DESIGN AND VALIDATION OF A CHASSIS DYNAMOMETER FOR
PRESENT AND FUTURE VEHICLE TESTING AND DESIGN

A Thesis Presented to
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Fritz J. and Dolores H. Russ
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Master of Science
by

Robert L. Wilson III
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Table of Contents

Lists of Tables ................................................................. v
Lists of Figures and Illustrations ........................................... vi
Symbols and Abbreviations .................................................. ix

Chapter 1
1 Introduction ........................................................................... 1

Chapter 2
2 Literature Review ............................................................... 9
2.1 Types of Dynamometers .................................................... 10
   2.1.1 Electric Generator ......................................................... 10
   2.1.2 Water Brake Dynamometer ........................................... 14
   2.1.3 Eddy Current Dynamometer .......................................... 16
   2.1.4 Inertia Dynamometer .................................................... 18
2.2 Types of Tests and Purpose of Tests .................................... 20
   2.2.1 Engine Dynamometer Testing ........................................ 21
   2.2.2 Transmission Dynamometer Testing .............................. 25
   2.2.3 Chassis Dynamometer Testing ....................................... 28

Chapter 3
3 Design .................................................................................. 30
3.1 Phase I ............................................................................. 30
   3.1.1 Roller Carriage ............................................................. 30
   3.1.2 Electric Generator and Power Transfer Case .................. 32
3.2 Phase II ............................................................................ 33
   3.2.1 Finding Dynamometer .................................................. 34
   3.2.2 Design of Power Transfer Case, Motor Mount, Tie Down System, and Leveling of the Roller Carriage System ................................................................. 34
3.3 Phase III ........................................................................... 38
   3.3.1 Selection of Dynamometer ............................................ 39
   3.3.2 Data Acquisition/Controller ......................................... 43
   3.3.3 Cooling Water System Design ....................................... 45
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3.4 Coupler Redesign for Roller Chassis</td>
<td>47</td>
</tr>
<tr>
<td>3.3.5 Half Coupler for the Power Transfer Case</td>
<td>49</td>
</tr>
<tr>
<td>Chapter 4</td>
<td></td>
</tr>
<tr>
<td>4 Construction</td>
<td>53</td>
</tr>
<tr>
<td>4.1 Stage One</td>
<td>53</td>
</tr>
<tr>
<td>4.2 Stage Two</td>
<td>59</td>
</tr>
<tr>
<td>Chapter 5</td>
<td></td>
</tr>
<tr>
<td>5 Validation of the Chassis Dynamometer</td>
<td>61</td>
</tr>
<tr>
<td>5.1 Benchmark Testing</td>
<td>61</td>
</tr>
<tr>
<td>5.2 Validating of Ohio University's Chassis Dynamometer</td>
<td>73</td>
</tr>
<tr>
<td>Chapter 6</td>
<td></td>
</tr>
<tr>
<td>6 Conclusion</td>
<td>81</td>
</tr>
<tr>
<td>Bibliography</td>
<td>89</td>
</tr>
<tr>
<td>Appendix A</td>
<td></td>
</tr>
<tr>
<td>A.1 Water Cooling System</td>
<td>91</td>
</tr>
<tr>
<td>A.2 Half Coupler Design</td>
<td>92</td>
</tr>
<tr>
<td>Appendix B</td>
<td>97</td>
</tr>
<tr>
<td>Appendix C</td>
<td>115</td>
</tr>
<tr>
<td>Appendix D</td>
<td></td>
</tr>
<tr>
<td>D.1 Calibration Procedure</td>
<td>132</td>
</tr>
<tr>
<td>D.2 Operation Procedure</td>
<td>133</td>
</tr>
<tr>
<td>Appendix E</td>
<td>136</td>
</tr>
<tr>
<td></td>
<td>138</td>
</tr>
</tbody>
</table>
## List of Tables

<table>
<thead>
<tr>
<th>Tables</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Table 5-1</td>
<td>75</td>
</tr>
<tr>
<td>Table 6-2</td>
<td>81</td>
</tr>
<tr>
<td>Table 6-2</td>
<td>84</td>
</tr>
<tr>
<td>Table 6-3</td>
<td>86</td>
</tr>
</tbody>
</table>
# List of Figures and Illustrations

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 1-1</td>
<td>1</td>
</tr>
<tr>
<td>Figure 1-2</td>
<td>5</td>
</tr>
<tr>
<td>Figure 1-3</td>
<td>6</td>
</tr>
<tr>
<td>Figure 1-4</td>
<td>8</td>
</tr>
<tr>
<td>Figure 2-1</td>
<td>11</td>
</tr>
<tr>
<td>Figure 2-2</td>
<td>12</td>
</tr>
<tr>
<td>Figure 2-3</td>
<td>14</td>
</tr>
<tr>
<td>Figure 2-4</td>
<td>15</td>
</tr>
<tr>
<td>Figure 2-5</td>
<td>16</td>
</tr>
<tr>
<td>Figure 2-6</td>
<td>17</td>
</tr>
<tr>
<td>Figure 2-7</td>
<td>18</td>
</tr>
<tr>
<td>Figure 2-8</td>
<td>18</td>
</tr>
<tr>
<td>Figure 2-9</td>
<td>22</td>
</tr>
<tr>
<td>Figure 2-10</td>
<td>23</td>
</tr>
<tr>
<td>Figure 2-11</td>
<td>24</td>
</tr>
<tr>
<td>Figure 2-12</td>
<td>24</td>
</tr>
<tr>
<td>Figure 2-13</td>
<td>25</td>
</tr>
<tr>
<td>Figure 2-14</td>
<td>26</td>
</tr>
<tr>
<td>Figure 3-1</td>
<td>31</td>
</tr>
<tr>
<td>Figure 3-2</td>
<td>35</td>
</tr>
<tr>
<td>Figure 3-3</td>
<td>37</td>
</tr>
<tr>
<td>Figure</td>
<td>Page</td>
</tr>
<tr>
<td>----------</td>
<td>------</td>
</tr>
<tr>
<td>Figure 3-4</td>
<td>38</td>
</tr>
<tr>
<td>Figure 3-5</td>
<td>41</td>
</tr>
<tr>
<td>Figure 3-6</td>
<td>42</td>
</tr>
<tr>
<td>Figure 3-7</td>
<td>43</td>
</tr>
<tr>
<td>Figure 3-8</td>
<td>44</td>
</tr>
<tr>
<td>Figure 3-9</td>
<td>46</td>
</tr>
<tr>
<td>Figure 3-10</td>
<td>47</td>
</tr>
<tr>
<td>Figure 3-11</td>
<td>49</td>
</tr>
<tr>
<td>Figure 3-12</td>
<td>51</td>
</tr>
<tr>
<td>Figure 4-1</td>
<td>55</td>
</tr>
<tr>
<td>Figure 4-2</td>
<td>56</td>
</tr>
<tr>
<td>Figure 4-3</td>
<td>57</td>
</tr>
<tr>
<td>Figure 4-4</td>
<td>58</td>
</tr>
<tr>
<td>Figure 5-1</td>
<td>62</td>
</tr>
<tr>
<td>Figure 5-2</td>
<td>64</td>
</tr>
<tr>
<td>Figure 5-3</td>
<td>65</td>
</tr>
<tr>
<td>Figure 5-4</td>
<td>66</td>
</tr>
<tr>
<td>Figure 5-5</td>
<td>67</td>
</tr>
<tr>
<td>Figure 5-6</td>
<td>69</td>
</tr>
<tr>
<td>Figure 5-7</td>
<td>70</td>
</tr>
<tr>
<td>Figure 5-8</td>
<td>71</td>
</tr>
<tr>
<td>Figure 5-9</td>
<td>72</td>
</tr>
<tr>
<td>Figure 5-10</td>
<td>74</td>
</tr>
<tr>
<td>Figure</td>
<td>Page</td>
</tr>
<tr>
<td>-----------------</td>
<td>------</td>
</tr>
<tr>
<td>Figure 5-11</td>
<td>76</td>
</tr>
<tr>
<td>Figure 5-12</td>
<td>76</td>
</tr>
<tr>
<td>Figure 5-13</td>
<td>77</td>
</tr>
<tr>
<td>Figure 5-14</td>
<td>80</td>
</tr>
<tr>
<td>Figure 6-1</td>
<td>86</td>
</tr>
<tr>
<td>Figure A-1</td>
<td>92</td>
</tr>
<tr>
<td>Figure A-2</td>
<td>107</td>
</tr>
<tr>
<td>Figure D-1</td>
<td>135</td>
</tr>
</tbody>
</table>
Symbols and Abbreviations

\( A_{\text{Bearing}} \) = the area in which the bearing force

\( A_{\text{Shear}} \) = shear area on the key

\( \sqrt{a} \) = Neubler’s constant

\( \alpha \) = angular acceleration

A&B = are given values based on the neck down ratio

\( c \) = ratio

\( C_{\text{load}} \) = correction factor for the type of load

\( C_{\text{reliab}} \) = correction factor for the reliability of the coupler

\( C_{\text{size}} \) = correction factor for the specimen size

\( C_{\text{surface}} \) = correction factor for the couplers surface

\( C_{\text{temp}} \) = correction factor for the temperature operation of the coupler

d = diameter of reduced section after being necked down

d\( w \) = diameter of water line

\( D \) = diameter of the coupler

\( D_1 \) = inside diameter of specimen

\( D_o \) = outside diameter of specimen

\( D_s \) = diameter of shaft

\( f \) = roughness equivalent or smoothness of pipe surface

\( F_F \) = fuel energy

\( F_k \) = force applied to the key

\( F_m \) = force applied to the end of the moment arm
g = gravity

h = height of the key

h_L = head loss

h_S = the amount of head that a pump supplies

Hp = horsepower

I = mass moment of inertia

I_c = current

J = second mass moment of a specimen a geometric property of the shaft that can be calculated

k_L = loss coefficients for bend, inlets, outlets, and valves

K_{fs} = final notch correction factor

K_{ts} = first notch correction factor

K_T = stress concentration factor

l = length of key

l_w = length of water line

L = lower boundary

n = number of data points

n_D = normalized data

\eta = efficiency

\eta_{bearing} = factor of safety in bearing

\eta_{shear} = factor of safety in shear

\eta_N = efficiency of vehicle
\( \eta_{1000} \) = factor of safety for a 1000 cycles

\( \eta_{\infty} \) = factor of safety for infinite life

\( P \) = power

\( P_E \) = power out of engine

\( P_{\text{ele}} \) = electric power

\( P_m \) = mechanical power

\( P_O \) = output power

\( Pr \) = the pressure of the water

\( P_t \) = power out of transmission

\( P_w \) = power to the wheels

\( q \) = notch sensitivity

\( Q \) = volumetric flow rate

\( r \) = length of the moment arm

\( r_g \) = radius of gear

\( r_{g1} \) = radius of gear one

\( r_{g2} \) = radius of gear two

\( r_g \) = groove radius

\( r_p \) = radius of the water line

\( r_1 \) = radius from the center of the specimen to stress location to be inspected

\( r_1 \) = radius of gear one

\( r_2 \) = radius of gear two

\( R_n \) = radius of neck
$S_e = \text{the correct allowed stress for the infinite life}$

$S_{m_{\text{corr}}} = \text{the correct allowed stress for the number of required cycles}$

$S_{m_{10^3}} = \text{the allowable stress in a material to allow for 1000 cycles}$

$S_u = \text{ultimate strength of material}$

$S_y = \text{yield strength of the material}$

$\sigma' = \text{effective stress in the coupler}$

$\sigma'_{\text{Alt}} = \text{alternating effective stress}$

$\sigma_{\text{max}} = \text{maximum stress that the coupler will undergo during loading}$

$\sigma_D = \text{standard deviation}$

$\sigma_m = \text{standard deviation of the mean}$

$t = \text{time}$

$t_i = \text{confidence interval factor}$

$T = \text{torque}$

$T_E = \text{torque out of engine}$

$T_1 = \text{torque out of transmission}$

$\tau = \text{torque}$

$\tau_{\text{max}} = \text{the max torsional shear stress applied to the coupler}$

$U = \text{upper boundary}$

$V = \text{velocity}$

$V_o = \text{voltage}$

$V_1 = \text{velocity of gear one}$

$V_2 = \text{velocity of gear two}$
\( V_{wa} = \) water velocity

\( w = \) width of key

\( W_{drum} = \) work done on the drums

\( \omega = \) angular speed (rpm)

\( \omega_1 = \) angular velocity of gear one

\( \omega_2 = \) angular velocity of gear two

\( y_{BE} = \) curve fit data point

\( y_i = \) actual data point

\( z = \) height difference between the inlet/outlet with respect to a fixed datum

\( \theta = \) degrees the drums traveled

\( \gamma = \) specific weight of water

\( \xi = \) number of degrees of freedom

\( \psi = \) confidence interval

Subscripts:

Out = water conditions at the dynamometer inlet

In = water conditions at the reservoir
1. Introduction

In this thesis many different aspects of an automotive chassis dynamometer test system will be talked about and explained in great detail. The topics that will be covered in this thesis will include: what a chassis dynamometer is; its importance; how it is used in industry; why Ohio University RCENT needs one; and the process of how to design, construct, and validate a chassis dynamometer. Figure 1-1 shows a picture of a chassis dynamometer.

