STOCK CAR SUSPENSION STIFFNESS RATIO ANALYSIS

by

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Submitted in partial fulfillment of the

requirements for Departmental Honors in

the Department of Engineering

Texas Christian University

Fort Worth, Texas

May 6, 2019

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Abstract

A racecar's suspension is one of the key contributors to its performance on a track. Each component – springs, shocks, links, etc. – can be dealt with as a variable within a mathematical model. There are hundreds of combinations of these variables, with each change affecting the overall stiffness ratio of the car. The stiffness ratio is a dimensionless cumulative indicator of the suspension's ability to absorb dynamic load in the front end of the car relative to the rear. The stiffness ratio is a driver-dependent condition that is heavily reliant on driver comfort level, with a higher ratio yielding a more tightly handing racecar and a lower ratio yielding a more loosely handling racecar, and the ideal stiffness ratio is one that is present in the car when the driver feels the most comfortable and can drive the car as quickly as possible. Using the sway bar as the alterable variable of interest and the employment of data acquisition and computer modeling, a mathematical was developed for predicting the stiffness ratio as a function of sway bar diameter. During test runs at Kern County Raceway Park, the driver felt most comfortable with a stiffness ratio of approximately 2.5, which was most closely predicted to be achieved with a 1.25" outer diameter sway bar, using the derived mathematical model. This model can simplify the time-consuming iterative process that is "racecar setup" by allowing a race team to plug numbers into an algorithmic equation to make predictions instead of conducting on-track test sessions to determine the results of each component change.

Equipment

1. RPM Motorsports Super late model racecar
    a. 2016 Hamke Racecar Manufacturing chromoly steel chassis
    b. 2016 Five Star Racecar Bodies Dodge Challenger fiberglass body
    c. Chevrolet 350 cubic inch motor
    d. Subsequent suspension and tires
2. Competition Data Systems data acquisition transducer system
3. Competition Data Systems TrackMaster computer software, Version 6
4. WinGeo Suspension Geometry computer simulation software
5. LISA computer aided design/finite element analysis software, Version 8
6. Microsoft Excel computer software

Introduction and Background

It is without question that automobiles are complex machines, regardless of what they are being used for. This complexity is increased when studying automobiles that are used to race competitively. From an engineering perspective, a racecar can be thought of as a dynamic system that is constantly accelerating and changing condition when on-track. There is an extensive assortment of systems that contribute to a racecar's performance, and one of the most important and integral pieces to performance is the racecar's suspension.

There is some crucial background information that must be explored in order to begin to understand the important aspects of this report. While the average observer may have a hard time distinguishing one type of racecar from another, many different kinds of cars are designed for different types of tracks and environments. The racecar of interest for this project is known as a "*stock ar*," hence the project's name, and the history o stock car racing is integral to understandng the basis for the name of these types of cars.

The basis of the name "stock car" and its subsequent sanctioning bodies were officially initiated in 1948 with the conception of the

National Association of Stock Car Auto Racing, otherwise known as NASCAR. During the prohibition era, people known as "moonshiners" would transport illegal alcohol around the southeastern United States in their everyday cars, and in order to evade law enforcement, they began to modify their cars to make them as fast as possible. Eventually, rumors began to spread about who had the fastest car, and these moonshiners began to race against each other. Eventually, this illicit practice gave way to formal organization at the hands of a man named Bill France, and American car racing was officially invented. Many years later, the term "stock car" was coined based on the nature of NASCAR's conception [3].

A "stock car" is a racecar that is designed to aesthetically look like an everyday streetcar that one might come across in the United States. Today, NASCAR's highest-level premier series hosts Toyota, Chevrolet, and Ford stock cars. At a more local, regionally based level of racing, super late models are a popular type of stock car, and that is precisely the type of automobile that is the subject of this report.

Figure 1a: Example of a Super Late Model Stock Car (Left Side)

Figure 1b: Example of a Super Late Model Stock Car (Right Side)

Figures 1a and *1b* show examples of two super late model stock cars. While the name implies that they should look like casual streetcars, it is apparent that this is not necessarily the case. Aside from the obvious decal differences, the red circles indicate some important aspects that differentiate these super late models from street cars. When these cars are simply sitting at a neutral position and not moving, they appear to be off-kilter, and not necessarily in a perfectly balanced equilibrium. This is not a result of broken components and is in fact the result of an intentionally manipulated *suspension* that is intended to help the cars perform better when they become dynamic systems on an oval racetrack with banked turns.

To describe it simply, the suspension is a network of links, connections, and fasteners beneath a car's body that keeps it suspended off the ground, inversely hanging the car's chassis and subsequently attached parts. The suspension is responsible for absorbing the various loads and changes in road condition in order to keep the car handling properly. This responsibility is amplified with a competitive racecar.

Figure 2: Super Late Model Chassis and Suspension

When dealing a street car, the suspension components are generally standardized, and are not subject to extensive change; *Figure 2* shows that these parts are relatively similar in appearance as well. However, manipulation of a racecar's suspension is integral to achieving peak performance. From an engineering perspective, each component of a racecar's suspension can be thought of as a variable for all intents and purposes. Race team engineers toil over how to properly select values for these variables, from springs and shocks to tire pressures.

The springs and shocks of a racecar's suspension are crucial to its performance. In engineering terms, the spring-shock combination acts as a *2nd-order spring-mass-damper system* at each of the four tires of the racecar. *Figure 3* shows the types of spring-shock systems used on a super late model.

Figure 3: Super Late Model Spring-Shock System

Depending on the type of stock car, the springs may be detached as a separate installment. A super late model uses a *coil-over* system, meaning that the shocks are inserted inside of the springs, so that the springs literally coil over the shocks. It is important to understand how a spring-shock assembly can behave as a variable. The springs can be approached as variables in the sense that different springs have difference *spring rates*. The spring rate is typically a linear constant for a spring in steady state operation and can be defined the amount of force is required to deflect the spring [2]. From the test session utilized for this report, the spring rates in clockwise order starting at the left front of the car were 350 lb_f/inch, 300 lb_f/inch, 175 lb_f/inch, and 150 lb_f/inch. The car's *ride height*, the height at which each corner rests when the car is sitting at neutral position, can be adjusted by compressing or decompressing the springs using the collar on the threaded part of the tower shown in *Figure 3* when the car is at rest.

The shocks' variable behaviors can be adjusted with the *damping ratio* through custom building of the shock itself. Damping is the ability of the shock absorber to bring the springs' naturally infinite oscillation tendencies back towards zero [2]. The damping ratio is responsible for the rate at which the shock compresses and decompresses when subjected to load.

As mentioned previously, a suspension contains numerous links and connections, some of which connect the tires to the frame, some of which connect the axles to the frame. An example of this is the A-arm shown in *Figure 4*.

Figure 4: Super Late Model A-arm

The A-arm is referred to as such because of its triangle shape. Links like the A-arm can be treated as variables because they can be changed in length and thickness, also contributing to the way a car handles.

