

2012-2013 WPI SAE Baja Vehicle

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

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Date: April 22, 2013

Abstract

The Baja Society of Automotive Engineers (SAE) competition is held annually in order to provide engineering students an opportunity to design and build a competitive off-road vehicle. This Major Qualifying Project (MQP) focused primarily on a major redesign of a previous WPI Baja SAE car (2008-2009 Design and Fabrication of a SAE Baja Race Vehicle), by determining its strengths and weaknesses through design reviews and field-testing. The MQP improved the following subsystems, identified as the weakest components of the car: drivetrain, steering, brakes, and front suspension.

The team removed the existing hybrid-hydraulic drivetrain and designed, fabricated and tested a new mechanical drivetrain. The steering geometry was designed according to the Ackerman principle and to balance the effects of caster and camber. The required braking force for the car was calculated, and a new front and rear braking system was installed with the ability to lock all four wheels at speed, as stated in the SAE rulebook.

The front suspension was addressed to provide proper ground clearance in accordance with the SAE guidelines and to maximize suspension travel. New components and sub-systems were designed using SolidWorks. SolidWorks Simulation was used to perform finite element analysis to optimize each component and subsystem to determine the necessary strength while minimizing weight. Material selection was based on design factors including weight, cost, and performance.

All work was performed in accordance with the SAE Baja guidelines to maintain the car's eligibility for competition.

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Introduction

The vehicle used for this MQP was originally built for the project titled “2008-2009 Design and Fabrication of a SAE Baja Race Vehicle.” The stated goal for the build of the car was the endurance race in the 2009 SAE Baja competition. The rigors of this race dictated all design decisions of the vehicle, which is important to understand when studying the vehicle. The suspension and ground clearance of the vehicle were designed to overcome the obstacles presented in this race; the drivetrain was designed to run at the speed ranges required to be competitive in the race; the chassis was designed with ergonomics in mind to reduce driver fatigue while competing in this long race, and the brakes were designed to provide the required braking forces and survive the punishments of frequent and forceful braking during the race. A top priority in design and material selection was always weight. The limiting factor of the Baja car is the low powered engine, which all teams are required to use; therefore minimizing weight gives the car a competitive advantage. Many of the design decisions were based on knowledge gained from studying the 2007 WPI Baja MQP car. Calculations for expected forces on components of the car such as the front suspension were often taken from data based on the 2007 car.

Another MQP was completed on the vehicle for the 2009-2010 school year, titled “Hydraulic Series Hybrid Baja Car.” The goal of this project was to install a hydraulic drive system in which the motor was powered by a pump, allowing the motor to always run at peak horsepower and store excess power in a hydraulic accumulator for use when needed. This complicated system never worked as designed due to a series of problems outlined in the “Results” section of said MQP report. Recommendations from that report included doubling the

size of the accumulator; an impractical solution considering the current system leaves no space for expansion in the rear of the car frame and already adds 150 pounds to the car. The decision was made to remove this drive system from the car and revert back to a mechanical drivetrain because the focus of this MQP is the redesign of the front suspension, steering, and braking systems.

Goals and Objectives

1. Ultimate goal: 2009 Baja car will be fully operational and ready for competition when this MQP is completed. All work will be completed in compliance with the Baja SAE Collegiate Design Series Rules. Perform whatever miscellaneous tasks are necessary to satisfy this goal.
2. Reinstall a mechanical drivetrain with a CVT and chain reduction and tune the CVT to maximize performance.
3. Design and install a new front suspension that has better travel and is lighter, yet just as strong as the current design.
4. Design and install a new steering system that properly balances the effects of caster and camber to improve the handling of the vehicle in an off road environment. Maintain the original design requirements set for the vehicle: steering wheel rotation limited to 180 degrees in each direction with maximum steering angle of 30 degrees.
5. Design and install a braking system with “at least two (2) independent hydraulic circuits... capable of locking ALL FOUR wheels, both in a static condition as well as

from speed on pavement AND on unpaved surfaces,” as stated in the Baja SAE Collegiate Design Series Rules.

Product Specifications

1. Vehicle must be capable of carrying one person 75 in. tall, weighing 250 lbs.
2. Vehicle must be safe for a 95th percentile male operator.
3. Width of the vehicle must not exceed 162 in.
4. The vehicle must be capable of safe operation over rough land terrain including, but not limited to, obstructions such as rocks, sand jumps, logs, steep inclines, mud and shallow water in any or all combinations and in any type of weather including rain, snow and ice.
5. No components of the vehicle must come loose during a rollover.
6. All wiring must be sealed, protected and securely attached.
7. Vehicle must contain front and rear hitch point along the longitudinal centerline.
8. There must be a firewall between the cockpit and the engine and fuel tank compartment. It must cover the area between the lower and upper lateral cross members on the Rear Roll Hoop.
9. The vehicle must have a 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.
10. The brake system must be capable of locking all four wheels, both in a static condition as well as from speed on paved and unpaved surfaces.
11. Vehicle must be capable of completing a four hour endurance test.

12. Vehicle must complete an acceleration event, measured as the time to complete a 100-150 ft. straight course.
13. Vehicle must be capable of climbing an incline from a standing start.
14. Vehicle must have a static negative camber of less than 2° , with a dynamic camber gain of less than 5° .
15. Vehicle must be safe for a 5th percentile female.
16. A safety harness system of at least 5 points must be worn by all drivers. The lap belt and shoulder belts must be approximately 3 in. wide. The fifth (“anti-submarine”) belt must be worn between the legs to prevent the lap belt from riding up along the driver’s torso.

Drivetrain

The drivetrain for a Baja car needs to be strong and reliable enough to survive the endurance completion, while being light and fitting into the given space.

Engine

All vehicles completing in the Society of Automotive Engineers’ (SAE) Mini Baja competition must use the same engine: the Briggs and Stratton OHV Intek model 20. This single cylinder, four cycle, air-cooled, 52 pound engine is rated for 10 HP at 3800 rpm. SAE uses this engine to level the playing field between teams. To be competitive, the car needs to be designed to maximize the output available from this engine. The power curve for this engine, provided by Briggs and Stratton, is shown below in Figure 1.

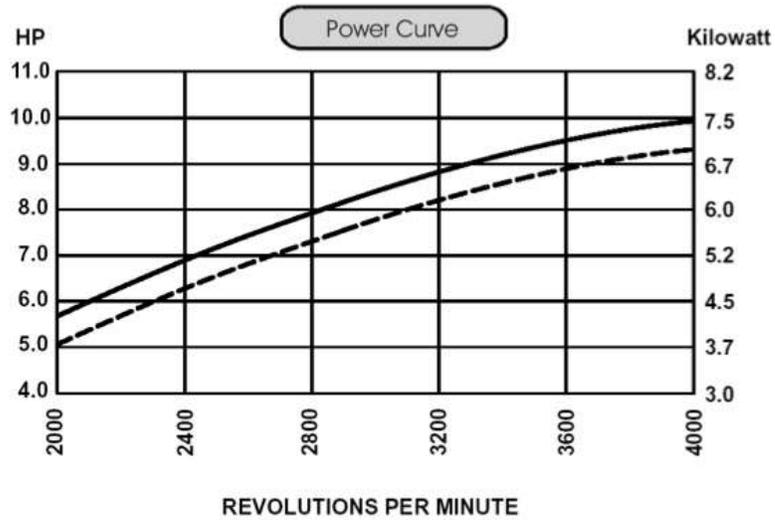


Figure 1: Briggs and Stratton Power Curve

The 2008 Baja MQP group took the actual engine and put it on a dynamometer to get real world data. The engine has a governor that limits the power at high rpm to protect the engine. The process is described in their report, “2007-08 WPI SAE Baja Vehicle.” Their findings are shown in Figure 2, which is the actual power curve of the engine.

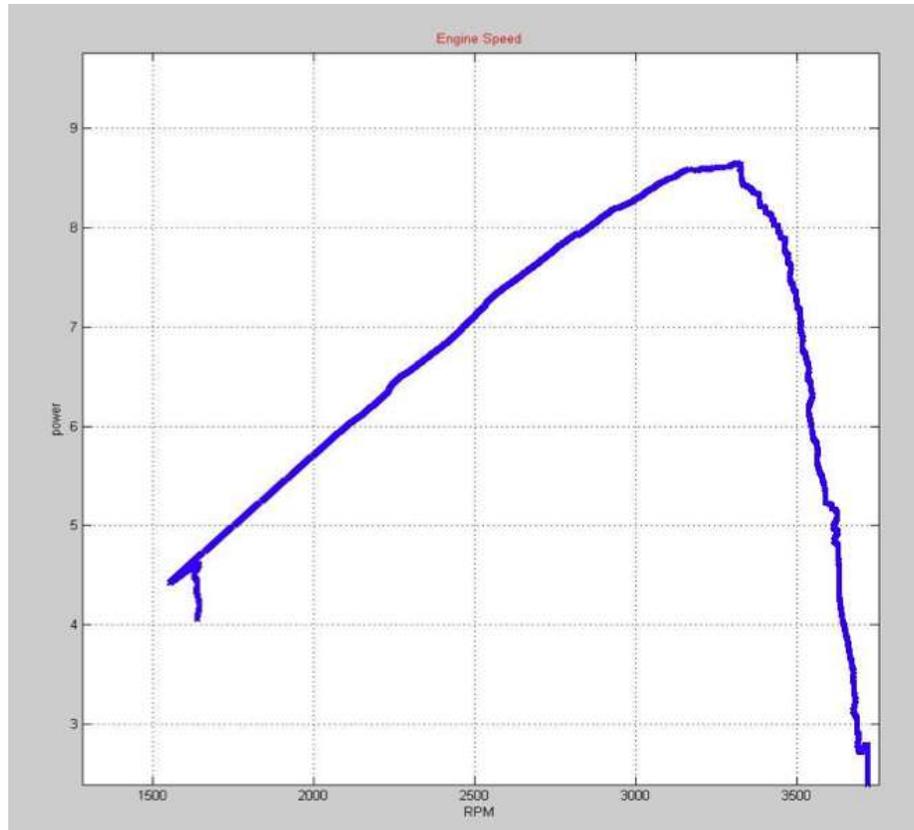


Figure 2: Real World Briggs and Stratton Power Curve

From this graph it can be seen that the maximum horsepower of the engine is actually around 8.8 horsepower and occurs at 3400 rpm. This is important information to know so that the rest of the drivetrain can be tuned to this optimum engine speed.

Original 2009 Drivetrain

The original drivetrain configuration and gear reduction were based on the average speeds expected in the endurance race and the rpm at which the engine produces peak horsepower. These calculations are taken from knowledge gained from the 2007 car and provided useful data on the engine. It was determined that all of this was completed correctly and implemented successfully on the vehicle. Therefore, the new drivetrain for this MQP will have the same gear ratio as the original. The 2009 MQP report gives descriptions of each component

of the double chain reduction drivetrain and provides information on why certain design decisions were made. It also explains which parts were purchased and what was manufactured. This MQP report also explains the process used to fabricate the drivetrain. The large side plates were water jet to relieve a majority of the material and then the assembly was welded together. The entire assembly was then sent out for tempering and stress relieving to bring the steel back to T6. Final machining was then done on the bearing holes and tensioner slots before assembly. Figure 3 below shows the CAD model used to design the original drivetrain and Figure 4 shows how the drivetrain sub-frame fit into the back of the vehicle frame.

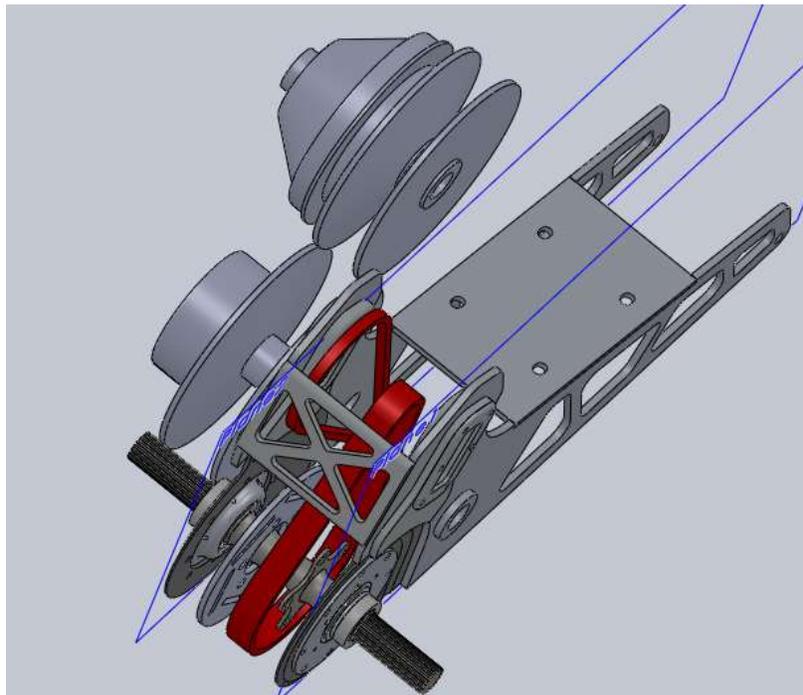


Figure 3: Original Drivetrain CAD

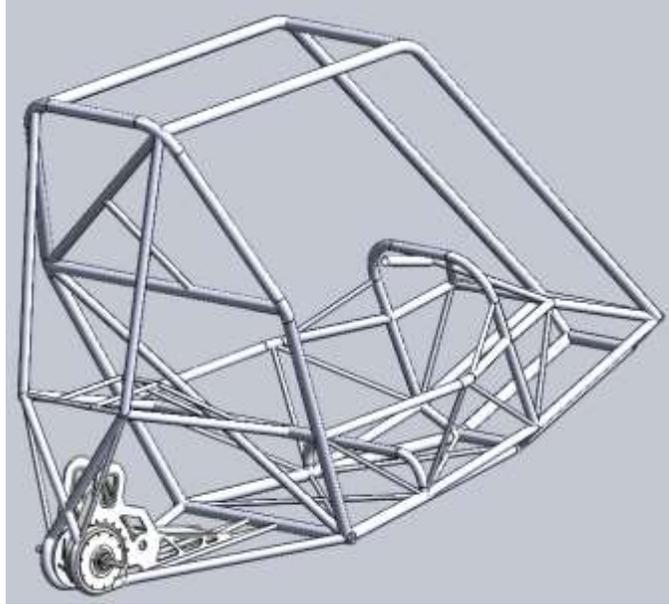


Figure 4: Original Drivetrain in Chassis

Drivetrain Design Goals

The first design goal set to improve upon the previous design is to lower the engine in order to lower the overall center of gravity of the vehicle, which will improve handling and reduce the chance of rollover. This will be accomplished by mounting the engine flatly on the bottom frame rails, rather than on an inclined plate as in the 2009 drivetrain sub frame shown in Figure 3. The second design goal is to reduce the weight of the whole system because weight reduction is an underlying design goal for this MQP's vehicle plan. Mounting the engine flatly will help with this goal as well because it will reduce the amount of aluminum plate needed for the gear frame. The third goal is to simplify the manufacturing process. The process to manufacture the original drivetrain involved water jet cutting, welding, tempering, and final machining. The goal is to eliminate the need for welding and tempering to simplify and reduce the cost to manufacture the drivetrain. The new design will only require water jet cutting and in-house machining. Finally, the last design goal is to have the drivetrain perform as well or better

than the original version in terms of drivability and reliability. The new CVTech transmission will accomplish this goal by providing more efficient power transfer and finer speed control.

