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Design and Optimization of an SAE Baja Vehicle

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Design and Optimization of a Baja SAE 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

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Date: April 27th, 2015
Abstract

The purpose of the 2014-15 Society of Automotive Engineers (SAE) Baja MQP was to design, develop, and manufacture a vehicle within the specifications of the SAE Baja Competition in order to enter a future competition. The chassis inherited from the 2013-14 SAE Baja MQP was utilized as a framework that the new designs had to integrate within. The success of the vehicle at competition will be driven by the individual subcomponents that make it unique including a dual differential four-wheel drive system, double wishbone suspension configuration in the front and rear, and five-speed manual transmission with a custom case that were all designed and developed by the team using software simulations, finite analysis, and optimization to test subcomponent interactions and packaging considerations. To manufacture the design, the team used multiple local fabrication shops as well as the resources available on WPI’s campus.
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**Introduction**

SAE International, a developer of technical standards and best practices, sponsors the Baja SAE series that allow collegiate chapters to primarily concentrate on the design and manufacturing of the chassis, drivetrain and suspension to improve the vehicle as the engine is standard amongst all teams. ("History of Mini Baja East,"

The purpose of the MQP was to design and manufacture a Baja SAE vehicle utilizing four wheel drive by the end of the 2014-2015 academic year for the SAE WPI collegiate chapter. The success of the vehicle at competition will be driven by the individual subcomponents that make it unique including a dual differential four-wheel drive system, double wishbone suspension configuration in the front and rear, and five-speed manual transmission with a custom case that were all designed and developed by the team using software simulations, finite analysis, and optimization to test subcomponent interactions and packaging considerations.

Collaboration and partnership with our project advisor, industry experts, and WPI collegiate SAE Chapter were essential in completing the Baja SAE vehicle. The necessary collaboration and partnerships developed expanded the reach of the Baja SAE series outside the engineering principles implemented from the classroom to the vehicle as it simulates a real world design project. The effective utilization of all the resources that Worcester Polytechnic Institute had to offer proved to be indispensable throughout the design, build and testing phases.

This vehicle will be utilized in Baja SAE competitions starting in the 2015-2016 academic year and provide WPI SAE members with an opportunity to develop and practice important engineering principles outside of the classroom with a hands on application. The MQP has a unique ability to encourage collaboration with the SAE WPI chapter, as this vehicle will
ultimately be theirs for future use by engineering students. The relationship offered the ability to
mentor new members and increase excitement of the Baja SAE series around campus and within
the chapter itself.
2013-2014 Baja SAE MQP

The 2014 Baja SAE competition underwent rule changes, regarding the specifications of the frame (Engineers, 2014). Page ten of the rules specifies that the vehicle must accommodate a male up to the 95th percentile, 186.7cm (6ft 1.5in) tall, 102kg (225 lbs), and an erect sitting height of 97cm (38.25in). Rule B8.3.12 of the Baja SAE 2014 rules requires the frame to be constructed out of steel that consists of .18% carbon content. Rule B8.3.1 of the Baja SAE 2014 rules splits the primary roll cage into primary and secondary members. This means that certain bars, front bracing members and rear roll hoop must be continuous making bending a requirement rather than welding, shown in Figure 1 (Atamer, Enjamio, Oliviera, Van Dale, & Wong, 2014). The new frame restrictions required SAE WPI to redesign and manufacture a new competition-eligible vehicle chassis. This was the main goal of the 2013-2014 Baja SAE MQP.

The five-member team had three overall major design goals for their MQP.

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 (Atamer et al., 2014).
Results

The primary outcome of the 2013-2014 Baja SAE MQP was a frame that was designed to be eligible for future Baja SAE competitions and inherited by the 2014-2015 Baja SAE MQP. This frame underwent a design review utilizing finite element analysis in accordance to SAE guidelines before being manufactured to ensure safety and design credibility (Atamer et al., 2014). The frame was fabricated by Assabet Valley Regional Technical High School as an educational opportunity for their students. The outsourcing of the frame allowed the 2013-2014 MQP the chance to gain experience interacting with outside contractors as well. The inherited frame provided challenges with mounting locations, as there were inconsistent angles as a result of the manufacturing.
Background Research

Control Arm Research and Analysis

Introduction

The role of a suspension within a vehicle is ensuring that contact between the tires and driving surface is continuously maintained, as well as providing a smooth and comfortable ride for the driver (Atamer et al., 2014). It must also absorb the vertical accelerations created by the wheels when the vehicle is in motion (Riley, 2014). In order to accomplish these goals a wide variety of components are utilized to achieve the desired vertical travel as well as compression for absorption. These components include things like shocks for the vertical travel, springs for compression, control arms for travel and flexibility, and lastly ball joints for mobility. A brief overview of the various types of control arms as well as how they function will be presented.

Control Arms & Parameters

Control arms serve as the connecting link between a vehicle’s chassis and suspension system (Nutt, 2014). Their role is to hold all the components within a vehicle together while undergoing vertical movement (Nutt, 2014). Vehicles typically have two categories for suspension; one being dependent suspension systems and the other being independent suspension systems (Vivekanandan et al., 2014).

Dependent suspension systems are essentially a beam connecting two wheels that transmit any movement or loads to both of the wheels (Vivekanandan et al., 2014). Although dependent suspensions add quite a bit of rigid strength to a vehicle as well as simplicity and lower cost there are many more disadvantages to their use. Today this type of suspension is usually only used in heavy industrial applications where that type of strength is typically needed (Longhurst, 2014). Some of the main disadvantages to using a dependent system are there...
excessive unsprung weight, which in turn need a heavy spring to be able to hold it (Longhurst, 2014). Also another major issue is there lack of adjustability since everything is rigidly attached meaning that once everything is set nothing can be adjusted or moved (Longhurst, 2014). This type of suspension will not be considered in the scope of this project largely due to its lack of adjustability.

Independent suspension systems provide more effective functionality in traction and stability for off-roading applications. Independent suspension systems provide flex (the ability for one wheel to move vertically while still allowing the other wheels to stay in contact with the surface) (Vivekanandan et.al, 2014). This is the type of suspension that is predominantly found within the Mini SAE Baja community, as it provides great functionality as well as adjustability, which are all key features for a Baja vehicle to possess (Chandler et.al, 2011).

There are many different versions and variations of independent suspensions, which include swing axle suspensions, transverse leaf spring suspensions, trailing and semi-trailing suspensions, Macpherson strut suspensions, and double wishbone suspensions (Vivekanandan et.al, 2014). Before the differences between each are discussed, a critical parameter needed for proper design of a control arm must be defined. Control arms are used for far more than just component support they can provide adjustments in not only the positions of shock assemblies, but also provide adjustments and flexibility in wheel alignment as well (Riley, 2014). The adjustment in wheel alignment that can be achieved through proper control arm design is known as camber (Riley, 2014).

**Defining Camber**

Camber is defined as the vertical alignment of wheels and is measured as an angle from the vertical axis of a suspension (Riley, 2014). A vehicle can have positive or negative camber
depending on the setup of the vehicle (Riley, 2014). Positive camber is taken to be when the top of a wheel is pointed outward away from a vehicle and the bottom of the wheel is pointed inward towards the vehicle (Nutt, 2014). Whereas negative camber is taken to be when the top of a wheel is pointed inward towards the vehicle and the bottom of the wheel is outward away from the vehicle (Nutt, 2014). A visual representation of the parameter is displayed in Figure 2.

A camber angle produces a camber thrust, which pulls the bottom of a wheel into the direction the top portion of the wheel is facing (Riley, 2014). In a Baja vehicle, positive camber is typically not used because it increases the chances of a rollover-taking place and is unstable in high speed and off-roading applications (Riley, 2014). Negative camber, however, can prove to be useful in off-roading applications. A wheel with negative camber can assist in the turning of a vehicle as it is able to maintain as much contact with the surface as possible, thus giving it more grip as well as stability (Riley, 2014). An example of this is when a car takes a left turn; the car will want to roll right. However if a negative camber is used on the right side wheels they will maintain maximum contact with the surface and provide stability throughout the turn. This
negative camber on the outward side of the car keeps the tire on the surface. Therefore maximizing the cornering forces experienced in a turn, as well as reducing the amount of rubbing there is between the surface and the tires of the vehicle. An example of the phenomena is shown in Figure 3 (Riley, 2014).

Control Arm Analysis

As mentioned previously there are quite a variety of different independent control arm designs that can be utilized however not all are completely suitable for a Baja car. The swing axle suspension was not considered because it cannot support large deflections and does not have very good handling capabilities (Wan, 2000). The transverse leaf spring suspension, although used in high performance vehicles like the Corvette, is not viable due to our selection of coil over springs and also the cost would be out of our budget. This suspension utilizes leaf springs made of composite materials, which are typically very expensive (Wood, 2014). The suspensions types that are suitable include the MacPherson strut, double wishbone, semi trailing and trailing arm suspensions.
Typically the semi trailing and trailing arm suspensions are used in the rear of a vehicle while the MacPherson strut and the double wishbone designs can be used in both the front and rear (Wan, 2000). The MacPherson strut is also called a single control arm suspension and consists of a strut or shock assembly, wheel hub and one control arm (Riley, 2014). Both the strut and control arm connect directly to the chassis of the vehicle. The control arm then connects to the bottom of the wheel hub, whereas the strut connects to the top of it (Riley, 2014). A visual representation is illustrated in Figure 4.

![Figure 4: MacPherson Strut](image)

The advantages of a MacPherson strut are its simplicity, low number of mechanical components that reduce failure points, low cost, light weight, and large amount of space allowing for ease of component integration (Riley, 2014). Its disadvantages, however, include an increased ride height due to its vertical space and lack of camber adjustments which means turning is more difficult (Wan, 2000).

Double wishbone suspensions utilize two control arms along with a shock assembly (Riley, 2014). The double wishbone consists of an upper arm and a lower arm; both of which are
connected to the wheel and chassis of the vehicle (Riley, 2014). A double wishbone setup is shown in Figure 5.

![Double Wishbone Setup](image)

**Figure 5: Double Wishbone Setup**

Before the advantages are discussed it’s important to note that there are two different types of double wishbone suspensions; equal length double wishbone and unequal length double wishbone (Wan, 2000). In the equal length double wishbone suspension, both the top and bottom arms are the same length making camber angle effects difficult to achieve (Wan, 2000). In an unequal length double wishbone suspension the top control arm is shorter than the bottom arm, thus giving it the ability to induce negative camber angles when a vehicle turns (Wan, 2000). The advantages of an unequal length double wishbone are its adjustability, camber abilities, versatility, and load handling abilities (Riley, 2014). The double wishbone provides increased negative cambers when a vehicle is taking a turn, thus providing greater control and stability (Riley, 2014). It also can easily be adjusted to increases parameters such as camber. Components can be moved around and the shock assembly can be mounted in various positions to achieve different goals. The use of two arms also makes the suspension resistive to high loads and, as
mentioned before, these unequal length arms provide a great means of handling the vehicle (Riley, 2014). The disadvantages of a double wishbone however are its weight, high cost, complex components that increase the chance of failure, and ultimately they can take up a substantial amount of space (Riley, 2014).