For the sake of this report, all of the previously mentioned variables were held as constants. The variable of interest was the *sway bar*, meaning that the sway bar was the variable that was manipulated as the dependent variable in research exploration. The sway bar is a hollow tube that extends from the left front corner of the racecar to the right front corner, effectively connecting the front corner suspensions together.

Figure 5: 1.25" Outer Diameter Sway Bar (left), 1.75" Outer Diameter Sway Bar (right)

The sway bar effectively behaves as a torsional spring when the racecar is in motion, specifically in the turns. This is best thought of intuitively. When the car is going around a left turn on a circle track, the right front tire travels a greater distance than does the left front tire, due to the greater displacement from the center of the imaginary circle that is the width of the racecar, and is therefore subject to different loads. However, due to obvious constraints, the right front tire must remain even with the left front tire. The sway bar torsionally twists to absorb the loads necessary to maintain the right front even with the left front, and then releases these loads when the car is out of the corner. In this sense, the sway bar is a torsional spring.

Clearly, each of these important variables present an opportunity to alter the way that a car performs. Racecar engineers will often reference the *stiffness ratio* to determine the state of the car's suspension with any particular setup. This ratio differentiates the front suspension from the rear suspension.

To put it simply, the value of *stiffness* is a measure of how much potential dynamic load can be absorbed by the overall suspension in either the front or the rear. The stiffness is essentially a value that accounts for each suspension component in each of the front and rear ends and expresses a cumulative sum of each of the suspension components. The stiffness is expressed in units of in.-lb_f/degree. The front stiffness is given by *Equation 1* and the rear stiffness is given by *Equation 2*, shown in the Analysis segment of the report.

It is important to understand that the stiffness ratio is a value that varies by driver preference and track. The higher the stiffness ratio, the more tightly the racecar will handle, and the lower the stiffness ratio the more loosely the racecar will handle. This again makes sense when approached intuitively. When the stiffness ratio is higher, more dynamic load can be absorbed by the front of the car, so the front end will be less inclined to turn around the corner; this is the definition of a tight racecar. When the stiffness ratio is lower, more load can be absorbed in the rear of the car, and the rear end will be more likely to spin out; this is the definition of a loose racecar [1]. Some drivers prefer a tighter racecar, others prefer a looser racecar, and will drive the car at its fastest potential at a level of stiffness ratio at which they are most comfortable.

The sway bar finds itself as an integral part of the stiffness ratio calculation, as it is a component that is unique to the front-end suspension. Because of the sway bar, the stiffness ratio will almost always be a value that is greater than one, save for a few unique circumstances. The sway bar is the variable of interest for this project because of its importance to the stiffness ratio.

The shocks and springs, links and connections, and sway bar are all integral pieces of the puzzle to maximize a racecar's handling and subsequently its performance. The value of stiffness is representative of the culmination of all of the suspension parts in the front and rear ends of the car and can be though of as the amount of potential dynamic load that can be absorbed by that end of the car. The "magic" in creating a fast car lies in the ability of a race team engineer to select components that will get the front-to-rear stiffness ratio to a value at which the driver feels the most comfortable.

This is more easily said than done, and racecar engineers work tirelessly to optimize the car for the driver. Currently, the mode of selecting a component variable value involves physically changing a component during a series of test sessions and seeing if it works. To use a relevant example, a racecar might be too tight on a track

for a driver's preference, so the engineer might suggest bringing in the car, installing a new sway bar, and sending the car back onto the track in the hopes that the change will have worked. Oftentimes, this kind of change does not make a significant impact immediately, and this iterative process can exhaust a lot of time, fuel, tires, and other important resources. Oftentimes, a driver will have to settle for a mediocre car in a race that they are not completely comfortable with because there was not enough time to test each variable change in the suspension. Enter the goal of this project, which is to create a predictive mathematical algorithm centered around the sway bar. With a known stiffness ratio that the driver prefers at a particular track, this predictive algorithm will be able to give an answer for the value of the outer diameter of a sway bar that can be installed into a super late model racecar, with all other suspension component variables held constant, to achieve peak performance. This will allow for the conservation of important racing resources, and more time to improve other aspects of the car besides the outer diameter of the sway bar installed.

Methodology

As with most research methods, this project began with the process of data collection. It is fair to wonder what kind of data could be collected in order to synthesize a mathematical algorithm to predict the ideal value of a sway bar at a particular track.

As mentioned in the background segment of this report, the stiffness ratio is a value that varies by track and driver preference. Data was gathered during a test session at Kern Country Raceway Park (KCRP) in Bakersfield, California. The super late model that was used in this test session is shown in *Figure 6.*

Figure 6: RPM Motorsports #33 Dodge Challenger Super Late Model

This super late model, owned by RPM Motorsports, is a 2016 Dodge Challenger fiberglass body manufactured by Five Star Racecar Bodies. This body sits on top of a 2016 Hamke Racecar Manufacturing chromoly steel frame. The car contains a 350 cubic inch Chevrolet motor that averages about 600 horsepower, and the car contains its own unique suspension that is constantly manipulated to achieve peak performance.

Kern County Raceway Park is a 0.5-mile paved oval track that has relatively high banking in the corners and straightaways. Nearly all circle track race courses have banking towards the center of the track, a physical necessity, like a banked freeway interchange. This is useful to know, but the derivation of how this baking works is outside of the scope of this report. For a super late model, a half-mile track is on the larger end, and they tend to be rather challenging because of the high speeds that drivers and cars experience and the forces present due to these high speeds. Half mile tracks become physically demanding for both the driver and the car. At Kern County Raceway Park, the driver that drove the car during the test session was able to drive the car at its fastest lap times and felt the most comfortable when the stiffness ratio was approximately 2.5, as determined from *Equations 1-2.* This means that, effectively, the car was able to turn the fastest lap times when the front end was comprised of components that were able to potentially absorb about 2.5 times the dynamic load than the rear suspension.

In order to collect data, a data acquisition system manufactured by Competition Data Systems was utilized. For automotive racing, a data acquisition system is a variety of transducers and sensors that are attached to a car at various pickup points. This is shown below in *Figures* 7 and *8*.

Figure 7: Components of a Racing Data Acquisition System

Figure 8: Transducer Attached to a Pickup Point

A sensor was placed at the start/finish line on a racetrack, and each time the car passed this sensor the data readings gathered by the transducers were sent to the sensor, which were then relayed to a form of memory system or directly to a computer. The data acquisition system utilized had transducers capable of reading temperature, tire slip angle, momentary loads, and a variety of other values. The readings that this project concerned itself with were the G-force values recorded, both lateral and vertical. For the sake of convention, vertical G-forces were recorded along the height of the car, and lateral G-forces were recorded along the width of the car. These G-force readings were shown using the Competition Data Systems TrackMaster Version 6 computer software, which allowed the user to navigate each transducer output.

To simulate a real lap using the eventually proposed mathematical model, a series of conditions were defined at which the data gathered by the transducers were defined as data points. There were five total conditions, each occurring at a certain distance beyond the start/finish line, including the neutral position of the car at rest. Visual representations of these conditions are shown below.

