Wind Tunnel Wheel Balance
Final Design Report

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Executive Summary

The Cal Poly Wind Tunnel Wheel Balance team has been asked to design, build, and test a wheel support for the interior of a subsonic wind tunnel for the Cal Poly Aerospace Engineering Department. The objective of the project is to reduce the wear on the wind tunnel rolling road belt by lifting the wheel so there is minimal contact between the belt and the aluminum frame of the rolling road. Actuation of the wheel is achieved through three independent axis controls—horizontal (x), transverse (y), and vertical (z). The horizontal direction actuation is achieved via rail translation, which is controlled by the user and is held in place by a hand brake. The transverse and vertical directions achieve movement via hand-powered and motorized lead screws respectively. The overall design accommodates wheels sizes ranging from 5” to 20” diameters. The final design of this project is a prototype to show the feasibility of such a design with the eventual goal of building four such mechanisms inside the tunnel in order to support a scaled Formula SAE vehicle or something similar.
1. Introduction

The wind tunnel is an indispensable tool in developing modern aircraft and vehicles. While the benefit of a wind tunnel is obvious in modern times, this wasn’t always the case. The advent of the modern wind tunnel began nearly 30 years before the Wright brothers took off at Kitty Hawk; scientists were doing aerodynamics experiments long before that as well. Philosophers and mathematicians alike have been capitalizing on wind since the beginning of the 1700’s. These great minds would test apparatuses near cliffs edges where wind speeds were high enough to produce repeatable results. After Orville and Wilbur Wright’s infamous 1901 Kitty Hawk disaster, they began testing airfoils by riding bicycles quickly back and forth on a street with airfoils attached. The crude data that the brothers gathered defied the conventional handbook information. They were encouraged by their findings enough to build the first ever wind tunnel. This tunnel was a small square channel (16 in\(^2\)) that had two balances holding the foil in the middle, which gave a reading of torque applied to the wing. The Wright brothers found that the heart of any great wind tunnel are the supports and devices that output the data. Wind tunnel technology continues to advance at unprecedented rates. The technology in this industry is highly coveted and very secretive in part because the formula racing industry is on the forefront. These racing companies hold their secrets very closely so they do not give their competitors any sort of edge over them. Meanwhile, research and design projects for wind tunnels and their supports are underway at California Polytechnic State University (Cal Poly) in San Luis Obispo, California.

The Cal Poly Aerospace Engineering department recently acquired a rolling road from Dan Gurney’s All American Racers Company to be used in a subsonic wind tunnel located on the campus. To complement this rolling road, a team of undergraduate engineering students led by Andrew Furmidge (Mechanical Engineering Undergraduate) designed and built a closed test section, shown in Figure 1, for the subsonic wind tunnel with help from Dr. Graham Doig, an Assistant Professor in the Aerospace Engineering department at Cal Poly. The rolling road that was donated is approximately 36 inches wide, by 42 inches tall, by 10 feet long and runs at a top speed of nearly 100 mph. This rolling road has been a significant contribution to the Aerospace Engineering department, but there are many additions needed in order to truly provide a state-of-the-art learn by doing educational experience for future students.

As a part of the capstone experience to the Mechanical Engineering degree at Cal Poly, senior engineering students are required to participate in a senior project. Our team has been assigned to make one of these major additions to the wind tunnel for our senior project. The team is comprised of Brady Hiob (Mechanical Engineering), Ryan Hamamura (Mechanical Engineering), and Samuel Fleet (Mechanical Engineering). Dr. Graham Doig is serving as both sponsor and the final customer.
Figure 1. Solid Works Rendering of Cal Poly Wind Tunnel with Test Section

We have been tasked with designing and building a prototype wheel balance to reduce the wear on the rolling road belt caused by the weight of the wheels as they roll along the belt. Without the wheel support system, there is excessive wear on both the belt and the aluminum frame. This results in frequent replacement of the belt which is extremely expensive. The overall scope of this project is to design, build, and test a prototype wheel balance to be installed in the wind tunnel. The completed design must be able to incorporate a standard mount for any given wheel and produce minimal flow interference. This wheel balance will be designed to support the following Cal Poly Clubs: Formula SAE/Electric, Human-Powered Vehicle (HPV), Prototype Vehicles Laboratory (PROVE Lab), and Supermileage.

This document will serve as a final design report that defines the design requirements and explains how we determined the appropriate modeling of the wheel support. In addition, this document describes the design process, timelines, and analysis for the final design.
2. Theory

2.1 Aerodynamic Forces

Aerodynamic forces occur on a body immersed in a fluid due to the relative movement of the body and the surrounding fluid\(^1\). The aerodynamic forces that result from the interaction between a moving body and the surrounding fluid can be given in terms of the forces at the fluid-body interface.

2.2 Drag and Lift

In this section, we discuss lift and drag as well as similarities between wind tunnel testing and the application of that knowledge in real world environments in terms of scaled models applying to full scale designs.

2.2.1 Drag

Drag can be defined as the resultant force in the direction of the upstream velocity, as shown in Figure 2 \([2]\). Drag force depends of fluid velocity and it decreases the fluid velocity \([3]\). Thus drag force is also referred to as air resistance or fluid resistance.

2.2.2 Lift

In contrast with the drag force, lift force exerts a force on the body normal to the upstream velocity. The following figure shows the lift force.

![Figure 2](image)

Figure 2. Forces from the surrounding fluid on a two-dimensional object: (a) pressure force, (b) viscous force, (c) resultant force (lift and drag) \([4]\).

2.2.3 Importance of Similarity in Wind Tunnel Testing

When considering aspects of experiments using scale models, results of scale model experiments may effectively be used to predict full-scale behavior. When a body moves through a fluid, forces arise that are due to the viscosity of the fluid, its inertia, its elasticity, and gravity. These forces are represented directly by the various terms given by the Navier-Stokes equation for the case of a viscous compressible fluid with body force of gravitational origin can be written as
\[
\rho \left( \frac{\delta V}{\delta t} + (V \cdot \nabla)V \right) = \rho g - V \left( p + \frac{2}{3} \mu \nabla \cdot V \right) + 2V \cdot (\mu \dot{S})
\]  

(1)

with pressure \( p \), coefficient of viscosity \( \mu \), rate of strain tensor \( \dot{S} \), velocity of the material element with respect to the same inertial frame \( V \), density \( \rho \), and the assumption that the bulk modulus is \( -\frac{2}{3} \times \) the coefficient of viscosity. The inertia force, corresponding to the left-hand side of the Navier-Stokes equation, is proportional to the mass of the air affected and the acceleration give that mass. Thus, the inertia force is the result of giving a constant acceleration to some “effective” volume. Then the inertia force can be given by

\[
\text{Inertia force} \sim \frac{\rho l^3 V}{t}
\]  

(2)

where \( \rho \) is the air density (slugs/ft\(^3\)), \( l \) is the characteristic length of the body (ft), \( V \) is the velocity of the body (ft/sec), and \( t \) is time (sec). Substituting \( l/V \) for \( t \), we get

\[
\text{Inertia force} \sim \rho l^2 V^2
\]  

(3)

According to this definition, the viscous force may be written as

\[
\text{Viscous force} \sim \mu Vl
\]  

(4)

where \( \mu \) is the coefficient of viscosity (slug/ft\(-\text{sec}\)). The gravity force is proportional to the volume of the body and can be written

\[
\text{Gravity force} \sim \rho l^3 g
\]  

(5)

The elastic force may be considered to be

\[
\text{Elastic force} \sim pl^2
\]  

(6)

Since pressure is related to density and the speed of sound \( a \) according to

\[
a^2 \sim \frac{p}{\rho}
\]  

(7)

so that we may write
Dividing the inertia force by each of the others gives three force ratios that can be seen in equations (9)-(11).

\[
\text{Reynolds number} = \frac{\text{inertia force}}{\text{viscous force}} = \frac{\rho V l}{\mu} \quad (9)
\]

\[
\text{Mach number} = \frac{\text{inertia force}}{\text{elasticity force}} = \frac{V}{a} \quad (10)
\]

\[
\text{Froude number} = \frac{\sqrt{\text{inertia force}}}{\sqrt{\text{gravity force}}} = \frac{\sqrt{V^2}}{\sqrt{l g}} \quad (11)
\]

These equations provide a foundation for designing scale experiments and interpreting the resulting data [5]. For wind tunnel experiments, the Froude number is an important similarity parameter only for dynamic tests in which model motion and aerodynamic effects are involved. These experiments are not within the scope of this project and thus the Froude number will not be a significant consideration. Matching of the Mach number usually applies only to flight vehicles in the high-speed flight region as Mach number effects predominate and the matching of Reynolds number effects is not as critical. The project will focus on ground vehicles at subsonic wind speeds which are more sensitive to Reynolds number effects.

According to the scaling relations for which the flow characteristics are a function of only Reynolds number, the force on a body of a particular shape is the same regardless of the combination of size and speed used to produce the particular Reynolds number (assuming the fluid, its temperature, and the free-stream pressure are unchanged). This relationship can be seen by writing the expression for a particular force component; taking drag for example

\[
D = \frac{1}{2} \rho_\infty V_\infty^2 l^2 C_d (R_e) = \frac{1}{2} \rho_\infty^2 V_\infty^2 l^2 \frac{\mu_\infty^2}{\mu_\infty^2} \frac{1}{\rho_\infty^2} C_d R_e \quad (12)
\]

This indicates that the drag on a particular shape with length of 10 ft at 20 mph is the same as the drag on the same the drag on the same shape with a length of 1 ft at 200 mph if the fluid and temperature are unchanged. In the context of this project, the force on an \( \frac{1}{8} \) - scale car model at 200 mph is the same as force on the full-scale vehicle at 25 mph.
2.3 Literature Survey

In order to fully understand the problem, we broke down our background research into existing wind tunnel technology, arm/support methods, and wind tunnel measurement methods.

2.3.1 Existing Wind Tunnel Technology

A wind tunnel works by moving air through a tunnel around a stationary object in order to model what would happen if the object were moving through the air. Engineers test the disturbance of the air flow by injecting smoke into the tunnel and observing the disturbance. Depending on the size of the wind tunnel, anything from tennis balls to full-scale vehicles and airplanes can be tested.

There are two major methods of supporting objects in a wind tunnel. The first is by a vertical force balance that supports the majority of the weight of the vehicle. The second method is by having horizontal force balances that extend out from the side of the wind tunnel. These horizontal force balances are used more for the application of supporting the wheels of a vehicle in the test section.

Inserting a rolling road into the wind tunnel introduces a very large effect on the object being tested. A vehicle being tested in a wind tunnel with no rolling road will experience different air forces once a rolling road is inserted because of the increased airflow introduced underneath the vehicle. This increases the lift of the vehicle typically because the boundary layer effect is significantly decreased.

2.3.2 Existing Arm/Support Technology

In order to gain more of an understanding of arm movement, we looked for inspiration from other products on the market. One of products we found was the swiveling mechanism on adjustable desk lamps. The swiveling desk lamp that is shown in Figure 4 has a very simple locking mechanism. This locking mechanism would be very easy to incorporate into the wind tunnel design and is extremely reliable.

We also looked into telescoping and multi-axis positioning found in a camera jib in Figure 3. This design inspired a counterbalancing concept. Counterbalancing is explained in further detail in later sections.

Figure 3. Swivel Lamp Inspiration
Figure 4. Camera Jib Inspiration
2.3.3 Wind Tunnel Measurement Methods

A load cell is a transducer that produces an electric signal with a magnitude that is proportional to the force being measured. Types of load cells include strain gauge load cells, hydraulic load cells, and pneumatic load cells.

A strain gauge load cell directly measures how much a mechanical component deforms (the strain) as a change in electrical resistance, which can tell us the applied forces. A load cell usually consists of four strain gauges in a Wheatstone bridge configuration, but one strain gauge (quarter bridge) or two strain gauges (half bridge) are also available. Strain gauge load cells are the most common in industry and these load cells are very stiff, have good resonance values, and have long life cycles in application.

