The Mechanical Design of a Suspension Parameter Identification and Evaluation Rig (SPIdER) for Wheeled Military Vehicles

THESIS

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By

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Abstract

This thesis discusses the design of selected components for a Suspension Parameter Identification and Evaluation Rig (SPIdER) for wheeled military vehicles specified by the Tank Automotive Research, Development and Engineering Center (TARDEC) of the U.S. Army. Suspension evaluation rigs have been used to effectively and efficiently determine suspension parameters. These parameters can then be used to accurately simulate vehicle behavior during driving maneuvers and aiding in suspension design and evaluation. Several components have been designed and analyzed according to the specifications given by the TARDEC of the U.S. Army and are described in detail in this thesis. First, is a discussion and analysis of two measurement systems used to measure wheel center motion with the intent of replacing a previous design using linear transducers and potentiometers. After analysis, it was decided to use one MicroSribe digitizing arm per wheel. This decision was based on the ability to obtain accurate wheel center motion measurements. Secondly, the detailed design of the custom wheel pads was discussed. The wheel pads allow motion in the x and y axes during kinematic testing and can be adapted apply road plane forces during compliance testing. They incorporate custom scales used to determine the wheel force and the center of the tire contact patch. Stress and deflections of various wheel pad components were calculated for maximum
loads occurring during kinematic and compliance testing with a maximum load of vehicles with a maximum weight of 100,000 lbs (45,359 kg) per axle (50,000 lbs (22,680 kg) per wheel). Finally, the roll constraint for the bounce and roll frame is discussed. An overview of the design is presented along with calculations of the contact stresses caused during compliance testing.
Dedication

To my parents, Bruce and Jennifer Wagner,

Their love, care and support made it possible to complete my undergraduate and graduate degrees.
Acknowledgments

I would like to thank Dr. Dennis A. Guenther and Dr. Gary J. Heydinger for giving me the opportunity to work with SEA, Ltd and their guidance and efforts in the preparation of this thesis. I would also like to thank Dr. Dale Andreatta for his assistance in design, Dr. Anmol Sidhu for his assistance in evaluating the measurement system, Dave Coover and Jim Nowjack for their assistance in creating and checking the design drawings and many others at SEA, Ltd. that made it possible to complete my research. Finally, I would like to thank Dr. John Merrill and the many faculty and staff of the Engineering Education and Innovation Center for giving me the pleasurable opportunity to teach and funding my graduate studies.
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Studies in Design and Manufacturing
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Chapter 1

Introduction

1.1 Motivation: Why Design?

Accurate computer simulation for design and analysis of vehicle and tire behavior requires suspension parameters obtained by vehicle suspension testing. Suspension parameters are very useful for suspension and steering wheel development, tire design, and vehicle dynamic simulations, as the suspension determines the position of the wheels on the road. Kinematic (movement of the wheel due to suspension displacement) and compliance (movement of the wheel due to road plane tire forces) characteristics must be known to accurately predict the movement and location of the vehicle chassis and wheels. Early on, the development of vehicle suspension and steering systems have been conducted mostly by subjective techniques during road and proving ground testing, but the trend is to use computer simulation to improve the effectiveness and reduce the time of the design and development of a new vehicle [1]. Suspension parameters have also proven useful for tire design as successful and efficient tire design and development requires an understanding of the combined tire and wheel system. Simulation tools have been used to predict wheel and tire behavior and these tools require suspension parameter data which can be obtained by suspension measurement devices [2]. Suspension
parameter data is also valuable for vehicle dynamic simulations for vehicle dynamics during rollover and driving maneuvers. The National Highway Traffic Safety Administration (NHTSA) has many applications for vehicle dynamics simulations for the offices of Crash Avoidance Research, Rulemaking, The National Advanced Driving Simulator, and Intelligent Vehicle Highway Systems (IVHS) studies [3]. Overall, computer simulations are widely used to predict vehicle response in real world scenarios for everything from design and research to accident reconstruction and rulemaking [4]. This thesis is specifically concerned with the need the U.S. Army Tank Automotive Research, Development and Engineering Center (TARDEC) of the Tank-automotive and Armaments Command (TACOM) at the Detroit Arsenal in Warren, Michigan has for a Suspension Parameter Identification and Evaluation Rig (SPIdER) designed by SEA, Ltd. By measuring suspension parameters and using them in simulation and design, TARDEC can improve the safety of military vehicles and trailers as they recognize that the system which causes the greatest dynamic structural behavior is the suspension system [5].

1.2 Constraints, Requirements and Objectives

TARDEC wants to develop accurate vehicle models and predict vehicle performance and behavior in the battlefield, which would increase soldier safety and reduce costs. The SPIdER would also allow other military vehicle programs to obtain suspension parameter data for use to improve vehicle stability, handling, and ride and to perform durability and failure analyses by computer simulation [5]. TARDEC currently has a Vehicle Inertia Parameter Evaluation Rig (VIPER), designed by SEA, which
provides them with the necessary inertial properties for the vehicle models, but do not have as suspension evaluation capabilities to provide the necessary suspension parameters. TARDEC wants to have the capability to test a wide range of vehicles and trailers including, but not limited to: HMMWV, MRAP, FMTV, PLS, MTVR, M915 Tractor, Stryker, M101, M129, and M1073 along with any future vehicles that meet the vehicle testing requirements [5]. The SPIdER should measure the following parameters and abide by the following constraints shown in Tables 1 and 2.

Table 1.1. SPIdER Measurements

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>•</td>
<td>Suspension spring force vs. deflection curve</td>
</tr>
<tr>
<td>•</td>
<td>Tire force vs. tire deflection curve</td>
</tr>
<tr>
<td>•</td>
<td>Suspension roll center</td>
</tr>
<tr>
<td>•</td>
<td>Suspension roll moment vs. roll angle characteristics</td>
</tr>
<tr>
<td>•</td>
<td>Roll steer</td>
</tr>
<tr>
<td>•</td>
<td>Auxiliary roll stiffness</td>
</tr>
<tr>
<td>•</td>
<td>Bounce Steer and Camber</td>
</tr>
<tr>
<td>•</td>
<td>Roll Camber</td>
</tr>
<tr>
<td>•</td>
<td>Steering Ratio</td>
</tr>
</tbody>
</table>

Table 1.2. SPIdER Vehicle Testing Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Static Weight per Axle:</td>
<td>50,000 lbs (22,680 kg)</td>
</tr>
<tr>
<td>Track Width (Min - Max):</td>
<td>55-110 in (1397-279 mm)</td>
</tr>
<tr>
<td>Vertical Suspension Travel:</td>
<td>±12 in (304.8 mm)</td>
</tr>
<tr>
<td>Maximum Roll Angle:</td>
<td>±5°</td>
</tr>
<tr>
<td>Maximum Wheel Width (Dual Wheels):</td>
<td>36 in (914 mm)</td>
</tr>
<tr>
<td>Longitudinal Wheel Travel:</td>
<td>±6 in (152 mm)</td>
</tr>
</tbody>
</table>

Other requirements requested by TARDEC are shown in Table 1.3.
Table 1.3. Other SPIdER Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allow for characterization of one axle at a time for multi-axle vehicles</td>
</tr>
<tr>
<td>Portable by crane and able to anchor to the current t-bed at TARDEC</td>
</tr>
<tr>
<td>All sensors must have less than 1% error</td>
</tr>
<tr>
<td>All measurements must have errors less than 3%</td>
</tr>
<tr>
<td>Designed to be easily modified for compliance testing</td>
</tr>
</tbody>
</table>

These constraints and requirements are used in the mechanical design of the SPIdER. With high loads it is important to carefully look at the structural integrity of each load bearing component to avoid the yielding of material and meet all requirements. The measurement accuracy and precision is also of great importance as suspension parameters are used for simulations that help improve vehicle safety and performance. This creates a need to examine the wheel measurement device’s accuracy and precision and attachment to the wheel, along with the deflection of load bearing members which can cause error in measurements.

1.3 Past Machines

The advent of the radial ply tire in the late 1940’s created a requirement for more precise control of suspension kinematics. Citroen, along side of Michelin, started to create machines to test vehicle suspensions. In the early 1960’s the development of a suspension evaluation rig was started at Cranfield Institute of Technology under Professor J R Ellis [1]. Then, in the early 1970’s one of the most well known test machines called the Chevrolet Vehicle Handling Facility (VHF), designed and built by General Motors (GM), became operational [1]. Since the 1970’s there have been several machines designed and built for various applications for light and heavy vehicles.
In 1987 the Suspension Parameter Measurement Device (SPMD) was designed by the Vehicle Research and Test Center (VRTC) of the NHTSA for light vehicle kinematic and compliance testing [6, 7, 8]. Soon after in 1990, SEA, Ltd. designed a modified version of the VRTC SPMD which was also for light vehicles called the SPMD II [3, 9, 10]. The VRTC also built a test rig called the Composite Parameter Measurement Device (CPMD) for light vehicles. This was specifically for their Vehicle Dynamics Analysis, Non-Linear (VDANL) simulations and sought to build a new suspension evaluation rig to replace their SPMD because it required the suspension to be assembled to a temporary frame called a buck and took too much time to complete tests. The VRTC compared a design proposal by MTS to the SEA SPMD II to see if either design would meet their needs and it was determined that both would [3]. MTS Systems Corp. later built their proposed suspension evaluation rig called the Kinematic and Compliance (K&C) Deflection Measurement System in the mid 1990’s and has since built another machine called the Dynamic K&C Measurement System [11, 12]. In 1996 MIRA, UK had what they call the MIRA K&C Rig designed by Anthony Best Dynamics Limited (ABD) for light vehicle testing. This test rig was designed based on the success of the SPMD and SPMD II previously mentioned [1]. A new method of suspension measurement was designed by SEA Inc. in the late 1990’s by using a side-pull test. Using the side-pull test eliminates the need to combine kinematic and compliance testing, which can become questionable for systems where there is a coupling between bounce and roll and allows for testing with larger roll angles [4]. Although all of the discussed suspension evaluation rigs were designed to test passenger vehicles, there also has been work done
regarding the testing and design of suspension testing machines for heavier vehicles, namely heavy trucks and trailers.

Chris Winkler has done a considerable amount of work with heavy vehicle suspensions at the University of Michigan with the Highway Safety Research Institute (HSRI) and the University of Michigan Transportation Research Institute (UMTRI). In the early 1970’s Winkler had a need for vehicle parameters and had to determine them experimentally. He developed a simple method used to determine a limited number of suspension parameters [13]. In the late 1970’s The HRSI has since designed and built a test facility for measuring all of the compliance, kinematic and coulomb friction properties of commercial vehicle suspensions for vehicle simulation and continues to be in use at UMTRI [14, 15, 16]. In the early 2000’s the Michelin Americas Research and Development Corporation have their own operational test rig called the Michelin Heavy Truck K&C Rig that is capable of measuring suspension parameters of up to three axles at a time [2].

1.4 Thesis Overview

Chapter two contains a background on suspension parameter testing methods, information on past testing machines and an overview of the basic function and operation of the SPIdER. Chapter three details the selection and evaluation of two wheel measurement devices. Chapter four covers the evolution and design of the wheel pad assembly along with the bounce and roll frame constraining method. Chapter five concludes the thesis with future work and possible design changes.
Chapter 2

Wheeled Military Vehicle SPIdER Overview

2.1 Past Design Methods

In general, suspension parameters are found through a series of tests that determine specific parameters by isolating different components. In order to find the spring stiffness of a suspension, it must be isolated from the damping effects of the suspension and because damping is a function of velocity and spring stiffness is a function of displacement, testing is not done dynamically. Therefore, testing for the suspension spring stiffness, roll stiffness and other parameters are done quasi-statically. During quasi-static testing, data is measured discretely between incremental displacements to the suspension. The process involves displacing the suspension a discrete amount, stopping to take the required vehicle and suspension force and displacement measurements. Separate dynamic testing must be done to determine parameters regarding the dampening effects of the suspension. Common tests conducted on suspension evaluation rigs are shown in Table 2.1. The only machines referenced that conduct dynamics test are the MTS Dynamic K&C Compliance system and SEA Inc.’s Side-Pull. MTS conducts both quasi-static and dynamic kinematic and compliance...
testing along with a variety of other dynamic testing including simulation of road and body force conditions [12].

