roll cage is split into primary and secondary members per SAE Baja rule B8.3.1. The primary members include the Rear Roll Hoop (RRH), Roll Hoop Overhead Members (RHO), Front Bracing Members (FBM), Lateral Cross Member (LC), and Front Lateral Cross Member (FLC). The secondary members include the Lateral Diagonal Bracing (LBD), Lower
Frame Side (LFS), Side Impact Member (SIM), Fore/Aft Bracing (FAB), Under Seat Member (USM), and any other required cross members. Figure 1 indicates where many of these bars are located on the roll cage. [1] More precise definitions for these members can be found in the Baja SAE rules.

**FIGURE 1 - ROLL CAGE STRUCTURE**

The rules specify that certain bars must be continuous. Therefore, bending must be used over welding. This includes the vertical and lateral cross members of the Rear Roll Hoop (RRH) and the Front Bracing Members (FBM). In all other locations, tubes may be joined by welding, but must be reinforced by welding sleeves.

### 2.2 Suspension

The suspension system of a vehicle maintains contact between the tire and the driving surface and provides shock absorption. Without a suspension system, the slightest bump in the road would provide an upward force that lifts the tire off the ground since it would be attached to the frame. This would make handling difficult. A suspension system allows the tire to move up with the surface of the road while maintaining the frame height. Generally, different types of suspension systems are used for the front and back. Double-wishbone and A-arm suspension systems are often used in the front, while trailing-arm and semi-trailing arm are often used in the rear. [3]
2.2.1 Double-Wishbone

Double-wishbone suspension consists of two wishbone-shaped arms which have one attachment point on the knuckle and two on the frame. This system also consists of a shock absorber to manage vibrations. The advantages of a double-wishbone suspension is that, depending on the design, it can allow for greater control over the camber angle gain of the wheel which is desirable and can provide better steering. This is one of the more common suspension systems used on the front of the vehicle in the Baja SAE competition. The trade-off for a double-wishbone design is that it is often heavier than other suspension systems. [4] A schematic of a double-wishbone suspension system is shown in Figure 2.

![Schematic of a double-wishbone suspension system](image)

**FIGURE 2 – DOUBLE-WISHBONE SUSPENSION**
2.2.2 Single A-Arm

Single A-arm suspension consists of one A-shaped control arm that has one attachment point on the knuckle, and two on the frame, making an “A” shape. It is similar to the double-wishbone suspension, only with one control arm. The major advantage for this design versus the double-wishbone is cost, weight savings, and simplicity. [4] An example of a single A-arm is shown in Figure 3.

![FIGURE 3 - SINGLE A-ARM](image)

2.2.3 Trailing-Arm

Trailing-arm suspension consists of an arm connecting the frame and the wheel, with the arm in front of the wheel so that it is “trailing”. This type of suspension is used on the rear wheels. The advantages to this suspension are that it allows for only the vertical movement of the wheel, which is ideal for the rear wheels, and it is light weight and compact. [3] An example of a trailing-arm suspension is shown in Figure 4.

![FIGURE 4 - TRAILING ARM SUSPENSION](image)
2.2.4 Shock Absorbers
Shock absorbers, as the name implies, absorbs shock impulses when traveling over rough ground, releasing it at a controlled rate. Hydraulic shock absorbers and pneumatic shock absorbers are the most common types of shock absorbers used. Shock absorbers turn the kinetic energy of the movement of the suspension into heat energy that can be dissipated, which dampens the oscillatory motion of the vehicle. Hydraulic shock absorbers work by converting the kinetic energy of the suspension traveling upwards into heat energy, and dissipating it into the hydraulic fluids. Hydraulic shock absorbers generally use springs to absorb the energy, while the hydraulic fluid dissipates the energy. Pneumatic shocks work similarly, but instead use gas to absorb the energy. As the shocks absorb the energy, the pressure builds and the gas acts like a spring. Shock absorbers are a vital component of any suspension system as it evens out the ride and improves handling. [5]

2.2.5 Ride Height
The Jeep Wrangler [6] and Toyota FJ Cruiser [7] have about 10 inches of ground clearance, and both are capable of tackling terrain similar to that found in Baja competition. A ground clearance in this area should be able to handle the environment found the SAE Baja competition events.

A greater ground clearance allows for more travel in the suspension and reduces the possibility of damaging the underside of the vehicle or getting stuck. However, there is a tradeoff between ride height and stability. The greater the ground clearance of the vehicle, the worse the stability tends to be. [3]

2.2.6 Approach, Departure, and Break-over Angles
The approach, departure, and break-over angles help determine how well a vehicle would be at traversing obstacles. The higher the break-over angels are, the better the vehicle maneuvers over obstacles. Both the approach angle and departure angle are measured from the base of the wheel to the bottom of the bumper. They determine the maximum angle slope or size of obstacle that you can approach or depart from without causing damage to the vehicle or getting stuck. As Baja vehicles do not have a bumper in front of the wheels, this does not really apply. In theory, this means that the vehicle can approach or depart from any size
obstacle. The break-over angle will determine whether the vehicle can go over an obstacle such as a peak without getting stuck. This angle is measured from the base of the front or back wheel to the middle of the base of the frame. Ideally, a Baja vehicle would have a break-over angle of around 40° or more. This compares favorably to a Jeep Wrangler (25°) [6] and the FJ Cruiser (27°) [7].

2.3 Steering

2.3.1 Rack and Pinion

A rack and pinion assembly is the most common mechanism used for steering in vehicles. It consists of two gears, the pinion, a circular gear, and the rack, a linear gear bar. The pinion is rotated when the driver turns the steering wheel of the vehicle, which is connected to the pinion by the steering column. As the pinion rotates, it moves the rack in the direction of rotation. The rack is connected to the wheel by the tie rods, which attach onto the knuckle. As the rack moves, so do the tie rods, which push or pull on the wheels to turn the vehicle.
2.3.2 Camber
Camber is a useful characteristic that influences the handling of a vehicle. Positive camber requires less force to turn the steering wheel. However, positive camber is not as stable as no camber or negative camber. With zero camber a vehicle will accelerate faster in a straight line and with negative camber, the wheels have more grip and the vehicle is more stable when turning. [8] Therefore, under ideal circumstances, the vehicle would have no camber in a straight line, but when the wheels are turned, the vehicle would have negative camber on the outer wheels. A diagram detailing positive and negative camber is shown in Figure 5.

![Figure 5 - Camber Angle](image)

2.3.3 Caster
Caster is another useful feature that should be considered for the steering of the vehicle. In addition to being used to alter the camber when turning, it can enhance the damping of the suspension depending on how the shocks are set up. With caster, as the wheel is turned, the vehicle will gain camber. [8] The camber gain is negative compared to the direction the wheel is turning and this is ideal. Caster can also improve the tendency for the wheels to self-center. Figure 6 shows positive and negative caster.

![Figure 6 - Caster Angle](image)
2.3.4 Toe

The toe out configuration can be used to gain stability in a straight line. Additionally, toe out can compensate for the rolling tendency of the negative camber wheels. As an example of rolling tendency, as a spinning coin slows down, it tends to form a cone like shape. The coin wants to keep turning in the same direction to keep creating this cone like shape. This is called camber thrust and it is a type of twisting moment. Avoiding or reducing the rolling tendency is sought after because rolling tendency increases rolling resistance and wear on the wheels. [9] Toe can be adjusted fairly easily by adjusting the lengths of the tie rods.

By using a toe out configuration, the steering is impacted because the two wheels do not turn by the same angle as measured from the direction of travel. Regardless, having the outer wheel turn less than the inner wheel is beneficial. This is the idea behind Ackerman steering, which is used in professional racing. Therefore, the toe could be used as a “crude” way to achieve Ackerman steering. Figure 7 shows positive and negative toe.

![Figure 7 - Toe Angle](image)

2.3.5 Ackerman Steering

Ackerman steering is a type of steering where the front wheels turn at different rates. The inner wheel turns more than the outer wheel. In doing so, this reduces the amount of slippage in the wheels. Combining this with camber and caster as described above would mean that the front wheels gain different amounts of camber and this would allow the front of the vehicle to “lean” into the corner. However, because of the limited amount of time available to
the project, as well as the necessary additional research required to appropriately implement Ackerman steering, it was decided that it would be left to future MQPs to incorporate into the vehicle. [10]

2.3.6 Maneuverability and Stability

A tight turning radius is beneficial to vehicle maneuverability. Maneuverability plays an important role in off-roading because it allows the driver to change paths and pick better lines more quickly and easily. It also means that in a racing situation, the driver will not need to reverse as much in order to make a tight turn or clear an obstacle.

Under ideal circumstances, the turning angle of a Baja vehicle should be up to 40° from the direction of travel. This should provide an inner turning radius of around 9 ft and a turning radius of 12.5 ft when measured from the outer wheels. To put this into perspective, the Jeep Wrangler [6] and Toyota FJ Cruiser [7] have a turning radius slightly under 21 ft, and the Scion iQ [11] has a turning radius of 13 ft with a turning angle of almost 45°. As a result, this vehicle should be very capable in maneuverability events.

Stability can be thought of as the likeliness of the center of gravity to fall within an area covered by the base. To increase stability, the center of gravity could be lowered or the base size could be increased. The ride height can only be lowered to a certain amount otherwise the vehicle’s ability to maneuver over obstacles would be compromised. However, when the wheelbase is increased, the turning radius is also increased and this is undesirable providing a trade-off scenario.

2.4 Drive Train

The drive train is the system that turns transfers the torque output from a vehicle’s engine to a vehicle’s drive wheels. The drive train in a Baja vehicle consists of the transmission, the gearbox, and the axle shafts. The transmission is connected to the engine by the drive shaft. It changes the gear ratios which transform the input RPM and torque from the engine into the desired RPM and torque. When going uphill, higher torque and lower RPM is desired; this is achievable by adjusting the gear ratios. To travel quickly on the highway, torque can be lower but a higher RPM is mandatory. The transmission allows for the ideal gear ratio to be achieved
for different conditions. The gearbox is the set of gears where different gear ratios are achieved. Most automobiles have four forward gears and one reverse gear. The gearbox is connected to the wheels by the axles, which then drives the vehicle.