A hydraulic load cell uses a conventional piston and cylinder arrangement. The piston is placed in a thin elastic diaphragm so the piston does not actually come in contact with the load cell. When the load is applied on the piston, the movement of the piston and the diaphragm increases the oil pressure which in turn creates a change in the pressure on a Bourdon tube connected with the load cells. As this sensor has no electrical components, it is ideal for use in hazardous areas and outdoor applications.

A pneumatic load cell is designed to automatically regulate the balancing pressure within the cell. Air pressure is applied to one end of the diaphragm and it escapes through the nozzle placed at the bottom of the load cell. A pressure gauge is connected to the load cell to measure the pressure inside the cell. The deflection of the diaphragm affects the airflow through the nozzle as well as the pressure inside the chamber. The gauge is used to calculate the load on the device.

2.3.4 Formula SAE Specifications

One of this project’s main customers is the Cal Poly Formula SAE Club. Andrew, the test section designer and member of Cal Poly Formula SAE specified that we should design the wheel support for their 1/3 scale car model which translates to a 19in track width. This meant that the wheel support would accommodate a track width of this size as a maximum track width (any larger wall effects might become noticeable). A 1/3 scale car model wheel and tire (Jongbloed and Hoosier) weigh approximately 7 lb, with a tire outer diameter of 7in.
3. Design Development

The overall scope of this project is to provide a prototype wheel balance/support for the recently acquired rolling road in the Cal Poly wind tunnel.

3.1 Requirements

The objectives of this project were developed through a technique called Quality Function Deployment (QFD) in order to turn customer requirements into engineering specifications. This is achieved by quantifying the importance of various customer needs and then correlating them to engineering specifications. The QFD matrix can be found in Appendix A.

The first step in the QFD process is to put a weighted score on each customer requirement with a value of 1-5, with 5 being the most important and 1 the least. Each of these scores were given after deliberation within the team and research of competitor's facilities. In layman's terms, if our proof of concept misses any specification of a value 4 or 5, the final device is not useful to our sponsor. Outlined below are the findings from QFD.

The first requirement is for the support arm to be capable of holding a sufficient load. This was expressly stated from our sponsor to be extremely important because this reduces the wear of the rolling road. The rolling road is capable of moving at over 100 MPH, therefore holding the load up for long durations of time is essential for valid wind tunnel data and the longevity of the rolling road.

The next customer need is low maintenance. This wind tunnel is a facility that will be used by students who will be testing their designs almost every week out of the year. This facility is also used as an instructional facility by other professors. Having unreliable wheel supports in the wind tunnel would make the job of the students and professors extremely difficult and potentially invalid due to support failure. When maintenance is eventually needed, it should be easily accessible and should not be hard for someone with little mechanical knowledge to perform.

Next, a standard mount design is important in order to have industry be able to utilize this facility. We need to communicate to outside customers the attachment requirements to test devices in this facility in order for them to come to the facility ready to test. Lockable adjustments and automatically balancing supports are also essential in the final delivery of this senior project.

All the above customer needs were compared against engineering specifications determined to ensure each customer requirement had a corresponding engineering specification. The first specification is to be aerodynamically neutral. This isn't necessarily within the scope of this project, but is still essential in the eventual installation in the wind tunnel--this may come in later iterations of this project. The next comparison was the maximum testing duration that the wind tunnel will operate. The maximum duration that the wind tunnel would realistically be operating would be around two hours. We will design our support to be under operating conditions for 4 hours in order to build in a factor of safety and to accommodate longer testing periods than anticipated.

In the case of loading requirements, our customer did not specify minimum loading requirements. However, he did state that the wheel balance prototype should be designed for the aforementioned Cal Poly clubs and organizations as well as testing of an individual full scale car wheel.

Due to the small overall size of the wind tunnel, there are tight parameters to fit into for maximum dimensions in the X, Y, and Z directions. The cross section of the tunnel is 46” wide (Y direction)
and 34" tall (Z-direction). This means the maximum values cannot exceed these values. The X-direction maximum dimension are less constraining at a length of ~10'.

In order to prevent too much movement from vibrations, wind resistance, and load forces, the system needs to be counterbalanced and damped. Some of the vibrations will be damped through the gear systems and mounting designs. This is covered in more detail in section 3.3.

The final specification that we determined for our balance was the required motor specifications to move the arm. Once the preliminary design of the system has been fully determined, determination of motor specifications will come after. However, all motion concepts have the ability to be human-powered or electrically powered. The beginning designs focus on human-powered designs which can incorporate electrical power in later iterations.

### 3.2 Additional Design Considerations

Several design considerations that were not quantifiable are outlined below. These specifications are no less important and need to be considered in all design decisions.

**This support must be easy to use:** The final design must be intuitive and simple to use. Ideally, the design should only require one person to operate.

**This support must be easy to service:** The design should not require regular maintenance more than quarterly. The design should last for several years without needing any replacements.

**This support must be affordable to design, test, and build:** While this seems like something that would fit into a standard QFD, this is an additional design requirement because we are only building a prototype. We understand that a prototype is going to be more expensive and the final design report will provide recommendations for keeping the cost down when building four wheel supports.

#### 3.2.1 Defining Specifications

The test section has dimensions of 46" by 34", which results in a cross-sectional area of 1564 in². Following the rule of thumb that the blockage area should be about 5% of the tunnel area\(^1\), the maximum cross-sectional area of the test model should be 78 in². At the maximum speed of 100 mph and assuming a drag coefficient of 1.0, this gives a maximum anticipated drag force of 13.6 lb.\(^r\).
We used the 5% to 10% blockage area calculations (shown in appendix B) of the test section to show the minimum and maximum allowable blockage areas of a scale model car. The minimum area is represented in our model as a green rectangle and the maximum area is represented as a blue rectangle.

Next, we took the base target zone dimensions and researched scale model RC cars that would fit into this model area. We looked at RC cars with slick, run-flat or on-road tires at 1/4, 1/5, 1/6, 1/8, 1/10 and 1/18 scales online to get an idea of how large they are.

After research we found that 1/8 scale RC cars fit best in our wind tunnel. Not only do RC cars at that scale fit in the model zone, they fit closely with the 5% ideal case.

Although a 1/8 scale model is 1/8 the size of a full size car, it is only approximately 1/5th of the size of the SAE car. This means that if the SAE team wanted to build a scale model of their car for use in this wind tunnel, they would have to scale it down to 1/5 of the normal size.
Formal engineering requirements, which have all been outlined in the QFD in appendix A, are shown in Table 1 and 2.

Calculations were developed using three different scenarios. The first scenario is assuming two maximum sized wheels of a scaled car are being supported by each support arm. The second scenario is assuming a single SAE wheel is being supported. The third and final scenario is assuming that a single full-sized Tesla Model S wheel is being supported. For each of these scenarios, the max loading condition in the X, Y, and Z direction were all calculated. After final conversations with the client, it was decided that designing for a full size Tesla wheel is beyond the scope of this project. Moreover, this project is serving to test the feasibility of this type of mechanism in the space.
<table>
<thead>
<tr>
<th>Spec. #</th>
<th>Parameter Description</th>
<th>Requirement or Target (units)</th>
<th>Tolerance</th>
<th>Risk</th>
<th>Compliance</th>
</tr>
</thead>
</table>
| 1       | Vertical (Z-direction) Load  
*Scale Model  
**Assuming SAE car wheel  
***Assuming Tesla Wheel | 1.15 lbs.*  
17.5 lbs.**  
90 lbs.*** | +/- .5 lbs.*  
+/ - 5 lbs.**  
+/ - 10 lbs.*** | H | A, T |
| 2       | Transverse (Y-direction) Load  
*Scale Model  
**Assuming SAE car wheel  
***Assuming Tesla Wheel | 1.15 lbs.*  
17.5 lbs.**  
90 lbs.** | +/- .5 lbs.*  
+/ - 5 lbs.**  
+/ - 10 lbs.*** | H | A, T |
| 3       | Longitudinal (X-direction) Load  
*Scale Model  
**Assuming SAE car wheel  
***Assuming Tesla wheel | 4.65 lbs.*  
28 lbs.**  
45 lbs.*** | +/- 1 lbs.*  
+/ - 5 lbs.**  
+/ - 10 lbs.*** | H | A, T |
| 4       | End of Arm Position | 0.1 in | +/- 0.001 in | M | T |
| 5       | Belt Pressure | 1.8 N/mm² = 13.0 PSI | Min | H | T, I |

High and medium risk specifications (denoted by an H and M respectively) are specifications, which are essential in meeting the overall objective of the project and are anticipated to be difficult to accomplish. A, T, S, and I all denote a method of verification of compliance via Analysis, Testing, Similar Existing Designs, and Inspection respectively. All tabulated values are based on estimated static drag forces with no factors of safety included.

Note: for scale models, loads are for two wheels on one side of a car on each support arm.

<table>
<thead>
<tr>
<th>Spec. #</th>
<th>Parameter Description</th>
<th>Requirement or Target (units)</th>
<th>Tolerance</th>
<th>Risk</th>
<th>Compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>Vertical Adjustability</td>
<td>15 in.</td>
<td>+/- 1 in.</td>
<td>L</td>
<td>T, I</td>
</tr>
<tr>
<td>7</td>
<td>Lateral Adjustability</td>
<td>24 in.</td>
<td>+/- 1 in.</td>
<td>L</td>
<td>T, I</td>
</tr>
<tr>
<td>8</td>
<td>Longitudinal Adjustability</td>
<td>48 in.</td>
<td>+/- 1 in.</td>
<td>L</td>
<td>T, I</td>
</tr>
</tbody>
</table>

Low risk specifications (denoted by an L) are specifications, which are not essential in meeting the overall objective of the project but are still important in the overall success of the device. A, T, S, and I all denote a method of verification of compliance via Analysis, Testing, Similar Existing Designs, and Inspection respectively.
3.3 Preliminary Design Development

In order to find three solutions that fit the needs of our customer, our team held several ideation sessions. We split ideation into three different sessions. The first session was brainstorming where we came up with various methods of powering the system and moving the system (via linkages, gears, etc.). We began this exercise by writing down ideas on sticky notes and placing them on the wall. Once all the ideas were on the wall, we each selected power methods and movement methods to further refine and develop. The next ideation session we had was called 3-3-6 where three people draw three picture each for six minutes. The final ideation method we employed was brainwriting where all members sat in a circle and after two minutes of either drawing or writing on a piece of paper, we stopped and passed our paper to the person next to us and repeated this process until our original sheet was back in our hands. This process allowed us to all build on one another’s ideas.

3.3.1 Concepts

After successfully completing all brainstorming sessions, we have narrowed our ideas down to several types of motion in the X, Y, and Z directions along with standard mount design and vibrational damping concepts. This section will go into detail about each concept.

X Direction (longitudinal) Motion:

We narrowed longitudinal motion down to three basic concepts of rail translation, a power-screw assembly, and a swivel arm.

The sliding and locking rail translation concept shown in Figure 5 below includes a locking mechanism along the rail in order to secure the location. This design is very intuitive, simple, and low maintenance. The drawback to this system however is that there is a finite amount of locations for the sliding mechanism to lock. The force acting on this design acts straight down through the pillar and against the pin holding the structure in place. The size of the pin can be designed to withstand very large forces. Overall, this is the most robust design in the longitudinal direction.

![Figure 7. Sliding and Locking Rail Translation](image-url)
The Power Screw and rail assembly concept shown in Figure 6 includes a power screw that runs along the inside of the bottom rail that, when rotated, pushes the assembly parallel to the side of the rolling road. This design offers fine incremental changes and inherently is self-locking, which is an advantage over other mechanisms. This idea however isn't low maintenance and works best with electrical power in order to drive the power screw (although hand-powered cranking is theoretically possible).

