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ABSTRACT

Vehicle dynamics simulation is one of the newest and most valuable technologies being applied in the racing world today. Professional designers and race teams are investing heavily to test and improve the dynamics of their suspension systems through this new technology. This paper discusses the testing of one of Clemson University's most recent Formula SAE racecars on a seven-poster vehicle dynamics simulator; commonly known as a "shaker rig." Testing of the current dampers using a shock dynamometer was conducted prior to testing and results are included for further support of conclusions. The body of the paper is a discussion of the setup and testing procedures involved with the dynamic simulator. The results obtained from the dynamic simulator tests are then analyzed in conjunction with the shock dynamometer results. Conclusions are formed from test results and methods for future improvements to be applied in Formula SAE racing are suggested.

INTRODUCTION

Formula SAE is an international competition where colleges and universities from around the world test their design skills and mechanical abilities as they compete against each other in Detroit, MI with racecars they have designed and fabricated.

Clemson University has been exceedingly successful in all its competitions finishing in the top ten in each of these, including Rookie of the Year in 1998 with a finish of 6th, which was repeated the following year.

A well-developed suspension system is very important to be successful in the Formula SAE competition. Clemson therefore decided to test their dampers on a shock dynamometer in order to determine the quality of the current dampers while also placing their recent racecar on a seven-poster shaker rig. This would corroborate the qualitative data received through driver feedback and lap times during testing with quantitative data from the simulation, while also confirming previous assumptions for suspension setup.

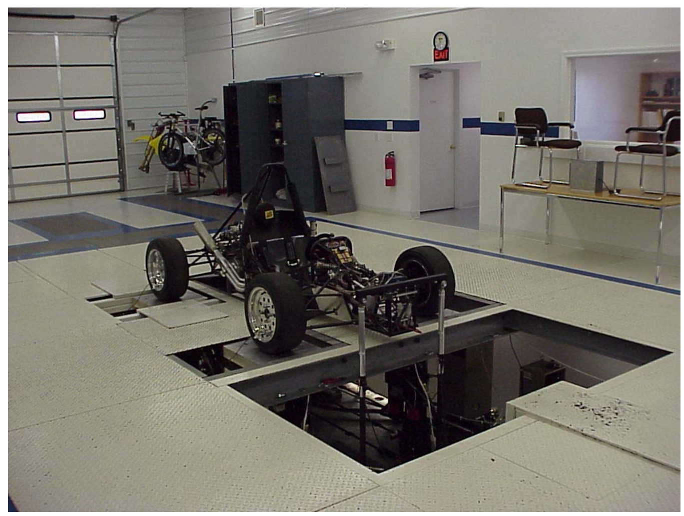

Figure 1. Clemson Formula SAE racecar setup on the seven-poster dynamic simulator.

EQUIPMENT

The seven-poster shaker rig can be used to test a broad range of vehicles from Formula One to NASCAR and even a Formula SAE racecar. The shaker rig simulates track surfaces and disturbances at each tire contact patch along with three downforce inputs simulating aerodynamic forces and inertial g-forces through hydraulic actuators.

The rig is operated on a seismic mass to isolate the instrumentation from outside vibrations and provide more accurate results. It is powered by a hydraulic pump and a 100-hp, water-cooled electric motor, all of which are housed in a separate room [2].

APPLICATIONS

One of the most advanced uses of the shaker rig is simulation of actual track surfaces captured from testing sessions on the racetrack. This allows teams to focus on troublesome areas of the track where large gains can be accomplished. Other teams such as Clemson use sine sweeps, impulses, white noise inputs, and simple road maneuvers for a more generic, yet still highly sophisticated test.

Another use of the shaker rig is determination of chassis stiffness with solid struts that can be applied to areas such as the entire chassis, suspension linkages, or mounting points. Chassis stiffness measurements can also be taken with the dampers in place, which can be used to determine how much friction is in the suspension through a total hysteresis.

