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Design, Analysis, and Verification of an Open-Wheeled Formula-Style Race Car Suspension System

Joshua Beckerman

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Joshua Beckerman

Candidate for Bachelor of Science
and Renée Crown University Honors
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Honors Capstone Project in Your Major

Capstone Project Advisor: ____________________________
Mark Povinelli, Professor of Practice

Capstone Project Reader: ____________________________
Frederick Carranti, Associate Professor of Practice

Honors Director: ____________________________
Chris Johnson, Interim Director

1 Abstract

This honors thesis presents the design, measurement, and analysis of an open-wheeled formula race car suspension system. This race car is the second iteration of Syracuse University's Citrus Racing student team competition vehicle. The race car's suspension system features several designs that enable geometric adjustability to impact the vehicle's dynamic performance. The purpose of this research is to find an analytic approach to verifying the correlation between suspension design tunings and their effect on vehicle handling and road holding capacity. This was done by analyzing measured data obtained from a system of damper-mounted travel sensors as the vehicle drives through numerous realistic competition scenarios.

2 Executive Summary

This honors capstone project thesis provides an analysis and evaluation of a race car's suspension system using sensors that take measurements during live testing. The car was designed and built by a team of students as part of the Formula Society of Automotive Engineers (FSAE) intercollegiate engineering open-wheeled race car national and international racing competitions. Part of this engineering design process includes the vehicle suspension system. The 2017 race car from the Syracuse University FSAE team, Citrus Racing, was developed solely on theoretical concepts from textbook readings and trial and error process. Typically, the final design, including the suspension system, is road tested and adjustments are made based on the drivers' subjective observations and experiences. This iterative cycle of design, build, and test is repeated until the optimal suspension settings can be found.

This thesis is an investigation to determine if a quantitative measurement and analysis approach could be used to improve the performance of the race car's suspension system. To achieve this, linear shock potentiometers were attached to the shock absorbers and connected to a data storage device. The data could then be downloaded to a computer for long-term storage and analysis. Sensor data was collected during the race car track test runs over the course of a single day to isolate environment-dependent performance variables. Data was collected across three suspension set-ups over a total of 30 laps. The first of which was a "neutral" set-up that was designed to be an ideal compromise based on theoretical readings. The second set-up was characterized by the maximum amount of straight line control arm loading geometry to take the wheel loads off of the shocks. The third set-up was designed to limit the amount of roll that the vehicle experiences in high-acceleration turns. This data was used to calculate the individual wheel loads as the car was accelerating, decelerating, and turning. The data showed that a line of best fit will change its slope as a function of the adjustable performance characteristics. The data was also compiled to show average peak acceleration endured for each set-up, which can prove to be useful in determining the "fastest" suspension set-up.

Results of this analysis show a relationship between each of the performance characteristics and their associated weight transfer. In addition, two of the characteristics (control arm loading geometry in the front suspension and geometry to alter the amount of roll that the vehicle experiences in a turn) showed a significant effect on the amount of acceleration the tires could endure. Sets of plots were generated for these conditions as a function of weight transfer and maximum acceleration. The analysis of maximum acceleration showed that a low percentage of control arm loading geometry was able to help the suspension hold the most longitudinal acceleration. It was also found that a larger amount of effective roll on the vehicle had a slightly positive effect on cornering ability.

This thesis finds that a correlation between geometric suspension parameters and changes in effective weight transfer trends in the direction anticipated. Specifically, this is a decrease in weight transfer associated with a decrease in effective vehicle roll or control arm loading geometry. This ultimately validates all initial assumptions made during the design process. From the data analysis, it is concluded that the suspension set-up that will allow for the fastest achievable lap time is one with a moderate amount of control arm loading geometry and a relatively high amount of effective vehicle roll. These results support the approach that using measured quantitative data would allow the design team to determine which suspension set up produced the best results in terms of maximum acceleration achieved. While the subjective view of the test driver is also important, these results open up other sensor data collection opportunities for other engineering systems on the car.

These results will help remove the uncertainty that exists between the test driver's feedback and the engineer's technical understanding. A test driver can feel certain vehicle behaviors that could indicate three or four different possibilities based on set-up characteristics. Even skilled race car engineers cannot always know how to sift through driver reported results and be able to accurately know how to translate that into beneficial suspension system changes. It can be finally concluded that relying solely on the qualitative observational input from the test driver cannot encompass all of the factors when dealing with an experimental vehicle.

From this project it is recommended that the Formula SAE team make changes in the suspension design for the 2018 Citrus Racing car based on the results presented here. Aspects of the design such as suspension attachment point location and suspension arm geometry are likely to be affected by these conclusions. This change will also affect the design of the frame layout. Going forward, Citrus Racing should continue to collect data on the suspension and find new ways to improve on the original theoretical design. It is also recommended that the team invest in individual wheel accelerometers to determine true wheel loads. This will help to characterize the difference between wheel loads and shock loads. This will also allow engineers to account for suspension compliance.

The limitations of the measured data analysis presented here center on the insufficient quantity of data to extrapolate the results to other suspension set-ups. The general principles explored in this experiment are universal, but particular values for the set-up are specific to this vehicle and cannot necessarily be applied to another car's set-up. There is also an opportunity to optimize other systems of the car (e.g. brakes or engine) to output more braking and accelerative power. This would be one way to push the limits on the suspension system. It would allow the team to understand if the vehicle is limited by the acceleration systems or the suspension system. System compliance is also always a concern and can take away from the performance of a system like suspension. "For engineers, 'compliance' is the inverse of stiffness. So a higher stiffness value corresponds to a lower compliance value."[i] Since every component has a stiffness to it, there's a certain amount of flex occurring during loading that adds to inaccuracies when compared to an ideal system.

Systems like steering must also be in harmony with the suspension in order to provide the driver with adequate feel of the vehicle. Both the driver's fatigue and the feel of the vehicle can affect the outcome of these tests. For this reason, human error and inconsistency play a role in minor discrepancies found in the data. The only way to minimize this source of error is to increase the amount of quantitative measured data collected. A larger sample size could also make anomalies in the data collection less prominent.

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4 Introduction

4.1 Purpose

This honors capstone project thesis provides an analysis and evaluation of a race car's suspension system using sensors that take measurements during live testing. Formula Society of Automotive Engineers (FSAE) is an intercollegiate engineering competition in which a team of students designs and builds an open-wheeled race car during the academic year, then competes against other teams at national and international levels. The program gives students the opportunity to hone their engineering skills, develop manufacturing competency, and iterate an entire design cycle from ideation to execution. These skills are incredibly valuable to employers and make members of FSAE teams some of the most desirable new hires out of college. During the competition, teams are awarded points in a range of events:

Table 1: Scoring Distribution for Formula SAE Competition

Static Events		**Dynamic Events**	
Cost and Manufacturing Report	100 pts	Acceleration	100 pts
Marketing Presentation	75 pts	Skid Pad	75 pts
Design Judging	150 pts	Autocross	125 pts
		Efficiency	100 pts
		Endurance	275 pts
Static Total	*315 pts*	*Dynamic Total*	*675 pts*

In order to be competitive in Formula SAE, a team must perform well in both the static and dynamic events. The dynamic events are the primary focus and are cumulatively worth more than twice the points of the static events. The logic follows that in order to do well at competition, a team must do well in the dynamic events.

It is well-established that a Formula SAE car will not be able to go faster than about 60 mph in the course, due to the tight turns and short straightaways. Because of the relatively low speeds in the course aerodynamic drag and downforce are minimal and the most important design element is the suspension's capacity for *grip*. Grip is the traction that the tires have with the ground, and, the more a car has, the faster it can navigate a course. This project will explore several aspects of the design process and verification of the 2017 Citrus Racing suspension system.

4.2 Historical Background

Formula SAE was started in 1981 by industry professionals within the Society of Automotive Engineers.[ii] This design program is part of a larger organization that encompasses numerous other countries, including Germany, Brazil, and Australia. "The concept behind Formula SAE is that a fictional manufacturing company has contracted a design team to develop a small Formula-style race car."[iii] Although every team functions differently from one another, the most well-established programs continue to be the best because of their constancy in design, build, and test cycle. The majority of the best teams in Formula SAE have been consistently producing a new car each year for up to twenty or thirty years. These teams will generally make incremental changes or upgrades to particular subsystems each year, rather than starting from scratch. The advantage of this is that knowledge retention within a team is easy to maintain as new members join and senior members graduate.

At Syracuse University, the first Formula SAE team was started during the early 1990s. The team was successful in its endeavors and managed to place 25th overall at competition one year. However, the team was disbanded in 2002 due to lack of student interest and university support. The organization would lay dormant for ten years, all accumulated knowledge either forgotten or outdated. In 2012, the team was brought back into existence, but was unable to produce a car for several years. It was not until 2015 that the team, now known as Citrus Racing, was able to bring a functional race car to a Formula SAE competition. Both the 2015 and 2016 vehicles did not pass technical inspection so the 2016-2017 academic year was the first time Citrus Racing had a race car that could be tested properly. Until this point, the Citrus Racing vehicle suspension systems have been designed and tuned through trial and error based on driver input. This is the first time that the performance of the suspension will be optimized using quantitative measurements.

4.3 Technical Concepts

4.3.1 Overview

There are many kinds of suspension systems that have been developed, but the one that is used almost exclusively in Formula SAE is the non-parallel, unequal length, double A-arm independent suspension. This system is characterized by two "V"-shaped links, called *control arms*, that attach the wheel to four *pick-up points* on the chassis. The *shock absorber* is mounted to the chassis and is connected to a *rocker,* which is actuated by a *push rod*. There is also either a *toe link* or *tie rod*, depending on whether or not the wheel is steered, that controls the toe setting or the steered input to the wheel.

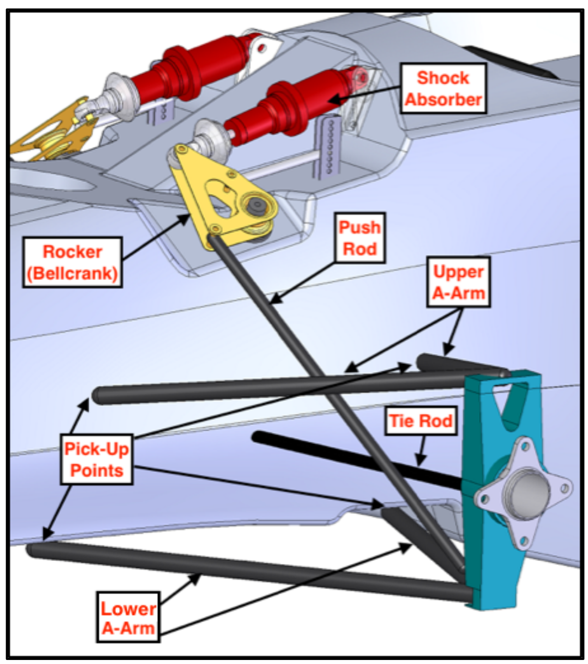

Figure 1: Identifying Key Suspension Components

In any suspension system, there are certain basic characteristics that define how the system behaves in a given situation. A few of the parameters that the engineer can alter are *shock spring constant*, "*anti-" geometry*, *roll center height*, and *rocker arm motion ratio*. These four variables can significantly alter the way a vehicle handles. The 2017 Citrus Racing car has been designed (using *OptimumKinematics*) to allow for a variable set-up through which a skilled engineer can tune the four parameters mentioned above.