As mentioned earlier, semi-trailing and trailing arm suspensions are typically used in the back of a vehicle. Trailing and semi trailing arm suspensions are almost identical as they are both pivoting arms attached to the chassis of the vehicle. These arms are placed in front of the wheel creating the trailing effect (Atamer et.al, 2014). The difference between the two arises in the position of the pivot points. Trailing arms have their pivots positioned perpendicular to the vehicles centerline while semi-trailing arms have their pivots positioned at an angle to the centerline (Wan, 2000). Both setups are displayed in Figure 6.

![Figure 6: Trailing Arm vs Semi-Trailing Arm](image)

In general both arms share the same advantages, which include a rigid design, simplicity, and very few mechanical components (Wan, 2000). However, a semi-trailing arm has slightly better steering properties than a trailing arm due to the angle found on the semi trailing arms.
This angle allows for camber effects, which the trailing arms are not capable of (Wan, 2000). In the case of the semi arm, its angles provide a trailing section and a transverse section, which provide under-steer and over-steer respectively (Wan, 2000).

Semi trailing suspensions create both over-steer and under-steer, which creates a neutral camber effect. Although this is not as great as a negative camber, it is still more ideal and stable than positive camber or under-steer (Wan, 2000). In terms of disadvantages, the trailing arm as described above is not ideal in handling since it creates quite a bit of under-steer (Wan, 2000). The semi-trailing arm however has its own disadvantage, which takes place when a bump is hit. After collision, the camber angles change providing an unpredictable response, which is dependent on the bump (Wan, 2000). This makes handling in off-road conditions more difficult. Additional disadvantages to both trailing arms are weight and high cost (Riley, 2014).

After initial research a team decision was made to move forward with a double wishbone design for the front suspension. The reasoning behind this decision was largely based on the vehicle being four-wheel drive, which means that all steering assistance available would need to be utilized. A double wishbone design will provide greater steering assistance along with camber effects. It also will add much more adjustability and has superior handling compared to the MacPherson strut as discussed earlier. The camber adjustments that a double wishbone design provides are crucial for improving handling capabilities. For the rear suspension, a double wishbone suspension will also be utilized. The reasoning for moving forward with this option was, again, based on optimizing handling and stability by utilizing negative camber capabilities.
Shock Assembly Research and Analysis

Introduction

Suspensions are one of the most critical aspects within a vehicle, without one, it would not be possible to safely control a vehicle. A suspension’s role in a vehicle is to absorb or dampen the vertical acceleration a wheel undergoes when it contacts uneven driving surfaces (Harris, 2005). This provides for a smooth ride and protects the vehicle from experiencing any unwanted forces in the vertical direction (Harris, 2005). Suspensions are designed around two main concepts, which are known as ride, and handling. Ride is the amount of vertical forces a suspension system can absorb or dampen out (Harris, 2005). Handling, on the other hand, is how well the vehicle can be controlled, which includes turning, braking, and accelerating. These are the concepts that shape the design behind our vehicle (Harris, 2005).

A suspension is a combination of several components that work together to successfully absorb or dampen forces, as well as support the vehicles weight and keep the tires on the driving surface (Harris, 2005). One of the most recognized components of a suspension system are known as springs, which expand and compress to dampen out the vertical forces presented by a wheel (Harris, 2005). There are a wide variety of different springs that can be used on vehicles, such as coil-over springs, leaf springs, torsion bars, and air springs. In terms of designing a Baja vehicle only air and coil-over springs are typically considered. This is largely due to them being relatively lightweight, low in cost, and high in functionality (Ansell, 2008).

Air Shocks vs. Coil-Overs

Coil-overs feature two main components; coil springs and shock absorbers (Douglas et.al, 2012). The coil springs provide a means of absorption for the vertical accelerations that a wheel undergoes when in contact with a rough surface or bump. This absorption is essentially the
transformation of the wheels kinetic energy into potential energy by compression (Ansell, 2008). Once the spring is compressed it will decompress and transfer the potential energy back down to the wheel. Thus causing the wheel to rebound back down to the surface (Ansell, 2008). The shock absorber controls and dampens the spring’s oscillation (Ansell, 2008). This prevents the springs from oscillating countless times until all energy is released and allows the wheels to stay on the surface (Ansell, 2008).

Air shocks function very differently. An air shock consists of a sealed air cylinder with a rod inside. The air cylinder is filled with both nitrogen and oil. The oil controls the compression and rebound of the shock and nitrogen controls the height. Essentially air shocks perform the functions of both springs and shocks in one unique system (Williams, 2006).

Air shocks are typically highly and easily adjustable, lightweight, and low cost (BajaSAE Forums, 2014). Adjustability is perhaps the biggest advantage as they are simply made by adjusting oil and nitrogen levels to acquire the desired ride (BajaSAE Forums, 2014). When compared to coil-overs they are much lighter in weight, which can be roughly 2.5 pounds for air and 8 pounds for coil-overs (BajaSAE Forums, 2014). They are also typically cheaper ranging anywhere from $200 - $250 per shock (BajaSAE Forums, 2014). Air shocks however have some disadvantages. Air shocks are less durable, require more maintenance, and are unable to rebound as quickly as coil-overs (BajaSAE Forums, 2014). Coil-overs offer better overall performance, durability, rebound rate, rebuild ability, and the ability to utilize the full length of the spring as opposed to air shocks (BajaSAE Forums, 2014). As far as disadvantages with coil-overs they are not as adjustable, and typically are more expensive with a wide range of prices depending on brand and model (BajaSAE Forums, 2014).
In order to reach a decision on the best springs for the Baja vehicle criteria was established on the functionality of the spring. One of the most important features needed would be adjustability. This is crucial because the conditions and challenges presented in a Baja competition are unpredictable. An easily adjustable suspension would be a great attribute to the competing vehicle. In evaluating the adjustability of both sets of springs the air shocks provide easier and a more versatile means of adjustment. Simply changing the levels of oil and nitrogen within the shock allow for easy and quick adjustments. The coil-overs require a special tool that alters the amount of tension or compression needed to yield the correct stiffness or softness in the shock assembly. The shocks within coil-overs are much less adjustable compared to air shocks. In a comparison of adjustability the air shocks provide more. The coil-overs, although less adjustable, are not completely fixed and do have some adjustment available.

The next criteria considered was what type of damping characteristics were needed in a shock. In order to have a fair amount of adjustability within the vehicle important concepts to consider were adjustments on preload, rebound, and compression of both high and low speed (Douglas et.al, 2012). At a minimum the team would like to acquire shocks that allow adjustability within these areas so that the team can have a multifunctional suspension that can be adjusted as needed. Fortunately both types of shocks have these capabilities.

The last criterion, and one of the most important, was the compression springs load. According to the team from last year, the vehicle was estimated to have a weight of roughly around 600 lbs including all components and driver (Atamer et.al, 2013). The assumption is made that the vehicle undergoes a 40% front and 60% rear weight distribution due to the vehicle having the engine mounted in the rear (Atamer et.al, 2013). Ideal spring loads can be calculated from these estimates and were found to be 240 lbs for the front and 360 lbs for the back. Next
was to divide these loads into two springs since both the front and rear of the vehicle would feature two springs. The results for the front were springs that could withstand 120 lbf/in and as for the back springs ones that could withstand 180 lbf/in. The calculations are equated in Appendix D: Suitable Spring and Shock Calculations.

**Comparative Analysis**

The first product found was under the Fox SAE Baja Powersports Program. This program featured an academic discount on air shocks made by Fox. Initially the team preferred this program due to the preference of purchasing air shocks because of their exceptional adjustability. The Float series shocks, which came in a set of two, were priced at $521.25. Purchasing two sets of these shocks would cost $1042.50, far above what was budgeted for shocks and springs. We began to look elsewhere for other options. Appendix C: Fox Springs and Shocks Price and Size Chart displays the information regarding these shocks.

Our next step was to find an alternative product that would fit within our budget limitations. The next company looked into was Rad-Flo and they offered 2.0” individual air shocks, which were comparable to Fox’s for around $235.00. This would amount to around $940.00, which still was over budget. Rad-Flo also happened to make 2.0” coil-over suspensions but they individually ranged from $255.00 - $385.00 depending on length, which pushed us even further over budget. Walker Evans was the last company looked at. They provided coil-over shocks at a value range of $176.00 - $196.00. Despite being the lowest yet they were still over budget coming in at $740.00 for a full set.

However Polaris had a special Baja sponsorship program that we were luckily able to participate in. We were successfully able to find coil-over shock assemblies within our price range. A full set of Polaris RZR 570 shocks would be only $362.00 after academic discounts.
The loads of the springs included within the RZR 570 shock assembly were 115 lbf/in for the front spring and 185 lbf/in for the rear springs, which is remarkably close to our calculated load rates. After additional research our team viewed these as a viable option despite having to sacrifice not getting the preferred air shocks for adjustability purposes. The Polaris RZR 570 shock assembly provides the mechanical performance, and adjustability that we need as well as meeting our financial situation as well. These springs satisfied all of the criteria mentioned above by allowing for adjustments in all the required categories as well as meeting calculated loads and price. Information on the product is detailed in Appendix B: Polaris Shocks and Springs Price Comparison. The table below displays the different specifications of each individual shock that was considered.

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Type</th>
<th>Diameter</th>
<th>Max Force</th>
<th>Total Cost</th>
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<td>Fox Air</td>
<td>1.659”</td>
<td>498 lbs</td>
<td>$1042.50</td>
<td></td>
</tr>
<tr>
<td>Rad-Flo Air</td>
<td>2.0”</td>
<td>900 lbs</td>
<td>$940.00</td>
<td></td>
</tr>
<tr>
<td>Rad-Flo Coil</td>
<td>2.0”</td>
<td>400 lbs (Without spring)</td>
<td>$1020.00</td>
<td></td>
</tr>
<tr>
<td>Walker Evans Coil</td>
<td>2.0”</td>
<td>500 lbs (Without spring)</td>
<td>$740.00</td>
<td></td>
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<tr>
<td>Polaris Coil</td>
<td>2.06”</td>
<td>300 lbs (Without spring)</td>
<td>$362.00</td>
<td></td>
</tr>
</tbody>
</table>
Power Delivery Research and Analysis

The final drive analysis explains how power will be taken from the engine and transferred to the transmission in the Baja vehicle. This will consist of a mechanical linkage from the engine output shaft to the transmission’s input shaft.

Engine Chain vs. Belt Drive

A manual Yamaha Big Bear 400 5-speed transmission has already been chosen for the vehicle. By choosing a manual the drive type out of the engine becomes much simpler. With a manual transmission a gear or belt reduction is used to deliver power from the engine’s output shaft directly to the input shaft of the transmission. However, if a CVT transmission were selected then there would have to be a planetary gear reduction or other form of reduction between the engine and the transmission which would have added weight and created a weaker area for failure within the drivetrain (McCausland, 2010). Therefore, the manual transmission helps to save weight in this regard and the decisions left to be made are whether to use a belt or chain drive and also what ratio should they be set to.

Chain Drive

A chain is the most reliable option for the drive in a Baja vehicle. The average power efficiency for a chain is 98% meaning that 98% of the power the engine outputs is transferred to the transmission through the chain and sprocket set up. However, chains are heavier than the conventional pulley system and would add additional weight to the back of the vehicle.