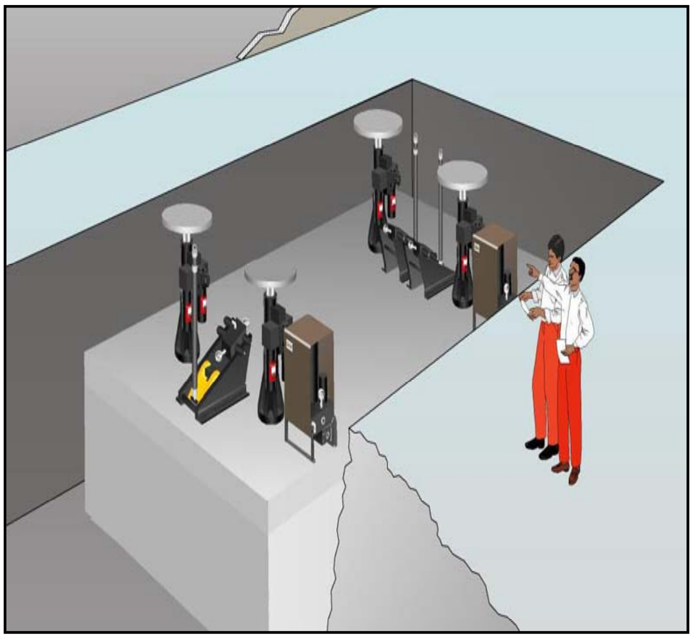

Figure 2. MTS schematic drawing of their 7-poster dynamic simulator.

Through data acquired from accelerometers mounted to the wheels and chassis of the car, three highly accurate load cells installed within each of the four tire pads, seven hydraulic actuators with fluid accumulators and control systems [2], performance parameters such as tire load variation for different chassis and suspension setups can be verified or optimized. This was Clemson's main focus for their first test session.

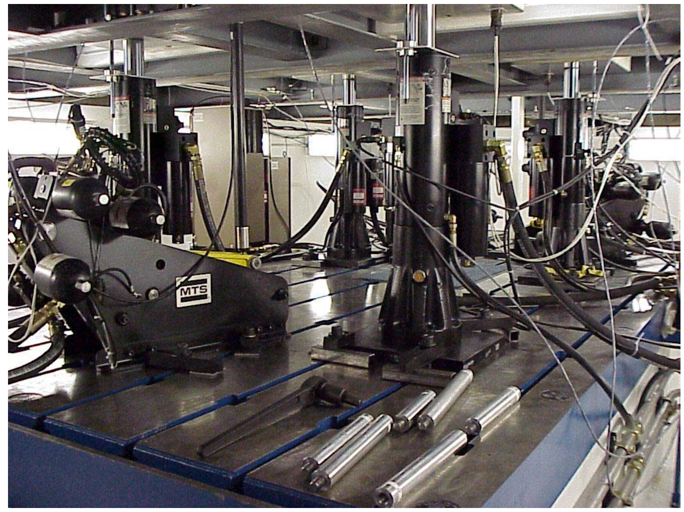

Figure 3. A view of the hydraulic actuators underneath the shaker rig, which create all motion and applied forces.

METHODS FOR SETUP

In order to create the aerodynamic forces and other movements in the racecar, the hydraulic actuators were rigidly anchored to the chassis via three brackets, one for each actuator.

After several different possible approaches, the front brackets were mounted to the top of the front chassis tube on the ends of a piece of square tubing. The square tubing raised the height to 15.5" above the tire contact patch and each side extended only 15" from the centerline of the chassis, minimizing deflection. Welding to the chassis was avoided through the use of an aluminum sandwich bracket and the mount was extremely rigid.

TESTING

There are many different testing procedures that may be performed using the shaker rig for analysis of specific dynamic characteristics. It was decided that for this first test session, an attempt would be made to validate and optimize the anti-roll bar and damper setup that would result in the best overall performance.

The following are the initial procedures taken before the extensive Roll Matrix and Shock Matrix testing was conducted:

1. Placed racecar on wheel pads and hooked up accelerometers to each wheel and various points on the chassis

2. Found unsprung weight by disconnecting pushrods and dampers from bellcranks

3. Maintained anti-roll bar positions at soft in the front and medium stiffness in the rear from

track testing with the dampers set at zero compression

4. Confirmed wheel weights distribution while adding 170 lbs driver weight to car

5. Several tests at 4ips and 10ips vertical with the car in a roll motion and accelerating at 0.2g and cornering at 1.4g were conducted to provide preliminary data.

6. During testing the tires are not anchored to the tire pads and the tires tend to slowly move in one lateral direction after extensive testing. With the movement, the bottoms of the tires tend to rotate inward over time, causing unwanted lateral forces to develop. Öhlins uses plastic sheets filled with a minimal amount of air that are placed underneath the tires to reduce the lateral forces on them and these sheets were used from this point forward

7. Next several sine wave sweeps at 6ips at no applied load were conducted

During these initial tests, it was discovered that there is a tremendous amount of friction within the suspension system. Most of this friction was discovered to be within the rod ends that are on the end of the push rods, but other areas are within the bellcrank and A-arm joints. It should be noted that in future designs a strong effort will be made to reduce as much of this friction as possible.