The final stage in the design process is verification of engineering choices. The most efficient way to verify that the car will behave as it was intended is to drive it on the track, take measurements, and accumulate data on overall vehicle behavior. The data will be collected using a set of four linear shock potentiometers (*shock pots*) that measure the displacement of each shock independently, as well as a built-in accelerometer. The data will then be stored using a data acquisition (DAQ) system made by AiM. From this data, simple computational methods can be used to extract the qualitative analysis necessary to verify the design. The conclusions of this data analysis will be used to choose an optimal suspension set-up for the Formula SAE competition. This approach can also be used to influence the starting design of the next Citrus Racing vehicle.

4.3.2 Longitudinal Load Transfer

There are two forms of weight transfer in a vehicle that contribute to the forces on a suspension system. The first is longitudinal load transfer, which occurs when the car is either speeding up or slowing down $\left(a_y = \Delta v_y\right)$ and is defined as the movement of force from the front of the vehicle to the rear or vice versa. The longitudinal transfer in force is a function of the longitudinal acceleration, the weight of the vehicle, the height of the center of gravity and the wheelbase (distance from the front axle to the rear axle):

$$Long\ Load\ Transfer = Long\ Accel * \frac{Weight * h_{CG}}{Wheelbase} \qquad (1.2.1)^{iv}$$

Reducing this load transfer is often an advantageous characteristic to have, but only within reason. Maximum longitudinal acceleration occurs under braking, which makes braking the more pertinent concern. For this reason, the design of an FSAE suspension must account for approximately 1.5g of braking versus only about 1.0g of acceleration before including a margin of safety.

4.3.3 "Anti-" Geometry

The term *anti-* refers to the part of the suspension that can resist certain behaviors. Depending on the design, a suspension can have more or less resistance to those behaviors. In this context, those behaviors are *squat* and *dive*, and they are both related to the longitudinal acceleration of a vehicle.

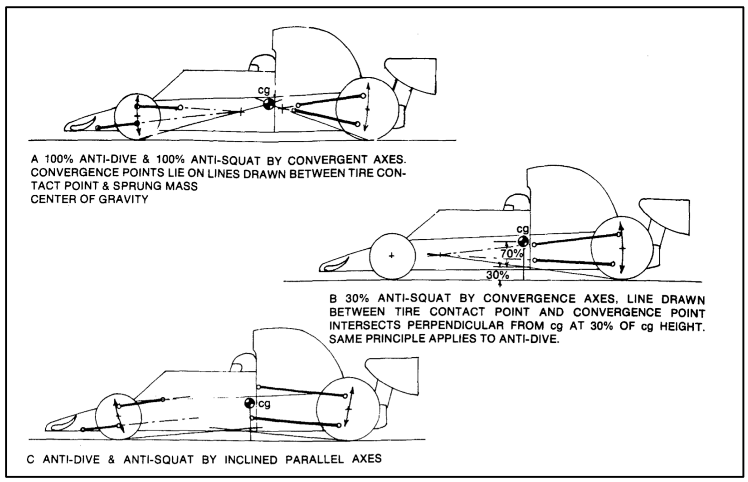

Figure 2: Illustration of Anti- Geometry
Tune To Win, Carroll Smith

Squat is what happens when a vehicle has a large forward acceleration, characterized by "giving it gas." In this instance, the vehicle will transfer some of the load on the front tires to the rear tires, causing the rear suspension to compress more than it would under *steady state conditions*. Dive is the opposite of squat and occurs under heavy braking. The vehicle will transfer load from the rear tires to the front tires, causing the front suspension to compress. In both cases, these behaviors can be reduced in magnitude by increasing the *percent anti-squat* or the *percent anti-dive*. It should be noted that anti-geometry is measured as a percentage in order to normalize the effect to a generic vehicle set-up. This is necessary because these characteristics are heavily based upon the location of a particular vehicle's side view instantaneous centers and center of gravity.

$$\%\ \text{anti}-\text{dive front} = \frac{(\%\ \text{front braking})\left(\frac{\text{svsa height}}{\text{svsa length}}\right)}{\left(\frac{h_{CG}}{Wheelbase}\right)} \qquad (4.3.3)$$[v]

The main effects of anti-geometry are as follows: (1) Anti-dive geometry decreases the amount of *bump* deflection in front suspensions during braking. Bump is when the wheel is raised with respect to the chassis. (2) Anti-squat geometry decreases the amount of *bump* deflection in rear suspension during acceleration on rear-wheel-drive vehicles. From *Race Car Vehicle Dynamics*, "the anti feature was assumed to be positive and therefore always working in such a way that the pitch deflections of the whole car would be reduced. It is possible to have the geometry arranged in such a way that the longitudinal forces actually increase suspension deflections. This is called *pro-dive, squat, or lift.*" [vi] This was deemed to be a poor option for race car performance.

4.3.4 Lateral Load Transfer

The second form of weight transfer occurs when a vehicle turns and is a more sensitive characteristic in Formula SAE compared to longitudinal load transfer. The load will shift to the outside of the turn due to the effects of centripetal acceleration $\left(a_x = \frac{v_t^2}{R}\right)$. Lateral load transfer is a function of the lateral (or centripetal) acceleration experienced by the vehicle, the weight, the height of the center of gravity, and the track width (the distance between the right and left wheel):

$$Lat\ Load\ Transfer = Lat\ Accel * \frac{Weight * h_{CG}}{Track\ Width} \qquad (4.3.4)^{vii}$$

The amount of speed that a car can carry through a turn is called its *cornering capacity* or maximum *cornering powering*. The higher the cornering capacity, the faster a car can go, which translates direction into quicker lap times.

This correlates directly with the stability of the vehicle through a turn. If a suspension is too soft, it will deflect and oscillate well after the exit of a turn. A slow response time is undesirable in the fast-paced setting of an autocross event. Alternatively, a vehicle that utilizes an overly stiff suspension has no feel and will likely disorient the driver. It is important to find a balance in this, which necessitates a validation in the height of a suspension's front and rear roll centers.

4.3.5 Roll Center Height

In order to maintain grip through the corners, the vehicle's suspension must be proficient when it comes to dealing with large lateral accelerations that lead to lateral load transfer. A competitive FSAE car must be set up for as much as 1.5g of lateral acceleration (or acceleration equal to 1.5 times the acceleration due to gravity). As the centripetal acceleration increases, the mass of the car will effectively shift to the outside wheel of the turn due to counteracting centrifugal force (see Figure 3). The discussion regarding vehicle handling in an autocross vehicle brings the focus to body roll for a

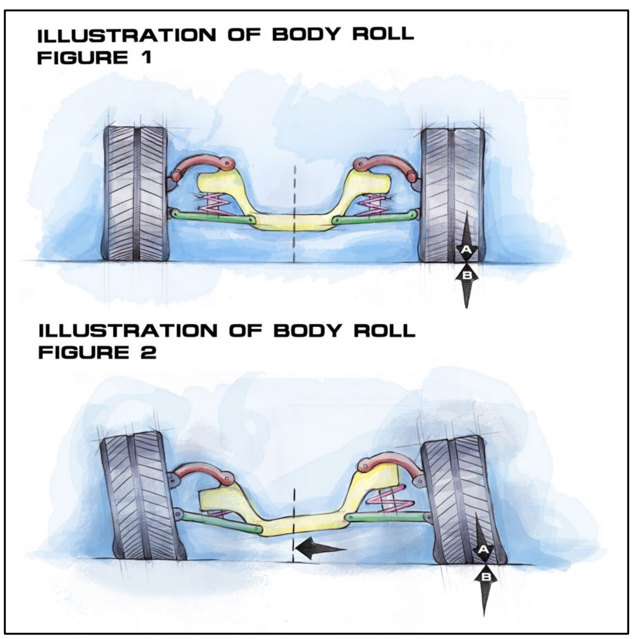

Figure 3: Visualization of Vehicle Body Roll in a Right Turn
http://chevymax.com/wp-content/uploads/2013/08/04.jpg

number of reasons. The rationale behind the relationship between vehicle roll response and handling characteristics is beyond the scope of this project because it involves an understanding of dynamic parameters such as control derivatives, a graduate-level dynamic concept.

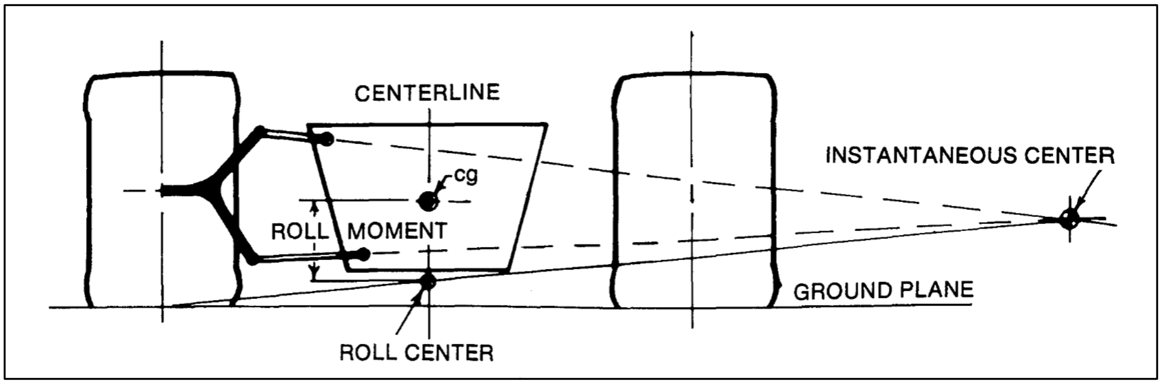

Figure 4: Depiction of Front View Suspension Geometry Including Roll Center & Roll Moment
Tune to Win, Carroll Smith

The suspension roll center can be found geometrically as shown in Figure 4. The roll moment is applied based on the height of the center of gravity relative to the height of the roll center. As the roll moment increases, so will the amount of body roll. These types of changes in vehicle orientation will directly apply to the stability of the system.

4.3.6 Other Concepts

Other concepts such as: contact patch, traction circle, compliance, suspension binding, heave/roll/pitch, bump, droop, and roll moment are necessary for a complete understanding of an open wheel suspension system. It is highly encouraged to become familiar with these concepts.

4.4 Importance of Measurement to Achieve Goal

4.4.1 Original Design Approach

The design of Citrus Racing 2 (CR2), the second vehicle to be designed by the Syracuse University FSAE team, began with very little knowledge of suspensions, vehicle dynamics, structural design, or even the proper sequence to follow for creating a complex race car system much like the first vehicle two years earlier. *The design process* had only been the subject of textbook discussion during the first two years of the engineering curriculum as shown in Figure 5. There was no hands-on experience in designing something completely from scratch.