2.4.1 Continually Variable Transmission (CVT)

Previous MQP teams have custom fabricated a Continually Variable Transmission (CVT), which achieves different gear ratios using a set of pulleys. The pulleys consist of two cone shapes facing each other. A belt connects both pulleys and as the cones move closer or farther to one another, the belt rides higher or lower on the pulleys. As the pulleys of the CVT are pulled apart, the belt rides lower and effectively creates a smaller gear. This way, many different gear ratios can be achieved. The pulley connected to the engine is called the drive pulley, and the pulley connected to the gearbox is the driven pulley. As the radius of one pulley increases, the radius on the other pulley decreases to keep the belt tight. Figure 8 illustrates how the pulleys of a CVT work. The two cones of the drive pulley are pulled apart, effectively making a low gear, while the two cones of the driven pulley are close together making a high gear, ultimately creating a low gear ratio. [12]

![FIGURE 8 - IMAGE OF DRIVE AND DRIVEN PULLEYS OF A CVT [12]]
These factors make a CVT ideal for the transmission of a Baja vehicle. The seamless transition between gear ratios can improve vehicle performance under varying circumstances as opposed to a single gear system, optimizing performance. Despite not using any gears, a gearbox is still required to provide a reverse gear as the CVT is not capable of switching the direction of rotation on its own. [13]
2.4.2 Rear Wheel Drive vs. Front Wheel Drive

Rear wheel drive is more common in Baja SAE due to it generally being easier to mount the motor in the rear rather than the front. Therefore, rear wheel drive is simpler to implement. Rear wheel drive requires fewer components for a vehicle with a rear-mounted engine, thus reducing the overall weight of the vehicle. Specifically, a rear wheel drive vehicle does not require a differential because the rear wheels tend to travel similar distances when turning as opposed to the front wheels. Vehicles with rear mounted engines also generally have better weight distributions than vehicles with front mounted engines, which can improve handling. Rear wheel drive vehicles also tend to have better acceleration because as the weight shifts to the rear wheels, traction is improved.

On the other hand, front wheel drive has considerable advantages in off-road conditions. Front wheel drive vehicles have better traction when turning and over slippery surfaces. This makes front wheel drive ideal for many dynamic Baja events such as the rock crawl which require traction on the front wheels. There is a major trade-off between front wheel and rear wheel drive, as front wheel drive can be better in some off-roading circumstances due to increased traction on the front wheels, but increases the overall weight of the vehicle due to the necessity of additional components such as the differential. [3] Table 2 shows the positives of each type of suspension.

<table>
<thead>
<tr>
<th>Rear</th>
<th>Front</th>
</tr>
</thead>
<tbody>
<tr>
<td>• More even weight distribution</td>
<td>• Better traction in off-roading conditions</td>
</tr>
<tr>
<td>• Weight transfer during acceleration</td>
<td></td>
</tr>
<tr>
<td>• Less components (no differential necessary)</td>
<td></td>
</tr>
<tr>
<td>• Easier to mount engine in rear of vehicle</td>
<td></td>
</tr>
</tbody>
</table>

TABLE 2 - REAR VS FRONT WHEEL DRIVE TRADE-OFF
Chapter 3: Methodology

Creating a new vehicle from scratch allowed for a high degree of freedom in the design of the vehicle; provided the finished product followed the Baja SAE rules. However, logical reasoning as to why the new design elements were an improvement over the previous design was required. As a result, the previous WPI SAE Baja vehicle needed to be analyzed and tested in order to determine areas where the vehicle could be improved upon.

The new design was split up into the following main categories: frame, body, suspension, steering, and drivetrain. Each of these categories was inspected on the existing vehicle in order to assess possible improvements that could be incorporated in the new design. Once design features were agreed upon, the design was realized using the computer-aided design software SolidWorks. Simulations were carried out on the design to evaluate the structural integrity of the frame and validate design decisions made.

3.1 Previous Vehicle Testing

The previous WPI SAE Baja vehicle was test drove in order to determine what areas of the vehicle needed to be improved upon. During testing, it was discovered that the original vehicle had a 40ft turning radius, which was excessive. This large turning radius was due to a combination of poor design in the frame and subpar shocks. As a result of the frame’s design, the tie rods hit against the frame if the steering wheel was turned too much in one direction and the tie rod would hit against the shocks if it was turned too much in the other direction. The tie rods hit against the shocks because of a poor mounting point with the frame. Furthermore, the shocks themselves leaked air and the front end dipped as the vehicle was turned, worsening the situation. This caused the vehicle to have severe positive camber of up to 20°, hindering the stability and the maneuverability of the vehicle. Ultimately, testing ended prematurely due to a tie rod snapping because of the issues described earlier.

Additionally, the previous vehicle was weighed at 425 lbs. The weight distribution was roughly 180 lbs in the front and around 245 lbs in the rear. This translates to a percentage weight distribution of 42% – 58% front-to-back. Interestingly, the left side of the vehicle weighed 15 lbs more than the right. While, the weight distribution is agreeable, the total weight
of the vehicle is considered excessive, especially considering the fact that the vehicle did not have a body.

3.2 Frame

3.2.1 Safety

Safety is the top priority in the design of the frame. Most of the SAE rules pertain to safety. Therefore, the existing frame was inspected for compliance with these rules. Modifications to the rules between the 2013-2014 competitions intensified the safety requirements, making the previous WPI SAE Baja vehicle ineligible for competition. For example, the existing vehicle had wall thicknesses in the tubes that were thinner than the minimum allowed thickness due to the rule changes. Additionally, certain clearances and angles were no longer permitted due to the new rules. Therefore, the existing vehicle could not be taken to competition and the design of a new frame was necessary.

First, to ensure that the new frame satisfied the competition rules, the wall thickness of the new frame was doubled to 0.062 in for all roll cage members. The minimum tube diameter on the new frame is 1 in. A thickness of 1.25 in was used on the primary members of the roll cage. These tubes replace several tubes that had an outer diameter of half an inch or less in the previous vehicle. As a result, the frame would have a higher stiffness which is desirable for both safety and performance.

Several potential safety issues were considered in the design of the new frame. One safety issue with Baja vehicles is that they do not have bumpers, so in the event of frontal impact, a substantial amount of the force will be transferred to the rest of the frame and driver. To reduce these forces, a sloped bar was placed at the front of the vehicle as shown in Figure 9 on the following page. This will help deflect certain obstacles that the driver may encounter. Furthermore, the wheels of the vehicle were moved 8in forward than in the previous design. If the vehicle were to crash into a large object, the wheels would be the first elements to strike. The tires could absorb some of the impact and distribute the forces along the rest of the suspension, effectively acting as a front bumper.
Deflecting rear impacts was also a consideration in the design of the new frame. The rear of the new vehicle is triangular and slanted by 30°, as depicted by Figure 10, in order to reduce the effects of rear collisions. This is accomplished because a triangle is structurally stronger than a square, therefore it increases the stiffness in the rear and because it is triangular and slanted, it is more likely to deflect a vehicle or object that crashes into the back of the vehicle.
Side impacts are another consideration that must be designed for. The side impact member is the bar that absorbs the majority of the impact in this type of collision. This makes up the widest part of the vehicle, connecting the rear roll hoop to the front of the vehicle, indicated in Figure 11. The side impact member is only considered a secondary member of the roll cage, however, it was treated as a primary member of the roll cage since it would be struck in the event of a side impact, and thus it is vital in protecting the occupant. By strengthening this member, the vehicle is 20% stiffer and 15% safer by factor of safety in the event of a side impact.

FIGURE 11 - SIDE IMPACT MEMBER, TOP VIEW

The side impact member forms the widest part of the vehicle, which can be taken advantage of to distribute the forces throughout the frame more evenly. A mounting point was designed for the rear suspension to connect to the side impact member and rear roll cage, shown in Figure 12. The width allows us to distribute the forces of the shocks more evenly throughout the frame.
To improve the rigidity of the entire frame, two tubes were added to the sides of the roll cage for added safety, shown in Figure 13, which was an important aspect of the new design. These bars are not strictly required by the competition rules, but despite this, they improve upon the structural rigidity of the frame, improving the overall safety of the vehicle. Although this adds to the weight of the vehicle, the improved safety of the vehicle outweighed this.

The last safety consideration was the amount of clearance between the tubes of the frame and the driver. All clearances were improved compared to the original vehicle. In the event of a crash, these tubes can now bend more without injuring the driver. For example, there is 4in more legroom and space between the frame and knees of the driver. There is 1in
more space between the shoulders of the driver and the frame, and most importantly, there is 1.5in more space between the helmet of the driver and other parts of the frame.

3.2.2 Weight

Weight was second in priority to safety in the design of the frame. Because every vehicle in the Baja competition has the same engine, the weight of the frame drastically impacts the performance. Therefore, a goal was to reduce the weight of the vehicle when the new frame was designed. However, various changes needed to be made in order to improve safety and comply with the SAE Baja rules as detailed in the previous section. This posed a major challenge: double the wall thickness of the tubes used for the frame, while also reducing the overall weight of the vehicle.

In order to reduce the weight while increasing wall thickness, as few tubes as possible had to be used. However, this also needed to be checked with the rules to ensure compliance. Furthermore, the vehicle had to be safer than the initial vehicle. Ultimately, there was a tradeoff between the amount of bars used and the safety of the vehicle. Still, in some areas, a few of the bars used for structural support serve a dual purpose. Figure 14 shows that one of the bars used in the front serves as the tow hitch, but also adds structural support and skid protection in the event of a collision.

![FIGURE 14 - FRONT TOW HITCH, TOP VIEW](image)
Another instance of this weight reduction tactic is visible in the rear of the vehicle. Previously, the vehicle had a large square rear where the drivetrain was placed. By switching to a triangular shape, fewer bars were used and the lengths of the bars were also reduced.

The previous vehicle also had several small bars running across different members. These were cut out and the design was simplified. The previous vehicle had a body with many variations in its cross section if viewed from the front to the back. The small bars were required to achieve multiple cross sections, however, they added complexity to the design, increased the weight, and created areas of stress concentrations.

3.2.3 Material Selection

Material selection impacts both the weight of the frame, its rigidity, and its safety. Therefore, this was an area of utmost importance. The original frame used AISI 4130, however, an analysis was conducted to determine which steel alloy provided the best combination of strength and lightness.

