Suspension Design and Vehicle Dynamics Model Development of the Venturi Buckeye Bullet 3 Electric Land Speed Vehicle

THESIS

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

By

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Abstract

For over a dozen years the Buckeye Bullet Electric Land Speed Racing Team has developed electric land speed vehicles that have set numerous national and international land speed records over 300 miles per hour. These vehicles have been powered by various battery chemistries as well as hydrogen fuel cells. The vehicles have been platforms for advanced vehicle research and training for future engineers. Taking such advanced vehicles to tremendous speeds has its difficulties associated with it. There are two related challenges that are covered in this document. The first challenge is the design, analysis and implementation of a custom fully-independent suspension for the Venturi Buckeye Bullet 3 which is targeting to set an international land speed record over 400 miles per hour. Many of the considerations that went into the design and analysis will be presented. The second challenge addressed is the development of a vehicle dynamics model of the land speed vehicle. There is a significant emphasis on the development of the models and obtaining the necessary vehicle parameters required to build a vehicle dynamics model. The model is developed to begin to quantify the vehicle performance, handling and stability. In the end an analysis of the vehicle stability in crosswinds is considered. The results of the work will be used to validate the vehicles performance and identify any areas of improvement.
Dedication

This document is dedicated to my family for their patience, support and encouragement.
Acknowledgments

I would like to first thank the Buckeye Bullet Electric Land Speed Racing Team members for their dedication, passion and tireless efforts. These vehicles and world records would not be possible without the countless hours, late nights and passion of the team. In addition, I would like to thank my family and girlfriend for their continual support through my education and involvement with the Buckeye Bullet. Their unwavering support and encouragement has allowed me to accomplish the challenges that have been presented to me.

The Venturi Buckeye Bullet 3 would also not be possible without several key important sponsors. The support from Venturi Automobiles, A123 Systems, The Timken Company, Cooper Tire & Rubber Company and MathWorlks along with many more sponsors have made this project possible. All the partners have contributed to my education and development to make me a better engineer.

Finally I would like to thank Giorgio Rizzoni for your support of my research. I would especially like to thank Jeff Chrstos for his continual support and advice through every step of the development of the vehicle as well as my research work.
Vita

January 8, 1989 .................................................. Born – Dayton, OH

2007 ................................................................. Graduate, Thomas Worthington HS

2007-2013 .......................................................... B.S. Mechanical Engineering, The Ohio State University

2007-2013 .......................................................... Buckeye Bullet Team Member

2013-2015 .......................................................... Buckeye Bullet Mechanical Team Leader

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2015-Present ...................................................... Buckeye Bullet Team Leader

Fields of Study

Major Field: Mechanical Engineering
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Nomenclature

\( a \)  
Linear Radius Due to Velocity Coefficient (s)

\( b \)  
Linear Stiffness Due to Velocity Coefficient (Ns/m²)

\( D_x \)  
Linear Deceleration (m/s²)

\( f_n \)  
Ride Frequency (Hz)

\( F_T \)  
Vertical Tire Force (N)

\( F_T^* \)  
Modified Vertical Tire Force (N)

\( F_{xmf} \)  
Maximum Braking Force for the Font Axle (N)

\( g \)  
Acceleration due to gravity (m/s²)

\( h \)  
Center of Gravity Height (m)

\( IR \)  
Installation Ratio (-)

\( K_{RR} \)  
Ride Rate (N/m)

\( K_s \)  
Spring Rate (N/m)

\( K_T \)  
Tire Stiffness (N/m)

\( K_{T0} \)  
Calculated Tire Stiffness at V=0 m/s (N/m)

\( K_{T200} \)  
Calculated Tire Stiffness at V=90 m/s (N/m)

\( K_T^* \)  
Modified Tire Stiffness (N/m)

\( K_{wr} \)  
Wheel Rate (N/m)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_{\phi_{ARB}}$</td>
<td>Roll Rate Contribution by Axle Anti-Roll Bar (Nm/deg.)</td>
</tr>
<tr>
<td>$K_{\phi_{DES}}$</td>
<td>Desired Roll Rate (Nm/m)</td>
</tr>
<tr>
<td>$K_{\phi_{F}}$</td>
<td>Front Roll Rate (Nm/deg.)</td>
</tr>
<tr>
<td>$K_{\phi_{R}}$</td>
<td>Rear Roll Rate (Nm/deg.)</td>
</tr>
<tr>
<td>$K_{\phi_{TARB}}$</td>
<td>Total Roll Rate Contribution by Anti-Roll Bar (Nm/deg.)</td>
</tr>
<tr>
<td>$l$</td>
<td>Length of Anti-Roll Bar Arm (m)</td>
</tr>
<tr>
<td>$L$</td>
<td>Wheelbase (m)</td>
</tr>
<tr>
<td>$M$</td>
<td>Mass of the Vehicle (kg)</td>
</tr>
<tr>
<td>$MR$</td>
<td>Motion Ratio (-)</td>
</tr>
<tr>
<td>$R$</td>
<td>Radius of Turn (m)</td>
</tr>
<tr>
<td>$R_0$</td>
<td>Unloaded Tire Radius (m)</td>
</tr>
<tr>
<td>$R_{0v_0}$</td>
<td>Calculated Unloaded Radius at V=0 m/s (m)</td>
</tr>
<tr>
<td>$R_{0v_{200}}$</td>
<td>Calculated Unloaded Radius at V=90 m/s (m)</td>
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<tr>
<td>$R_0^*$</td>
<td>Modified Unloaded Tire Radius (m)</td>
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<tr>
<td>$R_L$</td>
<td>Loaded Tire Radius (m)</td>
</tr>
<tr>
<td>$R_L^*$</td>
<td>Modified Loaded Tire Radius (m)</td>
</tr>
<tr>
<td>$t$</td>
<td>Track (m)</td>
</tr>
<tr>
<td>$V$</td>
<td>Vehicle Forward Velocity (m/s)</td>
</tr>
<tr>
<td>$V_T$</td>
<td>Tire Velocity (m/s)</td>
</tr>
<tr>
<td>$W$</td>
<td>Weight of the Vehicle (N)</td>
</tr>
<tr>
<td>$W_{fs}$</td>
<td>Front Axle Static Load (N)</td>
</tr>
</tbody>
</table>
\[ \delta \quad \text{Steering Angle (deg.)} \]

\[ \mu_p \quad \text{Peak Coefficient of Friction (-)} \]

\[ \frac{\phi_r}{A_y} \quad \text{Roll Gradient (deg./g)} \]
Chapter 1: Introduction

1.1. Land Speed Racing

For over a century, racers, gear heads, and racing enthusiasts have flocked to the Bonneville Salt Flats in Utah to see how fast they can go. This is a very unique region that consists of nearly 40 square miles of salt that is traditionally densely packed. Due to the pure size, surface hardness and flatness, the Bonneville Salt Flats has been a perfect location to run vehicles at high speeds. Even with the size of the Salt Flats the land speed racing tracks are traditionally 12 miles in length due to geological restrictions and quality of the salt. Over the past few years with changes in weather patterns the salt conditions have degraded causing racecourses to be limited in length and surface quality.
Every year the Bonneville Salt Flats have racecourses for national and international records. There are two governing bodies that oversee the national and international records respectively, The Southern California Timing Association (SCTA) and the Federation Internationale de l’Automobile (FIA). Many may think that the differences may be minor between the two governing bodies but their rules significantly impacted the design goals of the Venturi Buckeye Bullet 3. The rules that define how a record is established differ greatly.

The types of world records that are typically competed for in land speed racing are “flying mile” and “flying kilometer”. The rules established for this style of record apply across the world. For the flying mile and kilometer the records are measured over a mile or kilometer respectively with no limitations on track length. The vehicle can have as
long of an acceleration period as is wanted but the vehicle has to make two runs in opposing directions through the timed distance in an hour or less. Running a vehicle in opposing directions is done to account for grade and wind effects. The record speed is the average speed of the two timed distances.

A United States national record with the SCTA has a very different set of rules to receive a record. The racecourse is set at seven miles in length with the third, fourth and fifth miles being timed. If the vehicle qualifies for a record it is impounded for the rest of the
day and the team is allowed to service the vehicle for four hours. At the beginning of the next morning the vehicles are run in the same direction as the initial record attempt.

Since the Bonneville Salt Flats are so flat and vehicles are not allowed to run with any appreciable wind, the record can be established running the vehicle in the same direction. The two timed distances are then averaged and if the average is greater than the previous record it is a new national record.

Figure 1.3: SCTA Certified US Record Description
In future discussions a “run” is defined as one high speed run through the timed mile or kilometer. This also means that it would be one half of a national or international record.

1.2. History of the Buckeye Bullet Program

Electric racing has been going on at The Ohio State University for over 20 years which started with the Smokin’ Buckeye open-wheel racer. After the success of the Smokin’ Buckeye the team looked for greater challenges which lead them to try and create the fastest electric cars. The Buckeye Bullets have been a series of electric vehicles that have been targeting national and international records for battery and fuel cell vehicles. This ambition has led the team to numerous records over 300 miles per hour and has pushed them to target 400 miles per hour with the latest vehicle generation.

1.2.1. Smokin’ Buckeye

The Smokin’ Buckeye was the first modern electric race vehicle that The Ohio State University built. It competed in an intercollegiate open-wheel racing series called Formula Lightning [1]. The Ohio State University joined the series in 1993 with the Smokin’ Buckeye and won all three national championships that were offered. This vehicle used a spec chassis with 32 lead acid batteries powering an AC induction machine [2]. When the batteries would be sufficiently depleted the vehicle would return to the pit area and the team would swap all the batteries.
1.2.2. Buckeye Bullet 1 (BB1)

After the Formula Lightning racing series dissolved, the team was at a point where they had many talented engineering students that wanted to continue their electric racing experience. After deliberating with faculty, staff and team sponsors the team decided to set their sights on land speed racing. With that decision, the concept of the Buckeye Bullet 1 was created. The Buckeye Bullet 1 was an electric land speed streamliner that was battery power [3]. In its final configuration the vehicle was powered by a nickel-metal hydride (NiMH) battery pack using cells from the Toyota Prius with a custom AC induction machine powering the rear axle. The BB1 also weighed nearly 3,800 pounds. The vehicle raced from 2001 to 2004 when it reached a top speed of 321.834 miles per hour and set a US national record at 314.958 miles per hour. The vehicle was originally designed for 300 miles per hour and had reached its physical limits.
The team had again hit a point where they had to make a decision of what to do next. At the time lithium-ion batteries were still not proven so the team looked at what energy technologies were being heavily researched and the technology that was drawing a lot of attention were hydrogen fuel cells. This new challenge was taken on by the team and named the Buckeye Bullet 2. The team developed partnerships with Ford Motor Company, Ballard Power Systems and Roush Industries to develop the new electric streamliner. The vehicle was powered by two proton exchange membrane (PEM) fuel cell stacks producing 300 kilowatts each. The fuel cells powered the same custom inverter and AC induction machine from the BB1 that powered the front wheels. Due to the numerous components needed in the car to support the fuel cell systems it weighed 5,800 pounds. The vehicle raced from 2007 to 2009 and set an FIA Category A Group XIV Class 6 world record at 302.877 miles per hour in the last year of running.
1.2.4. Venturi Buckeye Bullet 2.5 (VBB2.5)

After pushing the fuel cell systems to their limits and only incremental improvements possible for the Buckeye Bullet 2, the team looked again at the emerging technologies. In the time that it took to develop and race the BB2, lithium ion batteries had made significant improvements in power and safety. To better prepare the team for the development of a new vehicle with much greater speed targets, the BB2 was retrofitted with lithium-ion batteries in place of the fuel cell systems and rebadged the car as the Venturi Buckeye Bullet 2.5. The batteries chosen for the vehicle were A123 System’s 32113 lithium iron-nanophosphate batteries. Other systems such as the controller, data acquisition and clutch were all changed to test the feasibility for the next land speed vehicle. With the change in power source the vehicle significantly reduced weight to 4,300 pounds. The vehicle ran in 2010 and set a FIA world record for Category A Group VIII Class 4 at 307.666 miles per hour.
1.2.5. Venturi Buckeye Bullet 3 (VBB3)

The VBB2.5 was designed to be a test bed for future technologies that would be integrated into the Venturi Buckeye Bullet 3. The next step for the team was to try and take a vehicle to even higher speeds with the latest battery and powertrain technologies available at the time. The goal of the VBB3 is to set a FIA certified world record over 400 miles per hour. To reach the target speed, 2,000 prismatic A123 lithium iron nanophosphate cells power custom inverters and two internal permanent magnet motors per axle. The vehicle is four-wheel drive with nearly all the systems similar front and rear just mirrored about the driver. The vehicle is capable of producing over 1.5 megawatts of power. Nearly every component was customized for the vehicle due to the unique application and goals. The vehicle was unveiled in 2013 and made its first runs on the Bonneville Salt Flats in 2014. Most recently the vehicle raced at the Salt Flats in August 2015 and was able to reach a top recorded speed of 288 miles per hour while setting a FIA certified record for Category A Group VIII Class 8 at 240.320 miles per hour.
Figure 1.8: Venturi Buckeye Bullet 3 (Photo Denis Boussard)
<table>
<thead>
<tr>
<th></th>
<th>BB1</th>
<th>BB2</th>
<th>BB2.5</th>
<th>BB3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>period</strong></td>
<td>2001-2004</td>
<td>2006-2009</td>
<td>2010</td>
<td>2011-current</td>
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<tr>
<td><strong>powertrain</strong></td>
<td>2WD</td>
<td>2WD</td>
<td>2WD</td>
<td>4WD</td>
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<tr>
<td><strong>vehicle power</strong></td>
<td>400 kW</td>
<td>425 kW</td>
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<td>200 kW (overload x 1.885)</td>
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<td><strong>motor maximum speed</strong></td>
<td>10500 rpm</td>
<td>10500 rpm</td>
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<tr>
<td><strong>transmission</strong></td>
<td>6 speed semi-automated</td>
<td>2 speed semi-automated</td>
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<td></td>
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<tr>
<td><strong>clutch</strong></td>
<td>hydraulic</td>
<td>mechanical overrunning</td>
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<tr>
<td><strong>ESS technology</strong></td>
<td>NiMH batteries</td>
<td>hydrogen fuel cell (PEM)</td>
<td>lithium-iron phosphate</td>
<td>lithium-iron phosphate</td>
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<td>20 kWh - 450kW</td>
<td>600 kW</td>
<td>17.8kWh - 650 kW</td>
<td>92.4 kWh - 2.2 MW</td>
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<td>0.82 m²</td>
<td>0.85 m²</td>
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<td>2600 kg</td>
<td>1950 kg</td>
<td>3600 kg</td>
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<tr>
<td><strong>length</strong></td>
<td>9.5 m</td>
<td>10.7 m</td>
<td>11.2 m</td>
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</tr>
</tbody>
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1.3. Research Motivation and Organization

Currently in today’s climate with battery electric vehicles starting to become available, there have been certain stigmas that are attached with them. One of the most prominent stigmas is that they lack power and performance. The previous Buckeye Bullets have proven that electric vehicles can be fast but they have never been competitive with the internal combustion engine counterparts. Currently, some of the fast internal combustion engine streamliners in the world are running over 400 miles per hour which is why the team has taken on such an ambitious goal of setting a world record with an electric land speed vehicle over 400 miles per hour. The Buckeye Bullet Racing Team was the first teams to have an electric car exceed 300 miles per hour now the new goal is to be the first team to exceed 400 miles per hour and compete with the internal combustion engine counterparts.

Making the jump from 300 miles per hour to 400 miles per hour is no small feat. It requires a significant amount of design, analysis and ingenuity to make a vehicle go 400 miles per hour let alone with such complex and unique systems. Every system of the vehicle must be designed for the sole purpose of making the vehicle reach its targeted speed goals. This requires that every system and component is heavily analyzed and scrutinized to insure that it is doing its part to reach that goal. The content of this document looks at the design and analysis of a completely custom suspension for the VBB3. It also looks into the development of a suspension and vehicle dynamics model that is used to insure that the vehicle performs as intended with the correct ride, handling and stability.
The suspension design section is an overview of all the suspension component goals, designs and analysis. This details all the major considerations that went into the design of the VBB3 suspension components and why the suspension was designed the way it was. This section is also meant to act as a knowledge book for future team members of the Buckeye Bullet and the development of future suspension designs.