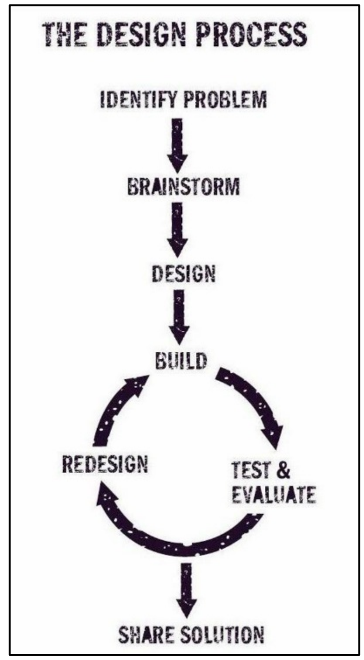

Figure 5: Iterative Design Process
http://makezine.com/wp-content/uploads/2008/11/designprocessjpg.jpg

The first lesson taught in both *Race Car Vehicle Dynamics*, by William Milliken and Douglas Milliken, and *Tune to Win*, by Carroll Smith, are that every great race car is designed around the tires. This is why the first chapter in each book is about tires, the contact patch, and the traction circle. Moreover, it is explained in these books that the sole purpose of a race car suspension is to maximize the contact patch of the racing tire at all times.[viii]

The first step in designing anything is to assess the goals that are intended for the design. For example, an off-road suspension will have at least a foot of wheel travel in order to avoid bottoming-out in a ditch, but this ability is useless on a NASCAR oval track. The basic goals of this FSAE competition race car system were as follows:

- Minimal aerodynamic loading

- Compromise between straight-line and lateral acceleration
- High reliability
- Reasonable tuning capability for performance optimization
- Avoid system compliance issues
- Avoid potential suspension binding

These goals guided the entire design from start to finish. Many engineering choices can be traced back to the original goals, however the only one with which this thesis is concerned with is that of the tuning capability of the suspension. It was decided that certain parameters would need to be tested in order to determine the optimal set-up, given the team had no previous experience with such design choices. The test parameters of anti-squat, anti-dive, and roll center height were chosen because each one helps to determine the location of the suspension attachment points on the chassis. There are a number of other parameters like these in suspension design, such as scrub radius and kingpin inclination, that affect vehicle performance in steering and other dynamic areas. These other parameters will remain beyond the scope of this thesis, but are important areas for exploration and will be discussed in the Future Work section of this report.

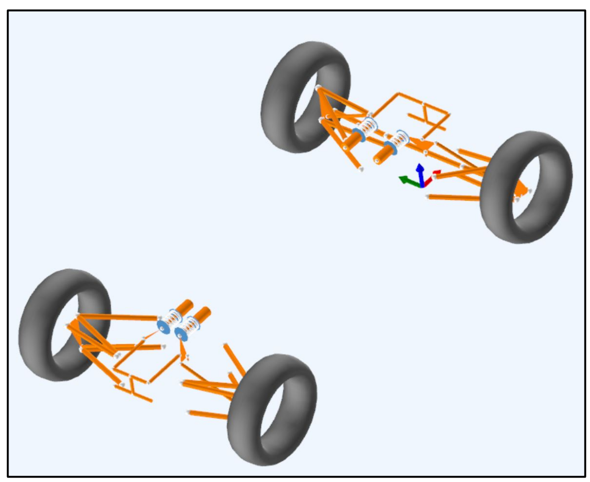

Figure 6: CR2 Suspension from OptimumKinematics

Once a conceptual suspension has been scratched out on paper, the next step is to start playing with the numbers and modeling the *kinematics.* Kinematics is a branch of mechanics that deals with pure motion of a system, without reference to the masses or forces involved in it[ix]. The kinematic model is a purely geometric model, focusing on position, velocity, and acceleration of bodies. CR2 kinematic studies were conducted using a program called *OptimumKinematics*, which models the vehicle as points, rods, and cylinders (see Figure 6).

OptimumKinematic allows the user to put in the location of every point of a given suspension system and run it through user-defined motion studies involving Heave, Pitch, Roll, and Steering inputs. The program applies these motions as a function of time (in time steps) to the physical body at the pick-up points and allows the suspension to move freely according to geometric linkage principles and physics. The output is a list of critical vehicle characteristics as they vary with time, such as physical suspension point locations, camber values, and roll center height to name a few. This helped to guide the initial determination of suspension values.

Once the initial set-up design was selected, components such as control arms, push rods, bellcranks, and uprights were designed, stress tested, and manufactured. One component that is easily overlooked or over-simplified is the pick-up point. This is the bracket that holds the control arms to the chassis and is critical to this thesis because it is the source of CR2's suspension adjustability. This bracket assembly has several components; the bolt, the rod end, the bracket, two T-spacers (green in Figure 7), and two quarter inch spacers (red in Figure 7).

Because the T-spacer offset from the top and bottom of the bracket provides the proper clearance in high bump/droop scenarios, the design of this assembly allows the engineer to vary the effective height of each pick-up by the width of a spacer in either direction. Instead of a spacer on either side of the rod end, there could be two spacers above the rod end, moving the effective pickup point down a quarter of an inch. Manipulation of these pick-up points allows the suspension set-up to be varied trackside based on driver feedback.

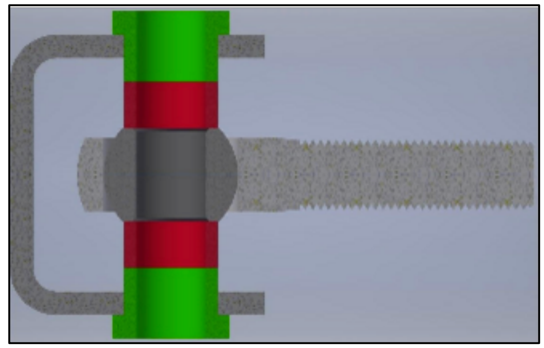

Figure 7: Cross-Section of Suspension Pick-Up Point

4.4.2 Alternative Design Approach

Although driver feedback is typically a crucial part of vehicle development, it is not without its faults. The most concrete approach to verifying design choices is through quantitative data collection and analysis whenever possible. For this reason, the vehicle was fitted with four linear shock potentiometers that connected directly to the on-board data logger. With the combination of data collected by the shock pots and the accelerometer built into the data logger, a great deal of information can be learned about this suspension system.

Through this alternative approach, Citrus Racing hoped to accomplish two objectives: (1) verification: to verify that adjusting the suspension pick-up points would affect vehicle performance in the ways that it was discussed in the textbooks and (2) optimization: determine the amount of anti-squat/dive and the height of the roll center that would result in the most achievable acceleration.

Both of these objectives require a range of suspension set-ups that would highlight the range of the adjustability during testing and give an engineer a minimum of two data points in order to identify trends. For objective 1, verification requires an expectation of what should happen when something is done. In the case of adjusting anti-squat/dive geometry, it is expected that increasing the percentage of each will take more load off of the shocks and transfer it onto the control arms. This transfer results in less shock compression for an equivalent wheel load. In any situation, the wheel loads are independent of anti-geometry and can be assumed to be directly proportional to the amount of longitudinal acceleration experienced by the vehicle. Therefore, the verification of anti-geometry can be conducted by measuring the amount of shock compression, or effective shock load, per *g* of longitudinal acceleration. It is expected that additional anti- will decrease the effective shock load per *g*. Similar to the verification of anti-geometry adjustments, the height of the vehicle roll center alters the roll moment applied when the vehicle experiences roll. This means that a higher roll center will be closer to the center of gravity. This should decrease the roll moment and therefore decrease the amount of effective weight transfer per *g* of lateral acceleration.

For objective 2, optimization of the suspension system relies on more complex metrics. Because tire contact patch cannot be realistically measured during testing procedures, the next best thing is to look at lap times to see which suspension set-up produces the fastest times. This requires consistency and repeatability from the vehicle, driver, and environmental conditions, which are all quite difficult to hold constant for an entry-level Formula SAE car tested outside in Syracuse, NY in early November.

Given the difficulties with objective 2, the analysis can focus on the accelerations achievable in a particular set-up as it will be a direct indicator of potential lap times. This is similar to when tire companies will boast skid pad improvements of 0.05g because it is a direct indicator of performance and ensures that the driver will be able to carry more speed through a particular corner. All other variables held constant, the best set-up will enable the greatest peak accelerations, thus expanding the vehicle's potential traction circle.

Another metric for finding the fastest set-up will be how fast the set-up allows the car to be driven. This seems obvious, but there are some critical factors at play that could affect the results of a test such as this. Because these tests were not conducted on a clearly defined, closed loop track, there are instances where the driver could go "outside the lines" and achieve speeds greater than possible within the track bounds. In order to avoid significant outliers collected from straight line testing, the median speed will be captured rather than the average speed. In addition, it is possible that set-ups will get increasingly faster simply because the driver will get more accustomed to the course and the limits of the vehicle.

4.5 Objective

It is expected that the physics will correlate with the suggestions made in the automotive engineering textbooks used to design the suspension system. It is unknown, however, how the performance characteristics in question will affect the overall performance of the vehicle. It is also important to note that one major purpose of the experiment is to prove that direct measurements are often better than test driver input due to accuracy and expediency.

4.6 Approach

In the next sections, the reader will find an overview of the method, procedure, and results of this experiment. These sections are meant to enable others to repeat the experiment and report the findings that came from the data collected. There is a discussion section that explores various trends observed from the data as well as posited explanations for those trends. The final sections outline conclusions drawn from the experimental results, recommendations for future experimental changes, and suggestions for future work on the subject.

5 Measurement Set-Up & Data Collection Methods

5.1 Instrumentation

The collection of data for this experiment is done through the use a number of sensors. The data is sorted, then saved by a data logger until it can be read and interpreted by the user. All instrumentation hardware used in this set-up was purchased from AiM to ensure hardware and software compatibility. AiM has created an interactive software package, called Race Studio 2, that allows users to access, analyze, and interpret large sums of data collected using their devices.

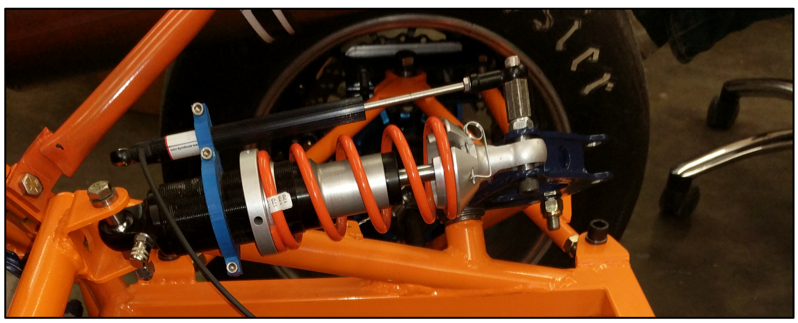

Figure 8: AiM Potentiometer Mounted to the Rear Shock

The primary measurement sensors used in this thesis are linear potentiometers. They are specifically designed to measure the compression and extension of suspension dampers. The specific sensors used have a diameter of 9.5 mm and measurable stroke length of 75 mm. These linear potentiometers are mounted along the length of the dampers with one end attached to the bolt connecting the damper to the rocker, while the other end is secured to the shock cylinder using a 3D printed, PLA clamp (see Figure 8).

Figure 9: AiM Evo4 Data Logger
http://www.aim-sportline.com/images/common/foto_evo4_290.jpg

The equipment used for sorting and storing the data is the AiM Evo4 Data Logger (see Figure 9). This onboard logger also has a built-in three-axial accelerometer, which collects acceleration data based on the car's behavior. The Evo4 is bolted to a laminate panel that is clamped to a frame member on the front of the car, oriented in such a way that the accelerometer can accurately collect data. The data logger is connected to the shock pots using 0.5 m 712-719 patch cords. The Evo4 connects to a laptop using a USB cable.

5.2 Procedure

Data collection for this experiment requires a large amount of preparation including instrument set up and calibration. In order to calibrate the linear potentiometers with the data logging system a step by step procedure must be followed as outlined in the user manual for the potentiometers[x]. Once the shock pots have been calibrated in Race Studio 2, the vehicle can be taken out for testing.

5.2.1 On-Site Preparation

Once on site, the first step in the experiment is to set up a track of cones that include at least two small radius corners (one for each side), a straight, a hard braking corner, and a hairpin. Also include a short section for figure-8 skid pad testing. After the track is set up, the car must be set up as well. Properly balance the resting wheel loads by scaling the car and adjusting the shock spring pre-load is done to even out the suspension's static loading. Also, dialing in the proper amount of camber, toe, and other settings that could affect the symmetry of the vehicle's handling.