In a chain a sprocket drive system the first component to break would be the chain. The equation below shows the direct load due to tensile load on a chain. F is defined as the total load on the chain (pounds force), P (watts) is the power output from the engine as a function of torque.
which can be taken from a graph when cross-referenced with the toque curve, \( v \) is the linear velocity of the chain as a function of RPMs that the engine is running at.

\[
F_t [lb] = \frac{P_1 [W] \cdot 1000}{v [RPM]} + \frac{P_2 [W] \cdot 1000}{v [RPM]} + \cdots + (Nth \ numbers \ of \ torques \ on \ system)
\]

Equation 1: Direct Load Due to tensile load on a Chain

There will only be one term in the equation for the team’s uses because the only torque that the chain is subjected to is from the engine output shaft. Also the centrifugal force put on the chain is equal to (Beardmore, 2013):

\[
F_c [lb] = M_c \ [lb] \cdot v^2 [RPM]
\]

Equation 2: Centrifugal force put on a chain

Where \( M_c \) (pounds) is the mass of the chain. Therefore the total load the chain must hold is equal to:

\[
F_{total} [lb] = F_t [lb] + F_c [lb]
\]

Equation 3: Total Load on the Chain

In order to deal with these forces acting upon the chain the proper chain must be chosen.

Although the exact chain cannot be chosen because the distance between the engine output shaft and the transmission input shaft is not yet know, general characteristics of the chain can be found using estimates. Using the equations above it was determined that a size 35 chain would be
necessary for the Baja vehicle. The size 35 chain has a 3/8 in. pitch, .200 in roller diameter, with a maximum tensile strength of 1500 lbs and a measuring load of 31 lbs.

Figure 7 shows a simplex, duplex, and triplex chain, which could all theoretically be used on the Baja vehicle. A duplex or triplex chain can be used when space constraint is necessary because of their higher tensile strength. They can also be used in situations when lower pitch or high speed chain rotation is necessary. However, although as already stated the team does not know the exact distance between the engine and transmission shafts an estimate can be used to determine that a size 35 simplex chain can be used for the vehicle application. This will cut down on cost and weight for both the chain and sprocket by using a simple simplex setup (SIT, 2014). The cost and weight are both reduced by just under 50% using standard pricing and weights off of McMaster- Carr. Another advantage that a size 40 chain yields is the ability to use aluminum sprockets. This will lessen the weight to help take some weight out of the back of the vehicle. Through the Polaris SAE sponsorship a complete chain and sprocket set with a gear reduction of 4:1 can be purchased for $80.00.

Belt Drive

A belt drive is the lightweight solution to the final drive that consists of two aluminum pulleys and thin lightweight flexible material for the belt. They have a slightly lower efficiency than a chain system, usually around 95%, but are susceptible to higher efficiency losses due to slips. Baja is a high vibration environment with almost continuous shocks going through the system which could drastically lower the efficiency of the belt (Stubs, 1994).

As with the chain and sprocket system where the chain would be the first failure point, the first component to fail in a belt and pulley system is the belt. The equation for the centrifugal force, \( F_c \), related to tensile force, \( T_c \), and angle of belt lap, \( \theta \), on the belt equals:

\[
F_c = \frac{T_c}{2} \left( \frac{\theta}{180} \right)^2
\]
\[ F_c [\text{lb} \cdot \text{rad}] = 2T_c [\text{lb}] \cdot \frac{d\theta [\text{rad}]}{2} \]

**Equation 4: Centrifugal force on the belt**

Where \( m \) is the mass of the belt and \( v \) is the linear speed of the belt. The tensile force of the belt are analyzed as a ratio of the two sides and is defined as:

\[ T_c [\text{lb}] = m [\text{lb}] \cdot v^2 [\text{RPM}] \]

**Equation 5: Torque**

where \( \mu \) is the coefficient of friction of the pulley (Beardmore, 2013).

\[ \frac{T_1 [\text{lb}]}{T_2 [\text{lb}]} = e^{\mu \theta t} \]

**Equation 6: Tensile Force Ratio**

After using the same estimates used in the chain analysis it was found that the forces are close to equal between the two with actually slightly lower forces acting upon the belt. However, whereas the chain is made from steel, the belt is made from a flexible rubber or polyester. This means that over time the belt will stretch and distort at a much greater rate than the chain which will cause increase in efficiency losses over time. A standard belt and pulley set that could be used on the Baja vehicle costs around $75.00 if no back up belts are purchased.

**Final Decision**

The team has decided to use a #35 chain in order to properly attach a centrifugal clutch (discussed further in the report) from the engine to the transmission. Chains are also more reliable than belts and are much less likely to either break or slip off track. A #35 chain is more than durable enough to run the vehicle application successfully and is a cost effective method driving power from our engine.
Brake System Research and Analysis

Purpose

There are three basic functions of a braking system are as follows:

- Decelerate a vehicle including stopping.
- Maintain vehicle speed during downhill operation.
- Hold a vehicle stationary on a grade.

(Limpert, 1999)

These functions listed above are a result of forces generated between the tires tread and the surface that it contacts. The determining parameters of a braking car and how a brake system performs are correlated to velocity, V (feet/second), deceleration, distance and time, t (seconds). Shown in Equation 7: Velocity & Acceleration.

\[
V = \frac{Distance [ft]}{Time [s]}, \quad a = \frac{\Delta V}{\Delta t} = \frac{V_2 - V_1}{t_2 - t_1}
\]

Equation 7: Velocity & Acceleration

The brake system chosen was chosen considering the following criteria:

- Reliability (safety and durability).
- Material selection (cost vs. strength, weight, life).
- Assembly (cost effective manufacturing, maintenance, serviceability).
- Economics (overall cost consideration).
- Failure Analysis (Effect of critical failure).

The significance of the parameters are variable and differ than that for a passenger vehicle, but the same design concepts and specifications were utilized to determine the best brake system for the singular purpose of the Baja SAE vehicle that can produce a sufficient
braking force greater than what Baja SAE competition requires, Appendix A: SAE Rules and Regulations. Most importantly the system must be “capable of locking all four wheels, both in a static condition as well as from speed on pavement and on unpaved surfaces,” (SAE Rules, 2014).

Per SAE regulations they require a foot brake that operates a hydraulic braking system and runs on two separate circuits increasing safety in case of a circuit failure, partial braking will still be provided.

The specific brake system selected is one part of the design not specified by SAE Baja regulations. This design decision comes down to two brake systems that utilize friction to slow the rotation of the wheels, either drum brakes or disk brakes.

**Drum vs. Disc Brakes**

The two systems convert kinetic energy into thermal energy in two different mechanisms. Drum brakes apply a force via brake shoes radially to the drum of the wheel creating friction. A disc brake applies friction via pads axially onto a rotor.

Disc brakes have been used and are more readily used in passenger vehicles on the road today, with drum brakes making up the majority of brake systems used in medium to heavy duty trucks and tractors in North America. Long repetitive breaking results in increase in overall temperature of both the plate and brake fluid with potential to cause brake fade and thermal cracks. However, 93.4% of the energy produced in disk brakes at the point of contact between the disk and the pad is absorbed by the disc. The absorption and free surface on the disc allows the heat to be dissipated via heat convection. (Talati, Jalalifar, 2009) Drum brakes tend to vary in braking torque depending on the road conditions because of the ability of water or debris to
interfere with the shoe and drum interaction. Brake torque is mathematically predicted via a dimensionless number, brake factor (BF). This number represents the brake torque of a certain system. Figure 8 displays, the BF for drum brakes is considerably higher than that of disc brakes.

\[ \text{BF} = \frac{F_d \ [\text{lb}] }{F_a \ [\text{lb}] } , F_d = \text{rotor drag}, F_a = \text{application force} \]

Equation 8: Brake Factor (Limpert, 1999)

![Figure 8: Brake factor vs coefficient of friction (What-when-how, 2014)]

**Brake System Selection**

The Baja SAE vehicle will utilize disk brakes rather than drum brakes. Disk brakes do not fade in brake ability at high temperatures where as in drum brakes they are highly temperature sensitive and cannot deal with the same type of temperatures that disc brakes can. Temperature may become an issue in the competition as repetitive braking will occur navigating around and over obstacles. Disc brakes are also much easier to assemble and service, disk brake adjustments are automatically occur in hydraulic systems. Also in the case of disc brakes the relationship between bake torque and pad friction coefficient is linear increasing the ability to specifically adjust to the required braking torque (Limpert, 1999). Table 2 shows the benefits of
a disc brake system because the brake (shoe) factor, although low, requires a low relative braking power and has high stability, meaning it maintains the same relative BF when conditions of the friction material are changing due to either wetness or temperatures.

<table>
<thead>
<tr>
<th>Type of brake</th>
<th>Shoe factor</th>
<th>Relative braking power</th>
<th>Stability</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single trailing shoe</td>
<td>0.55</td>
<td>Very low</td>
<td>Very high</td>
</tr>
<tr>
<td>Two trailing shoes</td>
<td>1.15</td>
<td>Very low</td>
<td>Very high</td>
</tr>
<tr>
<td>Disc and pad</td>
<td>1.2</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>Single leading shoe</td>
<td>1.6</td>
<td>High</td>
<td>Low</td>
</tr>
<tr>
<td>Leading and trailing shoes</td>
<td>2.2</td>
<td>Moderate</td>
<td>Moderate</td>
</tr>
<tr>
<td>Two leading shoes</td>
<td>3.0</td>
<td>High</td>
<td>Low</td>
</tr>
<tr>
<td>Duo servo shoes</td>
<td>5.0</td>
<td>Very high</td>
<td>Very low</td>
</tr>
</tbody>
</table>

Table 2: Brake Layout Comparisons (What-when-how, 2014)

Since disc brakes are sub divided into floating and fixed caliper. The use of floating calipers proves to be the better choice for the Baja SAE Vehicle. First and foremost they are compact and cheaper because they involve less parts, as they only have hydraulic pistons on the inboard side rather than on the outboard like fixed calipers do. Floating calipers, Figure 9, are proven to be as effective and require less power than a fixed caliper (Limpert, 1999).
Brake System Product Selection

After considering the various system types and concepts, WPI Baja SAE vehicle utilized a mechanical driver pedal effort with hydraulic brakes that are a dual circuit system that applies the friction using floating caliper disc brakes. The next step is to consider multiple manufacture options that will satisfy these desired specifications. Our desired system is a complete assembly designed for an all-wheel drive utility vehicle or All-Terrain Vehicle. These systems will be most consistent with the braking results we are looking to produce on the SAE Baja vehicle. John Deere, Polaris and Husqvarna vehicles were all manufactures considered.

Final Decision

The decision to go with a Husqvarna HUV front brake assembly was due to its high brake factor and independent four caliper system. Braking capabilities will not be an issue on the vehicle ensuring safety for the driver and security in case of a failure in a different subcomponent in the vehicle. The decision to purchase all four wheels was also supplemented due to the cost effective option we were offered at Morrison’s Power Equipment in Plymouth, MA. The ability to get all parts to the assembly at a discounted price allowed us to get a high quality brake system and allocate additional funds to other areas of purchase within the design and build.
Clutch Research and Analysis

Function

A clutch is the mechanism that connects and disconnects the engine from the transmission. A fully functional clutch, when a force is applied to the pedal, disengages the clutch allowing the engine to disconnect from the transmission when the desired gear ratio within the transmission is being shifted to. The most desirable application of the engine power to the transmission is gradual, to create a smooth transition limiting the shock on the driving parts as well as no slipping when the engine power is being transmitted to transmission. Friction is applied via a pressure plate engaging or disengaging the driven plate to the flywheel, shown in Figure 10. The clutch is engaged while there is no pressure being applied to the pedal but when the pedal is depressed this allows for the gears to be changed as the drive plate is not connected to the flywheel. The ability of a clutch to function properly and transmit torque relies on the following factors:

- Applied Pressure.
- Coefficient of friction of clutch frictional surfaces.
- Surface area of frictional surfaces.
- Internal and external diameter of driven disc (Thiessen, 1996).
Clutch Components

A clutch is comprised of multiple components that allow the power of the engine to be delivered to the drive wheels. These components are as follows:

- Flywheel.
- Clutch Disc.
- Pressure Plate.
- Control Linkages (Thiessen, 1996).