New Drivetrain Design

Upon inspection of the vehicle, it was determined that the CAD models for the frame from when the car was built do not reflect what was actually built. The first step to design the new drivetrain was to model the frame members that are relevant to the drivetrain in SolidWorks. The plates were then designed to mount the engine and the drive shafts, while keeping the following design goals in mind: to keep the center of gravity of the vehicle as low as possible, to reduce the weight of the whole system, and to simplify the manufacturing process.

The original drivetrain had the engine mounted up high and on an angle to give more room for the secondary drive shafts and chains. With the engine mounted flatly on the bottom frame rails, packaging becomes more complicated due to the inherent space restrictions. SolidWorks part models were downloaded from McMaster-Carr's website for the standard parts. The simplicity of the new design only requires the following parts to be machined: 2 identical upright plates and the engine mount plate. These parts are made from aluminum plate and require only simple shapes and holes.

This design has the center of the engine block located 3 inches to the left of center. This allows the intermediate sprocket and shaft to be moved down and back toward the engine, keeping it lower and more compact. The drawbacks of this design are the overhung portion of the intermediate shaft on which the CVT is mounted, tight clearances, and the fact that there is more material on the engine mount plate. The benefits include minimal material on the upright plates, a stiff mount for the intermediate shaft, a lower center of gravity, a compact package, and

short chains. The plates are also designed to be mounted to the frame using only U-bolts around frame members and existing mounting points to avoid any modifications to the frame. Figures 5, 6 and 7 below show the top view, side view, and isometric view with the engine, respectively.

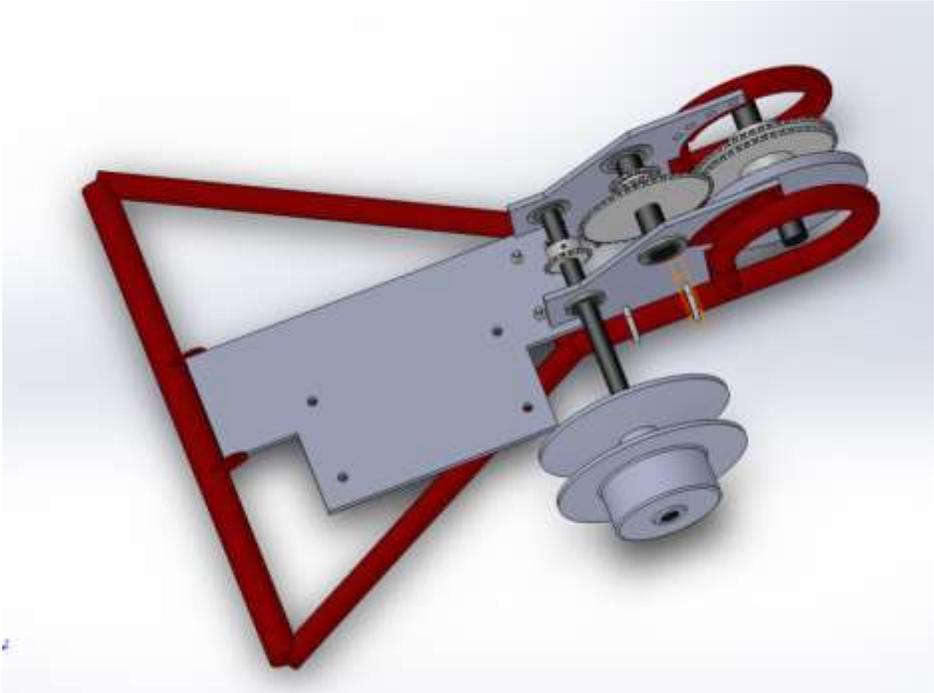


Figure 5: New Drivetrain CAD Top View

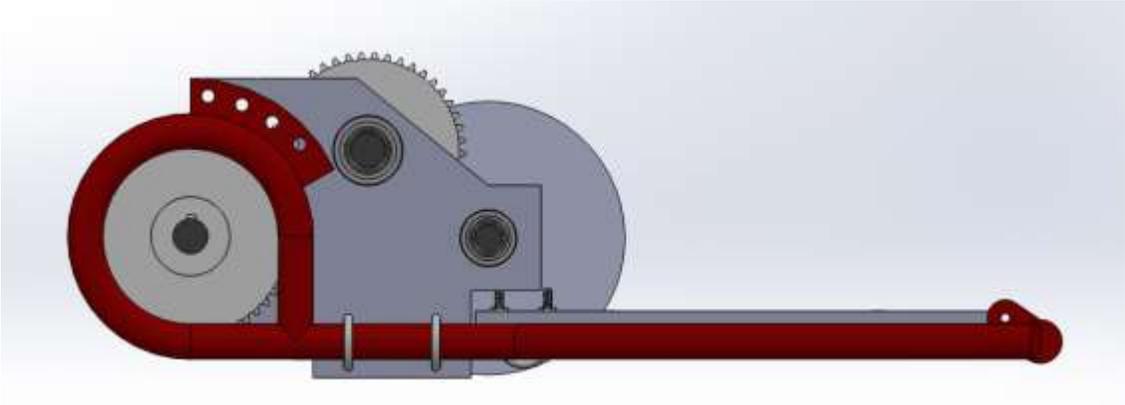


Figure 6: New Drivetrain CAD Side View

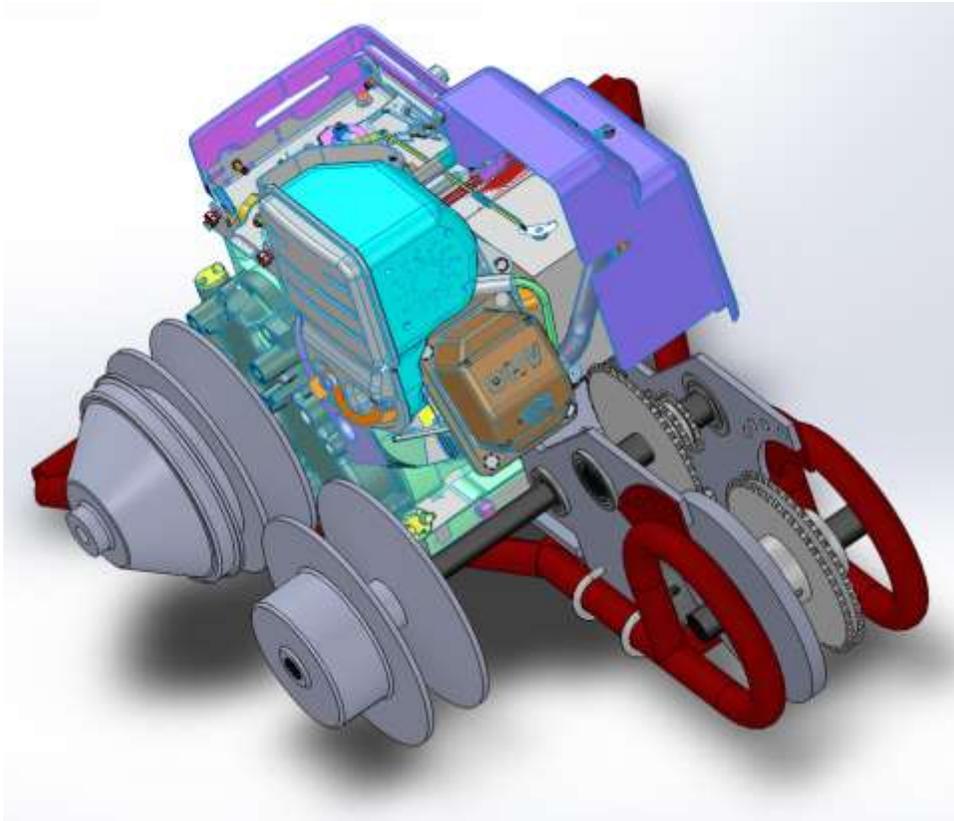


Figure 7: New Drivetrain CAD with Engine

Individual components were then analyzed to verify that the drivetrain would be strong enough to withstand the forces it will experience during competition. The area of greatest concern is the overhung portion of the CVT shaft that can be seen in black in Figure 7, between the driven CVT pulley and the bearing plate. The distance from the edge of the bearing to the edge of the CVT pulley is 6.19". Professor Norton's book, Machine Design, was consulted to find the deflection in the shaft. Chapter 10 provides the equations for power and the angular deflection of the shaft, and chapter 4 provides the equations for the linear deflection. The equations were entered into TK Solver, which allows the input parameters to easily be changed,

so that the effect on the other variables can instantly be seen. This code is shown in Figure 8 below.

Status	Input	Name	Output	Unit	Comment
	10	Hp		Hp	Horse Power
		P	330000	ft*lb/min	Power
		T	16.1603617	in-lbs	Torque
		w	20420.335	rad/min	angular velocity
	3250	rpm		rpm	engine rpm
		theta	.00033386	degrees	angular deflection
	7.38	l		in	shaft length
	11500000	G		psi	shear modulus
		J	.031063085	in^4	polar moment of inertia
	29000000	E		psi	elastic modulus
		Ymax	-.000172448	in	max shaft deflection
	194	F		lb	force on shaft
	6.19	a		in	overhung length
		Iner	.015531542	in^4	Inertia of shaft
	.75	d		in	shaft diameter
	3.14159	Pi			Pi
		r	.375		shaft radius

Status	Rule
Satisf	P=T*w
Satisf	w=rpm*2*Pi
Satisf	P=Hp*33000
Satisf	theta=(T*l)/(G*J)
Satisf	J=(Pi*(d^4))/32
Satisf	Ymax=((F*(a^2))/(6*E*Iner))/(a-3*l)
Satisf	Iner=(Pi*(r^4))/4
Satisf	r=d/2

Figure 8: CVT Shaft Deflection TK Solver Code

From these calculations, the maximum angular deflection and the maximum linear deflection are both negligible, confirming that the overhung $\frac{3}{4}$ " diameter shaft will be sufficient. The minimal deflections can be attributed to the low power of the engine.

Drivetrain Manufacturing

Engine Mount Plate

The engine mount plate was manufactured in-house in Washburn shops. The $\frac{3}{8}$ " thick 6061-T6 aluminum plate was bought from McMaster-Carr with the required 8" width and with a length of 36". The only machining that was necessary was to cut the plate in half on the band saw to a length of 18" and drill 12 holes on the drill press. No other machining was required thanks to the simple design approach. The plate mounts to the bottom frame rails with 4 U-bolts and the engine is then mounted to the plate with 4 bolts.



Figure 9: Mocking up the Engine Mount Plate



Figure 10: Engine Mounted

Bearing Plates

The bearing plates required fairly complex geometry, which included an arc, which mounts to the frame and allows the plates to fit within the design envelope. Vangy Tool



Figure 12: Bearing Plate Final Machining

The bearing holes were undersized by 5 ten thousands of an inch for each respective bearing size and a pattern of 4 holes was drilled and tapped around each to help retain the bearings. The bearings were then pressed in on the arbor press. A finished bearing plate can be seen in Figure 13 below.



Figure 13: Finished Bearing Plate

Sprockets

The #35 chain, single wide, 15-tooth sprocket used on the CVT shaft came from McMaster-Carr with the bore and keyway already machined and ready to mount. The #35 chain, single wide, 45-tooth aluminum sprocket had to be mounted to a steel hub in order to mount it to the intermediate shaft. The #35 chain, double wide, 16-tooth sprocket had a plain bore that was opened up to a 1” bore and manually keyed to mount it to the intermediate shaft.

CVT (Continuously Variable Transmission)

The 2-stage chain and sprockets provide a 9:1 reduction at all times. Another transmission is required to provide variable reduction at different engine speeds in order to balance the engine torque and speed in different driving scenarios. A clutch is also required to disengage the drivetrain from the engine at idling speed to allow the vehicle to stop with the engine running. A continuously variable transmission provides for both of these requirements. CVTech-IBC supports a mini Baja sponsorship program, through which they provide a CVT

specifically designed for the mini Baja competition at a discounted price. A new CVT was bought from CVTech and installed in the vehicle. Figure 14 below shows the CVT and completed drivetrain.



Figure 14: Completed Drivetrain

CVT Shaft Support

After testing, a secondary bearing block was added to provide extra support to the overhung portion of the CVT shaft because the engine mount plate provided a good location to do so.

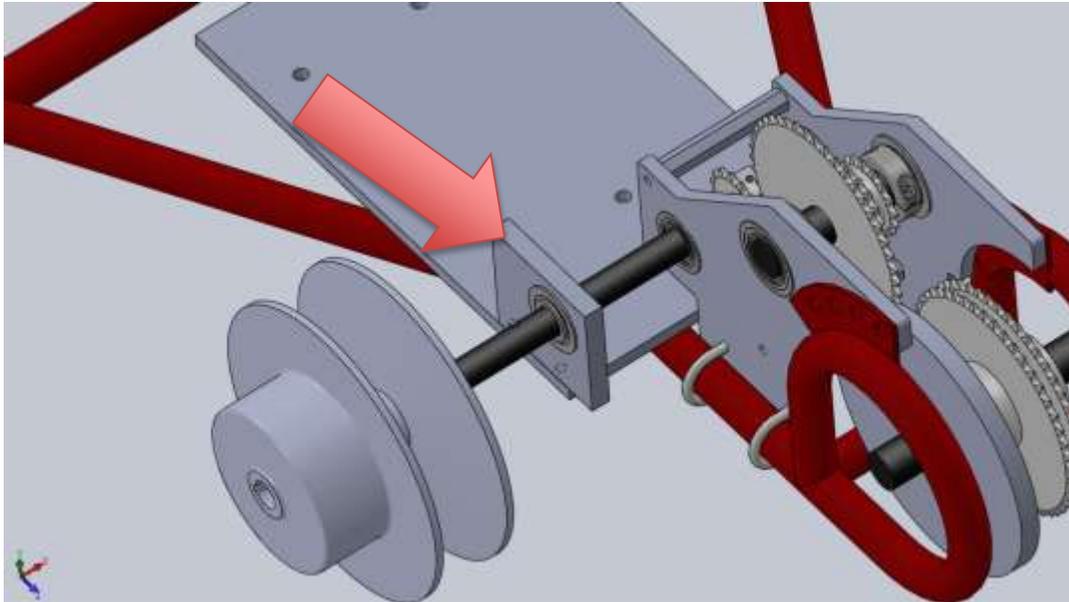


Figure 15: Support Bearing CAD Model

The block was machined out of a 5/8" thick piece of 6061-T6 aluminum. The bearing is the same as the others used in the bearing plates for this shaft. Holes are tapped into the bottom of the block to mount it to the engine mount plate. Excess material was removed from the block to clear the engine, as shown in the picture below.



Figure 16: Mounted Support Bearing

Brakes

The braking system of a vehicle is one of its main safety components. The goal of a braking system is to slow down the car in case of an emergency and allow for safer maneuvering whether being driven on a road or in an off-road competition. The Baja vehicle is equipped with disc brakes that are normally used on snowmobiles, and are controlled the same way most cars are, only smaller in comparison. The following is a discussion on the original design, new goals, and redesign of the overall brake system.

Original Brakes

The original brake design on this vehicle had a three-brake system. There was one brake on each front wheel, mounted as seen below, and one on the rear drive shaft, mounted to the final drive shaft within the drive train (original location also shown below).



Figure 17: Original Front Brake Mounts



Figure 18: Original Front Brake Caliper/Rotor Orientation



Figure 19: Original Location of Rear Disc Brake

When the hydraulic drive train was installed, the project team removed the rear brake setup. Braking power was instead obtained through the hydraulic system.

The original pedal assembly included a plate that would hold the brake and throttle cables as well as the master cylinder for the braking system. This setup is shown in Figure 20.