The vehicle dynamics modeling section gives a detailed look into the development of a kinematic model of the VBB3 in SuspensionSim™ as well as the development of a model for the VBB3 in CarSim™ [4] [5]. It discusses the methods of how vehicle parameters were gathered from testing as well as vehicle run data. The developed model is then used to consider crosswind stability of the vehicle when traveling down the race track at high speeds.

While these two parts are two distinctly different topics they are greatly tied to each other. The performance of the vehicle, especially the handling and stability are greatly affected by the design of the suspension.
Chapter 2: Background

2.1. History of Buckeye Bullet Suspension Designs

The Buckeye Bullet land speed vehicle suspensions have been an ever-evolving system with increased complexity and needs. The process has required increased understanding through every step. The BB1 was one of the more simplistic designs because of the lack of experience, finance and manufacturing ability. The BB2 was a substantial step forward in complexity because of the budget and manufacturing ability. The VBB3 has used the ten plus years of land speed racing experience and lessons learned in combination with modern manufacturing to produce the most advance suspension for any Buckeye Bullet land speed car.

2.1.1. Buckeye Bullet 1

The BB1 was a step in a new direction compared to the Smokin’ Buckeye suspension. There were no guidelines of how to design a suspension for a land speed car. The goal of the suspension was to be able keep maximum traction which led the team to an independent suspension. Both the front and rear suspensions were fully independent. The front track width was approximately 482.6 millimeters while the rear track was 711.2 millimeters. This made the packaging extremely difficult for the front suspension especially with the steering system.
The wheel hubs for the vehicle were kept as compact as possible and integrated a double row of bearings to support the spindle. In the rear the spindle was connected to off-the-shelf CV joints that transferred the torque of the rear differential to the wheels. The front and rear hubs differed mainly in the design of the attachment between the hub and the A-arm.

The A-arms were kept as simple as possible to reduce the need for complex manufacturing. Most of the designs were 2-D parts that allowed for most of the parts to be manufactured in house. The A-arms utilized rod ends to allow for adjustability in the suspension kinematics.

The sprung mass was supported by a pushrod system for the front and rear axles. The pushrod attached to the lower A-arm in the front while the rear pushrods attached to the wheel hub. The pushrods for the front and rear both connected to rockers that then transfer the loads to the coil-over shocks.
The vehicle originally used Goodyear Eagle tires for both the front and rear axles. When
the vehicle was beginning to reach higher speeds approaching 300 mph, the tires began to
show significant degradation on the driven tires. To alleviate this, the team switched to
the Mickey Thompson Bonneville tire on the rear axle while keeping the Goodyear Eagle
LSR tires on the front.

2.1.2. Buckeye Bullet 2

The BB2 took many lessons learned from the BB1 into consideration for the design of the
new suspension. One of the biggest improvements was with the manufacturing
capabilities that allowed from more complex and efficient packaging of all the suspension
components. The design also incorporated a significant amount of flexibility in
suspension kinematics and interchangeability between the various corners of the vehicle.
Many of the components on the suspension could be swapped with a component from
another corner.

The front and rear suspensions were fully independent suspensions that used a
nonconventional pull rod system that attached to the spring and damper through a rocker.
Both the rocker and the springs could be packaged under the 6-speed transmission in the
front while they there packaged under high-pressure hydrogen and Heliox lines in the
rear. The suspension also implemented a parallel and equal length double wishbone
system. This was done to help prevent bump steer and changes in camber, which could
potentially have negative effects to the needed performance of the suspension.
The A-arms were custom machined components due to the packaging constrained required for the braking system. They were made from 6061 aluminum billets and included the use of spherical bearings and rod ends to meet the required attachment needs as well as adjustability. The downside was that the rod ends were the weakest points of the A-arm designs.

The wheel hub was also a custom design that incorporated a custom spindle that had a CV joint bolting to the spindle. The spindle is supported by two radial ball bearings that are retained by a bearing nut and lock washer. The spindle and bearing assemblies are housed in a custom machined aluminum housing that connects the spindle assembly to the upper and lower ball joints.

The rockers were attached to the lower A-arm in a nonconventional location, the bottom of the lower A-arm. This was done due to the tight packaging space. The rockers were made from 6061 aluminum and integrated two radial ball bearings at the pivot point to reduce drag.

The rockers were attached to the dampers that were modified to fit in the available packaging space as well as customized for the specific damping characteristics required by the team. The dampers were something that the team did not have a significant amount of experience with so the team specified a ride frequency and some of the vehicle parameters to the supplier so they could make a suggestion on the damping. The team used dampers that allowed them to re-valve and make adjustments to the damping in both jounce and rebound.
Finally, the BB2 used Mickey Thompson Bonneville tires that were 24.5 inches in diameter. These tires were selected due to their high speed capabilities, load capacity and availability. The Goodyear Eagle LSR tires were considered but due to the issues that were seen with the BB1 and the fact most of the vehicles traveling over 300 miles per hour were using the Mickey Thompson tires.

2.2. History of Buckeye Bullet Vehicle Dynamics Analysis

There has been a significant amount of work that has been put into researching and analyzing earlier vehicles of the team. The BB1 had much more research done on the vehicle dynamics and handling compared to the BB2. This was due to timeline in which the vehicle needed to be manufactured. The BB2 relied on the knowledge gained from the BB1 to make the choices for the BB2.

2.2.1. Buckeye Bullet 1

The BB1 had two theses written about the vehicle dynamics and handling. The first thesis written by Thomas Sopko focused on creating a three-degree of freedom (3DOF) linear model that was modified to account for aerodynamic changes due to the rudder in the tailfin of the vehicle [6]. Sopko further analyzed the response of the vehicle in crosswinds and with the use of the rudder. Frequency response analysis was used to look at the stability of the vehicle in various situations to insure that it would not become unstable throughout a run. To validate the model an ADAMS model was developed for the full vehicle that also accounted for the additional rudder inputs.
The second thesis written on the BB1 vehicle dynamics was by Edward Hillstrom which focused on adding a fourth degree of freedom (4DOF) to Sopkos three degree of freedom model [7]. This additional degree of freedom was added to account for the torsional rigidity of the BB1 chassis. Hillstrom split the vehicle mass into two masses that had a spring and damper connecting them. Significant work was done to analyze the chassis through finite element analysis and validated it with physical testing. The data was used to apply to the four-degree of freedom model as well as an ADAMS model that was used to begin to validate the four-degree of freedom model.
2.2.2. Buckeye Bullet 2

The BB2 was a very different case when considering the analysis of the vehicle dynamics. Due to the short timeline that was required to design and build the vehicle, typical research areas had to be condensed significantly so that the vehicle could be manufactured in time. This meant that there was very little time for vehicle dynamics modeling and analysis. The team had to use the knowledge gained in the development and analysis of the BB1 to then make decisions for the BB2. The design left flexibility in the system to allow for minor modification when further analysis could be conducted. Most of the stability analysis for the BB2 was done in conjunction with the aerodynamic analysis of the vehicle while the kinematics of the suspension were developed in CAD but heavily referenced the BB1 designs.
Chapter 3: Venturi Buckeye Bullet 3 Suspension Design

3.1. Kinematics

The kinematics of a suspension describes the motion of the assembly due to the geometry of all the components. The kinematics affect everything from vehicle handling to ride quality as well as steering feedback. Some of the most common effects considered by the Buckeye Bullet team in the suspension kinematics are bump, roll and torque steer. The kinematic design for the Buckeye Bullet land speed cars has evolved over time and continually had to improve with more complex and powerful vehicles.

3.1.1. Buckeye Bullet 1

The Buckeye Bullet 1 used an independent front and rear suspension. The vehicle was rear-wheel drive, which eliminated the concerns of any form of torque steer. The front suspension was extremely narrow with a track of 482.6 millimeters, which made the motion capabilities more limited. The rear track was much greater at 711.2 millimeters. The goal of the front suspension was to eliminate bump steer so that little or no correction needed to be taken if a bump was encountered at high speeds. The team eliminated bump steer by using steering tie rods that were parallel and equal in length to the upper and lower control A-arms.
3.1.2. Buckeye Bullet 2

The Buckeye Bullet 2 had a fully independent suspension in both front and rear. The vehicle was front wheel drive for stability concerns but this added complexity to the kinematic design since powering through the steering wheels could induce torque steer. Similar to the BB1, the minimization of bump steer was also critical. Even with the Salt Flats being one of the flattest places on earth there are still imperfections. The suspension also needed to have three inches of travel so that there was plenty of suspension travel with ride height adjustments. The vehicle included 12 degrees of steering from lock to lock mainly for maneuverability at the test track and events. With the vehicle being front wheel drive torque steer was a concern so proper precautions had to be taken when designing the steering geometry. The torque steer was eliminated by have zero scrub radius as well as equal length CV joints. Length differences in drive shafts can induce torque steer due to differences in articulation of the joints of the drive shafts as well as torsional differences [8]. Another goal of the design was also to make as many of parts as common as possible so that they could be interchangeable so that less spares had to be made.

3.1.3. Venturi Buckeye Bullet 3

The VBB3 used many of the successful designs from the previous cars but with small changes that improve upon each vehicle’s designs. It was decided that a fully independent suspension was necessary so that the vehicle could attain maximum traction so that it can accelerate as quickly as possible. The track width of the vehicle was set by
the gearbox and chassis limitations at the minimum and the width of the battery compartments set the maximum track width. After working on packaging of the gearbox and chassis, the track width of the vehicle was set at 658 millimeters. This is significantly less than a normal consumer vehicle while still being 104 millimeters narrower than BB2. With the increase in weight and decrease in track width, roll stability could become a concern. To improve the roll stability countermeasures could be added to the vehicle. The major concern of the team was reducing the frontal area of the vehicle in order to reduce the aerodynamic drag as much as possible.

The wheelbase of the vehicle was purely constrained by the packaging space that was required to fit all the components of the vehicle in between and around the four wheels. Once the layout of the vehicle was decided the wheelbase could then be set. Knowing that the driver cockpit would be surround by the two battery packs and the orientations of the gearbox allowed for the wheelbase to be set at 6,850 millimeters.

There were several kinematic designs that were carried over from the BB2. One of those designs was the parallel wishbone suspension to prevent camber change and to simplify the design. This was done due to the limitations of space around the wheel. Since the tires expand at high speeds, if the vehicle had to a major weight shift the camber could potentially contact the chassis or body if proper precautions were not taken. One of the downsides of this design was that the tires would scrub more with suspension travel. This occurs because the track width would increase and decrease with suspension travel due to the wishbones being equal length. This was disregarded because the BB2 had
success with a similar geometry and the Bonneville Salt Flats are very smooth compared to many surface streets.

To prevent the wheels from being steered by bumps on the track, zero kingpin inclination was included. The kingpin inclination viewed from the front or back of the vehicle is the angle of the line that connects the upper and lower ball joints compared to the vertical axis. Traditionally, kingpin inclination tries to return the wheel to straight if deflected as well as reduces the need for a “dished” wheel. The decision to have little or no kingpin inclination was due to the steering force input needed to overcome imperfections and bumps on the track. With no kingpin inclination there is less steering input force needed to keep the vehicle heading in the correct direction. To eliminate any scrub radius the ball joints were put in line with the wheel center causing the allowed the kingpin to continue to have no inclination and zero offset. Having a scrub radius typically helps with lower speed maneuverability but since the design goal of the vehicle is to handle well at high speeds this was not a concern. Also, having a negative scrub radius can help keep a vehicle stable on split mu surfaces but since the VBB3 has such a narrow track width split mu surfaces tend to have a lesser effect on the vehicle.

To make the steering of the vehicle to self-center, 0.5 degrees of positive caster was added in the suspension design. Caster angle can be viewed from the side of the vehicle with a line connecting the upper and lower ball joints and compared to a vertical axis to give the angle. A positive caster angle makes the steering want to return to center. This effect can be seen when the steering wheel is turned and if let go while still rolling the steering wheel will attempt to return to center.
Stability of the VBB3 was extremely important in the design of the vehicle. To make the vehicle more stable at high speed, 25 millimeters of positive mechanical trail was added to the suspension design. This was done so that the wheels inherently want to point in the direction of the vehicle's travel. This effect can be seen on the front wheels of a shopping cart that has significant mechanical trailer to always point in the direction of the cart's travel.

Table 3.1: VBB3 Kinematic Settings

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suspension Travel</td>
<td>76.2 mm</td>
</tr>
<tr>
<td>Caster Angle</td>
<td>0.5 deg.</td>
</tr>
<tr>
<td>Mechanical Trail</td>
<td>25 mm</td>
</tr>
<tr>
<td>Kingpin Inclination</td>
<td>0 deg.</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>6850 mm</td>
</tr>
<tr>
<td>Track Width</td>
<td>658 mm</td>
</tr>
<tr>
<td>Steering</td>
<td>14 deg.</td>
</tr>
</tbody>
</table>
3.2. Wheel Hub

A wheel hub is the component that allows the wheel to rotate and can transmit the torque from the drive shafts to the wheels while supporting the load sprung mass loads of the vehicle. These components typically consist of a set of bearings, spindle, housing and seals.

3.2.1. Buckeye Bullet 1

The Buckeye Bullet 1 wheel hub was a very simplistic design that incorporated many off-the-shelf components. The design incorporated an off-the-shelf spindle that was mounted to a plate that acted as the wheel hub housing. The spindle used a pair of tapered roller bearing that have the preload set by a castle nut. The spindle also incorporated flanges to mount the brake rotor and wheel. The housing had the attachments for the ball joints and steering tie rod.
The wheel hub was designed for a vehicle to travel 300 miles per hour with a total vehicle weight of 4,000 pounds. Since it had similar weights to consumer vehicles and race vehicles, some off-the-shelf component for racing could be integrated into the design. The off-the-shelf components reduced manufacturing difficulties and made spare parts easier for the team to attain.

There were some issues with the design especially if the team were to go to higher speeds, which would generate greater loads in suspension. The biggest issue was that the spindle is pressed into an aluminum plate, which is then cantilevered out. The bearing sets were installed on the cantilevered spindle, this increases stress in the spindle as well as the plate that the spindle is secured to. If the vehicle were to go any faster or weigh
more, further considerations of the different bearings with higher static load capabilities would have to be investigated.

3.2.2. Buckeye Bullet 2

The BB2 wheel hub was significantly different in design compared to the BB1. The BB2 developed a custom wheel hub that allowed for a Learjet braking system to be incorporated inboard while still having all the steering and suspension characteristics that the team wanted. The spindle of the hub incorporated a bolt pattern to allow for a low-brow CV joint to be attached to the spindle. The spindle was supported by two large radial ball bearings that were sealed. The bearings were constrained on the spindle by a locking nut with washer, which could adjust preload on the bearings. Finally, the spindle assembly was installed into a custom aluminum housing that had all the attachments for the ball joints and steering tie rods.
The design was able to keep a fairly narrow package to allow for the braking system to be incorporated. It was also very flexible because the hub assembly could be used at any of the four corners of the vehicle by changing the location of the steering arm and spacers at the ball joint connections. There were also several downsides to the design as well. One of the biggest issues was that the sealed bearings were not able to keep all the particulate out of the bearing, which caused them to have to be replaced nearly every year. There were no seals integrated into the housing to slow the ingestion of contaminants. Another issue was the strength of the steering arms connection. Four bolts could remove the steering arm but the bolts threaded into an aluminum part that has steel inserts with minimal thread engagement. Finally, assembly of the hub on the vehicle had some
difficulties especially when attaching the CV joint. The brake assembly further exaggerated the assembly process due to the lack of space.