![Power Screw and Rail Concept](image1)

Figure 8. Power Screw and Rail Concept

The Swivel Arm concept shown in Figure 7 includes turn buckles which actuate the swivel arm back and forth. This provides a wide range in the longitudinal direction without having to move the carriage/frame at all. This design is simple, but may not provide the best structural support at the pin location. The shear stress introduced into this joint is large. There is also a problem with effectively designing it to accommodate a range in wheel locations and two scale car wheels at once.

![Swivel Arm with Turn Buckle Concept](image2)

Figure 9. Swivel Arm with Turn Buckle Concept

**Y Direction (lateral) Motion:**

Y direction motion has been narrowed down to an extendable screw or modular telescoping concept.
The Threaded Screw concept shown in Figure 8 has a large threaded rod that, when rotated, moves the standard mount at the end of the rod along the y-axis. This motion design, while providing very fine adjustment and being very simple suffers because we do not have the room in the wind tunnel to fit the back end of the power screw. Our client would prefer if we do not cut any holes in the side of the wind tunnel. This design also introduces stress concentration problems when the rod is experiencing deflection under the weight of the load.

![Extendable Screw](image)

Figure 10. Extendable Threaded Screw Concept

The idea that we kept coming back to in the lateral direction was a modular telescoping mechanism. This idea uses multiple smaller power screws to expand and retract a telescoping support structure for the wheels. This idea is rather complicated to build, but if designed correctly it should hold the load of the wheels while being able to expand and retract a great range. If we have trouble selecting mechanical components that are strong and robust enough to support a large load while still being able to fit to small-scale model tires then we can design for the whole telescoping mechanism to be modular so that it can be replaced with a more robust arm to support large wheels. The telescoping motion can be either electrically powered mechanism or a human powered system where the user sets the distance manually. Our initial prototype will most likely be mechanically powered which a future design suggestion of adding the capability of electric power. Figure 9 shows the basic design concept of telescoping motion.

![Modular Telescoping](image)

Figure 11. Modular Telescoping Mechanism
**Z Direction (vertical) Motion:**

Motion in the vertical plane was narrowed down to a power screw assembly, a counter weight and a hydraulic actuator.

The Power Screw Assembly concept includes a power screw moving in the vertical axis. As the power screw rotates, the carriage moves up and down which moves the arm in the z-direction. This system is very robust, would be able to withstand large perturbations to the system and would be self-locking. The drawback to this system is the initial investment in a motor and a power screw for the system. Figure 10 shows the concept.

![Power Screw](image)

**Figure 12. Vertical Power Screw Concept**

The Analog Counter Balance concept shown in Figure 11 is a very simple system in theory. A weight that is threaded moves back and forth on the opposite side of the wheel. As the weight moves further from the fulcrum, the induced moment lifts the tire off the rolling road. The problem with this design is both the space required and the potentially very large and heavy weights needed in order to counter balance.
The Hydraulically Actuated Counter Balance concept shown in Figure 12 is one of the most expensive designs in this report. The large cost comes from the hydraulic system. While this system is able to provide very fine actuation with very large pressures, the system is not feasible for the small space and small budget provided. It also has the potential to get very messy in an otherwise clean wind tunnel lab.

The ideas presented in figures 5-13 conclude the feasible concepts that came from ideation for this project. Section 5.3 discussed how the ideas were narrowed down to come up with our final design concept. Next, balancing and damping of the system is a large concern along with the standard mount design that our customer asked for. A torsional damper is a design concept.

**Damper System Design:**

With so many dynamic forces on our machine in the wind tunnel, it is inevitable that it will encounter significant vibration. In order to keep the wheels the machine is holding barely touching the belt and to protect the machine from destroying itself through vibration, it will require some kind of dampening system.
The Torsional Damper concept is shown in Figure 12. This concept is based on the idea of using torsional dampening grommets on the part of the machine that rotates around the X axis. This design would sufficiently dampen the max shock loads in the X, Y and Z directions of the wind tunnel, but would most likely require an additional actuator on the back side of the fulcrum in order to keep the desired amount of tension in the arm in order to keep the wheel barely touching the belt.

![Torsional Damper Concept](image1)

Figure 15. Torsional Damper Concept

The Traditional Damper concept is shown in Figure 13. This is an alternative to the torsion-dampening concept and it incorporates more traditional grommets and linear springs to dampen upward and downward vibration. Dampers that do well in shear force could also be incorporated in order to deal with any other kind of vibrational forces from the lateral and longitudinal directions. The key difference with this design is that it would not require any kind of active tensioning assuming that the overall structure can sufficiently support the static load of the scale models or wheels and any additional dynamic loads.

![Traditional Damper Concept](image2)

Figure 16. Traditional Damper Concept
Standard Mount Design:

A key feature of the design will be to have a standard mounting feature from which individuals or teams looking to use the tunnel can design a fixture to hold the wheels they are looking to test. This standard mount will also be the base from which our customer could potentially attach load cells and/or an independent motor control system. Our one design idea is to have a flanged casting or machined piece with the threaded holes to accommodate either four lug nuts or a singular lug nut.

Figure 17. Standard Mount Design Concept

At this point in the process, we will also complete motion and DOF studies to ensure that our design can complete its function without any collisions or obstructions. This is where we will do our initial engineering analysis in order to determine the sizes and types of specific parts. If need be, we will perform Computer-Aided Engineering (CAE) analysis and any material or specimen testing at this step, but we will try to keep the use of these tools to a minimum in order to keep the design process moving efficiently. We may also make prototypes of individual parts or subassemblies by utilizing the campus Rapid Prototyping capabilities at this point in the design process in order to improve communication of design ideas among team members and outside advisors and sponsors. We will adjust our CAD design based on the results of our analyses if necessary.

Our last major design step before the final build will be to make a smaller scale prototype (most likely out of laser cut wood panels, dowel rods and 3D printed parts) that can be placed inside the test section in order to spot any last design issues. Any lingering design issues will be fixed in our final design and then we will build and test the final machine in the second half of senior project.

3.3.2 Overall Design

In order to help supplement engineering analysis in making our decision, we used a Pugh Matrix in each direction of motion. The criteria that we used in our Pugh Matrix was holds sufficient load, low maintenance, easy to use, easily movable, flexible size/location, manufacturability, and cost. Holds sufficient load means that the concept is able to support our maximum conceived load. Low maintenance means that the concept does not need to be serviced and should not experience failure often. Easy to use means that the system is simple enough for an operator without any previous knowledge of the system to use. Easily movable means the operator does not have to exert a large amount of force to move the carriage or arm/support. Flexible size/location range means that the end of the support is able to reach a large area within the wind tunnel. Manufacturability means that the concept is easy to build and manufacture. Finally, cost means that the concept is not more expensive than the datum and will fit within the budget provided by our customer.
Table 3. X-Direction Pugh Matrix

<table>
<thead>
<tr>
<th>X Direction</th>
<th>Weight</th>
<th>Power Screw and Rail (datum)</th>
<th>Sliding and Locking Rail</th>
<th>Swivel Arm w/ turnbuckles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holds Sufficient Load</td>
<td>5</td>
<td>0</td>
<td>0</td>
<td>-1</td>
</tr>
<tr>
<td>Low Maintenance</td>
<td>5</td>
<td>0</td>
<td>1</td>
<td>-1</td>
</tr>
<tr>
<td>Easy to use</td>
<td>3</td>
<td>0</td>
<td>-1</td>
<td>-1</td>
</tr>
<tr>
<td>Easily Movable</td>
<td>3</td>
<td>0</td>
<td>-1</td>
<td>-1</td>
</tr>
<tr>
<td>Flexible Range</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>-1</td>
</tr>
<tr>
<td>Manufacturability</td>
<td>4</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Cost</td>
<td>4</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 4. Y-Direction Pugh Matrix

<table>
<thead>
<tr>
<th>Y Direction</th>
<th>Weight</th>
<th>Modular Telescoping (datum)</th>
<th>Single Variable Telescoping</th>
<th>Extendable Threaded Screw</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holds Sufficient Load</td>
<td>5</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Low Maintenance</td>
<td>5</td>
<td>0</td>
<td>-1</td>
<td>1</td>
</tr>
<tr>
<td>Easy to use</td>
<td>3</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Easily Movable</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>-1</td>
</tr>
<tr>
<td>Flexible Range</td>
<td>3</td>
<td>0</td>
<td>-1</td>
<td>0</td>
</tr>
<tr>
<td>Manufacturability</td>
<td>4</td>
<td>0</td>
<td>1</td>
<td>-1</td>
</tr>
<tr>
<td>Cost</td>
<td>4</td>
<td>0</td>
<td>-1</td>
<td>-3</td>
</tr>
<tr>
<td>Z Direction</td>
<td>Weight</td>
<td>Power Screw assembly (datum)</td>
<td>Weighted Counter balance</td>
<td>Hydraulic Actuator</td>
</tr>
<tr>
<td>-----------------------------</td>
<td>--------</td>
<td>------------------------------</td>
<td>--------------------------</td>
<td>-------------------</td>
</tr>
<tr>
<td>Holds Sufficient Load</td>
<td>5</td>
<td>0</td>
<td>-1</td>
<td>0</td>
</tr>
<tr>
<td>Low Maintenance</td>
<td>5</td>
<td>0</td>
<td>1</td>
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<td>-1</td>
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</tr>
<tr>
<td>Easily Movable</td>
<td>3</td>
<td>0</td>
<td>-1</td>
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<tr>
<td>Manufacturability</td>
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<td>1</td>
<td>-1</td>
</tr>
<tr>
<td>Cost</td>
<td>4</td>
<td>0</td>
<td>-1</td>
<td>-1</td>
</tr>
</tbody>
</table>

As seen from the Pugh Matrix above, the top concepts were the sliding and locking mechanism in the X-direction, modular telescoping concept in the Y-direction, and the power screw assembly in the Z-direction. Figure 18 shows the amalgamation of the top idea concepts.

**Figure 18. Final Pugh Matrix Design Concept**

The final design shown in the figure above contains all the top ideas from the Pugh matrix. Along the vertical axis, a power screw assembly controls the height of the carriage. This assembly offers very fine control of the end of the arm because we can mitigate backlash in the power screw by ordering a fine thread variety. A sliding and locking mechanism is going to be used for the longitudinal direction control of the apparatus.
Along the y-axis we have a modular telescoping concept where we have a telescoping mechanism that supports heavy loads (>10 lbs) and a telescoping mechanism that supports light loads (<10 lbs). An additional modular telescoping mechanism could be designed to support super heavy loads (>70 lbs) such as a Tesla wheel if necessary. This delineation between light and heavy loads comes from the sensitivity needed from different load cells meaning that if we are testing smaller wheels, we will likely be looking for smaller vibrations, deflections, and a drop in the overall height of the telescoping arm so we need a more sensitive load cell installed. The converse is true for large loads with larger movements where we would be worried that a small load cell would not pick up all of the movement. This design-for-testing layout and procedure is further explained in the testing procedure section below. There is also the issue that it could be hard to design mechanical components that sufficiently hold a large load such as an SAE wheel or a Tesla wheel and are designed to not fail from fatiguing at that particular load while still fitting to smaller RC car tires. If this issue comes up, then we will design for the lateral motion arms to be replaceable depending on the range of loads it is expecting to see for that particular test. Even if the system does require some modular substructures, we will design for it to share as many components as possible in order to keep the costs and amount of manufacturing to a minimum.

Our number one design also features a modular mount design at the end that can accommodate various types of bearings as well as motors for independent speed control and load cells for improved testing capabilities.