Table 2.1. Common Suspension Tests

| • Bounce (kinematic) |
| • Roll (kinematic) |
| • Longitudinal Compliance |
| • Lateral Compliance |
| • Aligning Moment Compliance |
| • Steering Ratio |

Kinematic and compliance testing is generally conducted in the same manner from machine to machine. During the suspension tests the lateral, longitudinal, and vertical motion of the wheel center is measured along with steer angle change (rotation about the wheel’s z-axis), camber angle change (rotation about the wheel’s x-axis) and caster angle change (rotation at the about the wheel’s y-axis). The vehicle coordinate axis defined by the Society of Automotive Engineering (SAE) is shown below in Figure 2.1. While caster and camber angles are shown in Figure 2.2 and Figure 2.3.

Most suspension evaluation rigs support the vehicle wheels on platforms or “wheel pads” which act as the road plane and were traditionally designed on air bearings [1]. The air bearings allow the wheels to “float” and freely move laterally, longitudinally and in steer during bounce and roll tests and actuators could be attached to apply road plane forces and moments during compliance testing. The GM VHF used air bearings in the design of their tire pads and attached actuators to perform compliance testing [17].
Figure 2.1. SAE Vehicle Axis [18]

Figure 2.2. Positive Caster Angle for a Right Front Wheel [19]
When VRTC designed their SPMD they used a combination of linear slides for frictionless motion in the x and y direction and used an air bearing to allow for frictionless rotation of the wheel pad [7]. The Michelin Heavy Truck K&C uses linear bearings for x and y motion also, but replaces the air bearing with a rotary bearing [2]. Not only is it important for the wheel pads to allow frictionless motion laterally, longitudinally and in yaw, but the wheel force must be known and scale systems are designed into the wheel pads to measure the wheel force. Since the location of the center of pressure of the tire is generally desired, the scale systems are built with multiple load cells. Knowing the location and forces on each load cell, the center of the contact patch of the tire is easily found. For compliance testing, the most realistic and convenient method of applying road plane forces to simulate braking and cornering forces is to attach
actuators to move the wheel pads in the longitudinal direction for braking forces, laterally for cornering forces and rotationally for steering compliance [9]. Although it is not convenient to move the vehicle chassis relative to the road plane for compliance testing, it can be convenient to do so for kinematic testing.

During kinematic tests the wheels must be moved in jounce, rebound, and roll relative to the vehicle chassis and is generally done in one of two ways, either by moving the vehicle frame or test buck (temporary frame fabricated and fitted with the vehicle suspension) or clamping the vehicle frame, fixing it to the ground, and moving the wheel pads. For kinematic testing, the most realistic method is to move the vehicle chassis [9]. If the wheel pads are moved while holding the chassis fixed, the wheel movement must be related to the chassis’ bounce and roll by some suitable method [9]. Bounce tests are fairly straightforward where either the wheel pads or the chassis is moved incrementally in jounce and rebound, taking measurements between each movement of the suspension. Roll testing is not as straightforward because of the roll center of each individual suspension and several methods have been used to conduct roll tests.

Each suspension type has what is called a roll center as determined by the geometry of suspension linkages and can be found by inspection of the suspension geometry or by the wheel center motion during a roll test. The roll center is defined as the point in a vertical plane through the wheel at which lateral forces can be applied to the sprung mass (vehicle body) without producing any suspension roll [18]. The suspension roll center determines the location of the roll axis of the vehicle, which is the instantaneous axis about which the unsprung mass (wheels) rotates with respect to the
sprung mass when a pure moment is applied to the sprung mass. In a two axle vehicle, the roll axis is defined by connecting the front roll center and the rear roll center with a straight line [18]. The roll axis is considered instantaneous because as a vehicle bounces or rolls, the suspension geometry changes, changing the location of the roll axis. As with bounce testing, roll testing can be conducted moving either the wheel pads or the vehicle chassis. Along with having the choice of moving the wheels or the chassis, there is also a choice as to where the roll axis of the roll device will be located. There is no restriction to the location of the roll axis as long as the wheels are free to move laterally. There is no lateral movement of the wheel when the vehicle is allowed to roll about its instantaneous roll axis [9].

Most suspension parameter devices already allow for the free movement of the wheel pad and are therefore able to choose the roll axis at any location. The benefit of rolling the vehicle about its roll axis is the ability to apply a pure roll moment to the vehicle, otherwise the roll moment applied must be “corrected” to determine the true roll moment [14]. Michelin’s Heavy Truck K&C Rig is an example of designing a roll apparatus allowing the vehicle to roll about its own roll axis. Michelin rigidly attaches a beam to the chassis of the truck without rigidly constraining the vertical motion of the beam relative to the ground and applies the roll moment through a system of cables and pulleys [2]. Other machines such as the VRTC’s SPMD, SEA’s SPMD II, and MIRA’s K&C Rig designed a roll apparatus which place the roll axis of the machine at a convenient location for design purposes [1, 7, 9]. The heavy vehicle suspension measurement facility at UMTRI was designed to move the wheel pads vertically during
both bounce and roll testing and the roll axis was placed on the ground plane (the plane on which the tires sit on the wheel pads) [14]. It should be noted that some suspension measurement devices are capable of conducting pitch tests for their kinematic testing, but pitch effects on the suspension are relatively minor compared to bounce or roll effects [9]. The MIRA K&C rig is an example of a machine which was designed with the intention of performing pitch testing but does not list any results obtained from pitch testing nor does it list pitch testing under the range of testing [1, 20]. As mentioned before, during kinematic and compliance testing, the wheel center motion must be measured and is done by attaching transducers to the wheel and through post processing, the movement of the wheel center can be determined.

The most common method of measuring wheel motion has been the use of string potentiometers and other linear potentiometers and encoders. The GM VHF, VRTC SPMD, SEA SPMD, MIRA K&C Rig, and UMTRI Heavy-Vehicle Suspension Measurement Facility all used several linear potentiometric transducers to measure wheel motion. In order to determine all wheel motions with linear transducers, several have to be used to resolve the wheel camber, steer, and longitudinal and lateral displacements. The MTS K&C System and the Michelin Heavy Truck K&C Rig use a single transducer for all wheel motion with MTS using an optical non-contact system and Michelin designed their own Wheel Plane Transducer Assembly (WPTA) which is a robotic arm consisting of five precision resolvers for angular motion and one precision linear potentiometer [2, 11]. Using a single transducer for all wheel motion reduces the set-up time and number of ports necessary for the DAQ system.
2.2 General Wheeled Military Vehicle SPIdER Design

What makes the SPIdER unique is its need to accommodate a wide range of vehicles. The lightest military vehicles and trailers could be tested on some of the passenger vehicle suspension evaluation rigs while the heaviest military vehicles could not be tested on UMTRI’s Heavy-Vehicle Suspension Measurement Facility. Not only does the SPIdER have to support heavier loads than most rigs, but it has to accommodate a wide range of track widths, wheel bases, number of axles, chassis designs and armor plating. Also, because TARDEC originally had a contract to build their suspension evaluation rig with another company, an attempt was made to use the components and parts already purchased and assembled by the previous company.

Early on it was decided that the vehicle body was to be fixed, moving the wheel pads to produce the necessary bounce and roll motion. This decision was made because it is easier to accommodate a wide range of vehicles as it is easier to design vehicle constraints to fit all of the vehicles rather than designing a bounce and roll apparatus to accommodate all vehicles. Wheel pads were designed using linear motion guides and a thrust type bearing to allow for independent lateral and longitudinal motion and ease of adaptation for compliance testing. The wheel pad scales purchased by the previous company could not be used as they were designed with dozens of load cells making it unreasonable to calculate the center of pressure of the tire contact patch and instead, custom scales were designed. A decision had to be made as to how the wheel pads would be actuated and the hydraulic cylinders and components purchased by the previous company were chosen to move the wheel pads in bounce and roll.
The initial concept was to create two separate axle assemblies movable by crane allowing TARDEC to place them on their T-bed and move them, when necessary, to accommodate a wide range of vehicle wheel bases and configurations. Each single-axle motion assembly consists of two vertical pylons to support the hydraulic cylinders oriented vertically supporting the roll and bounce frame which houses the wheel pads as seen in Figure 2.4.

![Figure 2.4. Single-Axle Motion Assembly](image)

The hydraulic cylinders have pin connections at both ends allowing the roll and bounce frame to rotate with respect to the main frame. For bounce motion, the cylinders simply...
move the same distance in the same direction, moving the roll and bounce frame in jounce and rebound. As for roll, the cylinders move the same distance but in the opposite direction. The roll and bounce frame is constrained, unable to move laterally or longitudinally, allowing pure bounce and roll motion with the roll axis located at the road plane.

The wheel displacements were originally planned to be measured with a motion tracking system, Polhemus: Liberty, purchased by the previous company. After evaluation it was clear it would not provide accurate measurements. Another measurement device had to be chosen. SEA wanted to investigate the possibility of using a digitizing arm by MicroScribe instead of using their traditional “Trident” system. The MicroScribe was evaluated for feasibility of use and accuracy. After evaluation, it was chosen to measure longitudinal, lateral, vertical, steer, and camber displacements. It was chosen primarily because it reduces the number of transducers that would have been needed for the Trident system which consists of string pots and other linear transducers. The initial design concept with wheel measurement devices and test vehicle is shown below in Figure 2.5. The wheel measurement evaluations, wheel pad design and some of the roll and bounce frame design are described in detail in the following chapters.
Figure 2.5. Single-Axle Motion Assembly with Wheel Measurement
Chapter 3

Wheel Measurement Evaluations

3.1 Polhemus Liberty Evaluation

The previous company had planned on using a Polhemus Liberty motion tracking system. The Polhemus Liberty system is an electromagnetic tracker consisting of a transmitter, or source, and several sensors. The Polhemus system provides the position and orientation of each sensor relative to the source’s coordinate axis orientation. One Polhemus system can measure all required wheel motions without using any other transducers. The drawback to using the Polhemus Liberty system is that using an electromagnetic source, any metal near the system will cause distortion in the results producing error in the recorded positions. Placing the Polhemus system in a position where metal does not cause distortion would be almost impossible as the SPIdER apparatus and frame is manufactured mostly of steel and the vehicles have metal wheels and armor plating. An evaluation of the system’s accuracy was initially conducted to determine if using the system for the SPIdER was plausible.

A simple test was conducted to test the repeatability of the measurements. Two sensors were fastened to a block of wood, keeping the distance between the two sensors
constant and placed on a plastic container placing the sensors two and half feet above the ground floor in the middle of an open room in an effort to minimize the possibility of distortion caused by metal. The source was placed on the same plastic container. The positions of the two receivers were recorded with the piece of wood at various locations and orientations, but always kept in the same quadrant of the source’s coordinate axis. Since the distance between the sensors was held constant, it should remain constant. The positions of the sensors for a sample of ten positions were recorded and the distances between the two sensors were calculated and shown in Figure 3.1.

![Figure 3.1. Calculated Distance Between the Two Sensors for Each Position](image)
With the ten data points it can be seen that there is a larger variance between the distances than expected. The average distance between the two sensors is 6.288 inches (159.7 mm) with a standard deviation of 0.067 inches (1.7 mm). There was not a precise method of measuring the distance between the sensors as the origin is within its body and its location is difficult to measure. Therefore, the error at each orientation of the sensors could not be determined accurately. Before an effective method to determine the accuracy of the system, the repeatability was checked, and if acceptable, a reliable method for determining the error would be pursued. To get a better look at the repeatability, every possible combination of two distances were compared and their differences are shown below in Figure 3.2. The largest difference in distance between the two sensors is 0.2182 inches (5.542 mm) while the average difference in distance is 0.0798 inches (2.027 mm). Of the 45 combinations, 13 of them had a difference greater than 0.1 inches (2.54 mm). Since the suspensions will see displacements of 1 inch or smaller, an error of 0.1 inches (2.54 mm) would results in errors of 10% or greater which is not acceptable and does not meet TARDEC’s requirement of 3% accuracy for all calculated measurements. No more data points were taken nor were other tests conducted as the large variance in the distance between the two sensors for the ten positions provided enough information to declare the Pohlemus Liberty system unacceptable and pursue other measurement systems. The second measurement device tested was a 3D digitizing arm by MicroScribe which was ultimately chosen to measure wheel motion.
3.2 MicroScribe Evaluation, Testing, and Measurement

SEA owns an older MicroScribe model and it was tested to determine if it would be capable of obtaining all necessary wheel measurements. The MicroScribe is a five or six (depending on model) degree of freedom (DOF) articulated digitizing arm commonly used for metrology, reverse engineering, and inspection. Physically, the MicroScribe consists of a base, shoulder, two arms, and a stylus as seen in Figure 3.3. The five DOF models allow rotation at the base, shoulder, elbow, and two wrist joints while the six DOF models add rotation of the stylus. The MicroScribe resolves the location of the
stylus tip along with the stylus’ orientation relative to the MicroScribe’s stationary origin by using the angles of each joint. The MicroScribe outputs the x, y, and z coordinates of the stylus tip along with the x, y, and z vector components of the stylus, which is treated as a unit vector.