Figure 9: Condition 0, Car at Neutral Resting Position

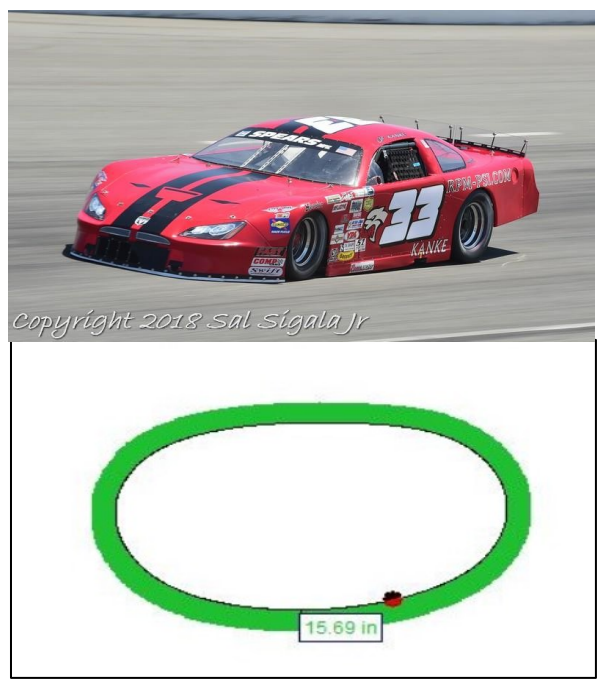

Figure 10: Condition 1, Car Braking into Turn 1 (262.5 ft.)

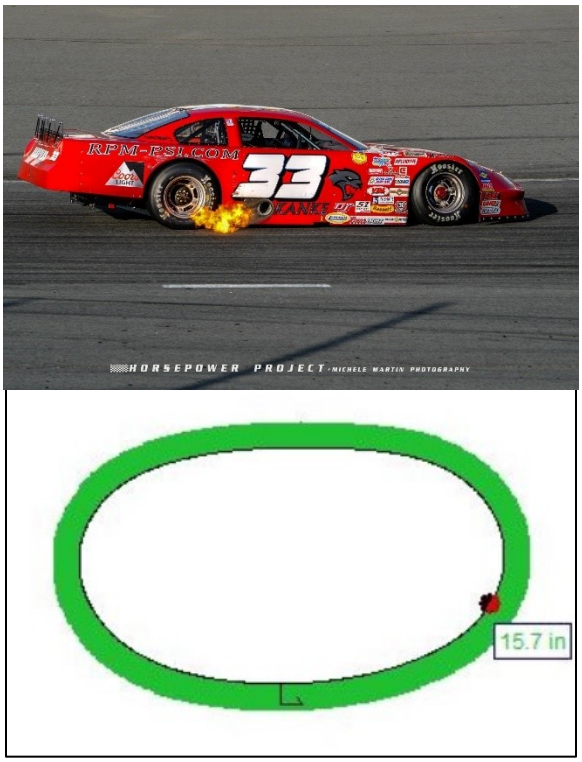

Figure 11: Condition 2, Middle of Turns 1 and 2 (606.1 ft.)

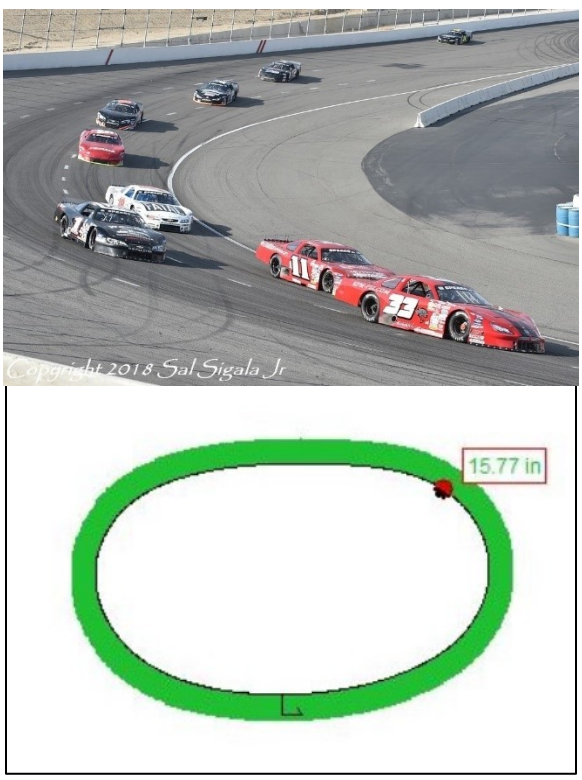

Figure 12: Condition 3, Accelerating off Turn 2 (1003.9 ft.)

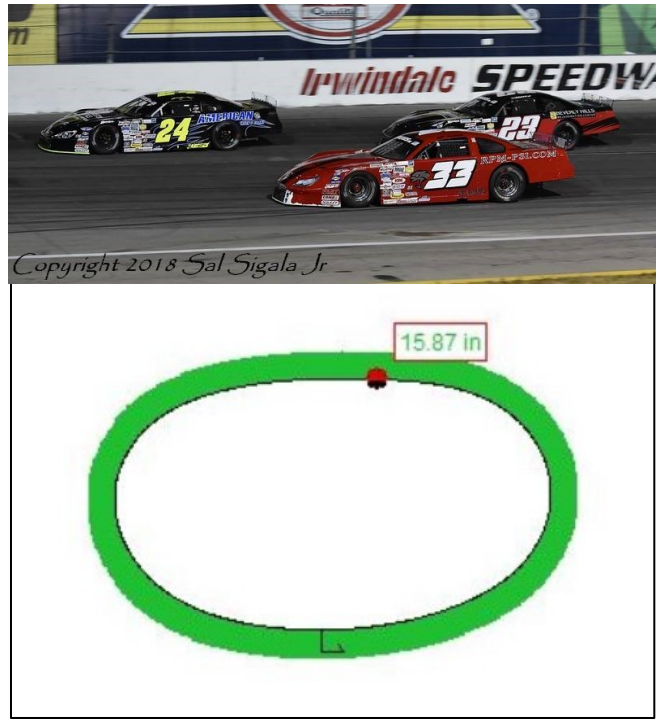

Figure 13: Condition 4, Full Throttle on Back Straightaway (1325.5 ft.)

Defining these conditions of the car on the track allowed the subsequently utilized computer programs to simulate a car going through a corner on the racetrack, the portion of the track that the suspension bears the greatest change in load. The G-force readings that were gathered at each of these conditions are shared in the Results segment of this report.

Due to the banking of the track, these G-forces acted on the racecar in multiple directions. Because of this, the G-forces recorded at each condition were able to be reduced into three-dimensional x, y, and z-direction forces that were modeled as acting at each of the four tires of the car. These reductions were conducted in Microsoft Excel and were applied to a computer aided design (CAD) of the car's chassis and suspension components.