3.3.3 Preliminary Design Considerations

The manufacturing process and cost constraints are two of the largest design constraints that limit the scope of this project. The final design concept developed in section 6.2 is a comprehensive design that includes motorized translation methods in two directions. Even though we would like to have the entire prototype powered by electric motors, an electric powered screw assembly in the longitudinal direction is too expensive and potentially too much work to incorporate into this design. In order to make this project more realistic, we decided to stick with the sliding and locking mechanism that led in the X-direction Pugh matrix. This design is also very space efficient and is very user friendly. If we have the time and money after the rest of the prototype is built and working properly, we could go back and upgrade the longitudinal direction mechanism to be motor and power screw powered.

The power screw assembly in the vertical direction is remaining in the final design because it offers the best method of translation for the user. While this is not the easiest method for us to implement, there is no other final design concept in the vertical direction that has fine incremental changes, automatic locking feature, and is user friendly. This system is not necessarily low maintenance either. However, we feel that the negative utility resulting from maintenance and cost is outweighed by the fine incremental changes, locking features, and user friendliness.

The telescoping mechanism in the transverse z-direction will be simplified as well. The original concept had the whole system being mechanized through an electric motor. We have removed the motor element and have inserted a simple cranking mechanism utilizing a worm gear and a worm mounted on a power screw shaft. This will simplify the overall telescoping mechanism design and make it easier to control and maintain as well as lower the cost.

Another concern is that there might be people or groups who want to use our prototype in the future to hold up wheels with a camber. If we have time, we would like to design in a modular mechanism
with small linear actuators that control a standard mount that are on a hinge in order to better mate to a cambered surface. If we do not have time to design a device like this, then the mounts that customers design to actually hold their wheels and mate to our standard mount, will have to accommodate for the camber of the wheels they are testing.

One last concern is that our prototype is not very aerodynamic, which could have a negative effect on the results of the vehicles or wheels we are testing. To negate this, we are planning on designing some kind of housing (most likely out of sheet metal) to reduce the amount of turbulence created by the subassemblies. This will probably include a neutral airfoil housing for the lateral motion mechanism that telescopes in and out with the mechanism. The housing for the other two subassemblies will also be aerodynamically neutral as possible while still allowing for the mechanisms to move freely.

### 3.3.4 Preliminary Solid Model

Below is our first Solid Works model of our preliminary idea. Most of the detailed components have been left out of this model, including the vertical carriage assembly, lateral assembly gear housing and any sheet metal for housing our prototype and reducing any negative effects it has on the airflow. However, all of the key mechanism components have been included in order to demonstrate how the prototype will move and mount to the wheels being tested. In this particular model, the module holding the wheel is designed for a single SAE wheel.

![Figure 19. Preliminary Solid Model](image-url)
4. Critical Design

The solution shown in section 3.3.4 had significant flaws in the overall design. One of the main flaws was installing a rail in the upper corner of the wind tunnel. One of the constraints that the client had was not drilling into the acrylic surroundings of the wind tunnel. The design would require significant amount of supports, which would require drilling through the acrylic to attach to the t-slot support system on the exterior. Next, motorized lead screws are either prohibitively expensive or not available when in the height of the wind tunnel.

The new design eliminates the top rail system and relies on a single rail system in the bottom corner of the tunnel. A single I-beam supports this new system with t-slot guide rails running along the top. A carriage system slides back and forth on the t-slots with a tower built on top. The tower has a cantilevered telescoping system, which has a standardized mount at the end to attach to various objects. Overall, the system provides almost a foot of travel in the z-direction, 6 inches in the y-direction, and 6 feet in the x direction. This final design is shown in the figures below.

Figure 20. Orthogonal Solid Works Rendering
Figure 21. Assembly looking down x-axis (collapsed telescope)

Figure 22. Assembly looking down x-axis (expanded telescope)
4.1 Functional Description of Design

The overall final design description will be broken down into the three directions (x, y, and z) and the mechanics and manufacturing analysis of each system will be discussed. Additionally, the electrical components will be discussed in detail.

![Figure 23. View down wind tunnel intake](image)

4.1.1 X-Direction Design

The main supporting feature is the A316 steel extruded I-beam. When originally iterating this design, we kept running into deflection problems because of the moment load due to the cantilevered wheel. We tried square tubing, solid steel beams, and composite material beams. In the end, steel I-beams provided the best structural support. Overall, the beam experiences very little deflection. With our most conservative calculations putting the whole mass as a point load at the center of the beam along with a point moment load, the beam only deflects 0.12 inches.
The I-beam is secured at the front and rear of the wind tunnel. Both ends of the beam are secured into t-slot cross members.

The original calculation we performed for the deflection of the horizontal support were with a basic beam deflection equation shown here:

\[ y = \frac{L^3 F}{192EI} \]  

(4.1)
Equation 4.1 assumes a worst-case scenario similar to our computer model with a 200lb vertical point load in the center of the beam. Here, I is the second moment area of inertia, E is the modulus of elasticity, and L is the overall beam length. Y is the amount of vertical deflection experienced in the middle of the beam. This calculation showed a deflection of 0.12” in total. This is significantly more than the model showed because this calculation does not accurately represent how the beam is fastened. However, mounted on top of the steel I-beam are aluminum t-slots, which will help provide additional support to counteract deflection. A cross-section of the t-slot mounted on the I-beam is shown in the figure below.
The above figure shows a cross-sectional view of the horizontal support system. At the base, we have an aluminum plate, which will be in direct contact with aluminum t-slots. There is a 0.0625” gasket between the steel I-beam and the aluminum plate to prevent corrosion (discussed further in section 4.4 – Material Selection).

Figure 28. X-Direction Mounting Method

Figure 29. Sliding Carriage Design

The figure above shows the mechanism that facilitates the translation in the x-direction. This carriage slides back and forth along t-slots. This carriage has two hand brake mechanisms that can be twisted.
when the user wants to lock the carriage in position. Each of these hand brakes is able to hold 60lbs of force, therefore, combined the brakes can withstand 120lb of force. The hand brake works by a plastic screw mechanism that rotates into the metal t-slot and prevents movement through friction. The carriage, brakes, and t-slots are provided by t-slots.com. The original concept did not include a hand braking mechanism and we were trying to design our own method of holding the carriage. However, for ease of manufacturing and design, we decided to implement this system that is already available off the shelf and for less than any system we could have designed and built ourselves.

To transfer electricity to the carriage there needs to be a safety feature that safely holds the wiring as the mechanism moves back and forth. Shown below is the small cable carrier supplied by McMaster-Carr.

![Small cable carrier](image)

Figure 30. Small cable carrier
4.1.2 Z-Direction Design

The z-direction mechanism can be seen in the figure below. The actuation depends on structural support from both a vertically oriented I-beam and two solid steel rods, which guide linear bearings up and down.

![Figure 31. Z-Direction Rendering](image)

The figure above shows the overall Z-direction actuation method for the assembly. The power for the movement is supplied through the motorized lead screw that is mounted on the rear side of the I-beam (not shown in figure). The details of the motor and the control system will be discussed in detail in section 4.1.4.

The initial calculations performed on the z-direction supports were to find the total angular deflection, compression, and Euler buckling. These calculations were done using the following correlations:

\[
P_{CR} = \frac{\pi^2 EI}{I^2}
\]

(4.2)

Where \( P_{CR} \) is the critical pressure that can be applied to the top of the beam without the beam experiencing first order buckling.

\[
P_{all} = \frac{P_{CR}}{F.S.}
\]

(4.3)

\[
\sigma_{all} = \frac{P_{all}}{A}
\]

(4.4)

\[
y_{max} = e[\sec\left(\frac{\pi}{2} \sqrt{\frac{P}{P_{CR}}}\right) - 1]
\]

(4.5)

\( y_{max} \) is the maximum amount of vertical compression that the I-beam will experience under extreme loading conditions. The maximum amount of compression seen would be 0.003 inches. This is negligible when compared to other stresses and deflections in the system.
\[
\sigma_{\text{max}} = \frac{P}{A}[1 + \frac{4c}{r^2}\sec\left(\frac{\pi}{2} \sqrt{\frac{P}{P_{CR}}}\right)]
\] (4.6)

\(\sigma_{\text{max}}\) is the maximum amount of stress the I—beam can see before failure (meaning deflecting beyond the elastic region). \(\sigma_{\text{max}}\) is found to be 530 psi which is nearly 5 times more that the most extreme case this beam would see.

The overall compression seen in the vertical I-beam tower was assumed negligible based off earlier calculations. The deflection analysis via Solid Works supports this assumption.

Buckling is a more valid concern in this case. However, since the force on the I-beam is so small, Euler buckling is also negligible. As seen in the above figure, the maximum deflection seen in the extreme loading condition would be 0.03 inches. However, since the I-beam does not experience eccentric loading in the conditions we tested, we are confident that the beam will experience significantly less buckling characteristics than shown.

In addition to the I-beam support in the vertical direction, two solid steel rods help guide the telescoping mechanism as it actuates up and down. Each of these rails carry less than half the weight regularly and even in extreme cases will never see more than 50% of the weight of the telescoping mechanism and tire. However, for calculations, we assumed failure of one rod and analyzed the rods carrying the full weight of the telescoping mechanism and tire. In this conservative loading calculation, it can be seen in the figure below there is a maximum of 0.055 inches deflection.
When the loading case presented is less extreme and a factor of safety of 2 is used for load sizes, the deflection for a more complex loading condition is less than the previous model. In the case of moment loading and compression loading combined, there is a total deflection of only 0.050 inches.

Precisely controlling and changing the z-position of the wheel is the main thesis of this project and has been carefully approached. The ideation process for this z-position control can be seen throughout Section 4.1.2. The final solution was to use the Thomson Linear NEMA23 motor with a 50-0100 lead screw and an XC Advanced Flanged Anti-Backlash nut shown in Figure 37 to control the telescoping assembly’s motion. The screw has a diameter of 0.5 inches and positions the telescoping mechanism up or down 0.1 inches per revolution (0.005in per step). The motor controller will be described in Section 4.1.4. In order to make the system as intuitive as possible to the user, we are going have control through simple up and down buttons. A stepper motor has several appealing qualities that made it the appropriate choice for this application. Firstly, they can achieve very large torque at low speeds with the torque capacity decreasing with increasing speed. The lead screw will always be running at relatively low speeds (<1 in/s). Secondly, great precision can be achieved in open loop (no feedback). This is due to the nature of a stepper motor where the energized phases essentially lock the rotor in a fixed position [9]. The size of this motor, lead screw, and lead screw flange hinged upon our customer’s requirements for this wheel support and the weight of the telescoping mechanism (27.24 lb). However, the total weight also includes the weight of the test model wheel and tire. The total weight was conservatively estimated to be 45 lb. The maximum thrust produced by the motorized lead screw was quoted to be 200 lb, which would be more than sufficient to raise the telescoping assembly and wheel. Back driving of the motor is not possible for two reasons: the flanged nut is self-locking because
of the fasteners used to mount it and the telescoping assembly is fixed about the lead screw’s rotational axis.

![Thomson Linear Motor and Lead Screw](image)

**Figure 36. Thomson Linear Motor and Lead Screw**

![NACA Air Foil Cover for Vertical Assembly](image)

**Figure 37. NACA Air Foil Cover for Vertical Assembly**
The final addition to the Z-direction assembly is the National Advisory Committee for Aeronautics (NACA) airfoil based on the equation for a symmetrical 4-digit NACA airfoil,

\[ y_t = 5tc \left[ 0.2969 \frac{x}{c} + (-0.1260) \left( \frac{x}{c} \right) + (-0.3516) \left( \frac{x}{c} \right)^2 + 0.2843 \left( \frac{x}{c} \right)^3 + (-0.1015) \left( \frac{x}{c} \right)^4 \right], \tag{4.7} \]

where:

- \( c \) is the chord length,
- \( x \) is the position along the chord from 0 to \( c \),
- \( y_t \) is the half thickness at a given value of \( x \) (centerline to surface), and
- \( t \) is the maximum thickness as a fraction of the chord.