Finally, a complete check of the car will ensure reliability and peak performance. Check all components to ensure that they do not have too much wear (e.g. brake pads) and check fuel levels, oil pressure, etc. to ensure proper operating conditions. Once this is done, the driver should perform several warm-up laps to get the tires to the appropriate temperature for maximum grip prior to data collection.

5.2.2 Testing Procedures

Once the vehicle has been prepared for testing, the first data collection is started by clearing existing data off of the Evo4 logger. Instruct the driver to operate the car as close to the limit of grip as possible, passing through the course, for about 10 minutes if possible. This should be enough data to create a large sample. After the data has been collected, bring the car back in and connect a computer to the Evo4 logger and download the data. Save this data, it will be used for the analysis.

Following the quantitative data collection, ask the driver how it felt when operating the car. Get as much information as possible about the understeer/oversteer characteristic, the overall handling of the vehicle, throttle response, braking response, etc. This will not only confirm that the car is operating at peak performance, but give a qualitative perspective to the data. After both quantitative and qualitative data sets are collected, change the vehicle set-up and repeat the testing procedures. For all subsequent data sets, include the information of comparisons to previous set-ups.

6 Results

The results of the measured data analysis will be presented starting with the neutral set-up data. Two alternative test set-ups will then be discussed, one with the greatest achievable anti-squat and anti-dive and one with the highest achievable roll center with the given tuning capabilities. Each set-up had a particular amount of anti- and a particular roll center height set into the system, as prescribed by the designers. These parameters were hypothesized to affect lateral and longitudinal accelerative capabilities. This correlation will be explored in the Discussion section. The results include plots of peak longitudinal acceleration, peak lateral acceleration, and peak speed. These plots can be found in the appendix. The results also report the effect of this geometry on load transfer using two plots of weight transfer versus acceleration for each set-up.

6.1 Neutral Set-up

The first few runs at operating conditions were performed under what was deemed to be neutral conditions. These are the settings that were chosen by the designers to be the most desirable, based on the theoretical research done on suspension. The neutral set-up is defined in Table 2 below.

Table 2: Netural Set-Up Parameters

Set-Up Parameters	**Neutral Set-Up**
Percent Anti-Dive [%]	25.6
Percent Anti-Squat [%]	18.7
Roll Center Height [in]	Front: 1.27 Rear: 1.49

Although these values do not mean much on their own, they provide a point of comparison to the other two alternative suspension test set-ups.

At the neutral position, the suspension permitted a widely varied range in peak braking acceleration while maintaining a far narrower range of peak throttle acceleration. This can be seen in Figure 10. The suspension also allowed a roughly symmetric amount of lateral acceleration, however, it seems that the maximum lateral acceleration was on average slightly higher above 25 mph, viewable in Figure 11. The peak speed had a large spread, but had a median peak speed far below the maximum at 49 mph, as shown in Figure 12.

The maximum lateral acceleration yielded approximately 300 lb. of lateral weight transfer in each the front and rear, as shown in Figure 13. There is a high density of data points around the lateral acceleration of zero because the driver spends more time in this area of the acceleration range due to the straights and transitions points on a track. Finally, the trend of the data in Figure 14 shows two different slopes, one for data points with greater than zero acceleration and one for data less than zero acceleration.

Here are the driver's comments after the first 10 minutes driving the car in its neutral set-up:

- *Tendency to understeer at low speeds.*
    - *Doesn't handle bumps.*
    - *Wants to throw the car.*
- *High Speed is okay for understeer.*
- *Nice initial rotation [of the steering wheel].*
- *After additional rotation, exaggerated roll.*
- *Slow roll response.*
- *No Dive. Need more weight transfer.*

6.2 *First Alternative Test Set-Up, Maximum "Anti-" Geometry*

During the second set of runs, the suspension was adjusted for the highest percent anti-dive and highest percent anti-squat that can be tuned into the system. This setting ultimately changes the effective load on the dampers by putting some of the transferred weight into the control arms. This change in geometry affects the location of the roll center as well. The roll center height in this set-up was not chosen, but is a consequence of the increase in anti- geometry.

The maximum "anti-" geometry set-up is defined in Table 3 below.

Table 3: Maximum Anti- Geometry Set-Up Parameters

Set-Up Parameters	Max Anti Set-Up
Percent Anti-Dive [%]	45.8
Percent Anti-Squat [%]	45.7
Roll Center Height [in]	Front: 0.72 Rear: 1.22

Compared to the settings reported in Table 2, the anti-dive and anti-squat percentages have both been nearly doubled in order to achieve almost 50% anti- effects. Also, the roll center location for the front and rear dropped by 0.55 inches and 0.27 inches, respectively.

In this alternative set-up, the vehicle seems to operate consistently both under braking as well as during throttle acceleration, which is seen in Figure 15. Again, the suspension responds symmetrically to lateral acceleration in the low speed turns, but in the high speed turns it does not. This can be seen in Figure 16. The peak speed had a relatively narrow spread, but the same median as in the previous set-up, as seen in Figure 17.

In Figure 18, there is high point density at the maximum and minimum weight transfer areas, which indicates that the driver spent more time on the edge of grip in lateral acceleration. When looking at high point density on Figure 19, it can be seen that point cloud for negative acceleration was quite light. This indicates that the driver was either not braking heavily or was unable to obtain sufficient braking force during these runs.

Here are the driver's comments after 10 minutes driving the car in its first alternative set-up:

- *Car was better [than the neutral set-up] through the slalom this time.*
- *The suspension was doing a better job transferring weight front to back [than neutral].*
- *Right to Left had too much roll.*
- *No warning when it's going to break loose. About the same as before, but getting used to it so I can get closer to the limit.*
- *Tires were working better this time. Good grip, could get them to screech.*
- *No change in braking. Still has a lot of understeer.*
- *Fronts were getting more traction.*
- *Straight line acceleration felt the same.*
- *Snap oversteer no change.*
- *Improvement overall in terms of drivability.*
- *Asymmetric R/L mid corner. Can't turn to the right, just understeer.*
- *Starts by plowing, then digs in and just wants to spin you.*
- *Possibly a camber issue.*

6.3 Second Alternative Test Set-Up, Highest Roll Center

During the third set of runs, the suspension was tuned to give it the highest possible roll center in the front and rear. This change effectively shortens the moment arm between the roll center and the center of gravity, thus decreasing the roll moment. A decreased roll moment translates into less rotation of the chassis in a given turn at a given speed.

The highest roll center set-up is defined in Table 4 below.

Table 4: Highest Roll Center Set-Up Parameters

Set-Up Parameters	**High RC Set-Up**
Percent Anti-Dive [%]	26.7
Percent Anti-Squat [%]	11.5
Roll Center Height [in]	Front: 2.34 Rear: 2.55

Compared to the settings reported in Table 2, the anti-dive and anti-squat percentages are relatively close, but anti-squat is somewhat lower. Also the roll center location for the front and rear was raised nearly doubled.

In this alternative set-up the vehicle seems to operate consistently under braking, but has some extreme outliers during throttle acceleration that are likely attributed to error. These trends are seen in Figure 20. The suspension responds somewhat symmetrically to lateral acceleration, however the relative difference between this set-up and the others may indicate a small amount of instability in the system. It can also be seen in Figure 21 that the average high speed acceleration is slightly lower than the average low speed acceleration. The peak speed had the most scattered data of the three set-ups and only a slightly higher median, as seen in Figure 22.

In Figure 23 there is a strong correlation to the data, which would indicate that much of the data was collected in pure lateral acceleration, rather than a combination of lateral and longitudinal acceleration. The driver spent an uneven amount of time on the edge of grip for one side of the vehicle versus the other, which can be concluded by looking at the point density on the plot. The data from Figure 24 shows a steeper slope for positive longitudinal acceleration than in the previous two set-ups.

Here are the driver's comments after 10 minutes driving the car in its second alternative set-up:

- *Felt a lot better. Improved Understeer.*
- *Could take the slalom faster. More front end grip.*
- *Rear is dodgy, wants to step out, but now you can control it.*
- *Lots of dirt, difficult to get grip.*
- *No warning still [on traction breakaway].*
- *Tires working best so far. Overall handling better.*
- *Figure 8 [track handling] improved.*
- *Throttle is getting weird. Possible not going as fast.*
- *Can Trail Brake Easily.*
- *Under braking load transfer was better.*
- *No difference in asymmetry*
- *Plowing is less.*

7 Discussion

In this section, the data for each tunable parameter is split up in order to look at comparisons across multiple set-ups. Anti-dive, anti-squat, and roll center height data are given in tables. All associated plots, from which the tabular data is drawn, can be found in Appendix A. It should also be understood that the data in this section includes values of average acceleration. Although it might seem like differences on the order 0.01 g of acceleration are insignificant, the cars within the same price range might only see a 0.01 or 0.02 g spread in cornering ability on the skid pad.[xi]

7.1 Anti-Dive Data

Tuning the anti-dive geometry was intended to affect the amount of load that the front shocks experienced per degree of negative longitudinal acceleration. This tuning parameter should therefore alter the slope of the best fit line on a plot of Braking Acceleration vs. Shock Load as presented in Figure 14, Figure 19, and Figure 24. Specifically, an increase in percent anti-dive on the vehicle should correspond to a decrease in absolute value of the slope, which will be referred to as the suspension's *inboard braking stiffness*. In Table 5 a comparison between Neutral, Max Anti and High Roll Center is presented. The concept of inboard braking stiffness holds true in comparison of anti max and high roll center. However, comparing the results of Neutral and High Roll Center shows a contradictory trend.

Table 5: Anti-Dive Data for Each Set-Up

Performance Characteristics	Neutral	Max Anti	High RC
Percent Anti-Dive [%]	25.6	45.8	26.7
Inboard Braking Stiffness [lb/g]	-138	-124	-167
Avg. Braking Acceleration [g's]	-0.76	-0.73	-0.77

Though the inboard braking stiffness is used as a metric in the validation of the vehicle, it does not directly affect the vehicle's performance. The average braking acceleration, however, does. If the vehicle has more stopping power with a particular set-up, it will be able to hold a high speed for a fraction of a second longer. These fractions of a second will add up to significant reductions in lap time. From Table 5, it is seen that the vehicle could achieve a greater average braking acceleration with a lower percent anti-dive, both in the Neutral set-up and High Roll Center set-up, than possible with the Max Anti set-up.

When comparing the feel of the vehicle from the Neutral set-up to the Max Anti set-up, the driver reported that there was no discernable difference in braking. It is shown that there was in fact a difference in average braking acceleration between these two set-ups. However, it is quite possible that this difference was not significant enough for the driver to be aware of. This is proof of the original purpose of this experiment in that the testing driver cannot always tell what is going on with the vehicle.

7.2 Anti-Squat Data

Similar to anti-dive, anti-squat geometry was intended to affect the amount of load that the rear shocks experienced per degree of positive longitudinal acceleration. This tuning parameter should therefore alter the slope of the best fit line on a plot of Throttle Acceleration vs. Shock Load presented in Figure 14, Figure 19, and Figure 24. Just as in anti-dive, an increase in percent anti-squat on the vehicle should correspond to a decrease in absolute value of the slope, which will be referred to as the suspension's *inboard throttle stiffness*. In Table 6 a comparison between Neutral and Max Anti inboard throttle stiffness holds true for the general trend, though the relationship seems to not be greatly affected by large changes in percent anti-squat. It also appears from the data that the relationship between anti-squat and inboard throttle stiffness is non-linear.