3.2.1 Data Collection for the MicroScribe

In order to evaluate the MicroScribe a windows application had to be created to record and collect the data. The application continuously displays the angles at each joint, x, y and z coordinates of the stylus tip, and vector components of the stylus. The user can choose to either continuously record data points by selecting “record” followed by clicking on “Connect” or collect a single data point by clicking on “Capture Values” after clicking “Connect” with “record” unselected. The values captured are saved as a text file which can be easily imported to Matlab or Excel. A screen shot of the windows application is shown in Figure 3.4. With the positions of the stylus tip and orientation of
the stylus exported the translation of the rotation of the stylus tip can be calculated and used in the determination of wheel center motion.

![Figure 3.4. Windows Application for MicorScribe Data Collection](image)

3.2.2 Initial Testing

It was important to see whether the data recorded by the MicroScribe produced accurate results and confirm that the stylus vectors recorded were indeed components of a unit vector representing the stylus and could be used to measure steer and camber. Translation of the stylus tip in the x, y, and z directions is simple to calculate as it is the difference between the recorded x, y, and z data points at each instance. The position measurements of the stylus tip are expected to be accurate as the MicroScribe specifies them to be accurate to < 0.003 inches (0.08 mm). A simple test was conducted to quickly confirm the data collected and translation calculations. A 12 inch (305 mm) steel scale
with engraved markings was taped to a table and recorded data while the stylus was
initially placed on an engraved inch marking and moved the stylus an inch (25.4 mm) to
the next marking. As expected, the distance between the initial position and the final
position was calculated to be 1.00 inches (25.4 mm), to the nearest 0.00 inches.

Although there was confidence in the position measurements, initially it was not
known if the accuracy of the stylus vectors and calculation of the angles would be as
accurate since the stylus vector components are not used in common MicroScribe
applications for which it was designed, but the assumption was they are accurate as the
angles of each joint needed to be accurate to determine the position of the stylus tip. By
moving the stylus in different orientations and observing the stylus vectors displayed in
the Windows application, it was confirmed that the vectors are the components of the unit
vector which corresponded to the stylus orientation in the default coordinate system,
which is established based upon the original orientation of the MicroScribe in the “home”
position. The vector components of the stylus can be easily used to calculate the angles
of the stylus with respect to the home coordinate system using sine and cosine functions.
The angle measurements were validated using a sine bar with one degree increments
from zero degrees to 8 degrees. The stylus was placed on the flat surface of the sine bar
and a data point was taken and the angle was calculated and compared to the angle of the
sine bar. The largest error in the measurements was less than 0.05° which confirmed the
assumption that the vector components of the stylus are accurate.
3.2.3 Using the MicroScribe for Wheel Measurements

After determining the MicroScribe is capable of measuring wheel motion accurately, the method and required calculations had to be determined. It is not possible to attach the stylus to the wheel center because the wheel center is located at the intersection of the wheel’s spin axis (the axis the wheel “rolls” about) and the wheel plane (a plane perpendicular to the spin axis that intersects the middle of the tire) [18]. It is most convenient to attach the stylus to the wheel’s rim or lug studs and depending on the type of wheel the stylus may be mounted up to 20 inches (508 mm) away from the wheel center. With the stylus tip located a fixed distance away from the wheel center, as the wheel moves in steer and camber, the longitudinal, lateral and vertical displacements of the wheel center will not be accurately measured. Since the wheel center is located a fixed distance away from the stylus tip, the increase, or decrease, in angle of steer and camber will cause the MicroScribe to measure displacements smaller than the true wheel center displacements. The steer and camber rotation of the stylus is accounted for using rotation transformation matrices.

3.2.3.1 Using Rotation Matrices

In three dimensional coordinate systems there are three rotational matrices. They are shown with variables from the SAE coordinate system.
Rotation about the z axis:
\[
\begin{bmatrix}
\cos(r) & \sin(r) & 0 \\
-\sin(r) & \cos(r) & 0 \\
0 & 0 & 1 \\
\end{bmatrix}
\] (3.1)

Rotation about the y axis:
\[
\begin{bmatrix}
\cos(q) & 0 & -\sin(q) \\
0 & 1 & 0 \\
\sin(q) & 0 & \cos(q) \\
\end{bmatrix}
\] (3.2)

Rotation about the x axis:
\[
\begin{bmatrix}
1 & 0 & 0 \\
0 & \cos(p) & -\sin(p) \\
0 & \sin(p) & \cos(p) \\
\end{bmatrix}
\] (3.3)

Rotational transformation matrices allow coordinates of a point to be translated between coordinate systems of different axes orientation but the same origin. This is useful for determining the wheel center location.

For determining wheel center motion, the coordinate system used for the wheels follows SAE’s convention, but with the origin located at the wheel center. It is important to note that the coordinate system does not rotate or translate with the wheel during suspension testing. The mounting of the stylus was designed so that its axis and the y axis are collinear. By doing this, steer is defined as rotation about the z axis, and camber as rotation about the x axis. As long as the wheel does not steer or camber, the axis of the stylus will remain parallel with the wheel’s initial axis of rotation and the change in
position of the stylus tip will be equal to the change in position of the wheel center.

Rotational transformation matrices become useful for determining wheel center
displacements when the wheel moves in steer and camber.

When the wheel steers or cambers, the stylus rotates in the same way. When this
happens, the axis of the stylus is no longer parallel with the y axis and the change in
position of the stylus tip is no longer equal to the change in position of the wheel center.
This happens because the wheel center position is relative to the stylus tip and stylus axis
and coordinates of the wheel center relative to the stylus are no longer equal to the
coordinates of the wheel center relative to the origin. In order to calculate wheel
displacements, the transformation matrices for rotations about the z and x axes are
needed. The transformation matrix for rotation about the y axis is not needed because the
wheel center lies on the stylus’ axis. Therefore, when the caster angle changes during
testing, the position of the wheel center relative to the stylus tip will remain unchanged.
The transformation matrices for rotations about the z and x axes are used to determine the
wheel center displacements relative to the origin.
\[
\begin{bmatrix}
W_{C_x} \\
W_{C_x} \\
W_{C_y}
\end{bmatrix} =
\begin{bmatrix}
\cos (r) & \sin (r) & 0 \\
-\sin (r) & \cos (r) & 0 \\
0 & 0 & 1
\end{bmatrix}
\begin{bmatrix}
1 \\
0 \\
0
\end{bmatrix}
\begin{bmatrix}
\cos (p) \\
\sin (p) \\
-\sin (p) \\
\cos (p)
\end{bmatrix}
\begin{bmatrix}
x_0 \\
y_0 \\
z_0
\end{bmatrix} +
\begin{bmatrix}
x \\
y \\
z
\end{bmatrix}
\]

(3.4)

Where:

- \(W_{C_x}\) = wheel center position
- \(W_{C_y}\) = wheel center position
- \(W_{C_z}\) = wheel center position
- \(x\) = stylus tip x coordinate
- \(y\) = stylus tip y coordinate
- \(z\) = stylus tip z coordinate
- \(x_0\) = initial distance between stylus tip and wheel center in x direction
- \(y_0\) = initial distance between stylus tip and wheel center in y direction
- \(z_0\) = initial distance between stylus tip and wheel center in z direction
- \(r\) = steer of the wheel recorded by the MicroScribe
- \(p\) = camber of the wheel recorded by the MicroScribe

The initial distance between the stylus tip and the wheel center will be measured during set-up by the operator. The initial distance in the x and z directions is recorded as 0.0 inches and the distance in the y direction can be as large as 20 inches (508 mm) depending on the vehicle. Using the measurements from the set-up, data recorded by the MicroScribe and Equation 3.4, the wheel center motion for each wheel can be calculated during testing.

3.2.3.2 Error Caused by Misalignment in Stylus Mounting

The MicroScribe meets TARDEC’s requirement that all measuring devices must have less than 1% error. TARDEC also requires all calculated values to have less than 3% error. As a result, the error in wheel center displacement calculations needed to be investigated. There are two factors of the stylus mounting that can cause error in the
wheel center displacement calculations: the stylus axis will not be perfectly collinear with the wheel axis and there will be error in the measured distance between the stylus tip and the wheel center. The stylus is mounted rigidly to a fixture which is attached to the wheel using the existing lug studs.

Theoretically, each lug stud lies on a circle concentric with the wheel center and each lug stud is of equal length. Error exists with the location and lengths of the lug studs. As a result, the slots in the fixture used to mount to the lug studs were designed to account for any imperfections in the lug studs. The unequal length of the studs will cause the fixture to be mounted at an angle relative the wheel plane. Therefore, the stylus could be mounted so that its axis is not collinear or parallel with the wheel’s spin axis.

This affects calculations as the stylus tip no longer lies on the y axis because the error causes the distance vector between the stylus tip and the wheel center to have an x and z component. This affects calculations as the recorded values of $x_0 = 0$ and $z_0 = 0$ are not accurate and produces error in $WC_x$, $WC_y$, and $WC_z$. Caster angle change creates more error in $WC_x$, $WC_y$, and $WC_z$. When the wheel rotates, the coordinates of the wheel center relative to the stylus are not constant. With error in $x_0$ and $z_0$ being relatively small and the assumed values zero, it was assumed that caster angle change would not create significant error and was not included in equation 3.4.

The location of the stylus tip relative to the wheel center in the y direction is recorded manually by the operator during test setup. The operator will measure the distance between the stylus tip and the front of the tire which is then added to half of the tire’s width and recorded. As the wheel changes in camber and steer, the error in the
recorded distance between the wheel center and stylus tip will create error in the wheel center displacement.

The largest source of error in wheel center displacement is expected to come from the difference in lengths of the lug studs causing the axis of the stylus to be mounted at an angle relative to the wheel’s spin axis because of large distance between the stylus tip and the wheel center. In order to investigate the affects of the errors on the wheel center displacement, MATLAB code was written to calculate the maximum error caused by the errors in stylus mounting and measurement.

Exact errors of the lug studs are not known because there are many wheel designs using various lug studs and lug stud patterns making the determination of the error in the lug stud lengths and error in the location tedious. The error in the measured distance between the wheel center and the stylus tip would also be difficult to determine as it is produced mainly by human error. Therefore, maximum errors are conservatively estimated and used to examine the worst case scenarios. The slots in the stylus fixture were designed with a tolerance of 0.01 inches (0.3 mm), allowing a maximum error of 0.01 inches (0.3mm) in either direction along the x and z axis. The lug stud lengths are assumed to be different by as much as 0.0625 inches (1.59 mm) and the average circular lug stud pattern is assumed to have a diameter of 20 inches (508 mm) based on observations of several military vehicles. A span of 20 inches (508 mm) and a difference in length of 0.0625 inches (1.59 mm) creates an angle of 0.2 degrees but a conservative value of 0.5 degrees was assumed to add some leeway for the 0.0625 inches (1.59 mm) in height and 20 inches (508 mm) in diameter.
These values are used to determine the expected maximum error during bounce and roll testing when the maximum steer, camber, and caster angles are 5 degrees. The total error is a combination of several components and because of this it is possible the error of one component can cancel with another. Eight cases were examined, each a possible combination of errors in the x, y, and z components of the distance between the wheel center and stylus tip. For instance, Case 1 determines the error of the wheel center measurement if only error in the mounting angle is present. Case 8 determines the error of the wheel center measurement if error is present in the x, y, and z components along with error in the mounting angle. The MATLAB code determines the maximum error in the x, y and z wheel center motion of the eight cases. In addition, the maximum error in the eight cases was found with every possible combination of positive and negative error.