To determine the best material for tube diameter and wall thickness, an iterative method was used to find the best strength-to-weight ratio for frame components. First, it is best to look at the rules and see what restrictions exist since these restrictions were most likely put in place for safety reasons. When selecting steels, one will find that the elastic modulus is always the same and there are only very minor variations in alloy density. The rules specify the bending stiffness as no less than that of a 1 in diameter steel tube made out of AISI 1018 alloy with a 0.12 in wall thickness. It becomes important to focus on this rule because the equation for finding the bending stiffness (M) is the product of the elastic modulus (E) and the second moment of area (I).

\[ M = EI \]

EQUATION 1

Since all elastic modulus for steel are the same, the only way to change this result is to change the second moment of area which is a property of the tube cross section.

The second moment of area “I” for a hollow tube is given by the equation
where “D” is the outer diameter and “d” is the inner diameter of the tube. The best way to increase the second moment of area while decreasing the cross-sectional area of the tube is to increase the outer diameter and decrease the wall thickness. To reduce the number of iterations needed, it is best to reference the rules which dictates the minimum wall thickness of a roll cage structural member be no less than 0.065 in. Next, one must consider the bending stiffness which must be no less than that of a 1 inch diameter steel tube made out of AISI 1018 alloy with a 0.125 in wall thickness. One can further limit the number of iterations by looking up the commercially available sizes of steel tube which move up in 0.125 in increments. Since the rules also state a minimum tube diameter of 1 inch and anything over 1.5 in is heavier than the 1 in due to the 0.062 in minimum wall thickness, the iterative solution will only need to focus on tube diameters of 1, 1.125, 1.25, 1.5 in. 1.375 inch tubing was omitted because it was not commercially available in the desired wall thicknesses in AISI 4130. Using the cross-sectional area as the indicator of tube weight, solution results showed a 1.25 in diameter tube with a 0.062 in wall thickness made of either AISI 1018 or AISI 4130 would satisfy the bending stiffness requirement while weighing the least.

To decide which alloy to use and confirm the eligibility of the material, the bending strength must be checked. The rules state the equation for bending strength to be

\[
B = \frac{S_y \times l}{c}
\]

**EQUATION 3**

where “\(S_y\)” is the yield strength, “\(l\)” is the second moment of area, and “\(c\)” is the distance between the load and the most extreme fiber which for the same component would be constant so a value of 12 in was assigned as “\(c\)” for all calculations involving bending stiffness. The yield strength of the control material, AISI 1018, is 52,939 PSI. After looking up the
materials AISI 4130 and AISI 1020 their yield strengths were found to be approximately 102,000 PSI and 50,800 PSI respectively.

After calculating the projected bending stiffness for each alloy using the selected tube dimensions produced in the previous calculations, it was evident that AISI 4130 was the best candidate and worth the extra cost. The AISI 4130 tubing with a 1.25 in diameter and a 0.062 in wall had a bending stiffness that was around 30% better than the standard given in the rules and a bending strength that was around 200% better than the standard given in the rules and the same 1.25 in tubing made out of AISI 1018 instead of AISI 4130. In addition to being much stronger than the other steels, AISI 4130 is slightly less dense. This allows us to use less material and maintain or improve the structural rigidity of the frame, while also reducing weight. The results of the calculations can be found below in Table 3.

<table>
<thead>
<tr>
<th>Material</th>
<th>D (in)</th>
<th>d (in)</th>
<th>Area (in^2)</th>
<th>Bending Stiffness</th>
<th>Bending Strength</th>
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<td>0.032711</td>
<td>0.331752115</td>
<td>971.5095292</td>
</tr>
<tr>
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<td>0.032711</td>
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<td>1.01</td>
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<td>0.425999875</td>
<td>2042.222634</td>
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<tr>
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<td>1.126</td>
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<td>1215.733859</td>
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<tr>
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<td>0.995</td>
<td>0.030516</td>
<td>0.216455689</td>
<td>906.3108594</td>
</tr>
<tr>
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</tr>
</tbody>
</table>

TABLE 3 - ITERATIVE CALCULATIONS FOR TUBE DIMENSION AND ALLOY SELECTION

For secondary tubing the rules dictate a minimum wall thickness of 0.032 in and a minimum tube diameter of 1 in. For ease of assembly, it is always best to use similar alloys when welding is used as the primary method of joining. By following the rules, looking to maintain maximum strength, and facilitating the welding team, it was decided that all
secondary frame members would be made out of AISI 4130 tubing with a 1 in outer diameter and a wall thickness of 0.032 in.

Before selection of the material could be completed it was important to notice the recent rule change which almost doubled the minimum wall thickness of primary roll cage members from 0.032 in to 0.062 in. This change was made in the pursuit of increased safety of the roll cage. The minimum tube diameter is an important rule to consider because while calculations are done assuming the tubing is uniform; the reality is there could be unknown defect in the wall thickness of a section of the tubing. As wall thicknesses get thinner the effect of these defects get amplified and if the wall is too thin, these wall thickness variations could cause an unexpected failure on a frame member that was designed with a small factor of safety. The previous WPI Baja frame utilized the 0.032 in wall thickness to save on weight while having an increased outer diameter to maintain bending stiffness. While the frame is structurally sound, the new rules make it ineligible for future competitions. As unlikely as major tube defects are it is always important to be prudent and make sure the material you are selecting has reasonable tolerances. When ordering the tubing from MSC Metals, a Mill Test Report was asked for giving information on the specific batch of steel that was received based on the testing of a random sample from the mill at the time of production. A copy of the report along with a visual inspection for defects upon delivery ensured that the material selected would perform as calculated.

Furthermore, the front suspension uses AISI 4130. Originally, aluminum alloys, either 6061 or 7075, were considered for use in the suspension. 7075 is stronger and lighter than the 6061, but it is not as easily welded as 6061, which was the chosen method of construction for the suspension arms. However, the original design planned on using a double A-arm configuration in the front of the vehicle. With the switch to a single A-arm suspension configuration, there are less arms to distribute the forces that the suspension will sustain when landing, requiring the arms to be stronger. Thus it became impractical to use either aluminum alloy, and AISI 4130 was chosen for its higher strength.
3.2.4 Structural Considerations
In addition to the material selection, the method used to create the frame was also considered as it would impact the structural rigidity. The SAE rules dictate locations that must have bends as discussed in the background of the report. Using a bend instead of welding the tubes produces a 20% sturdier frame. Consequently, the team elected to bend as much of the frame as possible.

Circles or semicircles are good shapes to work with since they lack corners. Corners tend to act as stress risers and by eliminating or reducing the corners, one is also reducing the stress risers. Welding involves joining two or more metal parts together. However, when this is done, it creates a corner or an edge, where stress concentrations can occur. Furthermore, by bending, one is not joining two separate bodies so the end result is tougher as the weld is something that is more likely to fail. Thus, by bending instead of welding, the loads are distributed more evenly amongst the tubes, improving the rigidity of the frame. This also increases the life of the frame, as fatigue cycles tend to wear down the parts under the highest stresses.

In areas of the frame where welds are needed, there are usually three tubes that join together to create a triangle. Like the circle, a triangle is a good shape from a structural standpoint as it is the next shape with the fewest corners. A triangle cannot deform without changing the length of one of its sides. What this means is that a triangle less likely to twist than a rectangle or other polygon. Another advantage of the triangle is that is can be slanted and this increases the likeliness of something to deflect off of it when it is hit.

Another feature used to improve the stiffness of the frame is the gusset plate. Gusset plates serve two purposes: they help join two different metal parts together and they also strengthen the joint. These are usually used in the trusses of bridges and other stiff structures. Gussets were used along the welds and in corners of the frame of the vehicle in order to strengthen the frame these areas. Figure 15 shows an example of a suitable location for a gusset plate.
3.2.5 Accessibility

The previous vehicle suffered from poor accessibility due to tight clearances and the positioning of the tubes behind the roll cage loop that connect to the drivetrain. While this has been discussed in the safety section of the report, these tight clearances also made it difficult for the driver to get out of the vehicle in less than five seconds, a requirement for the competition. Additionally, it was difficult at times to work on the drivetrain. Many metal tubes of the suspension arms were in the way and fitting tools into these tight clearances was a hassle. The easiest way to work on certain parts of the drivetrain was to either remove the seat and firewall or remove the arms for the rear suspension and then extract the necessary part once the engine was pulled out. To solve this, the area at the rear of the new frame was enlarged. The engine can be removed by lifting it upwards from the area shown in Figure 16. This setup makes it easier to tune components of the drivetrain or fix any parts in the rear of the vehicle.
3.2.6 Ergonomics and Comfort
To improve ergonomics, two major aspects were added to the design of the frame. First, the final design was lengthened to provide more legroom for the driver. Second, the rear roll hoop was tilted at a slight angle (as opposed to having it vertical) to make the driving position more comfortable. These alterations created a better and more comfortable driving position but added weight to the frame. Nevertheless, if the driver is not comfortable, the performance of the vehicle would suffer (especially during the endurance event) either because the drivers would need to be switched more frequently or the driver would be more careful going over obstacles so as not to harm themselves.

3.2.7 Manufacturability
Simplicity in the design was important in order to make it easier to manufacture. However, due to a combination of the material choice and bend usage instead of welds in certain locations, the overall manufacturability was negatively impacted. The steel that was chosen has a tendency to wrinkle when bending because of its high stiffness and the relative thinness of the walls of the tubes. To improve manufacturability, only the tubes that were 1.25” in diameter were bent, whereas the 1” diameter tubes were welded. However, this still
resulted in the need to mandrel bend the tubes, which was outsourced to Worcester Manufacturing Inc.

3.3 Body

Modern racecar bodies are usually made out of the lightest materials with only enough strength to accomplish the task of holding their shape for aerodynamic performance during the race. Composite materials accomplish this objective very well due to their low density and high tensile strength. The best composite material would be carbon fiber. However, carbon fiber materials are very expensive when compared to other materials and was not feasible due to budget. It should also be noted the carbon fiber fabrication would have involved equipment, like a large autoclave, that our team would not have had access too. For that reason, fiberglass was selected for the Baja body. Fiberglass follows the same composite theory as carbon fiber, where a material made out of glass fibers is soaked in a resin which is allowed to harden, forming a component that carries the properties of both the resin and the glass filaments.