Table 6: Anti-Squat Data for Each Set-Up

Performance Characteristics	Neutral	Max Anti	High RC
Percent Anti-Squat [%]	18.7	45.7	11.5
Inboard Throttle Stiffness [lb/g]	-282	-263	-323
Avg. Throttle Acceleration [g's]	0.50	0.51	0.50

From Table 6 the average throttle acceleration seems to be reasonably unaffected by the percentage anti-squat. This would indicate that the amount of anti-squat tuned into the suspension system does not affect its ability to accelerate from the motor's power. It is also possible that the suspension system is only slightly affected by the amount of anti-squat, due to the very weak (yet present) correlation.

7.3 Roll Center Data

The analysis of the roll center height requires a look at the front and rear suspensions individually and as a pair. This is due to the fact that the vehicle has two roll centers (front and rear), but that they are still connected to one another. The front and rear suspensions both have the same basic configuration, but the data shows inconsistencies that must be addressed. When looking at Figure 13, Figure 18, and Figure 23 it is clear that there is a strong, negatively correlated slope between lateral acceleration and effective load on the shocks. This slope will be referred to as the *cornering sensitivity* because it describes how the much effective load the shocks will see per lateral g of acceleration. This parameter can be used for comparisons because it can be assumed that the actual load on the wheel per lateral g of acceleration does not change. It is expected that the increase in roll center height will ultimately decrease the cornering sensitivity, translating into less roll of the body.

When looking at Table 7, it can be seen that the roll center height for the neutral and max anti set-ups are between 0.7 and 1.5 inches. These two set-ups have cornering sensitivities hovering around 190 or 200 lb/g. In contrast the highest roll center set-up has a roll center approximately 2.4 inches above the ground and a cornering sensitivity around 180 lb/g. Though this is not a dramatic change, it does follow the assumed trend for cornering sensitivity.

Table 7: Roll Center Data for Each Set-Up

Performance Characteristics		Neutral	Max Anti	High RC
Roll Center Height [in]	Front Rear	1.27 1.49	0.72 1.22	2.34 2.55
Cornering Sensitivity [lb/g]	Front Rear	-200 -185	-203 -190	-187 -174
Avg. Lateral Acceleration Under 25mph [g's]	Right Left	0.96 1.00	1.07 1.14	1.01 1.11
Avg. Lateral Acceleration Over 25mph [g's]	Right Left	1.03 1.04	1.08 1.06	0.98 1.03

Looking at the average lateral acceleration over and under 25mph in Table 7, a number of observations can be made. The first observation is that there is little correlation between the average lateral acceleration and the speed at which vehicle is traveling. This can be stated because there are an equal number of instances where the acceleration is greater for the under 25mph case as there are for the number of instances where the acceleration is greater for the over 25mph case.

The second and more critical observation to be made is that the Max Anti set-up, the configuration with the lowest roll center, consistently has the greatest average lateral acceleration. However, there is not a clear trend across all three suspension set-ups. In in Table 7, the average lateral acceleration of the Neutral and High Roll Center set-ups switch between being the worst performing set-up across the data set. For example, the average lateral acceleration under 25 mph for the Neutral set-up is lower than the High Roll Center set-up, but in the over 25mph case, the High Roll Center set-up is lower than the Neutral set-up. The expectation was that there would either be a linear trend across all three set-ups or no trend at all. There are a few possible sources for error in this data that can account for this inconsistency. These include asymmetry in the system during manufacturing of the suspension and insufficient amount of data collected to neglect circumstantial differences in the suspension.

The driver reported that the Neutral set-up and the Max Anti set-up had issues with understeer, but this was not mentioned in the High Roll Center. Also the driver believed that there was too much roll in the Max Anti set-up, which is not surprising given that this set-up had the lowest roll center. However, the driver also said that the drivability of the vehicle was vastly improved. This observation reflects the improvements in average lateral acceleration. It seems that the increase in roll allowed the driver to navigate the vehicle consistently closer to the limit of grip.

7.4 Overall Speed Data

It was the intention of this section to take a holistic look at the effect of altering the suspension set-up. Even though taking a look at median or average speed could be a performance indicator in a true track-tested scenario, it is not possible here. This is because the vehicle was not tested on a closed loop track. In our test the driver had the ability to speed up, slow down, or take a different path through the course each time. This is also why lap times are not collected and reported.

Table 8: Median Peak Speed for Each Set-Up

Performance Characteristic	Neutral	Max Anti	High RC
Median Peak Speed [mph]	49.1	49.0	50.5

This systematic dismissal of median peak speed data is supported by Table 8, which shows no significant difference between the median peak speed of each suspension set-up. Ordinarily, data like this would disqualify any other results from the experiment, but the previous results and conclusions still hold merit on their own. This is because the testing did not occur on an enclosed track. The driver could speed right past the course just to cycle through the gears at any point during testing just to get a feel for the acceleration. This type of variation does not affect the acceleration data, but does affect mean peak speed. For this reason, Figure 12, Figure 17, and Figure 22 are not an integral part of this discussion. However, as stated above, the rest of the data collected is still valid.

8 Conclusion

The most significant conclusion to be made from this experiment is that the basic physics and dynamics on which the original engineering design choices were verified using measured sensor data. In fact, these assumptions are so prominent that the experimental data resoundingly reflected all such trends. These trends included those of the inboard braking stiffness, inboard throttle stiffness, and cornering sensitivity. Overall, it was clear from the comparison between the driver's notes and the data collected, that some of the fine detail gets lost between the tires and the driver's notes.

Additionally, it can be concluded from the data that anti-squat geometry has little effect on the amount of throttle acceleration this vehicle can withstand. However, if a car with more torque at the wheels were tested, perhaps one more prone to wheel spin, the results may be different. This is because the race car engine does not have enough accelerative power or torque to make the tires spin. Wheel spin is effectively a car exceeding its capacity to translate power from the drivetrain into motion.

From the tests results of the anti-dive geometry, it was found that the amount of anti-dive tuned into the suspension affected the amount of braking acceleration that the car could hold. Furthermore, it was found that this effect was a negative correlation of braking acceleration with an increased percentage anti-dive. It seems that beyond a certain point, anti-dive is detrimental to the performance of the vehicle. Finally, it was determined that a low roll center height provided the highest level of performance from the vehicle and improvements in drivability. However, there were some unclear trends with the other two set-ups. Because these results are inconsistent, it is encouraged that this area be explored more in the future.

9 Future Work

It is recommended that the race car be tuned to have little or no anti-squat, some anti-dive (approximately 20%), and a roll center about 1 inch above the ground. These are all recommendations that are based upon the findings described in the discussion and conclusion sections above. In terms of taking these findings further, there are several possibilities that are open to engineers interested in the topic. The first opportunity is to explore a wider range of values for the height of the roll center. This test measurement would give a more complete and detailed picture of the effects of this performance characteristic. The next opportunity is to test how the location of the instantaneous centers effect the suspension performance. As discussed in the technical concepts section, the instantaneous centers are geometric constructs that ultimately determine the location of the roll center. However, the instantaneous centers have other effects on the suspension system's behavior as well. These effects include the amount of camber change per degree of roll.

The last opportunity for future work on this topic is to explore other possible performance characteristics that were omitted from this thesis. This include testing the tuning characteristics of camber values, Ackermann, and slip angles which are also critical to the performance of the vehicle. These parameters correlate directly with the steering performance and would focus primarily on the front end suspension of the race car.

10 Appendices

10.1 Suspension Set-Up Sheet

Suspension Testing Set-up Sheet

Front Suspension				Rear Suspension			
Config 1 (neutral setting)				**Config 1 (neutral setting)**			
	Fore	**Aft**	**(x,y,z)**		**Fore**	**Aft**	**(x,y,z)**
Top		FVIC	(0,141.19,8.829)	**Top**		FVIC	(0,141.674,10.454)
		RC	1.27			RC	1.49
			(x,z)				**(x,z)**
Bottom		SVIC	(106.111,12.359)	**Bottom**		SVIC	(-156.452,11.305)
		% AD	25.6			% AS	18.7
Comments:				Comments:			
Config 2 (max anti-dive)				**Config 2 (max anti-squat)**			
	Fore	**Aft**	**(x,y,z)**		**Fore**	**Aft**	**(x,y,z)**
Top		FVIC	(0,249.303,8.252)	**Top**		FVIC	(0,178.929.10.492)
		RC	0.72			RC	1.22
			(x,z)				**(x,z)**
Bottom		SVIC	(52.708,10.967)	**Bottom**		SVIC	(-58.595,10.362)
		% AD	45.8			% AS	45.7
Comments:				Comments:			
Config 3 (Highest Roll Center)				**Config 3 (Highest Roll Center)**			
	Fore	**Aft**	**(x,y,z)**		**Fore**	**Aft**	**(x,y,z)**
Top		FVIC	(0,68.179,9.088)	**Top**		FVIC	(0,72.261,10.384)
		RC	2.34			RC	2.55
			(x,z)				**(x,z)**
Bottom		SVIC	(110.026,13.335)	**Bottom**		SVIC	(-895.659,39.842)
		% AD	26.7			% AS	11.5
Comments:				Comments:			