Figure 20: Original Pedal/Master Cylinder Setup

Brake Design Goals

Per SAE Baja rules, the main objective of a braking system is to be “capable of locking ALL FOUR wheels, both in a static condition as well as from speed on pavement AND on unpaved surfaces” (2013 Baja SAE Rules). The goal for the new brake system was to be able to lock up all four tires from a speed of 30mph. In order to achieve this, a four-wheel disc-brake system with two circuits (front and back) was designed.

Brake Design

The new brake design has two separate circuits, as stated in the SAE Baja rules. The front circuit was not changed from its original design, aside from the replacement of all components involved, with new parts.

To control these new circuits, two (2) master cylinders from Wilwood were purchased, one for the front brakes, and one for the brakes in the rear. Lines were run from both master cylinders to the front and rear calipers. For the front, the brake line was split directly at the master cylinder and then run to each side of the vehicle. The rear brake line was run underneath the seat, up to the area of the final drive shaft, and then split at that location to both the left and right sides to minimize the length of cable needed. The route is displayed in Figure 21. The green lines indicate the front system, while the blue lines represent the rear system.



Figure 21: Brake Line Route



Figure 22: Wilwood Master Cylinder¹



Figure 23: New Master Cylinder Setup

Two (2) Wilwood PS-1 calipers were installed onto the front knuckles. These calipers were connected to the master cylinder using steel-braided brake line. The new rear braking system was designed to be similar to the front setup, with two separate disc brakes, one for each wheel. The challenge was to design a rotor and caliper mount that would be installed with

¹ <http://www.wilwood.com/Images/MasterCylinders/Master%20Cylinder%20Photos-Large/260-2636-lg.jpg>

minimum modification to the existing rear knuckles, use the same rotor and caliper as the front wheels, not interfere with any other systems, be simple to fabricate, and be as light as possible. There is enough room inside the rear knuckle to fit the rotor and caliper. The rotor is mounted to a simple hub that was welded to the driveshaft and the caliper is mounted to a small aluminum plate. These parts are illustrated in Figures 24 and 25.

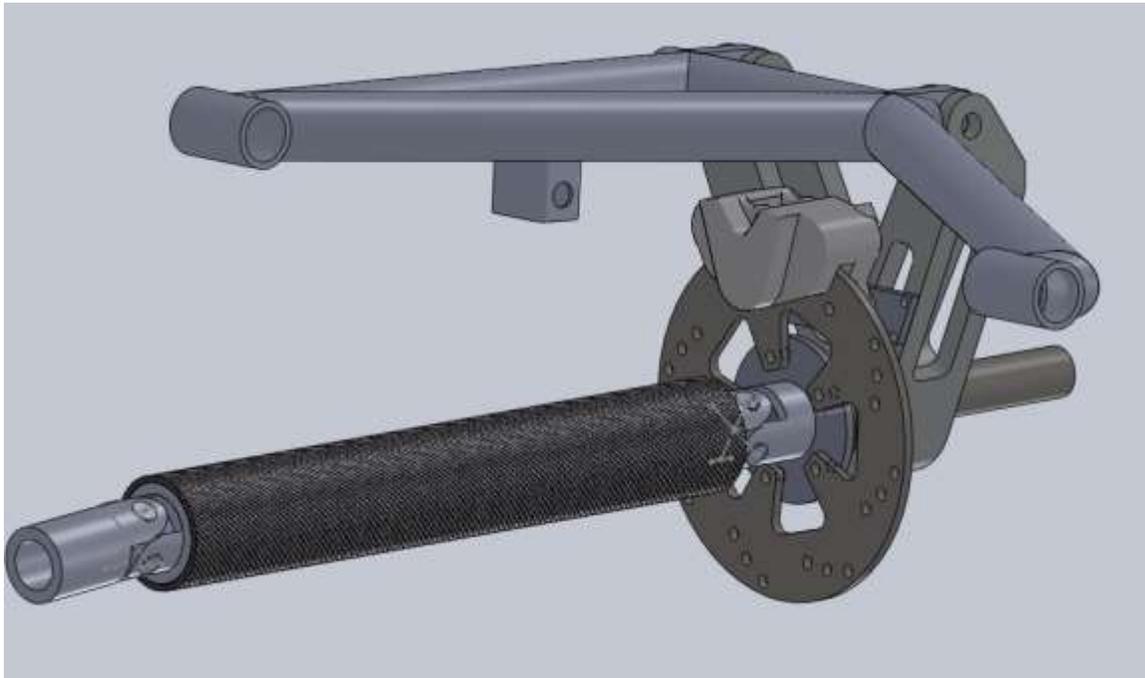


Figure 24: Rear Suspension Model with Rotor and Caliper

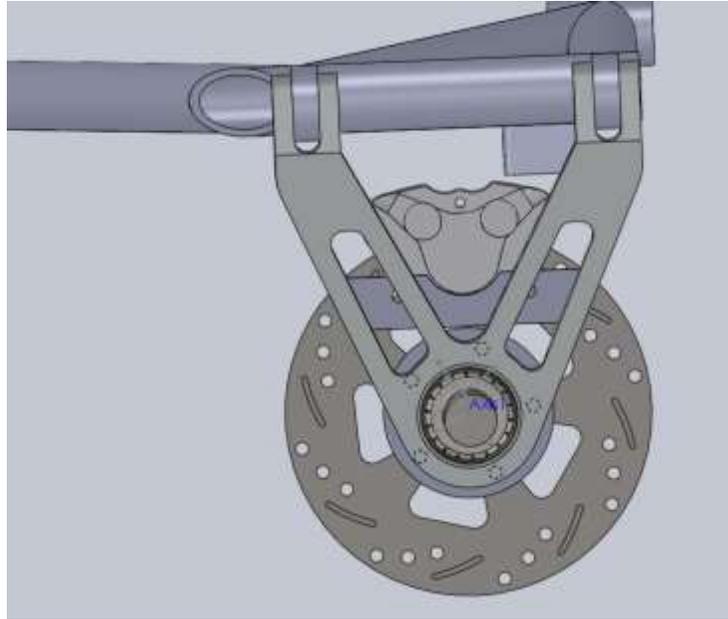


Figure 25: Rear Suspension Side View

A mount was manufactured out of aluminum that allowed the team to mount the additional Wilwood PS-1 calipers to the rear knuckles shown in Figure 26.



Figure 26: Current Rear Knuckle



Figure 27: Location of Caliper on Rear Knuckle

A hub was also machined and welded to the driveshaft before the knuckle to allow for the rotors to be mounted. There is not as much vertical clearance in the rear as there is in the front, so the original 6-inch rotors that were on the front wheels, will be used in this location instead.



Figure 28: Hub on Rotor



Figure 29: Welding Rotor Mount



Figure 30: Rotors Mounted to Rear Driveshaft

For the front rotors, we designed a new disc brake using SolidWorks. The part was then sent to Vangy Tool, Inc. to be water jet.

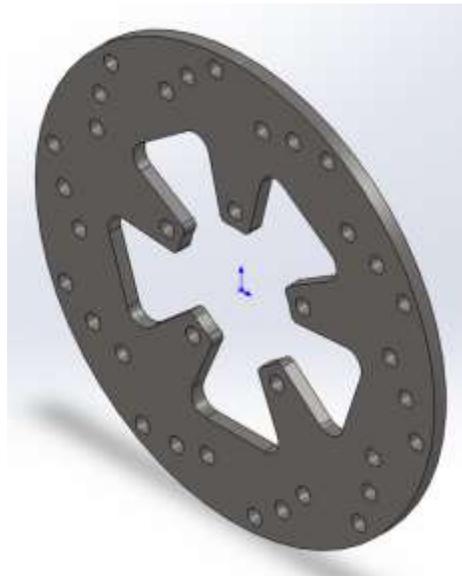


Figure 31: Original Rotor Design



Figure 32: Rotor after Waterjetting

The finished front and rear brake assemblies are shown in the following figures.



Figure 33: Finished Front Brake Assembly



Figure 34: Finished Rear Brake Assembly

Brake Testing

Field-testing the vehicle with the new braking system proved it was able to lock up all four wheels at top speed on dry pavement, as per the original design goals and SAE requirements. Brake response is very strong and predictable.

Front Suspension

One of the stated goals of this Major Qualifying Project was to re-design the front suspension and steering systems in order to improve handling and performance. Each component from the mounting points out was re-engineered. The mounting points could not be altered without extensive modification to the frame so the system was designed around this constraint. . In the course of designing an off-road vehicle, much attention must be paid to the terrain it will be navigating in order to develop a fitting suspension system. A Baja vehicle suspension must provide the car with the ability to compete in every event including the hill climb, endurance, and maneuverability competitions. A sufficient suspension will have the necessary practical

features such as adequate ground clearance and suspension travel to allow navigation of the terrain as well as provide comfort and control to the driver. The goal of any suspension is to maximize the contact between the tire and the track surface. Two basic methods of accomplishing this goal include reducing the weight of the suspension, which is called the un-sprung mass and increasing the stiffness of the mounting points on the vehicle or sprung mass. Reducing the un-sprung mass will decrease the effects of inertia in the system allowing it to react more rapidly to bumps. There are several different types of suspensions, each with their own advantages; however the double wishbone design allows for the most control of ride behavior and isolation of individual tire movement. For this reason, most performance vehicles employ double wishbone suspensions on the front axis and this design was no different. In off-road vehicle design, some attributes that provide necessary ride height and maneuverability must be prioritized over other parameters that might improve handling but cannot be optimized under the necessary design requirements. After researching and ranking the suspension characteristics discussed above, the team was able to define both static and dynamic goals for the new design. The design of the front suspension and steering will be explained as one since the two are closely related and changes made to one system can greatly affect the other.

Design Process

Different properties affect the performance of a suspension system and in order to design for a specific environment, they must be understood. There are both static and dynamic principles that characterize suspension movement. The more important of these will be discussed in order to explain the rationale behind the design changes.

Static Properties

Perhaps the most important factor to consider in the design of a suspension is the positioning of the tires. The tire is the link between the vehicle and the ground and it is important that the suspension maintains consistent tire contact with the ground. Camber and castor define the angular position of the kingpin, which is the center of rotation for the steering of the wheel. Camber is the angle between the kingpin axis and the ground when viewed from the front of the vehicle. When the bottom of the wheel is further out than the top, there is negative camber. Slight negative camber is desirable because it improves grip during cornering and stability when landing from a jump. Castor angle is the angle between the steering axis and the ground when viewed from the side of the vehicle. If the top of the kingpin is angled towards the rear of the vehicle, there is positive castor. Positive castor forces the front of the vehicle to be lifted slightly when steered, providing force feedback that centers the steering helping the driver to maintain a straight line. Unfortunately with a link type suspension, there can be no perfect situation where camber and castor are at the ideal angle all the time. Compromises must be made to balance the desired qualities of each. As the kingpin angle increases, the effect of castor angle diminishes. Therefore a balance must be found in order to maintain reasonable link and knuckle angles that will be able to handle the forces involved and be manufactured. Another important characteristic that is often seen in off-road vehicles is rake angle. Rake angle is the angle between the control arms and the ground when viewed from the side of the vehicle. Rotating the control arms counter clockwise in this plane allows more of the horizontal component of the force from large bumps to be applied directly to the suspension travel. Track width is another important characteristic because it affects control arm length and stability. The longer the control arms are, the more suspension travel can be obtained however the forces on the arms will increase. Generally the

track width should be as wide as is reasonably possible however for SAE competitions, track width must be limited to 64” in order to be able to navigate certain obstacles per SAE rules. It is fairly simple to find a compromise between these parameters under static situations however maintaining these characteristics during suspension travel is another challenge.

Dynamic Properties

An independent front suspension is a four bar linkage and due to the nature of this simple mechanism, link angles will change based on link lengths and driven angles. The angle of concern in suspension design is the kingpin angle, which defines camber. Camber angle will change during suspension travel and in order to maximize grip, negative camber is desired at all times. This is hard to achieve in off-road vehicles because there is significant suspension travel, which forces the linkage to have a large range of motion. Lengthening of the control arms will help to lower the angular displacement of the linkage, offsetting the effect of suspension travel. Often during suspension droop, some positive camber will have to be accepted in exchange for ground clearance and lower amounts of wheel scrub. This is acceptable, it is more important that camber remain negative during compression, which is when good traction is needed. Traction is often considered to be limiting wheel slip during longitudinal acceleration however, maintaining grip during lateral acceleration is also very important. In order to take advantage of static coefficients of friction between the tire and ground, the wheel must not slip. Any slip will initiate transition to dynamic friction, which will reduce performance. In vehicles this is called scrub. In order to minimize scrub, track width must be kept as constant as possible even during suspension travel. This means the instant center of the suspension must be kept at ground level as shown in Figure 35. In low, street racing vehicles this is possible but in Baja cars where large ground

clearance is necessary and control arms must be angled down towards the ground, it is more difficult.

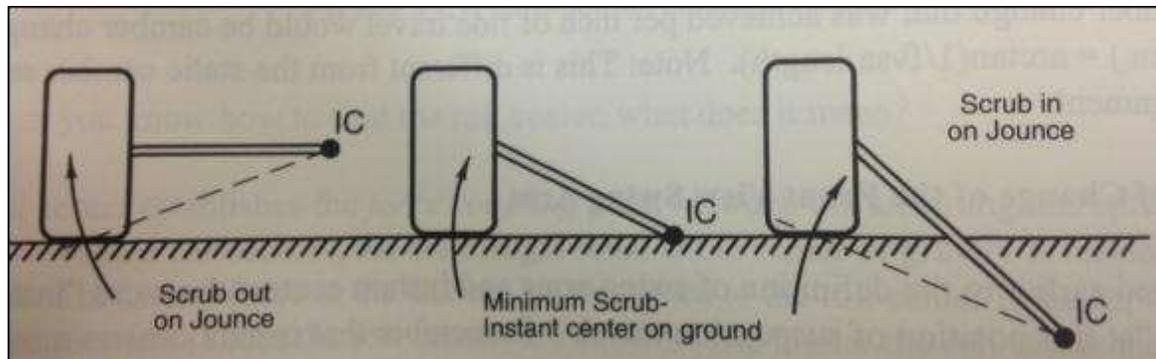


Figure 35: Tire Scrub

Some scrub will have to be accepted but because Baja competitions take place on slippery and uneven surfaces where traction is limited anyway, it will not be as detrimental to performance as it would be in road cars. The team decided that the most important qualities for off-road performance are ground clearance, suspension travel, and positive camber during compression and these were the focus of the design.

Original

The first step in the re-design was to analyze the original system to determine exactly what needed to be improved. The goal of the original suspension was to create large negative camber gains during steering in order to lean into each turn. This idea was inspired by motorcycle characteristics but this type of behavior did not translate well to a four-wheeled vehicle. Steering and suspension characteristics were adversely effective and the details of these problems will be discussed throughout this section of the report.

Many problems were found through simple inspection of the vehicle. As can be seen in Figure 36, the front wheels are tilted outward dramatically. This large amount of positive camber

can cause various problems, the most serious being torsional forces on the knuckle and control arms. A vertical force applied to the wheels in this position would produce a moment on the knuckle, magnifying the forces applied to the control arms. This could lead to catastrophic failure if the vehicle were to land from a large jump or encounter a rock at high speed.



Figure 36: Positive camber gain

The second problem with positive camber is that it cannot be maintained throughout the full range of suspension travel. This means that the distance between the contact patch of each front wheel will have to shift during droop and compression. The contact patch is the portion of the tire that is in contact with the ground and when the distance between them changes, it is called “scrub”. Scrub is an undesirable occurrence because it puts the contact between the tires and the road into dynamic friction, causing loss of traction.