The SAE team already had two 200 yd² rolls of fiber glass, which reduced the final cost of the body panels. Resin and hardener still needed to be acquired along with material to make molds and form the fiber glass. For mold construction it is best to use an ultra-high density urethane (UHDU) foam because it has a density similar to wood, allowing for a great surface finish and shape accuracy, while still being easy to form using a subtractive method like cutting and sanding. UHDU foam is very expensive at over $1000 per cubic meter and therefore not within the constraints of our final budget. Since every panel of our body was flat, our team was able to substitute 2 inch thick pink insulation foam sourced from Home Depot for $40 a sheet for the originally planned UHDH without much impact on mold performance.

The panels of the body were designed to be custom fit into the spaces of the frame to display the frame geometries without comprising vehicle aerodynamics and driver safety. This design theory also significantly reduced the amount of forming that needed to be done to the foam in order to create the molds. The bottom floor pan was made from a 0.035” sheet of aluminum instead of fiber glass it provide better energy absorption due to its more ductile properties. This decision was made because during the endurance race, a large amount of small
objects, mostly rocks, are expected to get kicked up underneath the vehicle favoring a material that provides a better toughness. The side panels were made of 3 ply fiberglass to save on weight while still being able to deflect small rocks kicked up by the tires which is expected to happen at a much less frequent rate than underneath the car. All panels will be made rigid enough to withstand all static and dynamic forces experienced during the competition.

The molds for the body panels were a positive of the shape required to fit in the frame openings. The mold was formed by cutting their general shapes from sheets of 2” foam, then coating the mold with a latex paint to protect the foam from the resin. Each mold was labeled to its specific frame location since the frames larger tolerance generated a scenario where each panel was slightly different from its counterpart. This also meant the final panels were not interchangeable. After the molds were used they were discarded because some of the resin got through their base and started to degrade the foam. A future recommendation would be to plastic wrap the molds to ensure the resin does not contact the foam and make final removal easier.

Creating the panels from fiberglass involved a lot of preparation including treating the molds with WD-40 so the final panels would not adhere to the mold, cutting the glass fabric shapes so they were ready to be laid on the molds, and ensuring the workspace was clean and well ventilated to reduce contamination and ensure no buildup of fumes while the resin dries. Once the prep work was completed and checked off, the resin and hardener were mixed. A proper ratio of 10 mL to 4oz was required to ensure the resin would cure properly but not cure too fast that there was not enough time to lay the fabric on the mold. Once the glass fabric was saturated with resin, it was laid on the mold and a flat scraper tool was used to remove any air bubbles between the layers of fabric and between the fabric and the mold. Once the resin had cured, the hardened panels were separated from the molds and excess material was cut off using an angle grinder. Panels then received small holes so they could be secured to the frame using hose clamps and screws. Panels were then checked to ensure proper fit in their designed locations.
After the panels passed their final fit check, they were readied for paint. Each panel was leveled and prepped with a layer of bond, 60 grit sandpaper and finished with 300 grit sandpaper before paint to ensure proper paint adhesion and a good final surface finish. An acrylic enamel primer and base color coat were used to paint the panels with the same color tone as the formula vehicle body to ensure the two WPI competition vehicles had the same color scheme. After the base color had completely out gasses a flexible matte finish clear coat was applied to protect the enamel and give the Baja vehicle an off-road style matte finish in contrast to the gloss finish on the formula vehicle. After paint and decals were applied, the body panels were stored to only be put on after the rest of the vehicle was assembled to ensure they were not damaged during the installing of mechanical components.

3.3.1 Mounting to the Frame

To secure the panels to the frame a method was employed which improved panel fit and endurance during the race. Hose clamps provided the ability to fasten the panels to the frame without having to add gusset plates or drill into the structural tubes. Panels were drilled so a bolt could pass through the hose clamp and panel, securing the panel to the frame. This fastening approach is very adaptable and easy to adjust during the competition.

3.3.2 Firewall

The Baja SAE rules specify that a firewall must be between the cockpit and fuel tank compartment for safety reasons. The firewall occupies the entire area between the lower and upper lateral cross members on the rear roll hoop, completely separating the driver from the engine and fuel tank. The firewall was made of a 0.5 mm thick Al 2024 aluminum sheet. For additional safety, the rules also stipulate that a fire extinguisher be present in the vehicle cockpit, located below the driver’s head on the right side of the firewall and half of it must be above the side impact member. A Baja Frame with a firewall is shown in Figure 17.
3.4 Suspension

Single A-arm and double wishbone type suspension arms were considered for use in the front suspension, while trailing arm and semi-trailing arm was considered for the rear suspension.

3.4.1 Single A-arm Design

The original suspension that was designed for this vehicle incorporated double A-arm in the front of the vehicle. The advantages to this type of suspension is that it distributes stresses more evenly because of the additional connection points compared to a single A-arm design, reducing areas of stress concentrations. This type of suspension can also have adjusted camber, caster, and toe depending on how it is designed. These characteristics could improve the handling of the vehicle and they are discussed in the steering section of the chapter.

However, the front double A-arm was modified to a single A-arm configuration. The advantages of the single A-arm over the double A-arm is that it weighs about 30% less and is easier to manufacture. Additionally, since there are fewer parts, there is less of a chance that it could fail. An Original Equipment Manufacturer (OEM) knuckle was used as this eliminates the amount of machining that would otherwise be required. Furthermore, in the event that the part fails, an identical, spare part can immediately replace it. A bracket was placed in order to hold the knuckle in place.
Camber and caster on the front wheels was added to the single A-arm design by adjusting the angles of the brackets that hold the knuckles in place. In a straight line the front wheels have no camber, but gain camber as the wheels are turned. This is the same idea behind the previous front suspension design. Figure 18 shows a render of the new front suspension arms.

![FIGURE 18 - FRONT A-ARM](image)

### 3.4.2 Trailing Arm Design

Since the vehicle is rear engine mounted, around 65% of the weight will be in the rear, and the vehicle will land on its rear after coming off a jump. This requires the rear suspension be able to cope with up to 5Gs worth of forces. For example, when the vehicle goes off a jump, it should land on its rear wheels to prevent the vehicle from rolling over forward. As a result, the rear suspension needs to be able to be strong enough to cope with these forces.

Trailing arm suspension was chosen to be used in the rear of the vehicle because of its strength relative to its weight compared to other suspension types. One of the advantages of using a trailing arm is that it can be stronger per unit mass than the double wishbone suspension. Furthermore, trailing arm suspension eliminates the need for a knuckle and ball joints that are otherwise required in a double wishbone setup. By eliminating these components, a trailing arm is also simpler than a double wishbone suspension.

A trailing arm type suspension could also provide greater clearance depending on how it is designed. Imagine removing the lower arm of the double wishbone suspension to achieve a
trailing arm design. By doing so, one can increase the distance between the ground and the arm that reaches out to the wheel from the frame.

3.4.3 Shocks

Much consideration went into the selection of the type of shocks to be used. The previous vehicle used pneumatic shocks. These shocks were pumped up with air and the more they were pumped, the higher the ride height should have been. However, because they had a leak, they never maintained their internal pressure and ultimately caused the front end to dip. When the front end dipped, the shocks interfered with the tie rods, preventing the vehicle from turning. To avoid this in the future, pneumatic shocks with a redundant spring back-up were chosen in the event that the pneumatics failed.

The shocks had to be strategically placed in order to ensure that they did not damage the frame or interfere with the tie rods like they did on the previous WPI SAE Baja vehicle. Accordingly, the shocks were mounted behind the tie rods. Although this exposes the tie rods in the event of a frontal impact, the risks associated with this are low. The wheels extend 3in ahead of the front of the frame, so they would be the first parts to hit a large obstacle in front of the vehicle. Furthermore, the lower arm of the double wishbone is more likely to hit an obstacle before that obstacle hits the tie rods.

The shocks are mounted on points of the frame that are well supported. For instance, the rear shock mounts to the rear roll cage loop and side impact members. The rear roll cage loop is one of the strongest parts of the frame and the side impact members help to distribute this load along the length of the frame. In the front, the shocks are mounted to the side impact member and also to another bar in between the two side impact members. This additional bar prevents or reduces the bending or compression at that point. Furthermore, this bar is used to hold up the steering wheel, saving weight.

3.4.4 Ride Height

The intended ride height of the vehicle is 14 inches in the front and 10 inches in the rear. This should be more than enough to clear the toughest obstacles during competition and compares favorably with other off-roading vehicles. The ride height is higher in the front than
the back to lessen the negative impact on stability that raising the ride height would have. The wheelbase and track was expanded to maintain good stability. By keeping the rear end of the vehicle lower down, the center of gravity is lowered, especially considering that most of the weight is in the back. Also, by slanting the vehicle slightly, the bottom of the vehicle essentially acts as a skid plate. This situation is ideal because it allows for greater travel in the front suspension in the front, as well as a greater ride height to clear obstacles while also lowering the center of gravity to 16.5 in above the ground. Since the trailing arm has a better ground clearance than the double wishbone in the front, the extra ground clearance in the back can be sacrificed.

3.5 Steering

3.5.1 Camber
As discussed in the background section, a vehicle with no camber while moving in a straight line, but negative camber on the outer wheels when turning is ideal. To achieve this, caster was used in order to give the wheels camber when turning, while maintaining no camber when the wheels are parallel to the direction of travel. This is similar to the front wheel of a motorcycle.

3.5.2 Toe
For the purposes of this project toe was ignored, as optimizing the amount of toe is very difficult. This can be a topic for future projects because toe also influences the acceleration and steering characteristics of a vehicle. The optimum value will range anywhere from no toe to negative toe of up to -10° as it can balance some camber thrust issues that might be encountered.

3.5.3 Maneuverability and Stability
To achieve a tighter turning radius and improve maneuverability, the arms of the front suspension were carefully designed. However, decisions made to increase the stability of the vehicle have some negative implications for the turning radius of the vehicle described below.

Designing for stability is difficult because the best ways to increase stability is to have a lower center of gravity or a wider wheelbase. However, the ride height can only be lowered a
certain amount, otherwise the vehicle’s ability to maneuver over obstacles would be compromised. Additionally, the roof cannot be lowered because of SAE rules so the remaining option is to increase the size of the base. The wheels are designed to be as far apart as the rules allow. The wheelbase is shorter than the maximum allowed by the rules of 162 cm (64 in), but this was done to improve the break-over angle as discussed previously. However, having a wider wheelbase increases the turning radius of the vehicle. This is compensated for by increasing the turning angle of the front wheels.