Using a CAD software called LISA, the car's chassis and suspension components were modeled. This software was also utilized to perform a finite element analysis (FEA) on these components. An FEA that is conducted on a model provides the ability for a user to see the various displacements, loads, and stresses that are present on any individual component that is part of the CAD by "slicing" the components into a

mesh that becomes manipulated by a simulated load. The loads at each condition that had been reduced from the G-force readings were the loads that were applied to the CAD during the FEA of the racecar. Although Condition 0 was not a relevant condition to the design of the mathematical model of the project, *Figure 14* shows an example of the CAD and FEA of the front suspension of the racecar when subjected to the three-dimensional loads at neutral position. As expected, these loads at neutral position were solely in the vertical direction, the normal forces present on the tires when the car was not in motion.

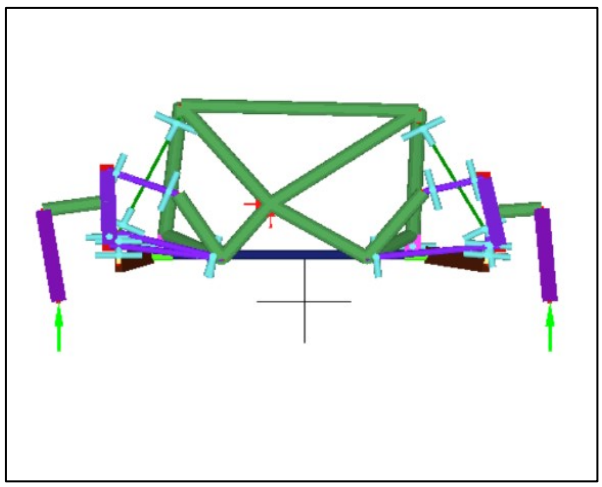

Figure 14: CAD and FEA of Racecar's Front End at Condition 0

In this CAD model, the tires were modeled as the purple rectangles at the outside edge of the frame, and the sway bar was modeled by the horizontal bar at the lower part of the model, highlighted in a navy-blue color. The green arrows on the tires acted as the simulated three-dimensional forces for the sake of the FEA.

Each suspension component was treated as a variable within this CAD model, just as they can be in the real-world manipulation of a suspension. Instead of physically swapping out the sway bar in the racecar during this test session, the value of the sway bar's outer diameter was altered within the CAD model in order to see the effects that such an alteration would have on the FEA results. To reiterate, the range of sway bar outer diameters tested was 1.25" to 1.75", and these values were altered by 0.0625" within the CAD model. While the FEA provided a large amount of useful data, like the data acquisition system, the values of interest were the readings that is supplied for shock length and tensile force present on the shocks. This, again, can be understood intuitively. The loads from both the weight of the car and its acceleration will transfer around the car dynamically as the car transitions between brake and throttle and the velocity vector of the car itself changes direction through each of the four on-track conditions. These changing loads will be absorbed by the front and rear suspensions in different distributions, so the physical loads on each of the shocks will change, resulting in a change in shock length. This demonstrates the importance of the damping ratio of the shocks and the rates of the coil-over springs mentioned in the Background segment of this report. Once these tensile force and shock length values were gathered from the FEA iterations with varying sway bar outer diameter, they were plotted as a function of the sway bar outer diameter. These plots are shown in the Results segment of this report.

The final computer program utilized after the CAD/FEA software was a suspension geometry simulation software known as WinGeo. This software allowed for a visual walkthrough of the suspension's physical movement through each of the four on-track conditions by way of the shock lengths simulated by the FEA. In order to use this software, the shock length values for each sway bar diameter through each condition were written into a type of computer code file known as a .pth file, pronounced "path file." WinGeo utilized this path file to create a visual path of conditions to simulate. This .pth file was uploaded into the suspension geometry simulation software, and a visual representation of the car transitioning through the four on-track conditions for each of the possible sway bar outer diameters was generated. Alongside the creation of this walkthrough simulation, the WinGeo software also generated a variety of values, including the steering angle and the camber angles for each of the front tires. Important to the goals of this project, the suspension geometry simulations also provided a value for the overall twist angle of the right side of the sway bar relative to the left side, simulating the sway bar's behavior as a torsional spring through a corner.

Figure 15 shows the WinGeo software's display at Condition 0, in the same state as the CAD model in *Figure 14*.

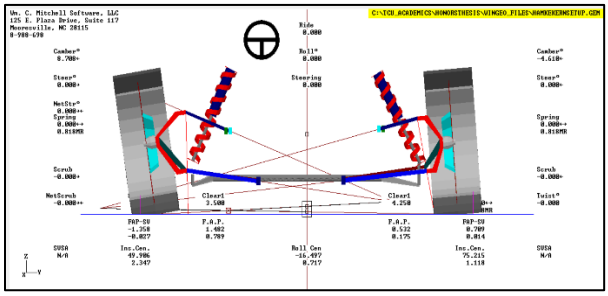

Figure 15: WinGeo Suspension Geometry Simulation at Condition 0

As with the shock tensile force and shock length values generated by the FEA simulation, the value of the sway bar's twist angle was plotted as a function of its outer diameter. This plot is shown in the Results segment of this report.

Each of these values generated by the FEA and the suspension modeling software furthered a necessary understanding of the suspension's behavior when subjected to a variety of different conditions. It is important to recall that the stiffness ratio, one of the main values of interest to this project, is a value representative of potential load bearing, and is a summation of the current setup that will not ultimately change when the car is on the track. However, these resultant shock tensile force, shock length, and sway bar twist angle values are necessary to consider in the practicality of achieving a certain stiffness ratio. Not only do they provide an understanding for what is physically occurring within the car's suspension, but they also may negate the possibility of altering a suspension component for the sake of the ideal stiffness ratio due to design restrictions. Understanding these simulation results is necessary in design in the sense that they help narrow down the realm in which a racecar engineer may work the most efficiently and realistically. For example, making a theoretical sway bar change that stretches a shock outside of its physical bounds is obviously not feasible.

Using *Equations 1* and *2* from the Analysis segment of the report to determine the front and rear stiffness values for each of the possible sway bar outer diameter values, the stiffness ratio at each of these sway bar outer diameter values was subsequently calculated. These calculations provided a set of data for the stiffness ratio at each of the possible sway bar outer diameters, and the stiffness ratio was plotted as a function of the sway bar diameter. This plot was fitted with a curve that best encapsulated the data points and is shown in the Conclusions segment of this report. Once the most accurate type of curve fit was selected using Microsoft Excel, an equation was developed to summarize the curve. This equation became the solution to this project, the predictive mathematical algorithm that would allow a racecar engineer to predict the sway bar that would achieve a driver's preferred stiffness ratio at Kern County Raceway Park.

Results

Figures 16-19 show the selected output readings of the data acquisition system in the TrackMaster Version 6 display software for each condition. This was effectively the data measurement of the project. The circular plots in the upper corners show where the car was on the track at each condition, and the plots in the lower corners show both the lateral and vertical G-force values recorded by the attached transducers.