3.2.3. Venturi Buckeye Bullet 3

The goal of the design was to develop a wheel hub that could support a vehicle traveling at 500 mph that weighed statically 2,000 pounds per corner. This equated to a wheel speed of roughly 6,800 revolutions per minute. It is an estimation because of the lack of information of tire growth at high speeds. The load on the hubs due to weight transfer and aerodynamics was estimated at roughly 2,600 pounds on the rear axle, which will see the highest amount of loading due to weight transfer from acceleration. Another major goal was to find a seal that could work for the application so that the bearings wouldn’t have to be replaced after every racing season. Finally, an improvement in installation compared to the BB2 wheel hub was essential.
3.2.3.1. Spindle

The spindle design was heavily constrained by the type of CV joint that was going to be used. The two options available for the CV joint were a splined design and a bolted design that transmits the torque to the spindle. The bolted design would allow for easier installation but it had a larger outer diameter than the splined CV joint. Also, the splined CV joint was more efficient in carry the torque as well as being a more compact design.
The wheel bearings of the hub were directly affected by the size of the CV joint since the CV joint was housed internal to the spindle. This meant that with a large CV joint, larger wheel bearings had to be selected. For this reason the splined CV joint design was selected and used to minimize the wheel bearing size. With the selection of the spline CV joint, the integration into the spindle was set by having to retain the CV joint with a shoulder and a retaining ring. The CV joint was sealed on one side with a flexible boot and on the other side sealed by an aluminum cap that uses an O-ring to seal to the spindle surface so that water or salt could not enter.

The spindle flange was kept similar to the BB2’s design so that the wheels from the BB2 program could be used for lower speed testing. The flange diameter was slightly changed to account for different wheel studs while a centering ring boss was created on the face of the flange to help hold concentricity between the wheel and spindle. The concentricity of the wheel is primarily determined by the wheel studs, which incorporate a tapered nut to bring the wheel in concentricity with the hub. This is the most common method of
holding concentricity between the spindle and the wheel. The studs that used were from ARP that came from their NASCAR Speed Stud line. These studs are made from high strength steel and are designed so that they cannot be cross-threaded.

The material of the spindle was selected by The Timken Company in collaboration with the team. The analysis of the spindle was also conducted by The Timken Company to determine the correct material for the application. Due to the braking capacity on pavement the CV joint could apply a braking torque through the spindle of 3,700 foot-pounds. With the team’s input Timken was able to analyze the spindle to insure that the design was adequate. From the analysis, the alloy of steel selected was hot rolled SAE 1045. The spindle was also surface hardened to a Brinell Hardness range of 217-255 with a core hardness of 57-63 Rockwell A. The increased surface hardness allows for slightly easier installation of the bearings and reduces the likelihood of galling during installation.

Figure 3.6: VBB3 Wheel Hub Spindle
3.2.3.2. Bearings

The bearing selection for the wheel bearing was a difficult task due to the load and speeds that the vehicle could achieve. Initial designs looked at radial ball bearings and angular contact bearing but none of the bearings were rated for the load and speed. These types of bearings have lower friction, which would reduce rolling resistance.

The difficulty in the selection process was that no off the shelf bearing was capable of the speeds and loads that the vehicle could see. The partnership between The Timken Company allowed for a custom bearing solution so that the bearings could withstand the loads and speeds.

First, the team and Timken had to decide on the type of bearing, then finding a range of inner diameters that would allow for the CV joint to be packaged inside of the bearing diameter. Once a range of bearings were identified, the static load capabilities were compared to the projected loads from the vehicle. The static load capacity was the greatest concern because the dynamic loads and speed could be compensated for by changing greases and coatings on the rolling elements. Also, the effects of exceeding the maximum rated speed and dynamic load effects the life of the bearing but due to the relatively short service life for the vehicle this was not a major concern. Finally, the fit of the bearing had to be selected. This determines if the bearings have a preload, line-to-line, or endplay fit.

The bearing type that was used in the wheel hub design was a tapered roller bearing. This type of bearing is commonly used in automotive wheel hubs because of its load carrying capacity radially as well as axially. This type of bearing inherently has higher
drag from friction due to the greater contact area compared to a ball bearing but some of this could be compensated by modifying some of the geometry as well as adding coatings and better greases.

The range of the bearing inner diameter was constrained by the size of the CV joint and the necessary material to make the spindle structurally strong enough. The inner diameter range of the bearing was set to 4.75 inches with a tolerance of -0.150 and +0.250 inches. This allowed for a significant reduction in the number of bearing that could fit. The positive tolerance range was set fairly small to try and limit the size of the bearing, which would increase drag as well as reduce the size of the spindle. An increase in spindle diameter would increase the mass and inertia of the part which emphasized the need to have as small of bearings as possible.

Figure 3.7: VBB3 Wheel Hub Tapered Bearing
With a range of bearings that fit the size criteria selected, the static load capacity was then considered. The bearings that met the static load criteria were identified and compared to identify the bearings that were the smallest package.

There were a couple modifications made to the bearing to deal with the potential speeds as well as reduce friction in the bearing. The first modification made was to grind a crown into the cup and cone of the tapered roller bearing races. This modification reduces the contact area between the rolling element and the races, which in turn reduces bearing drag. The other major modification to the bearing was to coat the part of the roller. The large end of the roller that occasionally comes in contact with the rib was coated in a tungsten carbide type material. This did two things, it acted as a dissimilar material that prevented micro-welding and adhesive wear at higher speeds, but it also reduced friction between the roller and rib, which is traditionally where a significant amount of drag originates from in tapered roller bearings.

Figure 3.8: Tapered Bearing Rib
3.2.3.3. Bearing Setting

Having the bearings selected, the setting of the bearing had to be specified. There are many tradeoffs to the three settings of preload, line-to-line and endplay. Preloading the bearing improves the stiffness of the assembly, which allows the assembly to spin true while loaded. Preloading the bearing to a point also improves the life of the bearing. The downside of preloading the bearing is that it increases the drag. Having endplay for the bearing assembly reduces the stiffness of the assembly as well as bearing drag. Endplay can also lead to reduced life of the bearing. Finally, line-to-line fits have many of the attributes of both endplay and preloaded bearings.

![Bearing Setting Tradeoff Graph](image)

Figure 3.9: Bearing Setting Tradeoff Graph [10]

For the vehicle, life of the bearing was not a primary concern but the stiffness and drag were very important. The stiffness of the assembly was critical due to the high rotational...
speeds. If the bearings allowed the assembly to deflect at high rotational speed it could introduce unique loads that were not intended originally in the design. The drag was also a sizeable concern as well because the drag in the bearing increases the overall vehicle rolling resistance. Considering the stiffness and drag, it was decided that the bearings fit would target a line-to-line fit to have a rigid assembly while trying to reduce drag. In practice the fit would most likely be a slight preload. This would occur because the large nut that sets the fit is held in place from loosening with a bearing lock washer. There are a certain number of orientations that would allow the washer to lock with the nut so the nut might have to be tightened passed a line-to-line state to allow the washer to lock into the nut.

3.2.3.4. Seals

With the VBB3 operating on the Bonneville Salt Flats it is critical to have seals to prevent salt and particulate from entering the bearings and grease. Previous designs relied on shielded and sealed bearings but these methods were never sufficient at keeping the contaminants out. The goal of the project was to implement seals into the wheel hub housing and not to rely on a seal on the bearing. When specifying a seal it traditionally looks at the surface speed of the rotating component that it will be in contact with, typically in feet per minute. There are two different seal sizes on the wheel hub and the outboard, or wheel side of the hub is the greatest diameter of the two. The diameter of the spindle at the outboard seal was 5.25 inches, which would correlate to a surface speed of 9,350 feet per minute at 500 miles per hour. All traditional contact seals have
maximum speeds significantly below this speed, approximately 3,500-6,500 feet per minute depending on material. Also, traditional contact seals add more resistance to the assembly so another option had to be investigated. The alternative to a contact seal is a non-contact seal. This type of seal relies on several internal channels closely spaced to prevent particulate from coming through. A common non-contact seal is the labyrinth seal. The non-contact seals have much higher speed rating but they are commonly more expensive than contact seals. A contact seal made from PTFE was considered because of its high operating speeds of to 6,500 feet per second and the duty cycles over the maximum speed would be rare and very short. In the end the seal that was chosen was a hybrid between labyrinth seal and a contact seal, it is called an isolator seal. This type of seal is used in harsh environmental applications traditionally. The seal relies on a winding path for particulate to have to travel as well as a contact seal that is part of the seal assembly to prevent contaminant from entering. Although there is contact in the seal that increases friction it is minimized with minimal contact between the seal and sealing face. The cost of the seals were also significantly less than labyrinth seals while still being more expensive than the traditional contact seal.
3.2.3.5. Housing

The wheel hub housing reacts and transfers all the road loads to the vehicle from the spindle to the ball joints. The loads from the wheel are reacted by the bearings and transferred to the wheel hub housing and finally carried to the A-arms at the ball joint connections. The majority of the loads seen by the housing are radial loads from the vehicle's weight as well as lateral forces during low speed turns. The goal was to reduce the weight as much as possible but still have a rigid housing that would be capable of reacting all the loads and minimize the compliance as much as possible.

The final housing design consists of two major parts that enclose the spindle assembly. The main housing reacts the entire radial and half of the lateral loads while the back housing reacts the other half of the lateral loads. The main housing incorporates the ball joint connections as well as the steering tie rod connection. The connections for the ball joints are ears that allow for minor adjustments in the suspension geometry. These adjustments include the caster and trail of the suspension. Steel bushings were integrated
into the housing to protect the housing from damage or deformation and to distribute the load more evenly through the ears. The main housing also includes the steering arm. A detachable steering arm was considered to make it like the BB2’s wheel hub that allowed the hub to be used at any corner but due to the force and the packaging area, the decision was made to integrate the steering arm into the housing. The inboard housing had relatively low forces to react so the goal was to minimize the weight while still being able to react the loads. Both housings were made from aluminum but the selection of the alloy of aluminum was not chosen until structural analysis was conducted.

Figure 3.11: VBB3 Wheel Hub Housing
3.2.3.6. Analysis

Analysis of the wheel hub housing was conducted by the VBB3 team. The primary analysis focused on the main housing of the wheel hub and looked at the radial, lateral and steering loads applied to the housing.

The loading scenario for the analysis was if the vehicle was braking at full steer. This is highly unlikely that the vehicle would ever be in the scenario but it would be possible on the test track. This meant that a radial load force distribution was applied to a set of bearing used in the analysis so that the contact with the housing could better represent real conditions. A lateral load was also applied to the bearings that were being reacted by the housing. Finally a moment was applied about the wheel axis on to the upright housing to represent the braking loads.

The results of the analysis showed higher stress around the lower ball joint since the lower A-arm transfers the wheel loads to the shocks. Also, there was significant stress at the base of the steering arm. This further confirmed the thought to integrate the steering arm into the housing over making it detachable. All the attachment points for the ball joints and steering tie rod all had stress concentrations around the bolt holes but that was to be expected. The steel bushings in the housing ears would help to distribute the load and decrease the stress. There was also higher stress at the base of the shoulder that the outboard bearing contacts since the radius could be no bigger than 0.030 inches because of the outer bearing race.
3.3. A-Arms

The A-arms connect the ball joints to the chassis. These parts react the vehicle accelerations as well as transmit the road loads to the spring and damper systems. In consumer vehicles the A-arms are traditionally cast or forged parts but in racing the parts are typically tubular, composite or machined components.

3.3.1. Buckeye Bullet 1

The A-arms of the BB1 are different from front to rear, this is due to the track width being drastically different in the front and rear axles. The A-arms were very simplistic in
design to allow for easy manufacturability. The designs used rod ends at the ball joints and bushings at the inboard attachment. The inclusion of rod ends allowed for the ability to make kinematic adjustments.

![BB1 Rear A-Arms](image)

Figure 3.13: BB1 Rear A-Arms

3.3.2. Buckeye Bullet 2

The BB2 had a fully independent suspension and had more complex A-arms due to the packaging constraints. The upper A-arm consisted of spherical bearings at the inboard connection while the ball joint connection used a spherical bearing. The lower A-arm was the opposite and used a spherical bearing at the ball joint and rod ends at the inboard connection points.
In both designs the use of spherical bearings and bushings allowed for more efficient designs. The spherical bearings also are able to carry more load compared to a rod end. The issue with some of the designs is the use of the rod ends. Since the A-arms react the loads of the vehicle from the vehicle’s acceleration and deceleration, the rod ends are put in bending which greatly limits the load carrying capacity. Also, due to the packaging constraints some of the rod ends did not have much thread engagement which further
weakens the component. Some of the A-arms, especially from the BB2, had stress riser due to rapid changes in geometry.

3.3.3. Venturi Buckeye Bullet 3

As mentioned earlier the independent suspension has parallel link lengths that are equal length. This means that the upper and lower A-arms have the same distance between the ball joints and the inboard connecting points to the chassis. The A-arms have to be able to support the various loads exerted on them from the vehicle accelerations as well as the lower A-arm having to support the vehicle sprung mass. The components had to accomplish this while avoiding interferences with the gearbox, chassis and wheel. The parts also needed to be very easily manufactured so that they could be produced in a short timeline. Also, to make the suspension system more manufacturable the A-arms needed to be as interchangeable as possible to reduce the number of distinct parts as well as reduce the lead time and cost.

3.3.3.1. Upper A-Arm

The designs of the VBB3 upper A-arm use spherical bearings for the inboard connection and the ball joint. Due to loads and the load ratings of the spherical bearings the inboard bearings use a 0.500 inch inner bore diameter while the ball joint spherical bearing has an inner bore diameter of 0.625 inches. The spherical bearings that were selected for the upper A-arms were high misalignment bearings. The high misalignment bearing was selected for ease of installation so that no spacers had to be made and higher load
carrying capacity for the inboard mounts. The high misalignment also allowed for the necessary rotational capability for the steering system. One of the limitations of the spherical bearing design is the retaining ring that prevents the spherical bearing from moving axially. Depending on the material selection and style of retaining ring can greatly change the thrust capacity of the retaining ring. Another factor the thrust capacity is the material of the material of the A-arm and how well the material supports the loaded area. Finite element analysis was used to insure that there was an appropriate safety factor for the retaining ring groove in the A-arm. The A-arm material selection of 6061 aluminum was confirmed using the FEA results and verifying that there was a minimum safety factor of 2.5.

Figure 3.15: VBB3 Upper A-Arm
3.3.3.2. Lower A-Arm

The lower A-arm had many of the same considerations in the design as the upper A-arm but the greatest difference was that the part was transferring the load of the road to the spring and damper. This meant that the component was being put in bending, so the force exerted on the part was much greater. The placement of the push rod attachment point also greatly effects the spring and damper design. The closer the connection is to the ball joint, the greater the motion ratio. The motion ratio is the damper motion divided by the wheel motion, which gives a ratio of the motion between the damper and the wheel. A greater motion ratio reduces the forces in the components while increasing the overall displacement. The final consideration of the design was that the attachment point to the push rod had to be along the centerline of the axle to allow the part to be interchangeable from front to rear. This would allow for the push rod and rocker geometry to not be effected.

Figure 3.16: VBB3 Lower A-Arm
3.3.3.3. Analysis

The analysis of the VBB3 suspension was conducted using theoretical loads as well as loading scenarios in worst-case conditions. The loads are all estimated from vehicle accelerations as well as possible steering inputs. Since the coefficient of friction changes significantly and there has never been any form of accurate measurements the range significantly varies. To insure that the vehicle components will be able to withstand the loads, all the forces were calculated for a vehicle on asphalt. This is a surface that the vehicle tests on in Ohio and has the greatest potential to generate higher forces. The loads assigned were applied in a scenario as if the vehicle was braking at the tractive limit. The vehicle’s braking deceleration capability is greater than the acceleration capabilities. This again is done to represent the worst-case loading scenarios.

To define the forces the maximum braking force was calculated for a given acceleration and peak coefficient of friction [11].

\[ F_{xmf} = \mu_p \left( W_{fs} - \frac{h W}{L g} D_x \right) \]

Where \( \mu_p \) is the peak coefficient of friction, \( W_{fs} \) is the static front axle load, \( h \) is the CG height, \( L \) is the wheelbase, \( g \) is the gravitational constant and \( D_x \) is the deceleration of the vehicle. The forces used for the analysis were for the front axle due to weight transfer effects. The weight transfer allows for greater tractive force as well as puts more load on the lower A-arms of the front axle.