3.5.4 Front Wheel Drive
The advantages of front wheel drive and rear wheel drive systems were researched in order to determine the best system for the Baja competition. Because the motor limits these vehicles, the main advantage of the weight shifting to the rear is not very significant. Moreover, at such low speeds, drifting is not a major consideration. Furthermore, front wheel drive is better than rear wheel drive on slippery surfaces like ice. Front wheel drive would allow the driver to change direction more easily through obstacles because the front wheels are driven. The old Minis, which were well known for their rallying performance, used front wheel drive. However, the main advantage of rear wheel drive it that it is simple. A differential is not needed in the rear wheel drive setup, but would be more desirable in a front wheel drive system. Adding a differential would in turn add weight and complexity. Also, the weight distribution would shift further to the front and this could be unwanted. Coming off a tall jump in a vehicle that is nose heavy would cause it to dip forward and land on the front whereas landing on the back is preferred. However, if the Ackerman steering were to be designed correctly, a differential could be eliminated and this poses a huge opportunity for future projects.

3.6 Drivetrain
Last year’s MQP team created a CVT that was optimized for the engine’s power curve characteristics. Since the rules dictate the same engine be utilized, the CVT from the previous vehicle was carried over. A gear box was also sourced to add a reverse gear to the vehicle which will help the WPI team get through the obstacle course. Similar to the previous vehicle, an aluminum metal plate will be used to secure all the drivetrain and engine components. The benefits of the metal plate are that it provides a strong base to resist engine and drive train...
torqueing and protects the mechanical components from debris during the dynamic competitions. Axels with CV joints were used to connect the gear box to the rear wheels. This configuration will provide only rear wheel drive which should be sufficient for racing requirements as a 4 wheel drive application would cause too much drivetrain drag for the supplied engine.

3.7 Budget

The plan to construct a complete vehicle put a lot of strain on this year’s budget. The new frame and body were given top priority this year as they needed to be completely redesigned, the frame to meet new regulations, and the body to fit the new frame. Secondary priority went to the suspension which we were looking to upgrade in order to make handling much better during the dynamic events than it has in the past couple of years. Next, money was allotted for new wheels and tires, as well as front and rear knuckles to mate the new suspension to the wheels. Finally, any extra money was devoted to purchasing mechanical components and connections that would need replacement.

Certain components like the driven pulley of the CVT, CVT belt, axels, and shocks had not aged well and would need to be replaced. Replacement parts proved difficult to track down but not too expensive to purchase when compared to buying a completely new system. Other components like the braking system, steering wheel and rack, racing seat and drive pulley of the CVT were still in great condition and could be reused on the new vehicle. There were also a new engine and functional gearbox which saved us from having to purchase these items. Miscellaneous expenses like shipping costs and testing supplies were budgeted for with whatever resources we had left to purpose. While tight, the budget was manageable because of the components that were reused and the sourcing of components and material from suppliers who offered free delivery to the school such as MSC Industrial Supply Co.

The production plan turned out to be flawed because it was not expected that Assabet valley would not be able to bend the tubes, so a vendor with a mandrel needed to be contacted to ensure our tubes were properly bent. After searching several vendors, it was discovered that Worcester Manufacturing had a rotary bender and could fit us in right away because the
mandrel for their current production run had broken. After some quick material movements and a day in the shop the tubes were bent and the frame back on schedule. Worcester Manufacturing was also kind enough to donate their services so our budget did not need to be adjusted to accommodate any service fees. The budget proved to be flexible enough to handle all the issues that have been presented throughout this project. Either by reusing extra components or making the parts that would have been purchased, enough resources were freed to complete our main objectives.
Chapter 4: Results and Analysis

4.1 CAD Simulations - Frame

Simulations were carried out using SolidWorks in order to evaluate the structural integrity of the frame and suspension to validate design decisions made. Below are the results of these simulations. Calculations were completed to determine forces that the vehicle could be expected to experience in worst case scenarios during the Baja competition. In all simulations, a fixed geometry was used on the opposite sides of where the forces were applied. These locations were where the suspension arms and shocks connect to the frame.

4.1.1 Frame Impact Force Calculation

For the purposes of the simulation, the worst-case scenario was considered. In such a case, the collision is perfectly inelastic. In a real world application, this would indicate that the shocks on the suspension are fully compressed at the time of the collision. As a result, they will simply hold certain parts of the frame in place during a collision. Furthermore, the assumption is that the tires will not slip and the suspension arms will remain rigid and not buckle under the load. As a result, all the forces from the collision will be exerted on the frame.

A few more assumptions were necessary in order to estimate the amount of forces that the frame experiences during a collision. First, it is assumed that after the frame collides with an obstacle or other vehicle, the time \( t \) it takes to come to rest \( (v_{final} = 0) \) from its maximum velocity is 0.1 s. This is an aggressive estimate. Most vehicles have crumple zones and impact attenuators, which this vehicle does not. Usually these are in the order of hundreds of milliseconds. The role of these elements is to prolong the duration of the impulse the frame experiences during an impact. Second, the maximum velocity \( (v_{initial}) \) of the vehicle before impact was 60 km/h or almost 40 mph. This is a bit higher than speeds for the National Highway Traffic Safety Administration crash tests but it is also close to the maximum velocity a Baja SAE vehicle is expected to experience. Finally, one must estimate the full weight of the vehicle. A heavy driver was estimated to weigh 195 lbs driver, making the total mass \( (m) \) of the system is 270kg (595lbs).
Given the conditions above, one can apply some basic formulas to reach an estimate of the forces the frame experiences.

First, we converted the velocity to m/s:

\[
\frac{60 \text{ kilometers}}{\text{hour}} \times \frac{1 \text{ hour}}{60 \text{ minutes}} \times \frac{1 \text{ minute}}{60 \text{ seconds}} \times \frac{1000 \text{ meters}}{\text{kilometer}} = 16.667 \frac{\text{meters}}{\text{second}}
\]

**EQUATION 4**

Therefore, in the 0.1 seconds the vehicle takes to come to rest, it will have travelled:

\[
16.667 \frac{m}{s} \times 0.1 = 1.6667 \text{ m}
\]

**EQUATION 5**

Next, work is defined as follows:

\[
W_{net} = \frac{1}{2} m v_{final}^2 - \frac{1}{2} m v_{initial}^2 = \frac{1}{2} m (v_{final}^2 - v_{initial}^2)
\]

**EQUATION 6**

But work can also be defined as:

\[
W_{net} = f \times d
\]

**EQUATION 7**

Once these equations are set equal to each other, one can solve for the force \( f \):

\[
f = \frac{1}{2d} m(v_{final}^2 - v_{initial}^2)
\]

**EQUATION 8**

Now, it is simply a matter of plugging in the known values to arrive at a force:

\[
f = \frac{1}{2 \times 1.6667 \text{ m}} \times 270 \text{ kg} \times (16.667 \text{ m})^2 = 22500 \text{ N} = 5058 \text{ lb_f}
\]

**EQUATION 9**
4.1.2 Vehicle Landing on Vehicle Force Calculations

In this case, the simulation assumes that a vehicle from another team is landing on the back of the vehicle. There are a few assumptions that must be made about the course and the other vehicle in order to be able to simulate for this event. First, the main assumption is that the other vehicle lands on top of the vehicle from a height difference of 6m (20ft). Next, the weight of both vehicles was assumed to be identical. Additionally, this force causes the shocks to compress by 0.25m (10in). This amount of travel is justifiable given the 14in of ground clearance in the front and 12in in the back. Finally, all of the forces are exerted on the upper rear of the vehicle, not just the top of it.

Here, the potential energy ($E_{pot}$) of the vehicle that will crash into our vehicle was calculated using the following equation:

$$E_{pot} = m \times g \times h = 270 \, kg \times 9.81 \frac{m}{s^2} \times 6 \, m = 15892J$$

EQUATION 10

The potential energy is the work performed on the frame of the vehicle so the equation of work can be set equal to the equation of potential energy:

$$W = f \times d = 15892J$$

EQUATION 11

$$f = \frac{15892J}{0.25 \, m} = 63568N = 14291lb_f$$

EQUATION 12

4.1.3 Frontal Impact Simulations

In the following simulations, the force above was applied to the front of the frame. The fixtures were the rear suspension tabs. This is illustrated in Figure 19 with the ovals representing where the forces were applied and the arrows represent where the fixtures were. This simulation would represent the frame hitting a large obstacle in a perfectly inelastic collision and coming to a complete stop.
The stress and strain simulations below, Figure 20 and Figure 21 respectively, show that the frame performs as desired in the case of a frontal impact. Most frame components will experience stresses of between 4MPa and 20MPa, well below the yield strength of AISI 4130 (460 MPa). The front corners and the two corners behind those are the most likely to fail due to a collision, and experience stresses of up to 60MPa. The roll cage remains rigid and the body of the occupant is protected. Furthermore, during a collision, the stresses that the frame experiences are well below the yield strength of the material that was used.
Figure 22 illustrates how much the frame is expected to deform in the event of a frontal impact. The deformation is very minor, about 1.1 mm, meaning the collision would be an elastic collision and the frame would not be permanently bent as a result of the collision. Furthermore, by looking at the factor of safety for the frame given by Figure 23, one can see that the frame
fares well in such a collision. The minimum factor of safety is almost 2.9 in such an event. To logically confirm that the results obtained from this set of simulations are correct (provided the inputs were correct), one can compare the factor of safety results to the stress and strain results. Areas with the highest stresses and strains line up with the areas of lowest factors of safety. This is logical as the material and its thickness is consistent throughout the frame. Therefore, by applying a greater force to the same material and same amount of material, it should achieve a lower factor of safety.

FIGURE 22 - FRONTAL IMPACT DEFORMATION

FIGURE 23 - FRONTAL IMPACT FACTOR OF SAFETY
4.1.4 Side Impact Simulations

In the following simulations, the force was applied to the left side of the vehicle along the side impact member and the lower frame side member. The fixtures were the suspension tabs and shock mounts on the right side of the frame. This is illustrated in Figure 24 with the ovals representing where the forces were applied and the arrows representing where the fixtures were. This simulation would represent the frame getting hit by a large obstacle or vehicle from the side in a perfectly inelastic collision. This assumes that the frame would not slide sideways when it is hit.

FIGURE 24 - FORCES AND FIXTURES LOCATIONS FOR SIDE IMPACT
Figure 25 and Figure 26 demonstrate that the stress and strain on the frame is greater for a side impact than a frontal impact. Still, all the values are within the yield strength of the material, therefore the frame should not break as a result of a side impact.