![A chassis dynamometer from Land and Sea](image)

Figure 1-1  A chassis dynamometer from Land and Sea

The importance of a dynamometer is to provide builders and designers of any power plant that produces power (such as internal combustion engines, electrical motors, jet engines, and etc.) the means to measure the power produced by that power plant. This is important, because although designers can predict the theoretical amount of power that a specific power plant should be able to produce, in reality the theoretical and actual power produced by the power plant are not equal, because of some unseen or unpredictable losses in the power plant. These losses come from numerous places, for example friction and internal resistances. Therefore how does a designer/builder find the
actual power produced by a power plant? This is done by the use of a dynamometer test system.

A dynamometer is a general term for any device that can apply a resistance load and measure that load and the resulting “loaded speed” of the power plant to determine power output. Dynamometers can measure powers ranging from 0.25 watt electric motors to 5000 lbs thrust jet engines, but a different type of dynamometer will have to be selected or designed for each specific power plant. In this thesis a power plant dynamometer will not be designed, but rather a chassis dynamometer.

What is a chassis dynamometer? A chassis dynamometer is used primarily in the automotive industry to measure the power that an automobile or other wheeled vehicle puts to the ground or to the tires. The power produced by an engine in an automobile does not directly go from the engine to the drive wheels, but instead the power goes through a drivetrain before reaching the tires or ground. A drivetrain consists of a transmission and sometimes a differential, which introduce a gear ratio that ratios the power and speed of the motor. Measuring the power at the tires gives the designers a better understanding of actual vehicle performance capabilities. Unlike an engine dynamometer a chassis dynamometer includes the transmission and differential losses and gearing effects to help designers better understand the end power and torque capabilities of a vehicle. Therefore the designers can decide if the current engine, transmission, and differential configuration are suitable to achieve the desired vehicle performance.

Besides finding the power being placed to the ground by a vehicle, with a chassis dynamometer the efficiencies of the subsystems of the powertrain can be found with
appropriate instrumentation of the test vehicle. Subsystems of a powertrain include the engine, transmission, and the differential. By knowing the losses in a powertrain an estimated efficiency can be calculated to inform the designer on the amount of power likely to be lost, which helps the designers in selecting the appropriate subsystems for the powertrain to obtain the desired vehicle characteristic performance. In other words a chassis dynamometer is more useful than a typical engine dynamometer, by its ability to test multiple configurations of powertrain subsystems as an entire powertrain system. Having the ability of testing an entire powertrain system and with appropriate instrumentation the individual subsystems of a powertrain allow the designers to create a vehicle design that transfers successfully from paper to reality.

Being able to test the powertrain of a vehicle is a crucial part in researching and developing new designs for the automotive industry. In our case it is also important to further increase the university's research capabilities by developing an automotive research faculty. Initially the chassis dynamometer will allow for extensive testing of the Electric Bobcat Race Vehicle, so that more information about the electric racecar can be known. After testing of the Electric Bobcat the chassis dynamometer will take an active role in developing future vehicles in joint research with the Ohio Future Car Consortium.

A chassis dynamometer system can be supplied in two fashions. The first choice is to purchase the chassis dynamometer system from a company that designs and builds chassis dynamometer systems. The second choice is to design and build your own. Instead of purchasing a chassis dynamometer system from a company, the decision was made to build our own. The reason for building our own is based on a couple of factors. Before the idea of using a chassis dynamometer for joint research with the Ohio Future
Car Consortium, a chassis dynamometer was already being designed for construction to test the Electric Bobcat RaceCar. This chassis dynamometer was very simple and inexpensive compared to buying one from a company, because the power needed to be absorbed was very low. Therefore when the idea to use the chassis dynamometer not only for the Electric Bobcat, but also for research on future vehicles was made part of the design was already done. Revamping the existing design would be cheaper than purchasing a new chassis dynamometer system from a company.

Before we get into more detail about chassis dynamometers a background is needed to understand a little more about how Ohio University’s chassis dynamometer came about. The development of the chassis dynamometer has gone through some redesign and changes from its initial conception to what exists today. The project has been developed in three major phases over a period of 4 years.

In 1998, two individuals, Marc Morgan and Brian Hintz, started the project under the guidance of Prof. Halliday. At that time the chassis dynamometer was to be designed to test the Electric Bobcat Race Car. How it was designed will be discussed in great detail in chapter 3 section 1. This phase of the project was not finished until the summer of 1999. From the time of the initial chassis dynamometer design to the completion of the design the Electric Bobcat Race Team received a donation of a new drive train. This new drive train increased the power of the Electric Bobcat RaceCar and had made the existing chassis dynamometer design obsolete, but not the whole design was lost. Figure 1-2 shows the frame and roller section of the chassis dynamometer designed by Marc Morgan and Brian Hintz.
The new drive train brought on the next phase of the project. Three individuals, Nathan Hagg, Matt Hannagan, and Robert L. Wilson III, undertook the second phase of the design under the guidance of Prof. Kremer, which consisted of two parts. The first part was to research power absorbers (dynamometers) that would be sufficient to absorb the power from the Electric Bobcat RaceCar with its new powertrain. In the midst of this research the criteria was enlarged to encompass the testing of other vehicles with potentially higher power levels. The second part of phase two was to design and build a power transfer case (Figure 1-3), tie down system, leveling system, and the design of an engine stand. These designs were implemented to support the chassis dynamometer, and the design will be looked at in greater detail in chapter 3 section 2.
After the completion of phase two, Robert L. Wilson III undertook the job to complete the design of the chassis dynamometer. Since the original conception of the idea of the chassis dynamometer, a lot of things had happened. So the first thing that needed to be done was the selection of a dynamometer and a data acquisition system. The dynamometer that was selected is the AE250 model eddy current dynamometer from Digalog Corp., and the data acquisition system that was selected is the Dyne-Loc IV Digital Dynamometer Control from Dyne Systems. Once the dynamometer was selected all previous design work had to be looked at to see if it was still valid and could be used. Parts from the first two phases that were no longer valid or could not be used had to be redesigned to comply with the new design criteria, which were based on the maximum capabilities of the AE250 eddy current dynamometer.

Once the design was brought up to the new specifications, the support parts and systems were redesigned to support the eddy current dynamometer. One part that was designed was a half shaft that allowed the connection between the eddy current
dynamometer and the power transfer case. Another system that had to be designed to support the dynamometer was the cooling water system. This system was designed to keep cooling water flowing into the dynamometer to prevent the dynamometer from overheating. This is the design aspect of the chassis dynamometer, but other aspects of phase three still exist and will be discussed in greater detail in chapter 3 section 3.

Another aspect of the dynamometer project is the construction of all the parts and subsystems to develop the entire system. Also testing of the individual sensors is necessary to verify that they are working properly, so that correct data is being collected. The last part of the project and the major part of the thesis are validating the chassis dynamometer to verify that the data being collected is correct. To do this an outside independent chassis dynamometer test of the Electric Bobcat was done to create power curves to compare with the power curves for the Electric Bobcat created from initial tests on our chassis dynamometer. Figure 1-4 shows the final construction of the chassis dynamometer.
Figure 1-4 Final construction of the dynamometer
2. Literature Review

Since the invention of the automobile, our history and society have changed significantly. The automobile has allowed society to expand, travel, and has allowed our economy to convert from a local dependent economy to a national economy. The automobile has allowed us to transfer people and goods from one part of the country to the other in a quick and efficient manner. The ability to transport people and goods across the country is dependent in part on the amount of power available for propelling vehicles containing people and cargo. Vehicles with more power can transport more cargo at the same speed or the same cargo at faster speeds. Therefore engineers attempting to increase engine power needed a way to test the power output of automobile engines, and hence the dynamometer was invented.

A dynamometer is any device that has the capability to absorb power from an external device (power plant) while at the same time collecting useful information to determine characteristics about the power plant[1]. There are numerous types of dynamometers for applications ranging from small electric motors to large jet engines, with varying levels of complexity depending on the total power that needs to be dissipated, the level of control desired for test parameters, and the desired accuracy of the torque/speed and power/speed measurements. Since two significant aspects of this research are selections of an appropriate load system for the chassis dynamometer and the collection and interpretation of the dynamometer test data, the remainder of this section will focus on the types of dynamometer load systems that are candidates for our application and the types that can be run with a dynamometer test system.
2.1 Types of Dynamometers

Dynamometers come in various forms, sizes, and designs. The four types of dynamometers that are reasonable alternatives for our application are the electrical generator, the water brake dynamometer, the eddy current dynamometer, and the inertia dynamometer. The reason that these four will be discussed is because they are the most common dynamometers in industry. They are by no means the only dynamometers used. In all cases the size of the dynamometer load system is dependent on the amount of power that needs to be dissipated. Although all four dynamometers power absorption capabilities can be increased to absorb larger amounts of power, the power absorption envelope characteristics is also important. The power absorption envelope tells how much power and torque can be absorbed at specific speeds.

2.1.1 Electrical Generators

Electrical generators in most cases are the same as electric motors. The only difference between the two is the way they are used. In the case of an electrical motor, electrical power is supplied to the motor to produce mechanical work in the form of shaft work[2]. An electric generator is like an electrical motor, except that instead of generating work it generates electrical power. In other words the electric generator converts work done on the shaft to electrical power in the form of the voltage and current. Figure 2-1 shows a picture of an electric motor.

One problem with generating electrical power is the need to dissipate that power. In most cases the power is dissipated by running the electricity through a series of electrical resistors in a procedure commonly known as a burnt off process[3]. The
dissipation of the electrical field can be controlled by manipulating the number of resistors in series, the strength of the current field of the generator, or both.

![Three-phase induction motor](image)

Figure 2-1 Three-phase AC electric motor that can be used as an electrical generator for dynamometer purposes.

Another way to dissipate the electrical power is to send it back into the electrical grid. The additional equipment needed to convert the power produced from the electric generator into the form of the power grid is very expensive. Unlike companies that produce large quantities of power on a regular basis to recover the additional cost of a power converting system, our system will produce small amounts of power on an irregular basis. Therefore sending power back into the grid using our system will not cover the additional costs of a power conversion system making this option nonfeasible.

There are two ways to calculate the power. The first method is to place the generator on trunion bearings, which will allow the generator to rotate around a defined axis[1]. As the generator rotates around the axis an angular velocity of the load shaft ($\omega$) and a reaction torque ($\tau$) required to hold the generator in place can be measured. Once the angular speed and torque are measured then the power ($P$) that the generator produces will be known.
To measure the torque, a moment arm of a known length is attached perpendicular to the generator. As the generator rotates it produces a force at the end of the moment arm, which can be measured by either strain gauges or a force transducer. Figure 2-2 shows a diagram of a trunion-mounted dynamometer with a torque arm. The torque is then calculated by the following equation [4].

\[ P = \tau \omega \quad (2-1) \]

\[ \tau = F_m r \quad (2-2) \]

\( F_m = \) force applied to the end of the moment arm

\( r = \) length of the moment arm

By substituting equation 2-2 into 2-1, equation 2-1 can be rewritten.

\[ P = F_m r \omega \quad (2-3) \]

The second way to measure the power produced by the generator is much simpler and more accurate than the first. By measuring the voltage (V) and current (I) of the electrical (\( P_{\text{ele}} \)) power produced by the generator. Then using Ohm’s Law the power is
known. This equation is only valid with electric generators and not with other
dynamometers.

\[ P_{ele}(t) = I_c(t)V_d(t) \] (2-4)

The one drawback with the second method is that only the power is known. The amount
of torque and speed that the generator is producing are unknown. The torque is as
important as the power for automobiles, because torque dictates the amount of
acceleration a vehicle has and the amount of road load that a vehicle can overcome. Both
methods for determining power can be performed to verify that the power being
measured is correct.