The theoretical lateral forces from cornering also had to be determined. To accurately determine forces requires cornering stiffness’s and suspension setup parameters that at the time were not identified so a more simplistic analysis was used to determine the
lateral forces. First, the outside turning radius of the front wheels was determined for maximum steer applied [11].

\[ R = \frac{L}{\delta} - \frac{t}{2} \]

Where \( R \) is the turning radius of the inner or outer wheel, \( L \) is the wheelbase, \( \delta \) is the steering angle and \( t \) is the track of the vehicle. With the front turning radius determined, the lateral force on the tires can be determined [11].

\[ F_{yf} = \frac{W_{fz} V^2}{g R} \]

Where \( W_f \) is the load on the front axle and \( V \) is the vehicle velocity. This gives an estimation of the lateral force but does not consider the effects of the tires and roll characteristics of the vehicle that affect the lateral force capabilities.

With the maximum tractive loads and theoretical lateral forces, simple free body diagrams were created to determine the loads on the upper and lower ball joints. Having the theoretical forces for the upper and lower ball joints, the analysis of the upper A-arm could be conducted. The analyses of both A-arms fix the inboard attachment points while the ball joints have the load applied to them.

The inboard attachments use spherical bearings to allow appropriate motion. To fix the upper A-arm, the radial surface that the spherical bearings are in contact with are locked from any rotation or radial translation. The thrust forces are reacted by the retaining ring or shoulder depending on the acceleration direction. Both loading scenarios were run to insure that there would be no failure in either orientation. To apply the longitudinal constraints either the shoulder that the spherical bearing would be in contact with or the
surface that the retaining ring would be in contact with would fix the longitudinal forces for the inboard attachment points.

The ball upper ball joint also utilized a spherical bearing so the loads on the ball joint were applied similarly to the constraints on the inboard points. The lateral force of the vehicle were applied using a distributed force on the radial surface of the ball joint which was distributed in the worst case scenario of putting the A-arm in tension. The longitudinal force from deceleration was then applied to the shoulder or the surface that the retaining ring was in contact with.

To analyze the lower A-arm the load of the vehicle being reacted by the ball joint has to be determined. Since many of the characteristics of the vehicle were not solidified so the analysis to determine the load became very generalized. To generalize the analysis, a model of the vehicle with no suspension was used in combination with the known lateral acceleration so a simple sum of the moments calculation could be used to find the increase in load due to the lateral acceleration.

Figure 3.17: VBB3 Axle Free-Body Diagram
The force from the lateral acceleration is then compared against the static wheel loads to find the change in force due to the lateral acceleration. That force is then added to the outside wheel of the front axle that is accounting for longitudinal load transfer from the braking.

This load is applied to the lower ball joint of the lower A-arm using a distributed load since the joint uses a spherical bearing. The thrust load from the deceleration is then added as well to the shoulder or the retaining ring groove face depending on the orientation. With the load of the A-arm calculated, the component then needs constraints to fix the part in the analysis. Like the upper A-arm, the inboard attachment points have similar constraints on the lower A-arm. The only additional constrain added is the radial constraint that is applied to the pushrod attachment point of the lower A-arm.

Using the constraints and loads described above for the A-arms allowed for the materials and the safety factors to be determined. Since the upper A-arm experiences significantly less force than the lower A-arm, it was able to be made out of 6061 aluminum while the lower A-arm which was put in bending due to the pushrod had to be made from 7075 aluminum to meet the required safety factor. The safety factors that were chosen were fairly high due to the uncertainty in the loads and application of the loads and constraints in the analysis. Also, there was not enough time to manufacture and test the parts to validate the analysis techniques.
3.4. Wheels

The wheels take the road loads from the tire and transfer them to the hub. In the case of land speed racing this occurs while the wheel is rotating at high rotational speeds. The wheel design has become considerably more complex for each vehicle due to the speeds, loads and design constraints.

3.4.1. Buckeye Bullet 1

The wheels BB1 used were aluminum wheels that were fairly simple in design. They incorporated a steel ring where the wheel lugs came through the wheel to distribute the load of the lug nuts across the steel ring. The wheels also had very little offset built into them due to the small size of the wheel hub. Very little analysis was done on the wheels due to the lack of experience and unknown loads.
3.4.2. **Buckeye Bullet 2**

The BB2 took the next step in design and manufacturing due to the increased weight of the vehicle and offset needed due to the hub design. To design the proper profile so that not only the land speed tires could be mount but also commercial tires could be mounted and seated properly on the wheel, a standard profile from the Tire and Rim Association was used. The wheel also had a 66.04 millimeter offset to allow for the packaging of the hub and ball joint attachment points. To center the wheel there was a recess that was mad that loosely centers on the outer diameter of the hub’s spindle. The wheel was truly centered on the spindle by the lugs that have tapered nuts that contact the tapered surface of the bolt holes on the wheels. Using a recess in the wheel introduced
stress risers in the wheel. Luckily, this region of the wheel was not highly stressed so it was not as much of a concern. There was still very minimal analysis done on the wheel to understand the wheel’s limits. To insure that the wheels would be strong enough, the wheels started as forged billets and near net machined. After the initial machining was done the wheels were then heat treated. Once heat treating was complete, the wheels were finish machined. This time intensive process of making the wheels allows for a better grain structure to be formed and a thorough heat treatment through the whole material.

Figure 3.19: BB2 Wheel
3.4.3. Venturi Buckeye Bullet 3

The wheels for the VBB3 had some very difficult requirements to meet. The wheel needed to be designed to go approximately 500 miles per hour which due to the tire growth makes the rotational speed difficult to identify. Using trends from data collected on the previous vehicles to estimate wheel speeds at 500 miles per hour it was targeted that the wheel needed to be able to spin at 6,800 revolutions per minute. The road loads were also difficult to quantify but from generalized analysis and previous data an estimation could be made. Similar to the A-arm analysis the worst-case loading scenario was assumed with the vehicle pitching and turning at the same time. The wheels also had to account for 90 pounds per square inch of air pressure for the pneumatic tire. The VBB3 wheel design started with the BB2 wheel as the basis of the design. The first thing that was modified was the offset of the wheel. Due to the suspension geometry and hub design, the offset of the wheel had to be increased. Also, the recess for the spindle flange alignment was removed for the VBB3 wheel to reduce the stress in the wheel. To align the wheel a centering ring boss was created on the spindle of the hub so a higher tolerance pilot hole had to be used to mate with the centering ring.

The internal geometry where the tire sits was kept the same except for the region where the valve stem was implemented. The two valve stems that are traditionally used are threaded into the wheel or is inserted through the back of the wheel flange and secured with a nut. The BB2 wheel used a thread valve stem that occasionally had issues maintaining a pressure rated seal and could unthread over time. The goal of the VBB3 wheel was to use a valve stem that could be installed from the backside of the flange and
be secured by a nut. It was also important to find a valve stem that was rated to handle the operating pressures of the tires. This led to the selection of a brass valve stem that is traditionally used for motorcycles but is rated for higher pressures. The advantage of the motorcycle valve stem was that it was smaller in diameter than most automotive valve stems making the necessary sealing area smaller. The challenge was to find a valve stem that had a small sealing surface area so that large stress risers didn’t have to be added into the wheel around the valve stem. To get sufficient sealing area some of the radii around the valve stem had to be reduced but they were still significant enough to reduce the stress concentrations. A small recess had to be cut into the external face of the wheel to allow for the nut of the valve stem to sit flat against the wheel. To make the recess smaller a custom nut for the valve stem was made that was flanged and minimized the diameter of the nut as much as possible.
3.4.4. Analysis

The analysis of the VBB3 wheel design was critical in the material selection. The wheel is one of the components on the vehicle that has a very low margin for error and any failure of the wheel has to be avoided at all cost. To assist in the analysis of the wheels, the team received technical support from Cooper Tire Company. The analysis of the wheel was a difficult task because of the difficulty in applying realistic loads and discretizing the wheel into the proper mesh to improve accuracy of the results while not using excessive amounts of computing power.

The rotational forces were very easy to implement with the software package that was used in the analysis. There was a load application that allowed for rotational loads to be
applied. All that had to be specified for the load application was the rotational speed and the axis of rotation.

The road loads also had to be applied to the wheel. The two road loads that were considered were the vertical loads as well as the tractive loads. Both of the loads were considered at the contact point between the wheel and tire. To apply the loads, the wheel was broken into six equal sections. The vertical force was then represented by a sine distribution over the one sixth of the section of the wheel with zero load at the two extremes and maximum load applied in the center of the distribution. This load is applied normal to the wheel and tire bead contact region. The tractive loads were applied over the entire tire bead to wheel contact area and that was applied tangent to the surface. The assumed load applied was equivalent to 1 g of acceleration.
Figure 3.21: Wheel Vertical Loading Distribution

Tire pressure loads were also included in the analysis. It was expected that the tire pressure would have a relatively small effect on the stress in the wheel but to be thorough with the analysis the load was applied. The load that was applied in this case was a constant pressure load across the whole interior area of the wheel flange where the air would be in contact with the wheel as well as the area where the tire bead comes in contact with the wheel as well.

The steering loads on the wheels were applied to the one sixth section of the wheel but laterally across the wheel. The load was applied perpendicular to the surface of the vertical bead of the wheel.
Figure 3.22: Wheel Analysis Internal Loading

In order to mesh the part and discretize the model several methods were considered. Initially, a constant mesh size was used but this required significant amount of computational time. The fine mesh that was needed for the higher stress areas were not needed in the lower stress areas and only added unneeded computational time. The next method considered was a course mesh for the lower stress regions while having a much finer mesh in the high stress regions. This would reduce computational time and would still give accurate results. The difficult with this methodology is getting the correct mesh control. The mesh can at times be difficult to control through the thickness of the part especially in unique geometry. With the difficulty in meshing, Cooper Tire Company was brought in for technical support and with some collaboration a new method was devised. This method used a section profile of the wheel that the mesh sizing could be determined and then revolved. The method used quadratic elements to better approximate linear bending. Using this method allowed for finer control of the mesh while still having a reasonable computational time.
The analysis considered two materials, 6061 and 7075 aluminum. After applying the material properties to the analysis could be computed. Once, the finite element analysis (FEA) was completed the Von Mises stress results were analyzed. The results showed that the region where the wheel profile spits into two directions and the same surface where the valve stem would be located was the highest region of stress.
To reduce the stress in the corner a greater radius could be used but it would also effect the valve stem location. If the greater radius was placed there, a larger pocket would have to be made to give the valve stem a flat surface to seal against. This pocket would also have significant stress risers since the pocket could not be radiused. A compromise had to be made between the radius of the corner and the size of the nut that retains the valve stem.

Figure 3.24: Wheel FEA Results
The previous wheels for the BB2 were 6061 aluminum. The wheels were forged and heat treated. This is a very time consuming process that requires cooperation of several companies but gives maximum strength to the part. Due to the short time frame the wheels for the VBB3 had to be produced in a much shorter time period. This meant that the two types of aluminum had to be used from round stock or plate without any heat treating. Since attaining rounds of the material were very difficult to find, the wheels had to be machined from aluminum plate. Material properties were attained for the various types of aluminum with various grain structures orientations. The lowest yield stress for the various grain structures and loading orientations were selected since the wheel is a rotational piece and at least two orientations of the wheel will have the lower yield stress associated with the region. Using this information the two material properties were applied and the results found that the two materials would not fail. However, the safety factor was near two for the 6061 wheels and the safety factor of 7075 was greater than
two. With the results of the analysis, the material for the wheels was selected to be 7075 machined from plates.

3.5. Spring and Damper System

The spring and damper system in vehicles control the ride of the vehicle and its ability to maintain traction. In high performance racing vehicles, the design of the spring and damper system focuses on the traction capabilities of the vehicle over the comforter to the passenger. In order to package the spring and dampers more efficiently in a fully independent suspension, a link that connects to the motion of the wheel is used to transfer the wheel loads to a rocker. The rocker is able to change the direction of the motion as well as modify the force and velocity seen by the spring and damper.

3.5.1. Buckeye Bullet 1

The BB1 had an independent front and rear suspension that incorporated a pushrod system that was actuated by the lower A-arm in the front suspension while the rear pushrods connected directly to the wheel hub. The pushrod translated vertically and connected to a rocker, which allowed the springs and dampers to be packaged more freely. The shock used was a coil-over shock, which is an efficient method of packaging the spring and damper and is commonly used in racing. The spring preload could be adjusted by adjusting the nut at the base of the spring. The damper was also adjustable for both jounce and rebound.
3.5.2. Buckeye Bullet 2

The Buckeye Bullet 2 also utilized a fully independent pushrod suspension in front and rear. The motion of the wheel was transferred to the pushrod by the lower A-arm. Due to packaging space around the gearbox and powertrain the coil-over springs and dampers had to be packaged underneath the gearbox in the front and high pressure gas lines in the rear. With this configuration the rockers were mounted parallel to the ground plane. The difficulty with this design is that there is very minimal travel in the shock and that the motion ratio through the full suspension travel is nonlinear. If compensated for in the design of the shock and vehicle design there is little issue. If the system is highly nonlinear and not compensated for the vehicle can handle very poorly in certain scenarios.
3.5.3. Venturi Buckeye Bullet 3

The design of the spring and damper system for the Venturi Buckeye Bullet 3 first looked at the packaging space available. Due to the space restrictions the system that best fit for both the front and rear axles was a pushrod system. This would allow for the most optimal use of the available space. The push rod was mounted to the lower A-arm and as close to the lower ball joint without causing any interference with other components. Another concern was to try and get the shock, rocker and pushrod on a similar plane throughout the full motion of the suspension to reduce side loads on the rocker that would increase stress in the mounts and friction in the system. Also, the anti-roll bar had to have the ability to attach to the rocker without interference as well as be oriented in an effective position through the whole range of motion.

To meet the requirements a significant amount of work was done to try and orient the shock, rocker and pushrod in the most optimal orientation. The push rod attachment was moved out as far outboard on the A-arm to allow for greater motion that would also
reduce the loads carried by the system. Once the general location of the pushrod attachment for the A-arm was determined the rocker and shock orientations were considered. Since the shock had minimal misalignment capability it was essential that the rocker and shock be nearly on the same plane of motion. The limitations of the orientations for the system were the mounts for the rocker and shock to the chassis. The areas of the mounts needed to be well supported since there was a significant amount of load being carried in those areas. With these constraints identified the orientation of the pushrod, rocker and shock could be adjusted to allow for maximum movement while remaining near the same plane of motion.

Figure 3.28: VBB3 Spring and Damper Setup
3.5.3.1. Pushrod

The pushrod transfers the load from the lower A-arm to the rocker. It also allows for adjustment of the vehicle ride height. By making the push rod longer it raises the vehicle ride height while if it’s shortened it decreases the ride height. This required that the pushrod be adjustable in length while having misalignment capabilities at the attachment points. It also has to carry the entire load so the design needed to be strong as well. One of the most common ways of making a pushrod is by using a tube that have welded-in threaded inserts at each end of the tube with rod ends threaded into them. This method was used to make the pushrods. The tube selected for the application was a 4130 steel tube that has good strength and weldability. The threaded inserts were left-handed thread on one end while the other end was right-handed thread so that the pushrod could be rotate to shorten or lengthen it with ease. High misalignment rod ends were used to allow misalignment at the connection points but they also were able to carry higher loads. Finally, jam nuts were used to fix the rod ends from moving. Quick calculations were done to insure that the tube would not buckle. Due to the length and diameter of the tube the safety factor was well beyond what was required.
The rocker connects the pushrod, anti-roll bar and the shock together. This meant that a significant amount of force was being transferred through the rocker as well as any misalignment would lead to unwanted loading and potential for binding. The attachment for the pushrod and shock were set by trying to maximize the motion of the shock so that the motion ratio would be as low as possible to reduce the forces. The only place that the anti-roll bar link had to attach to the rocker was between the shock attachment point and pivot of the rocker. This orientation would allow for the anti-roll bar to have an effective motion ratio as well as it could help to overcome any binding in the system. Once all the points were selected the next focus was finding a bearing that could handle the load as well as reduce friction between the rocker face and the rocker mount ears.
Initially, roller and ball bearings were considered but very quickly it became obvious that there was no traditional bearing that would be able to handle the load in such a small package. The next thought was to use metallic plain bearings. When calculating the bearing unit loads and sliding velocities for the necessary bushings it was found that the pressure was too high for a metallic plain bearing [12]. Finally, a plastic plain bearing was considered. The same pressure and velocity calculations were used for the selection of the plastic bearing and very quickly it was found that the plastic bearings could potentially handle the load and velocity. To insure that the plain bearing would work the team worked with Igus, which is a world leader in plastic bearings. With their support the team was able to identify a material and bearing that could work with the design.