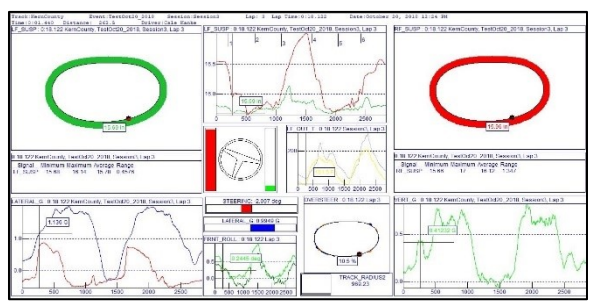

Figure 16: Condition 1 G-force Data

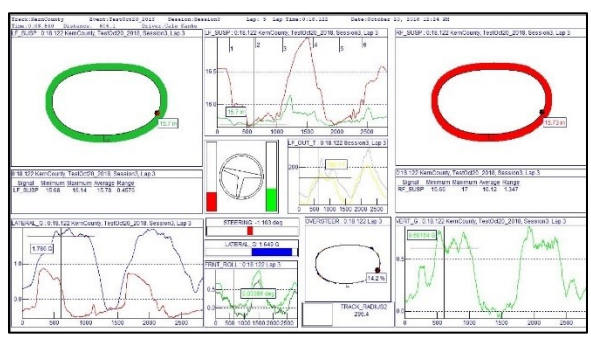

Figure 17: Condition 2 G-force Data

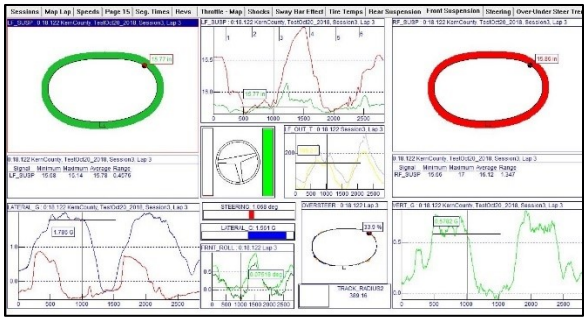

Figure 18: Condition 3 G-force Data

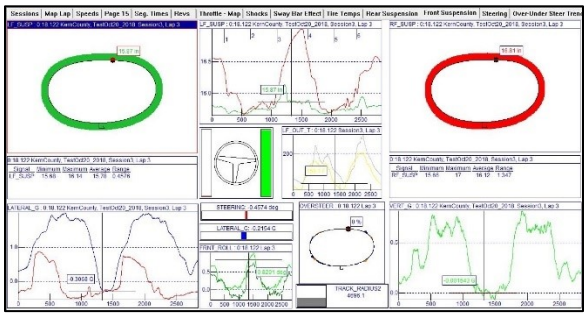

Figure 19: Condition 4 G-force Data

There are some trends in this data that are worth noting when the results are plotted and can be visually referenced. *Figure 20* represents the tensile forces present on each of the front shocks present when the car is moving through the corner with each potential sway bar diameter. The values for the tensile forces are negative values, essentially meaning that the shocks are in compression. By looking at the characteristics for each of the four conditions, the closer the racecar is to the center of the corner, the more compressive the force is on the shock. As the sway bar becomes wider in outer diameter, the left shock becomes subjected to a higher compressive load, while the compressive load on the right front shock decreases. When comparing *Figures 20* and *21*, it becomes apparent that the lengths of the shocks are directly proportional to the forces present on them; the plot characteristics follow the same shape.

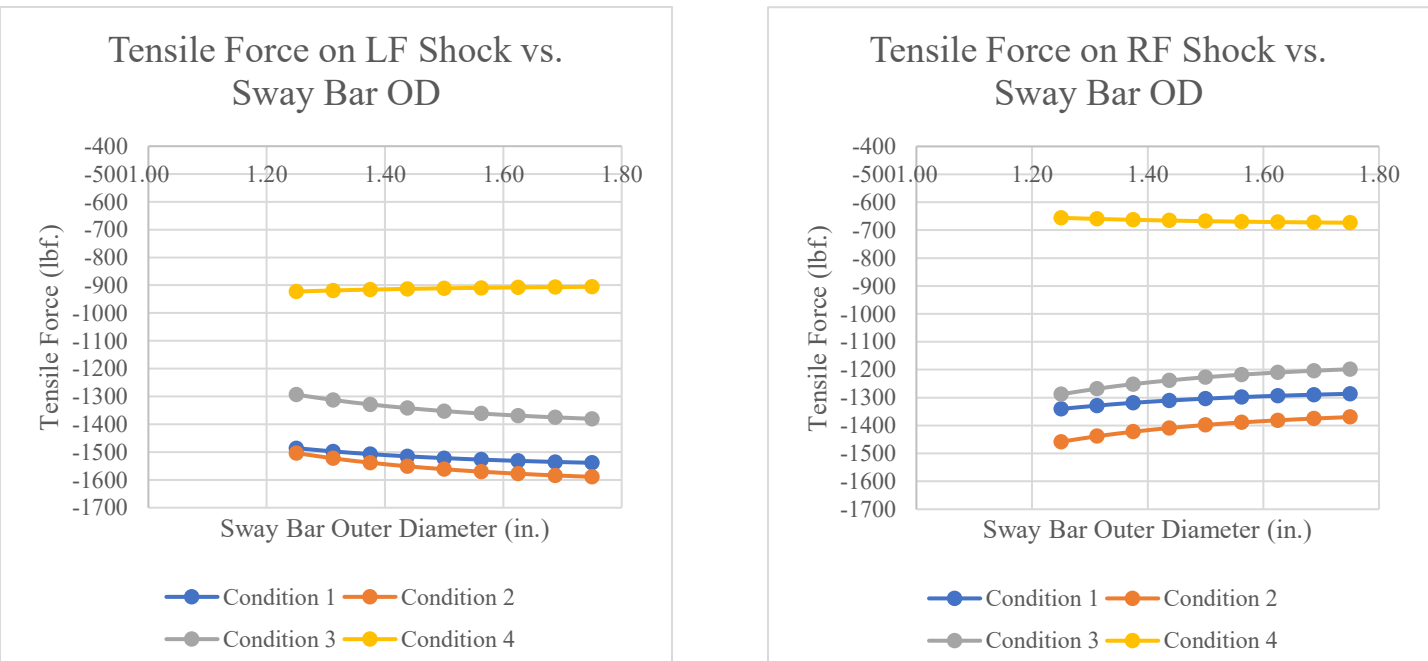

Figure 20: Front Shock Tensile Forces vs. Sway Bar Outer Diameter

Figures 20-22 are plots generated by Microsoft Excel that show the front suspension shock tensile forces, the front suspension shock lengths, and the sway bar twist angle for each of the four on-track conditions as a function of the theoretical sway bar outer diameter.

Figure 22 shows the plot of the left to right twist angle present in the sway bar for each of the conditions as a function of the sway bar diameter. While this concept was touched on previously, it is important to note that the twist angles converge towards zero for each condition.