The maximum errors in the x, y and z components of wheel center displacement are 0.041 inches (1.0 mm), 0.036 inches (0.90 mm), and 0.039 inches (1.00 mm) respectively. In order to achieve the maximum caster, steer, and camber angle of 5 degrees during bounce and roll tests the suspension most likely must travel at least 6 inches (152 mm) in the vertical direction. An error of 0.039 inches (1.00 mm) for a nominal value of 6 inches (152 mm) results in an error of 0.7% which is well below TARDEC’s maximum allowed error of 3%. The travel in x and y directions during bounce and roll tests will be significantly less than 6 inches (152 mm) and could approach and surpass the 3% error. However, since it is possible to have almost zero longitudinal and lateral wheel center displacement, it is unreasonable to expect them to be
measured with less than 3% error. The MATLAB code and a table of all maximum error
values are in Appendix A.
Chapter 4

Wheel Pad and Limited Bounce and Roll Frame Design

SEA has used wheel pads from Hunter Engineering Company in the past and considered purchasing custom made wheel pads from them. Custom wheel pads designed by Hunter would be reasonably priced, compact, and able to withstand the maximum loads. However, it would be difficult to make adaptations to allow compliance testing because of the design. The Hunter designed wheel pads are compact because they are primarily two plates separated by ball bearings. The ball bearings give the top plate three degrees of freedom, allowing the top plate to float, much like air bearings but without the need of a constant air supply. The disadvantage of this design is that it is difficult to constraint the x and y motion independently while still allowing steer. During compliance testing it is important that when forces are applied in the x direction, the motion in the y direction is constrained, and vice versa, while allowing the wheel to steer freely. Therefore, custom wheel pads were designed using linear rails and bearings. Using linear rails and bearings meant the height of the wheel pads had to be sacrificed, but was necessary to allow for future compliance testing. This chapter details the design of the wheel pad assembly.

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The wheel pads are placed within a bounce and roll frame which controls the bounce and roll of the wheel pads through the use of hydraulic cylinders. It is essential that the bounce and roll frame is able to create a pure roll motion about a pivot point on the road plane. Select aspects of the design of the bounce and roll frame are also presented in this chapter.

4.1 Conceptual Wheel Pad Design

The conceptual design of the wheel pad was created to present a general idea to TACOM. The design consists of a top plate, steer bearing, scale plate, load cells, scale mounting plate, linear guide rails and blocks for x and y motion, and a part for mounting the x-direction guide rail. A model with bill of materials is shown in Figure 4.1. All parts, except for the x-direction guide rails and blocks, were designed with general dimensions, ignoring maximum stresses and assembly considerations.
The top plate is simple in design as its only function is to be the contact point between the vehicle and the evaluation rig. During compliance testing, friction forces between the top plate and tire move the wheel in the lateral and longitudinal directions. The top plate must accommodate dual wheels with a maximum width of 28 inches (711 mm), but TARDEC wants the plates to be 36 inches (914 mm) to accommodate any future vehicles. The wheel pad must also be long enough to allow for the largest tire contact patch. The contact patch size can be estimated by assuming that the pressure on the contact patch is equal to the tire inflation pressure. If a 28 inch (711 mm) wide tire inflated to 100 psi (689 kPa) is subjected to 50,000 lbs (222 kN) of force (during a maximum 2g compression test), the tire contact patch will be approximately 18 inches (457 mm). The effective width of the contact patch will actually be smaller than 28 inches (711 mm) due to the gap between the single tires that make up a dual wheel and
the tread pattern. This means the length of the contact patch will be greater than 18 inches (457 mm). During testing, caster angle change will cause the wheel to rotate the top plate and must be accounted for in design. With both of these factors considered, the initial top plate was designed to be 24 inches (610 mm) long. The top is placed on the steer bearing, which allows the top plate to rotate, allowing the wheel to steer during testing.

It was known in the beginning that a thrust type bearing was needed to allow the top plate to steer and be able to withstand an axial static load capacity of 50,000 lbs (222 kN) with a small overall width. The steer bearing simply rests on the custom designed scale.

The scales inherited by TARDEC from their previous contract could not be used for two main reasons. The first being the scales were intended to be used while resting on flat solid surface. The inherited scales consisted of basically a bottom plate and top plate with dozens of button type load cells between them. The top and bottom plates are made of sheet metal and because of this; the scaled is required to be placed on a surface with little to no deflection as any deflection of the scale could cause inaccurate results. This would require a thick plate placed below the scale adding the overall height of the wheel pad assembly. The second reason the scale cannot be used is because the scale is not only used to measure the load on each wheel, but also to locate the center of pressure of the contact patch. In order to this, the load and position of each load cell must be known. The inherited scales were designed to output the total weight applied to the scale, not the load of each load cell and placement of the load cells were not know, both of
which make it difficult to adapt the scales to locate the center of pressure. Therefore, it was decided to create a scale of a steel plate and four individual load cells. The load cells were custom made by Delta Metrics Inc. with a load rating of 15,000 lbs (66.7 kN) per load cell.

The load cells of the scale rested on top of an arbitrarily designed plate. The plate was designed knowing it would have to change in the future as it would yield under a load of 50,000 lbs (222 kN). The intent of the design was to show a concern for the overall height of the wheel pad assembly and effort to keeping it compact. The plate is used to connect the scale to the x-direction guide blocks which allow the wheels to freely move longitudinally.

The x-direction guide rails and blocks inherited by TARDEC from their previous contract were initially planned to be re-used. They are manufactured by THK, model number SHS55C. The blocks have a static load rating of 197 N (44,288 lbs) which is more than adequate with four blocks bearing a total of 50,000 lbs (222 kN). The rails were cut to 686 mm (27.01 in) lengths which are long enough to allow the required ±3 inches (76 mm) of travel during testing. It is not possible to attach the rails directly to another set of guide blocks so a part was designed to mount the x-direction guide rails to the y-direction guide blocks.

The mount for the x-direction guide rail was also designed arbitrarily without calculating deflections or stress. It was simply used to show the concept behind attaching the x-direction guide rail to the y-direction guide blocks. For the initial design concept of
the wheel pad assembly, y-direction guide rails and blocks were not selected and possible rails and blocks were shown to convey the design concepts for the wheel pad assemblies.

4.2 First Wheel Pad Design

After the initial concept of the wheel pad was approved by TARDEC, the design of the wheel pad assembly was continued by evaluating the stress and deflection of each individual part. The design of each part started with hand sketches using simple hand calculations to estimate the necessary features and dimension of each part. The part was then modeled in SolidWorks and evaluated using SolidWorks’ built in Finite Element Analysis (FEA) software called SolidWorks Simulation. SolidWorks Simulation was used because of availability and non complex nature of the parts’ designs. The wheel pad parts are complex enough that is difficult to achieve accurate results through hand calculations but simple enough that more robust FEA software such as Ansys or Abaqus are not needed. The parts are designed using steel and keeping stresses under the yield stress and ignoring any thermal loading. Therefore, the simulations were all linear and suitable to be conducted using SolidWorks Simulation. After the simulations were conducted, simple hand calculations were made and compared to the simulations for validation.

The first wheel pad was designed using the method above and is shown below in Figure 4.2, but the details of the design and design process are not given because the wheel pad design was re-designed. However, general details and overall functionality of the first wheel pad design are given.
The first wheel pad design followed closely to the initial concept utilizing similar parts. The largest change was in the method used to rest the scale onto the x guide blocks and scale plate itself. While the scale plate remained largely unchanged due to the simplicity of its function.

The overall dimension of the top plate stayed the same at 36 inches (914 mm) wide and 24 inches (610 mm) long. The thickness was determined to be 1.25 inches (31.8 mm) with tapped holes added to allow the mounting of the steer bearing. Flanges were also added to each corner of the top plate to be used to attach the vehicle drive-on ramps. The steer bearing selected during the conceptual design of the wheel was used.
with a spacer added between the top plate and the bearing. The spacer was placed over the inner race of the bearing, which attaches the top plate, so that the top plate does not contact the outer race of the bearing allowing the top plate to rotate freely. The bottom of the top plate went through several design changes. Initially it was planned to use a large radial bearing that had an axial load rating larger than 50,000 lbs (222 kN) which required a multi-stepped surface to contact the inner race of the bearing on the inside and side surface. The multi-stepped surface sat inside the bearing, allowing the top plate to only contact the inner race and to keep the bearing in place. When the current bearing, which bolts to the top plate, was chosen a simple protrusion on the bottom side was initially designed to prevent the top plate from contacting the outer race, but because of machining costs, it was decided to use a separately designed spacer to limit material removal from the top plate.

The current bearing chosen, model RU297G, is manufactured by THK and has static load rating of 281 kN (63,200 lbs) and a width of 40 mm (1.57 in). The bearing was chosen because of its ability to bolt to the top and scale plate, removing the reducing the amount of machining needed to mount the bearing between the top plate and scale plate.

Initially, a recess with multi-stepped surfaces was designed to lower the overall height of the wheel pad assembly and mount the bearing to the plate. After the bearing model RU297G was selected, a simple recess without multi-stepped surfaced was designed to place the top of the bearing nearly flush with the top surface of the scale plate. This reduced the overall height of the wheel pad while keeping the scale plate
thick enough to withstand stresses and reduce deflection. A large hole was designed in the bottom of the recess for two reasons, to allow access to the mounting holes for the top plate, and to keep the surface of the recess from contacting the inner race of the bearing. Along with the large hole, there are tapped holes on the surface of the recess to mount the steer bearing. Flanges were placed on both the right and left sides of the wheel plate for the load cells to be mounted. The flanges are used to reduce the overall height of the wheel pad by reducing the distance between the bottom of the load cell and the top of the steer bearing. Simple feet for the scale were design which thread into the load cells. The feet are designed as two separate pieces. The threaded portion of the scale feet have a spherical radius bottom that sits on a disk with a spherical radius concave, allowing the top plate to tilt during vehicle drive on and non-centered loads on the top plate. The scale feet sit into recesses on the top of the scale carriers.

The scale carriers were needed because the distance between the scale feet was larger than the maximum center to center distance of the x guide blocks to allow for the required ±3 inches (76 mm) of travel during testing. If possible, simple parts would have been designed to place the scale feet at the center of the x guide block. Since this was not possible, scale carriers were designed to place the load of the scale feet on the center of the x guide block. This is necessary because the guide blocks are not able to withstand the moment of 37,500 in-lbs (4237 N-m) created by 12,500 lbs (55.6 kN) offset by 3 inches (76 mm). While the scale feet were designed to sit in recesses on the top surface of the scale carriers, protrusions were designed on the bottom surface of the carrier to insure the load from the scale is transferred to the center of the guide block. The scale
carrier drawing is shown in Figure 4.3. The scale carrier bolts the x guide blocks which ride on the x guide rails, both of which were inherited from TARDEC’s previous contract. The x guide rails are fastened to the x rail carriers.

![Figure 4.3. Scale Carrier for First Wheel Pad Design](image)

The x rail carriers are designed similar to the scale carrier as the x guide rails sit in a recess in the x rail carriers to decrease the overall height and protrusions are designed to distribute the load on the center of the y guide blocks to keep from producing too great a moment. The recess and the protrusions can be seen in the assembly drawing in Figure 4.2. The x rail carriers are fastened to the y guide blocks allowing the wheel pad to move in the lateral direction.
A different style of guide blocks and sliders were chosen for the y-direction. Instead of having a the guide block with mounting holes on top, a set of rails and guides with mounting holes on the side of the guide block were chosen. By using the HSR style of guide blocks and rails by THK, the distance between the two main beams of the bounce and roll frame could be decreased. The distance between the main beams is reduced because the overall width of the HRS type guides and rails is narrower and the orientation of the guide block and guides allow the guide rails to fasten directly to the main beams.

After calculations, simulations and validation and designs were completed, it was determined the first wheel pad assembly had to be redesigned for two reasons. First of all, in an effort to reduce the overall height of the wheel pad, extensive machining had to be performed and high grade steel used to reduce the thickness at the mating locations of parts while retaining the structural integrity of the part. In some parts, the stress became as high as 60,000 psi (414 MPa) to 80,000 psi (552 MPa), requiring high grade steel. Common, low-carbon 1020 steels have yield stresses ranging from 30,000 psi (207 MPa) to 51,000 psi (352 MPa) depending on whether it is hot-rolled or cold-drawn while in higher carbon steels the yield strength can approach 150,000 psi (1034 MPa) to 200,000 psi (1379 MPa) with a significant increase in cost per pound [22].