ROLL MATRIX

After the preliminary tests the racecar was run through a 3x3 roll matrix on the shaker rig that involved changing combinations of the front and rear anti-roll bars through their three different roll stiffness settings (See **APPENDIX A**). These tests were conducted through a roll drive file designed by Ohlins engineers where the racecar slowly rolls to the right and then back to its original position through application of the front hydraulic actuators. These tests yielded the positioning of the front and rear anti-roll bars that would result in the best handling performance for the racecar.

SHOCK MATRIX

With the anti-roll bars positioned in the settings found most suitable during the roll matrix, another 3x3 matrix was performed on the shaker rig that involved varying the compression stiffness of the dampers.

These tests ran at 10ips while the car moved through a left turn roll motion with 0.2g acceleration and 1.4g cornering. Clicking a single adjustment knob on the dampers made adjustments. Adjustments were from zero compression to medium to full compression, however it is highly unlikely that the valves on the dampers are proportional to the number of clicks made by the adjuster, leading to more focus on the zero compression and full compression results from testing.

These tests that were applied during the roll and shock matrices are ones that were developed by Öhlins through much research and development. They were chosen in agreement with Öhlins engineers because they would be most suitable for our first testing session purposes. The tests would be accurate and yield valuable results because the testing conditions were suitable for a Formula SAE racecar. Minimal aerodynamic downforce was involved, the accelerations experienced during testing were typical for a Formula SAE racecar, and the motions provided during the test runs would be enough to excite the entire racecar.

RESULTS

The results obtained from the shock dynamometer were helpful in providing conclusions concerning our current dampers. There was reasonable doubt before testing that all the dampers were close to the same. This is due to the fact that they were designed for mountain bikes and therefore overworked during racing.

Although, at the conclusion of testing, it was discovered that all of the dampers were within reason the same. As the damper curves display (see **Appendix B**), the peak velocities of the dampers are all nearly the same. The displacement curves also tend to follow the same curve path, further supporting the conclusion of very similar dampers on all four corners. Therefore, they might not be the best quality, but they are all at least very close to the same. As a result, the dampers would not skew the shaker rig results due to them being different across the board.

The results obtained from the roll matrix and shock matrix tends to reflect the qualitative information received from driver feedback and lap times during testing where the car performance was at its highest level. This involved the front anti-roll bar at its soft position and the rear anti-roll bar at the medium stiffness position.

Öhlins uses an analysis software program that provides colorful graphical representations of the front versus rear setup combinations. In all graphs the blue and cooler colors represent lower values and are more desirable in all testing situations, while the red and warmer colors represent the higher values. The main element analyzed for our testing was the grip disturbance, which is the mean of the vertical loads applied to each wheel pan.

ROLL MATRIX DATA

<u>Grip Disturbance</u>

The data for the overall grip disturbance showed its lowest value with the front anti-roll bar set at its soft position and the rear anti-roll bar set at its medium stiffness position. As the graph displays, any setting above the soft position on the front anti-roll bar causes grip disturbance to increase. However, the change is modest, which means it is possible that more performance gains could be achieved by making the front anti-roll bar even softer.

Roll Moment Phase Balance

The roll moment phase balance which is the load build-up on the car before it settles, also yielded its lowest values with the front anti-roll bar at its soft position and the rear anti-roll bar at medium to full stiffness.

Body Control

Using the accelerometers mounted onto the chassis of the racecar, very accurate body control measurements could be made. Body heave, pitch, and roll were all measured using the accelerometers mounted to the chassis of the racecar. The body heave is the vertical movement of the racecar experienced by simulated ride height changes. The pitch is body movement experienced by simulated longitudinal accelerations. The last body control tested roll, is the movement of the body rotating left or right across the centerline of the chassis. Overall, the body control characteristics were at their lowest levels with the rear anti-roll bar at its medium position or its stiffest position. The front anti-roll bar provides the racecar with the best body control at its stiffest position. This is not surprising since the motion of the body will be resisted more at the stiffness is increased, resulting in lower values. Instead of changing the anti-roll bars and other suspension settings which would compromise grip, it is recommended that an effort be made to lower the CG on cars in the future to increase body control because grip is more important for this application where body movements are relatively small.

SHOCK MATRIX DATA

Pitch Maneuver

Grip Disturbance

The overall grip disturbance, as with the anti-roll bar matrix was lowest with the front dampers at the zero compression setting and the rear dampers at soft to medium dampening.

Roll Moment Phase Balance

The roll moment phase balance was also lowest with the front and rear dampers at zero compression or at most medium compression. As the graphs from the shock matrix indicate, changing the front dampers has little effect on several of the dynamic characteristics tested, but changes to the rear dampers above medium compression show a considerable increase in grip disturbance and other areas analyzed. This is evident by the straight horizontal lines in the graphs.