This airfoil is necessary to minimize the disruption to the air caused by the tower. Disruption to the airflow is detrimental because it can invalidate the information about the object being tested. One of the best ways to decrease the disruption is to extend the tail of the foil because this helps slow down the deceleration of the air over the widest part of the foil and prevents flow separation after the maximum thickness of the foil. However, since there is limited space in the wind tunnel, the airfoils in this design are only 28 inches long. This restriction in length was due to the eventuality that there will be tower’s adjacent to one another to cover a car’s wheel base. The airfoil design will be a sleeve that slides over the top of the tower and can be easily exchanged for another one that has either a shorter or a longer tail based on the specific application. In order to keep the airfoil as light as possible,
it will be made from balsa wood and a lightweight skin called MonoKote, which is a lightweight adhesive shrink skin.

4.1.3 Y-Direction Design

Figure 40. Orthogonal View of Y-Direction Design

The final design in the y-direction is for a telescoping mechanism that overall travels 6 inches. This accommodates a scale car between 20 inches wide and 8 inches wide. The Cal Poly Formula SAE student car has a 57in track width which exceeds the test section width by 9 inches. For testing feasibility, the 1/3 scale model’s 19in track width was used in the design. This is the largest scale model that the wind tunnel can accommodate.

Figure 41. Telescoping Deflection Analysis Using Solid Works Simulation
The telescoping mechanism shown above is supplied by tslots.com and comes in a 2”, 1.75”, and 1.5” diameter sections. The combined deflection of the telescoping mechanism is 0.001 inches when loaded with a 45lb vertical force. This deflection analysis used in Solid Works is a very rudimentary analysis due to the programs inability to recognize a feature that is not securely fastened. The telescoping mechanism is only supported through friction and interference, which is an analysis Solid Works, cannot perform. In order to overcome this, the securely fastened each tube at the base. The deflection still compounds on each member as the load transfers towards the tower, just not to the scale it would normally with proper analysis. In order to find the overall deflection, we used Castiglione’s method:

\[
\delta = \frac{1}{E} \left[ \int_{0}^{l_1} \frac{1}{I_1} (-Fx)(-x) \, dx + \int_{l_1}^{2l_1} \frac{1}{I_2} (-Fx)(-x) \, dx + \int_{2l_1}^{3l_1} \frac{1}{I_3} (-Fx)(-x) \, dx \right]
\] (4.7)

Where \( \delta \) is the deflection at the end where the mechanism is loaded, \( E \) is the elastic modulus, \( I \) is the second moment area of inertia, \( F \) is the loading force, and \( x \) is the distance from the left where we would like to find the deflection amount. The deflection for a single support is 0.038” which with two supports comes to be 0.019” overall.

In addition to the telescoping design, there is a standard mount design at the end of the mechanism. This standard mount design is a steel-machined support that is designed for the user to attach their own attachment device to. The mechanism has a four-bolt attachment pattern in order to accommodate a wide range of wheels and devices.

![Figure 42. Orthogonal Standard Mount View](image)
As shown in the figure above, there is a double row angular contact bearing that is press fit onto the shaft in order to support this standard mount. We chose a double row angular contact bearing in order to prevent any sort of thrust loads from moving the shaft. Since this device could potentially be used to support something other than a shaft if the user chooses, this bearing must be able to withstand potential thrust loads. The bearing has a dynamic radial load capacity of 4855lb and a static radial load capacity of 3210lb as well as a maximum speed of 12,000 RPM. The maximum speed this shaft will see is 4,500 RPM and a static load of at most 200 lbs.

Figure 43. Orthogonal View of Standard Mount Assembly

Figure 44. Lead Screw Assembly from x-axis
As shown in the figure below, both guide housings are the same cross-sectional size. We designed the system this way in order to reduce the amount of parts we had to order. We will simply be able to order a single size of 2x6 in. square tubing, cut to size, and machine the necessary components.

![Telescoping Mechanism and Guide Housings](image)

**Figure 45. Telescoping Mechanism and Guide Housings**

The above figure is the final design for the actuation in the y-direction. This whole mount is designed from lightweight aluminum in order to prevent cantilevered mass from the I-beam base. All aluminum parts that are in contact with steel are being shielded with a gasket to prevent excessive corrosion.

### 4.1.4 Control System Design

To achieve the precise control in the z-axis translation, we will be using the motorized lead screw. Therefore, the motorized lead screw requires a controller to dictate the lead screw’s linear velocity. This will be accomplished with the circuit diagram shown in Figure 44.
Figure 46. Stepper motor control circuit

The notable circuit elements have been highlighted in green (z-limit switches) and blue (position potentiometers). The position potentiometers allow us to monitor the absolute position of the wheel in all three directions. These position potentiometers, also known as string pots, will be purchased as a kit from Andy Mark (see Appendix for part number). The kit includes a 10 turn 5k angular potentiometer and a coil spring from a disassembled retractable card holder (along with some screws). A picture of the completed string pot can be seen in Figure 45.
The spool (upper right white piece) will hold the length of wire tensioned by the coil spring wound on the inside of the spool. The ten turns of the potentiometer in conjunction with the circumference of the spool will allow for a linear measurement range of 27 in. The reference voltage for the potentiometers will be supplied by the Uno’s +5V pin. The analog voltage from the potentiometer is read by the Uno’s digital I/O pins with PWM capability.

Highlighted in green in Figure 44, the limit switch circuitry was intended as the home detector to determine the exact z position of the telescoping assembly by touching off on both the top and bottom limit switches to zero its position. The addition of the string pots removed the need for a home detector because the string pot will always represent the absolute position of the telescoping assembly regardless of circuit power state. Rather than remove the limit switches from the design, they have been repurposed as a safety precaution to avoid the motor stalling at the top and bottom range of the lead screw’s travel from misuse.

The microcontroller we will be using is the ATmega328P on the Arduino Uno rev3. As a simple entry level board, the Arduino Uno contains everything necessary to support the 328P. The benefit to using Arduino is having access to its community and thorough libraries. The Uno board can be powered via USB port or with an external power supply. We intend to be using the USB port because the user will be prompted through the serial monitor built-in to Arduino IDE. To drive the stepper motor, we chose the Adafruit motor shield v2 for the simplicity and ease of use. The Adafruit motor shield has a dedicated PWM driver chip to replace the data latch and PWM pin use. The TB6612 MOSFETs allow for 1.2 A per channel and 3 A peak current. One motor shield will power four DC motors or 2
stepper motors. The motor shield design is completely stackable with stack headers. The power supply we chose was a 12V, 25A, 300W power supply. This power supply will supply the motor shield which receives an input voltage of 5-12V. Since the peak current draw from the stepper motor is 3A, the power supply had to be rated for the peak current draw of four motors or a current rating of greater than 12A. Having a 25A power supply allows for future project expansion.

4.2 Analysis Findings

This section will discuss specific analysis of various parts of the support. Specifically, we will look into the effects of the standard mount and acrylic shield on the boundary layer and streamline velocity. Additionally, we will look in detail into effective stiffnesses of bolts and the analysis of the steel plate at the base of the mechanism.

![Velocity Contour at max wind tunnel speed](image1)

**Figure 48. Velocity Contour at max wind tunnel speed**

![Streamline profile of acrylic plate at wind tunnel max speed](image2)

**Figure 49. Streamline profile of acrylic plate at wind tunnel max speed**

Both figures above show different views of the side profile of the streamline on the side of the acrylic shield. On the contour diagram, the leading edge at x=0 is where the streamline is tripped into the turbulent region and is transitioning back and forth between laminar and turbulent flow until nearly
0.1m (3.93 in.). After 4 inches, the flow is fully turbulent as the boundary layer slowly increases. The benefit of the flow being in the turbulent region is it keeps the boundary layer smaller than if in the laminar region. This different in boundary layer thickness as the Reynold’s number decreases is demonstrated in the table below.

Table 6. Various Boundary Layer Thicknesses at Different Wind Tunnel Velocities

<table>
<thead>
<tr>
<th>Upstream Velocity (MPH)</th>
<th>Distance from leading edge (in)</th>
<th>BL Thickness (m)</th>
<th>BL Thickness (in)</th>
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<td>10</td>
<td>10</td>
<td>0.0046</td>
<td>0.1819</td>
</tr>
</tbody>
</table>

Figure 50. Boundary Layer Thickness at Various Velocities
The table above shows the boundary layer thickness on the acrylic shield at the standard mount. The boundary layer thickness is measured 10 inches from the leading edge where the standard mount connects to the tire. The thickest boundary layer is 0.18 inches, which means the flow is streamlined in between the tire and the shield because the tire sits further away from the surface of the shield than 0.18 inches. The behavior of the boundary layer can be seen to be increasing substantially as the upstream velocity decreases. This is a direct relationship with the Reynold’s number and the flow transitioning from turbulent flow to laminar flow. One of the main characteristics of turbulent boundary layers is they have substantially thinner boundary layers than laminar flow. The acrylic shield will have the smallest affect therefore at higher wind velocities.

Next, we want to make sure that the bolts that we are using to secure the entire assembly are strong enough to withstand the weight of the design in addition to the induced forces from the wind tunnel. Below are the calculations showing the effective bolt stiffness in our highest risk bolts in the base plate that connects the whole tower to the I-beam.

Tensile stress area of the bolt is:

\[ A_t = 0.1599 \text{ in}^2 \]

The minor diameter area of the bolt is:

\[ A_r = 0.1486 \text{ in}^2 \]

The grip length of the bolt is as follows:

\[ l = h + \frac{t_2}{2} \]

\[ l_t = l - l_d, \]

Where \( l_d \) is the unthreaded length of the fastener. For this fastener, the threads run all the way to the socket head so \( l_d \) is zero. \( t_2 \) is the area of the bottom steel plate of the aluminum, gasket, and steel plate stack. Here, \( h \) is the thickness of the washer and the top plate (aluminum and gasket). Shown below is the fastener length where \( d \) is the nominal bolt diameter.

\[ L = h + 1.5d \]

Finally, the overall effective bolt stiffness \( K_b \) is shown.

\[ K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} \]

\[ K_b = 10,292,414 \text{ psi} \]
4.3 Safety Considerations

Our system is designed to shield the user from any potential pinch points or electrical hazards. However, hazards are still a concern for the user. Both the user manual and large hazard stickers will notify the user of any hazards. Use of any of the parts outside of their intended use could present unforeseen harm to the user and the material. For a safety checklist, see appendix F.

One of the main hazards inherent in the system is electrical shock. There are several methods in place to shield the user from this hazard. The first one is placing all wiring in swiveling wiring harnesses along the base I-beam. Additionally, the system has low voltage and current—voltage to the board can be anywhere between 6-20 volts. However, if there is less than 6 volts, the board may become unstable or if more than 12 volts, the voltage regulator may become overheated and damage the board. If more than 500mA of current is applied to the USB port, a built in fuse will break the connection until the short or overload is fixed.

Next, pinch hazards will be overcome by placing shielding around moving components. There is a large foil around the tower, which has the motorized lead screw. There is also a foil around the telescoping mechanism where there is access to a keyway to rotate the horizontal lead screws. We will try to smooth out all sharp edges from manufacturing. Additionally, there will be a warning sign informing users to be careful of all sharp edges present.

4.4 Material Selection

Weight and strength are the two largest concerns for this mechanism. However, since the budget for this project is constrained, we had to find a way of decreasing the cantilevered weight while still maintaining strength at a reasonable cost. The most important place to shed weight is in the telescoping mechanism where the cantilevered weight puts stress on all downstream components. The easiest way to shed weight is to use 6061 aluminum alloy for the telescoping tubes and housings. However, where strength is imperative, we are using A36 stainless steel. However, when two different metals in the galvanic series (Magnesium, Zinc, Aluminum, Steel, Lead, Tin, etc.) are in contact, accelerated corrosion occurs. This happens because the metals form a bimetallic couple, where the metals have different affinities and in turn allows a current to flow. For instance, Galvanized steel is simple a steel coated with zinc, which acts as the sacrificial anode and experiences the accelerated corrosion instead of the steel.
The figure above shows that steel and aluminum alloys some of the most anodic metals available. Because of this, we will take as much caution in protecting the metals from one another.