Flywheel

The flywheel is an essential part of the clutch and the system delivering power to the transmission. First and foremost the flywheel is located on the back of the engine and is bolted onto the crankshaft and is responsible for absorbing vibrations and provide momentum for continuous engine operation while the clutch is disengaged. The flywheel has a flat surface that is machined so the clutch disc and pressure plate can be attached allowing the transmission and engine to be connected and disconnected.
**Clutch Disc**

The importance of the clutch disc is that this is the part of the clutch that provides the friction to the flywheel providing the power from the engine to the transmission. The clutch disc is a flat piece of metal with multiple facings that are wear resistant and have facing that are heat resistant because of the high heat production of friction. The disc as well utilizes steel segments that are twisted to create a gradual contact when the disc and fly wheel make contact minimizing the vibration. Minimizing the vibration in the crankshaft that comes from the clutch disc is a major concern because if not eliminated will cause the vibration to reach the powertrain, so a flexible center is used around the crankshaft to absorb the torsional vibration. (Thiessen, 1996)

**Pressure Plate**

The pressure plate is primarily designed to produce a constant force that is applied against the clutch disc holding the disc tight to the flywheel allowing for the power of the engine to go from the engine to the transmission. The pressure pate housing is mounted to the flywheel on the engine and the assembly inside contains the pressure plate that can be released from its tight fit against the flywheel using a release lever. Either coil springs or diaphragm springs produce the pressure of clutch disc against the flywheel. (Thiessen, 1996)

**Control Linkages**

Now that the components of the clutch are explained, the way these components are activated and either engaged or disengaged are described as;

- Mechanical Clutch Activation.
- Hydraulic Clutch Activation.
These two activation methods provide distinct advantages and disadvantages specifically to the implementation in the SAE Baja vehicle (Thiessen, 96).

A mechanical clutch, Figure 11, is much cheaper than that of a pneumatic or hydraulic clutches. The decrease in price comes with the decrease in range of torque application ability. More commonly than not mechanical clutch activation is used when the clutch itself is located close to the pedal because there is a series of either cables or rods directly connected to the release bearing lever. The distance however can be an issue because the friction loss may become high. A direct mechanical linkage is preferred over a cable one because of the immediate feedback received. Additionally, the state of wear on the clutch can be felt by driver, because the pedal free play indicates the need for clutch replacement. (Ebsco, 2014)

Hydraulic clutch activation utilizes a master and slave cylinder to deactivate the pressure plate. The main purpose of this type of activation, shown in Figure 12, is because of the unique

Figure 11: Mechanical Clutch Linkage (Ebsco, 2014)

Figure 12: Hydraulic Clutch Activation (Eaton, 2014)
ability of the hydraulic system to self-adjust its linkage system. This system requires a smaller applied force by the driver than that of a mechanical system. A hydraulic system is more compactable because the use of the fluid, but the loss of pedal free play is a disadvantage. (Eaton, 2014)
Throttle Research and Analysis

One of the most curious systems within SAE Baja buggies is the throttle assembly. This system includes a very intricate pedal system and a less intricate cable system to transfer the pedal motion to the throttle on the engine. Research shows that many teams spend a considerable amount of design time on the pedal assembly, but not enough on the cable assembly. An example of this can be observed within a post from a user on the SAE-Baja forums. A question is posed as to what is the most common area of failure within Baja buggies, and a post simply responds, “throttle cable… probably because it rarely [gets] the design time it deserves.” (BajaSAE Forums, 2014). In order to avoid this area of failure, our team plans to give the entire throttle system the necessary design time.

Throttle Cable

The throttle cable must be covered from the front pedal back to the engine (BajaSAE Forums, 2013). There are three practical options for our buggies throttle cable including basic bike cable, push/pull cable, or fully assembled throttle cable. Each cable has its own pros and cons as will be discussed.

Basic Bike Cable

Research from the SAE Baja forums revealed that many teams opt to use bicycle brake cable as their throttle cable. Bike brake cable is covered, very cheap, light, and can be modified very easily (easy to route around the buggy and can be easily cut to length). One drawback with this cable is mounting it to the pedal and engine throttle. Using a set screw to hold it in place has the potential to slip and fail, but this problem can be fixed by adding a solder joint (BajaSAE Forums, 2014).
**Push/Pull Cable**

A push/pull cable is very similar to bicycle brake cable. Instead of having the woven cable come out from each end of the cable, it is completely sealed and has linear sliders perform the visible movement. This cable is more durable than bicycle brake cable, and the friction between the woven cable and the insulator is much less than within bicycle brake cable. Jegs.com offers a completely non-friction model of this cable (Jegs.com). The linear slide can be mounted to bolt onto the engine and work with the throttle very effectively, as seen in Figure 13. Unfortunately, this cable is much less modifiable as it is thicker and has less freedom to bend and be routed around the buggy. It also poses an issue with being cut. Cutting it can be done, but requires a weld and retubing (BajaSAE Forums, 2014).

**Throttle Cable**

![Figure 13: Push/Pull Cable Mounted to Throttle (SAEBaja Forums, 2014)](image)

Another option for our throttle assembly is purchasing a fully assembled throttle cable. This design was utilized by the Northern Arizona University team and they commented on the success they had with this cable (Abdulrahman, 2014). A preassembled throttle cable saves design and implementation time by having most of the connections already set up. Besides fitting...
length, the cable is ready to be installed and used. The drawback with ordering throttle cable is the price, as they are the most expensive of the three options and are not always found in the correct length. They are also not as flexible to be fed through the buggy as the bike brake cable. Polaris offers throttle cable at a reasonable price, but only with three available lengths (Polaris). Lokar offers length customizable throttle cable, but is expensive (Lokar).

**Selection of Throttle Cable**

Polaris offers a push-pull cable to the length that the vehicle would require while still ensuring there is enough play in the cable. Therefore, due to the sponsorship established with Polaris and the reliability of the part, this cable will be used. A simple rig will be made to attach the cable to the throttle on the engine and the cable will be tied off with proper service loops to the frame.

**Throttle Pedal**

The SEA Baja competition mandates that only foot operated throttle controls are allowed (SAEBaja Forums, 2014). A foot pedal assembly used in tandem with any of the throttle cables listed above requires a pivot motion so that when the brake pedal is compressed downward, the top of the pedal moves upward, thus pulling the cable and the throttle on the engine.
Polaris offers a simple gas pedal assembly off of the Polaris Ranger. It utilizes a single pivot pedal with a return spring. A schematic of the Polaris gas pedal can be observed in Figure 14. The main issue with using this pedal assembly is its compatibility with our design. Without specific dimensions and its throw range listed, we cannot know if this assembly will fit in our frame and work with our engine throttle. The price of the assembly and its availability are further issues, as no price is listed.

Figure 14: Polaris Ranger Gas Pedal (http://parts.polarisind.com/Assemblies.asp)

Many teams, such as the Northern Arizona University team, scrapped the idea of buying and implementing an OTS pedal assembly, and decided to design and manufacture their own. This option is the better option for us as well. The process of designing a pedal assembly is underway that will fit the frame, work as a lever-pivot, and provide 1 3/8\text{th} inch of movement (the movement required for our engine throttle) (Abdulrahman, 2014). The materials needed for this design can be found in house, thus saving money.
Throttle Analysis

The team budgeted $200 for the throttle assembly in our original budget. Following this research and using brake cable with a custom throttle pedal, we believe that we can assemble the throttle assembly for much less than $200. Freeing up this money will allow us to spend more in other important areas including the shocks and springs.
Transmission Research and Analysis

CVT vs. Manual Transmission

In picking the transmission the team first had to decide what type of transmission should
be included in the vehicle. For a Baja vehicle it breaks down to two main options that are
compatible with the application, a Continuously Variable Transmission (CVT) or a manually
shifted transmission.

**CVT**

A CVT adjusts engine torque in an infinite number of gear ratios between a pre-
determined maximum and minimum value. This infinite range allows the output shaft to spin at a
constant velocity which in turn allows the vehicle to function at the most efficient rotations per
minute (RPM’s) at any given time (Rahman, 2014). This is ideal for a small car that is looking to
save fuel efficiency or to maximize power output. The faster the engine is allowed to run means
a higher performance output at a cost of reduction in fuel efficiency or vice versa where the
slower the engine turns means less power output but much greater efficiency.

The advantages of a CVT is the variability of the tuning options. This will allow the team
to set the transmission in the position that will give out maximum power without having to burn
through fuel. Another advantage is no clutch is needed in the vehicle with this transmission
(Rahman, 2014). A simple clutch could be added in order to create a neutral position for when
the car needs to be moved without being ran but would be an easy addition to the car.

The main disadvantages of a CVT come in its components and how it is tuned. A CVT
contains a torque converter, drive-neutral-reverse gear set, a variator, an actuation system that
can differ between models of CVT’s, and a final drive. The tuning is done through changes of the clamping forces of the sheaves of the main pulleys in the variator.

Figure 15 shows how the pulleys change between their minimum and maximum positions in order to have an infinite number of gear ratios. As can be seen in the clutch variator sheaves, the clamping forces can be manipulated in order to tune the maximum and minimum values for the pulley. The main power loss of a CVT comes from the clamping forces that are used to tune it (Meulen, 2012). The extra slip conditions that are generated from a higher clamping force between the pulleys and the actuators create power loses and instabilities in the transmission that can lead to unreliability of the system. Along with unreliability the CVT systems are used for low torque applications in car systems. Although the Baja vehicle will only be running a 10 HP engine the four wheel drive system that will be implemented will produce enough torque that a CVT transmission would not hold under the conditions.

![Figure 15: CVT Pulley Variability/Tuning (HighGain Tuning)](image-url)
Manual

Manual transmissions give the user the ability to switch between numerous gear ratios that are built into the transmission. The two main forms of manual transmissions are sequential and non-sequential transmission. A sequential manual is most commonly found on motorcycles and forces the driver to shift to only the previous or next ratio in the series. A non-sequential transmission which is found in most commercial cars and ATV’s give the user the ability to switch between any of the forward gear ratios and any point in the driving sequence. A non-sequential transmission would give the team two key advantages over a sequential one. The first is acceleration can be maximized by being able to rev at high RPM’s before launch because of the option to choose the revving gear ratio. Also, a higher rpm setting can be chosen easily while driving which would enhance the cars ability to climb hills which will add to the effectiveness of the 4 wheel drive system (George, 2013). Another major advantage that a manual transmission possesses is the option for a reverse gear. The team would not be able to find or afford a CVT with a reverse gear and this would lessen our chances of succeeding if the competition contains a rock garden event. This event would require the vehicle to pick the best possible path and possibly have to reverse if the vehicle gets stuck at any point throughout the event. A reverse gear would greatly increase the cars ability to receive a perfect score in this event.