**FIGURE 25 - SIDE IMPACT STRESS**

**FIGURE 26 - SIDE IMPACT STRAIN**
Figure 27 shows the displacement of frame members in the event of a side impact. The maximum displacement is estimated to be about 44 mm. This is more likely to be a permanent deformation. However, it should be noted that SolidWorks measures total displacement rather than deflection. The frame pivots up to 5° along the bottom right tube as the force is applied on the left tubes. As a result, the upper elements of the roll cage rotate more about the pivot axis. This shows up as the red areas with 44mm of deformation. The deflection of the tubes where the force is applied is closer to 35 mm.

![Figure 27 - Side Impact Deformation](image)

Figure 28 shows the factor of safety of the frame due to a side impact. The minimum factor of safety was 1.05. This is fairly low, but the frame should remain intact even in the worst possible situation. In reality, if the frame were to be hit with this much force, it is more likely for the wheels to lose grip and slide sideways. Therefore, these results are acceptable. The areas of lowest factor of safety match up with the areas where the forces were applied. Furthermore, these forces also match areas with the greatest stress, strain, and deflection of the tubes showing that the simulations were done correctly.
4.1.5 Rear Impact Simulations

In the following simulations, the force was applied to the rear tube at the tip of the triangle. The fixtures were the tabs and shock mounts where the front suspension would connect to the frame. This is illustrated in Figure 29 with the ovals representing where the forces were applied and the arrows represent where the fixtures were. In this situation, the vehicle would be hit from behind (possibly by another vehicle). Again, this assumes that the tires do not slip when the vehicle is hit from behind and the collision is perfectly inelastic.
The below stress and strain simulations in Figure 30 and Figure 31 show that the stress and strain on the frame when hit from behind is higher than when it is hit from the front, but lower than when it is hit from the side. This is because the front of the vehicle has more tubes that are able to split up the load whereas the back has fewer tubes. However, with the slanted tubes in the back, an obstacle is designed to bounce off the rear rather than collide directly into the rear of the vehicle with maximum force. This design consideration is not reflected in the simulations. Regardless, all stress and strain values remain within the range the material is capable of supporting.

FIGURE 30 - REAR IMPACT STRESS

FIGURE 31 - REAR IMPACT STRAIN
Figure 32 illustrates how much the frame is likely to deform as a result of a rear impact, showing that the deformation is minor; about 2mm. The frame therefore, should be capable of deforming this much without taking a significant loss to its structural rigidity. Figure 33 highlights the factor of safety of the frame. The minimum factor of safety for a rear impact is 1.4. This is sufficient for the conditions described above. The frame would still remain intact if hit from the rear with such a force. The roll cage remains safe for the driver as most areas of the roll cage have a significantly higher factor of safety.
4.1.6 Vehicle Landing on Vehicle Impact Simulation

This series of simulations represents another vehicle landing onto the vehicle. The first instance assumes that the other vehicle lands somewhere on the back of the frame. The second instance assumes that the other vehicle lands directly on top of the vehicle. The results for the first instance will be looked at first. In Figure 34, the ovals show the location where the forces are applied and the arrows represent the locations of fixtures. It is assumed that the vehicle does not move forward and remains fixed in place.

Figure 35 and Figure 36 on the next page show that the stresses and strains are close to the limits of the material. There are stress and strain concentrations on the upper portion of the rear triangle shaped frame member. This is an area where the welds could fail as a result of such high forces. However, this design directs the forces away from the roll cage elements. Additionally, many of the roll cage members experience significantly lower stress and strain.
The stress experienced by the 80% of the roll cage elements are below 240MPa and the strain for the equivalent roll cage members is 0.00240.
Figure 37 shows that the two bars in the back experience the greatest amount of deformation. They bend by nearly 40 mm and this is likely enough to cause permanent damage to these bars, if the welds holding them in place do not fail first. Otherwise, there is minimal deformation to areas occupied by the driver. To put this into perspective, by referencing the rules, the driver should have at least 2 in of clearance from edges of the roll cage elements. The simulation shows that the deformation is expected to be about 20 mm or slightly under 0.8 in.

![Figure 37 - Upper Back Impact Deformation](image)

Figure 38 illustrates the factor of safety across the frame in this type of collision. In this case it should be noted that the factor of safety is below 1, at around 0.6 in places colored red, which is in various areas near the rear of the vehicle where multiple bars meet. The areas with the lowest factor of safety are about the midpoints of the two bars in the back and where those tubes are welded to the frame. This means that those members are expected to fail in such an impact. However, it should be noted that the roll cage itself performs as expected and protects the driver. Most of the roll cage elements have a higher factor of safety. Also, this simulation
ignores the likelihood of deflecting the other vehicle because it is assuming a perfectly inelastic collision.
The second instance of a vehicle landing on the vehicle assumes that the other vehicle lands directly on top. In Figure 39, the ovals show the location where the forces are applied and the arrows represent the locations of fixtures. Again, it is assumed that the vehicle does not move forward and remains fixed in place.

![Figure 39 - Forces and Fixtures Locations for Vehicle Landing on Vehicle Impact Instance Two](image)

**FIGURE 39 - FORCES AND FIXTURES LOCATIONS FOR VEHICLE LANDING ON VEHICLE IMPACT INSTANCE TWO**
The simulations below show how the frame distributes the stress, Figure 40, and strain, Figure 41, across the tubes when another vehicle lands on top of it. As one can see the stress and strain are lower in this instance than the previous vehicle landing on vehicle instance.
Figure 42 shows how much the frame is expected to deform when another vehicle lands on top of it. The expected deformation is 14.5 mm for the overhead members. This is minimal and should be sufficient in protecting the occupant. 2 in of clearance between the driver’s head and the overhead members are specified by the rules, which is far greater than the expected deformation of these bars. The other roll cage members exhibit very small deformation.

FIGURE 42 - TOP IMPACT DEFORMATION
Figure 43 shows that the roll cage remains safe when another vehicle lands on it. Although the factor of safety is fairly low (1.01), it still passes for the worst case scenario. It should be noted that the corners have a low factor of safety and this could be improved upon by welding gusset plates to these corners to relieve the forces or provide additional contact area for the welds so that they are less likely to fail.

Overall, the frame should handle collisions very well. In most of these worst case scenarios, the frame remains very rigid due to the material that was chosen and the design. Even in the extreme cases used for this simulation, the driver will remain safe. The most likely collisions to occur would be the full frontal collision as the driver might not be able to turn or see an obstacle in time. In both the front and rear collision simulations, the frame performed very well. Although the frame does not perform as well in a side impact collision, the conditions used are highly unlikely to occur for a side impact and, regardless, the frame still passes with those conditions. Furthermore, the simulations indicate that the driver will remain safe even if another vehicle lands on it from a difference in height of 20 feet between the top of the frame and the bottom of the other vehicle. Additionally, having a stiff frame will help improve the performance as the vehicle is being driven throughout the racecourse. To further reinforce the
weak areas and improve the rigidity of the frame, gusset plates could be added along the corners where the tubes are welded together.

4.2 CAD Simulations – Suspension

Below are the results of the suspension simulations to validate the design. Calculations were completed to determine forces that the suspension would be expected to experience in worst case scenarios during the Baja competition.

4.2.1 Suspension Simulation Force Calculations

The suspension on a Baja vehicle must cope with strong forces. The suspension will distribute the load onto the frame, but it is critical that the suspension does not exhibit excessive deformation that could damage the system. The material selected for the suspension was the same steel that was used for the frame: AISI 4130. Again, the worst case scenario is being used to simulate how well the suspension will hold up under loads. For this case, Newton’s second law is being used:

\[ f = m \times a \]

EQUATION 13

The mass of the vehicle is expected to be 270 kg (or 595 lbs) and the acceleration due to gravity is 9.81 m/s\(^2\) (or 32.2 ft/s\(^2\)). However, this also needs to be multiplied by the G forces the suspension is expected to experience. According to *The Motor Insurance Repair Research Centre* this should be around 5 Gs. [14] For perspective, a typical person becomes unconscious at 5 Gs in the vertical direction. When these values are evaluated in the equation above, we receive the following force:

\[ f = 270 \text{ kg} \times 5 \times 9.81 \frac{m}{s^2} = 13244 N = 2977 \text{ lb_f} \]

EQUATION 14

Furthermore, although the shocks would allow the suspension arms to pivot along the tabs that are used to connect to the frame, the shocks in the simulations were assumed to be perfectly stiff. This concentrates the loads on the tips of the suspension arms. Effectively, this would mean that the wrong shocks were chosen for the suspension or the shocks have
completely locked up. Moreover, these simulations assume that only one suspension is subjected to these forces. Ultimately, if the simulations still show that the arms can cope with these forces under these conditions, they will be capable of withstanding any force in a Baja event.

### 4.2.2 Front A-Arm Simulations

After running the stress (Figure 44) and strain (Figure 45) simulations below, it was found that most of the stress is concentrated around the areas where the shocks are mounted to the front A-arms, which is logical. The tabs that connect to the shocks could be made thicker to reduce the stresses and strains on them.
Figure 46 shows the deformation that is expected on the A-arm as a result of the simulated forces. The maximum amount of deformation (2.42 mm) is fairly low. This means that the design should not experience plastic deformation as a result of such a load. Having a low deformation can only improve the performance of the vehicle.

The results from Figure 47 show how the factor of safety varies across this design. The results are expected since the greatest amounts of stresses and strain are along the shock tabs. These match up with the areas of lowest factors of safety. Still, the factor of safety is above 1 inch these areas, indicating that the design is well suited for these forces. To improve the factor of safety, the plates could be made thicker.
4.2.3 Rear Trailing Arm Simulations

Figure 48 and Figure 49 show the simulated stress and strain due to the forces described acting on the rear trailing arm. Predictably, the corners or edges along the shock tabs act as stress concentrators. By making these tabs wider, the stresses at the corner/edge would be reduced. It should be noted that the stresses and strains on the trailing arm are lower than the A-arms. This is advantageous as the vehicle is heavier in the rear and more likely to land on the trailing arms than the front A-arms.
Figure 50 displays the expected deformation of the trailing arm due to the impact forces. With a maximum predicted deformation of 0.91 mm, it deforms less than the suspension design in the front. Again, this should help performance as the vehicle is expected to land on the rear suspension as the vehicle comes off a jump.