Figure 3.30: Igus Plastic Plain Bearing

The plain bearing was able to carry the radial loads but due to the misalignment of the pushrod at the extremes of suspension travel, the pushrod would also impart a lateral force that was reacted by the face of the rocker and rocker mount. To prevent binding
thrust bearings had to be implemented. Again, the team worked with Igus to determine
the correct material for the washer that could then be manufactured by the team. The
thrust washers were recessed slightly into the rocker to help constrain them but protruded
past the rocker face to allow complete contact between the thrust washer and the rocker
mount face.

Finally, the material of the rocker had to be selected. To do this the rocker was analyzed
and very quickly it was determined that a higher strength material would need to be used
because of the loads and misalignment. Aircraft grade 7075 aluminum was considered
along with grade five titanium, Ti-6Al-4V. The aluminum was slightly lighter but was
not nearly as strong. It is also corrodes if it is not properly coated. Both materials were
considered in the analysis and found to be able to handle the loads applied. The deciding
factor was that the loading scenario was difficult to model this meant that the accuracy of
the loads and constraint might have been off slightly. Since the aluminum was close to
the allowable safety factor and knowing that the analysis was not highly accurate it was
decided that grade five titanium would be used for the rocker.
3.5.3.3. Shock

Many different shocks were looked at when comparing the options but the most compact and efficient for packaging was the coil-over shock. To reduce costs the team decided to reuse the shocks that the BB2 used with heavy modifications. The shocks were made to fit in the packaging area and included higher misalignment spherical bearings at the connection points. Also, the reservoirs were removed from the main body of the shock to being a remote reservoir that was connected by flexible line. This allowed for the reservoirs to rest underneath the gearbox.

There were obvious changes made to both the springs and dampers to account for the differences in VBB3 compared to BB2. The selection of the springs and dampers are difficult because there is not a guide for vehicle setup in land speed racing. The vehicle shouldn’t have a firm ride like an open wheel racer or an extremely soft ride either. To select the springs and dampers it would be best to look back at vehicle setups from previous vehicles that performed well on the Bonneville Salt Flats. A vehicle that performed and handled well was the BB2. To specify the VBB3 spring and dampers the
BB2 spring and dampers had to be analyzed and used as a basis for performance to set as performance targets.

To determine the spring rate for the VBB3 first the ride frequency was calculated using the spring rates for the front and rear axles along with their motion ratios to determine the wheel rate [7]. The ride frequency is the frequency at which the body is oscillating on the springs. The wheel rate is the stiffness of the springs at the wheel which is why the motion ratio is used in the calculation.

\[ K_{wr} = K_s(MR)^2 \]

Where \( K_s \) is the spring rate and \( MR \) is the motion ratio. With the wheel rate the ride rate can then be calculated. The wheel rate has given the stiffness of the spring at the wheel and so the ride rate is the stiffness of the whole system including the tire stiffness. If the tire was infinitely stiff the wheel rate and ride rate would be equal.

\[ K_{RR} = \frac{K_{wr}K_T}{K_T + K_{wr}} \]

Where \( K_T \) is the tire stiffness. The ride rate is then used to determine the ride frequency.

\[ f_n = 0.159 \sqrt{\frac{K_{RR}g}{W}} \]

Where \( W \) is the weight of the sprung mass. The ride frequency is determined for both the front and rear axles. With the ride rates for the BB2 front and rear axles the same equations can be reused with the BB2 ride frequency solving for the spring rate of the VBB3. In the end the VBB3 uses 900 pound per inch springs in the front with 1,000 pound per inch springs in the rear. The higher spring rate in the rear of the vehicle allows for a greater ride frequency compared to the front. This allows the rear suspension to
catch-up with the front suspension’s response so that the rear is stable and gives the vehicle the appearance of a flat ride.

To determine the damping for the VBB3, BB2’s dampers were analyzed. Once there was an understanding of the BB2 dampers the same calculations would be done for the VBB3’s dampers. Alterations would then be made to the force and velocity curves for the dampers to match the performance to BB2.

Since the system is a second order system with non-linear damping an equivalent damping had to be determined by calculating the work of the system [13]. Due to confidentiality the equations that were used will not be presented. The first step in determining the characteristics is to determine the equivalent viscous damping coefficient for a given force and velocity of the damper. To be able to determine the coefficient a sinusoidal input is used with a frequency of one hertz. The work of the damper is calculated for each velocity of the damper force-velocity curve. After determining the work of the damper the viscous damping coefficient is calculated for each of the velocities. Once the equivalent viscous damping is determined the effects of the damping can be seen by analyzing the sprung mass, roll, pitch and bounce damping ratios and natural frequencies. To determine these the sprung mass inertias as well as the pitch centers and modal inertias. Finally, after all these calculations the damping and natural frequencies can be determined for every velocity.

Using the results the BB2 and VBB3 damping can be compared. The goal was to make the VBB3’s damping ratios similar to the BB2’s by modify the force velocity curves of the VBB3 dampers. This was done by hand due to time so some of the results were not
exact. When looking at the individual damping ratios for each corner damping in the VBB3 suspension is significantly greater than the BB2 reference damping ratio.

Figure 3.32: Front Left Equivalent Damping Comparison

Even though the corner damping ratios for each corner are grossly different in magnitude the bounce, pitch and roll damping ratios for the VBB3 is nearly identical to BB2. This will insure that the vehicle should react in a very similar manor as BB2.
The unsprung mass damping is also critical to the vehicle performance. This damping affects the response of the wheel and hub assembly which affects the traction capability of the vehicle. The VBB3 unsprung mass damping ratios are very similar to the BB2 as
can be seen by the results. The front unsprung mass damping is slightly lower while the rear damping is slightly higher. These effects are relatively small differences that come from trying to get the VBB3 dampers to match the BB2 dampers. There are certain tradeoffs between the damping ratios for the bounce, pitch, roll and unsprung masses. Since the effects of the unsprung mass damping in land speed racing are not easily quantified the bounce, pitch and roll damping were of greater concern.

![Unsprung Damping Ratio's](image)

Figure 3.34: Front Left Unsprung Mass Damping Ratio Comparison

3.6. Anti-Roll Bar

An anti-roll bar is one of many methods that can alleviate a vehicle’s body roll. The anti-roll bar mechanically couples the left and right wheels of an axle through a torsion bar. The torsion bar spring rate can be tuned to meet the ideal vehicle roll characteristics.
The BB1 did integrate an anti-roll bar while the BB2 did not have to incorporate an anti-roll bar to reduce roll. The BB2 did not have to implement any countermeasures to mitigate roll of the vehicle due to the track width being 760 millimeters which is 102 millimeters greater than the VBB3. Also, the BB2 weighed significantly less at 5,800 pounds compared to the VBB3 at 8,000 pounds. The combination of a narrower track width and increased weight forced the implementation of a system to increase the roll stiffness of the vehicle.

3.6.1. Design Goals

The overriding goal of the addition of the anti-roll system was to reduce the roll capability of the vehicle so that maneuvering at the test track and racecourse was not of concern for the team or driver. By adding the anti-roll system the goal was to get the vehicle to have similar roll characteristics to BB2. Buckeye Bullet 2 was an extensively tested and raced vehicle. It was found to be extremely stable at the test track making maneuvers as well as driving at top speed on the Bonneville Salt Flats. This is why the BB2’s roll characteristics were set as the benchmark for the VBB3.

Not only did the anti-roll device have to be sized to give the vehicle similar roll performance as BB2 but it also had to fit in a very tightly packaged region of the vehicle. The gearbox was a fixed design that could not be modified and the placement of the springs and dampers could not be changed due to packaging limitations with the body of the vehicle.
There are many methods to add roll stiffness to the vehicle. Several methodologies were considered to reduce roll including a traditional anti-roll bar, tying the left and right dampers together as well as ferro-fluid dampers. Many of the options were considered but only the traditional anti-roll bar and damper coupling could be packaged or developed in the timeframe. After analyzing the feasibility of the two options it was decided that a traditional anti-roll bar would be the best option. Due to the space constraints, a U-shaped anti-roll bar design was selected for the application. The design also had ease of manufacturing and modifying its spring rates.

### 3.6.2. Anti-Roll Bar Sizing

To quantify the roll characteristics of the BB2 the roll gradient of the vehicle was considered [14]. This looks at the vehicles amount of roll for a given lateral acceleration.

\[
\frac{\phi_r}{A_y} = \frac{-Wh}{K_{\phi_F} + K_{\phi_R}}
\]

Where \( \frac{\phi_r}{A_y} \) is the roll gradient in degrees per g, \( K_{\phi_F} \) and \( K_{\phi_R} \) are the roll rates for the front and rear wheels, \( W \) is the vehicle weight and \( H \) is the distance from the roll axis to the center of gravity. It should be noted that the roll gradients from the front and rear both account for the installation ratio of the anti-roll bar. The installation or motion ratio of the anti-roll bar is a ratio that describes the relationship of the anti-roll bar motion to the wheel motion. The roll gradient is used since the corner stiffness could be greatly affected by the tire stiffness and to eliminate some complexities in the analysis the tires were not considered for either case.
Once the roll gradient of the BB2 was determined the desired roll rate could be determined. The roll rate is the torque that resists the body from rolling per degree of body roll.

\[ K_{\phi_{DES}} = \frac{W_{VBB3} \cdot H_{VBB3}}{(\frac{\phi_r}{A_y})_{BB2}} \]

Where \( K_{\phi_{DES}} \) is the desired roll rate of the VBB3, \( W_{VBB3} \) is the weight of the VBB3, \( H_{VBB3} \) is the distance from the center of gravity to the roll axis for the VBB3 and \( (\frac{\phi_r}{A_y})_{BB2} \) is the roll gradient of the BB2. To reach the desired roll rate the sum of the front and rear roll rates in addition to the contribution of the front and rear anti roll-bar roll rates were tabulated.

\[ K_{\phi_{DES}} = K_{\phi_{TARB}} + K_{\phi_F} + K_{\phi_R} \]

Where \( K_{\phi_{TARB}} \) is the total roll rate from the front and rear anti-roll bars. Since the vehicle has a nearly 50/50 weight distribution, the front and rear having small differences in roll rate and to reduces cost of manufacturing various parts it was decided that the front and rear anti-roll bars would be the same. Using this information the total roll rate from the anti-roll bars in front and rear can be rewritten in terms of the anti-roll bar roll rate.

\[ K_{\phi_{TARB}} = 2K_{\phi_{ARB}} \]

Using this information the roll rate for the individual anti-roll bar can be determined for the VBB3.

\[ K_{\phi_{ARB}} = \frac{1}{2}(K_{\phi_{DES}} - K_{\phi_F} - K_{\phi_R}) \]
With the roll rate for the anti-roll bar determined it has to be transformed into a spring rate for the anti-roll bar. Once a spring rate is determined analysis shifts from vehicle performance to mechanical component design [9].

\[
K_{\theta_{VBB3}} = \frac{K_{\phi_{ARB}} \cdot l^2}{IR^2 \cdot t_f^2}
\]

Where \(K_{\theta_{VBB3}}\) is the spring rate of the anti-roll bar, \(l\) is the length of the anti-roll bar arm from the torsion bar axis to the connection point of the link on the anti-roll bar arm, \(IR\) is the installation ratio of the anti-roll bar and \(t_f\) is the track width of the axle. The installation is a similar concept to the motion ratio except it is the motion of the anti-roll bar divided by the wheel motion.

The spring rate of the torsion bar is then used to determine the diameter and analyze the strength of the shaft. Since the VBB3 is so narrow the torsion bar was going to be fairly short while still requiring a large amount of deflection. This scenario would generate a large amount of stress in the torsion bar. A search for a material that would not yield from the high stresses as well as be able to deliver the required spring rate was required. The other consideration in choosing the material was finding a material that was less prone to fatigue failure.

Traditional steels that are easily acquired such as 4130 or 4140 are able to have the strength with a high enough heat treat but the materials become very brittle. To alleviate this, a higher strength steel was selected, 300M. 300M is very similar to 4340 with a slightly higher content of silicon and vanadium. These higher contents of the elements allow the material to be higher strength. Another group of materials that were considered
in the selection process were maraging steels. These are steels that are very high strength and have a much higher nickel content which make them more corrosion resistant. Although maraging steel would have met the design requirements, the cost and difficulty to acquire the material as well as the difficulty to manufacture removed it from the possible list of materials to use. In the end 300M was used as the torsion bar material because of the strength, manufacturability and availability. Other means would have to be used to alleviate the corrosion possibilities of 300M.

To get the higher strength properties of the material the torsion bars were heat treated to have Rockwell 52-54C. To further improve the strength, the torsion bars were then ground to final dimension and shot peened to reduce stress risers on the surface of the part.

3.6.3. Anti-Roll Bar Design

The anti-roll bar consists of three parts: the torsion bar, arms and links. The torsion bar is splined at both ends for the arms which are constrained by retaining rings. The anti-roll bar then ties into the rest of the suspension with the link the connects from the rocker to the anti-roll bar arm.

3.6.3.1. Torsion Bar

Many of the torsion bar parameters were selected from the analysis of the vehicle roll and determining the spring rate of the actual shaft. The connection between the torsion bar and arms still needed to be determined and methods to reduce the stress in the shaft were
also investigated. The connection between the torsion bar and arm was done using splines. This was done because splines are very efficient in transmitting torque and they are common in manufacturing. Keys were not used due to the size of the parts and the high stress. The spline selection was done so that the minor diameter of the spline would be greater than the shaft diameter. This was done so that there were no additional stress risers in the torsion bar. Once a minimum acceptable minor diameter was chosen, a range of spline options were selected from the Involute Splines and Inspection standards that conform to ANSI B92.1-1996 which is the standard for involute splines [15]. These ranges were checked to make sure that the splines would not fail under the loads. With the range of acceptable splines identified the machine shop that was selected to manufacture the parts was contacted and given the range of spline profiles. The goal of working with the shop was to determine their in-house machining capabilities to make the part turnaround time as short as possible and to reduce cost so that no additional tooling was required to manufacture the parts.

The spline fit also had to be specified to determine how loose or tight the fit between the mating splines would be. When considering the fit, the possibility of coatings and thermal variations have to be considered. Ideally, the fit between the splines would be line to line so that torsion bar and arms act like a single component. Since the shaft was made out of a material that is not very corrosion resistant the likelihood of a corrosion resistant coating being applied after manufacturing was good. To account for the coating thickness a couple thousandths gap was used between the mating splines.
3.6.3.2. Arms

The arms of the anti-roll bar arms have to connect the torsion bar to the anti-roll bar link. The arm is connected to the torsion bar using a spline but the arm still had to be retained on the shaft. The designed used to retain the arm on the shaft was to use retaining rings on each side of the spline of the arm. By putting the retaining rings in the arm it reduces any potential stress risers in the torsion bar and is in a region of low stress in the arm. Shims were used between the retaining rings and torsion bar to have better contact across the retaining ring face and controlling the tolerance stack up between the retaining rings of the arms. The inside shims were made into two part for assembly purposes. The connection between the arm and link needed to have very little play so shoulder bolts were used. The additional space could alter the performance of the vehicle as well as cause more force to be exerted on the bolts. Since the arms have to transmit the load of the link to the torsion bar, the arm is typically in bending. The other consideration of stress in the part was the splines that connect to
the torsion bar. Since the arms are fairly short the stresses and deflection of the parts were going to be lower so an exotic material did not need to be used. A range of materials including various steels, titanium and forms of stainless steels were considered. The wide range of materials was considered because the parts needed to be able to handle the stress but also be able to handle the corrosive environment. Most of the materials were able to meet the stress requirements. The deciding factors of the materials became corrosion resistance to the salt as well as manufacturability. All the steel alloys that were considered were not corrosion resistant which meant that they would have to have a corrosion resistant coating applied to them along with the torsion bar which would make holding the tolerances on the spline much more difficult. Titanium has excellent corrosion resistance and strength but it is more difficult to manufacture. Stainless steels typically have good corrosion resistance and strength while manufacturability differs for each alloy. To get the corrosion resistance and strength needed with good manufacturability 17-4PH was selected for the material. Traditionally 17-4PH is similar pricing with titanium but easier to manufacture.
3.6.3.3. Links

The anti-roll bar links are two-force members that connect the anti-roll bar arms to the rocker. To make the links easy to manufacture a simple design that uses tubing with weld-in threaded steel inserts with rod ends threaded in on each end. The rod ends allow for adjustability in length as well as misalignment because the connecting ends of the rocker and anti-roll bar arms are not on the same plane. For easier adjustability when the link is installed the weld-in inserts on one end of the link is right-handed thread while the other end is left-handed thread. This allows for the tube to be rotated and it will either thread in or out the rod ends.