Secondly, after consulting with Chris Winkler, TARDEC decided that the maximum longitudinal travel of the wheel pads should be increased to ±6 inches (152 mm). Winkler stated that the ±3 inches (76 mm) of longitudinal travel of UMTRI’s heavy vehicle suspension measurement facility has been marginal and said that if he was
to design it over again, he would allow for at least ±4.5 inches (114 mm) of travel. Winkler suggested that ±6 inches (152 mm) of travel may very well be necessary for heavier military vehicles. The added travel is not necessary to allow for longitudinal wheel displacements during bounce and roll testing, but longitudinal compliance testing. During longitudinal compliance testing wheel displacements are larger particularly for leaf springs because of “wrap-up”. When longitudinal shear forces are applied to a tire with leaf springs the shear force causes the leaf spring to rotate, or “wrap-up,” displacing the wheel longitudinally. An example of this rotation is shown in Figure 4.4.

![Figure 4.4. Leaf Spring Wrap Due to Sheer Forces](image)

The x-direction guide rail and blocks inherited from TARDEC’s previous contract do not allow for ±6 inches (152 mm) of longitudinal motion and can no longer be used.

4.3 Final Wheel Pad Design

The parts that saw the greatest re-design were the scale plate and the scale carriers as they had the largest stresses, tolerances, and features requiring more machining and material removal, all of which increased cost. New x-direction guide rails had to be purchased to accommodate the increased longitudinal travel. The opportunity was taken
to select a new model of guide block and guide rail to reduce the overall height of the wheel pad. The model chosen was SHS30V with a static load capacity of 66.6 kN (15,000 lbs) per guide block. The SHS30V guide blocks are 28 mm (1.10 in) shorter and 80 mm (3.15 in) narrower than the SHS55C guide blocks. The y-direction guide blocks and rails along with the steer bearing remained the same. The x-direction guide rail carriers remained similar in concept and the top plate saw little change. The final wheel pad design is shown in Figure 4.5 and the detail drawings are shown in Figure 4.6 and Figure 4.7.

Figure 4.5. Final Wheel Pad Design
Figure 4.6. Wheel Pad Detail Drawing 1
Figure 4.7. Wheel Pad Detail Drawing 2
4.3.1 Top Plate Design

The design of the top plate remained relatively unchanged from the first wheel pad design. The overall dimensions remained the same at 36 inches (914 mm) wide and 24 inches (610 mm) long and a thickness of 1.25 inches (31.8 mm). The changes from the previous design were the removal of the four flanges, addition of the circular bolt hole pattern, rectangular through hole pattern and the two tapped holes. The flanges were removed because the drive on ramp design changed and now uses the rectangular through holes to attach to the plate. The two tapped holes are used to mount the MicroScribe orientation fixture which is used to configure the MicroScribe axis. The circular bolt hole pattern is used to access the holes in the steer bearing to attach the bearing to the scale plate. This had to be done because the bearing was flipped upside from its orientation from the first design to create a larger area of support for the top plate. From the previous design, and initial FEA testing of the initial top plates, it was expected that 1.25 inch (31.8 mm) thick plate would be adequate and FEA simulations were run to confirm this.

Two cases were tested for the heaviest vehicle, one occurring during bounce testing the other occurring during drive on. It is during maximum compression during testing that the wheel will be seeing the largest load. The largest vehicles will weigh 50,000 lbs (222 kN) per axles, or 25,000 lbs (111 kN) per wheel. Maximum bounce occurs when either two of the following things happen, the suspension reaches the bump stops, or the compressive force per wheel reaches twice the magnitude of the curb weight per wheel. This means that the largest load per wheel pad is 50,000 lbs (222 kN). For
this case, a force of 50,000 lbs (222 kN) will be distributed over the entire surface of the top plate. This will create the largest stress and greatest deflections. A realistic vehicle of that weight is very likely to have dual tires with the maximum width of 36 inches (914 mm), covering the width of the plate. It is unlikely that contact patch will reach from front to back of the top plate, but to be conservative it is assumed to be true. Stress and deflection will decrease as the size of the contact patch decreases because a larger percentage of the pressure will be directly supported by the bearing underneath.

The second case examined is during vehicle drive on. When the vehicle is initially driven onto the wheel pad, the entire weight of the wheel will be supported near the edge of the top plate, with less support coming from the bearing under the plate. Once again, it is assumed that with the heaviest vehicle, 25,000 lbs (111 kN) per wheel, the wheel width will be 36 inches (914 mm), the entire width of the top plate. The tire contact patch is assumed to be 8 inches (203 mm) long, so a 25,000 lb (111 kN) force is applied to an area the width of the top plate starting at the front edge and extending 8 inches (203 mm) off the edge.

In both cases the top plate was constrained by fixing the 16 tapped holes that are used to attach the bearing. A contact set was also created to simulate the bearing support under the plate. This was done by first creating a work plane coplanar with the bottom surface of the top plate and a sketch of circles of diameters equal to the Inner Diameter (ID) and Outer Diameter (OD) of the outer race of the steer bearing. The sketch was used to create a surface where the outer race of the steer bearing will contact the top plate. The contact set was created with the surface and the work plane, treating the work plane
as a rigid wall. This keeps any node or element in the mesh from penetrating the work plane and applies a reactive force on those elements. This simulates support on bottom of the top plate provided by the bearing.

The top plate mesh was created using the h-method adaptive meshing which creates smaller elements in the in higher stress gradients. The p-method adaptive meshing is more desirable as it applies pseudo-mid-side nodes in the elements reducing the need to refine the meshing. The p-method could not be used as SolidWorks does not allow it to be used with contact sets. The h-method is beneficial as it automatically refines the mesh in large stress areas. The final stress and deflection plots for the two cases are shown on the next two pages.
Figure 4.8. Stress in Top Plate During Maximum Bounce

Figure 4.9. Deflection for Top Plate During Maximum Bounce
Figure 4.10. Stress in Top Plate During Drive On

Figure 4.11. Deflection for Top Plate During Drive On
The stress plots for both cases show that there is a maximum stress considerably larger than the yield stress of steel but is very localized. During the bounce test the maximum stress is about 82,000 psi (565 MPa) and the maximum stress during drive-on is about 70,000 psi (483 MPa). The high stresses are caused by stress concentrations and boundary conditions on the edge of the tapped holes and are not of concern. Away from the edges of the tapped holes, the stresses are around 25,000 psi (172 MPa) and 30,000 psi (207 MPa) in both cases, which are reasonable. The deflections are also reasonable because of the support given by the steer bearing. The largest deflection during bounce testing will be approximately 0.042 inches (1.1 mm) on the corners of the plate and 0.047 inches (1.2 mm) on the corners during drive on. Road plane motion in the z direction is measured during testing and used in calculations which reduce the concern regarding top plate deflection.

The FEA simulations were validated by taking a section of the plate and treating it as a cantilever beam. It would have been more desirable to use stress and deflection equations for plates from a text book such as *Formulas for Stress and Strain* by Roark and Young [24]. None of the text books had cases with a fixed support in the center of the plate. All of the cases involved different combinations of simple, fixed, and free edges with various loading conditions. Instead, the plate stress was approximated by a cantilever beam with a width equal to the bolt hole pattern diameter, 356 mm (14 in). The length of the beam is equal to the distance from the edge of the plate to the first tapped hole, 279 mm (11 in). The plate thickness of 31.8 mm (1.25 in) was also used. The pressure on the top plate caused by the 222 kN (50,000 lb) load is 400 kPa (58 psi) and
was used as the load on the cantilever beam. A beam thickness of 356 mm (14 in) results in a distributed load of 142 N/mm (812 lb/in) along the length of the cantilever beam.

The equation for the moment at the fixed end of a cantilever beam is given as:

$$M_{max} = \frac{wl^2}{2} \quad (4.1)$$

And the deflection is given as:

$$\delta = \frac{wL^4}{8EI} \quad (4.2)$$

Where

- $w =$ distributed load (lb/in)
- $L =$ beam length (in)
- $E =$ modulus of elasticity (psi)
- $I =$ moment of inertia ($\text{in}^4$)

Using these equations along with the equation for bending stress, the stress caused by the moment at the fixed end was 26,300 psi (181 MPa) and the deflection was 0.025 inches (0.6 mm). The stress values support the stresses of 25,000 psi (172 MPa) and 30,000 psi (207 MPa) at the tapped holes. The deflection is of the same magnitude as the deflection from the FEA simulation. The cantilever beam deflection was expected to be smaller as the cantilever beam does not take into account the deflection caused by bending in the direction orthogonal to the cantilever beam’s bending.

4.3.2 Bearing Spacer Design

The bearing spacers were the last parts of the wheel pad designed. Their thicknesses are arbitrary to the structural integrity as they will only see compressive
forces. It was desired to have the overall wheel pad height be such that the top plate is 2 inches (51 mm) above the top of the main beams of the bounce and roll frame. The 2 inches (51 mm) of height allows the drive-on ramps to be designed with the ramps flush with the top plate surface. The 2 inches (51 mm) also allow the clevis of the hydraulic cylinders to be mounted so that the pivot point of the clevis lies on the road plane, simulated by the top plate. Without the thickness of the bearing spacers, the top surface of the top is 1-3/8 inches (35 mm) above the main beams of the bounce and roll frame. This leaves 5/8 of an inch (16 mm) for the spacers. The spacers were designed to be 5/16 of an inch (8 mm), totaling 5/8 of an inch (16 mm) to place the top of the top plate 2 inches (51 mm) above the main beams. The OD and ID of both spacers were chosen to contact only the inner and outer race of the steer bearing allowing it to steer freely.

4.3.3 Scale Design

The design of the scale changed significantly from the first wheel pad design to the final design. It was necessary to reduce the machining time, material removal and thicken the scale to reduce the total cost. It was important to limit the height of the scale while increasing the thickness of the scale. With the model SHS30V guide blocks being 80 mm (3.15 in) narrower than the model SHS55C guide blocks, the scale is able to fit between the two sets of guide blocks while maintaining the same distance. The first design mounted the load cells to the bottom of the scale, keeping the scale plate above the guide blocks. In the second design, the load cells are attached to the top of scale, allowing the scale to hang between the guide blocks. With this design, the thickness of
the scale plate no longer has any affect on the overall height of the wheel pad assembly and eliminates any need to machine a recess for the bearing to sit. With the steer bearing flipped upside down and access holes added to the top plate, the hole in the middle of the scale plate was removed. Although the geometry of the scale plate is simple, the load applied and boundary conditions are unique.

The load from the wheel is transferred to the scale plate through the bearing and bearing spacer. For simulation purposes, the load is assumed to be uniformly applied on an annular surface on the top plate with the same dimensions as the bearing spacer. The largest stress and deflection occurs during maximum bounce when the largest possible force is 50,000 lbs (222 kN). The scale is supported on each of the four corners by a load cell. Each load cell is attached to the scale by two bolts and is simulated in SolidWorks with bolted connections. A “no penetration” contact condition was added to the scale and the four load cells which keeps any nodes of the load cell from passing through the scale. The contact also distributes forces to the scale. The holes in the end of the load cells which the scale feet thread into are held fixed, fully constraining the scale. The scale was also meshed with h-method adaptive meshing because contact sets were used and the stress and deflection plots are shown below.
Figure 4.12. Stress in Scale During Maximum Bounce

Figure 4.13. Deflection of Scale During Maximum Bounce
The largest stress appeared in the load cells but was ignored because the load cells designed by and purchased from Delta Metrics are rated up to 15,000 lbs (67 kN) per load cell with a safe overload of up to 150%. By looking at the stress plot it can be seen that there are not any significant stress concentrations in the scale plate. The largest stress values occur at the edges of the bolt holes in the middle of the plate with values as high as 20,000 psi (138 MPa). Stress reaches as high as 15,000 psi (103 MPa) at the corners as some moment is carried because of the bolts attaching to the scale plate. The total deflection at the center of the scale plate is 0.033 inches (0.84 mm) and primarily caused by deflection in the load cells which cannot be controlled. However, the deflection dependent on the thickness of the scale plate can be changed. It is desirable to keep the deflection of the scale low to allow better accuracy of wheel plane motion. The 2.5 inches (64 mm) scale plate deflection was found to be about 0.015 inches (0.38 mm) by subtracting the deflection at the end of the load cell from the deflection at the center of the scale plate. The deflection of 0.015 inches (0.38 mm) was deemed acceptable and the thickness of the scale plate was not increased.