FIGURE 50 - TRAILING ARM IMPACT DEFORMATION

Figure 51 shows local areas of weakness. The factor of safety is above 1 in all areas indicating that it is well suited for an event as described in the suspension simulation forces section. Furthermore, the factor of safety is higher for the rear suspension than it is for the front suspension, meaning that the back can handle even greater forces than the front, which is as designed.

FIGURE 51 - TRAILING ARM IMPACT FACTOR OF SAFETY
4.2.4 Simulation Outcomes

Overall, both suspension designs are well suited for the vehicle and perform as desired even in the worst case scenarios which the suspension is unlikely to experience in competition. To improve upon the rigidity of these designs, the plates and/or tubes could be made thicker, but this would add weight. There is always a compromise between weight and structural rigidity and these designs achieve a good balance.

4.3 Thermal Fluid System Analysis of Different Exhaust Pipes

The Baja vehicle is powered by a 305 cc Briggs and Stratton gasoline internal combustion engine which can be modeled under ideal conditions as a Carnot heat engine. In a Carnot heat engine, maximum efficiency is defined by the equation:

\[ \eta = 1 - \frac{T_c}{T_H} \]

EQUATION 15

Where “\( \eta \)” is the maximum efficiency, “\( T_c \)” is the absolute temperature of the cold sink, and “\( T_H \)” is the absolute temperature of the hot reservoir. Assuming the heat of the combustion gases (the hot reservoir) stays constant between each cycle, theoretically the efficiency of the engine can be increased by reducing the temperature of the exhaust pipe (the cold sink). A more efficient engine will slightly increase the amount of available torque and/or reduce the gasoline consumption of the engine. The benefits of extra available torque and reduced gas consumption would provide better vehicle performance in both the hill climb and endurance race respectively, improving the teams overall score.

The exhaust pipe material will be a 22 gauge stainless steel with a 1.25 in inner diameter which is lightweight and compliant with SAE Baja rule B2.5.11A. [1] The exhaust system requires a minimum of three 90° bends to articulate the driveline components and frame while finally extending back to the edge of the frame to ensure no exhaust gases get drafted back into the engine compartment. The bending process chosen to manufacture the system will dictate the internal and external surface contours of the pipe bends and will. The three types of bends that will be analyzed include, CNC mandrel bent with a smooth finish and continuous diameter,
press bent with a smooth finish but slightly constricted tube diameter through the bends, and crush bent tubing with a rough surface finish and a varying diameter through the bends.

The bend characteristics are what differentiates the three exhaust models and can be seen in the figures below. In Figure 52, the type of bend shown is the result of a crush bent process which works by moving a die, powered or manually, over the outside of the metal tube to create the desired bend angle and radius. This method crushes the inner surface of the pipe causing it to buckle to compensate for the decreased inner radius as there is no support to keep the shape uniform. In Figure 53, the type of bend being shown is the result of a press bending process, where a die is pressed into a pipe to create the desired bend angle and diameter. This process keeps the metal tube surface smooth but creates an indent on the inner surface of the bend due to the pressure used to create the bend. In Figure 54, the type of bend being shown is the result of a rotary CNC mandrel which draws the bend using a die and an internal support called a mandrel. This process creates the most uniform bend and should allow for the most efficient airflow.

FIGURE 52 - A 90 DEGREE BEND TYPICAL OF BEING MADE USING A CRUSH BENDER
A flow analysis was conducted to find the bending process that produces an exhaust system that minimizes the amount of heat lost to the exhaust pipe via convection. The thermal conductivity of 316 stainless steel is 16.3 W/(m*K) at 100 °C and 21.5 W/(m*K) at 500 °C. The expected exhaust gas inlet temperature and pressure is 454.4 °C (727.6 K) and 1.72 MPa respectively with a volumetric flow rate of 0.183 m³/s. [15] The exhaust exit pressure is assumed to be 0.101 MPa with the same mass flow rate that it is entering with. The gas composition is 71% nitrogen, 14% carbon dioxide, 12% water vapor, and 3% residual oxygen. The exhaust system with the highest exit exhaust temperature and least turbulence will be considered the best system as the least amount of heat is conducted from the gas to the metal tube.
After an initial SolidWorks Flow Analysis has been conducted on all three models to determine the most efficient, hand calculations will be used to further investigate the heat losses due to convection in the exhaust pipe. Figure 55, Figure 56, and Figure 57 detail the results of this analysis below.

**FIGURE 55 - SIMULATED TEMPERATURE DROP IN THE CRUSH BENT SYSTEM**

**FIGURE 56 - SIMULATED TEMPERATURE DROP IN THE PRESS BENT SYSTEM**
After setting up the initial parameters and running all three models it was clear that the CNC mandrel bent exhaust system was the most efficient at removing exhaust gases. The exit temperatures for the crush bent, press bent, and mandrel bent systems were 157.1 degrees Celsius, 236.87 degrees Celsius, and 248.17 degrees Celsius respectively. The results returned by the analysis were mostly expected since crush bending provides the most flow obstructions increasing turbulence and therefore the heat transfer rate, and the mandrel bent tubing provided the least restriction and a smooth contour to aid in the flow of the exhaust gases. The surprising result was how efficient the press bent system proved to be even with the flow restrictions at each of the 3 bends, meaning the exhaust flow rate generated by the engine was not high enough to utilize the full capacity of the mandrel bent exhaust system. Depending on the cost of production and the observed power benefit, it may become more cost effective to use a press bent exhaust system and use the money saved to purchase better components for other aspects of the vehicle.

The hand calculation that followed required a little extra background research. The density of the exhaust gases at 727.6 K was calculated to be around 8.3121 kg/m³. [15] While the viscosity, conductive heat transfer, and heat capacity of the exhaust gases were calculated
to be $1.664 \times 10^5$ Pa*s, 0.0217 W/(m*K), and 0.850 kJ/(kg*K) respectively. [15] The Reynolds number was calculated to be 478,220 which is indicative of a fully developed turbulent flow. The Nusselt number was calculated to be 10,504.9 for the straight lengths and 37,957.7 for the CNC bends which provided a total heat loss rate of 649 kW. This value is higher than the actual value because it was assumed the exhaust temperature stays constant throughout the exhaust system when in reality, as heat is lost the temperature decreases along with the density and viscosity. An iterative solution could generate a better result however one fact would stay the same. The heat transfer rate in the 90° bends is 361% higher than the heat transfer rate in the straight sections under the ideal CNC bent conditions, showing that the best way to further increase exhaust frequency would be to ensure only the minimum number of bends were used in the exhaust system. The full hand calculations can be seen in Appendix A – Hand Calculations for Fluid Flow Analysis of this report.

For the final vehicle configuration a new exhaust system would be required to ensure exhaust gases do not build up in the vortex behind the firewall. After running several simulations on three popular exhaust system configurations using different pipe bending methods followed by calculations to validate the simulations, the best exhaust system was found. Utilizing a CNC rotary mandrel bent exhaust system would ensure the most efficient exhaust flow and maximize engine power output to possibly give the WPI vehicle an edge during the dynamic competition events.
Chapter 5: Conclusions and Recommendations

There were a few decisions this year’s Baja team had to make due to constraints. These decisions could be revisited as there are some alternative designs that were created during this project, but ultimately not implemented in the final vehicle. These designs will be made available to next year’s team for their consideration.

The first compromise was the single A-arm in the front suspension. Initial plans were to use a double A-arm. Although a single A-arm was used, a double wishbone system with a custom knuckle could be an improvement. This is a more complex system, but it could be fine-tuned to optimize front suspension performance. As it distributes the loads more evenly along both arms and on the frame, it relieves some of the stress concentration points that are present on the single A-arm design. Furthermore, lighter materials like aluminum could be used for the suspension.

The steering is another area that could use improvement. Camber, caster, and Ackerman steering were considered for the front suspension and steering. However, Ackerman steering was something that was out of the scope of this project but would make a great addition to the steering and suspension system.

Finally, a gearbox with a reverse gear (and high and low gears) was secured during the project. Nevertheless, incorporating it into the final vehicle proved to be an unanticipated challenge. Being able to select a gear rather than let the CVT pick the gear was difficult. The reverse gear could be added to help with the maneuverability challenge during competition. Furthermore, besides fine tuning the CVT, it might be advantageous to have the driver able to have greater control on the gear they are using. For example, the characteristics of the engine might indicate that there is greater torque at a higher RPM value and remaining in a low gear would help with a hill climb challenge.

The last set of advice is geared towards team dynamics and collaboration. The team should focus on smaller components of the entire vehicle in order to deliver on quality. Furthermore, the composition of the people on the team makes a significant difference. At
least half the team members need to have great hands-on experience. The remaining half needs to have strong skills in computer aided design, report writing, testing, and extensive knowledge about cars. Collaborations with other organizations are essential. The SAE club helped test the previous vehicle. Assabet Valley Regional Technical High School welded the frame. These ties should not be cut off. However, the tasks that are handed out to third parties should be carefully considered. The capabilities and time it takes some third parties to accomplish tasks should dictate the tasks they are given.
Appendix A – Hand Calculations for Fluid Flow Analysis

To start the analysis of the exhaust system it must first be understood that due to the operation of the combustion engine and valve system, all the exhaust gases are flowing in a fully developed turbulent state. For this analysis it is also assumed that the exhaust gases are exiting the combustion chamber at a temperature of 727.6 K, volumetric flow rate of $0.183 \frac{m^3}{s}$, and a pressure of 1.72 MPa. Finally the exhaust gas composition is determined to be 71% $N_2$, 14% $CO_2$, 12% $H_2O$, and 3% $O_2$ as documented here: http://www.volkspage.net/technik/ssp/ssp/SSP_230.pdf. It should also be stated that the exhaust pipe has a diameter of 0.03175 m and a bend radius of 0.0762 m.