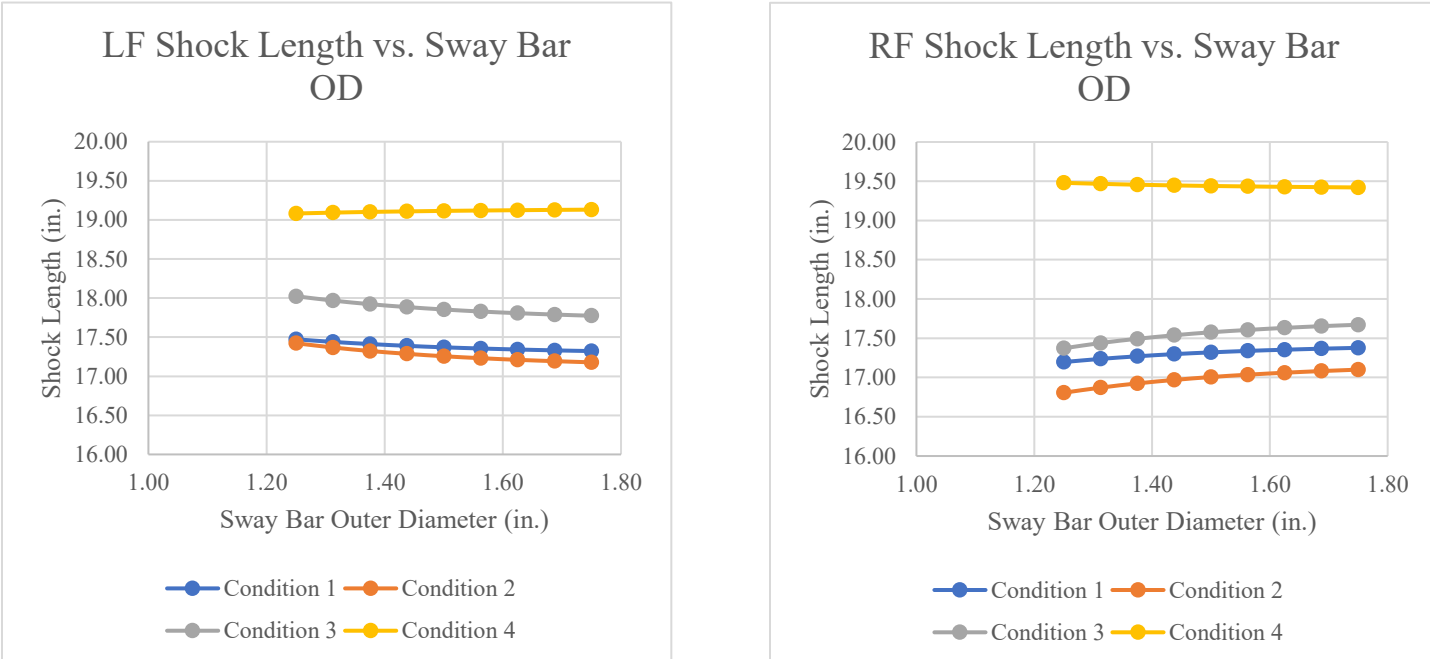

Figure 21: Front Shock Lengths vs. Sway Bar Outer Diameter

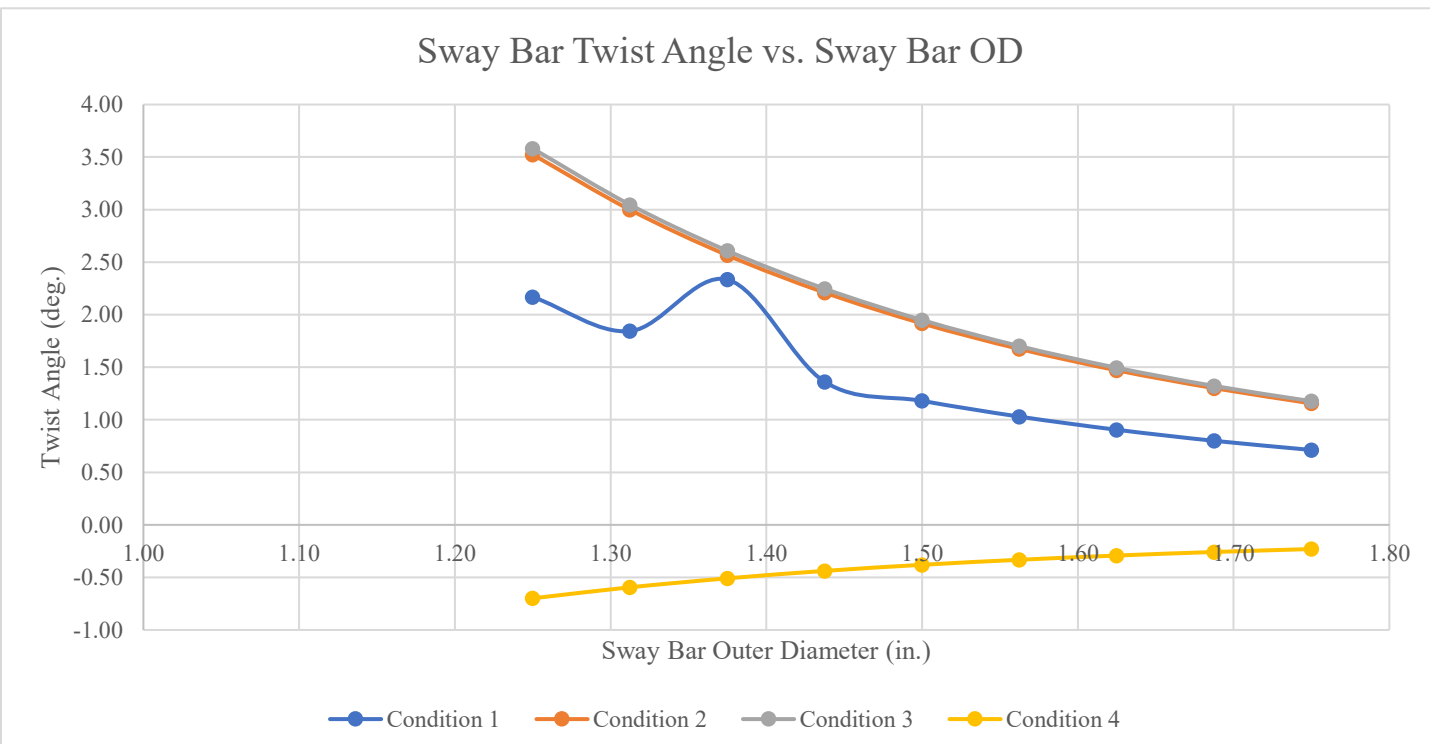

Figure 22: Sway Bar Twist Angle vs. Sway Bar Outer Diameter

The convergence towards zero apparent in *Figure 22* makes sense intuitively. As more material becomes present in the sway bar, it develops a higher resistance to a twisting motion, (i.e. higher torsional stiffness) so the simulated amount of twist present will decrease with an increase in sway bar outer diameter. It is apparent that there is an outlier data point for the twist angle at Condition 1. This outlier point is demonstrative of the imperfections present in an FEA conducted on a CAD model. Another important feature of this plot is the nearly identical nature of the twist angle trend at Conditions 2 and 3. These plotted characteristics lie nearly directly on top of each other. This can be attributed to the fact that at both of these conditions, the racecar has transitioned from a braking condition to an accelerating condition, but while this change in speed is occurring the car is still accelerating by changing its velocity vector by way of directional change, as the driver is still turning around the corner in both of these conditions. The driver is nearly full throttle at both Conditions 2 and 3 but is still turning the steering wheel to get the car around the corner.