10.2 Neutral Set-Up Plots

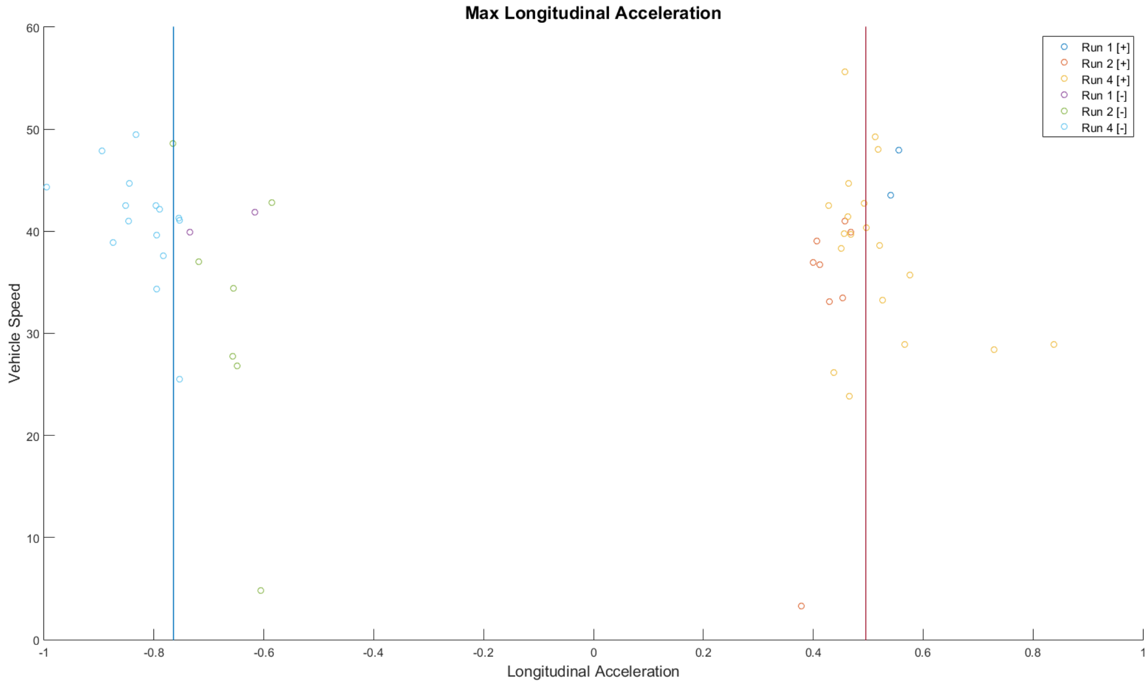

Figure 10: Maximum Longitudinal Acceleration for Neutral Set-Up

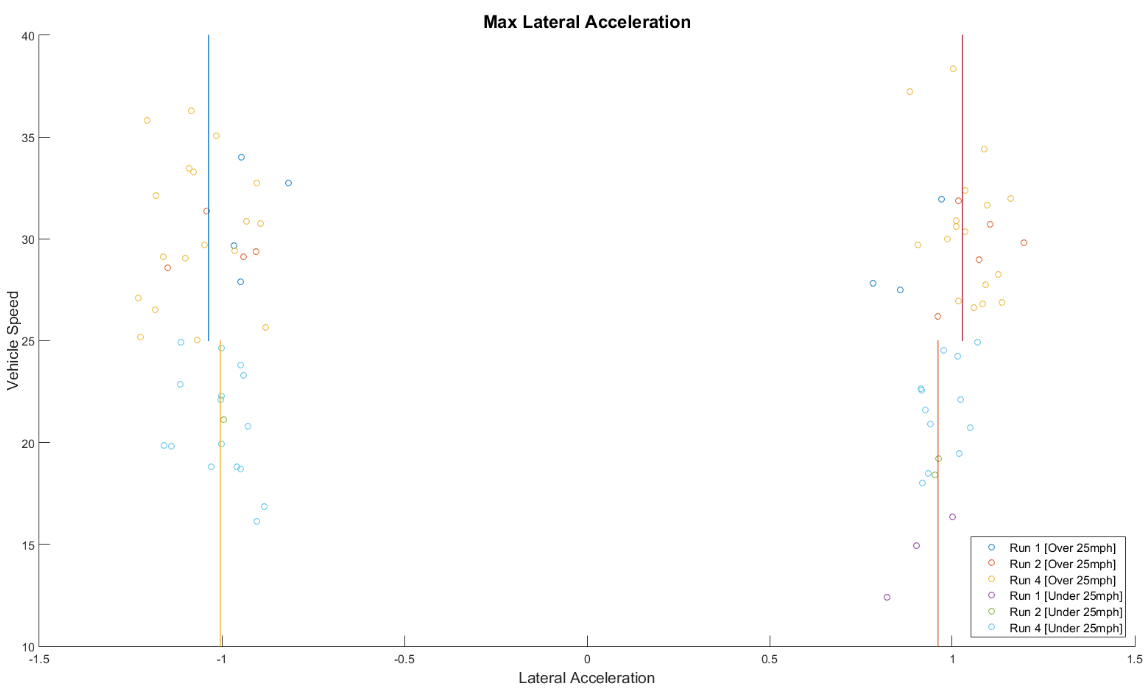

Figure 11: Maximum Lateral Acceleration for Neutral Set-Up

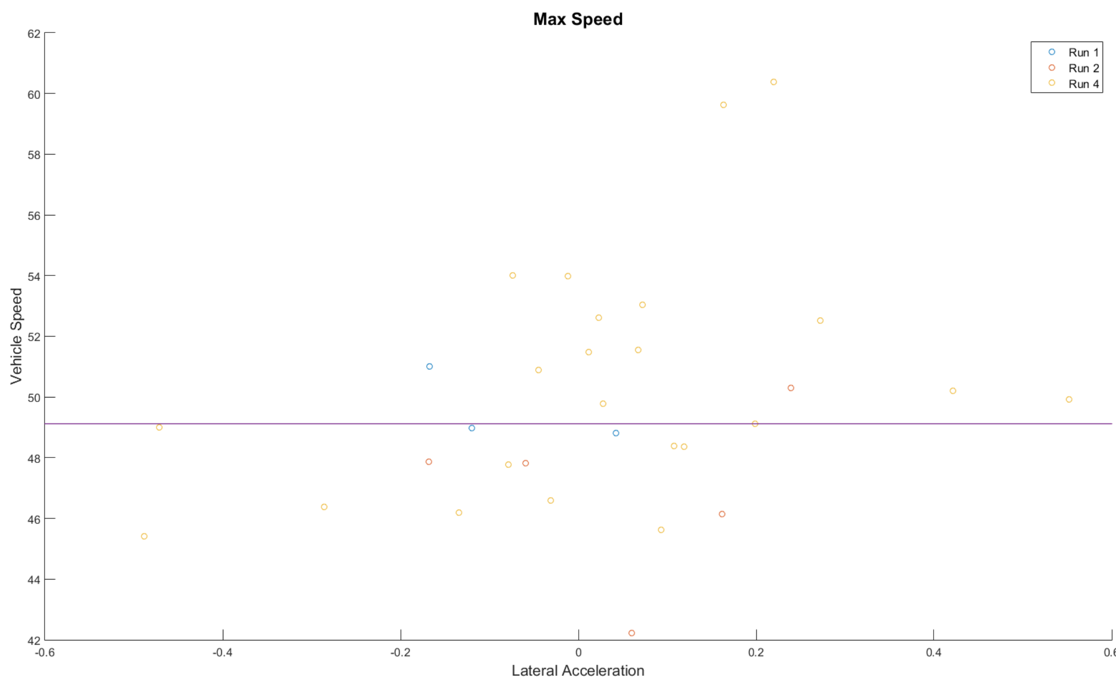

Figure 12: Maximum Speed for Neutral Set-Up

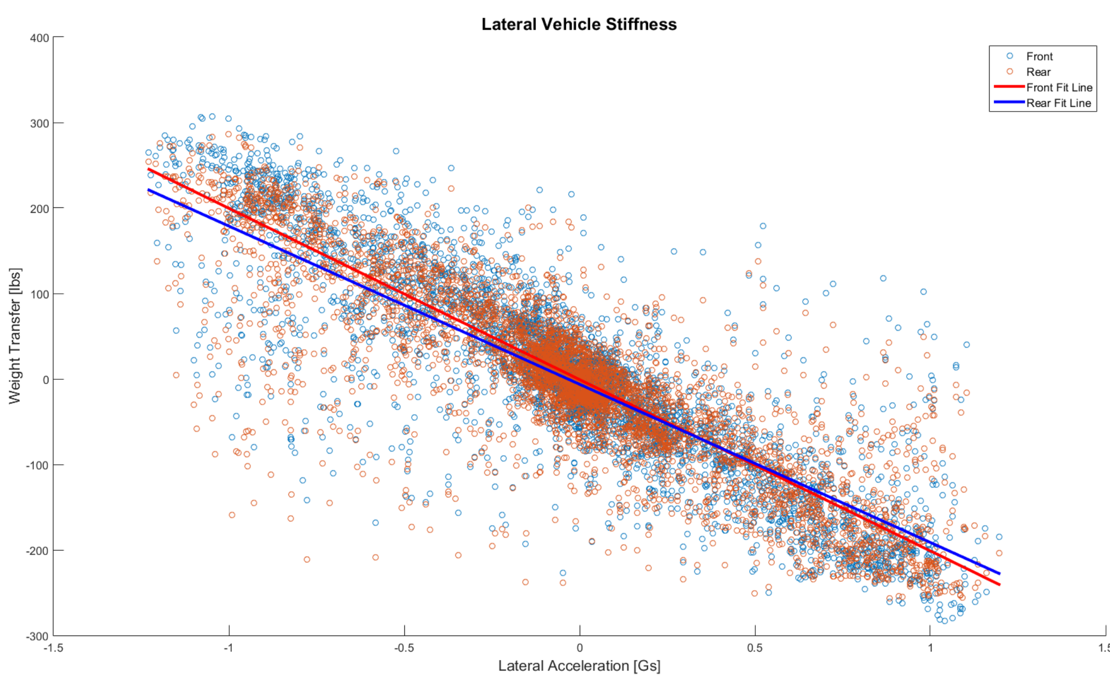

Figure 13: Lateral Vehicle Stiffness for Neutral Set-Up

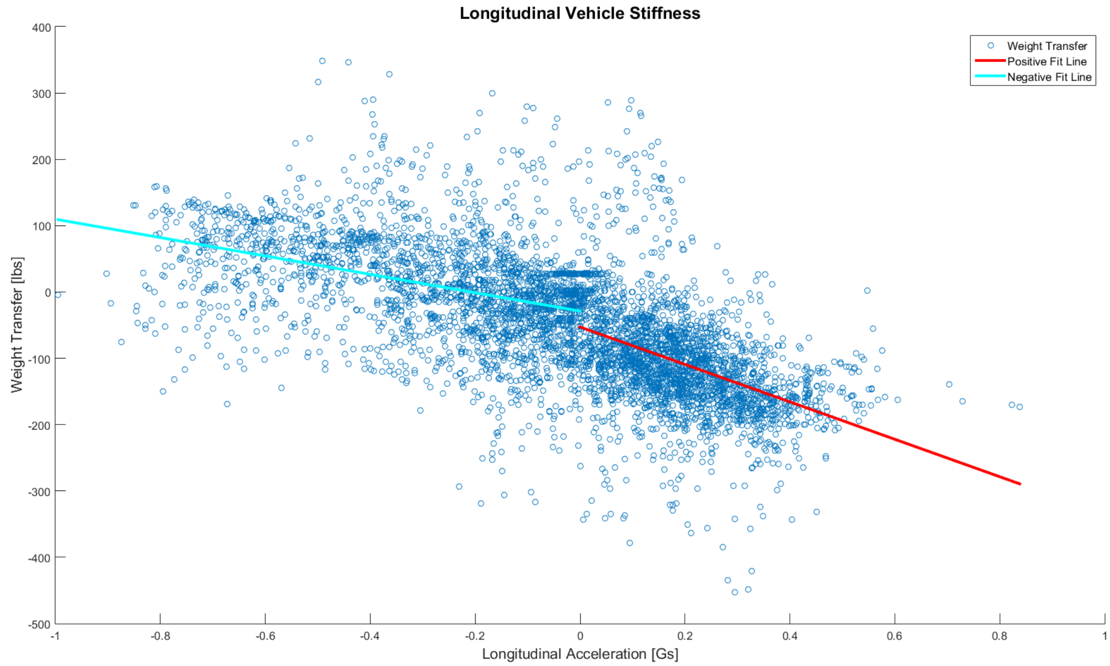

Figure 14: Longitudinal Vehicle Stiffness for Neutral Set-Up

10.3 Alternative Set-Up 1 Plots