The density and viscosity of the exhaust gas are the first values that will be determined utilizing the following online resources: (http://yeroc.us/calculators/gas-density.php, http://www.engineeringtoolbox.com/gas-density-d_158.html, http://www.lmnoeng.com/Flow/GasViscosity.php)

Determining exhaust gas density at 727.6 K:

$$\rho = \sum \frac{0.71 \times 7.982}{m^3} \frac{kg}{m^3} + \frac{0.14 \times 12.539}{m^3} \frac{kg}{m^3} + \frac{0.12 \times 5.133}{m^3} \frac{kg}{m^3} + \frac{0.03 \times 9.117}{m^3} \frac{kg}{m^3} = 8.312 \frac{kg}{m^3}$$

Determining exhaust gas viscosity at 727.6 K:

$$\mu = \sum \frac{0.71 \times 3.293 \times 10^{-5}}{Pa \cdot s} \frac{Pa \cdot s}{m^3} + \frac{0.14 \times 3.189 \times 10^{-5}}{Pa \cdot s} \frac{Pa \cdot s}{m^3} + \frac{0.12 \times 2.400 \times 10^{-5}}{Pa \cdot s} \frac{Pa \cdot s}{m^3} + \frac{0.03 \times 3.890 \times 10^{-5}}{Pa \cdot s} \frac{Pa \cdot s}{m^3} = 3.189 \frac{Pa \cdot s}{m^3}$$

Next the Reynolds number can be determined.

$$Re = \frac{Q \times D}{\gamma \times A}$$

Where: $Q$= the volumetric flow rate
D= the diameter of the pipe

\[ \gamma = \frac{\mu}{\rho} \]

A= cross sectional area

Hence the Reynolds number is discovered to be:

\[
Re = \frac{0.183 \frac{m^3}{s} \cdot 0.0318 \text{ m}}{\left( \frac{3.189 \times 10^{-5} \text{ Pa} \cdot \text{s}}{8.312 \frac{\text{kg}}{\text{m}^3} \cdot \frac{\pi \cdot 0.0159^2 \text{ m}^2}{} \right)} = 1.913 \times 10^6
\]

The equation for the Nusselt number in a straight piper exhaust system was derived in the following study and listed below: (http://www.mie.uth.gr/labs/ltte/grk/pubs/exhsysht.pdf)

\[
Nu = \frac{\frac{f}{8} \cdot (Re - 1000) \cdot Pr}{1.07 + 12.7 \cdot \sqrt[8]{\frac{f}{8} \cdot \left( Pr^2 - 1 \right)}}
\]

Where: \( f \) = the friction factor of the pipe

\( Re \) = the Reynolds number

\( Pr \) = the Prandtl number

To find the Nusselt number for the bends, the study provides the following relation:

\[
\frac{Nu_{\text{Bend}}}{Nu_{\text{Straight}}} = 1 + \frac{21 \cdot D_{\text{pipe}}}{Re^{0.14} \cdot D_{\text{bend}}}
\]

To further evaluate this problem, some information was needed on the friction factor of a smooth pipe. This information was obtained from the Flow of Fluids handbook.

From Page A-29:

if \( \frac{\text{Bend Radius}}{\text{Pipe Diameter}} \) is between 2 and 3, then the friction factor of the bend is: 12*\( f \)

From Page A-25:
For a 0.03175 m (1.25 in) pipe with a flow rate of 0.183 \( \frac{m^3}{s} \) (6.46 \( \frac{ft^3}{s} \))

The friction factor of the pipe is determined to be 0.026 based off the chart

This means the friction factor of the bend is:

\[ f_{Bend} = 12 \times 0.026 = 0.312 \]

Now the Prandtl number must be calculated:

\[ Pr = \frac{\frac{\mu}{\rho}}{\frac{k}{\rho \cdot C_v}} \]

Where: \( k \) = the thermal conductivity of the exhaust gas

\( C_v \) = the specific heat of the exhaust gas

The thermal conductivity and specific heat must be calculated next

To calculate the thermal conductivity, the values for each gas were found utilizing the following resource: (http://www.engineeringtoolbox.com/)

\[
k = \sum 0.71 \times 0.024 \frac{w}{m \cdot K} + 0.14 \times 0.0146 \frac{w}{m \cdot K} + 0.12 \times 0.016 \frac{w}{m \cdot K} + 0.03 \times 0.024 \frac{w}{m \cdot K} = 0.02172 \frac{w}{m \cdot K}
\]

To calculate the specific heat, the values for each gas at constant pressure were found utilizing the following resource: (http://www.engineeringtoolbox.com/)

\[
C_v = \sum 0.71 \times 0.743 \frac{kJ}{kg \cdot K} + 0.14 \times 0.655 \frac{kJ}{kg \cdot K} + 0.12 \times 1.760 \frac{kJ}{kg \cdot K} + 0.03 \times 0.659 \frac{kJ}{kg \cdot K} = 0.850 \frac{kJ}{kg \cdot K}
\]
From these newly found values, the Prandtl number can now be calculated:

\[
Pr = \frac{\left(\frac{\mu}{\rho}\right)}{\left(\frac{k}{\rho * C_p}\right)} = \frac{\left(\frac{3.189 \text{ Pa} * s}{8.312 \frac{\text{kg}}{\text{m}^3}}\right)}{\left(\frac{0.02172 \frac{w}{\text{m} * \text{K}}}{8.312 \frac{\text{kg}}{\text{m}^3} * 0.850 \frac{\text{kJ}}{\text{kg} * \text{K}}}\right)} = 124.8
\]

Now the Nusselt number of the straight and bent portions of the exhaust system can be calculated:

\[
\begin{align*}
\text{Nu}_{\text{Straight}} &= \frac{0.026}{8} * (1.913 * 10^6 - 1000) * 124.8 \quad 1.07 + 12.7 * \sqrt{\frac{0.026}{8}} * \left(124.8^2 - 1\right) = 42100 \\
\text{Nu}_{\text{Bend}} &= \frac{0.312}{8} * (1.913 * 10^6 - 1000) * 124.8 \quad 1.07 + 12.7 * \sqrt{\frac{0.312}{8}} * \left(124.8^2 - 1\right) = 152000
\end{align*}
\]

Now to find the heat loss from the exhaust gas to the exhaust system walls using the relation of the Nusselt number to the heat transfer:

\[
Nu = \frac{h * L}{k}
\]

Where: \( h \) = the convective heat transfer coefficient of the fluid

\( L \) = the characteristic length, which is the diameter for a round tube

The convective heat transfer coefficient can now be solved for:

\[
\begin{bmatrix}
\text{Nu}_{\text{Straight}} \\
\text{Nu}_{\text{Bend}}
\end{bmatrix} = \begin{bmatrix}
h * 0.03175 \\
0.02172 \frac{w}{m * K}
\end{bmatrix} = \begin{bmatrix}
42100 \\
152000
\end{bmatrix}
\]

\[
\begin{bmatrix}
h_{\text{Straight}} \\
h_{\text{Bend}}
\end{bmatrix} = \begin{bmatrix}
28800 \frac{W}{m^2 * K} \\
104000 \frac{W}{m^2 * K}
\end{bmatrix}
\]

82
Please note that the heat transfer is 361% higher in the bends which is expected due to the extra drag from changing the exhaust gas flow direction. This result is also reflected in the SolidWorks Flow Simulations.

To find the heat loss, the areas of the straight and bent sections needed to be measured from the model and are listed below:

Straight tube area total: $111.74 \times 10^{-4} \text{ m}^2$

Bent tube area total: $58.29 \times 10^{-4} \text{ m}^2$

$$\dot{q}_{\text{loss}} = 28800 \frac{W}{\text{m}^2 \cdot K} \times (111.74 \times 10^{-4} \text{ m}) \times (727.6 \text{ K} - 293 \text{ K}) + 104000 \frac{W}{\text{m}^2 \cdot K} \times (58.29 \times 10^{-4} \text{ m}) \times (727.6 \text{ K} - 293 \text{ K})$$

$$\dot{q} = 403 \text{ kW} = 403 \frac{kJ}{s}$$

The heat rate of the exhaust gas can be calculated utilizing the volumetric flow rate, density, specific heat, and the absolute temperature.

$$\dot{q}_{\text{heate rate}} = 0.183 \frac{m^3}{s} \times 8.312 \frac{\text{kg}}{m^3} \times 0.850 \frac{\text{kJ}}{\text{kg} \cdot \text{K}} \times 727.6 \text{ K} = 941 \frac{kJ}{s}$$

The exhaust gas temperature can be calculated utilizing the heat rate, heat loss of the exhaust gas along with the volumetric flow rate, density, and specific heat to find the absolute temperature.

$$\text{Exhaust Temp} = 941 \frac{kJ}{s} - 403 \frac{kJ}{s} \times \frac{1}{0.183 \frac{m^3}{s}} \times \frac{1}{8.312 \frac{\text{kg}}{m^3}} \times \frac{1}{0.850 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}} = 416.1 \text{ K}$$

This exhaust temperature seems very low because of one of the models initial assumptions.

This model assumes the exhaust gas stays at 727.6 K through the whole length of the exhaust system which inflates the total heat transfer rate. This assumption was made because the properties of a gas change in relation to its temperature and pressure and make it impossible to generate a perfect result utilizing only hand calculations. Iterative analysis and flow simulations generate much better exit temperature values due to the vast amount of temperature iterations that a computer can compute in a much more efficient manner when compared to standard hand calculations.
## Appendix B– Budget

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Misc. Fasteners $200.00

- Nut-Lock (6) $27.96
- M10 X 1.5 X 20 8.8 TXCS FL (8) $10.00
- Washer-Flat (4) $2.88
- Washer-Cone (4) $17.68
- Nut-M18 X 1.5, Castle, Zinc (2) $2.88
- Nut-Wheel, Castle (2) $8.62
- Bolt-Flange, Hex, W/Spring (4) $7.88
- Cotter Pin (6) $1.98
- Washer (2) $1.16
- Washer-Post, Steering (4) $8.32
- Washer, 3/8 FW $5.36
- Washer -Flat (2) $4.28
- Collar-Lockering (4) $9.56
- Pin, Cotter (2) $2.88
- Ring, Retaining (2) $17.98
- Pin-Cotter 10 (2) $0.70
- M10 X 1.25 X55 10.9 ZOD (4) $9.04
- Bolt (2) $11.98

Brakes $0.00

- Bolt-Brake Disc (12) $11.16
- Disc-Brake, Front (2) $131.98
- Disc-Brake, Rear, 8.625 X 0.188 (2) $103.98
- ASM-Caliper, Brake, 1.5, RH,R,MET $231.99
- ASM-Caliper, Brake, 1.5, LH,R,MET $231.99
- ASM-Caliper, LH INCL 2,3 $225.99
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