4.5 Fabrication and Assembly

One of the goals for this project is to supply parts for the project through our own manufacturing using campus facilities. The CNC machines on campus and mills are the primary method of manufacturing. McCarthy Steel is the main supplier of raw aluminum and steel stock. The table below shows the parts that need manufacturing on campus.

Table 7. Parts Being Partially or Fully Manufactured on Campus

<table>
<thead>
<tr>
<th>Part</th>
<th>Coating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel Mounting Plates</td>
<td>Gear Keyway</td>
</tr>
<tr>
<td>Aluminum Housings</td>
<td>Tower Carriage</td>
</tr>
<tr>
<td>Balsa Wood Frames</td>
<td>Gaskets</td>
</tr>
<tr>
<td>Standard Mount</td>
<td>Acrylic Shield</td>
</tr>
<tr>
<td>Standard Mount Flange</td>
<td>Gear Mounts/Stiffener</td>
</tr>
</tbody>
</table>
4.5.1 Fabrication

A majority of the parts for this project are being sourced from McMaster-Carr, Fastenal, or locally at McCarthy Steel. There should be minimal manufacturing needed on our end. However, below is a brief description of the parts that do need to be manufactured on campus and how we plan to do that. The manufacturing drawings for each of these parts is in Appendix D.

_Aluminum mounting plate:_ The mounting plate is a 3/4” aluminum plate. The plate is fastened to an aluminum t-slot via 5/16” diameter screw. Apex Industrial Supply supplies the t-slots and fasteners. Between the aluminum plate and the I-beam is a 1/16” neoprene gasket, which prevents the corrosion when steel and aluminum are exposed to one another. Again, between the I-beam and the existing aluminum t-slot supports is a 1/16” neoprene gasket.

_Standard Mount:_ The standard mount will be two pieces welded together. We decided to go with this approach in order to save material. If the mount were made from one piece of steel, we would have to remove a large amount of material in order to get the diameter down to the correct shaft size to press fit the bearing.

_Standard Mount Flange:_ The standard mount flange is made from two pieces of steel welded together. This is done in order to prevent an excessive amount of machining. There will be one piece 0.5x1.2x8 inches and another piece 3x1.2x4 inches.

_Guide Housing 1&2:_ The guide housing will be 2x6x8 inches. We will mill holes in the housing in which we will insert the telescoping tubes then weld them together. Since both of the housings have the same outer sizes, we will order just one 2x6 inch tube and cut into 8 inch increments. We will individually fabricate each housing after that.

_Balsa Wood Foil:_ The airfoil surrounding the tower is made from a balsa wood frame. In order to cut the wood to the desired shape, we will it using the large laser cutters in the Mustang '60 shop. Once the Balsa wood frame is assembled, we will wrap it in a MonoKote skin and apply heat to shrink-wrap the entire assembly. We will be using a laser cutter on campus to cut the balsa wood. This is done by importing an Adobe Illustrator file and inserting your part into the machine. The machine simply cuts the part loaded into illustrator.

One difficulty in performing any sort of manufacturing on campus is the delay caused by high demand of the facilities. In order to help mitigate any delay as much as possible, we will be using the CNC machines as soon as our parts arrive since these are the most highly impacted machines that this project will need. While the parts are being shipped, we will write the g-code via Solid Works and test the code on other material. This will ensure that when our material finally arrives, the code is ready to run and we run no risk in destroying material and re-ordering.

4.5.2 Assembly

Assembly will be one of the most complex parts of this project. Because there are so many components with such fine tolerances, the assembly is a high-risk portion of this project. An important aspect of assembly is the order in which we assemble the entire mechanism. For example, if assembled in the wrong order, we will not have access to the holes for certain bolts or the joints for certain welds. Due to this issue, we will need to have all of the parts for each subassembly before we can start permanently attaching the parts together. Everything that has a precise mate to another part of the assembly and is being welded, will be attached first. We will attach them first to ensure that the parts...
mated via a permanent attachment method will be properly aligned. We will then attach the main parts that are added via non-permanent attachment methods such as fasteners as they slightly less critical. Any accessory or less critical parts will be fastened last.

The order of assembly will somewhat overlap, however, the X-direction subassembly is what we will begin with. We will first machine the aluminum mounts and then hand drill holes in the flange of the I Beam to fasten our main X-direction structure to the aluminum T-slot frame of the existing wind tunnel. We will then use the additional aluminum mounts to attach the T-slot rail that our carriage for our X-direction movement is mounted on.

We will then start assembly on the Z-direction tower assembly. Using our machined plate on the bottom, we will weld in our rails and attach the main structural member of the tower followed by attaching the linear bearings and fastening on the top plate.

We will then be able to assemble the entire Y-direction assembly and attach it onto the vertical linear bearings. Lastly, we will make the airfoils for the Z-direction and Y-direction and put them over the mechanical assemblies.

4.6 Maintenance and Repair Considerations

We designed the wheel support to require minimum maintenance. Our design encloses all of the electronics and bearings in order to prevent corrosion and a buildup of debris. Overall, this is a small concern since this is an indoor facility.

The largest risk for this product is due to user error. The entire mechanism is designed to withstand higher forces than will ever be tested, however this does not mean it was designed to withstand the user leaving on the telescoping mechanism of using impact forces to move in different directions.

In the event of electronic failure, the first component to break would be capacitors. However, since the capacitors are very small, the failure would result in a small popping noise, a small puff of smoke, and a loss of power to the actuation system. Since the actuation system has reverse locking mechanism, the arm will not fall towards the ground in the event of loss of power. If these capacitor does blow, they are easily replaceable.

One of the most common mistakes with an Arduino Uno PCB is overloading the board with voltage. This means sending greater than 5.0 voltage to the I/O pins or 5.5 volts to the source pins. If this happens, the Arduino will start to smoke and melt. In order to prevent excessive voltage, a voltage divider will be places before the Arduino.

4.7 Cost Analysis

The initial budget given by our client was to not exceed $1,500. However, he stated that if there were a large enough need, he would be able to find alternative funding to cover any additional costs. A more detailed breakdown of the cost of this project is included in appendix E. A complete Bill of Materials (BOM) is shown in the tables below.
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<th>Assembly</th>
<th>Item</th>
<th>Name</th>
<th>Drawing/Part No.</th>
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Table 9. Tower Subassembly BOM

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<tr>
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<th>Quantity</th>
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<tr>
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<tr>
<td>35</td>
<td>Square Flange Linear Bearing</td>
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<tr>
<td>36</td>
<td>Square Flange Linear Bearing Fastener</td>
<td>371</td>
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<td>7.42</td>
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<td>37</td>
<td>Linear Bearing Fastener Nut</td>
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<td>Linear Bearing Gasket</td>
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<td>Motor/Lead Screw/Nut</td>
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<td>Large Gasket</td>
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<td>T Slot 15-30 Series Aluminum Extrusion @ 6'</td>
<td>221</td>
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<td>67.03</td>
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<td>Hex Nut Linear Brake</td>
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<td>Double Economy T-Nut</td>
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<td>Tower Top Plate</td>
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<td>Motor Bracket</td>
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Table 10. Tower Airfoil Subassembly BOM

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<tr>
<td>51</td>
<td>Air Foil Rib</td>
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<td>0.00</td>
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<tr>
<td>52</td>
<td>Air Foil Spar</td>
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<td>Air Foil Leading Edge</td>
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Table 11. Large Telescope Airfoil Subassembly BOM

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<td>18.13</td>
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<tr>
<td>55</td>
<td>Air Foil Rib</td>
<td>470</td>
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<td>0.00</td>
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<tr>
<td>56</td>
<td>Air Foil Spar</td>
<td>471</td>
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Table 12. Small Telescope Airfoil Subassembly BOM

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<td>Air Foil Spar</td>
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<td>60</td>
<td>Total Balsa Sheets</td>
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<td>61</td>
<td>Balsa Dowel</td>
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### Table 13. Base Rail Subassembly BOM

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<tr>
<td>62</td>
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<td>63</td>
<td>Mounting Plate Gasket</td>
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<tr>
<td>64</td>
<td>5/16&quot;-18x1-1/8 Hex bolt</td>
<td>233A</td>
<td>5.00</td>
<td>3.10</td>
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<tr>
<td>65</td>
<td>5/16&quot;-18 Stainless Steel Finished Hex Nut</td>
<td>233B</td>
<td>18.19</td>
<td>18.19</td>
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<tr>
<td>66</td>
<td>5/16&quot; 18-8 Stainless Steel Small OD Flat Washer</td>
<td>233C</td>
<td>8.44</td>
<td>8.44</td>
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<td>67</td>
<td>Grade B8, 1/2&quot;-13 Thread, 1-1/2&quot; Long</td>
<td>211A</td>
<td>26.10</td>
<td>52.20</td>
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<td>68</td>
<td>Grade 8 Type 18-8 Stainless Steel Hex Nut (packs of 10)</td>
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### Table 14. Electronics Subassembly BOM

<table>
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<td>71</td>
<td>Adafruit Stepper Shield</td>
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<td>1.00</td>
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<td>72</td>
<td>String Potentiometer Kit</td>
<td>510</td>
<td>3.00</td>
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<td>73</td>
<td>Potentiometer Housing</td>
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<tr>
<td>74</td>
<td>Potentiometer Spool</td>
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<td>Potentiometer Cover</td>
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</tr>
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<td>76</td>
<td>Snap Acting Limit Switch</td>
<td>531</td>
<td>8.88</td>
<td>17.76</td>
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<td>77</td>
<td>Upright Mount Pulley</td>
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<td>11.77</td>
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<td>78</td>
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<td>79</td>
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<td>7.16</td>
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<td>80</td>
<td>Inverting Schmitt Trigger</td>
<td>532</td>
<td>0.45</td>
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<td>24V Single Output AC/DC Switching Power Supply</td>
<td>533</td>
<td>51.95</td>
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<td>82</td>
<td>Capacitor</td>
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<td>0.59</td>
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</table>

The final cost of the above BOM came to $1728.41. However, this price doesn’t include shipping nor manufacturing costs. Our sponsor did mention his ability to get t-slots for free, so that may be able to reduce our price.

### 4.8 Critical Design Review

The critical design (Section 4) was not accepted by our sponsor due to aerodynamic concerns. As shown in Figure 23, the test section area blockage produced by our support is very significant and was the main reason for a major project re-scope.

The bulkiness of the support was a consequence of the necessary rigidity in the Y-direction sub-assembly. To increase simplicity and thus reduce the blockage, the project was re-scope to focus on...
X- and Z-direction sub-assemblies. The Y-direction sub-assembly would be a rigid, non-adjustable sting for the purposes of demonstration. This re-scope led to the final design detailed in Section 5.

5. Final Design

This final design is the product that has been delivered to our client, Dr. Doig. This section will detail the final design changes and provide an overview of manufacturing and assembly.

Figure 52. Final Balance Device Delivered to Client

5.1 Functional Description of Design

The biggest success achieved in this design when compared to the previous design is the overall cross-sectional area. Since this was the area that our client wanted us to improve the most in, there were certain design sacrifices that were made in order to achieve this more important goal. Each direction (X,Y, and Z) will be discussed in detail as well as the electrical controls.