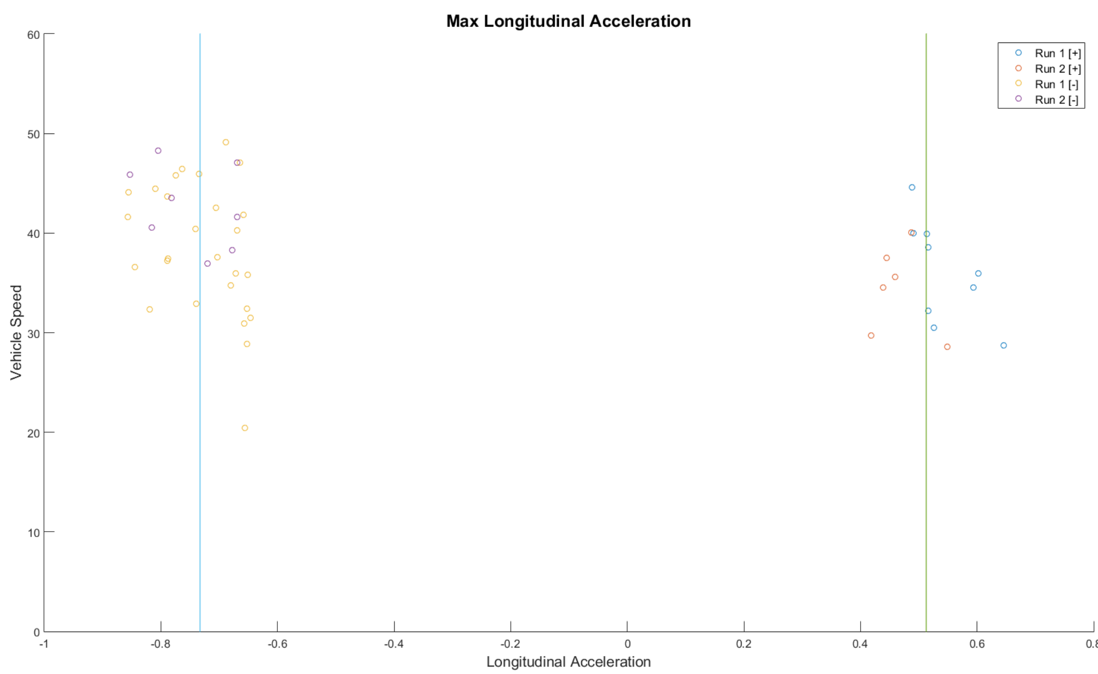

Figure 15: Maximum Longitudinal Acceleration for Alternative Set-Up 1

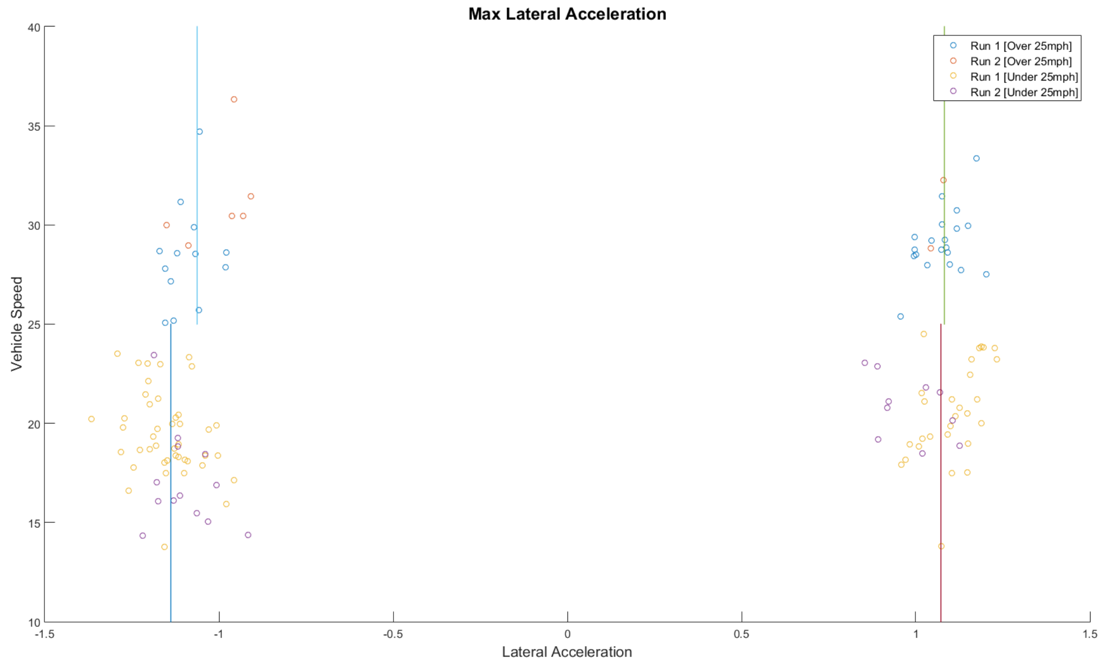

Figure 16: Maximum Lateral Acceleration for Alternative Set-Up 1

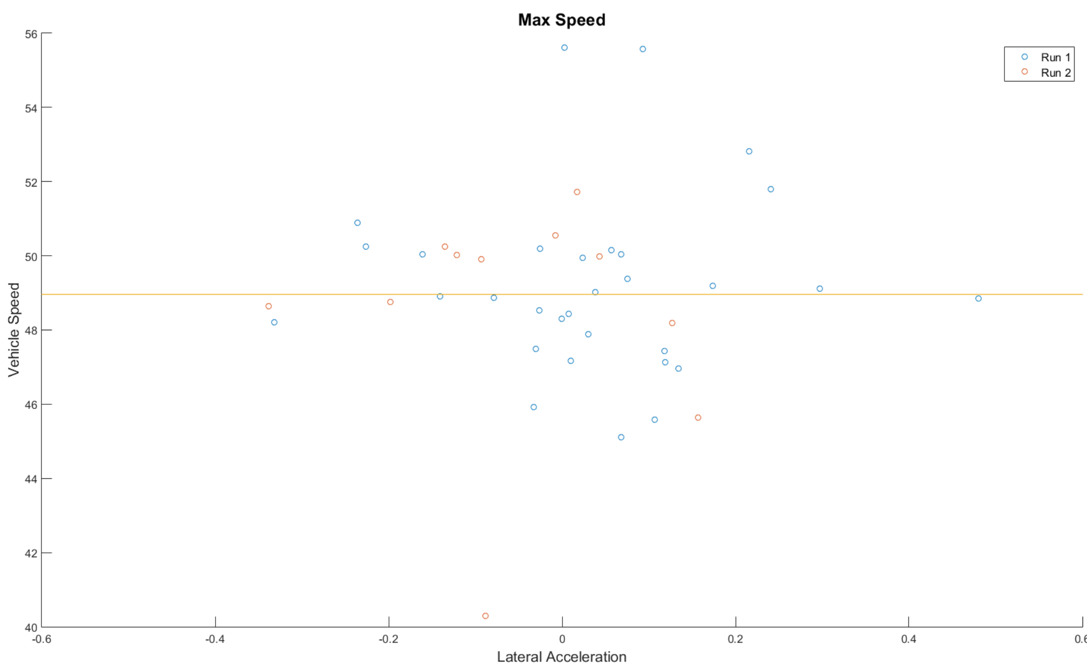

Figure 17: Maximum Speed for Alternative Set-Up 1

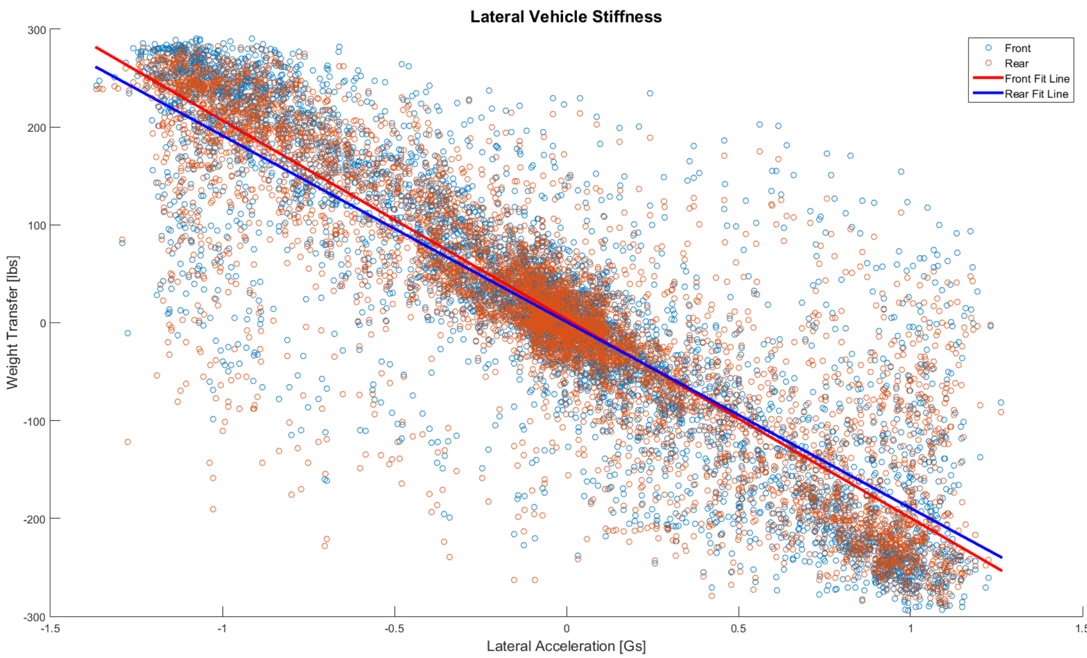

Figure 18: Lateral Vehicle Stiffness for Alternative Set-Up 1

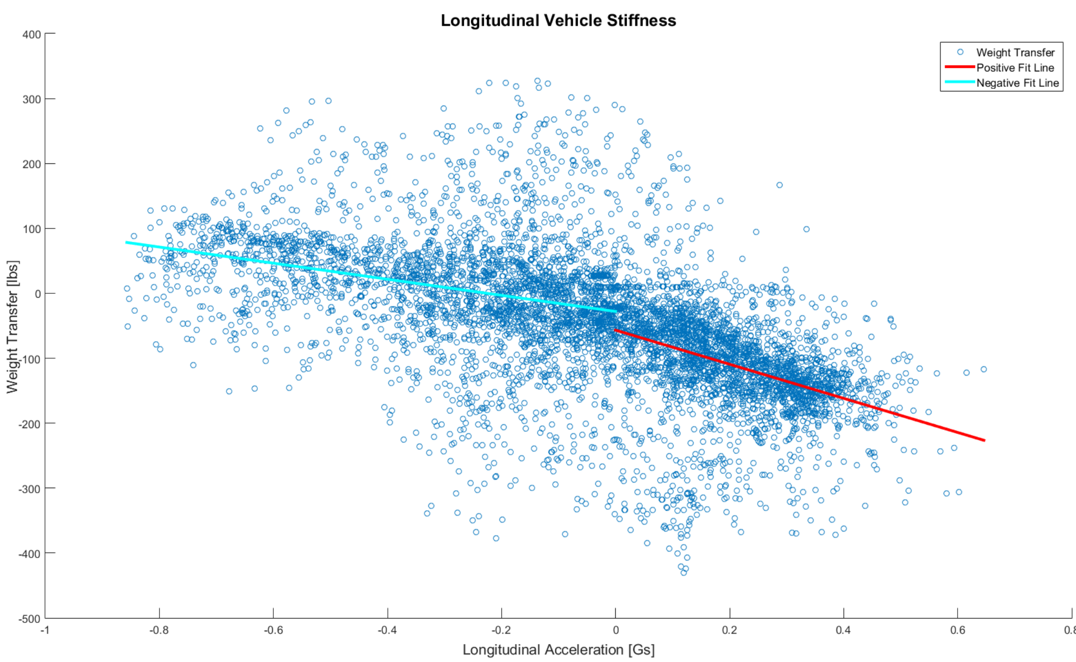

Figure 19: Longitudinal Vehicle Stiffness for Alternative Set-Up 1

10.4 Alternative Set-Up 2 Plots