Corner Maneuver

During cornering, grip disturbance levels were calculated and the data shows that there is less grip disturbance with the dampers set at zero compression in the front and medium dampening in the rear. In conjunction with the pitch results, the rear grip disturbance is not affected by changes made to the front dampers, but is highly affected by changes to the rear dampers to anything above medium dampening.

Body Control

In all aspects of body control (heave, pitch, roll) the lowest values were obtained with the front dampers at zero compression and the rear dampers at medium to full compression. Along with other maneuvers discussed, any compression on the front dampers, particularly in body heave and body roll, results in unwanted conditions. As mentioned with the roll matrix, lowering the CG of the racecar in future designs could reduce these values instead of compromising grip.

CONCLUSIONS

The seven-poster dynamic simulator is a device that can quickly provide large improvements in the setup and suspension components of a Formula SAE racecar. Nearly three testing sessions on the track were used to determine the most suitable anti-roll bar setup, whereas it only took one day on the shaker rig to provide those same results, along with more information and analysis. Through tests on the shaker rig and analysis of the output, our qualitative data from track testing was confirmed, as the performance was best with the anti-roll bar at its soft stiffness position in the front and its medium stiffness in the rear. It was also discovered that the front dampers should be set at zero compression and the rear dampers should be somewhere between soft to medium compression (10 clicks on the adjustment knob) in order to provide the best grip and handling, a fact not known to team members until after testing. For future designs, it is also suggested that another front anti-roll bar be made that can be adjusted to even softer settings where more performance might be found. As previously mentioned, a strong attempt will also be made to reduce friction and lower the racecar's CG in order to provide better suspension handling and more body control.

FURTHER WORK

In addition to the work completed with this paper, Clemson FSAE hopes to continue researching various applications of the seven-poster dynamic simulator in Formula SAE racing. Some other areas of interest are testing different suspension setups with camber angle, toe in, and caster while also investigating further the dynamics of the tire under different test conditions.

Further analysis of future suspension components will also be completed with a more detailed study the new set of dampers that will be used on the next Clemson University Formula SAE racecar. This will be accomplished through extensive shock dynamometer testing and analysis.

ACKNOWLEDGMENTS

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

CG: Center of Gravity

IPS: Inches Per Second

ARB: Anti-Roll Bar

APPENDIX A Anti-Roll Bar Calculations

Table 1: Anti-Roll Bar Characteristics		
	Front	**Rear**
Outer Diameter (in)	0.5	0.5
Wall Thickness (in)	0.03	0.04
Length (in)	8.00	8.00
Weight (lbs)	0.09	0.12

The anti-roll bar stiffness values were determined from the general equation for torsional stiffness and the equation for the moment of inertia of the cross section:

$$k_\phi = \frac{JG}{L} \quad (1)$$

$$J_{tube} = \frac{\pi}{32}(d_o^4 - d_i^4) \quad (2)$$

where,
k_ϕ =The torsional rigidity of the anti-roll bar in Nm/rad
J =The moment of inertia of the cross section in m^4
G =The modulus of rigidity in Pa (G [4130]=79e9 Pa)
L **=**The length of the bar in meters

This equation, the moment of inertia equations, and the values for the material modulus of rigidity were all taken from [Juvinall 2000]. Through dividing the torsional rigidity of the bar by the anti-roll bar arm length, the stiffness of each roll bar setting can be determined:

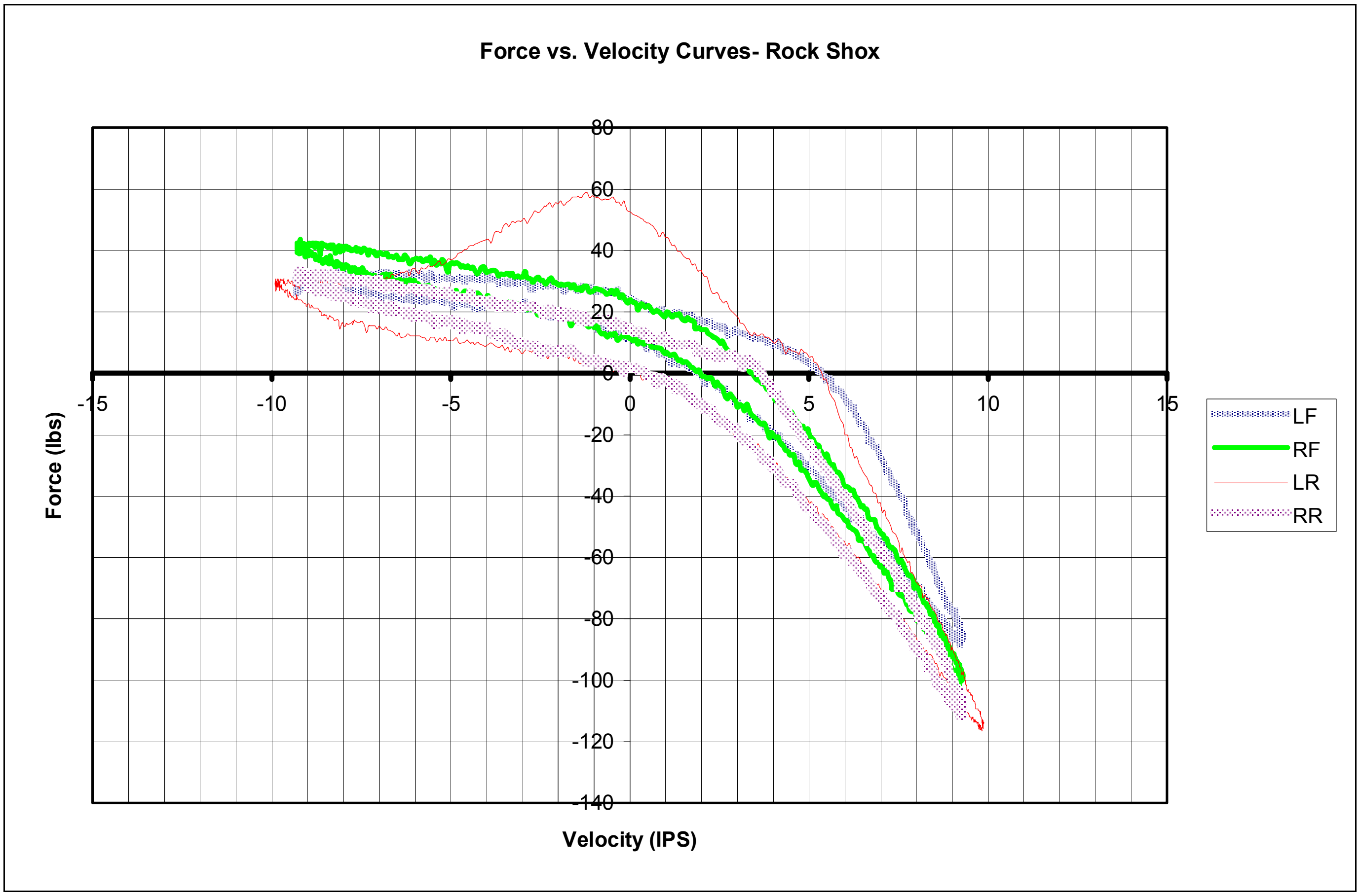
$$k_i = \frac{k_\phi}{a_i} \quad (2)$$

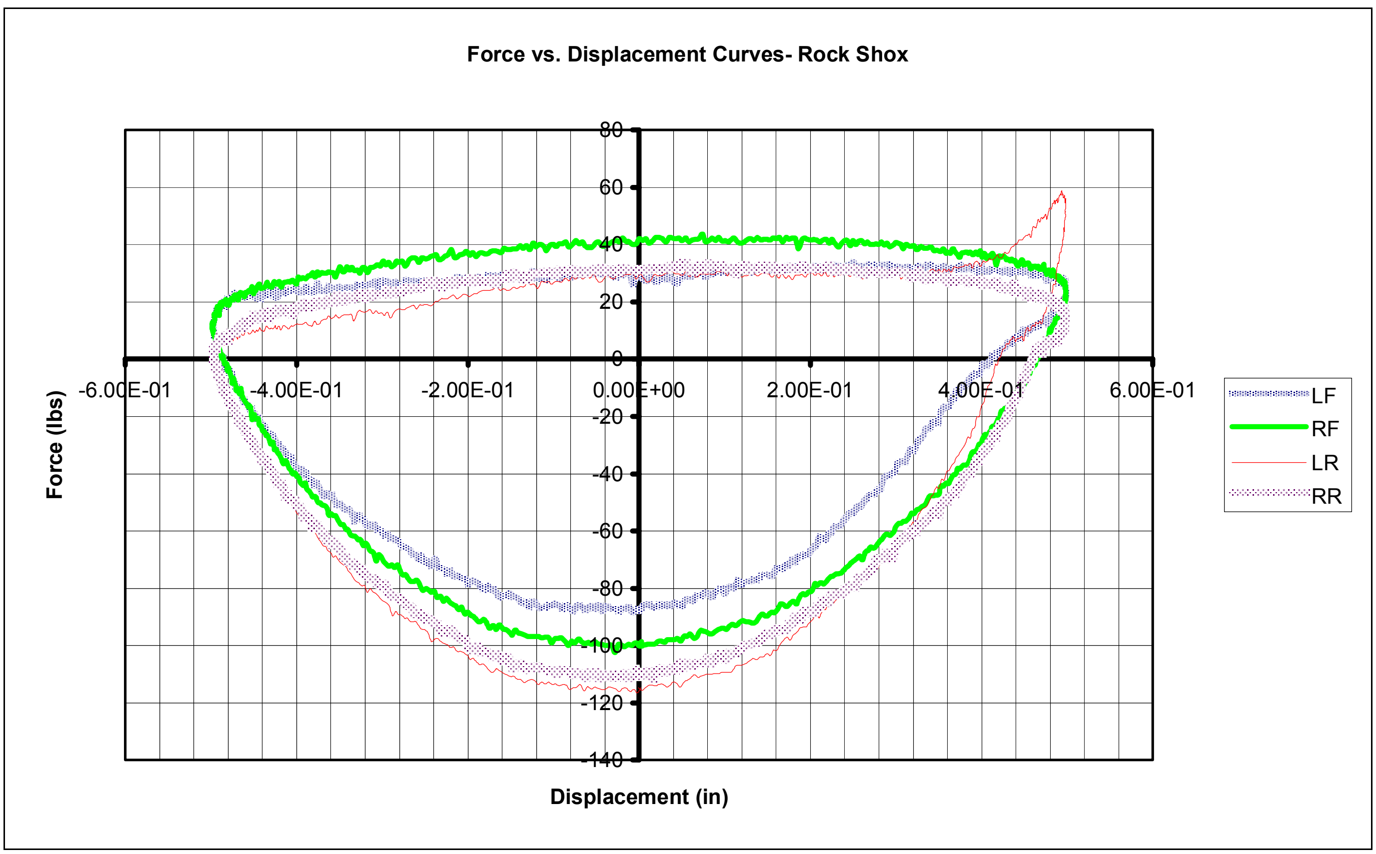
where,
k_i =The stiffness of the roll bar in N/rad
a_i =The length of the anti-roll bar arm in meters
i **=**The roll bar setting number (1, 2, 3)

Equations (1) and (2) were used in an iterative process to calculate the anti-roll bar stiffness values using the outer diameter, the wall thickness, and the anti-roll bar arm length as the design variables. The resulting anti-roll bar stiffness values are shown in **Table 2**.

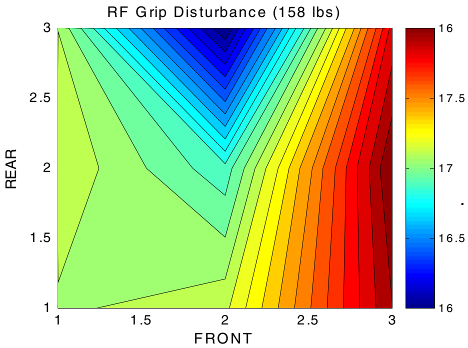
Table 2: Roll Bar Stiffness Values	
	K_θ (lbs/deg)
Arm Length (in)	**Front**
1.5	38.81
2.5	23.29
3.5	16.63
Arm Length (in)	**Rear**
1	69.74
2	34.87
3	23.25