There were a few structural considerations in the design of the link. The most likely failure mode of the tube would be from buckling so an appropriate diameter and wall thickness had to be selected. Also, the rod ends had to be sized to handle the loads. High misalignment rod ends were used in this situation since the two connecting points were
on different motion planes as well as the higher load capacity. Finally, the bolts needed to be checked to insure that the loads would not yield the bolts. The design of the rocker and anti-roll bar arm had the bolts in double shear, which greatly reduced the bolt diameter requirements. A larger bolt was selected because the rod end thread diameter was the weakest point in the design. The thread diameter of the rod end is proportional to the bore diameter of the rod end so the bolt had to be increased to increase the thread diameter of the rod end.

3.7. Tires

There are two typical types of tires that have been used in racing; bias-ply and radial tires. There are several tradeoffs to each type depending on the application. Most consumer tires today are radial tires due to their life, low rolling resistance, grip and reduction in cabin noise. The bias-ply tires on the other hand tend to have lower impact harshness.

Bias-ply tire construction consists of several layers of cords laid at alternating angels diagonally across the contacting face of the tire. The angles of the plies typically range from 25-40° from the plane of the wheel. The smaller angles are traditionally used for stability and speed requirements. Due to the construction of the tire, the stiffness of the tire is more dependent on inflation pressure than a radial tire.

Radial tire construction predominantly has cords running between the bead wires, perpendicular to the plane of the wheel. On top of the radial plies are belts of cords that typically have adjacent layers with alternating angles ranging from 10-30° from the plane
of the wheel. The tread layer is then applied on top of the belts. This construction allows for a greater contact area between the tire and the road which allows for better traction capabilities.

![Bias-ply Tire and Radial-ply Tire](image)

Figure 3.37: Bias and Radial Ply Construction Comparison [11]

There are two tires that are traditionally used by land speed racers on the Bonneville Salt Flats. The two most common tires are the Goodyear Eagle Land Speed tire and the Mickey Thompson Bonneville tire especially for vehicles attempting to run over 300 miles per hour. These tires are both bias-ply tires that have no tread and have differing sizes available.

3.7.1. Buckeye Bullet 1

The Buckeye Bullet 1 initially used Goodyear Front Runner drag racing tires on the front axle and the Goodyear Eagle Land Speed tires on the rear axle. The Front Runners are
treaded tires that are used as steering tires in drag racing. The Eagle Land Speed tires were used on the rear because the rear axle was powered while the front axle was not powered but used to steer. When the vehicle began to reach higher speeds closer to 300 miles per hour, the team began to have issues with the tires blistering. Because of the tire concerns the rear tires were switched to the Mickey Thompson Bonneville tires which was the tire that was most commonly used for the driven tires on other land speed vehicles that were traveling over 300 miles per hour. The team used the 24.5 inch diameter tires due to the packaging constraints of the body.

3.7.2. Buckeye Bullet 2

The BB2 design incorporated the Mickey Thompson Bonneville tires from the beginning. Due to the issues that the BB1 had seen previously with blistering on the Goodyear Eagle Land Speed tires and the success of using the Mickey Thompson tires at higher speeds in combination with BB2 weighing significantly more than BB1, initial designs incorporated the Mickey Thompson tires. The BB2 used the smallest version of the Bonneville tire, which is nominally 24.5 inches in outer diameter, 7.5 inches in width and uses a 16 inch wheel. The tires were run with great success on the BB2 with no major issues.

3.7.3. Venturi Buckeye Bullet 3 Tire Selection

When it came to selecting the tires for the Venturi Buckeye Bullet 3, the selection of land speed tires were the same as they were with the previous vehicles. The Goodyear Eagle
Land Speed tire and the Mickey Thompson Bonneville tire were compared again for the latest vehicle. One of the deciding factors in the selection of tires was the overall size of the tires. To reduce aerodynamic drag the frontal area of the vehicle needed to be reduced as much as possible. Besides the drivers dimensions and the battery pack sizes the tires were the next largest factor in increasing the frontal area of the vehicle. If the team could use a smaller tire, the height and potentially the width of the vehicle could be reduced. The total travel of the suspension would have to be considered as well as the growth of the tires at high speeds. For the Mickey Thompson tires, the team had seen growth of up to 1.25 inches of growth in diameter when at 300 miles per hour. At 400 miles per hour the growth would most likely be greater so adequate spacing between the body and tire had to be added. Due to the battery pack size and driver compartment configurations the large tires from both Goodyear and Mickey Thompson had to be eliminated from the selection. This left the 21, 23 and 25 inch Goodyear Eagle Land Speed tire and the 24.5 and 26 inch Mickey Thompson Bonneville tire. With further analysis of assumed tire growth for the tires, the 25 inch Goodyear and 26 inch Mickey Thompson tire would add a significant increase in frontal area in order to have proper clearances. This limited the selection to the 21 and 23 inch Goodyear Eagle Land Speed tire and the 24.5 inch Mickey Thompson Bonneville tire.

One of the biggest considerations in the tire selection was finding a tire that could support the load of the vehicle. Early on in the development of the vehicle it was realized that the vehicle would weigh between 7,000 and 8,000 pounds. This would mean that the static corner weights of the vehicle would be between 1,750 and 2,000 pounds. When
considering the spec sheets for the two tires the only Goodyear Eagle Land Speed tire that was not close to being able to support the weight was the 21 inch that had a max load of 1,200 pounds [16]. Even though the maximum load for the tires was still below the projected corner weights, testing and collaboration with Goodyear engineers could potentially prove the tire to be capable in the application. The Mickey Thompson tires did not have a maximum load listed but after consulting the tire engineers with Mickey Thompson it was found that the load would not be an issue. After looking at the size and load capabilities of the tires, the selection was narrowed down to the 23 inch Goodyear Eagle Land Speed tire and the 24.5 inch Mickey Thompson Bonneville tire.

The last major consideration in the tire selection was the speed limit of the tires. The Goodyear tires have a limit of 300 miles per hour but there have been vehicles that have run the tires at speeds higher but they were also lighter vehicles than the VBB3. The Mickey Thompson tire does not have a maximum speed rating. It is the most common tire used by some of the world’s fastest wheel-driven vehicles. Most of the land speed cars that are running over 400 miles per hour use the Mickey Thompson Bonneville tire. Due to the weight of the vehicle and the high speeds, the tire that was selected for the VBB3 was the 24.5 inch Mickey Thompson Bonneville tire.

3.7.4. Land Speed Tire Considerations

There are many effects that need to be considered for land speed racing tires compared to consumer tires. The tires are affected by the high rotational speed, which cause the tire to grow as well as increase in stiffness. Also, the tires are highly reliant on the inflation
pressure for their stiffness. An understanding of these effects will allow for more consistent and predictable performance.

3.7.4.1. Tire Growth

Bias-ply and radial tires both exhibit tire growth with increased vehicle speed but the bias-ply tire sees a greater increase. This means that the rolling radius of the tire is changing with the velocity of the vehicle speed. The increase in rolling radius is caused by a decrease in tire deflection at the contact patch and the tire circumference increasing.

The Buckeye Bullet land speed vehicles have seen tire growth in all the previous cars so special consideration had to be taken with the VBB3 and its higher speeds. The tire growth was calculated for the BB2 and BB2.5 to begin to understand the increase in
rolling radius. The calculated rolling radii for the various runs were very inconsistent and trends were difficult to identify. Fitting the data and even trying to make functions to predict the tire growth of the vehicle were nearly rough estimations that had a significant amount of error in their formulation.

![Cooper and BB Filtered Tire Radius Calculation](image)

**Figure 3.39**: Tire Growth Results [17]

To try and further understand the tires it was determined that the tires needed to be tested in an attempt to control some of the variables that effect the growth of the tire. The team initially looked at the possibility of using a belts tire testing machine to test the tires since it best represented a flat surface. The difficulty in trying to use a belt tire tester is the limitation of speed. Some of the highest speed belted tire testers are still limited to
roughly 250 miles per hour. After further investigation the only way that the team would be able to test the tires loaded to high speed was by using a drum type tire test system. These types of systems are not as ideal because it doesn’t truly represent the contact between the ground and tire because the tire is contacting a round surface. With a larger drum diameter the effects are decreased but there are still concessions that have to be made. Significant effort was put into finding a facility that could potentially test the tires up to 400 miles per hour while loading them with the projected vehicle loads. The only facility that consistently came back as a result was the Landing Gear Test Facility at Wright Patterson Air Force Base in Dayton, Ohio. This facility had the capability to load the tires while spinning the tires to 350 miles per hour. This speed is still below the vehicles targeted top speed but it would allow for a significant increase in understanding of the tire dynamics and wear. After visiting the facility and formulating a test plan it was determined that the facility could not do the testing in an adequate amount of time that fit into the team’s budget. Finally, the team was able build a relationship with Cooper Tire Company to help with the testing and analysis of the tires. Cooper Tire was willing to conduct testing free of charge as long as they could conducted the tests on their timeline and on their own machines. The machine that was used to conduct the tests was an external drum test machine that had a top speed of approximately 215 miles per hour. The team then supplied an estimated velocity profile for the vehicle under acceleration that the machine would try to mimic. It was also decided to load the tires to 2050 pounds of force to account for the aerodynamic loads on the vehicle. The tire pressure was set at 90 pounds
per square inch. This same style of test was also run for the BB2 and VBB2.5 corner weights as well.

The testing showed that the rolling radius increased in a non-linear manor. The tire test machine was able to test the tires up to 217 miles per hour and held close to top speed with nearly constant load for several seconds. An interesting occurrence to note was that when the tire was being held at nearly constant top speed of the machine and constant load the rolling radius continued to increase. This phenomenon could represent that the tire has a “break-in” period where the cords and rubber are attempting to find an equilibrium state that allows the tire to grow more significantly. This occurrence was also in rolling radius data from previous vehicles. This phenomenon has been verified by other land speed racing teams that use the Mickey Thompson Bonneville tire as well as speaking with IndyCar engineers that dealt with the bias-ply tires before the radial tires were used.
Most of the land speed cars that are going over 400 miles per hour mount their tires and then spin them unloaded to roughly 7,000 revolutions per minute. This process exposes the tires to the high centripetal forces, which force the plies of the tire to find an equilibrium point. Once the equilibrium point is reached the tire is maintained at constant pressure. If there is a significant reduction in pressure the effects produced by the spin testing are lost. This methodology allows for the dominating force, the centripetal force to be exerted on the tire but is still does not load the tire.

Another method for “breaking-in” the tire used by IndyCar when they were using bias-ply tires was to put the wheel and tire on a machine that would load the tires to maximum expected loads and rotating them slowly. This would deform the tire, which would then

Figure 3.40: Comparison of Two Tires Tested Identically
allow the plies of the tires to move into an equilibrium state. The downside of this method is that the tire is not exposed to nearly all the forces that it would see at top speed. This method could be used on the Mickey Thompson Bonneville tires but further analysis would be needed to determine the effectiveness of the methodology.

3.7.4.2. Tire Matching

The VBB3 does not use a differential but locks the left and right rotation together with a spool, or locked differential. This can add complexity to the vehicle setup to try and reduce the wear of the tires from one side to the other. With the spool, the rolling radii need to be ideally the same from left to right to prevent significant tire slip. This would mean that neither tire would have to slip while the other was rotating. It is very difficult for the team to get identical tires due to manufacturing differences so a method had to be determined to match the available tires to sets for an axle. Since the tires are so expensive as well as the limitation in number of wheels available to have tires mounted on, the team has to make do with what is available to them. The differences in tire diameters were never truly noticed until the VBB3. This could be due to the fact that the tires are more heavily loaded or the manufacturing process was changed.

The process that was implemented in the 2015 racing season was to compare the circumference of the inflated tires while no load was being applied. Ideally, the best method to match the tires would be to check the rolling radius while loaded to the vehicles static weight and match the rolling radius but this was not easily accomplished.
This is why the measurement of the circumference was the quickest and most cost effective method at the time.

There were several procedures used to maintain consistency in the preparation and measurement of the tires. The first step was to inflate all the available tires to 90.0 pounds per square inch +/- 0.2 pounds per square inch. The tires would then be allowed to rest for five to ten minutes to allow them to stretch and find their equilibrium point. This required that the tires all rest in the same environmental conditions. Once the resting time was over the pressures were all checked again to make sure that all the tires were still within tolerance. Once all the checks were done the tire circumference was measured using a flexible tape measure that measured to the sixteenth of an inch. This was deemed to have good enough resolution because the diameter of the tire would only be off by approximately 0.020 inches in diameter. This was suitable since the measurement of the circumference is already an estimation of the rolling radius. Once all the radii of the available tires were checked, pairs of similar circumference tires were then put together. Ideally, the circumferences would be within one eighth of an inch with maximum differences being approximately a quarter inch. The reason for the fairly large tolerance was due to the number of wheels and tires that the team had available.

3.7.4.3. Inflation Sensitivity

Another way that the rolling radius can be altered is by adjusting the inflation pressure of the tire. As mentioned earlier because of the construction of the tire, the bias-ply tire is
much more reliant on the inflation pressure to carry the load. This means that adjusting the inflation pressure could alter the rolling radius. To study the effects of the inflation pressure on rolling radius and tire stiffness a similar testing procedure that was used to find the stiffness of the tires at various loads was used for various inflation pressures. The tests that were conducted adjusted the inflation pressure by 10 pounds per square inch starting with an initial inflation pressure of 90 pounds per square inch and finishing at 120 pound per square inch.

![Tire Load vs. Deflection](image)

**Figure 3.41: Tire Load as a Function of Tire Deflection**

The testing shows that the tire stiffness increases with inflation pressure. This was to be expected but what was interesting was that the increase in stiffness was not linear. If the
tire stiffness was linearly related to inflation pressure it would be expected that the four different inflation pressures curves would be evenly spaced with slightly different stiffness’s. Some of the non-linear effects could be because of the tire being loaded on a large diameter drum but this most likely plays a smaller role compared to the tire inherently acting non-linearly.

![Tire Stiffness as a Function of Inflation Pressure](image)

Figure 3.42: Tire Stiffness Sensitivity to Inflation Pressure

### 3.8. Conclusions

The development of the VBB3 suspension has pushed the capabilities of the team and sponsors to deliver unique solutions to very difficult problems. The entire suspension had to be packaged in an extremely small track with higher speeds and loads than any of
the previous Buckeye Bullets had to deal with. It also pushed the team to gain a better understanding of every component in the system and its effect on the vehicle performance.