Positive camber present in the original design was caused by different factors including knuckle design and manufacturing errors. The latter of these caused more problems than might normally be expected. Camber angle when the car is in a neutral position varies between the two

front wheels and can clearly be seen in Figure 37. This is likely caused by differing dimensions between the components on the left and right sides of the vehicle. Great care was taken in the fabrication of the new components to adhere to specified dimensions in order to avoid this issue.

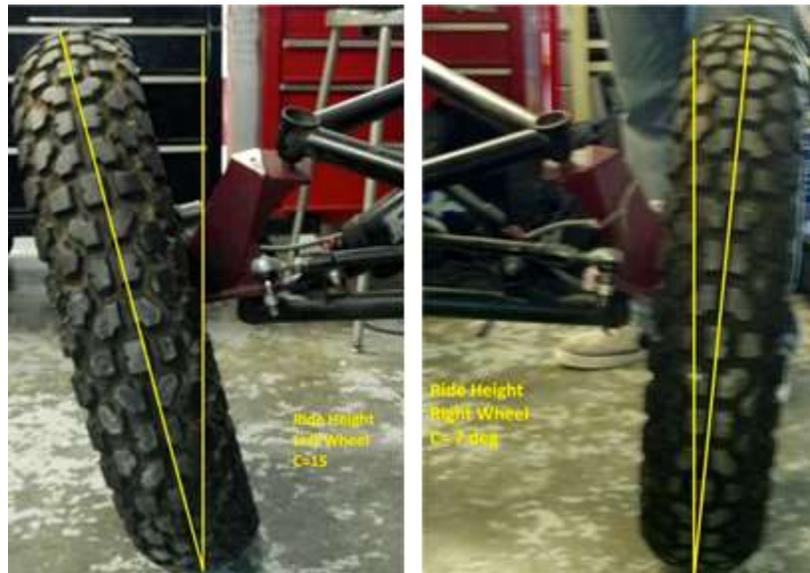


Figure 37: Camber of front wheels in neutral position

Perhaps the most influential parameter that affects camber is the design of the knuckle, specifically kingpin angle. Kingpin angle, the angle between the axis of the ball joints and the vertical plane, dictates the amount of camber gain. The original design calls for a kingpin angle of 20 degrees, which is quite dramatic compared to most systems. To achieve a kingpin angle of 20 degrees and a castor angle of 25 degrees, a radical knuckle design was required with large spindle offsets. The spindle was offset 3.5 inches laterally and 4 inches longitudinally. These offsets result in the unnecessary concentration of lateral forces on the upper control arm as well as a radical knuckle design that was difficult to manufacture. By reducing the kingpin angle to a more conservative value, the camber can be corrected and the components can be simplified.

The effects of kingpin angle are greatly increased during turning which makes it difficult to control the steering angle of each wheel. In order to avoid wheel scrub, the outside wheel must follow a circular path of larger radius than the inside wheel during the completion of a turn. This concept is called the Ackermann steering principle. Figure 38 shows the vehicle in a left turn and it is easy to see the different angles. The problem is that the difference in angles is far too great and because more force is applied to the outside wheel during a turn, the inside wheel is effectively dragged laterally across the track surface.



Figure 38: Incorrect steering angles

Turning also causes the camber of each wheel to respond differently. The effects of kingpin angle are increased when the knuckle is pulled inward by the tie-rod. This means that during a turn, the inside wheel will experience very large positive camber gains in excess of 28 degrees or more while the outside wheel will remain nearly vertical as shows.



Figure 39: Camber at max compression and full-lock left

Dramatic scrub occurs as the camber increases and in a high-speed turn, this can cause the wheels to slip. Wheel slip results in under-steer, which means the car will not actually turn when the wheels are turned. This effect was actually experienced during testing and can be quite unsettling. When the driver cannot expect the vehicle to respond to steering input, a loss of confidence can lead to poor performance and potentially unpredictable

From observation and testing, it was determined that the two greatest weaknesses in the original design are the camber gain and the difference in steering angles. Correcting these issues was the focus of the re-design, which will be explained in the following section.

New Suspension Design

A quality front suspension is vital to achieving good handling in any vehicle. If the driver cannot predict how the vehicle will respond, he or she will not be able to operate the vehicle at the limits of its capability. The front suspension must provide a smooth ride while maintaining

traction and driver control. Being an off-road vehicle, the suspension for this Baja vehicle was designed with simplicity and ruggedness in mind.

Preliminary Design Parameters

During the beginning design phase of the front suspension, a list of specifications was developed to define goals for static dimensions at ride height and also the dynamic capabilities of the system. Some of these goals were set based on known challenges that would be faced in SAE Baja competitions and others were determined from testing of the original suspension. These specifications allowed a suspension to be built that was within design constraints and would be able to improve upon the original design.

Static Parameters

Dimensions and characteristics for the Baja vehicle at ride height were determined from our design specifications and the SAE rules. SAE competition limits the track width to 64" and this was the goal because the wider the track width, the better the vehicle will handle and the more stable it will be. Figure 40 below summarizes the track width and other static values discussed below. The static camber angle was set between negative 2 and 3 degrees to ensure steering would be crisp and responsive. Variations in ride height caused by different shock absorber settings could cause the camber to change slightly so the slight negative camber was chosen to be certain the camber would always remain negative at ride height. With the wide track width, a large amount of ground clearance could be reasonably designed for without forcing the control arms to protrude at large angles. Twelve inches was chosen because it would allow the car to easily maneuver large bumps and ruts and even some obstacles such as tree limbs. The large tires chosen by the original MQP team will serve to help raise ground clearance and reduce shock to the suspension when bumps are encountered.

Dimensions at Ride Height	
Track width	60" to 64"
Camber	negative 2 to 3 deg.
Ground Clearance	12"
Tire Size	25"
Wheel Size	17"

Figure 40: Dimensions at Ride Height

Dynamic Parameters

With these dimensions finalized, the dynamic characteristics of the car could be defined as shown in below. One of the most important qualities in an off-road vehicle is suspension travel. Suspension travel is the amount of vertical wheel displacement allowed by the given system. Large amounts of travel are necessary for keeping all four wheels on the ground while traversing rough terrain. The goal for wheel travel was seven inches up and five inches down. With seven inches of upward travel, the suspension could absorb shocks from large bumps or obstacles and the five inches of downward travel would provide adequate length to maintain tire contact with the ground during droop. Next, dynamic camber must be considered. As discussed earlier, negative camber is desired during compression of the suspension in order to improve grip and stability. Although negative camber is desired in moderation, positive camber must be minimized as much as possible while still maintaining negative camber during compression. The original design resulted in too much camber gain during suspension travel and the aim was to reduce this effect in the new design.

Kinematic Suspension Performance	
Wheel Travel	
Up	7"
Down	5"
Camber	
Compression	10 deg
Droop	minimize
Track Width	minimize

Figure 41: Kinematic Suspension Performance

Control Arm Design

A double wishbone suspension is composed of an upper and lower control arm, which effectively makes up a three-dimensional four bar linkage. The driven angles and the ratio between the link lengths have the greatest impact on how the suspension will behave. Preliminary synthesis of the linkage was done in AutoCAD. Two-dimensional sketches were created of each view of the suspension. The front view of the suspension determines link lengths and kingpin angle. Castor angle, castor trail, and spindle offset are defined in the side view sketch. Figure 42 shows the sketches drawn out in AutoCAD, the front view synthesis is on the left and the side view is on the right.

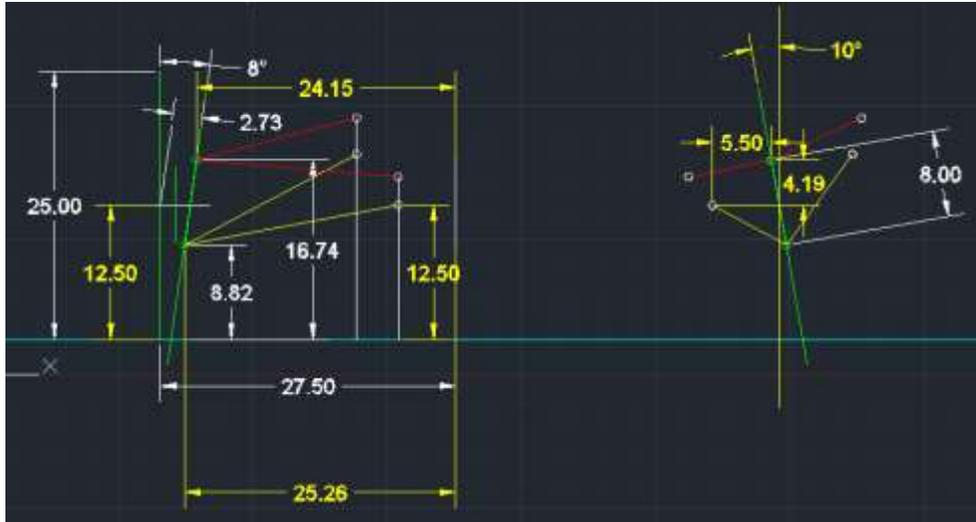


Figure 42: AutoCAD Suspension Sketches

The first step in the process is to establish the mounting points on the frame. Since the design was being built off an existing frame, these points were already fixed. Next, the desired track width and ground clearance must be sketched out. With these two values and the distance between the outside of the wheel and the kingpin axis, the length of the lower control arm is defined. From there, the length of the kingpin must be chosen. Kingpin length has an effect on the rate of camber change as well as the force loading on the control arms. A short kingpin length will result in higher rates of camber change and larger loading forces on the control arms. For this reason, the longest practical kingpin length is desirable. Eight inches was chosen as this length because it is short enough to be practical and provide adequate camber gains without being so short that it would place undue forces on the control arms forcing the metal tubing thickness to be increased. Once the kingpin length was defined, the upper control arm was the only unknown length. This was set to allow for a beginning kingpin angle of 8 degrees.

Once the two sketches were completed, the views were combined with simple geometry to determine the exact link lengths and angles for the control arms as well as the kingpin and

caster angle. With these initial values, a sketch of the front view was created in SolidWorks. Although it was two dimensional, the link lengths and angles were adjusted to account for the location of each length in the z-direction. The SolidWorks sketch allowed the linkage to be moved throughout the range of motion it would experience in service. The camber change during travel must be plotted in order to calculate the ideal upper control arm length which was left unsolved in the two dimensional analysis.

In order to determine the dynamic camber change as a function of the lower control arm angle, a Mathcad file was created to perform a four-bar linkage analysis. Professor Robert Norton's text book entitled *Design of Machinery* served as a guide in producing formulas for camber angle in terms of the chosen link lengths and the driven angle of the linkage which was plotted from 35 to 65 degrees as shown in below. The full Mathcad code can be found in the appendix.

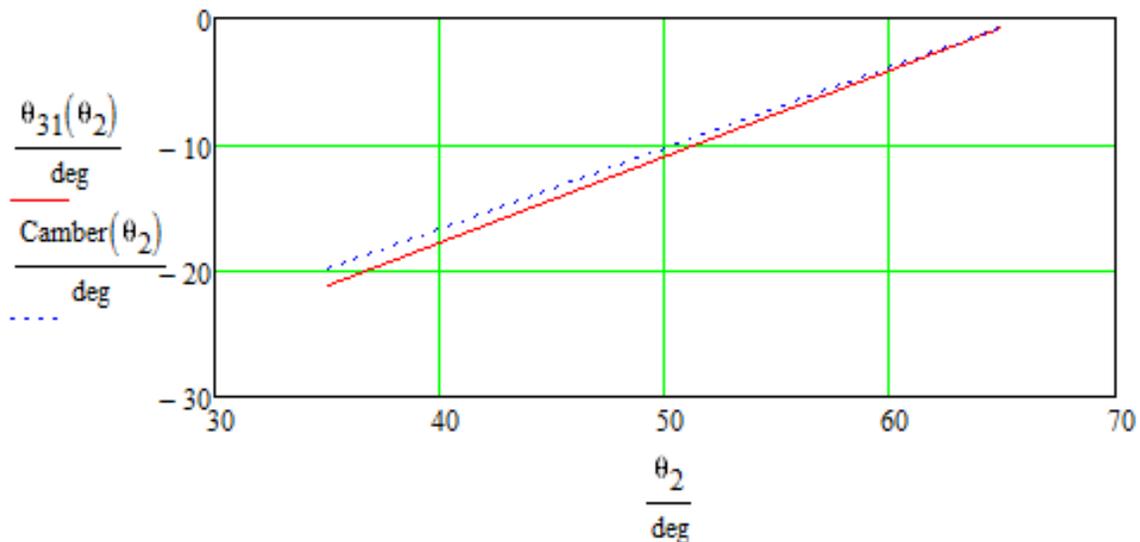


Figure 43: Plot of Camber vs. Suspension Travel

The dotted line in the plot is the un-adjusted angle of the kingpin length throughout travel. The solid red line is adjusted for the rake angle of 20 degrees, which is a forced parameter because of the fixed geometry on the frame of the vehicle. Optimal values of the upper control arm length and kingpin angles were determined through iteration. An upper control arm length of 15.25 inches was deemed most beneficial. The kingpin angle was set to 12 degrees and the castor angle was then found to be 10 degrees. Castor is necessary in order to create mechanical trail. Trail causes the wheels to angle during turning, lifting the vehicle slightly. This lift provides force feedback in the steering, which helps the driver to maintain a path without making constant corrections. A final sketch of the front view suspension was created in SolidWorks to verify the design. Figure 44 displays the final upper control arm length and kingpin angle as well as the ground clearance. The height of the kingpin is fixed by the diameter of the tire so the chosen ground clearance defines the angle of the angle and length of the lower control arm.

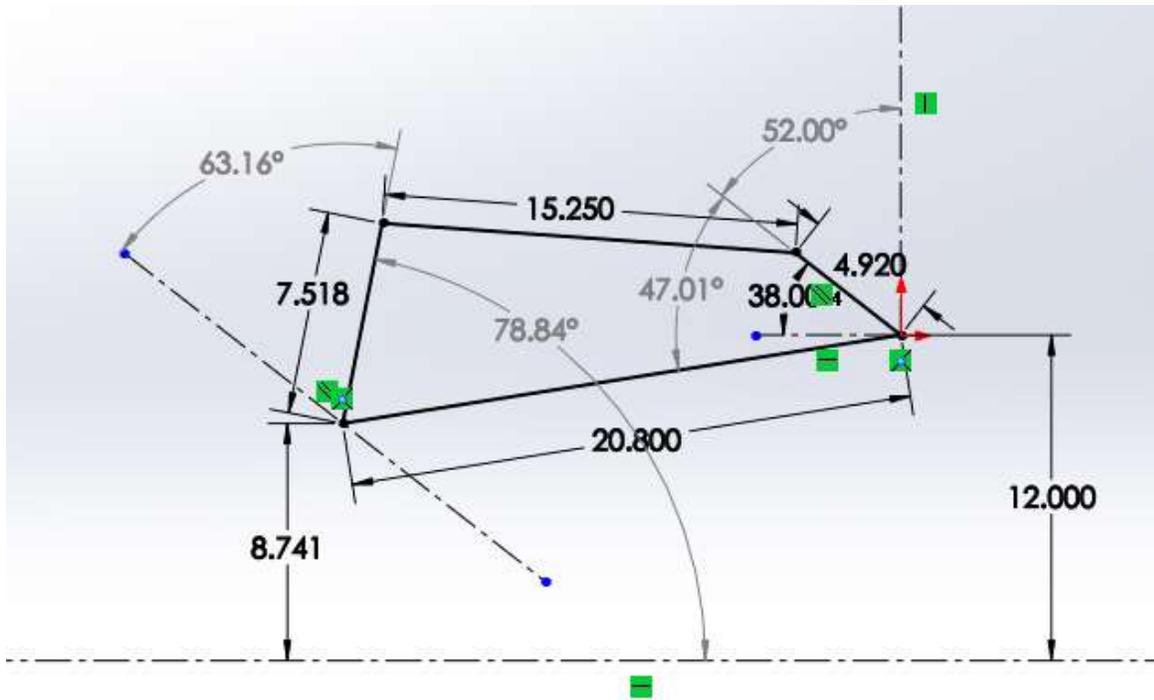


Figure 44: Final Four-Bar Values

With the final control arm dimensions confirmed, a solid model could now be constructed in order to confirm fit and function as well as to perform finite element analysis. The first step in this process was to determine the location of the four suspension mounts in 3D space. This was accomplished by working with the solid model of the frame and confirming dimensions on the actual vehicle. These dimensions along with the given ground clearance of 12 inches fixed the four points in space. The two kingpin joints were sketched next. These points were set given the required track width and the castor and kingpin angles determined in the 2D synthesis. The completed 3D sketch was used to create a part file for each control arm. SolidWorks' weldments feature was used to create the tube lengths. At this point, the tube thicknesses from the original design were used until a finite element analysis could be performed on the new control arms. An assembly was created as shown in by mating the frame mounting points of each control arm

together. A distance mate was created between the two kingpin joints to allow the assembly to travel up and down as it would with a knuckle in the assembly.