Analysis

While there were a variety of calculations conducted by the computer programs utilized for this project, including the reduction of the G-force readings to three-dimensional localized forces and the calculations conducted within the FEA software, the value for stiffness had to be calculated "by hand" for each of the sway bar component changes. The equations for both front and rear stiffness are given in *Equations 1* and *2*.

$$S_F = 0.01745[{P_{LF}}^2(W_{LF} + W'_{LF}) + (T_F + P_{LF})^2(W_{RF} + W'_{RF})]$$

Equation 1: Front Stiffness

$$S_R = 0.01745[({P_{LR}}^2 T_R) + W_{RR}(T_R - P_{LR})^2$$

Equation 2: Rear Stiffness

In each of these equations, the subscripts denote the racecar's end or corner of reference. F and R denote rear and front, respectively. LF is left front, RF is right front, LR is left rear, and RR is right rear. The value *W* is the value of what is known as the *"wheel rate"* within a corner of the suspension. The wheel rate is what allows the stiffness values to be associated with the load bearing capacity of the suspension. The derivation of the value of the wheel rate is outside of the scope of this project, but it is important to know that the wheel rate at each corner is a value that accounts for that corner's tire air pressure, shock damping ratio, and coil-over spring rate. The prime value of the wheel rate is only applicable to the calculation of front stiffness, and this is because of the presence of the sway bar. Because the wheel rate is the value that essentially accounts for the bulk of the load bearing capacity, the sway bar's torsional capacity must be considered as well. For this reason, the prime value of the wheel rate is the wheel rate due to the torsional load bearing capacity of the sway bar that is unique to the front suspension of the car.

The value of *T* is the value of the *track width* of the racecar. This is a measured value that can be altered using the variability of the rods and connections in the suspensions. The track width is essentially the width of the car, the value in inches spanning the outside edges of the front tires for the front track width and the outside edges of the rear tires for the rear track width. The value *P* is the value of *pivot* that is present on the left side of the car. Because the racecar of interest was tested on a circle track, only left turns were made, so only the left side pivot was relevant in setup calculations. The value of pivot is an analysis of the track width compared to the wheel rates and can be described as the amount of travel that the left side of the car experiences relative to the right side due to the load bearing capacity represented by the wheel rates.

The calculations for each of these values, and eventually the stiffness values, can be tackled by knowing the values for each suspension component variable. To reiterate, the goal of this project involved holding each of these component variable values constant with the exception of the sway bar outer diameter, which affected the prime value of the front wheel rates.

In order to obtain the stiffness ratio, the front stiffness was divided by the rear stiffness, affirming the stiffness ratio's existence as a dimensionless ratio value.

Conclusions

The creation of a mathematical algorithm to give the necessary sway bar outer diameter value based on a driver's known stiffness ratio was contingent on the creation of a characteristic plotting the stiffness ratio as a function of the sway bar outer diameter. The calculation of the stiffness ratio is covered in the Analysis segment of this report. This plot is shown in *Figure 23*.

For the value of the upper limit of sway bar sizes simulated, 1.75" in outer diameter, the stiffness ratio was 6.145. This means that the front end of the car would be susceptible to absorbing slightly over six times the dynamic load than that of the rear, a value that would make the car handle overly tightly for the vast majority of drivers. This kind of tightness would make the car nearly undrivable through a corner.

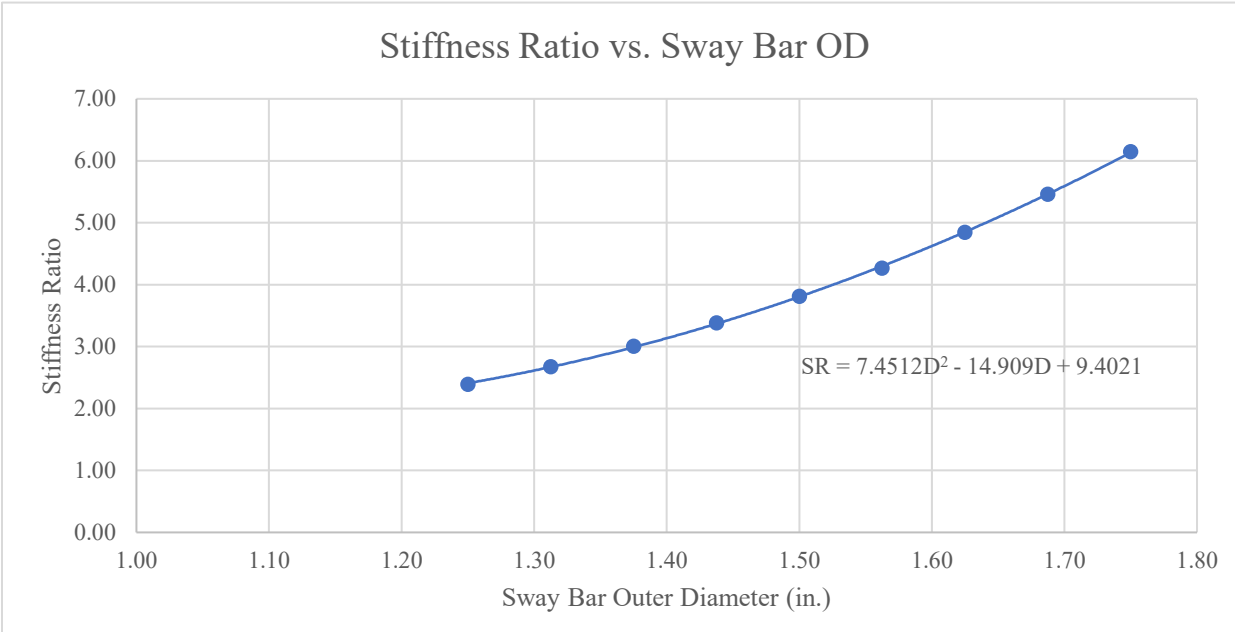

Figure 23: Stiffness Ratio vs. Sway Bar Outer Diameter

Figure 23 demonstrates a clear positive correlation between a racecar's stiffness ratio and the outer diameter of the installed sway bar; the greater the outer diameter of the sway bar, the greater the stiffness ratio. Once each of the data points were plotted, the developed curve fit that best summarized the trend of the data was somewhat surprising. The correlation for the stiffness ratio as a function of the sway bar diameter was not a linear change, rather best approximated by a second-order polynomial equation. The equation that was generated by Microsoft Excel to characterize this polynomial trend is given by *Equation 3*.

$$SR = 7.4512D^2 - 14.909D + 9.4021$$

Equation 3: Stiffness Ratio as a Function of Sway Bar Outer Diameter

If the value for the upper diameter limit was increased in the simulations, the second-order polynomial solution suggests that eventually, the stiffness ratio would reach a point of nearly infinite increase with little increase in the actual sway bar diameter. This is because at very large diameters, the sway bar would no longer be able to twist or absorb any load. Because the sway bar is a torsional spring, this can be analogous to the behavior of a linear spring. If a linear spring becomes fully compressed, it will behave as an incompressible block that absorbs load as a solid piece of metal as opposed to a spring.