The vehicle debuted in the summer of 2013 where it began its initial low speed testing. By the summer of 2014, the vehicle was testing at higher speeds and eventually made its debut on the Bonneville Salt Flats hitting a top speed of nearly 270 miles per hour. Finally, in the summer of 2015 the vehicle had done extensive high speed testing on asphalt at the Transportation Research Center to prepare for the salt flats. By August, the vehicle was again able to run on the Bonneville Salt Flats but in very bumpy conditions. The vehicle in the end reached a top recorded speed of 288 miles per hour. After some initial changes after the first year of racing the suspension has been one of the most consistent systems in the vehicle that has reached its design goal. The vehicle has been tested much more extensively and harder than any of the previous vehicles. The team is able to do the more extreme testing because of the consistent and predictable performance of the suspension that the team and driver are confident in. The racing in the summer of 2015 further validated the suspension performance by maintaining good traction and handling while on a very rough track. The vehicle encounter several very rough patches of track that did not give any unwanted steering feedback while the drive was still able to maneuver the vehicle across the course with confidence to try and find the smoothest route down the track.

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Chapter 4: Vehicle Dynamics Model Development and Stability Analysis

4.1. Modeling and Simulation

The use of modeling and simulation tools is critical to the success of most vehicles produced today. These mathematical tools apply the equations and laws that describe the various subsystems and looks at their interaction as a complete system. Since only one vehicle is produced, it is critical to be able to understand how the vehicle will perform and how to control it before it’s ever built. The simulations are also used to define goals and targets for the design of the vehicle. Having only enough time and money to build one racecar further emphasizes the need for accurate models that describe the vehicle so that no major modifications have to be done. To insure that the models are accurate, quality data has to be gathered from physical testing and analysis.

4.1.1. Simulation Objectives

The VBB3 has required a much more in depth development of modeling and simulation. A significant amount of effort was placed in the development of models for the powertrain, aerodynamics and the full vehicle. The full vehicle simulator that was created for VBB3 begins to consider the vehicle dynamics but lacks the depth of analysis to consider the stability and ride characteristics of the vehicle. This is why a more
extensive model needed to be developed, especially with the vehicle going significantly faster than any of the other vehicles.

The first goal of the model was to make a comprehensive vehicle dynamics model of the VBB3 that could be used to run various vehicle maneuvers for racing or test track scenarios. This model will consider all the systems of the vehicle that affect the handling and performance of the VBB3.

When developing the vehicle model, a major goal is to develop a modular vehicle model that will be able to not only allow easy modifications to the vehicle model but is also robust so that the model will be able to accept various models. An example of this is if the team would be interested in looking at the performance of a new powertrain or suspension design.

In order to better predict the vehicle handling and stability, a much more comprehensive model of the vehicle suspension and kinematics is developed. The models incorporate the full vehicle suspension geometry and kinematics as well as incorporate the shocks, anti-roll bars and tires that heavily effect the vehicles overall performance and handling. This allows for accurate kinematic models that can be used to determine displacements and forces in members.

To develop all the various models, significant amounts of data are utilized from previous vehicles, track testing, Bonneville racing events as well as data collected from numerous bench tests of vehicle systems and components. The data also includes data from computational analysis as well. This data will be used to develop vehicle parameters as well as maps to more accurately model the performance of the various systems.
Since a great deal of effort is being put into a much more encompassing vehicle dynamics model it would be beneficial to be able to put the vehicle in various driving or racing conditions such as adding crosswinds or running the vehicle at different tracks or facilities. This will allow for a better understanding of how the vehicle will handle and perform before the team decides to attempt any situation with the actual vehicle as well as know the vehicle limits before attempting a specific maneuver.

Finally, the end goal of the vehicle model is to make a model that can be a basis for future Buckeye Bullet projects as well as other student projects at The Ohio State University. The modularity of the model design will allow significant changes to the vehicle designs which will allow the teams to very easily introduce changes and see how the proposed changes effect the vehicles performance in a given situation or event.

4.1.2. Simulation and Modeling Tools

There are many mathematical modeling tools that are available to use to reach the goals presented previously. There were many considerations that went into the choice of tools selected. One of the most significant limitations was the availability of the software. There are many software packages that are available to conduct the simulations but they were not readily available. Also, there was not one particular software offered that met all the various needs so more than one piece of software needed to be utilized. The search had to be expanded to see which software were able to work together well while still meeting all the goals. What was settled on was to use MATLAB™, CarSim™ and
SuspensionSim™. Using the combination of these three pieces of software will allow for all the various goals to be met.

SuspensionSim™ is a multi-body simulation software that is used in the analysis and modeling of the suspension kinematics. This software allows for easy changes in geometry that can then be analyzed for their effects on the vehicle performance. The software is also designed to interface with CarSim™ and MATLAB™ which makes tying all the programs together more seamless.

CarSim™ is a program that looks at vehicle system interactions as well as the performance of the vehicle. This is a program that is commonly used in industry to analyze the vehicle’s performance capabilities for a given procedure. The software allows for numerous events or procedures for the vehicle to be tested in including acceleration, braking, double lane changes as well as user defined procedures. The program brings in all the submodels and information from MATLAB™ and SuspensionSim™ and uses that information to run the vehicle simulations. The data from the simulation can then be exported back to MATLAB™ for quick data processing.

Finally, MATLAB™ is being used for the implementation of the various powertrain models as well as its ability to be used as a post processor of the data generated from the simulations. The majority work completed in MATLAB™ will be develop models for the batteries, inverters and motors to more accurately determine the power capabilities during the simulations.
4.2. Model Configuration and Parameter Identification

Parameter identification is critical to the development of an accurate model. Identifying and gathering parameters can be one of the biggest challenges in developing models due to the complexities of gathering the information needed. In most cases this means the data collection from experiments, vehicle testing and racing to further improve the accuracy of the models. This is further accentuated by the high speeds of the vehicle, which require high speed, power and load testing to be done which those capabilities may not exist at the time.

4.2.1. SuspensionSim™

The use of SuspensionSim™ is due to its ability to be a flexible multi-body solver. This allows for very unique suspension geometries to be implemented with relative ease. Conveniently, the program supplies many examples that can be modified for whatever
application or a model can be created from scratch using points in three dimensional space. To develop the model for the VBB3 a specially developed model for an open-wheel race vehicle was used due to the similarity of the suspension setup of being a fully independent suspension that use rockers to connect to the shocks. Since SuspensionSim™ is so flexible, the X, Y, and Z coordinates just had to be specified for all the connections and rotation axes and the program was able to animate the suspension as it was designed in CAD. This flexibility to adjust the various points in the suspension so easily makes it very helpful in the design and analysis of a vehicle’s suspension.

The open-wheel racecar model that was used also implemented a third spring and T-bar. This is very common in high performance open-wheel racing and is used to improve suspension performance and improve roll characteristics of the vehicle. To improve the roll characteristics of the VBB3 a U-shaped anti-roll bar was implemented in both front and rear of the vehicle. This meant that the model needed to be modified to represent the U-shaped anti-roll bar. To do that new points and bodies needed to be defined.

In order to integrate the anti-roll bar into the SuspensionSim™ model there were two bodies that needed to be created, the anti-roll bar and the anti-roll bar link that connects to the rocker. This was done in the model by creating two new joints. The first joint for the anti-roll bar arm and link was created using a dataset with two joints. The link was connected to the rocker body with a specific location, which was then connected on the other side of the link to the anti-roll bar arm. The link was specified to have no stiffness and was kept at a constant length. The anti-roll bar arm was connected to the chassis of the vehicle at the rotation axis of the anti-roll bar. The arm also had the connection to the
link as well. Another joint was then created for the anti-roll bar torsion bar to connect the two anti-roll bar arms. This was given no stiffness because there is greater interested in developing an understanding of the motion in SuspensionSim™, which can then be used to develop a force calculation generated by the anti-roll bar in CarSim™.

Figure 4.2: VBB3 Front SuspensionSim™ Model

Once the kinematics of the suspension were properly implemented into SuspensionSim™, calculations and tables had to be generated that would then be exported and used by CarSim™. CarSim™ has predefined suspension models that allow for coefficients or functions to be defined for the camber, caster and toe but it does not have the capability to model the kinematics which is where SuspensionSim™ takes over. Kinematic models are developed in SuspensionSim™ and then used in CarSim™. This is where certain parameters have to be defined and calculated in SuspensionSim™ for
each of the various suspension models. Also, since there is an addition of an anti-roll bar it requires that additional tables are developed to then be exported to CarSim™ where further calculations are made to determine the forces that they impart. The major parameter that was needed was the rotation of the anti-roll bar arm about the torsion bar axis as a function of jounce. This allows for the anti-roll bar forces to be calculated and then added to the shock forces at the wheel. Other parameters that were needed in CarSim™ included suspension camber, toe, lateral translation, spring deflection, damper deflection and bump stop motion all as functions of suspension jounce.

4.2.2. CarSim™

As described earlier, CarSim™ is used to analyze the interaction between the various subsystems and how it effects the vehicle performance. It incorporates models from SuspensionSim™ and eventually Simulink to more accurately model the performance of the VBB3. The software will be used to validate the performance and handling of the VBB3 and in the future be used as a basis for future Buckeye Bullet projects. There were many parameters that were determined through analysis and modeling techniques but there were still parameters that needed to be determined from testing the vehicle. Some of these parameters include center of gravity location as well as the roll, pitch and yaw inertias.
4.2.2.1. Vehicle Center of Gravity and Inertias

The vehicle mass and center of gravity are some of the most critical pieces of information when looking at vehicle dynamics. The mass of the vehicle was determined by placing vehicle on scales that were level with one another, which also allowed for the longitudinal position of the center of gravity. The lateral position of the center of gravity was considered to be the centerline of the vehicle since the vehicle is very symmetrical in design. The height of the center of gravity was a much more difficult task to determine. The most common method of determining a vehicle’s center of gravity height is by raising one of the axles of the vehicle and measuring the weights on each axle. Doing force and moment calculations the center of gravity height can be determine. This method could have been used but due to the low ground clearance and the need to determine the roll, pitch and yaw inertias of the vehicle another methods was used. Since the VBB3 is so large it makes it more difficult to get some of the centers of gravity or inertias. Luckily, the team was able to find a facility that does vehicle testing that could handle vehicles as large as the VBB3 and bigger. The system that was used to determine the parameters was developed by SEA Ltd., which has supported the team before with similar testing before. The test bed that was used was designed for large heavy-duty military vehicles. To measure the vehicle center of gravity, the vehicle was lifted and secured to the deck of the test rig. The deck of the test rig was then lifted by a carriage and allowed to reach equilibrium. Weights were then added to one end of the deck at a known distance and the platform angle and displacement were recorded. Using
this information and adding various weights to insure the accuracy of the result the height of the VBB3 CG was found to be 351.5 millimeters above the ground plane.

![Figure 4.3: VBB3 on SEA Ltd. VIPER II](image)

After the CG location was determined the process of determining the vehicle inertias could be conducted. In the same orientation that the vehicle was in for the center of gravity height measurement the vehicle was then excited and the period, angle of motion and displacement of the vehicle were measured to determine the pitch inertia. This process was done several times in order to guarantee the accuracy of the results. The roll inertia was calculated with the same philosophy but the test rig deck was lowered onto a platform that allows the deck to oscillate from side to side. The platform was sprung so multiple oscillations could be captured in one testing cycle. The yaw inertia would have been calculated in a similar method to the roll inertia but with the deck on a platform that allowed the deck to yaw with the vehicle. This was unfortunately unable to be calculated.
due to the vehicle only resting on one side of the deck. Since the vehicle has similar proportions in height and width, it was determined that the yaw and pitch inertias would be considered the same.

<table>
<thead>
<tr>
<th>Roll Inertia (kg-m^2)</th>
<th>Pitch Inertia (kg-m^2)</th>
<th>Yaw Inertia (kg-m^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>81.7</td>
<td>29696.5</td>
<td>29696.5</td>
</tr>
</tbody>
</table>

4.2.2.2. Aerodynamics

The aerodynamics of the VBB3 are very critical and play a major role in the top speed capability of the vehicle as well as the stability. In order to develop the aerodynamics of the vehicle, extensive research was done in computational fluid dynamics (CFD) to accurately model the aerodynamics of the vehicle. Due to the high speed goals no wind tunnel had a fast enough rolling road in combination with the wind speed capabilities for wind tunnel testing to be a viable solution or method of validation. The aerodynamic design of the vehicle had to rely solely on CFD analysis. Since the analysis was so critical there were two master’s theses written on the topic of the vehicle aerodynamics. One area of analysis was to begin to parameterize the aerodynamics for a given slip angle [18]. The analysis was run from neutral to slip angle of eight degrees. The coefficients of drag, side force, and lift were determined as well as the pitch, roll and yaw moment coefficients for the various degrees of slip and speeds.

Also, needed for the analysis was the frontal area of the vehicle, which is the area of the vehicle when looking at the vehicle in a front or rear view. This is used to calculate the aerodynamic forces of the vehicle. CarSim™ also needed the density of air. The density
can greatly effect of aerodynamics of the vehicle. If the vehicle was run at high altitude there would be a reduction in drag on the vehicle as compared to if it was run close to sea level.

4.2.2.3. Powertrain

The VBB3 powertrain is a very complex system that is an area of ongoing research and development. For the purposes of vehicle dynamics modeling the powertrain was kept internal to Carsim™. In the future the powertrain will be used from an external Simulink model that incorporate the motor, inverter and battery dynamics.

The initial powertrain consisted of a motor model that used the torque speed curve from the motor spec sheet at one hundred percent throttle. The torque was quadrupled since the internal motor model of CarSim™ only has one motor the power of both VBB3 axle motors had to be combined into one. The power is transferred through the transmission and into a transfer case that splits the power evenly front to rear. The power is then finally put through differentials at the front and rear and transferred to the wheels. Along with the motor torque speed curves of the motor, the inertia of the rotors were needed to look at the dynamics of the motor more accurately.
A clutch model was included but since the model for the clutch was developed for a normal friction clutch it was difficult to implement the overrunning mechanical clutch without creating a custom clutch using VS commands in CarSim™ or using an external Simulink model. For the cases of getting a modeling to understand the vehicle dynamics and stability the clutch dynamics were not critical so a high torque capacity was specified for the clutch.

A two-speed transmission and its gear ratios were specified as well as the inertias of the gear sets in the transmission model of CarSim™. In the transmission model the shift duration is also selected. The team is targeting to have shift times between 0.5 and 1 second.
In order to make the model four-wheel drive, a transfer case was integrated in the model. In reality there is not transfer case and there are separate powertrains for the front and rear axles but due to simplicity and the lack of need for a more realistic powertrain model the transfer case was implements. The transfer case was specified to be always locked in a gear ratio of 1:1 with 100% efficiency. This would allow a somewhat simplified model to represent the four-wheel drive capabilities of the VBB3.

After the transfer case the power is then diverted to the front and rear differentials. The VBB3 has a type of differential but it is part of the transmission, commonly known as a transaxle. The final drive ratios and inertias were identified for both the front and rear differentials. The differential was set as locked to mimic the solid spool in the VBB3.

4.2.2.4. Tire Model

The tires of the VBB3 are very specialized for land speed racing as described earlier. The tires are bias-ply pneumatic tires that are inflated to 90 pounds per square inch. There are many parameters that are needed to begin to model a tire but due to the extreme speeds and forces in combination with the construction of the tire, has made the task even more difficult. Due to the lack of data from vehicle and tire testing it was decided for the time being to use a generic tire model that has modifications in the tire growth and stiffness. With more time and continual work with testing facilities, more data can be procured about the tire to eventually make a model.

One of the parameters needed for the tire model is the tire stiffness especially when considering tire growth. Some tires are very consistent in various conditions that a
constant spring rate can be assigned to them but due to the extreme conditions of the VBB3 and construction of the Mickey Thompson tire, the stiffness is not constant. Instead the tire stiffness changes significantly with the rotational speed. This presents two very different issues when the effects are considered. First, the stiffness changes which alters the handling of the vehicle because springs and dampers account for the stiffness and damping of the tires. The other effect is that tire rolling radius changes with the velocity of the vehicle. To begin to understand the effects of the tire stiffness controlled tire testing needed to be done. The first challenge was to find a facility that was willing to help test the tires and the second challenge was to locate a group that could test the tires to over 400 miles per hour. As previously mentioned there were very few facilities that could test the tires to high speeds. Due to cost, time and availability, Cooper Tire Company was able to do some initial testing of the tires. Some of the initial tire testing included fixed load tests from zero to 200 miles per hour, static tire stiffness and stiffness sensitivity to inflation pressure. These tests were beneficial in helping the team understand the tires but there was still not enough testing to be able to build a tire model. Since there are limitations of tire tests and vehicle testing, it was decided to use an existing tire model in CarSim™ and modify the tire force calculations to account for the increased tire stiffness and the tire growth due to vehicle velocity.