There are numerous advantages to the use of electrical generators as load systems.
One advantage is that it is easy to measure power by measuring current and voltage, and
it is not overly difficult to measure torque and speed compared to the other dynamometer
systems on the market. Also the ability to control the dissipation of power quickly and
accurately is relatively simple. Another advantage to a generator is how compact the unit
is, which is handy in a confined space. One of the major disadvantages to the electrical
generator is the price tag. Electrical generators become more expensive as larger power
absorption requirements are needed. Figure 2-3 shows a picture of a representative
power/speed curve characteristic of an electric generator.
2.1.2 Water Brake Dynamometers

A water brake dynamometer is basically an inefficient pump. In other words it absorbs power by pumping a fluid [4]. The power device is connected to the water brake shaft, which is connected to an impeller. The impeller rotates and moves a fluid, generally water that is supplied to the brake. Unlike a normal fluid pump, the fluid does not flow freely. Instead it has to flow around stationary vanes. These stationary vanes impede the movement of the water, making the pump inefficient enough to cause large power losses [5]. The power being lost from stream is transferred back into the water in the form of heat. Therefore the water being used as the working fluid has to be cooled or discarded. The amount of fluid flow to the brake is controlled by a control valve, which increases the potential of resistance load while preventing the water brake from over heating. Figure 2-4 shows a picture of a water brake.
Measuring the power being absorbed by a water brake is done in the same fashion as shown in equations 2-1 to 2-3 for the electric generator. The water brake is mounted on trunion bearings and a moment arm is attached to measure the resistance force and resistance torque of the brake. If the angular velocity of the water brake load shaft is also measured, the power can be calculated by equations 2-1 to 2-3.

The advantages of a water brake dynamometer are numerous, but different from that of the electric generator. One of the advantages is that it is compact and relatively inexpensive. This is important due to the limited space and budget available for this project. One of the main drawbacks is its inability to absorb high amounts of power at relatively low speeds. Figure 2-5 shows a power/speed curve characteristic for a water brake dynamometer. This is important for the testing of the Electric Bobcat, because the electric motor that powers it produces high torque at low speed. Another drawback is
controllability of the water brake. The less precise control of the amount of load causes higher uncertainty in the measurement of the power and torque. Also the need for a water system to keep the water brake cool and supplied with water is another disadvantage of the water brake.

![Graph of power and torque absorption characteristics of a water brake dynamometer.](image)

**Figure 2-5 Power and the torque absorption characteristics of a water brake dynamometer. [1]**

### 2.1.3 Eddy Current Dynamometer

An eddy current dynamometer is similar to an electrical generator, except for the fact that it is designed to be very inefficient, so it absorbs power via losses rather than via electrical power generation. Eddy currents are purposefully produced between stationary magnets and the rotating toothed rotor. The production of eddy currents causes high inefficiencies and loss of power [1]. The power is converted to heat and has to be carried away to keep the dynamometer cool. Measuring the mechanical power being absorbed by an eddy current dynamometer is done in the same fashion as in the other dynamometers, using reaction torque and load shaft speed. Figure 2-6 shows a picture of an eddy current dynamometer.
There are many advantages to an eddy current dynamometer system. One of the advantages is the ability to absorb high amounts of power at relatively low speeds as required for our application. The eddy current dynamometer is also compact. Like the electric generator the eddy current dynamometer is very controllable, which reduces the uncertainty in the measurement of the power and torque. There are also disadvantages associated with the eddy current dynamometer. The first disadvantage is cost; it is more expensive than a water brake but significantly less expensive than an electric generator for the same level of maximum power absorption. Another is the need for a water system to keep the eddy current dynamometer cool and supplied with water. Figure 2-7 shows a representative power/speed curve characteristic for an eddy current dynamometer.
2.1.4 Inertia Dynamometer

Unlike the previous three-dynamometer systems the inertia dynamometer has no active power absorption (brakes). Instead of using brakes to dissipate power, it uses the inertia of large drums to dissipate the energy. For example, a Dynotec inertia dynamometer uses two drums, which are 4 feet in diameter and weigh about 2700lbs a piece [7]. Figure 2-8 shows a picture of inertia dynamometer.

Figure 2-8 An inertial dynamometer from DynoJet. [7]
Like the previous dynamometers the inertial dynamometer is calculating power and torque. Unlike the previous dynamometers the inertia dynamometer is not calculating the horsepower by measuring torque and speed. Instead the inertia dynamometer directly measures power and calculates torque. Power is measured based on the acceleration of the inertia of the drums, and it is this inertia that absorbs the power. The inertia of the drums is a quantitative value that can either be calculated or measured prior to the test. Once the inertia of the drums is known then power can be calculated based on the following equations [7]. The torque exerted on an object is equal to its mass moment of inertia (I) times angular acceleration (\(\alpha\)).

\[
\tau = I\alpha \quad (2-5)
\]

\(I\) = mass moment of inertia

\(\alpha\) = angular acceleration

Where the work (W) done on the rollers is equal to the torque times degrees of rotation that the drum moved.

\[
W_{\text{drum}} = \tau \theta \quad (2-6)
\]

\(W_{\text{drum}}\) = work done on the drums

\(\theta\) = degrees the drums traveled

by combining these two equations we get

\[
W = I\alpha\theta \quad (2-7)
\]

Once the work is known than the power can be calculated by dividing work by time (t).

\[
P = \frac{W}{t} \quad (2-8)
\]
By rewriting equation 2-8 and dividing power by 550 to convert lbf-ft/s to horsepower we get.

\[
Hp = \frac{I\alpha\theta}{550r}
\]  

(2-9)

Using these equations will allow one to calculate the power and torque of a vehicle.

There are many advantages of the inertia dynamometer compared to the other three systems. One is cost. The cost of a fully operation inertia chassis dynamometer will run about $80,000. This cost includes the entire cost of the inertia chassis dynamometer mounted on a trailer to allow the dynamometer to be hauled to any location for testing. Another is the fact that it is a passive power absorber so no resistor banks or cooling system is required. Therefore the system is self-contained. Also the inertia dynamometer is very accurate and reliable. The one disadvantage to it is that you cannot easily direct connect an engine or transmission up to the system to be tested. Also another disadvantage to an inertia chassis dynamometer is its size.

2.2 Types of Tests and Purpose of Tests

The previous sections talked about the different types of power absorbers that can be used in a dynamometer system. Once a dynamometer has been selected, it can be attached to a power plant so that data can be collected about that power plant. In automobiles the power plant is the engine, where most automobile engines are internal combustion engines. Recently electrical motors have come to the forefront as a replacement for or to be used along side internal combustion engines.

Common types of dynamometer tests that are conducted in industry are engine testing, transmission testing, and vehicle drivetrain testing on a chassis dynamometer [1].
All three of these tests are similar but very different. They all three look at power and torque output over a range of engine speeds, but for the engine dynamometer the output is the engine shaft, for the transmission dynamometer the output is the transmission output shaft, and for the chassis dynamometer the output is the driving wheels. As society has grown and became more aware of their environment, dynamometer testing has expanded to include the testing of fuel efficiency and vehicular pollution.

2.2.1 Engine Dynamometer Testing

In the early days of the automotive industry the only things that the companies were interested in were the amount of power and torque that an engine could produce. Companies spent large amounts of money, time, and manpower to understand how engines work. All of this effort was done in attempts to build more powerful engines, so that the automotive industry could build bigger and faster automobiles. One of the key aspects of engine design is the need to be able to test specific engines to measure their power and torque output.

In an engine dynamometer system a power plant, the engine is directly connected to a power absorber, like one of the four options discussed earlier. The engine is run through its operating speed range and at each designed increment of engine speed the torque is recorded [1]. Figure 2-9 is a simple diagram of a dynamometer system, while Figure 2-10 is a sample of a power/torque vs. speed plot.

The torque/speed plot is important for designers, because it tells them at what engine speed the peak power and torque are located. Also knowing how the rest of the power/torque vs. speed curve looks helps a car designer choose a correct transmission with the correct gear ratio. For example a lot of torque is needed to overcome road losses
and provide the acceleration characteristics required for a specific vehicle. After the vehicle reaches a constant velocity less torque is needed to maintain constant speed since the acceleration load is zero and the only loads that must be overcome are rolling resistance, air drag, and grade load. In addition to these cases where torque limits the maximum speed, the maximum speed of an automobile can also be limited by the speed of the engine, known as speed limited. Speed limiting is where enough torque is provided by the engine to overcome the present road loads, but the vehicle can’t go any faster. The vehicle cannot achieve faster speeds, because the engine has reaches its maximum speed and cannot go any faster.

![Diagram of a simple dynamometer system](Image)

**Figure 2-9 Diagram of a simple dynamometer system**

During the fuel shortage of the 1970’s the emphasis in engine design changed somewhat from power and torque to fuel economy. The idea was to produce engines that would use less fuel, while producing relatively the same amount of power and torque as their predecessors. To do this a dynamometer test cell had to be developed as seen in Figure 2-11. The basic dynamometer system still exists, but the additional instrumentation and hardware is added. The additional hardware is enclosed in a control box. The control box is basically a containment area that allows nothing to flow in or out of it without being measured. A control box allowed for additional sensors to measure the energy
states of incoming and outgoing air and fuel. By knowing the energy states of incoming fuels and air and knowing the energy states of outgoing air along with the amount of power produced by an engine with in the control box, the efficiency of the engine could be determined. To find the efficiency of an engine, an energy balance is performed on the energy entering and exiting the box as seen in Figure 2-12. The energies entering and existing is the box may be in the form of fuel, air, heat, and etc.

![Torque & Power vs Speed (3K6)](image)

**Figure 2-10** A sample power/torque vs. speed curve for an electric motor.

The following equation shows the how to calculate the efficiency of an engine ($\eta$)[1].

$$\eta = \frac{P_O}{F_E} \quad (2-10)$$

$P_O$ = output power

$F_E$ = fuel energy
Figure 2-11 Dynamometer test cell. [1]

Energy Balance

<table>
<thead>
<tr>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
<td>Exhaust gas</td>
</tr>
<tr>
<td>300 kW</td>
<td>60 kW</td>
</tr>
<tr>
<td>Ventilating fan power</td>
<td>Engine cooling water</td>
</tr>
<tr>
<td>5 kW</td>
<td>90 kW</td>
</tr>
<tr>
<td></td>
<td>Dynamometer cooling water</td>
</tr>
<tr>
<td></td>
<td>95 kW</td>
</tr>
<tr>
<td></td>
<td>Ventilation air</td>
</tr>
<tr>
<td></td>
<td>70 kW</td>
</tr>
<tr>
<td>Electricity for cell services</td>
<td>Heat loss, walls and ceiling</td>
</tr>
<tr>
<td>25 kW</td>
<td>15 kW</td>
</tr>
<tr>
<td>320 kW</td>
<td>430 kW</td>
</tr>
</tbody>
</table>

The energy balance for the engine, see Chapter 11, is as follows:

<table>
<thead>
<tr>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
<td>Power</td>
</tr>
<tr>
<td>300 kW</td>
<td>100 kW</td>
</tr>
<tr>
<td>Exhaust gas</td>
<td>90 kW</td>
</tr>
<tr>
<td>Engine cooling water</td>
<td>90 kW</td>
</tr>
<tr>
<td>Convection and radiation</td>
<td>20 kW</td>
</tr>
<tr>
<td>300 kW</td>
<td>300 kW</td>
</tr>
</tbody>
</table>

Figure 2-12 Balance table of all energies entering and leaving the test cell. [1]

Besides fuel economy, other things can also be measured by the use of a control box. For example by sound proofing the outside of the control box and placing microphones in the box, acoustic testing can be done to find out where additional energy
is being lost and the noise level of the engine. Noise can come from many different things, including things rubbing due to too tight tolerances or knocking. Besides noise testing, vibration testing can be done with a control box. Vibration testing involves looking for engine imbalances. For example the imbalances can come from the cam, pistons, and rocker arms, which rob the engine of power. By finding the source of noises and vibrations and getting rid of them, power losses can be reduced thus increasing fuel economy.