The main disadvantage to a manual transmission is the loss of drivability compared to a CVT transmission. Whereas a CVT is constantly shifting between an infinite range of gear ratios, a manual transmission is limited to however many are built into the system. This creates the loss in drivability and is also what leads to the manual’s lower fuel efficiency (Forth, 2013). Also, the inability to tune a manual transmission as much as you can a CVT creates a less form fitting system. However, for Baja purposes only two forward gears, a high and low gear, will
have to be utilized. This means that the two best ratios can be picked from the 3-6 forward gears that come in manual transmissions. A manual transmission, unlike a CVT, will also be able to handle the amount of torque that will constantly put on the system. This gives the manual transmission a much greater reliability over a long period of time (i.e. the endurance test) than a CVT.

A manual transmission will also require a clutch system to be able to shift between gears. This is not too much of a restraint however because there are multiple ways that this can be done. The two simplest methods would be either a hand or foot clutch that is linked to the clutch mechanism on the transmission by a simple cable. The choice between a hand or foot clutch would rely almost solely on driver/team preference and comfort level. However, space constraints could force one of the options to be used so the team will not make a final decision on this until further along in the build.

**Final Decision**

The team has decided to use a manual transmission in the Baja vehicle. A manual transmission will have a greater reliability than a CVT. Also, the ability for a reverse gear will be instrumental in making sure the car can receive the most points possible in any events that are required in the 2016 competition. Also, the manual transmission will allow for a better score in the acceleration event and any event that requires moving uphill. The loss of fuel efficiency and tuning options will be small in comparison to the gains that we will receive from a manual transmission.

**Manual Transmission Options**

Now that the team has decided on a manual transmission multiple options were looked into. The options below were not the only transmissions considered, but they were the final
choices among the team. They also show the different range of manual transmissions that were considered to ensure that we selected the proper system for our goals.

Yamaha Big Bear 400 4x4

The Yamaha Big Bear 400 4x4 Transmission is a constant mesh 5-speed manual with a reverse gear. The transmission is advertised as a “super low” first gear ratio (the exact gear ratio is not released by Yamaha). Although the gear ratio has not been given we have encountered numerous reviews and reports that prove the torque output at in first gear is very high compared to other ATV models. This will be a huge help if there is a sled pull event in the 2016 competition.

A downside to this transmission, Figure 16, was the weight of it. Because it comes off of a 400cc four wheel drive ATV model is heavier than other transmissions that we have found. This transmission would also have to be ordered without a case because it is a part of the engine assembly when mounted in the ATV. If the team were to buy the engine assembly, it would be required to remove the transmission anyways. Therefore, the best course of action would be to buy it in either individual pieces or with the shafts already constructed as displayed in Figure 16. An engine assembly out of the Big Bear would cost around $900.00 whereas buying individual components will be around $400.00.

Figure 16: Big Bear 400 4x4 Transmission
Whether the team were to buy an engine assembly or individual components, a transmission case will have to be made so the components can be easily mounted onto the chassis. This transmission case will be designed by the team once the transmission is received and dimensions can be taken. The case will be a completely enclosed structure so transmission fluid can be properly used and will also have mounting brackets built into the design.

**Mahindra Alfa**

The Mahindra Alfa transmission is a 4 speed with reverse manual transmission. The Alfa uses a 395cc on a vehicle that weighs 1015 lbs. The four gear ratios can be found in Figure 17.

<table>
<thead>
<tr>
<th>Gear Type</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chain-Sprocket reduction</td>
<td>1.187</td>
</tr>
<tr>
<td>Initial input reduction</td>
<td>2.869</td>
</tr>
<tr>
<td>First gear ratio</td>
<td>4.6</td>
</tr>
<tr>
<td>Second gear ratio</td>
<td>2.733</td>
</tr>
<tr>
<td>Third gear ratio</td>
<td>1.667</td>
</tr>
<tr>
<td>Fourth gear ratio</td>
<td>1.115</td>
</tr>
</tbody>
</table>

*Figure 17: Mahindra Alfa Gear Ratios*

For Baja use the 4.6:1 first gear ratio is fairly high. This would mean that first gear would rarely be used and acceleration would be lost by starting in second gear. This first gear will simply add weight to the transmission that would never be used. An advantage the Alfa transmission has over the Big Bear is the Alfa transmission comes in a transmission case that could be used. This means no case would have to be made and the only concern would be to use the existing mounting capabilities already found on the transmission in order to mount it to the frame. Figure 18 shows the Mahindra Alfa transmission case.
Figure 18: Mahindra Alfa 4-Speed Manual Transmission

The major downside to this transmission is the pricing. The transmission alone cost $600.00 which is the entire amount allocated by the team for a transmission. However, this part could only be shipped from India and the shipping and handling could cost roughly another $100.00 which would put the component over budget. Also, the transmission is known to have some overheating and unreliability issues due to fast component wear down. This would mean the team would have to order parts from India at the cost of high shipping and handling in order to fix even a small problem in the transmission.

Harley Five Speed

The next transmission that we reviewed was a Harley Screamin’ Eagle five speed transmission. This model Harley can come with both a five or six speed transmission and the gear ratios can be found in Figure 19.

<table>
<thead>
<tr>
<th>Gear</th>
<th>5-Speed</th>
<th>6-Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>3.210</td>
<td>3.210</td>
</tr>
<tr>
<td>2nd</td>
<td>2.109</td>
<td>2.209</td>
</tr>
<tr>
<td>3rd</td>
<td>1.572</td>
<td>1.572</td>
</tr>
<tr>
<td>4th</td>
<td>1.226</td>
<td>1.226</td>
</tr>
<tr>
<td>5th</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>6th</td>
<td>NA</td>
<td>0.885</td>
</tr>
</tbody>
</table>

Figure 19: Harley Davidson Screamin’ Eagle Flight gear ratios
Throughout the Baja competition there would not be a point where the vehicle would have to shift all the way up to a sixth gear. Also, a lower second gear ratio in the five speed would be better for hills and acceleration so the five speed was chosen over the six speed option. This transmission would cost the team $650.00 dollars, which is $50.00 over the allotted amount.

A Harley transmission would be extremely reliable and an extremely smooth clutch system but there is a major drawback in this transmission. Because the transmission is found on a motorcycle it does not have a reverse gear. With a four wheel drive vehicle, a reverse gear is instrumental in getting the largest advantage we can on the rock crawl event which is where four wheel drive will excel. It is possible to purchase a kit to add a reverse gear but the cheapest one found cost $1,050.00. This means that even though we would not need a custom transmission case, like with the Alfa transmission, in order to add a reverse we would have to spend a significant portion of our budget.

**Final Decision**

The team has decided to go with the Yamaha Big Bear 400 4x4 transmission. Even with having to design and machine a transmission case the Big Bear transmission will come in at least $200.00 under the other two options assuming the team would only have to pay for stock for the transmission case. Also, the low first gear ratio will allow 4 wheel drive to be fully utilized in events such as the sled pull and rock garden events. By choosing the Big Bear transmission the team is enhancing the areas of strength the car is designed to have in order to receive maximum points in the large majority of events.
Methods

Suspension Design Process

Based on the control arm research detailed earlier both the front and rear control arm designs would utilize a double wishbone allowing us to optimize the handling ability of the vehicle. Figure 20 depicts the preliminary designs that utilized bushings and bolts to attach to the frame allowing for vertical travel. A ball joint was utilized for the wheel hub attachment which allows for vertical travel, camber, and horizontal travel. The design meets the allowable 64” track-width whether the arms are compressed or stretched.

The same concept used on the front arms was used on the back control arms just with minor modifications to accommodate the rear geometry. The length of the arms and the width of the upper arm were altered to maximize space at the top of the control arms and reduce the amount of strain possible on the upper member of the frame. Figure 23 illustrates the dimensions of the rear control arms along with the preliminary CAD design of the rear control arms.
The wheel hub and rotor assembly design was reliant on the flexibility of the bearings that were utilized. The flexibility of a ball joint is critical as it provides improved handling of the vehicle with a greater range of movement. The initial bearings that were chosen could travel up to 65 degrees, however the 75-degree movement of the ball joints that came included within the hub of the Husqvarna HUV were viewed as a better option. Measurements of the hubs were taken followed by the bearings being pressed out of the hub. After that measurements of the bearing were taken to recreate it as well as make a fixture within the arm to house the bearing. These measurements allowed us to reengineer the wheel hubs and ball joint in solidworks to properly fit the wheel hubs and bearings shown in Figure 21 and Figure 22.
The design change noted in Figure 24 and Figure 25 caused for a redesign of both control arms. The previous design had a round bar in the middle; this was changed to a rounded plate with a hole through it so that the bearings could be pressed into the arm. This redesign occurred for both control arms as seen in Figure 22 and Figure 25. This change in design needed minor adjustments to dimensions in order for the wheel hub and ball joints to fit simultaneously.

The original placement of the shock in Figure 24 was problematic because of the interference it created with the axle going through the middle of the arms. Another minor issue was that the shocks consumed more space than originally planned for. Thus making the shocks
attach to the bottom control arm difficult. Initially to overcome this issue the idea was to move the shock to the upper control arm. However this approach required an additional bar to be placed on the frame so that the shock could be attached properly, Figure 26.

After having a design review this idea shown in Figure 26 was not feasible. Further research of similar Polaris vehicles with similar shocks led to a concept that provided a simple way to connect shocks to the frame while still allowing the axels to go through the middle of both arms. The concept involved widening the control arms so the shocks could be attached to one side of the arms with the axel going through the middle of both arms, Figure 27. The complete view shows that the components fit into this third control arm design and provided a robust system to connect the shocks to the frame.
Fourth and Fifth Design Iterations

After defining the location of the shocks the next iteration of the control arm design mainly saw changes on how the shock was mounted to each arm. The change in mounting was that the shocks were now mounted to two tabs that were welded onto a plate that runs in between the bottom arm. The previous iteration had the tabs on each end of the arm. Changing how the shocks were mounted was needed to fit the axel within the middle of the arms. The front and rear designs can be seen in Figure 28.
The fourth iteration was nearly a complete design however the only issue arose with the rear arms not being wide enough for the axle to fit. Following that realization a fifth and final design iteration was needed. The front control arms remained the same as the previous iteration. However the rear arms changed quite drastically as can be seen in Figure 29, several bends were added to widen the arms so that the rear axles would fit properly.

After finalizing the fifth design iteration the drawings of each arm were sent to Worcester Manufacturing for bending. Afterwards the arms were sent to City Weld to be welded finalizing
the manufacturing process. This process is discussed in much more detail in the suspension results section.
Steering Design Process

The most important criterion for the vehicles steering system was to produce a fully functioning system at the lowest cost possible. The first step of the process involved looking at in house components and determining if any of them were acceptable. The reason cost was so critical was largely due to the welding of the control arms taking out a much bigger portion of the budget than originally anticipated. Luckily after evaluating the in house components within the SAE club shop useful components were found that would significantly lower the cost. The main component that was found was a previously used steering rack which easily saved $200 - $300. The next step after evaluating that the rack was functioning was to measure its dimensions and reengineer the rack into CAD.