Based on this characteristic, the best sway bar outer diameter for the known stiffness ratio of 2.5 happens to be the lower limit that was simulated. When the sway bar outer diameter is

1.25", the stiffness ratio comes out to be 2.39, which is about 4.5% lower than the approximately ideal ratio of 2.5 that was known going into the testing procedure. This makes the 1.25" outer diameter sway bar the clear choice to install into the racecar. In order to achieve a stiffness ratio of exactly 2.5, the sway bar outer diameter would have to be approximately 0.7274", a size that is not manufactured. This precise size would also likely have too little material to stand up to the test of the G-force loads that the racecar experiences on the track, to further emphasize the understanding of the loads present when designing a realistic suspension setup.

This equation that defines the plot characteristic is the solution to the initial goal of this research project. Instead of guessing which sway bar will result in the best performance and using the resource-negligent "trial-and-error" method of component selection, this characteristic equation can be utilized to make the variable selection. Track conditions play a large role in the way that a racecar handles. Simply put, the car will typically handle more loosely on a hotter day and tighten up on a colder day. The derived algorithm can define the methodology for selecting a sway bar diameter at Kern County Raceway Park with all the other variables held constant. If there is a race weekend at KCRP during a hot summer day, a race team's engineer can immediately know that the car will behave more loosely, so they may automatically opt to try a sway bar that is one increment larger than the one that satisfies the equation for the driver's most comfortable stiffness ratio that was determined during a test session.

This mathematical model can be particularly useful for the RPM Motorsports team when the driver prefers a car with a stiffness ratio of 2.5. However, if is wholly possible that this same driver becomes more comfortable in a looser car after accumulating more driving experience. In this case, the model still holds beneficial because the preferred stiffness ratio can be lowered until a sway bar size fits the equation.

Again, this is the mathematical algorithm that was derived based on the loads that the racecar was subject to at Kern County Raceway Park; this equation will not necessarily be valid at a track of a different size or turn radius. However, the methodology for determining the same type of equation at a different track remains the same. This type of computer and mathematical modeling is also not exclusive to the determination of a sway bar outer diameter. An important aspect of racecar engineering is the ability to apply mathematics to the selection of each of the many variables that make up the suspension.

Competitive automotive racing is trending towards the standardization of the implementation of these kinds of models, with the technology of data acquisition and computer modeling becoming better with each new racing season that comes along. For this reason, it is important to be able to develop these kinds of mathematical and computer models in order to remain competitive and relevant. While this equation is helpful in allowing one team to predict an ideal sway bar outer diameter for one track on the schedule, the future possibilities for these types of algorithms are seemingly infinite when considering the large array of components in a suspension and the number of tracks on which super late models compete.

Acknowledgements

I would like to begin by extending my gratitude towards the RPM Motorsports racing team, specifically Dennis Olsen, P.E. As the race team's engineer, Mr. Olsen provided me with a wealth of knowledge that I was able to utilize for this project and opened my eyes to the intensity of what it really means to apply engineering to the world of competitive automotive racing. I hope to further learn from and collaborate with him in the construction of a winning racecar.

I would also like to thank Dr. Mike Harville, my academic sponsor in this endeavor. Without his help, I would not have had the opportunity to even undertake such a project. Because of his suggestions in the presentation of my findings, I was able to share the complexity and fascinating nature of stock car racing with a group of both fellow engineers and engineering laypeople, all of whom enjoyed the project enough to nominate me for the TCU John V. Roach Honors College Boller Award.

Glossary

- *Coil-Over*
  A spring shock system in which the shocks are inserted inside and connected to the spring, existing as a joint unit as opposed to two separate parts.
- *Damping Ratio*
  A characteristic of the oil used inside of the shock, affecting the rate of the shock's compression and decompression and preventing infinite oscillation of a spring.
- *Pivot*
  A value representative of the track width relative to the wheel rates. In the case of a super late model on a circle track, this value represents how much the left side of the car moves relative to the right side due to the load bearing capacity represented by the wheel rates.
- *Ride Height*
  The height, in inches, at which each corner of the racecar is above the ground when the car is sitting at neutral position (Condition 0).
- *Spring Rate*
  The amount of force required to compress a spring a distance of one inch, typically a linear characteristic of the spring.
- *Stiffness*
  A measure of the amount of dynamic load that can be potentially absorbed by either the front or rear suspension of the racecar that cumulatively accounts for each suspension variable present, expressed in in.-lb_f/degree.
- *Stiffness Ratio*
  A dimensionless ratio of the front stiffness to the rear stiffness, indicative of the state of the setup present in the racecar.
- *Stock Car*
  A racecar designed after the everyday streetcars seen on roadways in the United States.
- *Suspension*
  A network of links and connections, springs and shocks, torsional bars, and other components that connect a car's frame to its tires and have a major impact on the car's handling performance.
- *Sway Bar*
  A hollow, cylindrical rod spanning the front end of a stock car's suspension that connects the right front and left front corners and behaves as a torsional spring.
- *Track Width*
  The value representing the width of the car for both the front and rear ends of the car, spanning the outer edge of the left tire to the outer edge of the right tire.
- *Wheel Rate*
  The load bearing capacity portion of the stiffness ratio, accounting for the tire pressure, shock damping ratio, coil-over spring rate, and, in the case of the front end, torsional load bearing capacity of the sway bar. Each corner of the racecar has its own wheel rate.
- *2nd-Order Spring-Mass-Damper System*
  In the context of a racecar, a mass represented by the car itself that is attached to a spring whose oscillatory motion trends towards zero due to the damping mechanism, in this case the shock.

References

T. D. Gillespie, *Fundamentals of Vehicle Dynamics*. Warrendale, P.A.: Society of Automative Engineers, 1992.

P. V. Valkenburgh, *Racecar Engineering and Mechanics*. New York, NY: Dodd, Mead and Company, 1976.

S. Wise, "History of Stock Car Racing," *Special Collections at Belk Library*, 2019. [Online]. Available:http://www.collections.library.appstate.edu/stockcar/history. [Accessed: 26-Feb-2019].