2.2.2 Transmission Dynamometer Testing

A transmission dynamometer is along the same lines as an engine dynamometer, with one major difference. The difference is that a transmission is not a power plant, or in other words it does not produce power. It is a gearbox, and all it does is transfer power from an engine to the wheels by reducing or increasing the torque. This is done by creating a ratio between two gears. The purpose of this is to allow for more torque to be applied to the wheels at a lower speed than what the engine can produce. Figure 2-13 shows a picture of a transmission dynamometer while Figure 2-14 shows how to calculate the torque and angular speed in a gear ratio. [6]
\[ V = \omega r \]  
\[ V = \text{velocity} \]  
\[ r_g = \text{radius of gear} \]

\[ V_1 = V_2 \]  
\[ V_1 = \text{velocity of gear one} \]  
\[ V_2 = \text{velocity of gear two} \]

\[ \omega_1 = \omega_2 \frac{r_2}{r_1} \]  
\[ \omega_1 = \text{angular velocity of gear one} \]  
\[ \omega_2 = \text{angular velocity of gear two} \]  
\[ r_{g2} = \text{radius of gear two} \]  
\[ r_{g1} = \text{radius of gear one} \]

Using equation 2-3, the new torque and angular speed coming out of the transmission can be calculated.

\[ P_E = P_r \]  
\[ (2-14) \]
\[ P_E = \text{power out of engine} \]
\[ P_t = \text{power out of transmission} \]

\[ T_t = \frac{T_e r_2}{r_1} \]  \hspace{1cm} (2-15)

\[ T_t = \text{torque out of transmission} \]
\[ T_e = \text{torque out of engine} \]
\[ r_2 = \text{radius of gear two} \]
\[ r_1 = \text{radius of gear one} \]

Theoretically a designer can calculate what the torque and new speed is going to be after it goes through the transmission, but in reality this is not the torque and speed. Due to inefficiencies in a transmission, especially an automatic transmission, a loss of around 5-10\% can be seen. While the entire drivetrain including the differential can see losses around 30\%. This is important because if a designer thinks there are 100ft-lbs of torque available from the transmission and there are only about 90-95ft-lbs of torque, the vehicle may not perform at the level expected by the designer. Therefore a transmission dynamometer tests the efficiency of the transmission and to see the actual output of the transmission. By knowing the transmission efficiency a correction in gear ratio can be made to provide the required torque/speed characteristic. If the correct gear ratio cannot be made to provide the correct torque/speed profile, then the transmission efficiency helps in selecting bigger engine to achieve the required characteristics. Also like in the engine testing vibration and noise testing can be performed on a vehicle to find problems, in an attempt to better the performance of the transmission.
2.2.3 Chassis Dynamometer Testing

A chassis dynamometer is the last type of testing that a vehicle will go through and is the most important test. The reason why it is the most important is based on the fact if you can only perform one test, an engine, transmission, or a chassis dynamometer test, only the chassis dynamometer test will tell you the actual capabilities of a vehicle. The other two will tell you what the vehicle systems are capable of doing, but not necessarily what the entire vehicle can do.

The purpose of chassis dynamometer testing is to find the actual power and torque that a vehicle places to the ground [1]. The reason this is important is that once you know this information, a designer can accurately determine the amount of torque a vehicle has for pulling or towing and the top speed of a vehicle. The designer knows this because what is placed to the wheels is all the power the vehicle has to use for propulsion.

The information obtained by this test tells a couple of different things. The first thing it tells is the overall efficiency of the vehicle. If both the power out of the motor and the power at the wheels are measured, then one will know how much power was lost and could calculate the power transfer efficiency. Now by knowing how much power is lost overall in the vehicle a search can be done to see where it is lost.

\[
\eta_v = \frac{P_e - P_w}{P_e} \times 100
\]  

(2-16)

\( \eta_v = \) efficiency of vehicle

\( P_w = \) power to the wheels
The transmission and differential are the biggest robber of power in the drive train. By knowing how much power the transmission and differential robs from the system and comparing the losses to the total power loss measured by the chassis dynamometer, a designer can tell how much additional power is lost due to bearing and friction within the vehicle. By doing the other tests, such as bearing test and etc., and adding up the total losses of each component should roughly equal the losses found by the chassis dynamometer. If two losses are not the same then the designer, then knows to look for what is causing the discrepancy. Also vibration and noise tests can be performed with a chassis dynamometer to find any additional problems or losses.
3. Design

The design of the chassis dynamometer has been a multi-group and a multi-year project. The design of the chassis dynamometer has been done in three distinct phases, with each phase beginning and ending with a particular group. The design of the chassis dynamometer has changed from its original conception to what exists today. The reason for the changes has come from the needs and requirements of the dynamometer. The design process and evolution of the chassis dynamometer will be explained starting at the beginning phase I.

3.1 Phase I

Phase I started with the two individuals C. Brian Hintz and Marc B. Morgan. This group started the design process under the direction of Professor Halliday. The original idea of the design was to design and construct a fully operational chassis dynamometer to test the electric bobcat racecar.

The design specifications for the chassis dynamometer were based on the configuration of the electric bobcat racecar at the time of phase I, and primarily its 90Hp DC motor. Therefore the dynamometer had to be able to absorb and dissipate at least 90Hp worth of energy. The group also placed a max speed rating for the dynamometer at 120mph equivalent wheel speed. With these specifications the group started to design the system. The chassis dynamometer was designed in three distinct sections [8].

3.1.1 Roller Carriage Structure

The first section of the chassis dynamometer system was the roller carriage structure (See Figure 1-2)[8]. The roller carrier structure was designed to make it possible to place the electric bobcat racecar on set of rollers. The rollers would be
connected via shafts and couplers and supported by bearings. The design of this roller carriage system was done to allow the power at the wheels to be transferred to a dynamometer.

To design the roller carriage structure, many different parts had to be specified and put together to build the entire system. The design starts off with the rollers. Four rollers were constructed of 12" dia. schedule 40 pipe with end plates welded on. The rollers had 1 3/4" dia. shafts running through them to be mounted to bearings, which sat on top of 6" C channel. The shafts were connected together with 1050T10B Steelflex Couplers. All of these parts are connected to produce the roller carriage system and can be seen in Figure 3-1.

![Figure 3-1](image)

Figure 3-1 The actual picture of the finished product of Phase I in addition to the power transfer case, which will be done in Phase II.
3.1.2 Electric Generator and Power Transfer Case

The second section of the design was the selection of a power absorber or the dynamometer [8]. At the time of phase 1, money was very tight and the construction costs for the roller carriage system were very expensive. Due to the expensive costs of the roller carriage system, and the fact that the electric bobcat race team had two DC electric motors to power the electric bobcat racecar, the plan at the time was to use the motor that was not in the racecar as an electric generator, primarily due to the money savings in not having to purchase a power absorber. The electric generator would allow for an easy way to measure the power of the car over a range of speeds.

Once the power absorber and the roller carriage system were conceptualized, a way to attach the two systems was needed. Therefore a third system needed to be designed to transfer the power from the roller carriage system to the electric generator. A power transfer case was thought of. The power transfer case consisted of two pulleys, one on the roller carriage system and the other connected to the generator shaft, with a belt wrapped around the two pulleys. In other words as the racecar turned the rollers, the rollers would turn the pulley connected to the roller carriage system, which would turn the pulley connected to the generator. Look at Figure 1-2 shows the drawing of the pulleys and generator.

Although the electric generator and power transfer case were conceptualized, they were never implemented. Building the roller carriage system took a considerable amount of time. In fact it took so long that this group graduated before the roller carriage system
was completed. By the time the roller carriage system was completed changes were in the works for the electric bobcat racecar, and hence phase II.

3.2 Phase II

Phase II was undertaken by a second group of individuals, Nathan Haag, Matt Hanagen, and Robert L. Wilson III. When this group undertook the job of finishing the chassis dynamometer, a couple of things had and were changing in the design criteria for the chassis dynamometer. The first change was that Professor Kremer had assumed responsibility for the design of the chassis dynamometer. This was done for a couple of reasons.

The first reason was that Professor Kremer was the new faculty advisor for the electric bobcat and he knew that the electric bobcat was going to receive a major overhaul. The overhaul entailed the replacement of the 90Hp DC motor with a 125Hp AC motor. The overhaul of the motor made certain aspects of the original design obsolete, namely the 90 Hp electric generator. Another reason was that Professor Kremer wanted to broaden the scope of the chassis dynamometer design to include testing of other vehicles besides the electric bobcat racecar for research application.

The first thing that group II had to do was locate and find adequate alternative dynamometers that could be used for the chassis dynamometer. The style and source of the dynamometer was not of any importance in finding adequate dynamometers. Besides finding adequate dynamometers, the group was in charge of finishing the design of the chassis dynamometer[9].
3.2.1 Finding Dynamometers

The selection of the proper dynamometers is the most important aspect in the redesign of the chassis dynamometer. Due to this fact a majority of the time was spent on finding and locating appropriate dynamometers. The dynamometers that were located varied from water brake dynamometers to eddy current dynamometers. As wide ranging as the many different types of dynamometer that were located, so were the vendors. This variety was good, because it gave the group numerous possibilities in selecting the appropriate dynamometer.

The main problems which prevented the group from buying a dynamometer were money and time. At the beginning of the project Professor Kremer had placed a proposal for money to purchase the chassis dynamometer in an attempt to buy and complete the chassis dynamometer that year. By the time that the proposal was approved and the money awarded, not enough time existed for the group to complete the project. Therefore the group had to settle on completing other tasks, which would aid in the completion of the chassis dynamometer. The aspects that the group undertook were the design and completion of the power transfer case, a motor stand, the leveling of the roller carriage system, and a tie down system to hold a car on the roller carriage system.

3.2.2 Designing of Power Transfer Case, Motor Mount, Tie Down System, and Leveling of the Roller Carriage System

The first thing that was designed was the power transfer case. The power transfer case was designed to transfer 150Hp at 120mph. The power transfer case was also designed to be as adjustable as possible, so that any dynamometer could be hooked up to
the power transfer case. Based on these design considerations, the group completed the design as seen in Figure 3-2.

Figure 3-2 The power transfer case. [9]

The power transfer case’s base consists of two W6x15 I beams at 32” lengths, with four pieces of 14” long C8x13 ¾” channel, and one C8x13 ¾” channel 21 ¼” long. The four 14” channels are placed back-to-back and welded to form an I-beam. The two channel I beams are now welded to the back of the 21 ¼” channel to form a U. This U structure is placed on top and bolted down to the two I-beams. This structure forms the base of the power transfer case, which supports the components that actually transfers the power.

The part of the power transfer case that transfers power consists of two Gates P56-8M-85 SK HTD Sprockets, one Gates 1760-8MGT-85 Powergrip Belt, two Woods SK 1 ¾” Bushings, 4Dodge P2B-DL-112 Pillow Block Bearings, one 1¾” 1045 ground and polished shaft at 41 ¼” long, two 2996T24 ¾”-10 Turn Buckles, and four 60645K38
Female Spherical Rod Ends, R.H. Shank. The powergrip belt wraps around the two sprockets, which rest on the bushings. One advantage of choosing sprockets is that by selecting different radius sprockets, gear ratios can be made to bring the torque and speed of a vehicle into the dynamometers working specification. One of the sprocket and bushing sets is supported by the roller carriage system, while the other set sits on the 1 3/4" shaft connected to the rear roller set. The shaft is attached to the bearings, which are bolted to the base of the power transfer case. Proper belt tension is maintained using a MVL Vertical Mount with a SO-6 shoulder stud and a P4F idler Pulley from Brewer Machine and Gear Company.

The second thing that was designed was a motor stand. The intent of the motor stand is to allow a motor to be tested directly without having to run it through a car. By having the ability to test just the motor by itself or in the car, allows for estimation of the power losses in the drive train. Knowing the power losses allows for better understanding of the car performance characteristics for simulations and better prediction of the amount of power that will been seen at the wheels when selecting different rated horsepower motors. Also knowing the power losses helps in selecting adequate gear ratios for a desired performance. The design of the motor mount can be seen in Figure 3-3.

The last thing that this group undertook was to devise a way to level and permanently install the roller carriage system and to tie down a vehicle to the roller carriage system. The leveling and permanent installation of the roller carriage system is necessary to prevent it from becoming torqued and binding up parts to prevent vibration.
at high speeds and to prevent movement of roller carriage system. Therefore the group's proposed design solution was to pour a level concrete pad to set and anchor the chassis dynamometer on.

Figure 3-3 A motor stand designed for our dynamometer. [9]

The tie down system consists of ratchet straps tying the car to the roller carriage system. The ratchet straps accomplish two goals. The first is to provide a safety net for the car, by preventing it from driving off the chassis dynamometer and causing damage to the building or hurting individuals. The other goal is to provide sufficient downforce on the drive axle to prevent slippage between the wheels and the rollers. This is important, because if the wheels slip then the data that is obtained during a test is invalid. The reason
is as the wheels slip some power is being lost and is not transferred to the dynamometer.

Figure 3-4 shows a picture of the strap location and tie down system.