Figure 45: Control Arms- Solid Model

The front suspension design synthesized in this project possesses the static and dynamic properties desired to maximize performance in an off-road environment. Compromises were made between handling characteristics such as camber gain and scrub, and required static traits including ground clearance and track width. The handling characteristics of the vehicle were

synthesized as ideally as possible given the limitations imposed by the realities of off-road competition.

Knuckle Design

In order to connect the new control arms to the spindle, a knuckle is required. The new knuckle was designed with two things in mind, ruggedness and simplicity. In an off-road environment, durability takes precedence over other design goals. Simplicity was desired because the knuckle would be the most difficult component to manufacture. Geometries were kept as simple as possible and unnecessary features were eliminated.

Two options for the knuckle were considered during the design process. Either the knuckle would be CNC machined from an aluminum billet or it would be weldments of steel sheet metal. There are advantages and disadvantages to both. Aluminum is easy to machine and the one-piece component would eliminate the risk of failure due to faulty welds. Steel is more difficult to machine but it is easier to weld than aluminum and has a higher yield strength. It was decided that 4130 steel would be the material of choice because the team had more experience with welding than machining and steel does not have the finite fatigue limit that aluminum possesses. The 4130 alloy is also capable of attaining yield strengths of 130 ksi or more after heat-treating while still remaining flexible enough to absorb the impacts encountered in an off-road environment. With material selection complete, the knuckle was designed with sheet metal fabrication in mind.

An initial concept was created using .25” thick sheet stock. The original spindle housing was re-used in order to save time in manufacturing. Figure 46 shows the simple design with holes to fit the ball joints, a tab to mount the brake caliper and a simple rectangular upright at the

required 8-degree kingpin angle and 10-degree castor angle. This was the simplest design that could be devised. Some machining would be required in order to achieve the 8 degree angle bore for the spindle mount however the castor angle would be produced by the ball joint offset in the control arms. Another option would have been to angle the ball joint tabs and bore a straight hole through the upright for the spindle tube. It was decided that although machining would be simpler, this would be more difficult to fixture and clamp correctly for welding. The angled bore could be machined for a light press fit eliminating the need for fixturing before welding.

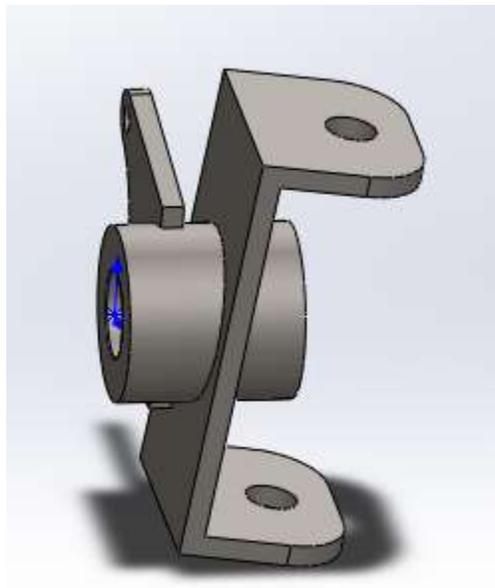


Figure 46: Initial Concept

Stress analysis will be discussed in a future section of the report however in order to maintain an orderly explanation of the design work, some of the results of the analysis will be discussed here. Finite Element Analysis revealed high stress locations in the upright around the spindle housing. In order to better distribute these forces, triangular gussets were added to the knuckle as shown in Figure 47. Gussets were also added to the ball joint tabs to reduce bending stresses.

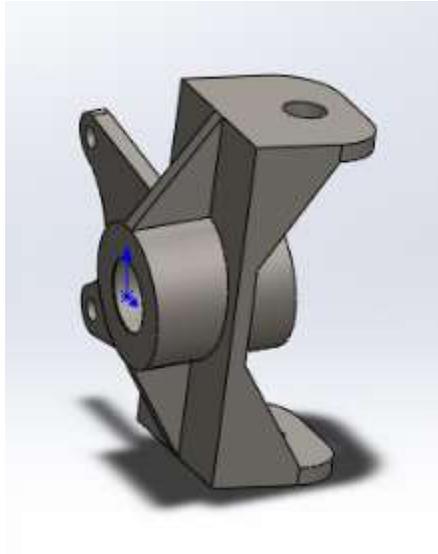


Figure 47: Triangular Gussets Added

The new knuckle complete with triangular gussets, was able to shave 1 lb of weight off the original design. Compared to the original, the new design also contained much simpler geometries, which would make manufacturing much easier. Many different bends were required for the original knuckle including some in three dimensions and varying angles. The improved version required no bending and only minor machining to produce the needed angles and clean edges for welding. In order to be able to steer the car, tie rods and mounting locations on the knuckle would be required, this was the next step in the design process.

Steering Design

A linear and predictable steering system is vital to a competitive Baja vehicle. The driver must be able to maintain control of the vehicle at all times. A poor steering system can lead to crashes and potentially injury.

The principles that govern steering are fairly straightforward compared to suspension design. The two main goals are to obtain correct steering angles for each wheel and to achieve

slight camber in the direction of the turn. Figure 48 portrays the Ackermann principle, which was discussed earlier.

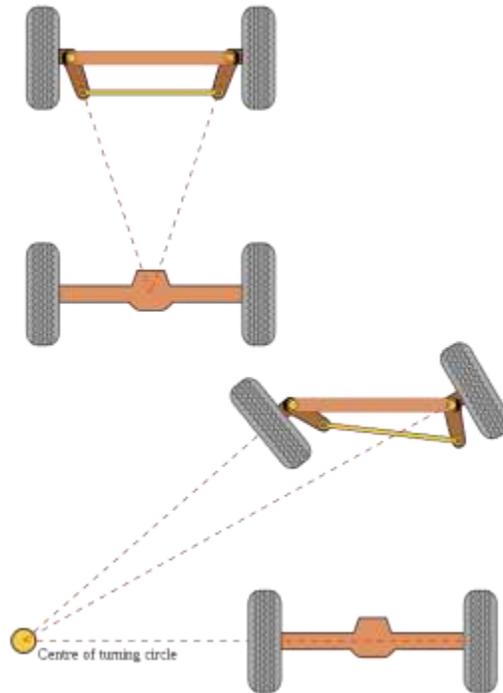


Figure 48: Ackermann Principle

The lower part of the figure explains how the property works. The centers of rotation of each of the front wheels must intersect the axis of the rear axle at the same point. A good approximation of this for design purposes is for the two steering arms to intersect at the center of the rear axle. It does not matter if the steering arm itself points towards the center of the rear, what is important is that a line drawn through the kingpin axis and the tie rod mounting hole intersects the rear. The improper location of this mounting hole was part of the reason why the previous design produced odd steering angles that caused scrub and under steer.

Another issue that comes in to play when large amounts of suspension travel are involved is a change in the distance between the steering mechanism and the tie rod mount on the knuckle.

In the case of the Baja car, during compression this distance actually becomes shorter thereby pushing the knuckle out and causing toe in. This also causes wheel scrub and instability. In order to reduce this, the mounting point on the rack and pinion must remain the instant center of rotation of the knuckle during as much of the suspension travel as possible. Unfortunately, it was not possible to move the rack and pinion without modifying the frame. Any change would have made it very difficult for the driver to get his/her legs into the vehicle to reach the pedals. Optimizing the location of the mounting point on the knuckle was the best option. The new knuckle sits about 3 inches lower than the original due to the spindle being in the center of the knuckle rather than an inch below it. This meant that the best that could be done was to mount the steering arm as high up on the knuckle as possible as shown in Figure 49.

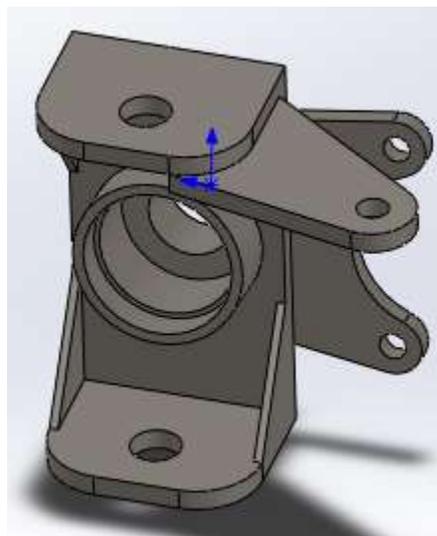


Figure 49: Steering Arm Added

The tie rod lengths would also have to be shortened due to the new control arm and knuckle geometry. Another important parameter is the distance between the kingpin axis and the mounting hole for the tie rod. This distance along with the ratio of the rack and pinion, dictates the overall steering ratio. In Baja competitions, it is required to have a harness holding the

drivers wrists to the steering wheel. This makes it difficult for the driver to take his hands off the wheel to complete tight turning maneuvers. To alleviate this problem, a goal of the new steering system was to increase the ratio so that the car could be turned from lock to lock with 360 degrees or less of steering wheel rotation. Through the manipulation of the SolidWorks assembly discussed in the following section, it was determined that a distance of 3 inches would create the desired ratio. With this distance and the angles calculated for Ackermann Steering, the geometry was completely defined and the knuckle model could now be finalized.

Assembly in SolidWorks

A SolidWorks assembly was created of the entire system in order to verify the kinematics before fabrication. The model, shown in Figure 50, was also used to determine some parameters including tie rod length and steering arm angle. Three-dimensional sketches created during the control arm synthesis were used as the base for the assembly. Control arms were imported first followed by the knuckle and ball joints. The ball joints are from a Polaris ATV and the model was found online. Rod ends connect the upper control arm to the frame to allow for camber adjustment after fabrication. The shock mount location on the lower control arm was determined based on a 6-inch travel range for the shock. Two thirds of the travel was devoted to compression and one third was left for droop. At static ride height, the shock is compressed two inches. The hub, spindle, and wheel were imported from the model of the original car. In order to leave more room to mount the new brake calipers, the spindle housing was shortened by 1 inch over the original design. This effectively lengthened the spindle allowing the hub assembly to be pushed outward.

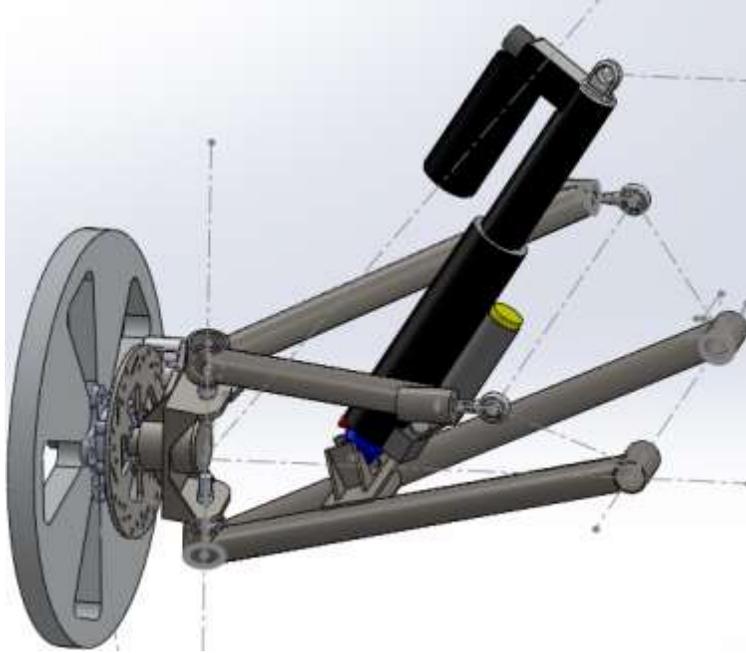


Figure 50: Right Side Assembly

The locations along the spindle of the various hub assemblies was determined through experimentation in the model. The caliper was mounted first, leaving enough room to not interfere with the knuckle. By locating the caliper, the location of the disk and hub were fixed as shown in .

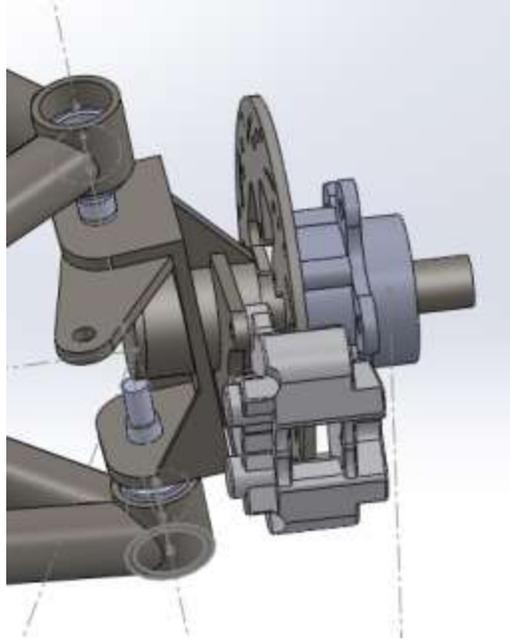


Figure 51: Knuckle and Hub Assembly

The spacing between the hub components was created using distance mates. In the assembly of the actual vehicle, steel spacers were used to achieve proper location. Each hub contains two ball bearings pressed into the main bore and can be seen in Figure 52. These bearings are 2 inches in diameter with a 1 inch bore. They are 9/16 of an inch in width. The spindle is pressed into the spindle housing and makes a slip fit with the two bearings. Washers and a castle nut hold the entire hub assembly onto the spindle. Snap rings prevent the ball joints from pushing out of the control arms.

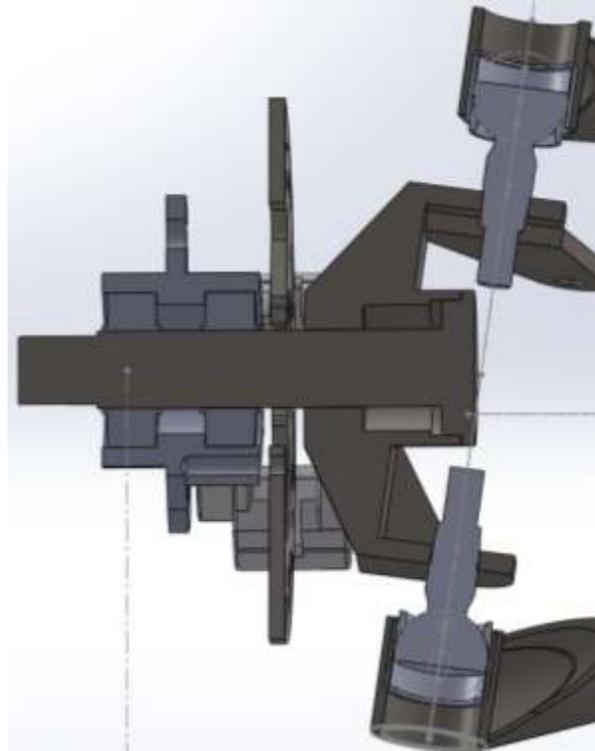


Figure 52: Hub Assembly Section View

With the geometry of each component verified in the solid model, the final iteration of the design was confirmed. It was now time to perform a Finite Element Analysis on the major components to verify material thicknesses and minimize stress concentrations.