Once the rack was successfully transitioned into CAD the first issue in the design process was made very clear. The first issue encountered was the extremely tight space found within the foot well of the vehicle. This severely limited the positioning of the steering rack. The front of the foot well was predominately taken up by the pedals while the middle was taken up by the T-Box. In order to overcome this issue a mounting bracket was modeled in solidworks. The mounting bracket was designed to mount directly over the T-Box to use the remaining space within the foot well efficiently. Figure 30 shows the purple steering bracket that mounts over the T-Box.
After successfully positioning the steering rack within the foot well the next part of the design process was to design tie rods, which would connect the wheel hubs to the steering rack. Luckily the hubs that were purchased prior to the school year featured a little bracket hanging off of the hub meant directly for a tie rod to attach too. The tie rods that were designed consisted of a steel rod that would then have two nuts welded at each end. The nuts would then be utilized as fixtures for the two ball joints that would be attached at each end. Two types of ball joints were utilized for the tie rods. On the hub end a ball joint with a pre-assembled shank was utilized to allow for vertical travel. However on the steering rack side a ball joint with a bolt through the middle was utilized to assure the tie rods are securely fastened to the rack.

For the rear very similar tie rods were designed however instead of attaching to a rack they were directly attached to the control arms. This locks the rear wheels in place so they cannot rotate in any way whatsoever. In addition to these components a steering wheel was purchased and a simple collar was turned on a lathe to attach it to the steering rod. Afterwards the steering rod was cut to a length suitable for a driver. Once the steering wheel mount was made the
steering design was essentially complete. There was only one design iteration due to the components already existing and the hubs already being made to have tie rods attach directly to them.
Transmission Design Process

As discussed in the Background Research section of the report the team has decided to use a 5-speed manual AVT transmission out of a Yamaha Big Bear 450. The team purchased the shafts, gear, shifting forks, and collar of the transmission from a reputable user on eBay for $30.00. By buying the parts of the transmission without a casing meant that the team would be required to design and build a case in order to house the transmission and transmission fluid. This meant that complex geometries and bearing systems would have to be designed in order for the transmission case to be able to completely operate the transmission while also sealing it so there would be no fluid drips resulting in automatic disqualification at competition.

The original goal for the transmission case was to design and cast an aluminum casing with the capabilities found at WPI by using the Foundry and 3-D printing technologies that can be found on campus. In order to fully understand what this would require out of us we contacted Pat Guida (WPI Class of 83’) who has experience in casting and automotive design and manufacturing techniques. He instructed us that it would be possible to Green-Sand Cast a case on campus as long as the team could design the case in a way that a 3-D model could printed and used in the process. However, he warned us that designing a case to fit these design goals would be a very in depth process that may not be plausible within the scope of our MQP. After further looking into what designing a case like this would entail with the geometries and methods involved the team decided to move away casting a case in order to stay within the budget and time restrictions of the project.

Pat further suggested that the team attempt to find an old scrapped transmission case for the transmission in order to be able to take dimensions and concepts off of it. This would mean that a simpler transmission case could be manufactured from plates that would basically make a
box around the transmission. O-rings and rubber housing for shafts could be used along with RFD in order to seal the case and it would be both cheaper and easier to manufacture than a casted case. However, the team was unable find a housing for the transmission that can be modeled off of. The major issue is that in Yamaha ATV’s the transmission is housed in the same casted piece as the engine so it is a large block that people scrap for money if they were to have one from a broken ATV. Also, motorsports shops are hesitant to allow us to disassemble a working ATV.

Therefore, the team decided to manufacture a case that would essentially act as a box housing for the transmission. In order to properly align the holes of the transmission iterations of laser cut acrylic was used to ensure that the final CNC milling of the case would be done correctly. Figure 32 and Figure 31 show the models that were used to laser cut the acrylic mock ups.
Oil seals, rubber housings, and sealed bearings were still used in order to seal fluid into the case to ensure that there would be no leaks from the transmission case. The final design would consist of bearings ordered directly from Yamaha to ensure they fit the required shafts and also standard bearings used where shafts were altered or had to be extended. Design considerations for the transmission case also had to be taken into account for the new drivetrain design which is described below in Drivetrain Design Process. The final SolidWorks assembly of the transmission case was then competed to include major shafts and gears to insurance properly tolerated fits and gear meshing.
Drivetrain Design Process

The drivetrain required the vehicle to power all four wheels in a way that would keep the maximum efficiency of engine output as well as ensuring that the driver was safe in the case of an unforeseen shaft failure or joint rupture. Therefore, multiple iterations had to be completed to ensure driver safety and efficiency throughout the vehicle while still maintaining the overall goal of a four wheel drive Baja vehicle.

Initially, the plan for the driveshaft was to run to either the left or right side of the driver. This would require multiple U-joints or CV joints to divert the axle around the seat while also accessing the transmission. This would require the main driveshaft to be driven by one of the two 90 degree output shafts of the T boxes. That shaft would then be diverted with multiple joints to the front T box resulting in a decrease in efficiency with each joint. The cost of the required joints and the extreme loss of efficiency would make this an infeasible idea for the group. A new method of power transfer had to then be determined that still utilized the two 90 degree T boxes and the Yamaha Big Bear transmission that were major components in the vehicle design.

Through discussion with the team and consolation with Professor Planchard and members of SAE the decision was made to use the transmission output shaft extended in both directions as the overall vehicle driveshaft. This meant the transmission had to be altered and other components of the vehicle had to be taken into account to ensure packaging of the vehicle worked while also maintaining safety and efficiency.

The engine mount required a 90 degree rotation while the seat had to be raised to ensure that the shaft could safely run under it. The transmission output shaft also had to be extended in either direction in line with the two T boxes to ensure that it was a straight shaft without
unnecessary stresses and efficiency losses. Figure 33 displays the new engine mount design so that the transmission can extend in either direction. The mount features ¼” 6061 aluminum that is bent to 90 degrees and then welded to a base plate that is U-bolted to the frame. The transmission sits directly below the engine.

![Engine Mount SolidWorks](image)

Figure 33: Engine Mount SolidWorks

The power from the T boxes will then be distributed to the axles through spacers that will have bushing on the insides to ensure proper fits. The team purchased Husqvarna HUV axles because of their CV joint configuration, to minimize efficiency loss, and their length sizing which matches the length needed. The spacers will be welded on the axle connection side and through bolted on the T box side. This will ensure no shearing of connection points will occur and the vehicle will operate safely. A complete overview of the design can be found in the Drivetrain Results section of the report.
Wheel Adapter Design Process

The team inherited four sets of wheels from last year’s Mini Baja MQP team. The set of wheels included two different sized tires and rims, for the front and rear of the vehicle. The supplied rear wheels are DWT 6061 [.190] rims with AT489 [AT23 X 8-12] tires while the front wheels are DWT 6061 [.125] rims with AT489 [AT23 X 8-10] tires. The two different rim sets we inherited have different bolt patterns.

The team purchased Husqvarna HUV hubs to match the previously mentioned Husqvarna HUV axles. These hubs have a different bolt pattern than either set of DWT rims. In order to attach the wheels to the vehicle, hub adapters were designed to space the rims away from the hubs to avoid collision and also match the varied bolt patterns. Two sets of these adapters were designed to match both sets of rims.

The adapters were designed with quarter inch aluminum because of strength and availability. They were made to be circular to fit inside of the rims while avoiding collision and have holes cut in the center to allow the axle nut to be accessible. They also have holes cut to match the bolt pattern of the rims and hubs. The holes cut to line up with the rims were countersunk on the backside to prevent bolt collision with the brake caliper. Figure 34 shows the SolidWorks design of the front and rear hub adapters.
Figure 34: Front (left) and Rear (right) Hub Adapters SolidWorks
Footwell Design Process

The footwell was designed to package all of the necessary components, including the throttle pedal, brake pedal, T-box and front axles, and steering assembly, while maintaining maximum leg and foot mobility for the driver. The first step in the design of the footwell was to design a base, or footplate. This plate was dimensioned to overhang the frame on the left and right side to allow it to be fastened to the frame with U-bolts. The front of the plate was tapered to allow clearance with the front pipes of the frame.

The team decided to eliminate the clutch, thus the clutch pedal, leaving the brake and throttle pedal to fit in the footwell. The two pedals were centered on the front of the plate; throttle on the right and brake on the left with the brake cylinders protruding out of the front of the vehicle. By positioning the pedals with the brake cylinders protruding out, the limited foot

Next the T-box was placed in line with the axles and centered on the footplate. A slot was cut into the footplate to allow the bottom mounting tab of the T-box to sit inside and be bolted on below the plate. A mounting bracket was designed to bolt onto the footplate and top mounting tab of the T-box, with a three inch hole in it to allow the T-box to sit inside of it. As described in the previous Steering Design Process section of the paper, the steering rack mount was designed to mount directly over the T-box. This mounting technique was necessary due to the tight space encountered in the footwell. Figure 35 displays a SolidWorks depiction of the footwell assembly.
Figure 35: Footwell Assembl SolidWorks
Brake Design Process

As previously stated in the Background Research section of the paper, caliper brakes were selected as the optimal design choice for the vehicle. Based on the purchase of the Husqvarna HUV hubs, Husqvarna HUV brake calipers were decided on due to their direct mounting to the hubs.

A brake pedal and its associated cylinders were provided to the team by the WPI SAE chapter. The brake pedal and cylinders were mounted offset left from the center of the footwell and drove the design of the brake system. The farthest left cylinder was designed to control the front brake calipers while the right cylinder controls the rear calipers. A “T-split” is utilized on both cylinders to split the line to send to each caliper. No SolidWorks model of the brake line was created.
Results

Suspension Results

The manufacturing of the control arms certainly did not go as planned. However the control arms were still successfully implemented into the vehicle. The first issue experienced with the manufacturing of the control arms was acquiring the necessary material. Originally the plan was to create all of the arms from left over stock of 4130-steel left behind from last year’s project. When discussing design with the head of engineering at Worcester Manufacturing, Jamie Gilman, he critiqued the ability of bending the 4130 steel that was planned to be bent for both the rear and front control arms. After multiple meetings with Jamie a redesign was had and the front arms were mitered at the locations of the bends because the tight radius on the front arms would not have been successful and crinkling of the inner radius would have occurred. Along with this decision Worcester Manufacturing reviewed the possible use of 1010 14 gauge steel for the rear arms. An analysis of 1010 14 gauge steel was done to ensure the material would provide sufficient strength in the suspension before submitting the drawings of the redesign to Worcester Manufacturing. Figure 36 below shows the jig made to weld the butt joints of the front control arms.

Figure 36: Front arm without welds
Worcester Manufacturing donated the material and labor in the manufacturing of the control arms. The next objective was to find a company or an individual that could weld the arms together, fish mouth the control arms, mount the pivot tubes and weld tabs onto specific locations per SolidWorks drawings. This process was in parallel with the manufacturing of the vehicle tabs. McNamara Fabricating Co. donated their time and material to manufacture all 52 vehicle tabs. The original plan to get the welding done was to go through Howard Manufacturing and the welding was to be done over spring break. Unfortunately due to a breakdown in communication with Howard a set date to drop off components was never made and we had to seek out alternatives. The first alternative that was suggested by Pat Guida, who has been one of our contacts throughout the project suggested to the team about utilizing the services of City Weld who are right down the street in Worcester.