![Figure 3-4 The tie down system.](image)

### 3.3 Phase III

Phase III of the project was undertaken by myself as a Master's Thesis to ensure that the project would be finished. Phase III entailed the finishing of the chassis dynamometer. The finishing touches for the chassis dynamometer included the selection of a dynamometer load system and control system, checking and verifying that the existing parts were still valid, design of any extra systems that may be needed, and validation testing with the chassis dynamometer. The design work will be discussed in this chapter while the validation will be talked about in chapter 5.
3.3.1 Selection of the Dynamometer

The selection of the dynamometer was based on information gathered by the second group. Looking at the different types of dynamometers, the group narrowed the choice down to three types of dynamometers. The three dynamometers that held the best potential were the water brake, eddy current, and electrical generator dynamometer. The inertia brake was eliminated, because the rollers had already been built and all inertia systems come with their own rollers. The selection process for choosing the best dynamometer was based on the following criteria: cost, amount of power it could absorb, and size. A more in depth review of the dynamometer specifications can be seen in Dr. Kremer’s request for quote located in Appendix B.

With the criteria defined, the selection process started. The first thing that was looked at was the amount of power and the speed ranges that the dynamometer could absorb. Looking at all three types they all could handle large quantities of power, but as more and more power absorption is needed the size and price of the dynamometers went up. Looking at the water brake it was quickly eliminated because of its inability to absorb high quantities of power at low speeds (a review of Figure 2-5 will show this). This left the electric generator and the eddy current dynamometer as the only two choices. Looking back at Figure 2-3 and 2-7 the electric generator and the eddy current can be said to have similar load absorption characteristics, with the electric generator having a little better load absorption capabilities. Therefore the electric generator was looked at first, but due to the expense of an electrical generator compared to the eddy current dynamometer, the electric generator was eliminated. This left the eddy current dynamometer as the best choice for our application.
Once the proper type of dynamometer was selected, the search was narrowed down to finding adequate eddy current dynamometer vendors. Two vendors that were found were Digalog Inc. and Dyne Systems. Digalog sells two types of eddy current dynamometers appropriate for this application, the AE150 and the AE250. Dyne Systems offers many dynamometer systems but they selected the 1014WHS from Midwest Dynamometers & Engineering Company in their price quote. The price and size of all three dynamometers were in the correct ballpark. The determining factor for selecting one over the other was based on the power absorption capability of the dynamometer. The criteria for the desired amount of power absorption at particular speeds was based primarily on the power/speed curve at the wheels of the electric bobcat with its current motor and transmission, and secondarily on projected power/speed curves for a 200Hp electric motor. Figure 3-5 shows the torque absorption of all three-candidate dynamometers and the torque curve of the current AC motor, with and without the transmission in all four gears, as well as a new AC motor rated at 200Hp.

Looking at Figure 3-5, one can see that all three dynamometers have an acceptable torque envelope to test the current motor if it were directly connected to the dynamometers. But when the power is transferred through the wheels to the chassis dynamometer the 1014WHS (WHS Figure 3-5) and AE150 cannot handle large portions of the vehicle torque output in 1st and 2nd gears and parts of 3rd and 4th gears. The dynamometer that best satisfies the torque requirements is the AE250 from Digalog. The torque envelope of the AE250 includes almost all of the vehicle torque output in 2nd, 3rd, 4th gears, and the stand-alone motor torque output, while only missing a small portion of the 1st gear output.
Figure 3-5 The Torque/Speed Characteristics of the Motor in all Four Gears as well as different Dynamometers

The only downfall for the AE250 is its inability to absorb sufficient power at low speeds for the Electric Bobcat fitted with a 200Hp motor. Other higher power dynamometers exist, but the price differential between the AE250 and the AE400 from Digalog ($13,200) made the AE250 the best compromise between cost and performance, and its size was also acceptable (a constraint based on our limited lab space).

The AE250 is an air gap eddy current dynamometer with the capability to dissipate 335hp (250KW) at 2000rpm, while the max torque is 886ft-lbs at 950rpm, with a max speed of 8000rpm. The AE250 (and all EC dynamometers) works as an inefficient generator. The power is input to the AE250 via a shaft, which is connected to a rotor. The rotor rotates pass a set of stators. Unlike an electrical generator, where electricity is produced, in the AE250 heat is generated by high inefficiencies. The heat is produced by a significant gap between the rotor and stator, and it is this gap, which also causes the
high inefficiencies between the stator and rotor. The rotor is manufactured from a high magnetic material.

The heat is carried out of the dynamometer by cooling water that passes through cooling passages. The cooling is monitored by embedding a thermocouple into a loss plate to measure the temperature to verify that enough water is being supplied. If insufficient water is supplied to the AE250 for cooling then thermal loading along with the cyclic loading can cause catastrophic contact with internal parts.

The torque and speed measurements come from different sensors. The speed is read by a 60-tooth wheel that produces a pulse, which is converted into an electrical signal. The torque is measured by a strain gauge load cell, which also produces an electrical signal that can be sent to a data acquisition system. The electrical signal is generated on the principle that the electrical resistance of a conductor changes as it is subject to mechanical deformation [4]. In other words by attaching a strain gauge to a load cell the force can be calculated. This is achieved by applying a load to the strain gauge to deform the material. As the material is being deformed the internal resistance in the material changes, and by measuring this change and knowing the relationship between the change in resistance and force the force can be calculated. Figure 3-6 shows a picture of the AE250 and Figure 3-7 shows the interior diagram of the AE250.
3.3.2 Data Acquisition/Controller

Besides the selection of the dynamometer, a controller/data acquisition system is also needed. The search for a controller/data acquisition system begins the same way as the search for the dynamometer did, looking for qualified vendors. Looking for a controller/data acquisition system was a little bit more difficult than the dynamometer. The reason was that most vendors that sold a controller/data acquisition system also sold dynamometers, where their controller/data acquisition system was designed specifically for their dynamometer.

The controller/data acquisition system that was sold by Digalog was too expensive for our budget and offered no great advantage in capabilities over alternative systems. Therefore a harder look for adequate controller/data acquisition systems found Dyne Systems. Dyne Systems makes a relatively inexpensive controller/data acquisition
system with tremendous capabilities and flexibility. Another plus with the Dyne System controller is that the system can be run alone or with a computer.

The system that was chosen to be the controller/data acquisition system from Dyne Systems was the Dyn-Loc IV Digital Dynamometer Control. Figure 3-8 shows a picture of the Dyne-Loc IV system. This digital has a lot of advantages over an analog system, such as zero percent regulation and drift. The system also has RS232, parallel, and analog interface options to allow for computer and test cell interface. The controller aspect allows for set point testing or more important sweep test with precise accuracy. In sweep testing the controller takes a starting rpm point and a finishing rpm as inputs and makes steady state measurements at numerous points between the starting and finishing points.

Figure 3-8 A front view of the Dyne-Loc IV data acquisition controller.
The data acquisition part of the system allows for accurate measurement. It uses a crystal oscillator based reference generator that can be set by using the rpm and torque set points from the dynamometer span and zero. The crystals allow for quick, precise, and accurate measurements of the rpm and torque. The digital output can be directly input into a computer to create torque speed curves.

3.3.3 Cooling Water System Design

As stated before the AE250 Eddy current dynamometer produces a lot of heat in efforts to dissipate energy. The cooling water system design criteria were based on the manufacturer's cooling requirements. The cooling requirement for the dynamometer (for the maximum power absorption case) is to supply a minimum of 45gal/min of cooling water at 24psi with the outlet water temperature not exceeding 140°F.

The first alternative design was the simplest and easiest to implement. The idea was to tie the dynamometer into the existing building water supply by running a line from the incoming building water supply to the dynamometer and running the exit water from the dynamometer to drain. The only problem with this idea was cost. The expense to have OU facilities make the modifications to the automotive lab (Stoker 012) to allow a direct connection to the dynamometer were considerably too expensive ($4,041) and put this idea out of the budget. Therefore other design alternatives were investigated, including a self-contained or a semi contained recirculating water-cooling system.

The self and semi contained recirculating systems both were based on the same concept with one exception. They both relied on the use of a pump-to-pump water from a reservoir source to the dynamometer. The exception is on the return side. The self-contained system would use heat exchangers to cool the water down before pumping it
back into the reservoir. The semi-contained system would not use heat exchangers; instead it would either recycle the used water or dump it to a drain and replace it with new cold water based on its temperature. Due to the expense of heat exchangers and the amount of energy that would need to be expelled at maximum power for a self-contained system, the semi-contained cooling system was selected. Figure 3-9 shows a diagram of the semi-contained cooling system, while Figure 3-10 shows a picture of the actual system.

The semi-contained cooling system is comprised of a water reservoir, pipes, and a pump. The water reservoir is a holding tank that holds a large quantity of water to be used for cooling. The pump is used to push the water from the reservoir through the pipes to the dynamometer. The pump had to overcome a head loss of approximately 75 ft and a power loss of approximately 0.85 Hp and provide the minimum flow and pressure to the dynamometer (the calculation can be found in the Appendix A). The pump that
was selected was a 1½ Hp pump with an inlet water of 1¼” and a water outlet of 1” from M’Master Carr Catalogue number 106 model #4291K65.

![Image of the final cooling water system]

Figure 3-10 The final cooling water system.

### 3.3.4 Coupler Redesign for the Roller Chassis

Since a new AC motor is being used to power the electric bobcat racecar, an overview of the previous work needed to be done to verify that all existing parts were still valid. After looking at bearings, sprockets, couplers, etc., the only thing that had to be changed was the couplers on the roller chassis. The couplers that were currently on the roller chassis were Falk 1050T10B Steelflex Couplers. These couplers had two problems with them.
When they were originally selected the required speed was 5000rpm; the new speed requirement was 8000rpm to allow the AE250 dynamometer to be used to its full capacity. Also the new motor produced a higher torque, which was greater than the torque that the couplers were rated for. The search for new couplers started with Falk and ended with Falk, because no one else had couplers that would handle the job for a reasonable price. Falk had many different types and styles of couplers to choose from, which helped the selection process.

The new coupler that was selected was a Falk G52-1010G model that can handle a torque rating of 10080in-lbs at 8000rpm. The reason that this was selected was based on two factors. The first reason was that it fit in the price range. The other reason was that it fit the design criteria, which were based on the dynamometers max torque and speed capabilities. The design was based on the dynamometer because it can handle more torque and speed than the current electric bobcat can produce. Designing the couplers around the dynamometer allows for maximum use of the chassis dynamometers. The maximum torque for the dynamometer is 10632in-lbs at 2250rpm, but due to the division of power going through each coupler, the design will allow for 70% of the torque (7443in-lbs) to go through one coupler. The reason for designing 70% of the torque to travel through one coupler is based on a conservative approach, where 70% of the torque is not expected to be transferred by one coupler. What is going on is the power is being divided up from the transmission to the two tires. Ideally each wheel will see half of the torque, but in reality this may not hold true. One wheel will see more torque than the other thus applying a larger torque to one coupler. Also one coupler may see more torque than the other due to tire slippage between the wheel and roller, this hold true in a
positive differential. See Figure 3-11. The equations for calculating the maximum torque allowable by the dynamometer are in Appendix A.

![Diagram showing the torque path through the couplers](image)

Figure 3-11 A schematic of the torque path through the couplers

### 3.3.5 Half Coupler for the Power Transfer Case

The last thing that needed to be designed before the assembly of the chassis dynamometer could occur was a half coupler that attached the dynamometer to the power transfer case. All that was needed was half of the coupler because the other half was already attached to the dynamometer. This coupler was first thought to come with the dynamometer, but after conversations with Digalog it was found that only a half coupler was supplied. They recommended that the coupler could be purchased through Applications Engineering Inc. out of Michigan. After numerous conversations with Application engineering, it was clear that the company wanted too much money to design
and build a coupler for our application. Therefore the only option that was left was to design our own coupler.

To start the design process for a coupler, the first question that was asked was to build a flexible or a rigid coupler. Ideally a flexible coupler would be ideal to all for misalignment, but to manufacture a flexible coupler would cost too much. Also looking at the coupler that came with the dynamometer, it too was a rigid coupler. To use a flexible coupler, the coupler that came with the dynamometer would have to be replaced, and with conversations with Digalog this was highly discouraged. Therefore a rigid coupler design was the only option remaining. To use a rigid coupler great detail will have to be paid attention to when aligning the two shafts is done.