Finite Element Analysis

The principles of Finite Element Analysis were applied to the major components of the front suspension in order to verify geometries and material thicknesses. SolidWorks Simulation was used to perform the various studies laid out in this section. Before any stress analysis could be performed, some basic forces were calculated using free body diagrams and information from previous Baja projects. Based on the weight of the original vehicle and the average weight of the driver, the total weight of the system was estimated to be around 400 lbs. In a previous graduate study at WPI, it was determined that the maximum acceleration a Baja vehicle is likely to endure

during competition is 3g's. With this knowledge, 1200lb forces were distributed on each component in various directions in order to simulate the stresses encountered during landing from a jump, lateral acceleration due to turning, and frontal impact. The resulting stress plots are presented below.

The knuckle was tested in order to identify any concentrated stresses in the design. Fixed geometry constraints were added to the ball joint holes at the top and bottom of the knuckle in order to prevent the part from moving when forces are applied. The knuckle was first loaded with a simple vertical force through the spindle similar to the vehicle landing from a jump. The resulting Von-mises stress plot is shown in Figure 53. In the original study, the triangular gussets shown below were not present. High stress areas were found at the top and bottom of the spindle tube where it meets the main plate of the knuckle. After this discovery, the gussets were added to spread out the force and reduce the stress concentration. The maximum stress was found to be approximately 24 ksi, which is well below the yield strength of 67 ksi for normalized 4130. After heat-treating, the yield strength would increase to around 140 ksi.

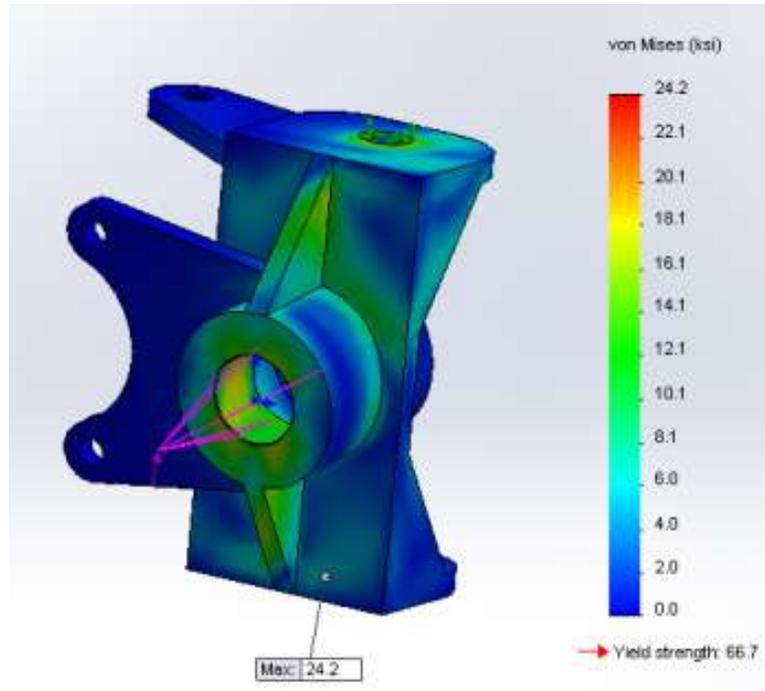


Figure 53: Knuckle Von Mises During Landing

The safety factor for the knuckle was plotted for this loading. Figure 54 shows the areas with the lowest factor of safeties in green. The minimum factor of safety was roughly 2.8, which was more than acceptable.

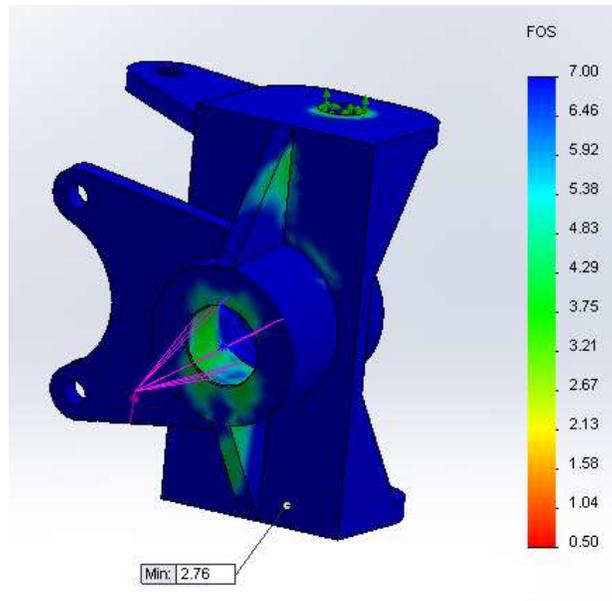


Figure 54: Knuckle Safety Factor

Besides vertical loadings, the knuckle would also be subject to bending moments due to side impacts with the tire and lateral acceleration during turning. For this case, the 1200 lb force was applied at a 12.5 inch moment arm which is the distance between the center of the spindle and the ground (the radius of the tire). This case produced a maximum stress of approximately 112 ksi, the highest found in the knuckle for the various loadings. This value is still more than 20 ksi less than the maximum yield strength after heat-treating.

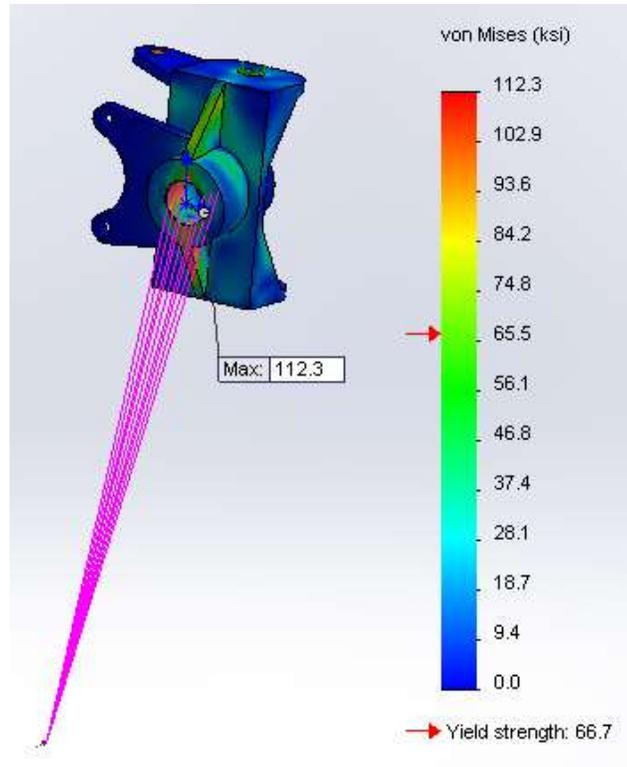


Figure 55: Knuckle Stresses During Side Impact

With the addition of the triangular gussets to support the spindle tube housing, the knuckle was deemed acceptable and ready for manufacture. Next, the control arms were analyzed to determine the correct balance strength and lightweight.

The lower control arms would endure higher stresses than the upper since they transfer any vertical loads from the ground into the shocks. A 1200 lb vertical force was applied to the ball joint housing and the other two ends of the arm were fixed. The resulting stresses are shown in Figure 56. The maximum stress was about 77 ksi, which is well below the yield strength and allows for a factor of safety of nearly 2.0.

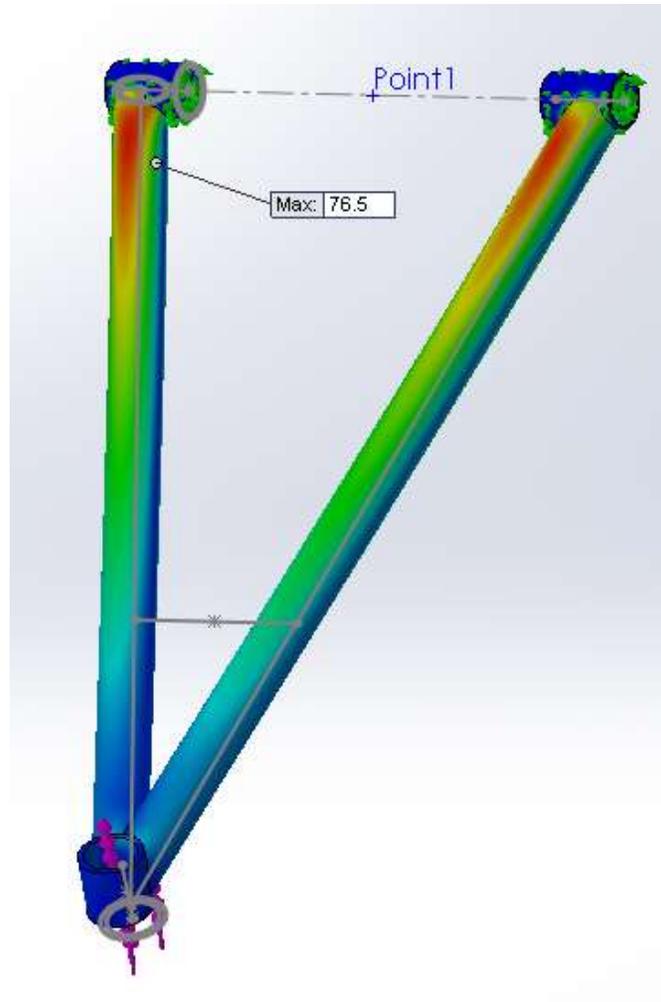


Figure 56: Lower Arm Stresses During Landing

The lower arm was also tested for strength during a frontal impact. Being an off-road vehicle, the car must be durable enough to withstand minor collisions and bumps without failure. A 1200 lb force applied at the ball joint housing towards the rear of the vehicle produces a maximum stress of about 60 ksi, as shown in Figure 57, which is even less than the vertical loading case. As in previous years, 0.065” thick tubing was found to be the thinnest allowable for the lower control arms.

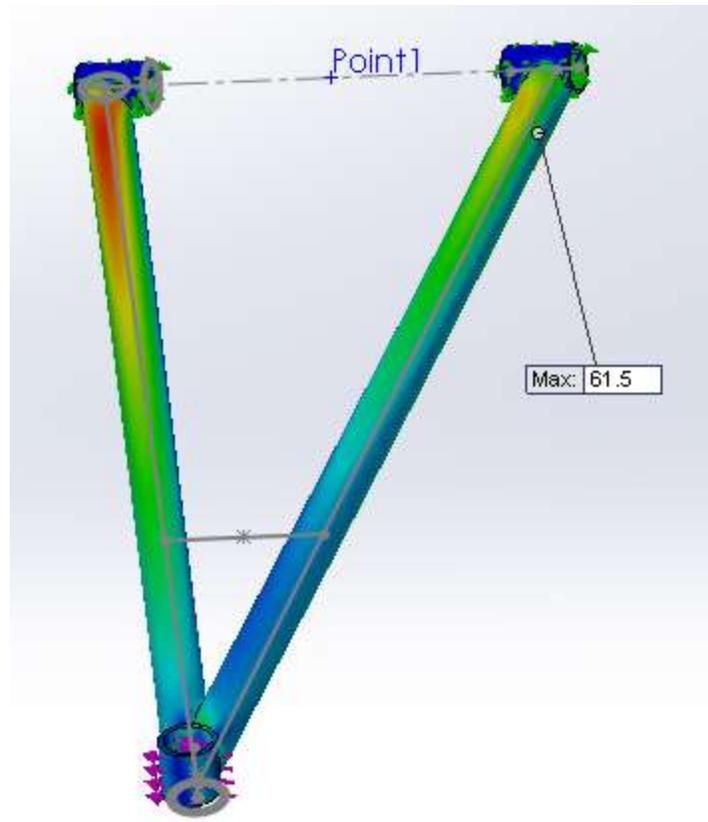


Figure 57: Lower Arm Stresses During Frontal Impact

The upper control arms were tested in both compressive and tensional loading cases. The lack of shock mount on the upper arm means that the member is always either in pure tension, or pure compression. Through experimentation, it was found that both cases produce nearly identical maximum stresses. As would be expected, the maximum stress occurs at the corners where the main tube segments meet the rod end housings. At these locations the stress reaches about 42 ksi with 0.035” thick tubing. Figure 58 shows the dynamics of the loading including the force applied at the ball joint housing and the fixed geometry constraints applied to the rod end housings. This is well below the yield strength as is the maximum stress during tension.

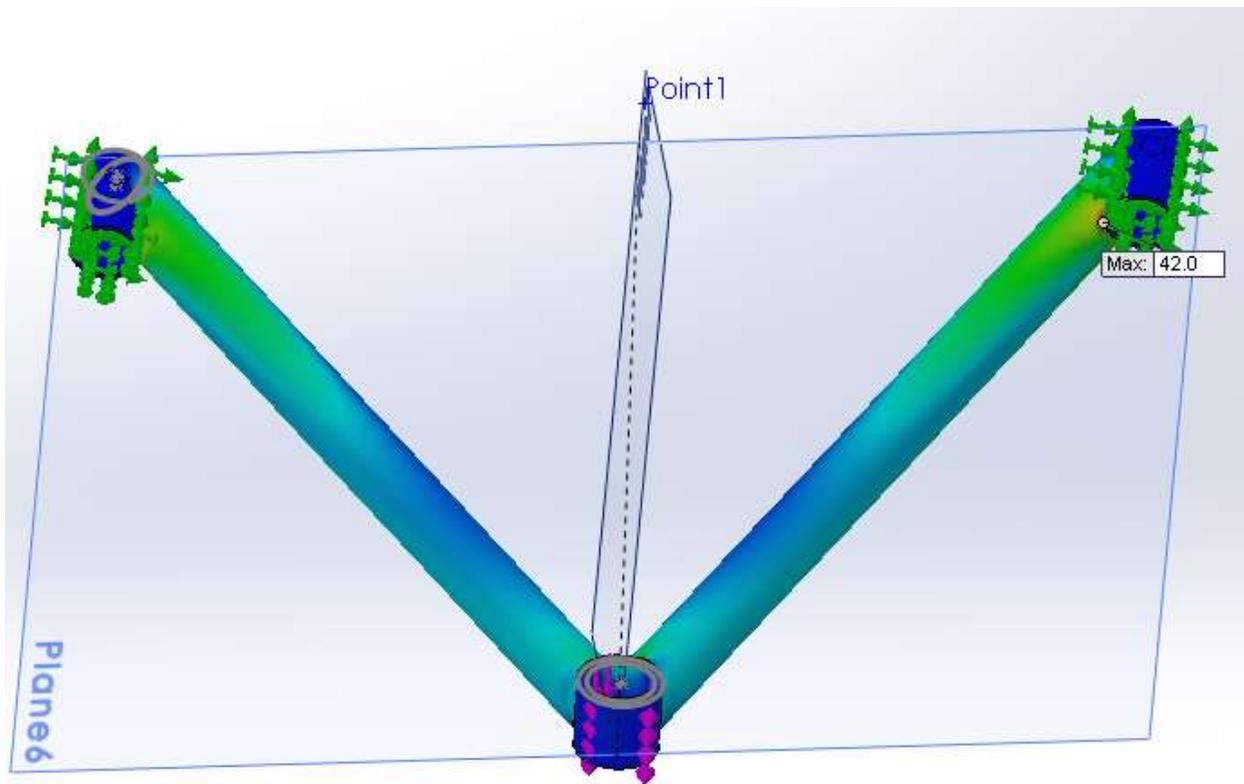


Figure 58: Upper Arm Stresses in Tension

With the Finite Element Analysis complete and the strength of the components verified, the team could now begin the process of manufacturing the new parts. It is important to note that the stresses found above are accurate provided the components are manufactured correctly and so, great thought and care were put into this process.