5.1.1 Final X-Direction Design

The previous critical design had an I-beam stretching the distance of the wind tunnel. The final design has a square extruded steel beam stretching the complete length. The design change happened because the I-beam was too tall and was placing the overall design several inches above the rolling road. This final design sees more deflection, however that is countered by the vertical actuation from the motor.
Movement is still achieved in a similar manner to before—the carriage slides on a linear guide rail which is mounted to the square extruded steel beam.

![Figure 53. Final X-Direction Carriage and Guide Rail](image)

The above figure shows a rendering of the final X-direction design. The overall height profile of the design is significantly more compact compared to previous designs. The steel extruded beam is painted, so no galvanic series corrosion will be experienced. The rails were purchased from McMaster-Carr as well as the carriage and a majority of the fasteners.

![Figure 54. Beam Deflection Analysis Using SolidWorks FEA](image)

The finite element analysis of the beam indicates that the beam, when supported with fixed ends, will deflect a maximum of 0.0279 inches. In all visual tests we performed with the ends simply supported, the beam deflected a maximum amount of nearly one inch. With each end securely fixed, the FEA model should be a good approximation of the final deflection under the weight of the system.
The beam is attached to the wind tunnel structure via 3” x 1” aluminum L-bracket. This L-bracket is attached to the wind tunnel via t-slot drop-in mounting fasteners. The specific fasteners we purchased are made to install in t-slots that are already capped at the ends.

On the previous design we had the operator manually translating the carriage and using the hand brakes to lock the device in place. On this current design, the actuation is controlled by a Thomson Linear NEMA 23 linear actuating lead screw. This both moves and holds the carriage in the desired location and is accurate to within 5/1000ths of an inch. This eliminates the need for the user to crawl into the wind tunnel to move the carriage as well as gives the end user the ability to move the system while the tunnel is operating.
5.1.2 Final Z-Direction Design

The final z-direction design is similar to the original design. It is controlled by a single Thomson Linear NEMA 23 linear actuating lead screw. This stepper motor is capable of moving as small as 5/1000ths of an inch per step.

As shown in the figure above, there is a single spinning nut motor that is contained in the steel housing. This housing moves up and down sliding on two linear guide rails. These guide rails and guide rail carriages were purchased from McMaster-Carr online. The lead screw is mounted in the aluminum plate perpendicular to it underneath. There are two aluminum plates that we installed on the interior of the C-channel in order to mount the top lead screw to as well as to mount the entire tower to the carriage. Each aluminum shelf is 0.5” and is mounted to the c-channel with M6x1 socket head screws. The motors are mounted to the body via M5x1 bolts. The base aluminum plate is mounted to the base rail carriage via M6x1 screws.
Figure 57. Deflection Analysis of C-Channel Using SolidWorks FEA

The deflection analysis in the figure above indicates a maximum deflection of 0.001 inches will occur in the C-channel support. After building the device and securing all parts and attaching a wheel, the FEA estimate seems accurate in magnitude. There was no visible deflection indicated.
5.1.3 Final Y-Direction Design

This aspect of the wheel balance device is where the largest changes have occurred. All methods of translation have been removed and the arm is now a fixed length. This change happened due to manufacturing concerns and the large cross-sectional area.

As shown in the figure above, the arm is covered in a wing to diminish the affects on the wind flow. The standard mount is press fit on a bearing at the end of the steel arm. The arm is mounted at the base of a steel plate that is screwed on to the steel carriage that acts as the motor housing.
Figure 59. Deflection Analysis of Y-Direction Arm Using SolidWorks FEA

The FEA deflection analysis above indicates that the total amount of deflection the Y-Direction arm should see is 0.05 inches. After building and loading the design, this magnitude of deflection is also reasonable. The fixed end is securely attached via weld to a steel plate. The free end of the beam is where the bearing is press fit on to the shaft, which is also press fit into the standard mount.

Figure 60. Universal Mount and Arm Design
The final arm design shown above is mostly a proof of concept design. Our client specified that he and his students would design arms to meet their own specific needs for experiments. In order to accommodate different arms, the bottom of the large steel plate has holes drilled in order to attach different arms. The fasteners we suggest are low profile socket heads in order to clear the aluminum shelf as the tower actuates up and down the Z-direction.

5.2 Safety Considerations
The wind tunnel is utilized by a large variety of students—anywhere from a new student in their first year to Masters level students writing their thesis. The equipment in the tunnel needs to be usable and safe for all users alike. This was a goal of ours through the entire project and was continually being considered when changing ideas and concepts. Below are several safety considerations for this design:

*Limiting Carriage movement:* Limit switches are placed at both ends of each lead screw so that the carriage will collide with it when it reaches the end. These limit switches are then activated and send a voltage to the arduino control system, which then shuts off the power to the motor. Despite using these limit switches, there are still pinch points that the user is in danger of if around the unit when it is being operated.

*Voltage/Current Limits:* Voltage being supplied to the Arduino Uno and stepper shield should not exceed 12 volts. The current should be limited to 1.5 amps. These levels should not be harmful to any user. However, because motors draw 1.5amp and the microprocessor is limited to 1.2 amps, the unit will get very hot. This should not be a problem if the unit is not insulated. We recommend that the electronic control unit is placed on the prototype tower in the wind tunnel so the flowing air helps cool the unit.

*Prototype Removal:* Removing the prototype from the wind tunnel is a very hazardous job. The tower should be removed from the x-direction guide rails

*Burrs:* Because of the amount of end milling that has occurred with mildly worn tools, there are burrs throughout the entire assembly. While we tried to mitigate these burrs with various deburring tools, we inevitable missed some. These burrs are sharp enough to cut the user if they slide their hand over them.

5.3 Material Selection
Material selection is something that must be strongly considered in this project. Because this project will be in a wind tunnel under high forces, all parts must be self-contained. One of the main areas we had to find a balance with was strength vs. weight. Steel is incredibly cheap and strong but is very heavy. The unit cannot be too heavy since it is placed in the middle of a 14’ beam and cannot create too much deflection.

Proper material selection is also critical because as discussed earlier in the report, when two different metals in the galvanic series (Magnesium, Zinc, Aluminum, Steel, Lead, Tin, etc.) are in contact, accelerated corrosion occurs. This happens because the metals form a bimetallic couple and the metals have different affinities and in turn allows a current to flow. For instance, galvanized steel is steel coated with zinc, where zinc acts as the sacrificial anode and experiences the accelerated corrosion instead of the steel.
5.4 Fabrication

Fabrication in total for this project took an entire ten weeks to complete. All fabrication was done on campus and all raw resources (bar stock, aluminum stock, etc) were purchased at the local steel mill. Fasteners, gaskets, electronics, and other various parts were all sourced from McMaster-Carr, Arduino, and Adafruit respectively. The fabrication took ten weeks due to many unforeseen problems. Some of the main problems are enumerated below:

Manufacturing Errors: Several delays were caused when members were in the machine shops working on specific parts and errors were made. One specific setback was encountered when a team member was drilling holes in the motor mount carriage and was using an incorrect speed and not moving the bit through the material fast enough and melted the steel to the bit. The work-hardened the steel housing which was then impossible to work with again. This specific piece has very precise tolerances and must be carefully machined in order to align the linear guide rail carriages. This process added several days to the fabrication stage.

Electronics Coding: The coding of the arduino took several days longer than expected due to equipment availability and coding errors. We were attempting to source a power supply on campus due to high cost to purchase them. It happened that in the middle of our search our client acquired one which was he gave us to use. After this we were able to debug the code and wire the system and test the capabilities of the motors and computers.

5.5 Maintenance and Repair

Maintenance on this system should be very minimal. This is something that our client had specified in original meetings. The main components that will need servicing will be the stepper motors, guide rails, and fasteners. Each system’s maintenance is described below.

Stepper Motors: Maintenance should be mostly visual to ensure there isn’t any debris near the moving components. Each of the stepper motors is enclosed in a steel housing and should be mostly protected from different environmental conditions.

Guide Rails: Grease should be applied to the guide rail once a year in order to prevent the balls from freezing up in the linear bearing.

Fasteners: The fasteners should all be regularly checked to ensure nothing has come loose. This systems experiences significant vibrations, which cause fasteners to experience accelerated loosening.

5.6 Final Cost

The final costs for the project came to $1,545. The table below shows a break down of where these costs came from. A complete list of cost breakdown and part number association can be found in Appendix E.
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<thead>
<tr>
<th>Item</th>
<th>Name</th>
<th>Description</th>
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<th>Supplier Part No.</th>
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<th>Qty.</th>
<th>Unit Price</th>
<th>Total Price</th>
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<td>15</td>
<td>Socket Head Cap Screw</td>
<td>M4 x 0.7 20mm Low Profile, Type 316</td>
<td>McMaster-Carr</td>
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<td>$10.51</td>
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<td>$3.37</td>
<td>$3.37</td>
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<td>6&quot; x 6&quot; x .125&quot;, I-beam, Al6061</td>
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<td>23</td>
<td>T-Slot Bracket</td>
<td>3&quot; x 1-1/2&quot; Single Profile, Framing Extrusion</td>
<td>McMaster-Carr</td>
<td>47065T241</td>
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<td>$6.96</td>
<td>$13.92</td>
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<td>For ideation, wind tunnel mockup, expo easel</td>
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<td>Solvent, paint, sandpaper, drill bit, ...</td>
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</table>
6. Design Verification

Testing for this design mostly consisted of ensuring the accuracy of the lead screws. Both lead screws were verified in the mechatronics lab to ensure the motor can move by an individual step. Outside of this, no other verification happened due to time constraints and resource contraints. Given more time, we would order string potentiometers to measure the distance the motor moves along the lead screw per number of steps.

Our original design verification plan also included testing the carbon fiber wing for strength, however, as the design progressed, the carbon fiber wing moved from being a structural component to only being needed to decrease the impact of the arm on the flow.

All arduino code that has been written is for display and demo purposes to highlight the capabilities of the device. Any additional code that is needed for experiments will be dependent upon the end-user to develop.
7. Project Management

In order to successfully complete the entire design and build of the project on time, we thought it prudent to break up the project into primary responsibilities for each team member so multiple aspects of the project can be completed simultaneously. We decided that it would be wise to both break up the over-arching year-long responsibilities for the project as well as the subsystems of the actual prototype machine between team members.

Brady Hiob is the Technical Writing, Project Management and Manufacturing Lead for the entirety of the project. His responsibilities include being chief technical report writer as well as editor. He is the team member who is primarily in charge of communicating by email with our sponsor and suppliers. He is also in charge of making sure that both the team and project is on track. He is also in charge of developing and documenting our manufacturing plans and processes as well as organizing all parts, drawings, and presentations. His primary responsibility on the machine itself is the X-direction motion.

Ryan Hamamura is the Chief Researcher as well as the Mechatronics, Vibrations and Testing Lead. He is the team member in charge of gathering background research on wind tunnel testing, aerodynamics theory and relevant mechanisms that have previously been developed. He will be in charge of developing the power systems and controllers for the wheel balance/support machine as well as ordering, installing and testing them. He is also in charge of vibrational analysis. He is also in charge of testing the prototype, which means that he will be in charge of developing a testing process to determine if the prototype works properly and designing in any components, such as single-axis load cells, to record data if necessary.

Sam Fleet is the Chief Designer as well as Mechanical Analysis and Prototyping lead. He is in charge of sketching out concepts and then rendering them in CAD software such as Solidworks. He is also in charge of performing engineering hand calculations in order to select the correct mechanical components. If necessary, he will also be in charge of utilizing any Finite Element Analysis (FEA) or Computational Fluid Dynamics (CFD) software to solve any design issues that can potentially happen. When all the analysis and designing is done, he will be in charge of ordering many of the components, fabricating a lot of the components that cannot be sourced from vendors and overseeing the assembly of a lot of the mechanical components. He is also in charge of any prototyping that might need to be done before the final build. The primary subcomponent of the machine he will be in charge of is the Y-direction motion as well as the standard mount design.