The physical dimensions for the half coupler came from the dimensions given in the dynamometer pamphlet from Digalog. The dimensions of the coupler that came with the dynamometer were 210mm in diameter, with eight 10.1mm diameter holes on a 92.1mm diameter centers, and a recess of 3mm at 196.7mm diameter and greater. Also a keyway and a center hole of 44.45mm diameter had to be placed in the center of the coupler. Also a key and keyway size had to be calculated for the coupler and the power transfer shaft. See Figure 3-12 for a picture of shaft coupler, and a detailed manufacturing drawing of the coupler, key, and keyway in the shaft can be seen in Appendix C. The design calculation for the half shaft can be seen in Appendix A.

From the calculation a couple of things can be found. First looking at the key, the size of the key is found for a safety of factor of 1.25. This safety of factor was selected for two reasons. The first reason is to allow the key to withstand the stresses that it would see without failing. The other reason is to allow the key to be the first component
to break in an over load case to protect the more valuable components such as the half coupler and the dynamometer. The key is designed to be at least $\frac{3}{8}'' \times \frac{1}{4}'' \times 1\frac{1}{4}''$. Also the calculations show that the key will probably fail in bearing before shear, because of the lower factor of safety in bearing.

![Figure 3-12 The half shaft.](image)

The power transfer shaft was already designed and supplied by group two, and to change the shaft size of the power transfer case would be expensive because a new shaft and bearings will have to be ordered. The calculations show that the power transfer shaft strength is drastically reduced when a keyway groove is made in it. From the calculation we can see that the safety factors are reduced from 5.5 to 2.0 for max load, while fatigue safety factor was reduced from 8.4 to 1.7 for a 1000 cycles and from 4.7 to 0.9264 for infinite life.

Designing the coupler around the physical dimensions of the dynamometer coupler the safety factor was 26 for max load. This safety of factor is extremely high, but is not reduced on the fact that we have to start with a large piece of steel that has to be
turned down, and there is no benefit in turning it down to reach a lower safety of factor. Since the coupler is relatively expensive compared to the key and shaft, we do not want it to fail and have to be replaced frequently. The calculations show that the steel for the coupler can range from 1020 or higher, but due to the availability of 1045 steel it was selected to produce the coupler. 1045 steel has a yield strength of 55000psi and an ultimate strength of 90000psi.
4. Construction

The construction of the chassis dynamometer was done in two distinct stages. The first stage was the layout and physical connection of all the subsystems to create the chassis dynamometer. The second stage was the connection of the water system for cooling and the electrical hookup of the chassis dynamometer. To discuss the construction of the dynamometer in greater detail, I will start with the first stage.

4.1 Stage One

Before any connections of the subsystem can be done a general layout of where the chassis dynamometer is going to rest must be done. With this in mind the first task was to find a location in the lab that would allow for easy access to the chassis dynamometer, but would be out of the way when not in use. Taking these two criteria the best location in the lab for the chassis dynamometer was in the far back corner of the lab on the right side. Besides the two criteria, this location put the chassis dynamometer beside the sink to give us easy access to water for both refilling our water reservoir and dumping water from our water reservoir.

After locating the general area for the chassis dynamometer, it was time to move all the subsystems to their final resting place. The first thing that was to be moved was the roller carriage system. This was to be moved first because it was the largest piece of equipment and would take up the most amount of room. The roller carriage system location was determined by a set of parameters. The first parameter was to find a location that would allow all four corners of the subsystem to touch the ground. The importance of all four corners touching the ground is that it eliminates the need for shimming of the system. The second parameter was to place the roller carriage in a
location that would leave adequate space for the other equipment and leave accessibility
to the sink.

Once this location was found for the roller carriage system the rest of the
equipment could be placed in their final locations. The position of the power transfer
case and the load system were predetermined by their relationship with the roller
carriage. Since the power transfer case connects directly to the roller carriage and the
load system connects directly to the power transfer case their locations were found. With
the location of the equipment found, the next step was to connect the equipment together
to transform each subsystem into one working system. Figure 1-4 shows a picture of the
locations of the load system, roller carriage system, and power transfer case

The connections of the equipment start with the connection of the power transfer
system to the load system. The load system and the power transfer system are connected
by a shaft coupler. The one coupler half came with dynamometer, while the other is the
one that was designed. To connect the two half shafts using the couplers the shafts needs
to be aligned both vertically and horizontally.

The vertical aligning of the shaft is accomplished by creating a parallel plane to
the dynamometer coupler. This is by using a level to measure the rise and run in the
dynamometer to find the parallel plane to the coupler. Once the plane is established,
measurements from the shaft to the plane can be made. The difference between the plane
and shaft should be constant. Figure 4-1 shows a picture of the vertical alignment setup.
Therefore 2" aluminum square stock was placed under both power transfer case I-beams
with a 1/8" thick steel plate placed under the closes I-beam to the dynamometer. The
horizontal alignment was much simpler. Two bolts located on the same radius from the
center of rotation of the dynamometer and were located on the horizontal plane through the center of rotation of the dynamometer, then all that was needed was to measure the diagonals from the bolts to the center of the shaft. Movement of the shaft was needed until both diagonals were equal. Figure 4-2 shows a picture of the horizontal alignment system. Now the two shafts are aligned, the two shafts are connected by eight \( \frac{3}{8} \)" dia. bolts. The half shaft that I designed sits on the power transfer case shaft and is held in place and is turned by a key.

Figure 4-1 The horizontal alignment system.
Once the power transfer system and load systems are connected, then the power transfer case and roller carriage system needs to be connected. The power transfer case is connected in two fashions. The first fashion is by a belt that goes around two sprockets. One sprocket is located on the shaft of the power transfer case and the other is located on the shaft of the roller carriage system with a belt tensioner located in the center to provide adequate belt tension to prevent belt slippage. Figure 4-3 shows a picture of the belt tensioner. The tension is comprised of three components, which were ordered from Motion Industries in Lancaster, Ohio. The first two components are the MVL Vertical Mount from the Brewer Machine & Gear Co., and a SO-6 idler shaft & stud attached to
The third component is the P4F idler pulley from the same company as the previous two parts. The tensioner is strictly to increase the power transfer capability, it is not for stability or anchoring purposes. Therefore another connection is made between the power transfer case and roller carriage system for anchoring purposes. Two ½” bolts are used to connect the power transfer case and the roller carriage together, but before this can be done the shafts have to be double-checked for alignment.

![Figure 4-3 The belt tensioner.](image)

Once the alignment of the shafts is satisfied then all of the components can be anchored to the floor. Also 4 x 6 x 5/16” at 3½” long angle iron will be connected to the roller carriage and power transfer systems with ½” bolts. The angle iron is attached to
the base of the systems to allow for \( \frac{5}{8} \)" dia. x 4½" long anchor bolts to be placed in the concrete to bolt the systems to the ground to prevent the systems from moving. Figure 4-4 shows a picture of the angles anchoring the systems to the ground.

![Image of anchor angles](image)

Figure 4-4 The anchor angles that fasten the roller carriage system to the ground.

The last thing that needs to be done is to attach a tie down system to the roller carriage. The tie down system is comprised of four I-bolts attached to the frame of the roller carriage system. These I-bolts are designed to have ratchet straps connected to them to allow for proper downforce to the car to prevent slippage between the tires and rollers. Also the straps are to prevent the car from rolling off of the rollers. Figure 3-5 shows a picture of the tie down system.
4.2 Stage Two

Stage two of the construction consists of two distinctly different jobs. The first is the connection of the water pipes for cooling purposes, and the second is the electrical power and data acquisition connections. The cooling water system was very simple. All it entailed was connecting 1½" copper pipe from the water reservoir to the pump and load system, and back again. The system can be seen in Figure 3-11.

The system consists of 1½" diameter copper pipe that was soldered together.Reducers were used to convert the 1½" diameter water pipes to and from the smaller water connections of the pump and water reservoir. The water reservoir is a Rubber Maid 150gal water trough. This reservoir was selected because it was the correct size for an open recirculation system and it already had the built in pipe connection, which saved time on installation.

The second part of stage two consisted of the electrical connections. The electrical connections consisted of two things. The first was to provide the appropriate power to the system, and the second consisted of connecting the data acquisition up to the load system. The connection of these two things were very simple.

The power connection consisted of connecting three phase 208V power supply coming out of the electrical bus to a transformer to convert the voltage to 240V and 120V. The 240V branch goes to power the Dyne-Loc IV data acquisition system, while the 120V runs to a relay and then to the dynamometer. The relay acts as an automatic shut off system. The way it works is that it receives signals from the dynamometer and the data acquisition system. The signals that it receives from the dynamometer are water temperature and pressure, while the signals it receives from the controller is over/under
speed. What the relay does is shut down both the dynamometer and the car when one of these signals is sent to the relay the system is shut down to prevent damage to the dynamometer system.

The data acquisition system reads torque and speed from the dynamometer. The dynamometer uses a load cell for the torque and a 60 pulse per revolution speed sensor. These connections are made in compliance with the installation directions that came with the Dyne-Loc IV. For more information on the electrical connection of the Dyne-Loc IV refer to the manual.
5. Validation of the Chassis Dynamometer

Now that the chassis dynamometer is constructed and operational, validation of the system must be done before testing can commence. To do this the electric bobcat was first sent to another facility to be tested on a different chassis dynamometer. The test data that was collected at the other facility will be used as a benchmark in determining if our chassis dynamometer is calculating valid data.

5.1 Benchmark Testing

The benchmark testing was done on the Electric Bobcat RaceCar in Cleveland, Ohio by a company called DynoTec Motorsports Inc. DynoTec Motorsports Inc. uses an inertia dynamometer from Dyno Jet. The Dyno Jet is said to have less than 1% error in accuracy and less than 1% error in repeatability in the data that is collects. Each test run on the Dyno Jet inertia dynamometer consists of accelerating the large rollers from zero speed out to some maximum speed. The tests are usually run with the throttle wide open to determine maximum system capabilities and the user has the option of remaining in one gear throughout the test run or shifting through all gears. For our application we chose to run the car wide open in one gear, until the horsepower fell off. In our testing we ran 33 different tests with three different sets of batteries and two different controller cards through all the gears. During the test a big difference in mechanical power was found between the cards. Thus one of the cards with all of its data was neglected from this report leaving a data set of 21 runs. Figure 5-1 shows a plot of all 21 runs.
Once we completed our testing we had an array of data from both the Dyno Jet and from our onboard data collection system (Corsa). The Dyno Jet data showed the mechanical data of the electric bobcat, specifically the power at the wheel as a function of linear vehicle speed. Corsa told us the electrical power, or how much power was being drawn from the batteries as a function of time. Both of these pieces of data were needed to validate Ohio University’s chassis dynamometer.

The reason for both sets of data is based on the variations in battery condition from test to test. Each time a battery set is charged up the amount of electrical power
stored in them may change slightly. Also when the electric motor is run the batteries start to become drained and have less electrical power. These differences in electrical power affect the amount of mechanical power that the electric motor can produce. Therefore both sets of data are needed to normalize the data in efforts to account for the effects of the electrical power on the mechanical power.

As stated before, the normalization of the data was done to account for the affects that the electrical power has on the mechanical power. To accomplish the normalization the following relationship will be used (equation 5-1).

\[ n = \frac{P_m}{P_e} \]  

(5-1)

\[ n = \text{normalized data} \]

\[ P_m = \text{mechanical power} \]

\[ P_e = \text{electrical power} \]

This relationship compares the amount of mechanical power produced with respect to the amount of the electrical power available from the batteries at the same moment in time. Therefore to use this normalization equation a relationship between the mechanical and electrical power needed to be established to find the normalized power over the entire test run.

Since the mechanical and electrical power data collected was done by two different systems, no direct time correlation between the two sets of data existed. Therefore a correlation between the two sets of data had to be made based on the time of some common event identifiable in both data sets. The first correlation was to set the time frame based on when the peaks of the electrical and mechanical power occurred.
By setting the peak powers together, the time frame for the two sets of data was established. By knowing when the mechanical and electrical powers occurred with respect to their peaks a range of normalized power could be computed. Once this was done a graph of the normalized data vs. motor speed could be produced (Figure 5-2).

![Normalized Power vs. Motor Speed](image)

Figure 5-2 Graph of the normalized data vs. Motor speed.

Looking at this graph one thing sticks out at you. That is the dip that occurs in the 3000rpm range. This dip can be seen in all of the runs, but the severity of the dip varies with each run. At first this does not seem correct, because what the dip is saying is that the mechanical power lags the electrical power. Therefore another correlation had to be made between the two sets of data.

The second attempt correlated the request for power evident from both electrical and mechanical data. In other words the lag in mechanical power was eliminated by aligning the ramp-up regions in both data sets. As before once these two points were
found a time frame was established and a range of data points could be interpreted. A plot of normalized data vs. motor speed based on aligning ramp-up regions was shown for one representative test run in Figure 5-3.