The FEA simulation was validated using stress and deflection formulas for flat plates from *Formulas for Stress and Strain* by Roark and Young [24]. Stress and deflection equations for a flat plate simply supported on all four edges with a uniform loading over a concentric circle with a constant radius. This was the closest approximation to the actual scenario. The deflection was expected to possibly be significantly higher in the simulation since the edges between the load cells are allowed to deflect. However, the formulas from Roark and Young were expected to provide close
approximations of the maximum stress at the center of the plate as the actual constraints of the load cells allow the edges to freely bend, acting similarly as simply supported edges. Although the loading of the scale plate is uniform over an annular surface, it was estimated by a circle of equal area. The equations for stress and deflection are given below.

\[
Max \sigma = \frac{3W}{2\pi t^2} \left[ (1 + \nu)\ln \frac{2b}{r_0} + \beta \right] \quad (4.3)
\]

\[
Max y = \frac{-awb^2}{Et^3} \quad (4.4)
\]

Where:

- \( W \) = load (lb)
- \( t \) = plate thickness (in)
- \( b \) = length, selected as the smallest value between the width and length of the plate (in)
- \( \beta \) = Non-dimensional parameter based on the width to length ratio
- \( \alpha \) = Non-dimensional parameter based on the width to length ratio
- \( \nu \) = poison’s ratio
- \( E \) = Young’s modulus of elasticity (psi)
- \( r_0' \) = equivalent radius of contact load (in)

The values of \( \alpha \) and \( \beta \) are given in a table based on the ratio between the plate width and length. Both values had to be interpolated. The equivalent radius of the contact load used for small radii is determined based on the thickness of the plate. According to Roark and Young, the equivalent radius of the contact load is equal to the radius of the
contact load in this case because the radius is more than twice the thickness of the plate.

The values used to calculate the maximum stress and deflection are given below.

\[
\begin{align*}
W &= 50,000 \text{ lbs} \\
 t &= 2.5 \text{ in} \\
 b &= 21.5 \text{ in} \\
 \beta &= 0.539 \\
 \alpha &= 0.137 \\
 \nu &= 0.3 \\
 E &= 30 \times 10^6 \text{ psi} \\
 r_0 &= 3.4023 \text{ in}
\end{align*}
\]

Using the values above and Equations 4.3 and 4.4 the maximum stress in the plate is 9,000 psi (62 MPa) and the maximum deflection is 0.007 inches (0.2 mm). The maximum stress in the middle of the plate from the FEA simulation is about 10,000 psi (69 MPa) which is close to the estimated stress of 9,000 psi (62 MPa). The estimated deflection is about half of that from the FEA simulation, which was expected. The boundary conditions from Roark and Young assumes all edges are simply supported constraining the edges in the z direction. In the simulation, the edges are allowed to deflect, which increased the total deflection.

4.3.4 Scale Legs, Feet, and Feet Retainer

The scale feet and legs were re-designed to decrease the required machining and decrease the amount of friction between the scale legs and scale feet. The first design consisted of a threaded leg with a spherical radius on the end which rested in the spherical radius of the scale foot. This allowed the scale leg to rotate as the scale
deflected keeping the scale leg and x-direction guide block from carrying any moment. This was replaced with a scale leg without a spherical radius end and a swivel bearing from McMaster-Carr (part number 2459K14). The swivel bearing has as a static load rating of 17,600 lbs (78 kN) which is large enough to accommodate the maximum load of 12,500 lbs (56 kN) per scale leg during maximum bounce. The dimensions of the swivel bearing are shown in Figure 4.14. The scale leg is machined from a piece of hexagonal bar stock. One end is machined round then threads are cut allowing the feet to be threaded into the load cells. The other end is machined round to fit inside the swivel bearing. The hexagonal section will rest on the inner race of the swivel bearing. The hexagonal section allows the leg to be threaded into the load cell with an open end wrench. The reduced friction in the swivel bearing allows the scale leg to be easily turned for scale leg height adjustments and easily rotate as the scale plate deflects.

Figure 4.14. Scale Foot – Swivel Bearing (McMaster Part Number 2459K14)
The scale foot transfers the load to the guide block and it is important that the foot is centered on the x-direction guide block to avoid any moment being carried. It is not important that the scale foot is rigidly attached the guide block. A simple plate was designed with a four hole pattern matching the threaded holes of the guide block. Another hole was placed at the center of the four hole pattern with a 0.03125 inch (0.794 mm) clearance for the scale foot to sit. This ensures the scale foot is centered on the top of the guide block.

4.3.5 X Rail Carrier Design

The design for the x rail carrier changed more than slightly from the first design. The entire carrier sits above the y-direction guide block. This is possible because the overall height of the wheel pad was reduced with the new x-direction guide rails and guide blocks and scale design. This allows the x rail carrier to have a constant cross-sectional area by eliminating the flanges. The flanges are no longer necessary to reduce the total height of the wheel pad assembly. FEA simulations would not be very beneficial because of the simplicity of the geometry and loading conditions.

For all calculations, the x-direction guide rail and the x rail carrier were assumed to be rigidly attached and treated as one member. The cross-section of the x-direction guide rail and x rail carrier was drawn in AutoCAD and the area and moment of inertias were calculated and shown below in Figure 4.15.
Two distributed loads totaling 12,500 lbs (55.6 kN) each are applied to the x-direction guide rail by the x-direction guide blocks during maximum bounce. The guide blocks are approximately 4 inches (102 mm) in length making each distributed load 3,125 lb/in (547 N/mm). All calculations are conducted with the wheel pad centered, causing symmetrical loading. It is assumed to be simply supported as the y-direction guide blocks should not carry any moment as they are only rated for about 0.905 kN-m (8,000 in-lb) and could be at risk of failure. This can only be assumed if the clearance between the guide block and the rails allows the guide block to rotate about the sliding axis at an angle larger than the angle of deflection of the x rail carrier and guide rail. It is not known how much clearance exists between the guide block and rail, allowing it freely rotate. The x rail carrier is rigidly attached to the y-direction guide block. Therefore, if the angle of deflection at the ends is greater than the amount of rotation allowed by the guide block, it will be forced to carry a moment possibly resulting in failure. The angle of deflection at
the ends were calculated to determine if there is concern and need to take measures to accommodate it. The beam loading for the calculations is shown below in Figure 4.16.

Figure 4.16. X Rail Carrier and X-Direction Guide Rail Loading

The beam is taken to be 864 mm (34 in) to account for the support by the y-direction guide blocks under the x rail carrier. Superposition has to be used to analyze the beam because of the two distributed loads. This means the two loads have to be handled separately and the results added together. The benefit of symmetrical loading is that one set of equations can be used for both distributed loads. This is possible as the right side of the beam for one distributed load will behave the same as the left side of the beam for the other distributed load and vice versa. Therefore, only the equations calculated for a beam with the loading condition shown in Figure 4.17 are necessary to determine the behavior of the beam shown in Figure 4.16.
For simple cases like the one above, the deflection, slope and moment for each point in the beam can be solved using the mathematical representation for the curvature of a beam caused by bending, Hooke’s Law, the flexure formula, and discontinuity functions. Combining Hooke’s Law, the flexure formula, and curvature, the moment in the beam can be simply expressed by the following [25]:

\[
\frac{M}{EI} = \frac{d^2u}{dx^2} \quad (4.5)
\]

Discontinuity functions allow the moment in the beam to be expressed as a single function of distance along the beam. Using the discontinuity equations for moment caused by a point load

\[
M = -P(x - a)^1 \quad (4.6)
\]
And by a distributed load

\[ M = -\frac{w_0}{2} (x - a)^2 \quad (4.7) \]

Where

- \( P \) = point load (lbs)
- \( x_0 \) = distributed load (lb/in)
- \( x \) = distance along the beam (in)
- \( a \) = location of the load (in)

The moment equation for the beam in Figure 4.17 is:

\[ M = P(x - a_1)^1 - \frac{w_0}{2} (x - a_1)^2 + \frac{w_0}{2} (x - a_3)^2 \quad (4.8) \]

Where:

- \( P = 2573.529 \) lbs
- \( w_0 = 3125 \) lbs/in
- \( a_1 = 0 \) in
- \( a_2 = 27 \) in
- \( a_3 = 29 \) in
- \( E = 30E6 \) psi
- \( I = 8.07 \) in\(^4\)

After substituting Equation 4.8 into Equation 4.5, and integrating twice using the boundary conditions of zero deflection at both ends of the beam, the equation for moment, slope of deflection and deflection are:
\[ M = 2573.529x - 1562.5 < x - 25 >^2 + 1562.5 < x - 29 >^2 \quad (4.9) \]

\[
\frac{dv}{dx} = \left(\frac{1}{2}\right)(2573.529)x^2 - \left(\frac{1}{3}\right)(1562.5)x^2 + \left(\frac{1}{5}\right)(1562.5)x - 473100.41 \quad (4.10)
\]

\[
v = \left(\frac{1}{6}\right)(2573.529)x^3 - \left(\frac{1}{12}\right)(1562.5)x^4 + \left(\frac{1}{15}\right)(1562.5)x - 473100.41x \quad (4.11)
\]

For this loading condition the maximum bending moment is 87,500 in-lb (9886 N-m). This bending moment results in a maximum bending compressive stress of 21,600 psi (149 MPa) on the top surface of the x-direction guide rail. With a yield stress of 50,000 psi (345 MPa) the maximum bending is not of concern, but is not small enough to warrant a reduction in size especially with the concern of beam deflection. The maximum deflection at the center of the rail and rail carrier is 0.049 inches (1.3 mm) and the deflection at the guide block locations is 0.031 inches (0.8 mm). The deflection of rail and rail carrier at the guide blocks is of more interest. The deflection at the guide blocks will directly affect the overall deflection of the wheel pad. The combined deflection of the scale and the x-direction guide rails and carrier is about 0.060 inches (1.5 mm). This deflection is not a major concern, but requires the use of string pots to measure the road plane deflection. However, the slope of deflection at the ends of the x rail carrier could be of concern as it was mentioned earlier that guide blocks should not be forced to carry any moment.
The slope of deflection at the end points of the x rail carrier was calculated to be ±0.00484 which results in an angle of deflection of ±0.28 degrees. In order for the guide block to not carry any moment, the guide block must be able to rotate 0.28 degrees about the axis of travel. THK claims that their linear motion guide rails are capable of absorbing an error of 0.3 mm/200 mm which results in an angle of approximately 0.09 degrees [26]. Any angle larger than that would force the guide block to carry moment. With an angular deflection of 0.28 degrees, it is likely that if the x rail carrier is mounted rigidly to the y-direction guide blocks, they will fail. With the current design, modifications had to be made to allow the beam to deflect without forcing the guide blocks to carry moment. It was decided that if the bolts fixing the x rail carrier to the y-direction guide block are fastened loosely, there would be enough clearance between the bolts and the x rail carrier to allow an angular deflection of 0.28 degrees. Loctite or similar product would need to be used to ensure the fasteners do not back-out. Other possible solutions requiring more extensive redesign are presented in Chapter 5.

Not only does the stress and deflection of the x rail and carrier have to be evaluated, but the shear flow between the x rail and x rail carrier has to be calculated to determine if the number and size of fasteners are adequate. Shear flow is defined as:

\[ q = \frac{VQ}{l} \] (4.12)
Where:

$q = \text{shear flow (lb/in)}$
$V = \text{shear (lb)}$
$I = \text{moment of inertia (in}^4\text{)}$
$Q = A'y' \text{ (in}^3\text{)}$
$A' = \text{area above or below the shear plane (in}^2\text{)}$
$y' = \text{distance between the centroid of } A' \text{ and the centroid of the total area (in)}$

Using the geometry in Figure 4.15 and the loading condition in Figure 4.16

$V = 12,500 \text{ (lb)}$
$I = 8.07 \text{(in}^4\text{)}$
$Q = 1.368 \text{ (in}^3\text{)}$

The shear flow is calculated to be 2120 lb/in (371 N/mm). With the length of the x-direction guide rail and rail carrier being 36 in (914 mm) long and a total of 12 bolts, the shear force per bolt is 6360 lbs (28 kN). The fasteners used for the rail are M8 bolts which have a cross-sectional area of 37 mm$^2$ (0.058 in$^2$). This area results in a shear stress of 109,000 psi (752 MPa) which creates the need for a re-design. It was decided that a quick fix would be to keep the fasteners loose, allowing separation between the x-direction guide rail and x rail carrier. This would decrease the shear force in the bolts, but would increase the slope of deflection. The deflection should not increase significantly as the majority of the moment of inertia comes from the x rail carrier. More substantial possible re-designs are presented in Chapter 5.
4.4 Initial Roll Constraint Design

It is important that during roll testing the bounce and roll frame is subjected to pure roll to achieve accurate results. It is impossible to do this by simply actuating the hydraulic cylinders in the opposite direction at the same time. A constraint must be designed to force the bounce and roll frame to roll about a pivot on the road plane. The initial design concept is shown in Figure 4.18 and Figure 4.19.