Although the primary parts of this project have been divided up so that they are the primary responsibility of certain group members, we all intend to share the project management, design, manufacturing and testing responsibilities as well as manual work.

A general timeline for this project has been provided in the form of a Gantt chart in Appendix B.

8. Conclusions and Recommendations

With the completion of this project, it is very evident that this is just the first of many iterations of this design. Our client understands the complexity of this mechanism and will be working with his managers and students to improve the design and alter it to meet the exact needs of future experiments.
in the wind tunnel. Recommendations for each direction of motion are discussed in their respective sections.

As this is an early prototype of a finalized product, Dr. Doig and his PROVE lab team will need to continue developing our project to get exactly what they need for accurate, full-range scale model testing on the rolling road. We therefore have some further recommendations on how to improve upon our project to meet this end goal.

The primary limitation of our project is that it is very limited in the Y direction. Whereas in our CDR design, the X, Y and Z directions are all adjustable, in our final design only the X and Z directions are adjustable. For our final design, we decided to remove any adjustability and simply have a solid spar because it required much less parts for our limited manufacturing time, took up much less blockage area. However, this also meant that it is limited to testing one size of scale model car on the rolling road. To account for this we recommend that either the teams testing their vehicles in the tunnel, develop a set of four arms based on our design that span the exact distance from our tower to their wheels. The other option we recommend is that a group from the PROVE lab team look into designing an adjustable Y-direction arm like our team did for CDR except focus on using higher strength-to-weight ratio materials like carbon fibre to reduce the overall size of the arm mechanism and therefore the blockage area compared to our original design. Applying such an arm mechanism device to all four proposed towers would allow for easy rolling road testing by any team with any sized car or model.

Other recommendations we have relate to what we showed in the original CAD of our final design, there should be a carbon fibre fairing that sits fore and aft of the tower to reduce negative aerodynamic effects. We recommend that they be kept as small as possible to enable the towers to be moved as much as possible without interfering with one another. We also recommend that these carbon fibre fairings be designed to fit over the tower like a sleeve and then be clamped to secure and unsecure for easy removal. We found that wrapping carbon fibre over blue foam and letting it cure naturally in the air is an effective way of making fairings cheaply and easily, however, it does require a lot of time.

One of the other suggestions for this project that was requested by our sponsor, but we were unable to get to in our limited manufacturing time was the ability to output the exact location of each mechanism so that location could easily be found again at a later time. To solve this problem we planned on using simple homemade string potentiometers that we explained in our electronics section. We recommend that the PROVE lab team add these device to our machine so they effectively locate the tower and carriage (and maybe later the standard mount) without having to enter the tunnel test section and measure.
References and Notes

2. See Munson (2009), Section 9.1.
4. Figure 9.3 in Munson (2009).
5. See Barlow (1999), Chapter 1 for equations for fluid motion in nondimensional form, and Section 2.1 for derivation of dimensionless coefficients.
6. Barlow (1999), Section 2.2.
# Appendix A – QFD

![Quality Function Deployment Method](image)

Figure 61. Quality Function Deployment Method
Appendix B – Project Timeline

Table 15. Gantt Chart Timelines for Senior Project

*To view in high resolution, double click timeline.
Appendix C – Test Section Parameters

Table 16. Test Section Parameters

| Test Section Parameters |
|---|---|
| $\rho^*$ | 0.07 \text{ lbm/ft}^3 |
| $\mathbf{u}$ | 100.00 \text{ mph} |
| CD | 1.00 |
| A | 1,564.00 \text{ in}^2 |

Table 17. 5% Blockage Section Parameters

| 5% Blockage Area |
|---|---|
| $\rho^*$ | 0.07 \text{ lbm/ft}^3 |
| $\mathbf{u}$ | 146.67 \text{ ft/s} |
| CD | 1.00 |
| A | 0.54 \text{ ft}^2 |
| FD | 13.59 \text{ lbf} |

Drag Force per wheel

| FD$_{,\text{w}}$ | 0.85 \text{ lbf} |

Table 18. 10% Blockage Section Parameters

| 10% Blockage Area |
|---|---|
| $\rho^*$ | 0.07 \text{ lbm/ft}^3 |
| $\mathbf{u}$ | 146.67 \text{ ft/s} |
| CD | 1.00 |
| A | 1.09 \text{ ft}^2 |
| FD | 27.19 \text{ lbf} |

Drag force per wheel

| FD$_{,\text{w}}$ | 1.70 \text{ lbf} |
Appendix D – Drawing List and Part Numbers

100 – Top Level Assembly
   101 – Exploded Top Level Assembly

200 – X Direction Assembly
   201 – Base Rail
   202 – Sleeve Bearing Carriage
   203 – Sleeve Bearing Carriage Rail
   204 – T-slot bracket
   205 – Lead Screw Mount

300 – Z Direction Assembly
   301 – Ball Bearing Carriage Rail
   302 – Ball bearing carriage
   303 – C-channel
   304 – Rail Spacer Plate
   305 – Motor Housing Mount Plate
   306 – Motor Housing
   307 – Middle Shelf
   308 – Base Shelf

400 – Y Direction Assembly
   401 – Ball Bearing
   402 – Front Mount Plate
   403 – Foil Mount Plate
   404 – Air Foil
   405 – Outer Foil Mount Plate
   406 – Sting Rod
   407 – Wheel Mount

500 – Controls
   501 – Arduino Uno
   502 – Adafruit Motor Shield
   503 – Stepper Motor
   504 – Limit Switch

600 - Hardware
   601 – M6 Socket Head Screw
   602 – M6 Hex Nut
   603 – M6 Flat Washer
   604 – M4 Socket Head Cap Screw (20mm)
   605 – M4 Hex Nut
   606 – M4 Flat Washer
   607 – M4 Socket Head Cap Screw (50mm)
101 – Exploded Top Level Assembly
201 – Base Rail

ALL DIMS. IN INCHES
MATERIAL: STEEL

9 X 4.72
10 X 0.25
36.00
1.00
168.00

2.00
1.00

Call Poly Mechanical Engineering
ME 430- SPRING 2016

Lab Section: Assignment #: Title: BASE RAIL

Dwg. #: Nat Asb: Date: 6/29/16 Scale:

Drawn By: SAMUEL FLEET

Chk'd By:
203 – Sleeve Bearing Carriage Rail

For use with M6 Socket Head Cap Screws

Extra-Wide Sleeve-Bearing Guide Rail
204 – T-slot bracket
301 – Ball Bearing Carriage Rail

**McMaster-Carr**: 6688K31
Steel
Vern-Mount Guide Rail
302 – Ball bearing carriage

Guide Rail 6688K33 Sold Separately

M6 Thread
8 mm Thread Depth

48 mm

64 mm

24 mm

3 mm

15 mm

47 mm

13 mm

3 mm

15 mm
303 – C-channel
304 – Rail Spacer Plate

ALL DIMS. IN INCHES
MATERIAL: STEEL

Call Poly Mechanical Engineering
ME 430 – SPRING 16

Lab Section: Assignment #: Title: SPACER

Dwg. #: Nat Adb: Date: 5/20/16 Scale: 1:2

Drawn By: SAMUEL FLEET

81
307 – Middle Shelf
401 – Ball Bearing
87
405 – Outer Foil Mount Plate

ALL DIMS. IN INCHES
MATERIAL: STEEL

1/4-20 Tapped hole

Cal Poly Mechanical Engineering
ME 430 - SPRING 2016
406 – Sting Rod

---

Cal Poly Mechanical Engineering

ME 430 - SPRING 2016

Lab Section:  
Assignment #:  
Title: STINGROD

Dwg #:  
Nat Asb:  
Date: 6/2/16  
Scale:  
Chkd By:  

Drawn By: SAMUEL FLEET
501 – Arduino Uno

![Diagram of Arduino Uno (Rev3)]
502 – Adafruit Motor Shield
### 503 – Stepper Motor

![Stepper Motor Diagram]

- **Dimensions Projection**

  **Maximum Stroke (Smax)**
  
  $S_{max} = L_m - L_n - 0.05$ in (1.6 mm)

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<th>Value</th>
<th>Unit</th>
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<td>[0.8]</td>
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<td>[1.6]</td>
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<tr>
<td>$D_{in}$ in [mm]</td>
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<td>[38.1±0]</td>
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<tr>
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</tr>
<tr>
<td>$G_{in}$ in [mm]</td>
<td>2.25 NAX [57.2 MAX]</td>
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<td>$H_{in}$ in [mm]</td>
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<tr>
<td>$I$</td>
<td>#22 AWG</td>
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<tr>
<td>$J_{in}$ in [mm]</td>
<td>12.0±0.05</td>
<td>[305±13]</td>
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<td>Flying leads are standard. Custom connection solutions possible. Contact Thomson for more information.</td>
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<td>$M_{i}$</td>
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<tr>
<td>$N$</td>
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<td>$\bar{O}_{in}$</td>
<td>0.375</td>
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</table>
504 – Limit Switch
601 – M6 Socket Head Screw

M6 x 1 mm Thread

5 mm Hex

10 mm

6 mm

25 mm
602 – M6 Hex Nut

M6 x 1 mm Thread

10 mm

5 mm
For M6 Screw Size

Washer may vary from 1.4 mm to 1.8 mm in thickness.

603 – M6 Flat Washer
604 – M4 Socket Head Cap Screw (20mm)
605 – M4 Hex Nut

M4 x 0.7 mm Thread

7 mm

3.2 mm

M4

94150A335

Hex Nut
606 – M4 Flat Washer

For M4 Screw Size

Washer may vary from 0.7 mm to 0.9 mm in thickness.
607 – M4 Socket Head Cap Screw (50mm)
### Appendix E – Cost Breakdown

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<th>Supplier Part No.</th>
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<th>Unit Price</th>
<th>Total Price</th>
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<td>Miscellaneous</td>
<td>Solvent, paint, sandpaper, drill bit, …</td>
<td>Miner’s, Home Depot</td>
<td></td>
<td>1</td>
<td></td>
<td>$50.00</td>
<td>$50.00</td>
</tr>
<tr>
<td></td>
<td></td>
<td><strong>Grand Total</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td><strong>$1,545.45</strong></td>
<td><strong>$1,545.45</strong></td>
</tr>
</tbody>
</table>
Appendix F – Safety Checklist

ME428/429/430 Senior Design Project 2015-2016

SENIOR PROJECT CONCEPT DESIGN HAZARD IDENTIFICATION CHECKLIST

Team: CP Tunnelers
Advisor: Sarah Harding

Y N

☐ ☑ Will any part of the design create hazardous revolving, reciprocating, running, shearing, punching, pressing, squeezing, drawing, cutting, rolling, mixing or similar action, including pinch points and sheer points?

☐ ☑ Can any part of the design undergo high accelerations/decelerations?

☐ ☑ Will the system have any large moving masses or large forces?

☐ ☑ Will the system produce a projectile?

☐ ☑ Would it be possible for the system to fall under gravity creating injury?

☐ ☑ Will a user be exposed to overhanging weights as part of the design?

☐ ☑ Will the system have any sharp edges?

☐ ☑ Will any part of the electrical systems not be grounded?

☐ ☑ Will there be any large batteries or electrical voltage in the system above 40 V either AC or DC?

☐ ☑ Will there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids?

☐ ☑ Will there be any explosive or flammable liquids, gases, or dust fuel as part of the system?

☐ ☑ Will the user of the design be required to exert any abnormal effort or physical posture during the use of the design?

☐ ☑ Will there be any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?

☐ ☑ Can the system generate high levels of noise?

☐ ☑ Will the device/system be exposed to extreme environmental conditions such as fog, humidity, cold, high temperatures, etc?

☐ ☑ Is it possible for the system to be used in an unsafe manner?

☐ ☑ Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any “Y” responses, add a complete description, list of corrective actions to be taken, and dates to be completed on the reverse side.