![Normalized Power vs. Motor Speed](image)

**Figure 5-3** The same data as in Figure 5-2, but alignment of the data sets is with respect to the power requested instead of peak power.

Now looking at this figure the data seems to be more realistic, because the curve is nice and smooth with a nice band power band. Since this graph looked more realistic all the data from each run was normalized in this manner. The data from each run was placed into two groups. Each group of data was placed into a group depending on the gear that that run was done in. The second group that each data run was thrown into was an overall group. This overall group included all valid run data (Figure 5-4 shows a diagram of this group breakdown).
Now each data set corresponding to a particular gear was evaluated to eliminate any bad data. The first test for eliminating bad data was based on mechanical power with respect to battery charge. In other words some of the data from a run was thrown out because batteries were dying and produced mechanical power well below the other runs, which in turn would influence the overall data. This can be seen in Figure 5-5.

Looking at this data a couple of things can be noticed. The first thing is that all the runs except for the bottom three follow the same ramp path, but have different peak horsepower. The different peak horsepower can mainly be attributed to the different battery power. The second thing is that the bottom three runs are considerably different than the other runs. These three runs were done at the end of a battery run, when the batteries were dying. Therefore due to the big difference in the bottom three runs and the rest, the bottom three runs where excluded to prevent their influence in an uncertainty analysis.
Once all the invalid data was removed then an uncertainty analysis was performed on the data. The first step was to plot all the data from one run and curve fit the data in Matlab by using the least squared error method.

$$\sigma_D = \left[ \frac{\sum_{i=1}^{n} (y_i - y_{BE})^2}{n - \xi} \right]^{1/2}$$  \hspace{1cm} (5-2)

$\sigma_D$ = standard deviation

$y_i$ = actual data point

$y_{BE}$ = curve fit data point
\[ n = \text{number of data points} \]
\[ \xi = \text{number of degrees of freedom} \]
\[ \sigma_m = \frac{\sigma_D}{\sqrt{n}} \quad (5-3) \]
\[ \sigma_m = \text{standard deviation of the mean} \]

This analysis was computed for each run and once each run was computed it was put into its group as shown in Figure 5-4. The data was then put into a Matlab code to produce a graph with all of the curve fit data, which included a mean curve and upper and lower bounds based on a confidence interval of 95%. The 95% confidence interval was calculated by defining the "error" as the deviation between the curve and the data point for each rpm, then using Matlab to compute the standard deviation of the data relative to the curve fit. Once the standard deviation was calculated, it was then multiplied by a confidence interval factor. The confidence interval factor depends on the degree of confidence wanted and the number of degrees of freedom. Once confidence interval is known, then it could be added to and subtracted from the mean value of the data set to get the upper and lower bounds of the data. The following equations shows how to calculate the confidence interval and Figure 5-6 shows the results for one gear while Figure 5-7 shows the results for all the data.

\[ y_m = \frac{\sum_{i=1}^{n} y_i}{n} \quad (5-4) \]
\[ \sigma_D = \left( \frac{\sum_{i=1}^{n}(y_i - y_m)^2}{n-1} \right)^{\frac{1}{2}} \]  
(5-5)

\[ \psi = t \sigma_D \]  
(5-6)

\( \psi \) = confidence interval

\( t \) = confidence interval factor (Appendix D)

\[ U = y_m + \psi \]  
(5-7)

\[ L = y_m - \psi \]  
(5-8)

\( U \) = upper boundary

\( L \) = lower boundary

Figure 5-6 95% confidence interval for the normalized power curve for third gear by setting the power at the power request point. The solid line (bold) represents the best-fit curve, the dashed lines represent the data trends for each individual run, and the upper and lower solid lines represent the 95% confidence interval.
Figure 5-7 95% confidence interval for all the runs by setting the power at the power request point. The solid line (bold) represents the best-fit curve, the dashed lines represent the data trends for each individual run, and the upper and lower solid lines represent the 95% confidence interval.

Looking at Figure 5-6 everything looks fine, the confidence interval follows the same form as the curve fit data and each run in the third gear follows the same trend. Looking at Figure 5-7 we see something different. Figure 5-6 shows that the peak efficiency point for the motor varies between 2650rpm and around 4250rpm. In all reality this cannot happen. The motor has only one maximum efficiency point. Therefore by looking at this Figure 5-7 tells us that matching the powers up at the requested power point is not the correct way of correlating the data. Therefore we went back to matching the electrical and mechanical peak powers together to see if it made more sense. The process of calculating the uncertainties of the peak powers was done in
the same manner as in the power request data. Figure 5-8 shows the graphs of the all the runs, but with the power being matched up differently.

Figure 5-8 A piecewise function of the runs based on the alignment of peak powers. The solid line (bold) represents the best-fit curve, the dashed lines represent the data trends for each individual run, and the upper and lower solid lines represent the 95% confidence interval.

The reason for choosing a piecewise function for this set of data is because of the dip. To curve fit the dip the polynomial has to be of a high order, which causes a wavy effect in the curve. The wavy effect in the data is not a true representation of what is happening. Therefore a piecewise plot is used to keep a more realistic representation of the data by allowing the curve fit to remain as a relatively low order polynomial. In doing this, the dip region is excluded leaving the piecewise function useful only in the
1500-2500 rpm range and 4000-5000 rpm range. These two ranges are the only ranges valid to check for validation of the Ohio University's Chassis Dynamometer. All other speed ranges will be excluded for an uncertainty analysis.

As we can see here all of the curve fit data's peak efficiency falls roughly at the same point (3800 rpm). This is more realistic than having two efficiency points. Therefore this graph indicates that this is the better method of correlating the data, and there is no other valid way to correlate the two forms of data without further information.

The additional information needed to help correlate the data is the motor speed data from the Corsa electrical data. Unfortunately, the tachometer system for the Corsa data was not functioning properly during the test, so that speed data is not available for these test runs. Figure 5-9 shows a plot of the upper and lower confidence interval bounds, with the actual data plotted.

![Normalized Power vs. Motor Speed](image)

Figure 5-9 The actual data with the uncertainty bounds for the ranges of 1500-2500 and 4000-5000 for the peak power.
5.2 Validation of Ohio University’s Chassis Dynamometer

The validation of the chassis dynamometer is based on two factors. The first factor is the ability to correctly and accurately calibrate the system for both torque and speed measurements. The second factor is running some tests to accumulate data from Ohio University's chassis dynamometer to compare with the testing done at Dyno Tech. Therefore we will begin with the calibration of the chassis dynamometer.

The calibration for torque on the system is performed by hanging known weights on calibration arms out to the side of the AE250 Eddy current dynamometer. Each 25Kg plate creates a torque of 200Nm per the manufacture’s specification. With both calibration arms installed and balanced to achieve a zero torque effect (see Figure 5-10), the calibration proceeds starting with one side of the dynamometer and placing one weight at a time on that side arm. The torque measured by the Dyne-Loc IV is not the same as the known torque being applied. Therefore setting the Dyne-Loc IV to the proper torque based on the 200Nm calibration standard and repeating this for each plate that is added, the system’s torque reading will be calibrated to read the correct torque values. Then repeat the procedure for the other side. To see more information about calibration of the chassis dynamometer system, see Appendix D.

The speed sensor is already calibrated by the manufacturer and therefore no calibration is needed. But a check to see if the Dyne-Loc IV is reading correctly was done using a stroboscope to measure the dynamometer roller speed to compare the two readings. By doing this the two speed readings were within 0.5% for a random selection of speeds, validating the dynamometer speed measurements.
Once the calibration was completed then testing could begin to validate the system. The validation was done by testing the Electric Bobcat RaceCar on the chassis dynamometer to compare the results of Ohio University's dynamometer and DynoTec Motorsports Inc. dynamometer. The testing on the car with Ohio University's Eddy current dynamometer is different than that of DynoTec's inertia dynamometer in that data from our dynamometer is in the form of steady state torque values at a few speeds, while the DynoTec calculates power at many small speed increments based on the angular speed and acceleration of the rollers. The speeds selected to be tested were 1040, 2000, 3000, 4500, and 5500rpm at the motor. The following table (Table 5-1) shows the equivalent dynamometer speed for these motor speeds (which is the control variable in these tests) as well as the measured torque, and calculated horsepower at the dynamometer.

Taking this data and converting it back to the motor output instead of at the dynamometer and normalizing it we can come up with the following plots. The
normalization process done on this data was the same as for the DynoTec data. The only difference between the two sets of data is that the Ohio University dynamometer gathers one data point per run instead of a range of data points. Therefore the normalization is a little simpler with the Ohio University data than with the DynoTec data, because all that is needed is to find the maximum electrical power and set it at the same time as peak mechanical power to obtain the normalized data.

Table 5-1 The results of testing the Electric Bobcat in third gear with Ohio University’s chassis dynamometer.

<table>
<thead>
<tr>
<th>Speed (rpm)</th>
<th>Speed (rpm)</th>
<th>Measured Torque (ft-lbs) at Rollers</th>
<th>Calculated Horsepower (Hp) at Rollers</th>
</tr>
</thead>
<tbody>
<tr>
<td>1040</td>
<td>500</td>
<td>296</td>
<td>28</td>
</tr>
<tr>
<td>2000</td>
<td>960</td>
<td>295</td>
<td>54</td>
</tr>
<tr>
<td>3000</td>
<td>1440</td>
<td>286</td>
<td>78</td>
</tr>
<tr>
<td>4500</td>
<td>2160</td>
<td>168</td>
<td>68</td>
</tr>
<tr>
<td>5500</td>
<td>2640</td>
<td>115</td>
<td>57</td>
</tr>
</tbody>
</table>

Looking at these two graphs we notice two significant things. The first thing we notice is that the torque and horsepower are lower for our dynamometer than what was measured with Dyno Tech’s dynamometer. Before we investigate this difference, first let’s look at the second aspect to the data. That is the data follows the same trend as that was measured with the Dyno Tech dyno, but with an offset. Also, the data is repeatable from run to run. Therefore the data seems to be correct, but with losses due to the dynamometer system creating a bias error in the measurements (Figure 5-11 & 5-12).
To test the theory that the offset could be a battery related problem, we first need to normalize the data as before and plot it to see if the batteries are affecting the power
and torque levels, see Figure 5-13. The second set of data will be the only set of data to be normalized based on the fact that it is the only run for which Corsa electrical data was collected.

Figure 5-13 Graph of the normalized power vs. speed curve for the Electric Bobcat using Ohio University’s and Dyno Tech’s dynamometer.

Looking at this figure we see that the normalized power with Ohio University’s dynamometer is also less than that of the Dyno tech dynamometer. Since this is true then the batteries are not the reason for lower values. Trying to understand where these differences are can be difficult, but explainable. The first place to look at to understand the differences is the system itself. The Ohio University system has many more components to it than that of the DynoTec dynamometer. Therefore power and torque losses can be accounted for here, but this does not account for all the losses. Therefore another place for losses has to be looked for.
There are losses between the tires and the rollers because slipping is occurring. How do we know this? By taking stroboscope measurements of the speed of the wheels and the speed of the rollers and taking into account the gear ratio between them, we find that the wheels are moving faster than the roller. The difference in speed implies slip. The belt is new and as you apply tension the belt starts to stretch allowing for slippage to occur. Therefore tension has to be reapplied every so often. Also after the belt is retensioned the torque reading on the Dyne-Loc IV go up implying less slip.

To fully understand how much power and torque is being lost to slippage and inefficiencies a study with torque sensors placed on the different shafts needs to be done. By placing torque sensors on two different shafts, the torque loss can be calculated by taking the difference in the two readings.

Another possibility for the loss in torque and power is vibration. Vibration is happening in the system based on imbalances in shaft alignment and roller alignment. The shaft misalignment is created by the uncertainties in the alignment procedure. A better procedure to align the shafts will reduce vibration and increase torque and power. The rollers can be creating vibration by a misalignment of the shafts or a non-homogeneous weight distribution in the rollers. This offset of weight and misalignment will cause loss in torque and power. To eliminate this problem the rollers will have to be checked for alignment and weight imbalances, and then corrected by adding weight to the rollers.

Another possible reason for the difference in the torque and power results is due to tire deflection. Tire deflection will not create a physical loss in torque and power, but it affects the gearing ratio. In other words as the tire deflects the effective radius of the
tire changes causing the assumed gear ratio of 2 between the tire and roller to change causing a change in the torque and power.