Pat sat down and gave us a design review of the front and rear control arms before the final control arms were sent out to be welded. He pointed out the weakness of the butt welds that would be needed for the front arms and expressed his concern about what effect this could have on the vehicle. Pat suggested the use of gussets or large plate that could be used to add fillet welds to the arm and significantly increase its strength. One other critique Pat suggested was the manufacturability of the arms he felt that some of the design choices specifically the ball joint blocks made the arms more complex and expensive. In order to address these concerns adjustments to the design of the front arms were created. The changes involved adding a large plate to the front arms that would be welded to the bottom of the arms and a small pipe that would be milled out to press fit the ball joint into. The small pipe would then be welded to plate replacing the old ball joint block.
The ball joint block for the back was also eliminated. Figure 37 below displays the changes that were made to the arms after Pat’s suggestions.

Once the changes for the top and bottom arms were finalized, the next was to make sure that City Weld would be able to weld and fish mouth the arms in the three to four day timeline that we really needed. Luckily when we spoke with City Weld they were more than willing to work with us and could get started as soon as we could send City Weld drawings as well as some more details about what we needed to be done since we had already made jigs for the arms.

After sending them the drawings they gave us an estimate on cost which was to not exceed $3,000 dollars and they said the work should not take more than a week to get done. Following the agreement with City Weld, the next step was to manufacture the plates so that they could be welded. Fortunately the plates for the arms were made in a timely manner and were made through the Nypro tool room as a donation to WPI. The components were dropped off and were all welded in about four days coming to a total cost of $2,680. Figure 38 shown below shows the arms after they were welded.
Once the arms and frame were received the next step was to manufacture the pipes for the ball joints as well as all of the bushings for the control arms. These were all made on a lathe over
the course of a week in one of Nypro’s machine shop. In addition to that work the ball joints were also press fitted into the pipes through the use of a two-ton press as well as using a tig welder to weld the pipes onto the arms. Upon completing the bushings and pipes the control arms were complete and were placed on the vehicle. Figure 38 above shows how the arms fit within the vehicle.

Through multiple iterations of the control arms the final control arms were manufactured successfully. The only issues that were experienced throughout the process were design setbacks via design reviews by industry experts and the lead time of professional manufactures. As mentioned previously the delay in manufacturing was largely due to a breakdown in communication with Howard manufacturing and then finding a replacement within the tight time frame.
Steering Results

In the steering design process it was mentioned that a previously used steering rack was found within the SAE shop. Due to it still being fully functional and most importantly free the steering was designed around this rack. The first step in the realization of the vehicles steering was to bend the 0.125” aluminum bracket to which the steering rack would attach too. The plate was cut to the required shape and then bent manually to a 45-degree angle. The bracket is displayed below in Figure 40.

![Figure 40: Steering Rack](image)

After the steering bracket and steering rack were mounted the next step was to manufacture both the tie rods as well as the steering wheel mount. For the tie rods the steel stock was bought and cut to length by the use of a lathe. After the tie rods were cut the nuts were then welded to each end in order to properly house the ball joints. The steering wheel mount was initiated by cutting the steering rod to a much shorter length. The next step was too mill a collar that would attach the steering wheel to the steering rod. The steering wheel was then mounted to the rod. The last part of the process was the rear tie rods which were cut to length welded and then bolted directly to the rear control arms. Both the front and rear tire rods are pictured below, Figure 41.
Overall the steering was fully implemented as planned and was done on time as well. One issue that arose was that the steering wheel was not fixed to be centered. In order to be properly done this will have to be completed by next year’s team. The last issue found was that the aluminum bracket ended up being too weak and will need to be replaced in order for competition.

Figure 41: Front and Rear Tie Rods
In retrospect, the steering went pretty much as planned and functions exactly as we wanted. One
other additional issue that will need a redesign of the foot well is moving the rack forward to
minimize bump steer that is experienced by the vehicle.
Transmission Results

As discussed in the drivetrain methodology the decision was made to extend the transmission out in either direction to be used as the main driveshaft. This meant that the case must be an open system in order to extend the driveshaft while also being sealed so that no SAE rules on leaks are broken throughout the competition. The output shaft was extended by welding two separate shafts to either end. The shaft extending to the rear T box is ¾” OD solid low-carbon steel shaft that was lathed to the outer diameter and depth to create a press fit to further secure the weld. The shaft extending to the front is a 1” OD, .825” ID low carbon steel shaft that arrived pre-sized for a tight fit on the shaft. Low carbon steel was selected for its high shear stress capabilities and also for its ease to weld to the already low carbon and hardened steel gear shaft. Figure 42 displays the output shaft extended in either direction out of the transmission case which is fastened to the engine mount.

The inner components of the Big Bear transmission that were bought did not include an input shaft but did include the input gear. Therefore, a shaft was bought and lathed down to the ID of the input gear and then welded on to deliver power into the system. This shaft was also
made of low carbon steel with a 1” OD and .5” ID given a very large wall thickness to dissipate the torque put on it from the engine. This shaft was selected with a safety factor of 1.5 to ensure that no failure would occur in this critical part.

After the input and output components of the transmission were determined bearings were then selected for the transmission case. The gear reduction shaft was unchanged on either end and did not have standard shafts OD’s so the bearings out of the Yamaha transmission case were ordered for either end. The shift collar utilized a Yamaha bearing on one end and then a standard 1.5” ID bearing was used in order to fit the shaft that was used to extend the collar the length of the shaft. This was done so that there would not have to be more than four walls to enclose the transmission case in order to avoid more weak points in the geometry. Standard steel ball bearings rated to a 1,384 dynamic stress capability were chosen for either end of the output shaft. The input shaft utilized the same bearings as the output shaft because of the high load capabilities and ease of purchase/ budget considerations. Each of the standard steel ball bearings were purchased for $12.04 and met the required capabilities with a safety factor of 1.3. Figure 43 displays the bearings used within the transmission case in their proper layout within the milled end walls.
The next and most crucial step was milling the end walls of the transmission case. The hole placement is critical to assure the gears properly mesh and there is no excess stress on the gears, shafts, and bearings in the case. Laser cut pieces of acrylic were used to ensure that the CAD models were correct once cutting into metal. The initial plan was to cut the end walls out of 5/8” 6061 Al due to its easy machinability, welding, and strength qualities. However, the first iteration of the end wall cuts produced a geometry where the output shaft was .030” to close to gear reduction shaft. This meant that case would not properly assemble and a second iteration had to be completed. The team believes that since only 1/8” acrylic was used it was deformed when being cut and handled which gave the improper dimensions. Therefore, for the second iteration the CAD models were fixed and re-milled in order for the transmission to properly function.
These models were then imported into the CAM software Esprit to ensure nominal dimensioning when milling these press fit holes. The bearings were all rated for a .003-.001” interference press fit so the holes were all dimensioned to fall within that range.

Once the end plates were milled correctly the bearings were pressed in using an arbor press. This allowed the shafts to be properly aligned and the shafts for the shifting forks pressed into their location. The last step was to machine four plates down to the overall outside dimension of the case to enclose the case and allow for fluid to be inserted. The case is assembled by first inserting the shafts into the end walls. The outer plates are then bolted into the end walls to create a tight fit and sealant is used to ensure no fluid is leaked. Figure 44 shows the case without these outer plates and extension shafts in order to show the purchased transmission sit within the system without alteration.

Figure 44: Transmission in Case without Extension Shafts and Outer Walls
**Drivetrain Results**

**Power Distribution**

The initial power output from the vehicle begins at the engine. A centrifugal clutch is mounted to the output shaft of the engine. The clutch is a Hilliard Extreme-Duty centrifugal clutch rated for an engagement speed of 2300 RPM’s and a max operating speed of 5000 RPM’s. The Hilliard clutch is equipped with a 17 tooth sprocket for a #35 chain and can be seen in Figure 45.

![Hilliard Heavy Duty Centrifugal Clutch](image)

**Figure 45: Hilliard Heavy Duty Centrifugal Clutch**

The #35 chain then runs from the centrifugal clutch to a 15 tooth sprocket on the transmission input shaft giving an initial gear reduction of 1.133. This ratio is then enlarged through the varying gear ratios found within the transmission. Figure 46 shows the chain engagement from the clutch to the transmission on the vehicle.
Figure 46: Engine to Transmission Chain Drive

Once the power is passed to the transmission it is then diverted in two directions to the front and rear mounted T-boxes through extension shafts. The transmission output shaft then mated with U-joints using 3/8” fasteners on either side to compensate for up and down movement when going over bumps and landing from jumps which can be seen in Figure 47.

Figure 47: Transmission Output to U-Joint
The U-joints then run a galvanized steel reinforced driveshaft to both T-boxes. The main galvanized steel driveshafts are 1” OD pipes rated for wind turbine shaft use, thus giving a safety factor well above what was needed to ensure driver safety. The cost associated with this increased safety factor was negated due to the shaft being donated by Industrial Communications for our use. These driveshafts are mated to the T-boxes and U-joints using the same 3/8” hardware as what was used for the transmission to U-joint connection.

Now that the power has been successfully transferred to the T-boxes it must then be diverted to either axle. This is done the same in both the front and rear of the vehicle with 4130 steel spacers being used to connect the axles, purchased off a Husqvarna 4120 HUV, and the T-box connections. The spacers interact with the T-boxes with 3/8” hardware while spacing bushings were created and welded to the splines on the axles. These bushings were then welded to the spacer itself and were made of a soft enough metal to roughly form into the shape of the spline that is found on the axles. An example of the U-joint to T-box and T-box to axle spacers can be found in Figure 48.

Figure 48: Rear T-Box to U-Joint Connection
The power is then distributed from the axles to the wheel hubs which are taken from the same HUV as the axles to ensure that the splines interact perfectly. This system thus allows a four wheel drive with optimized efficiency and safety for the driver.

Assembly

The drivetrain was designed and manufactured based upon the center driveshaft decision that was made to ensure that four wheel drive would work with the most efficiency in a way that was also safe for the driver. This meant that the first thing placed in the vehicle was the engine mount. The final manufactured engine mount was manufactured to the initial rotated design and can be seen in Figure 49.

Figure 49: Assembled Engine Mount
The T-boxes were then placed in their mounts and centered within the vehicle so the driveshaft runs straight along the centerline of the vehicle frame. Figure 51 and Figure 50 show the T-boxes in their mounts and bolted to the frame to maintain position of the driveshaft.

Once the T-boxes were centered the transmission and engine mounts were designed to provide linear adjustability to ensure a straight linear fit with the T-boxes. Once this linear adjustment is made the transmission extension shafts are connected to the U-joints. The U-joints then run the main drive shafts out of their other ends to the T-boxes which distribute power to the axles and thus the wheels which completes the assembly of the drivetrain.

Safety Considerations

Major safety issues are brought up when having a driveshaft run directly under the driver as it does from the transmission to the front T-box. In order to ensure that the driver is safe at all times safety factors were taken into account for the shafts. “Safe- failures” were built into the
design so that if something were to break it will break in a way that does not put the driver in immediate harm. For example, the safety factor for the main galvanized steel driveshaft was nearly 2 because it runs directly under the driver. If a shaft were to fail it would be the one connecting the transmission to the rear U-joint. This is the furthest shaft from the driver and would allot the driver plenty of time to brake if it were to fail. However, a safety factor of 1.2 was still used on this shaft so it should not break even under the vehicles operating conditions in rocky terrain that will put a lot of torque on the system.