Manufacturing

Of the many different stages of this project, manufacturing was the most unpredictable as far as amount of time required. The process of going from solid model to actual component was full of unanticipated obstacles. A recommendation for future Baja teams is to allow extra time for manufacturing and to outsource as much of the fabrication as possible. The three main components that were manufactured for this project were the upper and lower control arms and

the knuckles. The goal of this section of the report is to provide as much detail as possible so that future teams will not have to start off at square one. Any further questions about the process may be directed towards this year's team members.

Control Arms

The first parts to be fabricated were the four control arms. During the notching and welding of these arms, the most important goal was to be consistent so that the two uppers were identical and the two lowers were identical. This is important in avoiding issues with differing properties like the unmatched camber angles on the original design. Machining the end pieces, notching the tubing, welding, and heat-treating are four main steps to completing the control arms.

Ball Joint Housing and Rod End Housing Machining

Each control arm has a housing to hold a ball joint. This housing was machined from a length of 4130 round stock with a diameter of 1.5 inches. Normally this part would be turned on lathe however they were milled in one of the Haas Mini-Mills due to a tight time schedule and lack of tooling for turning. The parts were made in two simple operations using special aluminum soft jaws milled to the correct diameter for fixturing. The first operation, shown in Figure 59, involves facing the top surface for a good finish and boring the clearance bore for the ball joint to slide through. This is a deep bore, almost two inches, and it required an extended length four-flute end mill with excellent coolant flow.

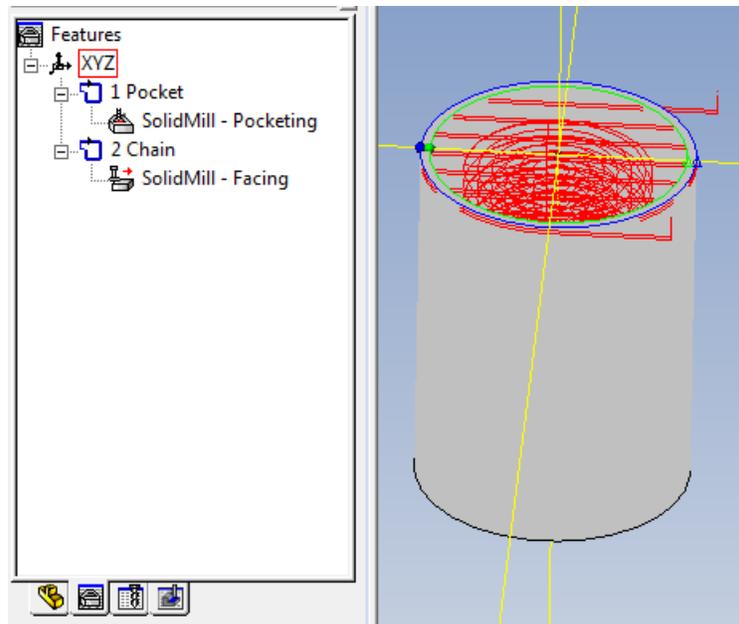


Figure 59: Ball Joint Housing OP1

Next, the other side of the housing was faced and bored as well. This bore required much tighter tolerances. A light press fit was required between this bore and the ball joint so the allowable variation was only .0005 inches. This was hard to achieve especially because four of these housings had to be made. It would definitely be recommended that in the future, parts like this be turned. The facing and boring tool paths for this second operation are shown in Figure 60.

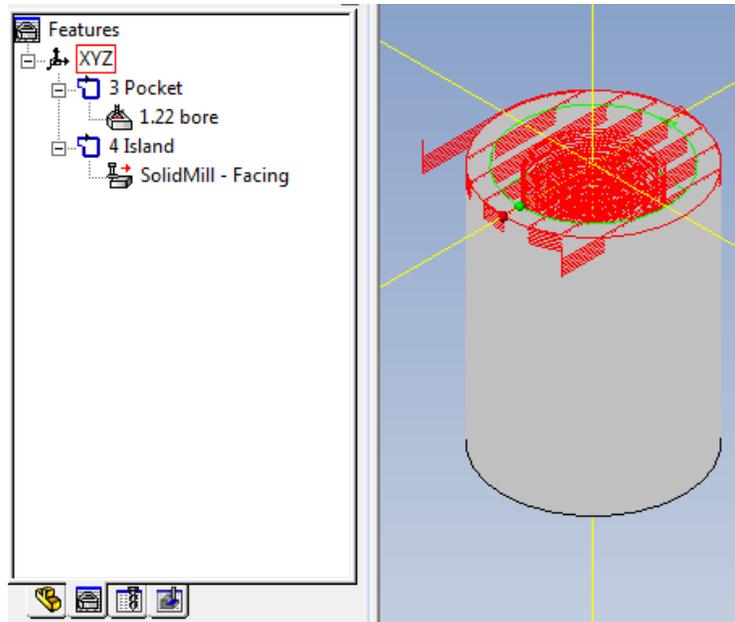


Figure 60: Ball Joint Housing OP2

Each control arm also required two fittings in order to attach to the frame of the car. The upper arms have a threaded housing for the rod ends and the lower arms have pieces of tubing for Delrin bushings to be pressed into. The rod end housings were made in a very similar fashion to the ball joint housings. They were faced on both sides and a hole was drilled and then tapped by hand. With all of the housings complete, it was now time to notch the tubing for each arm.

Tube Notching and Fitting

Before the control arms could be welded, each section of tubing required notching to fit. Tube notching is not a simple task and there are various ways it can be accomplished. CNC machining produces the best cut for a seamless fit however it is difficult to orient the profile on each end of the tube because both ends cannot be machined in one operation. The use of a hole saw in a drill press is another option and some experimentation was done with this method. Cutting thin tubing such as is required for the control arms was difficult, the hole saw would not cut smoothly and was often damaged. The method that was eventually chosen was to grind each

tube by hand. In order to make this as accurate as possible, a template of each control arm was created using the SolidWorks sheet metal features. It is important that the scale of the printed template is verified as it is very easy to accidentally print in the wrong size which can lead to incorrect control arms and a great deal of extra work. The template was then wrapped around the tube and used to mark the cut. Each tube was then cut and finish sanded to fit. In order to achieve correct tube location, the tubes for each control arm were laid out on full-scale drawings before tacking as shown in Figure 61 below.



Figure 61: Layout for Welding

The greatest difficulty in tacking the control arms was achieving the correct angle for each of the ball joint housings. It was difficult to determine when the correct angle was achieved and a great deal of hand grinding was required. Eliminating gaps between the main tube lengths and the housings was hard to achieve. Figure 62 below shows the process of hand fitting and grinding the control arm for each ball joint housing.



Figure 62: Hand Grinding for Fit

Tacking of the tube segments on the lower control arms for attachment to the frame was accomplished using a piece of pipe to align the segment on each side of the control arm. The rod end housings were aligned on the upper control arms using a straight edge and a flat surface to ensure parallelism.

The final step in fabricating the control arms was to create shock mounts for the lower control arms. A length of tubing, indicated in Figure 63, was used to support the force and was CNC machined from .065 inch tubing.



Figure 63: Shock Mount Support Tube

The rest was composed of 1/8" and 1/16" sheet metal. Each piece was cut using the break in Washburn Shops. Clamps and squares were used to fixture each piece for tacking and then final welding. The hole for the shock mount was drilled after final welding to ensure the hole in each side was aligned properly. The final shock mount can be seen in Figure 64 below.



Figure 64: Shock Mount

With the welding complete, the control arms were sent out to Bodycote in Worcester for heat-treating. Details of the heat-treating process were discussed at the end of the Finite Element Analysis section. The specifics of the welding process will be discussed in the section below.

Welding

The control arms were fabricated out of 4130 steel and TIG welded with the Miller Syncrowave 250 in the Washburn weld shop. A 3/32" 2% ceriated tungsten electrode (grey) was used in the torch, sharpened to a point. Alternatively, a 2% thoriated electrode (red) could be used, but the ceriated electrode will provide better arc control. ER70S-2 1/16" filler rod was used for the 0.065" tubing and 0.045" filler rod used for the 0.035" tubing. A #6 cup was used on the torch. To make the welding process easier it is essential to get the fits as tight as possible at all joints to minimize gaps. Prep work is also very important, clean all joints and filler rod immediately before welding to remove contaminants.



Figure 65: Welder Settings

The following settings were used on the welder:

- DC electrode negative
- Pure Argon cover gas flowing around 15 CFH
- Pre- flow time: 0.5 seconds
- Post- flow time: 12 seconds
- AC Balance: 2
- Crater: off
- Arc Control: off
- Amperage Control: remote
- Output/ Contactor: remote
- High Frequency Start
- Amperage: around 125 maximum for the tubing

The control arms were welded by first tacking all the joints together and checking the geometry on the printed out weld templates. The final welding was done in a number of passes on each end, alternating the side of the joint and working on the different joints all at the same time to minimize the amount of heat put into one spot to minimize warping or distorting the shape of the control arms. This figure shows one of the control arm weld joints on the ball joint end.



Figure 66: Ball Joint End



Figure 67: Lower Control Arms After Welding

Knuckle

The knuckle was the most complex part fabricated by the team for this project. Both CNC machining and TIG welding were required. The spindle tube for each knuckle was modified and

re-used from the original design and the rest of the components were new. 4130 steel sheet metal was used for each piece. Both 1/4" and 1/8" thick sheet were used. Unfortunately, these thicknesses were too thick to be cut on the break and too thin to be cut on the band saw. For this reason, the pieces were rough cut with a saws-all and then finish machined. The processes for machining the more complex components are explained in the next section.

Machined Components

Machining for the knuckle was completed in the Washburn Shops on the Haas Mini-mills and the SL-10 lathe. Facing of edges for fit was done manually on the Mini-mills however some of the operations required programs in ESPRIT.

The main plate of the knuckle required a bore to be machined to accept the spindle housing. This bore had to be at exactly an eight-degree angle in order to produce the desired kingpin angle. In order to accomplish this, a sine table was used in conjunction with a standard vice. Once a sine table is set and clamped, it will not move. This ensured that both knuckles would have exactly the same angle. Figure 68 shows the tool path for the pocketing operation. A half-inch four-flute end mill was used to mill the hole. Feed rates were set to 400 surface feet per min and .0027 inches per tooth.

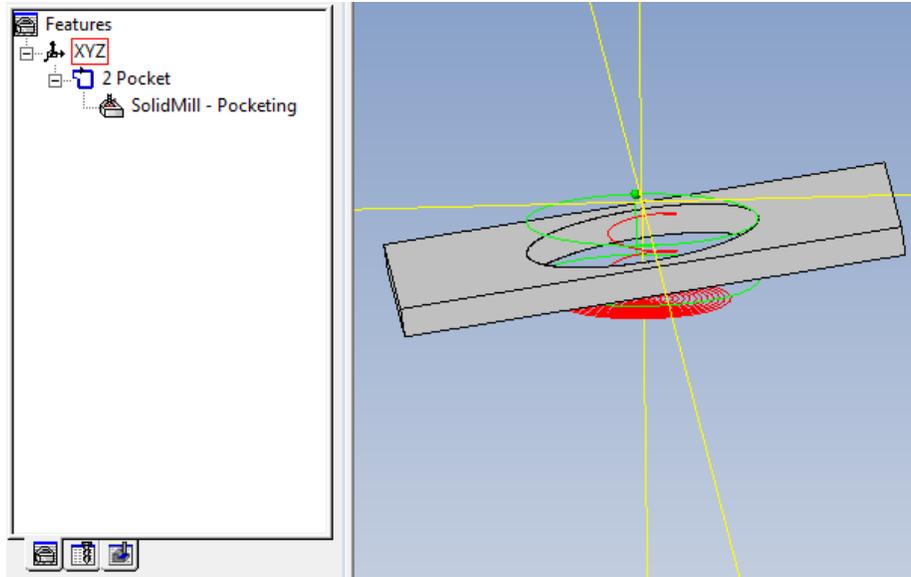


Figure 68: Spindle Housing Bore

In order to re-use the original spindle housings, the housing needed to be shortened by half an inch. Ideally, the proper way to machine these would be to face and bore the parts in a lathe. This was not possible because the OD (outside diameter) surface was too rough for the part to spin on center when held in a chuck. Instead, soft jaws were made in the Mini-mill to clamp the parts. The parts were then milled using the same half-inch end mill and the tool paths in Figure 69.

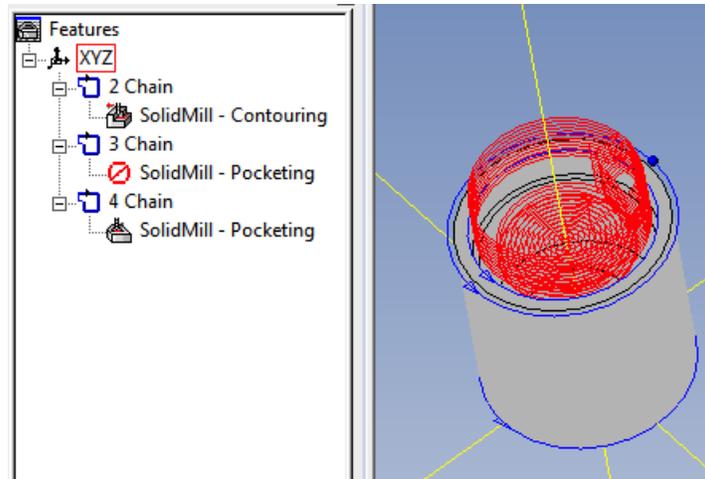


Figure 69: Spindle Housing Re-Work

The front caliper mounts required the most precision of all the knuckle components. The distance between the two mounting holes as well as their position in relation to the spindle housing was vital to ensuring proper rotor position. This component required profile machining around the entire outside edge. To accomplish this, two separate operations were needed. Figure 70 shows the trochoidal pocketing operation for one side of the part.

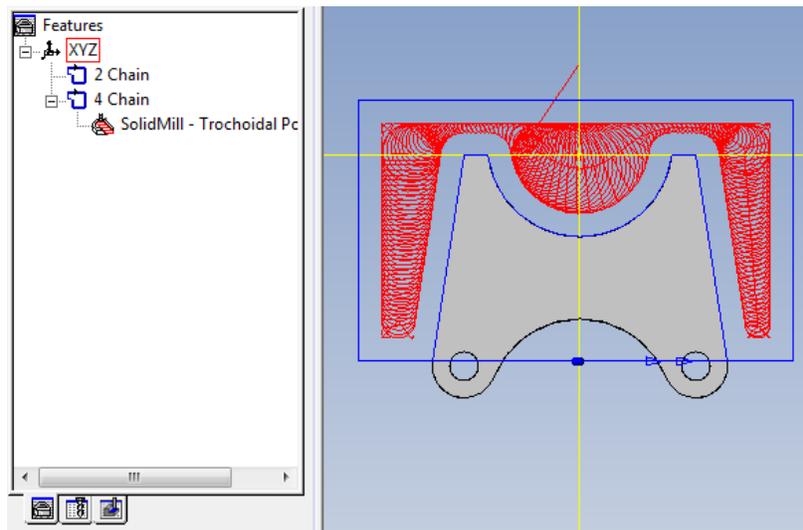


Figure 70: Front Caliper Mount Profiling

The other half of the profiling was performed in the same manner except with the part clamped from the other side. The two holes were not drilled until after the caliper mount was welded to the spindle tube. In order to ensure the two holes were located properly with respect to the centerline of the spindle housing, the assembly was clamped in soft jaws and the part origin was probed and set to the inner bore of the spindle housing.