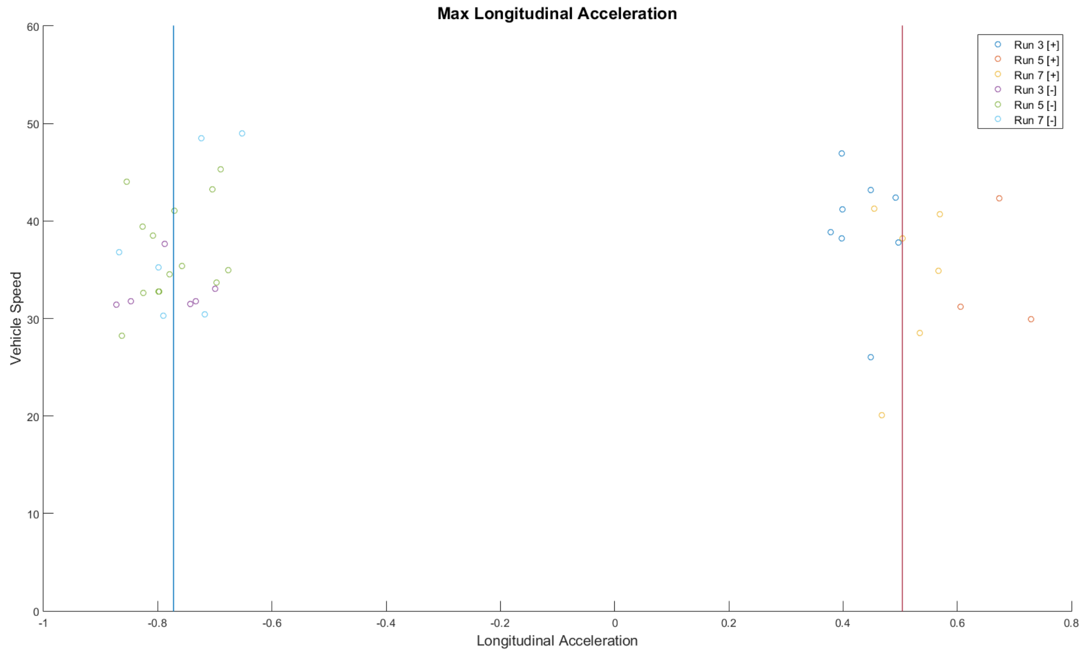

Figure 20: Maximum Longitudinal Acceleration for Alternative Set-Up 2

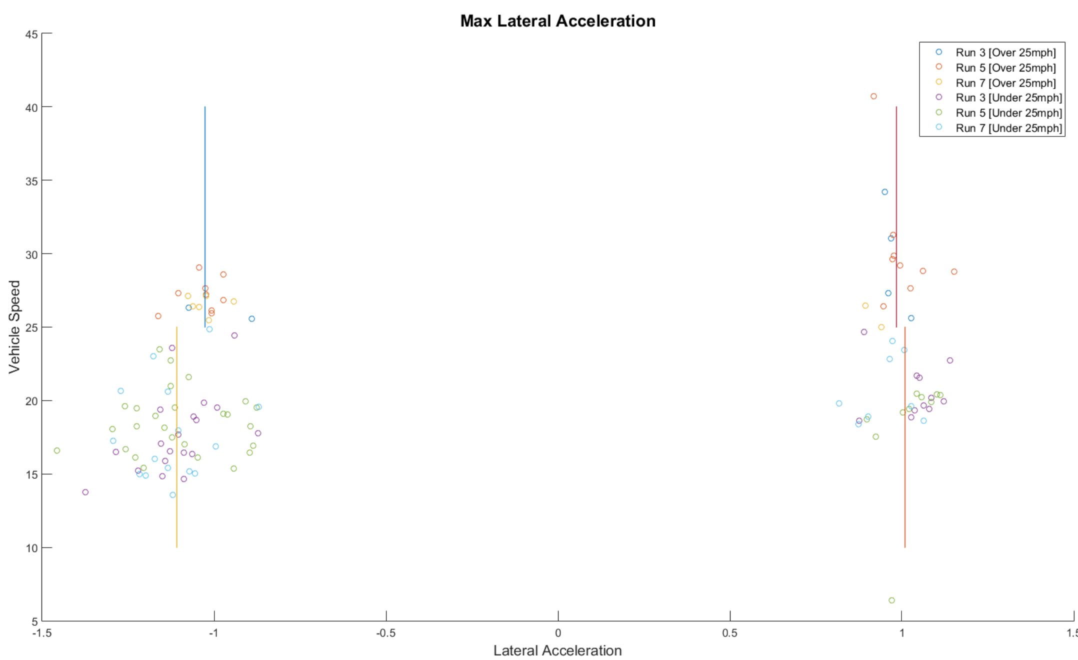

Figure 21: Maximum Lateral Acceleration for Alternative Set-Up 2

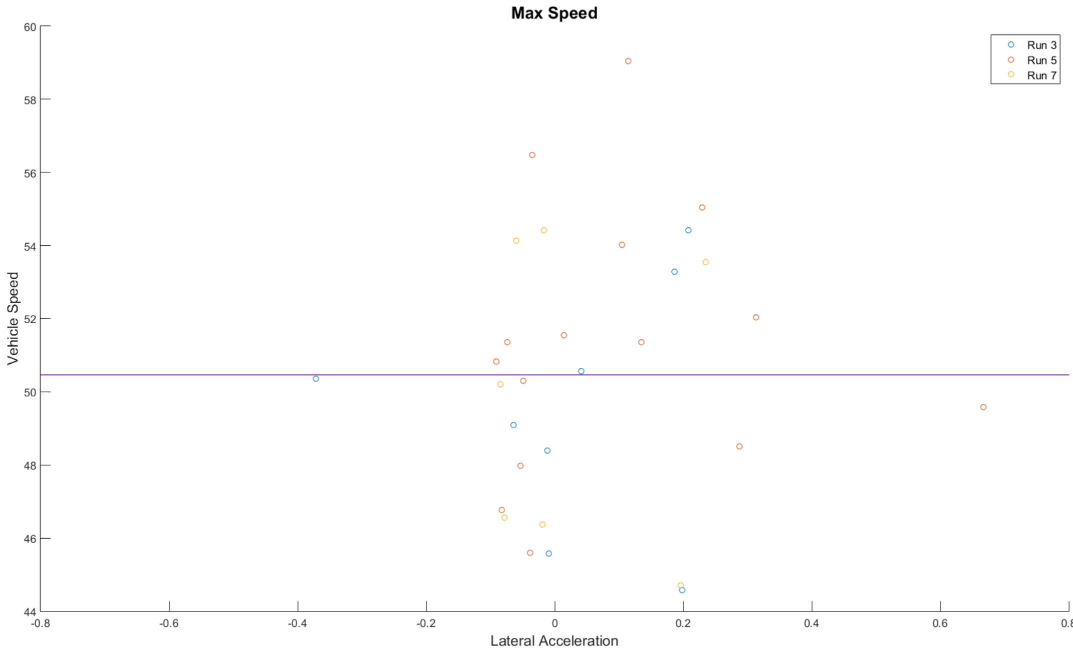

Figure 22: Maximum Speed for Alternative Set-Up 2

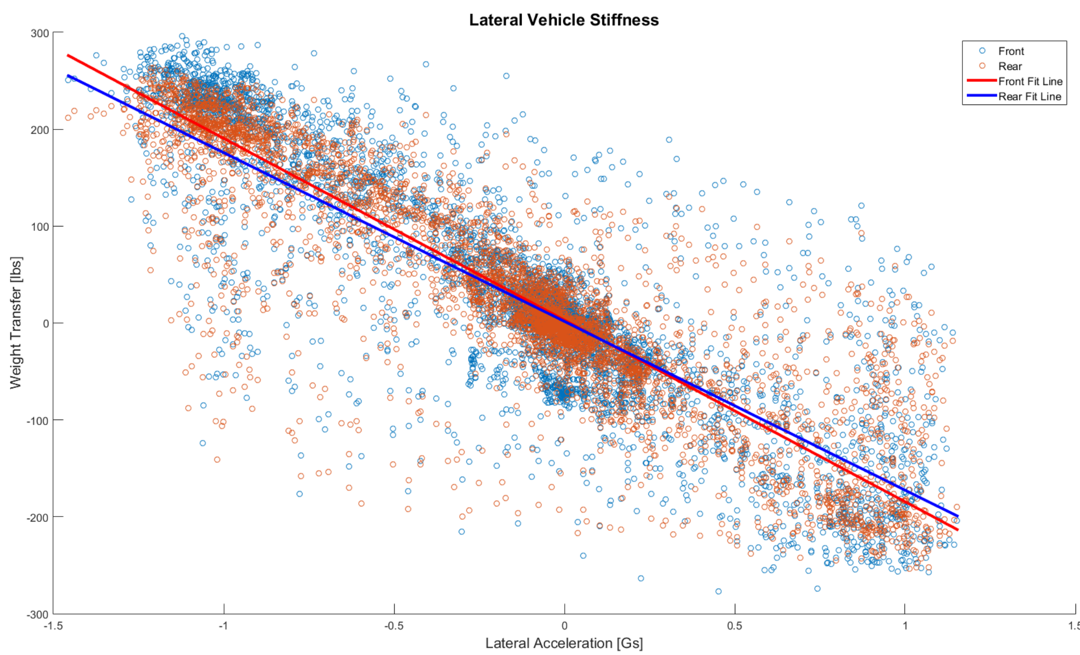

Figure 23: Lateral Vehicle Stiffness for Alternative Set-Up 2

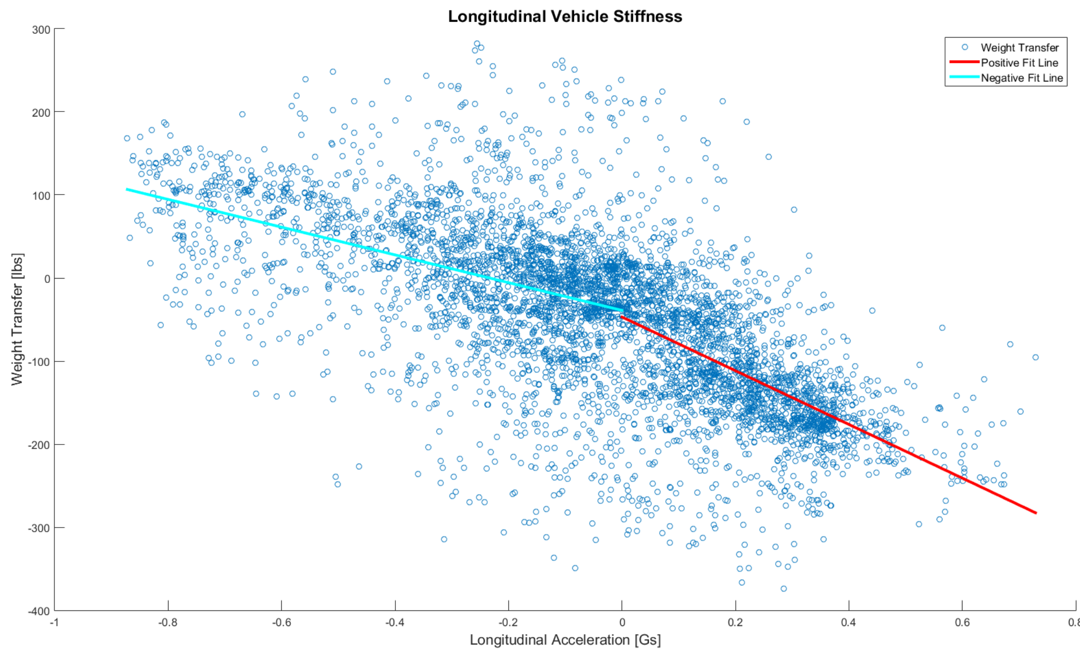

Figure 24: Longitudinal Vehicle Stiffness for Alternative Set-Up 2

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