The model selected as the basis for the Mickey Thompson Bonneville model was the Touring Car model internal to CarSim™. Due to the lack of tire data a generic tire model was selected for the time being until further tire testing is conducted. With a base model in place modifications could be integrated to account for tire growth. There are two
different effects that are encompassed in the tire growth; the physical changes in the radius as well as the change in tire stiffness due to the centripetal forces. To account for these two different effects, estimations for radius change and stiffness due to speed had to be developed. This was accomplished by using tire test data collected by Cooper Tire Company on a large diameter roller. The test procedure specified to hold the tire at a fixed load and accelerated the tire up to over 200 miles per hour from stand still while measuring the loaded radius of the tire. This was conducted three separate times with the same tire all at different loads. The loads specified were for the corner weights of the BB2, VBB2.5 and the VBB3.

Figure 4.5: Tire Rolling Radius at Varying Loads
In a tire model the force of the tire is described by the stiffness of the tire multiplied by the change in the initial unloaded tire radius and the loaded radius. Since the tire growth effects both the loaded radius and the stiffness of the tire a model for the tire radius and stiffness had to be created.

\[ F_T = (R_0 - R_L)K_T \]

To make a model that accounts for the velocity of the tire, the radius and loads had to be considered at various speeds. The radii of the three loading conditions were plotted as a function of the forces at two speeds, stationary and 200 miles per hour. First, the plots were analyzed to determine if the characteristic were linear or nonlinear. In the case of both plots at the two speeds they both appeared to be very linear.

![Figure 4.6: Tire Characteristics at V = 0 m/s](image-url)
Since the plots are very linear the data points were linearly fit. The y-intercept of the curve fit would represent the effective unloaded tire radius at the two speeds. The slope of the curve fit would represent the inverse of the tire stiffness as well. With both the curves being linear at zero and 200 miles per hour a linear relationship was assumed for the velocity and its effects on tire stiffness and loaded radius. This is a very large assumption that could prove to change with higher speeds but with the data that is available, the linear assumption was made. The linear assumption allows for linear equations to be developed for the initial tire radius and tire stiffness.

\[ R_0^* = R_0 (1 + aV_T) \]
\[ K_T^* = K_T(1 + bV) \]

Where \( a \) is the linear radius due to speed coefficient, \( V \) is the velocity of the tire and \( b \) is the linear stiffness due to velocity coefficient. The linear coefficients were determined by comparing the values of the radii or stiffness for the two velocities analyzed.

\[
a = \frac{R_{0V200} - R_{0V0}}{V_{200} - V_{0}}
\]

\[
b = \frac{K_{T200} - K_{T0}}{V_{200} - V_{0}}
\]

The linear radius coefficient compares the initial radii at zero and 200 mph and is divided by the change in velocities. The same idea was applied to the tire stiffness coefficient calculation.

Now with the ability to calculate the modified radius and stiffness, the new tire force can be calculated.

\[
F_T^* = (R_0^* - R_L^*)K_T^*
\]

The new tire force calculation replaces the original force calculated by CarSim™ for every tire. This again is not the most ideal method of implementing the growth model especially with the minimal data available to build a model off of. Also, it should be noted that the tire testing was conducted on a large roller which deforms the contact patch of the tire differently than the flat ground. This could affect the overall performance of the tires and the parameters that were gathered from the tests.
4.3. Crosswind Stability Analysis

Having a vehicle model allows the team to be able to analyze different characteristics and performance metrics of the VBB3 in various situations. This enables the ability to analyze various tracks and conditions. One condition that has been of interest is the effects of crosswinds on the VBB3’s stability while racing across the salt flats.

The VBB3 has run on the salt flats for the past two summers with inclement weather conditions that have left the track less than desirable. In 2014, the VBB3 was making some higher speed runs over 200 miles per hour and the driver found that the vehicle appeared to be “hunting” across the track. He mentioned that it was a similar experience that he had with the BB1 when the center of pressure and center of gravity were not space correctly. During the run there was a recorded crosswind at the timing station that could indicate a fairly strong crosswind greater than 10 miles per hour.

For 2015, the vehicle was reanalyzed aerodynamically to determine if the relationship between the center of pressure and the center of gravity needed to be adjusted. To gain slightly higher stability the tailfin was lengthened by an inch. Effects of the crosswind further emphasized the need to understand the crosswind effects on the vehicles stability through the duration of a run. This could also be used to determine when an unsafe point was to run the vehicle.

To analyze the effects of the crosswind two different types of tests were developed: a fixed steer test and a vehicle targeted steering. These two approaches would begin to show the effects of the crosswind on the vehicle.
First, a test procedure had to be created in CarSim™, which by chance had some built in crosswind tests. The crosswind speed and direction were defined as well as if gusts of wind wanted to be included. For the purposes of the analysis the wind was held constant through the test. Also selected was the direction of the wind. For the analysis the wind was applied perpendicular to the track.

The track coefficient of friction could also be modified to be constant or varying over a given distance. Since the salt flats surface varies but in general has a relatively low coefficient of friction the model selected a constant coefficient of 0.5. This is a fairly large assumption since the track varies significantly but with the lack of information it seemed the best method to see the effects of crosswind on the vehicle. Finally, the throttle control is specified for the test. There were many options but many of the throttle controls did not fit the needs of the test. Ideally, since the goal is to make the vehicle performance similar to what would actually happen in a run on the salt flats the vehicle should try to deliver the maximum amount of power to the ground without slipping the tires. Since there is no traction control to implement many of the models when specifying a target speed would slip the tires often for the first mile. To prevent tire slip
from occurring in the test procedure a ramp to full throttle was used. This limits the capabilities of the vehicle and doesn’t accurately represent the vehicle’s acceleration but the model would still give accurate enough results to begin analyze trends. Another method that could be implemented with more time would be to specify a distance or time and a targeted speed using other simulations that have a more refined throttle controller. Eventually, to get more accurate results a different throttle model will need to be implemented.

With all the constant settings for the test selected the steering controller had to be selected. For the fixed steering test it was very simple to specify zero steering wheel input. For the steered test, a targeted vehicle path was selected with settings that could adjust how far ahead the steering controller could see and react to the track and vehicle. For the test, generic settings for the reaction and look ahead capabilities were applied. Since the vehicle is being maintained straight and there are no major dynamic events that occur to the vehicle in the test the assumption of not changing the steering model was logical.

4.3.1. Results

The two different tests were run at two different crosswind speeds, 10 and 15 miles per hour. This was done to begin to see the relationships between wind speed and the effects on vehicle stability. Many of the results were as expected but some of the general trends were of importance to understand while the model is still being refined.
4.3.1.1. Fixed Steering

The fixed steering kept the steering wheel locked and had the vehicle accelerated with a constant crosswind. The effects of the crosswind were quite predictable with the vehicle veering off course. What was surprising was the magnitude of the vehicle veering off course. The vehicle veered nearly 250 meters off the straight line so much so that the vehicle eventually rolled. The crosswind also induced vehicle yaw. The yaw didn’t begin to occur until the end of the test, which helped in causing the vehicle to eventually roll. In both cases the vehicle eventually yawed over 150 degrees which induced the vehicle roll. These results show how great of an effect the crosswind has on the vehicle.

![Vehicle Trajectory](image)

**Figure 4.9:** Vehicle Trajectory in Fixed Steer Crosswind Analysis
4.3.1.2. Vehicle Target Steering

The vehicle target steering test was done to see how much driver input would be required over the duration of the run. The target was to keep the vehicle going on the same track with no deviation. This would show how much steering input would be required as well as the amount of torque the driver would have to provide to keep that steering input. The two different crosswind speeds were considered again in the simulations to understand the trends that occur with various wind speeds.

For the simulation it was determined that the vehicle had nearly 250 more Newtons of force acting laterally on the vehicle in the 15 mile per hour wind compared to the 10 mile per hour wind when at 680 kilometers per hour.

Figure 4.10: Vehicle Yaw in Fixed Steer Crosswind Analysis
This additional lateral force required additional steering input. The 10 mile per hour crosswind required a maximum of 5.5 degrees of steering wheel input where the 15 mile per hour wind required a maximum of 7.8 degrees at top speed. Also, there is a noticeable change in steering wheel input 30 seconds into the run which is the shift from first to second gear. The change in steering was approximately 0.4 degrees, which is fairly small but still noticeable. This shows that the shift does have an effect on the stability because of the steering correction that has to be input.

Figure 4.11: Aerodynamic Lateral Forces in Targeted Steering Crosswind Analysis
The steering wheel torque was analyzed to better understand the amount of effort the driver is having to input into the steering wheel. The steering wheel torque had very similar trends as the steering wheel angle. The torque applied to the steering wheel increases with speeds. The maximum steering torque occurs at top speed with the 10 mile per hour case requiring a maximum torque input of about 5 newton-meters while the 15 mile per hour case had a maximum steering wheel torque of 8 newton-meters. The dynamics that occur during the shift also produce a noticeable change in steering of over 1 newton-meter. These values may not be highly accurate due to the generic tire model that may have higher tractive capabilities. The results may also differ with improved and more accurate mapping of the vehicles aerodynamic characteristics. The importance is in
understanding the trends. Even with a 33% increase in wind speed there was a nearly 60% increase in required torque at top speed when comparing the two cross wind speeds.

Figure 4.13: Steering Wheel Torque in Targeted Steering Crosswind Analysis
Chapter 5: Future Work

5.1. Vehicle Future Work

The VBB3 has been running since the summer of 2013 and has run on the Bonneville Salt Flats two times at speeds approaching 300 miles per hour. To this point the suspension has operated as expected but there are always improvements that can be made to be able to analyze and improve performance of the vehicle.

Figure 5.1: VBB3 on the Bonneville Salt Flats
5.1.1. Tire Break-in

As mentioned previously there is a break-in period on the tires. Once the tires have been broken-in they appear to have more consistent and predictable characteristics. Being able to have predictable tires has been an area that the team has strived to understand and with time a process for breaking the tires in would be developed. This could include a combination or spinning the tires up to high speeds unloaded as well as loading the tires and rolling them at lower speeds. These tires could then be used to do tire testing which could provide more accurate tire parameters for the models.

5.1.2. Vehicle Ride Height Sensor

In the past the growth in the loaded radius was back calculated from the vehicle speed and wheel speed while making some assumptions of tire slip. To more easily calculate the actual loaded radius of the vehicle, a method of measuring the vehicle ride height would be necessary. This in combination of knowing the suspension travel would allow for a more accurate measurement of the loaded radius. The sensor would also improve the measurement of the sprung mass motion as well. The difficulty comes with finding a sensor that is not affected by the harsh and highly reflective surface of the salt flats while still having a high enough sampling rate for the vehicle traveling at high speeds. Development of a system or a partnership with a company may be required to be able to find a solution that fits all the needs.
5.1.3. Tire Temperature Sensor

The temperature of the tires has always been an area that has never been explored and with the high speeds goals it could be useful in understanding the tire dynamics and the limits of the tires. The temperature of the tires could potentially play a role in the dynamics of the tires but there is no data to prove or disprove its effects. At the very least the tire temperature sensors could be used to determine if the tire is approaching an unsafe temperature that could lead to rapid degradation or even failure. This knowledge does not come without problems of implementing the sensors. The sensors that would be used would have to be able to standup to an extremely harsh environment where salt is being sprayed all over the enclosed wheel-well at high speeds. This presents a problem with the durability of the sensor and being able to measure the temperature accurately. The placement of the sensor will be critical to reduce damage and improve the accuracy of the measurement. It has been considered to measure the sidewall of the tire which would put the sensor in the least harsh area of the wheel-well but it wouldn’t be able to measure the temperature of the contacting surface of the tire.

5.2. Simulation Future Work

There are many areas of the vehicle dynamics modeling and simulation that could be improved. Some of the most important improvements include fully implementing the kinematics model, developing a tire model, introducing a more realistic VBB3 powertrain and validating the vehicle model with the physical vehicle. These areas would allow for the most significant gains in accuracy of the vehicle model.
5.2.1. Kinematic Model

There are many improvements that can be made to improve the accuracy of the vehicle and suspension model. To improve the accuracy of the suspension kinematics and roll characteristics, a complete SuspensionSim™ model needs to be exported to CarSim™ through tables. These tables are used to calculate motion and forces in CarSim™. Currently, the SuspensionSim™ model is not able to output tables for the anti-roll bar due to programming issues. This has required that an auxiliary roll moment be applied in CarSim™ for the front and rear axle that is input as a constant rate. Troubleshooting will have to be done to determine the issue and integrate the kinematic model into CarSim™. Once the issue is resolved CarSim™ will then be able to calculate the displacement and forces of the anti-roll bar, giving a more accurate representation of the suspension performance.

5.2.2. Tire Parameterization and Testing

One of the greatest areas of improvement for the vehicle model would be in the modeling of the tires. Since the tire model that is being used in current simulations is a generic model for a radial tire with modifications for tire growth and stiffness changes as a function of velocity. Further testing is required to build a tire model for a bias-ply tire that can more accurately model the stiffness, growth, traction and dynamics of the tires. To gather the parameters extensive tire testing as well as vehicle testing on the track and salt flats will be required. It would be ideal to parameterize the tire characteristics for both the asphalt test track as well as the salt flats. Some of the tire testing and
characterization that would be of interested would be for higher speed testing with fixed loads, lateral and longitudinal slip, lateral tire stiffness and rolling resistance. These parameters will be the basis of a more accurate tire model that represents the Mickey Thompson Bonneville tire.

5.2.3. Venturi Buckeye Bullet 3 Simulink Powertrain

Currently, the powertrain model in CarSim™ uses an internal powertrain model that is better suited for a traditional vehicle powertrain. Since the VBB3 powertrain is unique in the fact that it has two different motors for the front and rear axles as well as a battery, inverter and motor for those powertrains it makes it difficult to implement these models internal to CarSim™. Instead the powertrain models of the VBB3 that were designed in Simulink will be integrated into CarSim™. This will allow for a much more accurate estimation of the instantaneous power availability. This would also allow for great flexibility of battery, inverter and motor models for future simulations.

Figure 5.2: I/O Structure of VBB3 Simulink Powertrain Model
5.2.4. Crosswind Stability Analysis

To further improve the results and effectiveness of the crosswind stability analysis improvements could be made on the throttle control, aerodynamic parameterization and tire model. The throttle control was an issue, to get the appropriate acceleration of the vehicle near the tractive limit of the surface. The current models are all open-loop controls that are not effective at representing the maximum potential of the vehicle. One method would be to specify a velocity for a given distance or time. This would improve the accuracy of the analysis for a specific vehicle setup but would need to be modified if characteristics of the vehicle changed. To make the simulation more robust a closed-loop control system with traction control capabilities would be beneficial in improving accuracy and make the model more robust. The aerodynamics model of the vehicle could use further analysis for a greater range of slip angles as well as coefficients for the vehicle at higher speeds to insure the aerodynamic force are accurate through the various speeds. Analysis will need to be done on the aerodynamic effects of increased body height and pitch angle due to the tire growth and tire stiffness change. Finally, improvements to the tire model will allow for better estimates of tractive capabilities. All these improvements to the analysis will have the greatest impact on the accuracy and quality of the vehicle simulations and analysis.

5.2.5. Model Validation

There has been significant work done to implement model parameters from analysis, component testing and vehicle testing to try and develop an accurate vehicle model. This
requires that the model be validated and altered to more accurately represent the vehicle performance. To do this, controlled vehicle tests will have to be conducted with proper instrumentation to then allow for comparison to the model. The validation will begin to show the strengths and weakness of the model that may need additional work to more accurately model the vehicle performance.
Bibliography


[13] Confidential, "Vehicle Damping Ratio Spreadsheet Definition".