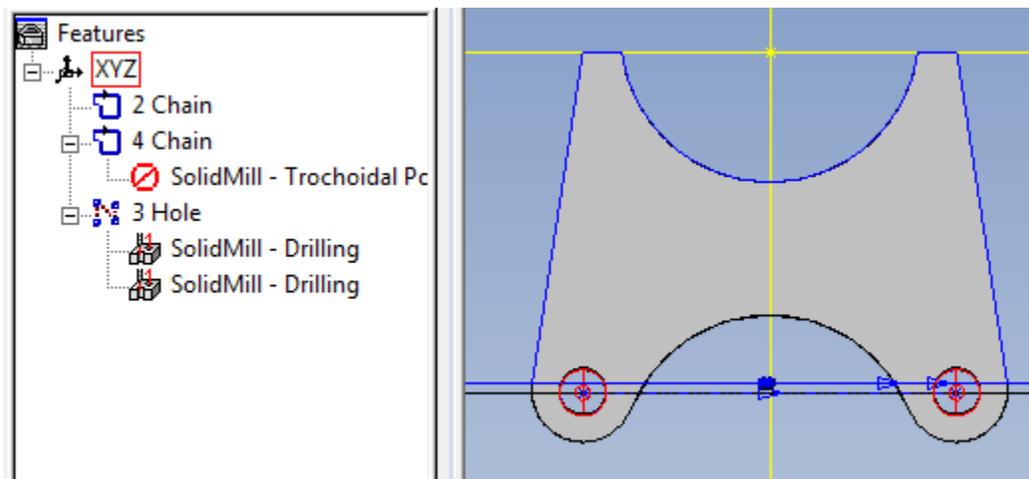


Figure 71: Front Caliper Hole Drilling

With the caliper mounts complete, the only parts remaining to be cut for the knuckle were the triangular gussets. These were rough cut with a saws-all as well and then finish machined to achieve a clean edge.

Welding

The same welder settings were used to weld the knuckle as previously described in the control arms welding section. The only differences are turning up the amperage to 150 and always using the 1/16" filler rod. The first step was to weld the caliper mounts to the spindle tubes.



Figure 72: Caliper Mounts Welded to Spindle Tubes

Next the body of the knuckle was welded together, including the tie rod mount.



Figure 73: Body of Knuckle Welded Together

Finally, the spindle tubes were welded into the body of the knuckle with appropriate gussets to finish the part.



Figure 74: Finished Knuckle

Hub and Spindle Re-work

The front hubs were the last major component that required machining before the front suspension could be assembled. The original hubs were used however the bores for the bearings had to be opened up to allow for new slightly larger bearings. This was more challenging than it would appear because the bore had to remain perfectly concentric to the bolt pattern for the brake rotor as well as the bolt pattern for the wheel itself. If the location was wrong, the wheel and rotor could spin off center causing problems with rotor/pad interaction and wheel wobble. To further complicate the problem, the bore for both bearings could not be milled in the same operation due to a smaller counter bore in the center of the hub. One part would have to be milled and then the part flipped over to finish the opposite side. In order to ensure the bore was perpendicular to the face of the rotor, special blocks were milled in the machine and then the hub

was clamped on these. The part was clamped on the face that the wheel bolts to; this meant the new bore would also be perpendicular to this surface. As shown in Figure 75, four clamps were used to hold the hub solid during the machining process.



Figure 75: Hub Fixturing

During the milling of the first bore, a light skim pass was also taken on the central counter-bore. This provided a feature to probe when the part is flipped over so that both bearings are perfectly concentric.

Another important factor to take into consideration when machining a bore for press fit is tool flex. An interference fit for a two inch bearing in steel is only 0.0007 inches. If there is excessive tool flexing, the bore may be tapered and the bearing will not be held evenly. Generally, shorter tools are used when flexing is a concern however, due to the skim pass on the counter-bore; a cutting length of 1.5 inches was required. In order to minimize tool flexing, a

spiral milling operation was used which would allow much of the cutting to be done by the bottom of the end mill. This meant that much of the cutting force would be along the axis of the tool rather than perpendicular to it. This spiral tool path, shown in Figure 76, helped reduce taper and provided a consistent bore size for multiple cuts on each hub.

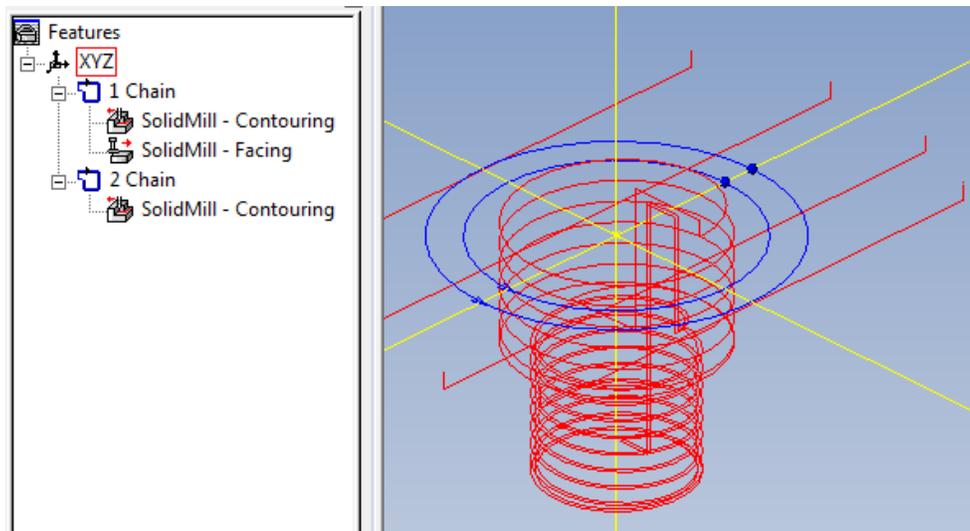


Figure 76: Front Hub Re-Work

Each bore was milled using a rough cutting pass and a spring pass, both with identical tool paths. A face cut was also made on the hub where the rotor locates. The only reason for this was to confirm that the rotor would be perpendicular to the bore. With the hubs complete, a bearing was pressed into each bore using an arbor press. It is very important that the proper interference fit be achieved. If the bore is too small, not only will it be very difficult to press the bearing in, but also excessive stresses on the bearing could lead to failure.

Testing

Once the new front suspension was completely assembled, testing was performed to evaluate the performance and determine if the design goals had been met. The two major design

goals were to minimize camber during steer and to reduce the camber gains throughout suspension travel.

Before the results of testing are discussed, a brief overview of the fine-tuning process for the suspension is prudent. Due to the location of the rack and pinion, it is crucial that the front suspension remains at the intended natural ride height. The rack is positioned about 4 inches above the centerline of the front spindles. Because the rack is not at the instant center of rotation of the knuckle, there is toe-out during suspension droop and toe-in during compression. Maintaining the correct ride height minimizes this toe effect. In order to achieve desired ride height, the air pressure in the front shocks were adjusted through trial and error until the vehicle tended to remain at ride height during most maneuvers and on different terrains. Once this height was achieved, the tie rods were then adjusted to provide slight toe-in at ride height. This procedure will likely have to be repeated before any future testing, as the shocks will lose pressure over time. A good pressure to start with for the shocks is about 30-40 psi in the main chamber and 80-90 psi in the “evol” chamber. With the suspension riding correctly, the car could now be evaluated based on stated goals.

First, some static performance specs were compared to the original design. Figure 77 shows the difference in positive camber gain during droop for the original and the new front suspension system. Positive camber was practically eliminated, only a few degrees remained. This helped to reduce the amount of scrub and will also improve stability and reduce stresses during the completion of landings. Although there is no qualitative data for ride smoothness, during testing the new design absorbed bumps much better than the original. This is partly due to the reduced camber during suspension travel and also the camber change during turning.



Figure 77: Difference in Droop Camber

Perhaps the most significant problem with the original design, was near 30 degree positive camber gain on the inside wheel during turning. Figure 78 shows the left wheel of the vehicle during a full lock left turn. There is a clear difference between the before and after pictures of the new car. The positive camber has been reduced to less than 5 degrees at full lock. This change greatly improved the turning performance of the new car and helped to reduce the magnitude of moments on the spindle during landing.



Figure 78: Difference in Turning Camber

The car felt much more responsive with these changes. Steering was very crisp and easy to predict. Unlike the original design, the new system did not suffer from constant under steer, which greatly reduces the confidence of the driver and their ability to compete. Before, high-speed turning maneuvers were very difficult because the wheels would turn and tilt and the car would continue going straight. The car is much more agile now that the camber gains have been eliminated.

Overall Results and Testing

A number of tests were set up to evaluate the overall performance of the car. Tests on the individual subassemblies were described previously in their respective sections of the report. These tests use the original state of the car as a benchmark for evaluation after improvements were made.

- Test: 100 steps, straight line, wet grass, rolling start
 - Original Time: 12.4 sec, 12.1 sec
 - Improved Time: 11.9 sec, 11.6 sec
- Test: 100 steps, straight line, pavement, dead start
 - Original Time: 10.4 sec, 9.9 sec
 - Improved Time: 10.1 sec, 9.7 sec
- Test: Short trail loop in Brimfield
 - Original Time: 48 sec, 46 sec
 - Improved Time: 43 sec, 42 sec

These improved times can be attributed to better tuning the drivetrain, the car riding and handling better on the new suspension, and the car being lighter.

Conclusion and Recommendations

The upgrades made to the Baja vehicle improve performance, manufacturability, and strength of the drivetrain, front suspension, steering, and braking system. The center of gravity was reduced by lowering the drive train, and the overall weight of the vehicle was reduced to make best use of the available power from the engine. Through analysis and iteration, the knuckle was made lighter and much simpler to manufacture along with the control arms to correct steering geometry. Stopping power and reliability of the braking system were increased significantly by upgrading from three brake calipers to four and increasing rotor diameter.

It is recommended that this car is taken to competition the way it is to evaluate performance in a competitive environment. This is the best way to truly evaluate all the components of the car and determine what parts still need improvement before another redesign is undertaken.

The car can only be improved to a certain degree while still using the original frame. Another iteration of this vehicle would benefit from a major redesign of all subsystems, including a new frame to accommodate the new design. This MQP was designed around the original frame but some components could be better designed with a new frame. For example, the front suspension was designed around the limiting factor of the mount points provided on the frame, but an even better suspension could be designed if the mount points could be moved. The drivetrain could also be improved if it was better integrated into the frame itself, which could eliminate parts and make the car lighter. The large diameter dirt bike tires provide good ground clearance, but should be evaluated in competition to determine if they truly provide an advantage, or if a more traditional mini Baja tire should be used.

Works Cited

Baja SAE Rules. SAE International. 2013 Collegiate Design Series. <students.sae.org>.

Milliken, William F., and Douglas L. Milliken. *Race Car Vehicle Dynamics*. Warrendale, PA, U.S.A.: SAE International, 1995. Print.

Norton, Robert L. *Machine Design: An Integrated Approach*. Boston: Prentice Hall, 2011. Print.

Racing Car Technology. Web. 11 Dec 2012. <<http://www.smithees-racetech.com.au/>>.

Smith, Neville, and Dale Thompson. "Camber, Caster and Toe." *Race Car Suspension Set-Up*.

Appendix

Link Lengths:

$$a := 20.8 \text{ in} \quad \underline{L} := 65 \text{ deg} \quad x := \text{atan}\left(\frac{3.88}{3.45}\right) = 48.357 \text{ deg}$$

$$b := 8 \text{ in}$$

$$\underline{c} := 15.25 \text{ in} \quad \text{will be varied}$$

$$d := 4.92 \text{ in}$$

Link Angles:

$$\theta_2 := 35 \text{ deg, 1...L} \quad \text{must be determined}$$

Link Ratios:

$$K_1 := \frac{d}{a} = 0.237$$

$$K_2 := \frac{d}{c} = 0.323$$

$$K_3 := \frac{a^2 - b^2 + c^2 + d^2}{2 \cdot a \cdot c} = 0.986$$

$$K_4 := \frac{d}{b} = 0.615$$

$$K_5 := \frac{c^2 - d^2 - a^2 - b^2}{2 \cdot a \cdot b} = -0.866$$

Intermediate Parameters:

$$\underline{A}(\theta_2) := \cos(\theta_2) - K_1 - K_2 \cos(\theta_2) + K_3$$

$$B(\theta_2) := -2 \sin(\theta_2)$$

$$\underline{C}(\theta_2) := K_1 - (K_2 + 1) \cos(\theta_2) + K_3$$

$$D(\theta_2) := \cos(\theta_2) - K_1 + K_4 \cos(\theta_2) + K_5$$

$$E(\theta_2) := -2 \sin(\theta_2)$$

$$\underline{F}(\theta_2) := K_1 + (K_4 - 1) \cos(\theta_2) + K_5$$

Determine θ_3 and θ_4 :

$$\theta_4(\theta_2) := 2 \cdot \operatorname{atan}\left(\frac{-B(\theta_2) - \sqrt{B(\theta_2)^2 - 4 \cdot A(\theta_2) \cdot C(\theta_2)}}{2 \cdot A(\theta_2)}\right)$$

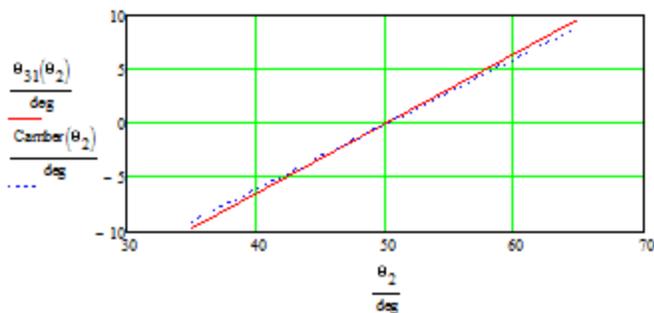
$\text{King}_p := 10 \text{ deg}$ does 10 degrees need
to be adjusted for rake
angle?

$$\theta_3(\theta_2) := 2 \cdot \operatorname{atan}\left(\frac{-E(\theta_2) + \sqrt{E(\theta_2)^2 - 4 \cdot D(\theta_2) \cdot F(\theta_2)}}{2 \cdot D(\theta_2)}\right)$$

$$\theta_{31}(\theta_2) := 2 \cdot \operatorname{atan}\left(\frac{-E(\theta_2) - \sqrt{E(\theta_2)^2 - 4 \cdot D(\theta_2) \cdot F(\theta_2)}}{2 \cdot D(\theta_2)}\right) + 52 \text{ deg} + \text{King}_p$$

$\text{Camber}(\theta_2) := \theta_{31}(\theta_2) \cdot \cos(20 \text{ deg})$ multiply by cosine of rake angle

$$\theta_{31}(50 \text{ deg}) = 1.167 \text{ deg} \quad \theta_3(50.6) = 136.321 \text{ deg} \quad \text{Camber}(47.01 \text{ deg}) = -0.742 \text{ deg}$$



At static ride height, θ_2 is approximately 50 degrees. At maximum compression, the angle reduces to about 40 degrees. The angle at maximum droop has not yet been determined because the location of the shock has not been decided.