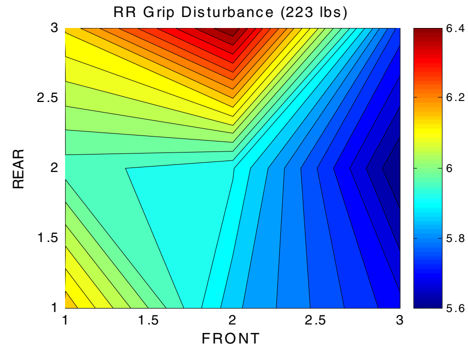
APPENDIX B Shock Dynamometer Results

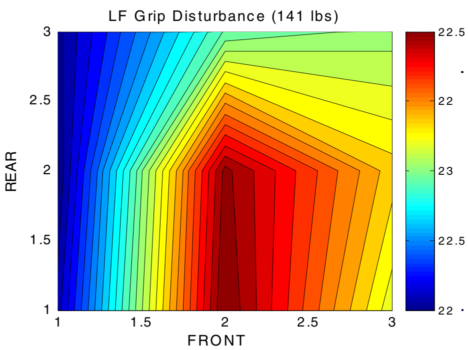


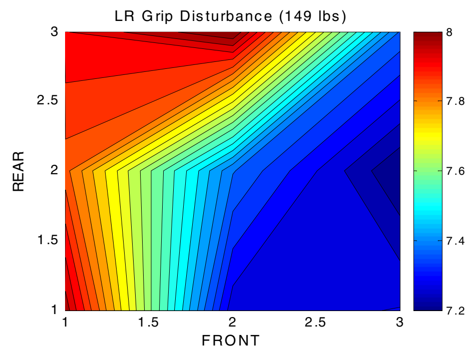


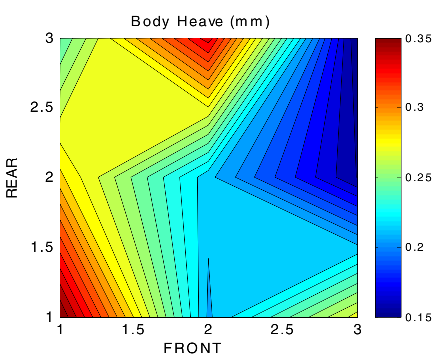
APPENDIX C Roll Matrix Results

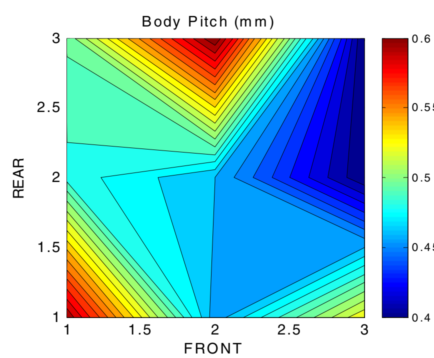


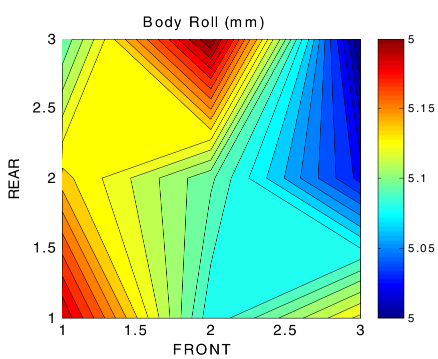




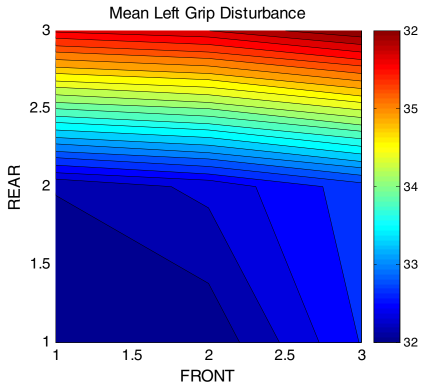


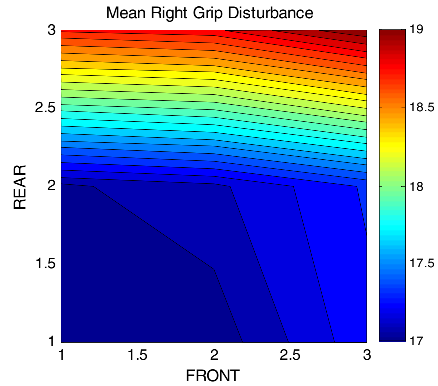


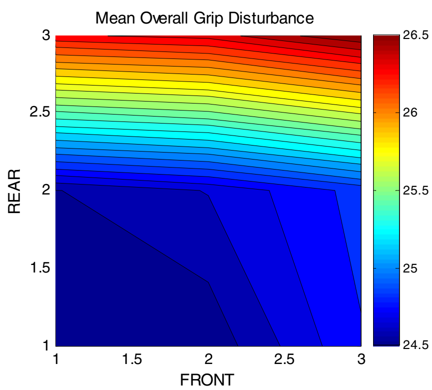