![Figure 4.18. Bounce and Roll Beam Roll Constraint](image)

This constraint design allows the bounce frame to freely move vertically, but limits the roll motion to rotate about a pivot point in the road plane. In the figure above, the magenta member is one of the bounce and roll frame’s main beams and the black members are fixed and grounded to the t-plate. The red plate, or roll plate, fits between the two black members in grooves in either sided that allow the plate to move vertically. The clearance on the right and left side of the roll plate is small enough to constrain the plate from rotating and a lubricant is applied to reduce friction between the grounded members and the roll plate. The roll plate has two slots which fit over the roll studs which welded to the main beam and are shown in gray in Figure 4.19. The radius and
location of the curved roll studs place the center of the arc on the road plane allowing the bounce and roll frame to rotate in pure roll.

Figure 4.19. Bounce and Roll Frame Roll Constraint Close-Up

TARDEC was concerned with this design because when the bounce and roll frame is in full rebound, the grounded members would extend up past the top plate and possibly interfere with a suspension member or something on the undercarriage of the vehicle. Therefore the roll constraint method was redesigned.
4.5 Final Roll Constraint Design

With the concern of objects interfering with components under the vehicles’ frame between the wheels, the roll constraint had to be placed outside of the maximum track width. It was decided to place the roll constraint components on the ends of the roll frame to ensure there would be no interference with any vehicle components while keeping the overall height of the roll frame the same.

Two plates were designed to attach to the ends of the roll frame that would slide vertically within a guide on the main vertical supports of the support frame. The guide is a simple piece of steel with a groove. The roll plate rides in the groove of the guide bar with a lubricant added to reduce friction between the two parts. The ends of the roll plates have a large radius with its center in the middle of the roll frame on the road plane. This allows the bounce and roll frame to freely move vertically and roll about a pivot point creating a pure roll moment as long at the pin connection between the hydraulic cylinder and the bounce and roll frame also lies on the road plane. The guide bar and roll plate are shown in Figure 4.20 and Figure 4.21.

Aside from the hydraulic cylinders, the only restraint of the bounce and roll frame is from the roll plate and guide bar. When lateral wheel plane sliding forces are simulated during lateral compliance testing, the force from the cylinder attached to the bounce and roll frame is transferred to the support frame through roll plates.
Figure 4.20. Roll Plate Detail Drawing

Figure 4.21. Roll Plate Guide Bar Detail Drawing
Contact stresses needed to be examined to determine if the plates could withstand the lateral force. During lateral compliance testing, a lateral force of 60% of the curb weight on each wheel is applied. Since the maximum curb weight per axle is 50,000 lbs (222 kN), the maximum lateral force will be 60% of 25,000 lbs (111 kN). The contact stress can be calculated using equations for contact stress between two cylinders, one with and infinite radius.

In *Mechanical Design of Machine Elements and Machines* by Jack Collins the contact pressure between two cylinders is defined as [22]:

\[ p_{max} = \frac{2F}{\pi bL} \quad (4.13) \]

\[ b = \sqrt{\frac{2F\left[\left(\frac{1-v_2^2}{E_1}\right) + \left(\frac{1-v_1^2}{E_1}\right)\right]}{\pi L\left(\frac{1}{d_1} + \frac{1}{d_2}\right)}} \quad (4.14) \]

Where:

- \( F \) = Normal force (lb)
- \( E_1, E_2 \) = Modulous of Elasticity for Cylinders 1 and 2 (psi)
- \( d_1, d_2 \) = Diameter for Cylinders 1 and 2 (in)
- \( L \) = length of contact between cylinders (in)
- \( v_1, v_2 \) = Poisson’s ratio for cylinders 1 and 2

The values below result in a contact pressure of 18,200 psi (125 MPa).

\[ F = 15,000 \text{ (lb)} \]
\[ E_1, E_2 = 30E5 \text{ (psi)} \]
\[ d_1 = 189.94 \text{ (in)} \]
\[ d_2 = \infty \text{ (in)} \]
\[ L = 2.5 \text{ (in)} \]
\[ v_1, v_2 = 0.3 \]
In Collins’ book the maximum principal stresses occurs at the contact between the two cylinders and the maximum shear stress occurs at approximately $0.75b$ below the contact surface [22]. The maximum stresses are defined as:

\[ \sigma_1 = -2\nu p_{max} \quad (4.15) \]

\[ \sigma_2 = \sigma_3 = -p_{max} \quad (4.16) \]

\[ \tau_1 = 0.3p_{max} \quad (4.17) \]

And Von-Mises stress is defined as:

\[ \sigma = \sqrt{0.5[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]} \quad (4.17) \]

Using $p_{max} = 18,207$ the principal stresses and Von-Mises stress are calculated to be:

$\sigma_1 = -10,900$ psi (-75 MPa)
$\sigma_2 = -18,200$ psi (-126 MPa)
$\sigma_3 = -18,200$ psi (-126 MPa)
$\tau_1 = 5,500$ psi (38 MPa)
$\sigma = 7,300$ psi (50 MPa)

All of the stresses are less than half the yield strength of steel common low carbon steel and not of concern.
Chapter 5

Future Work and Possible Design Changes

5.1 Adaptation for Compliance Testing

One of TARDEC’s requirements for the SPIdER is the ability to be adapted for lateral and longitudinal compliance testing. This means that actuators must be attached between the bounce and roll frame and the wheel pad assembly to apply road plane forces to the tires. This requires a minimum of two actuators per wheel pad; one for the longitudinal direction and one for the lateral direction. Hydraulic cylinders would most likely be used because of the already existing components to power the hydraulic cylinders for bounce and roll. Servo motors with belts and pullies would be another option to consider.

Constraints to consider when selecting an actuator are the ability to apply a maximum force of 67 kN (15,000 lbs) and clearance under the wheel pad assembly and between the two main beams of the bounce and roll frame. When designing the attachment method of the actuators to the wheel pad assembly, care must be taken so that the guide blocks are not required to carrier any moment. For instance, if the lateral force actuator is attached to the x rail carrier, the force will be transferred through the x-
direction guide blocks to scale legs and eventually to the top plate. This places a moment on the x-direction guide blocks. However, if the actuator is attached directly to the scale the only force carried by guide blocks is the force required to overcome the friction between the y-direction guide rails and guide blocks. This force will be very minimal and will not create a moment larger than the guide blocks’ capacity.

It is important that the actuators are able to produce pure longitudinal and lateral forces for accurate measurements. If the longitudinal or lateral actuators have force vectors with multiple components the compliance parameters will be inaccurate. This happens because the measured force by the actuator will not equal the force in the longitudinal or lateral direction. Another issue arises if the actuators apply forces with multiple components. If the actuator force vectors have a z component, the readings from the scale will no longer be valid and could possibly affect some compliance characteristics if they are dependent on the tire’s normal force.

It is also important that the actuators are able to be kept from applying any forces to the wheel pad. During bounce and roll testing it is important that the wheel is able to freely move longitudinally and laterally for accurate results. It is also important that during longitudinal compliance testing the wheel pad is able to freely move in the lateral direction and vice versa. If not, compliance characteristics will also be inaccurate.

Other than the actuator selection, mounting, and installation, no other modifications should be necessary to the wheel pad assembly. Additionally, the current wheel motion measurements should be adequate for compliance testing as there should not be any additional wheel measurement requirements.
5.2 Wheel Pad Design Changes

As mentioned in Chapter 4 there were two aspects of the x rail carrier design that could be improved upon. One aspect pertains to the slope of deflection at the ends of the x rail carrier and the other being the shear in the bolts fastening the x-direction guide rail to the x rail carrier. Both issues can be solved by increasing size of the x rail carrier. Increasing the height of the x rail carrier would have a more significant impact as it would obviously create a greater increase in moment of inertia. The down side to increasing the height of the x rail carrier is that the overall height of the wheel pad assembly would also increase in height. This would force a major redesign as the roll plate, hydraulic cylinder mounting, and drive on ramps, among others, have been designed for the current wheel pad height. As a result, it would not be convenient to increase the height of the x rail carrier. Increasing the width of the x rail carrier would also decrease the slope of deflection and shear stress in the bolts. Doubling the width of the x rail carrier would approximately cut the slope of deflection and shear stress in half. Doubling the width of the x rail carrier would significantly increase the cost of material and would cause clearance issues with the scale plate fitting between the two x rail carriers resulting in more re-design. Instead of making one change to the x rail carrier to solve both issues, separate changes could be made.

The shear force in the bolts could be decreased by doing two things. Additional holes could be drilled in the x-direction rail and x rail carrier. Doubling the number of fasteners would decrease the shear force by 50%. Another option would be to increase
the size of the bolts. This would require enlarging the existing holes, but would not require any additional design changes. Doubling the bolt diameter would have a greater impact on the reduction of shear stress as the area is a function of the radius squared, but the width of the x-direction guide rail limits the fastener size to 11M or smaller. The largest decrease in shear stress would obviously be a combination of both design changes. The concern of the x rail carrier slope deflection could be eliminated if the attachment between the x-direction guide rail and the x rail carrier was designed as a true simple support. It could be possible to design a clevis type attachment which fastens to the y-direction guide block and is pinned to the x rail carrier allowing the ends of the x carrier to freely deflect eliminating the moment carried by the y-direction guide blocks. This re-design would be significantly more extensive and difficult than previous suggestions, but could eliminate the need to re-design the other components.
References


Appendix A:

MatLab Code Table of Results for MicroScribe Accuracy and

clc;
clear;
close;
format compact
%Script file was created to compute the error in the microcribe

x0 = 0; %distance from wheel center to stylist tip in x
y0 = 20; %distance from wheel center to stylist tip in y
z0 = 0; %distance from wheel center to stylist tip in z

d_e_x = 0.01; %maximum error in x
d_e_y = -0.25; %maximum error in y
d_e_z = -0.01; %maximum error in x

e_s = -.5; %maximum error in steer [degrees]
e_c = -.5; %maximum error in steer [degrees]
caster = -5; %caster angle

T_steer_e_0 = [cosd(e_s) sind(e_s) 0; -sind(e_s) cosd(e_s) 0; 0 0 1]; %Steer Transformation matrix with initial error
T_camber_e_0 = [1 0 0; 0 cosd(e_c) -sind(e_c); 0 sind(e_c) cosd(e_c)]; %Camber Transformation matrix with initial error
T_caster = [cosd(caster) 0 -sind(caster); 0 1 0; sind(caster) 0 cosd(caster)];

%% Case 1

wc0 = [0 y0 0]'; %initial distance from stylist tip to wheel center [x y z]
wc0_e = (T_steer_e_0*T_camber_e_0)*[x0 y0 z0]'; %initial distance from stylist tip to wheel center with error [x y z]

i=0;
j=0;

for camber = 1:2:5;
i=i+1;
    for steer = 1:2:5;
j=j+1;
        T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1];
%transformation matrix for steer
        T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)];
%transformation matrix for camber
        WC = T_steer*T_camber*wc0;
        WC_e = T_steer*T_camber*T_caster*wc0_e;

        Change = WC-wc0;
        Change_e = WC_e-wc0_e;

        Error = Change - Change_e;
        Error_xyz_case_1(i,j) = Error(1);
        Error_xyz_case_1(i+1,j) = Error(2);
        Error_xyz_case_1(i+2,j) = Error(3);
    end
    j=0;
i=i+2;
end

Error_xyz_case_1
xlswrite('microscribe error calculation',Error_xyz_case_1, 'sheet1','C3')

max_1_x =
    max([abs(Error_xyz_case_1(1,:)),abs(Error_xyz_case_1(4,:)),abs(Error_xyz_case_1(7,:))]);
max_1_y =
    max([abs(Error_xyz_case_1(2,:)),abs(Error_xyz_case_1(5,:)),abs(Error_xyz_case_1(8,:))]);
max_1_z =
    max([abs(Error_xyz_case_1(3,:)),abs(Error_xyz_case_1(6,:)),abs(Error_xyz_case_1(9,:))]);
%%% Case 2

\( wc_0 = [0 \ 0 \ 0]' \); initial distance from stylist tip to wheel center [x y z]