Also, all rotating parts within the seat and footwell areas are covered with Lexan to protect the driver from getting entangled. This is found running from the engine mount to the front T-box and also across the top of the front T-box to axle spacers. Along with these precautions, the seat is raised 4” to allot for the required 6” head clearance from the top of the frame that SAE requires while also being furthest away from the rotating shaft. The underside of the seat is made from a steel plate as an extra aspect of precaution to protect the driver. Figure 52 displays the raised seat that is used to protect the driver. The Lexan covers were removed for this picture to show the driveshaft clearance from the seat.

Figure 52: Raised Seat Assembly and Driveshaft Clearance
Wheel Adapter Results

The wheel adapters were manufactured out of quarter inch aluminum. Two 9x9 inch and two 7x7 inch square plates were cut and a 2.5 inch hole was cut into the center of each. The plates were then milled on a turn table to be circles with radii of 9 and 7 inches each. The holes for both bolt patterns were then cut into the plates using the mill. To finish the adapter plates, one side of the rim bolt pattern holes were countersunk. Figure 53 displays one of the completed rear hub adapter plates.

After manufacturing the adapters, spacers were lathed out to space the rim away from the hub and prevent interference. These spacers were lathed out of steel pipe and were dimensioned to be 1.1 inches and 1.55 inches for the rear and front respectively. The final modification needed for the wheels was modifying the front rims. The bolt pattern of the front rim and hub

Figure 53: Rear Hub Adapter Plate
both fall within the wall of the rim, meaning that holes needed to be drilled in the rim to allow lug nut clearance. Figure 54 depicts the final wheel assembly for the front and rear tires.

Figure 54: Front Wheel Assembly and Rear Wheel Assembly
Footwell Results

The first step in assembling the footwell was to cut out an unnecessary pipe running over the top of the footwell space. Once this was removed, the footplate was cut to size out of quarter inch aluminum and mounted to the frame using U-bolts. Once in place, the pedals were mounted to the plate and the slot for the T-box was cut out. The T-box bracket was made out of 1/8 inch aluminum and was cut, drilled, and bent to size. It and the T-box were then mounted to the footplate. After this, the steering rack bracket was mounted directly over the T-box. Finally, and not in the initial design, slots were cut out on the sides of the plate for the axles to prevent interference when rotating. Figure 55 displays the completed footwell assembly.

Figure 55: Footwell Front-view and Top-view
Brake Results

The brake calipers were mounted onto the hubs using the provided hardware and fit perfectly because of the Husqvarna compatibility. Brake line was run from each cylinder to “T-splitters” where the brake lines then went to each caliper. The brake line was bent and formed using a brake line bending tool and the ends were flared using a brake line flaring tool. Flexible brake line was used for the connection to the calipers to allow movement with the suspension. The brake cylinders were filled with brake fluid and the brakes were then bled. Figure 56 displays the connected front and rear brake calipers.
**Throttle Results**

The WPI SAE chapter provided the team with a throttle pedal that had a throttle cable already attached. This cable was utilized and mounted to run from the pedal in the front to the rear of the car. An attachment was made to connect the throttle cable to the throttle on the engine. A return spring was added because of the simplicity of the provided cable. Figure 57 displays the throttle cable to engine connection.

![Figure 57: Throttle Cable Connection](image)
Knowledge Acquired

Feasibility

The design process became a real test of imaginative and ambitious ideas that were turned into practical drawings that then could be implemented correctly. We experienced designs that proved to have a relative degree of being easily done and others that proved to be very difficult and time consuming. This lesson learnt was most evident when we came to the mammoth task of designing a transmission case around the gears itself of a Yamaha Big Bear 400cc transmission. We spent many man-hours trying to track down a previous transmission case that would provide a design once reversed engineered. Also many hours were spent researching and communicating with experts on campus of the feasibility of sand casting either at Worcester Polytechnic Institute or the Palmer Foundry via investment casting or sand casting.

Although an educationally exciting opportunity to reverse engineer and aluminum cast our transmission case, the feasibility of such a task was unrealistic within the time frame of our project and the current location our project was within our Gantt chart. We were ambitious at first and it had been a valuable lesson learnt, that something may be the best approach fundamentally, however there are other constraints such as time that make that specific approach impractical.

Parallel Plans

Throughout the research and purchasing of specific systems within the Baja SAE vehicle many original plans had to be scrapped due to change in price or change in the general design requiring other changes in other systems. It became evident the need to have parallel plans
alongside “plan 1a” there had to be a feasible “plan 1b” that would of accomplished the same design goals and objectives in a slightly different way.

The design of the front and rear control arms experienced the use of a parallel design plan. The preliminary design was based off the preconceived size of the shocks that we were receiving. However the size of the shocks hindered the ability for the axle, control arms and wheel hubs to line up correctly. This however although an issue was quickly addressed as a parallel design, shifting over the shocks and changing the mounting points, already existed in the case this event had happened.

**Design Review Sessions**

On multiple occasions throughout the semester the use of a design review proved to be a valuable resource that allowed us to address fundamental problems within our design in a more direct and efficient manner. We held a design review session with the Worcester Polytechnic Institute chapter of Society of Automotive Engineers. We presented our design that had a fundamental problem with diverting power from the rear T-box to the front drive axles of the vehicle. Through discussion with the SAE organization it was decided to move the center driveshaft in line with the transmission output shaft along the centerline of the vehicle. This allows the transmission output shaft to be extended in both directions to the front and rear T-boxes that then split the power to the left and right drive axles. Although a simple fix this only came about because of the new perspective provided by the WPI SAE chapter members.
Future MQP Recommendations

Vehicle Improvements

- Establish proper spline connection between the axles and T-boxes.
- Purchase and install proper front rims and adaptors.
- Reinforce steering mount bracket.
- Add structural support to rear frame shock tabs.
- Design and install body panels for vehicle.
- Include reverse gear in transmission.
- Optimize differentials to increase steering capability.

Testing

A series of tests both static and dynamic need to be undertaken. All systems of the vehicle should be tested to ensure safety and reliability before the vehicle can be handed over to the WPI SAE chapter to be used in future Baja SAE competitions starting in 2015-2016 academic year. Other tests to establish the specific capability of the vehicle to the set requirements specified by SAE international for endurance, suspension and traction and hill climb are all feasible tests. In order to do this effectively a detailed testing protocol will need to be established abiding by SAE international rules and regulations. This will ensure that the static and dynamic tests are organized and efficiently conducted.
Appendix A: SAE Rules and Regulations

ARTICLE 11: BRAKING SYSTEM

B11.1 FOOT BRAKE
The vehicle must have hydraulic braking system that acts on all wheels and is operated by a single foot pedal. The pedal must directly actuate the master cylinder through a rigid link (i.e., cables are not allowed). The brake system must be capable of locking ALL FOUR wheels, both in a static condition as well as from speed on pavement AND on unpaved surfaces.

B11.2 INDEPENDENT BRAKE CIRCUITS
The braking system must be segregated into at least two (2) independent hydraulic circuits such that in case of a leak or failure at any point in the system, effective braking power shall be maintained on at least two wheels. Each hydraulic circuit must have its own fluid reserve either through separate reservoirs or by the use of a dammed, OEM-style reservoir.

Note: Plastic brake lines are not allowed

B11.3 BRAKE(S) LOCATION
The brake(s) on the driven axle must operate through the final drive. Inboard braking through universal joints is permitted. Braking on a jackshaft through an intermediate reduction stage is prohibited

B11.4 CUTTING BRAKES
Hand or feet operated “cutting brakes” are permitted provided the section (B11.1) on “foot brakes” is also satisfied. A primary brake must be able to lock all four wheels with a single foot. If using two separate pedals to lock 2 wheels apiece; the pedals must be close enough to use one foot to lock all four wheels. No brake, including cutting brakes, may operate without lighting the brake light.
## Appendix B: Polaris Shocks and Springs Price Comparison

<table>
<thead>
<tr>
<th>Shocks and Springs Description</th>
<th>University Discount Price</th>
<th>Retail Price</th>
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<tbody>
<tr>
<td><strong>Front Suspension:</strong></td>
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<td></td>
</tr>
<tr>
<td>7043761 Shock – SACHS 16.3&quot; (Ext) x 5.3&quot; (Stroke) - 2.0&quot; Body</td>
<td>$68.00</td>
<td>$112.99</td>
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<tr>
<td>7043809-458 Spring 115#/IN, 11.75&quot; Length, 2.06&quot; ID, Black</td>
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<tr>
<td>525376 Cam, Adjusting Spring</td>
<td>$2.00</td>
<td>$4.66</td>
</tr>
<tr>
<td>5251753 Retainer Spring</td>
<td>$3.00</td>
<td>$5.99</td>
</tr>
<tr>
<td><strong>Rear Suspension:</strong></td>
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<td></td>
</tr>
<tr>
<td>7043759 Shock – SACHS 19.0&quot; (Ext) x 6.7&quot; (Stroke) - 2.0&quot; Body</td>
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<td>7043760-458 Spring, 185#/IN, 12.25&quot; Length, 2.06&quot; ID, Red</td>
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<td>5253768 Cam, Adjusting, Spring</td>
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<tr>
<td>5251753 Retainer, Spring</td>
<td>$3.00</td>
<td>$5.99</td>
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<tr>
<td><strong>Total:</strong></td>
<td><strong>$181.00*2 = $362.00</strong></td>
<td><strong>$347.26*2 = $694.52</strong></td>
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Appendix C: Fox Springs and Shocks Price and Size Chart

<table>
<thead>
<tr>
<th>Description and Size:</th>
<th>Part Number:</th>
<th>Price:</th>
</tr>
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<tbody>
<tr>
<td>FLOAT 13.0” X 2.8”</td>
<td>830-10-200</td>
<td>$521.25 (SET)</td>
</tr>
<tr>
<td>FLOAT 16.2” X 4.5”</td>
<td>830-10-201</td>
<td>$521.25 (SET)</td>
</tr>
<tr>
<td>FLOAT 19.8” X 6.2”</td>
<td>830-52-200</td>
<td>$521.25 (SET)</td>
</tr>
</tbody>
</table>
Appendix D: Suitable Spring and Shock Calculations

Front & Rear Loads:

\[(\text{Total Weight} - 600 \text{ lbs}) \times (\text{Weight Percentage} - 0.60) = 360 \text{ lbs} - \text{Rear load}\]

\[(\text{Total Weight} - 600 \text{ lbs}) \times (\text{Weight Percentage} - 0.40) = 240 \text{ lbs} - \text{Front load}\]

Individual Spring Load:

\[(\text{Rear Load} - 360 \text{ lbs}) / (\text{Number of Springs} - 2) = 180 \text{ lbs Per Spring}\]

\[(\text{Front Load} - 240 \text{ lbs}) / (\text{Number of Springs} - 2) = 120 \text{ lbs Per Spring}\]
Works Cited


George, B., & Basava, V. B. (2013). Identification and Optimization of Diagonal Shift Failure in


Analysis of heat conduction in a disk brake system - Faramarz Talati and Salman Jalalifar - Received: 3 July 2008 / Accepted: 5 January 2009 / Published online: 27 January 2009 Springer-Verlag 2009