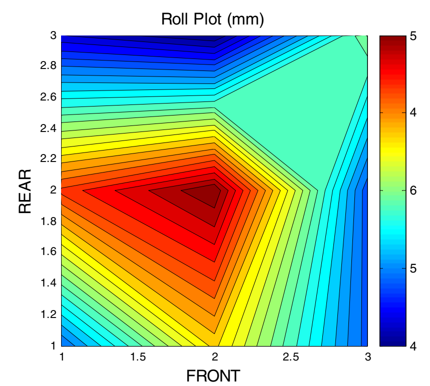


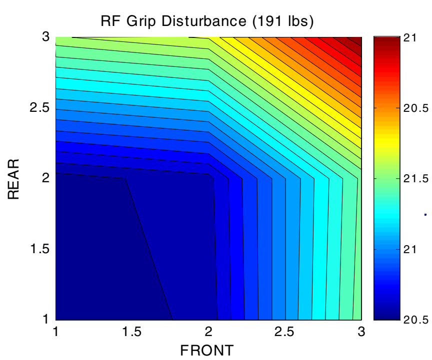
APPENDIX D Shock Matrix Results

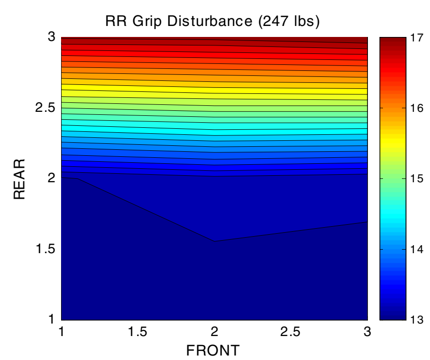


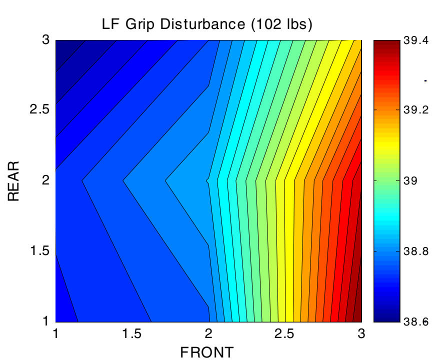


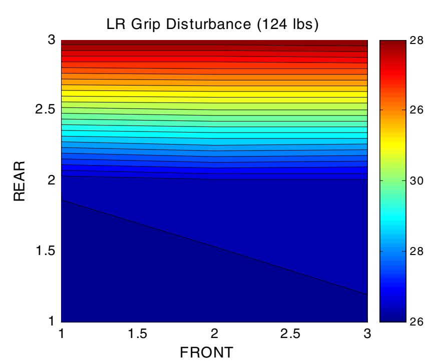


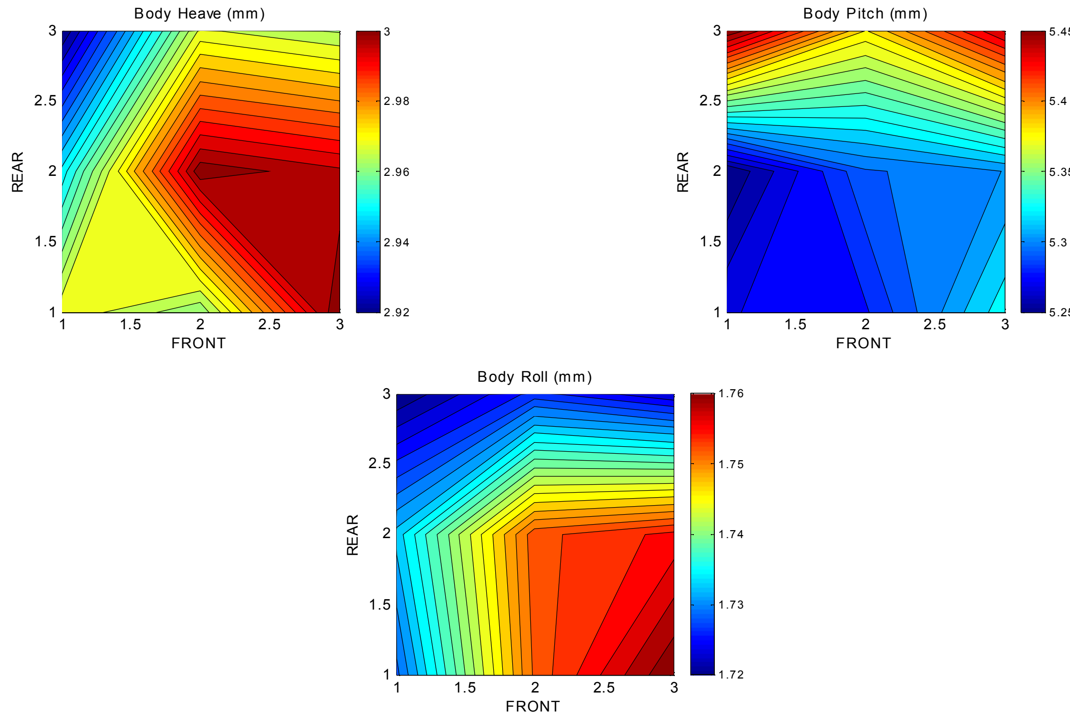











Body Heave (mm)
REAR
FRONT
1
1.5
2
2.5
3
2.92
2.94
2.96
2.98
3
Body Pitch (mm)
REAR
FRONT
5.25
5.3
5.35
5.4
5.45
Body Roll (mm)
REAR
FRONT
1.72
1.73
1.74
1.75
1.76