\( wc_0_e = T_{\text{steer e_0}} * T_{\text{camber e_0}} * [x_0 + d_{e_x} \ y_0 \ z_0]' \); initial distance from stylist tip to wheel center with error [x y z]

\[ \begin{align*}
i &= 0; \\
j &= 0;
\end{align*} \]

\[ \begin{align*}
\text{for} \ & \text{camber} = 1:2:5; \\
& \quad i = i + 1; \\
& \quad \text{for} \ & \text{steer} = 1:2:5; \\
& \quad & \quad j = j + 1; \\
& \quad & \quad T_{\text{steer}} = [\cos(\text{steer}) \ \sin(\text{steer}) \ 0; -\sin(\text{steer}) \ \cos(\text{steer}) \ 0; 0 \ 0 \ 1]; \\
& \quad & \quad \% \text{transformation matrix for steer} \\
& \quad & \quad T_{\text{camber}} = [1 \ 0 \ 0; 0 \ \cos(\text{camber}) \ -\sin(\text{camber}); 0 \ \sin(\text{camber}) \ \cos(\text{camber})]; \\
& \quad & \quad \% \text{transformation matrix for camber}
\end{align*} \]

\[ \begin{align*}
WC &= T_{\text{steer}} * T_{\text{camber}} * wc_0; \\
WC_e &= T_{\text{steer}} * T_{\text{camber}} * T_{\text{caster}} * wc_0_e;
\end{align*} \]

\[ \begin{align*}
\text{Change} &= WC - wc_0; \\
\text{Change}_e &= WC_e - wc_0_e;
\end{align*} \]

\[ \begin{align*}
\text{Error} &= \text{Change} - \text{Change}_e; \\
\text{Error}_{xyz\_case\_2}(i,j) &= \text{Error}(1); \\
\text{Error}_{xyz\_case\_2}(i+1,j) &= \text{Error}(2); \\
\text{Error}_{xyz\_case\_2}(i+2,j) &= \text{Error}(3);
\end{align*} \]

\[ \begin{align*}
\text{end} \\
j &= 0; \\
i &= i + 2;
\end{align*} \]

\[ \begin{align*}
\text{end} \\
\text{Error}_{xyz\_case\_2} \\
\text{xlswrite('microscribe error calculation',Error}_{xyz\_case\_2}, 'sheet1', 'C15')
\end{align*} \]

\[ \begin{align*}
\text{max}_2\_x &= \max([\text{abs}(\text{Error}_{xyz\_case\_2}(1,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(4,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(7,:))])); \\
\text{max}_2\_y &= \max([\text{abs}(\text{Error}_{xyz\_case\_2}(2,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(5,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(8,:))])); \\
\text{max}_2\_z &= \max([\text{abs}(\text{Error}_{xyz\_case\_2}(3,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(6,:)), \text{abs}(\text{Error}_{xyz\_case\_2}(9,:))]));
\end{align*} \]
%%% Case 3

wc0 = [0 y0 0]'; %initial distance from stylist tip to wheel center [x y z]
w0_c_e = T_steering_e_0*T_camber_e_0*[x0 y0+d_e_y z0]'; %initial distance from stylist tip to wheel center with error [x y z]

i=0;
j=0;

for camber = 1:2:5;
i=i+1;
  for steer = 1:2:5;
    j=j+1;
      T_steering = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1];
      %transformation matrix for steer
      T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)];
      %transformation matrix for camber
      WC = T_steering*T_camber*wc0;
      WC_e = T_steering*T_camber*T_caster*wc0_e;
      Change = WC-wc0;
      Change_e = WC_e-wc0_e;
      Error = Change - Change_e;
      Error_xyz_case_3(i,j) = Error(1);
      Error_xyz_case_3(i+1,j) = Error(2);
      Error_xyz_case_3(i+2,j) = Error(3);
  end
j=0;
i=i+2;
end

Error_xyz_case_3
xlswrite('microscribe error calculation',Error_xyz_case_3, 'sheet1', 'C27')

max_3_x =
max([abs(Error_xyz_case_3(1,:)),abs(Error_xyz_case_3(4,:)),abs(Error_xyz_case_3(7,:))]);
max_3_y =
max([abs(Error_xyz_case_3(2,:)),abs(Error_xyz_case_3(5,:)),abs(Error_xyz_case_3(8,:))]);
max_3_z = max([abs(Error_xyz_case_3(3,:)), abs(Error_xyz_case_3(6,:)), abs(Error_xyz_case_3(9,:))]);

%%% Case 4

wc0 = [0 y0 0]'; % initial distance from stylist tip to wheel center [x y z]
wc0_e = T_steer_e_0*T_camber_e_0*[x0 y0 z0+d_e_z]'; % initial distance from stylist tip to wheel center with error [x y z]

i=0;
j=0;
for camber = 1:2:5;
    i=i+1;
    for steer = 1:2:5;
        j=j+1;
        T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1];
        % transformation matrix for steer
        T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)];
        % transformation matrix for camber
        WC = T_steer*T_camber*wc0;
        WC_e = T_steer*T_camber*T_caster*wc0_e;
        Change = WC-wc0;
        Change_e = WC_e-wc0_e;
        Error = Change - Change_e;
        Error_xyz_case_4(i,j) = Error(1);
        Error_xyz_case_4(i+1,j) = Error(2);
        Error_xyz_case_4(i+2,j) = Error(3);
    end
    j=0;
    i=i+2;
end
Error_xyz_case_4
xlswrite('microscribe error calculation',Error_xyz_case_4, 'sheet1', 'C39')

max_4_x = max([abs(Error_xyz_case_4(1,:)), abs(Error_xyz_case_4(4,:)), abs(Error_xyz_case_4(7,:))]);
max_4_y = max([abs(Error_xyz_case_4(2,:)),abs(Error_xyz_case_4(5,:)),abs(Error_xyz_case_4(8,:))]);
max_4_z = max([abs(Error_xyz_case_4(3,:)),abs(Error_xyz_case_4(6,:)),abs(Error_xyz_case_4(9,:))]);

%% Case 5

wc0 = [0 y0 0]'; %initial distance from stylist tip to wheel center [x y z]
w0_e = T_steer_e_0*T_camber_e_0*[x0+d_e_x y0+d_e_y z0]'; %initial distance from stylist tip to wheel center with error [x y z]
i=0;
j=0;

for camber = 1:2:5;
i=i+1;
for steer = 1:2:5;
j=j+1;
    T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1]; %transformation matrix for steer
    T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)]; %transformation matrix for camber
    WC = T_steer*T_camber*wc0;
    WC_e = T_steer*T_camber*T_caster*wc0_e;
    Change = WC-wc0;
    Change_e = WC_e-wc0_e;
    Error = Change - Change_e;
    Error_xyz_case_5(i,j) = Error(1);
    Error_xyz_case_5(i+1,j) = Error(2);
    Error_xyz_case_5(i+2,j) = Error(3);
end
j=0;
i=i+2;
end
Error_xyz_case_5
xlswrite('microscribe error calculation',Error_xyz_case_5, 'sheet1','C51')
max_5_x =
max([abs(Error_xyz_case_5(1,:)), abs(Error_xyz_case_5(4,:)), abs(Error_xyz_case_5(7,:))]);
max_5_y =
max([abs(Error_xyz_case_5(2,:)), abs(Error_xyz_case_5(5,:)), abs(Error_xyz_case_5(8,:))]);
max_5_z =
max([abs(Error_xyz_case_5(3,:)), abs(Error_xyz_case_5(6,:)), abs(Error_xyz_case_5(9,:))]);

%%% Case 6

wc0 = [0 y0 0]';
%initial distance from stylist tip to wheel center [x y z]
wc0_e = T_steer_e_0*T_camber_e_0*[x0+d_e_x y0 z0+d_e_z]';
%initial distance from stylist tip to wheel center with error [x y z]

i=0;
j=0;

for camber = 1:2:5;
i=i+1;
for steer = 1:2:5;
j=j+1;
T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1];
%transformation matrix for steer
T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)];
%transformation matrix for camber
WC = T_steer*T_camber*wc0;
WC_e = T_steer*T_camber*T_caster*wc0_e;
Change = WC-wc0;
Change_e = WC_e-wc0_e;
Error = Change - Change_e;
Error_xyz_case_6(i,j) = Error(1);
Error_xyz_case_6(i+1,j) = Error(2);
Error_xyz_case_6(i+2,j) = Error(3);
end
j=0;
i=i+2;
end
Error_xyz_case_6
max_6_x = max([abs(Error_xyz_case_6(1,:)),abs(Error_xyz_case_6(4,:)),abs(Error_xyz_case_6(7,:))]);
max_6_y = max([abs(Error_xyz_case_6(2,:)),abs(Error_xyz_case_6(5,:)),abs(Error_xyz_case_6(8,:))]);
max_6_z = max([abs(Error_xyz_case_6(3,:)),abs(Error_xyz_case_6(6,:)),abs(Error_xyz_case_6(9,:))]);

%% Case 7
wc0 = [0 y0 0]';
wc0_e = T_steer_e_0*T_camber_e_0*[x0 y0+d_e_y z0+d_e_z]';

for i=0;
    j=0;
    for camber = 1:2:5;
        i=i+1;
        for steer = 1:2:5;
            j=j+1;
            T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1];
            T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)];
            WC = T_steer*T_camber*wc0;
            WC_e = T_steer*T_camber*T_caster*wc0_e;
            Change = WC-wc0;
            Change_e = WC_e-wc0_e;
            Error = Change - Change_e;
            Error_xyz_case_7(i,j) = Error(1);
            Error_xyz_case_7(i+1,j) = Error(2);
            Error_xyz_case_7(i+2,j) = Error(3);
        end
        j=0;
        i=i+2;
    end
max_7_x =
max([abs(Error_xyz_case_7(1,:)),abs(Error_xyz_case_7(4,:)),abs(Error_xyz_case_7(7,:))]);
max_7_y =
max([abs(Error_xyz_case_7(2,:)),abs(Error_xyz_case_7(5,:)),abs(Error_xyz_case_7(8,:))]);
max_7_z =
max([abs(Error_xyz_case_7(3,:)),abs(Error_xyz_case_7(6,:)),abs(Error_xyz_case_7(9,:))]);

%% Case 8
wc0 = [0 y0 0]'; %initial distance from stylist tip to wheel center [x y z]
w0_e = T_steer_e_0*T_camber_e_0*[x0+d_e_x y0+d_e_y z0+d_e_z]'; %initial
distance from stylist tip to wheel center with error [x y z]

i=0;
j=0;

for camber = 1:2:5;
i=i+1;
    for steer = 1:2:5;
j=j+1;
        T_steer = [cosd(steer) sind(steer) 0; -sind(steer) cosd(steer) 0; 0 0 1]; %transformation matrix for steer
        T_camber = [1 0 0; 0 cosd(camber) -sind(camber); 0 sind(camber) cosd(camber)]; %transformation matrix for camber
        WC = T_steer*T_camber*wc0;
wC_e = T_steer*T_camber*T_caster*wc0_e;
        Change = WC-wc0;
        Change_e = WC_e-wc0_e;
        Error = Change - Change_e;
        Error_xyz_case_8(i,j) = Error(1);
        Error_xyz_case_8(i+1,j) = Error(2);
        Error_xyz_case_8(i+2,j) = Error(3);
    end
j=0;
i=i+2;
end

Error_xyz_case_8
xlswrite('microscribe error calculation',Error_xyz_case_8, 'sheet1','C87')

max_8_x = max([abs(Error_xyz_case_8(1,:)),abs(Error_xyz_case_8(4,:)),abs(Error_xyz_case_8(7,:))]);
max_8_y = max([abs(Error_xyz_case_8(2,:)),abs(Error_xyz_case_8(5,:)),abs(Error_xyz_case_8(8,:))]);
max_8_z = max([abs(Error_xyz_case_8(3,:)),abs(Error_xyz_case_8(6,:)),abs(Error_xyz_case_8(9,:))]);

max_x = max([max_1_x max_2_x max_3_x max_4_x max_5_x max_6_x max_7_x max_8_x])
max_y = max([max_1_y max_2_y max_3_y max_4_y max_5_y max_6_y max_7_y max_8_y])
max_z = max([max_1_z max_2_z max_3_z max_4_z max_5_z max_6_z max_7_z max_8_z])
Table A.1. MicroScribe Accuracy Results

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<th>Negative and Positive Error Combination</th>
<th>Maximum Error of all 8 Cases</th>
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<td>X Y Z Steer Camber Caster Max X (in)</td>
<td>Max Y (in)</td>
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</tr>
<tr>
<td>- + + + + + +</td>
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<tr>
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</tr>
<tr>
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<tr>
<td>- + + + + + +</td>
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Average Maximum Error 0.0272 0.0234 0.0270