Finally another difference between the torque and power for the two cases could be based on the controller itself. The controller can control the amount of torque that the motor is able to produce and in turn the amount of horsepower (see Figure 5-14). Since our dynamometer tests steady state values instead of transient values as the DynoTec dynamometer does, the controller may not allow torque to reach its maximum potential. Therefore from Figure 5-14 we can see that the power and torque can be reduced from what is expected and seen from testing at DynoTec. Since not a lot is known about the controller, and what is actually happening with the motor and what mode of operation the motor is in is unknown. Therefore this issue will need to be explained and looked at in greater detail with the help of SatCom (is the designers of the controller).
Figure 5-14 A plot of the horsepower vs. speed of the motor as the controller is controlling the torque.
6. Conclusion

At the beginning of this project was the idea to design, develop, and construct a fully operational chassis dynamometer for the use of testing automotive vehicles. Now that this goal has been accomplished what are the capabilities of this chassis dynamometer. In our chassis dynamometer system, the capabilities of the load system are greater than what the whole chassis dynamometer is certified for. In other words our chassis dynamometer is only rated and certified to the level of the weakest component. What is our chassis dynamometer certified to do?

Before certifying the chassis dynamometer a look and understanding of the components must be made to understand the maximum capabilities of the system. To do this we will first look at the maximum load absorption capabilities of the AE250 Eddy Current Dynamometer, and then list all the components that are less than the AE250 capabilities. This can be seen in Table 6-1.

Table 6-1 A table of all the components that are not up to the maximum capabilities of the load system when the system is being run as a chassis dynamometer.

<table>
<thead>
<tr>
<th>Component</th>
<th>Max Speed (rpm)</th>
<th>Required Speed (rpm) for full capability</th>
<th>Max Torque (in-lbs)</th>
<th>Required Torque (in-lbs) for full capability</th>
</tr>
</thead>
<tbody>
<tr>
<td>AE250</td>
<td>8000</td>
<td>n/a</td>
<td>10632 @ 950rpm</td>
<td>n/a</td>
</tr>
<tr>
<td>1050T10B Coupler</td>
<td>4500</td>
<td>8000</td>
<td>3850</td>
<td>7442</td>
</tr>
<tr>
<td>G52 Coupler</td>
<td>8000</td>
<td>8000</td>
<td>10080</td>
<td>7442</td>
</tr>
<tr>
<td>Bearings</td>
<td>5000</td>
<td>8000</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Sprockets and Belt</td>
<td>4500</td>
<td>8000</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

From this table we can see that the chassis dynamometer is certified to run between 0-4500rpm at the load system(corresponding to a linear speed of 0-160mph for the Electric Bobcat Race Vehicle), with a maximum torque of 3850 in-lbs at 950rpm with
the 1050T10B couplers. Once the G52 couplers are installed the chassis dynamometer torque validation will be increased to 10632inlbs. Besides the maximum capabilities of the system being certified, the chassis dynamometer system in its current state is also validated for relative testing, but not for absolute testing until the power losses causing the measurement bias can be found and quantified or eliminated.

The reason for relative testing is based on the offset in torque readings. If we look at Figure 5-11 we can see that an offset in the torque readings exists. Although an offset exists, the data collected from Ohio University’s dynamometer and DynoTech dynamometer follow the same trends. Since the two sets of data follow the same trend with just an offset in torque readings, we know that the torque losses prevent us from measuring the absolute torque of a vehicle. Instead it allows us to measure the torque that a vehicle produces minus the torque losses due to the chassis dynamometer. Although relative testing will not allow us to test what the actual torque of a vehicle is, it will allow us to test how the performance of a car is affected by design changes in the vehicle.

The next question is what can be done to improve the system. In fact a couple of things can be done to improve the system by increasing the capabilities of the system and enabling its automotive test capabilities.

- Locate and Identify Torque/Power Losses
  - Belt System
  - Wheel Slippage
  - Roller Imbalances
• Retest Ebobcat with Working Tachometer to Verify Best Normalization Procedure
  • Peak Electrical/Mechanical Power
  • Request Power
• Replacement of Belt System & Upgrade of Components
• Automate the Water Cooling System
• Connect Shut Off System for the vehicle
• Loading/Unloading System
• Converting the Chassis Dynamometer into a Engine and Transmission Dynamometer

The first thing that needs to be done is to do some testing to find out where the torque and power losses in the system are located. By finding these losses then the dynamometer can become more accurate by taking into consideration these losses. Also once these losses are found then steps can be taken to eliminate the losses.

The torque losses come from various systems and problems. The first problem is the efficiency of the belt system. Usually belt systems are on the magnitude of 96-98% efficient.[11] With this in mind an alternative system to transfer the power from the rollers to the power transfer case could take care of this problem and recoup some of the losses. Belt losses are speed dependent and increase as belt speed increases.

Another area where torque is lost is due to tire slippage between the rollers and the tires of the vehicle. Slippage was verified by taking a stroboscope and measuring the speed of the wheels and rollers. Since slippage occurs torque is being lost. Methods for preventing include knurling the rollers or placing a knurl polymer around the rollers to increase the coefficient of friction to increase the slip limit in efforts to eliminate slip.
Also roller imbalances are another source of torque losses. Roller imbalances create vibrational loads that are transmitted to the ground causing torque losses. The more out of balance the rollers are the more the torque that is being lost. Like the belt system the vibrational losses are speed dependant, where the vibration worsens as the speed of the rollers increases until the rollers reach their resonance. To correct the imbalance problem with the rollers, they need to be taken off the roller carriage and dynamically balanced.

Another area where torque is being lost is by the bearings due to friction. Like the belt system and roller imbalances frictional losses are speed dependent. In other words as the speed of the rollers increase the torque losses due to frictional losses increase. There is not much that can be done to correct this problem, except possibly replacing the existing frictionless bearings with even lower frictional bearings. Table 6-2 shows possible magnitudes of torque losses due to these four factors just discussed, that would fully explain the torque offset between the DynoTec data and the Ohio University chassis dynamometer data.

Table 6-2 Indicates torque losses of the chassis dynamometer. The points are relative to Figure 5-11, where point I is the first data set and point IV is the fourth data set starting from left to right.

<table>
<thead>
<tr>
<th>Point I</th>
<th>Point IV</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel Slippage = 15ftlbs</td>
<td>Wheel Slippage = 0ftlbs</td>
</tr>
<tr>
<td>Belt Inefficiency = 3ftlbs</td>
<td>Belt Inefficiency = 5ftlbs</td>
</tr>
<tr>
<td>Vibrational Losses = 3ftlbs</td>
<td>Vibrational Losses = 30ftlbs</td>
</tr>
<tr>
<td>Frictional Losses = 4ftlbs</td>
<td>Frictional Losses = 10ftlbs</td>
</tr>
<tr>
<td>Total Losses = 25ftlbs</td>
<td>Total Losses = 45ftlbs</td>
</tr>
</tbody>
</table>

Looking at table 6-2 we see two things. The first thing that is seen it wheel slippage from point I to point IV, which goes from 15ftlb to 0ftlbs. The reason for this is
that the torque slip limit was stated to be at 150ftlbs. Since the torque limit was set at 150ftlbs that is why point I had wheel slippage losses and point IV didn’t. The second thing that we noticed was that the remaining three losses, belt inefficiency, vibrational losses, and frictional losses, all increase as the speed increase. This is based on the fact that all three of these losses are speed dependant, not torque dependant. In other words as the speed increases so does the torque losses. This table is not an approximation of torque losses, but rather a representation of torque losses.

As besides locating the torque losses it would be beneficial to retest the electric bobcat with a tachometer. Retesting the electric bobcat with a tachometer would allow us to verify which normalization procedure is the best method, normalizing based on peak powers or normalizing based on power request. A tachometer allows us to correlate torque data from the dynamometer and Corsa, instead of just the dynamometer.

Another improvement of the system would be to increase the capabilities of the system. The first thing would be to upgrade the bearings and belt system to a level of the maximum load absorption capabilities of the AE250 listed in Table 6-2. By doing this the chassis dynamometer system could be increased from a range of 0-4500rpm to 0-8000rpm, allowing the system to measure a larger range of vehicle capabilities. Doing this also requires a higher precision in aligning shafts, and to do this a transit is recommended. Also the design and addition of a system to prevent the rollers from rolling would be of great benefit in putting a vehicle on and off the chassis dynamometer.

Besides upgrading components another feature that could be implemented is to allow testing of engines and electrical motors without running them through the drive train. This can be done by attaching the engine/electrical motor to the other end of the
power transfer case shaft. To do this a coupler will need to be designed to attach the motor shaft to a coupler that connects to the power transfer case shaft. Figure 6-1 shows a diagram of this system and table 6-3 shows a table of components needed for upgrade. The flexible coupler allows for misalignment of the two shafts, and if only low speed testing (<4500rpm) is required the coupler that could be used is the 1050T10B coupler that will be replaced on the roller carriage system.

![Diagram](image)

**Figure 6-1 Engine direct connect system.**

Table 6-3 A table of all the components that are not up to the maximum capabilities of the load system when the system is being run as a motor/engine dynamometer.

<table>
<thead>
<tr>
<th>Component</th>
<th>Max Speed (rpm)</th>
<th>Required Speed (rpm) for full capability</th>
<th>Max Torque (in-lbs)</th>
<th>Required Torque (in-lbs) for full capability</th>
</tr>
</thead>
<tbody>
<tr>
<td>AE250</td>
<td>8000</td>
<td>n/a</td>
<td>10632 @ 950rpm</td>
<td>n/a</td>
</tr>
<tr>
<td>1050T10B Coupler</td>
<td>4500</td>
<td>8000</td>
<td>3850</td>
<td>1980</td>
</tr>
<tr>
<td>Bearings</td>
<td>5000</td>
<td>8000</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Besides these improvements automation improvements could be made in the cooling water system. The cooling water system could be converted from a manual operation to a computer control system. By implementing control parameters such as exit
water temperature, the computer can automatically control the amount of recirculating water to fresh water instead of a person doing it. This will reduce the amount of people needed to run a test.

Also right now we have a safety shut off system for the dynamometer, but there is not one for the vehicle. A system is already in place with the relay box, but questions remain about how to tie the relay box into the controller to shut the vehicle off. This is something that may need to be explored by the Electric Bobcat Race Team for the Electric Bobcat RaceCar.

Another project that should be done to make it easier to load and unload the car onto the dynamometer is the design of ramps, placement of a winch, and a lock down mechanism for the roller. The ramps will allow for greater ease of loading and unloading cars. The ramps that are in place now are very crude, but do work. A winch, which can be used to pull a car off or onto the rollers, will provide a safety measure by eliminating people from having to push the car. Also a winch will reduce the number of people needed. The last thing and probably the most important for loading is the addition of a roller lock. A roller lock will prevent the roller from spinning when the car is being loaded and unloaded. This is important because the car wants to spin off the rollers when it is being placed onto the dynamometer or taken off of the dynamometer.

Another point that needs to be done is verify that the values on the calibration arms are correct by testing them. The assumption was made that the calibration value of 200Nm stated on a plate attached to the torque arm is correct. Digalog was not able to verify that these numbers were correct, but the answer received was that they should be.
Therefore a check should be made on the arms to verify that the calibration value is correct.

The last thing that needs to be done is to get the tachometer in the electric bobcat working to get a new data set and recorded by Corsa. By recording tachometer data then the normalization process can be reevaluated to see which procedure is the best procedure, matching the data up by peak powers or by matching up the requested powers. By doing this, one method of normalizing can be selected as the correct method to set a benchmark for how the process should be done. This also gives the ability to compare losses between different tests and design changes made on a vehicle with certainty that the normalization is correct.

In conclusion, I feel that all the goals that where established for this project have been met and accomplished. The chassis dynamometer is operational and ready to be used for relative system testing up to its certified limits (Appendix D shows the calibration and operational procedures). The components (Appendix E shows a price assessment of the project) that need to be replaced to achieve maximum testing capabilities of the system have been identified, and possible reasons for the offset in torque values from the benchmark data have been discusses.
Bibliography


A.1 Water Cooling System

One of the more important subsystems to the chassis dynamometer system is the water-cooling system. The system is comprised of a reservoir, pump, piping system, and instrumentation equipment. The water-cooling subsystem does not come with the dynamometer; instead it has to be designed by the individual users. Therefore to design this system a couple of things need to be considered. The first thing is a conceptual idea of how the system will be laid out and work. The second is to design and "spec out" all the parts and accessories to meet the manufacturer’s specifications to prevent damage to the dynamometer.

The conceptual design is for pump-to-pump water from a 150gallon reservoir to the dynamometer by a piping system. The reservoir will be resupplied with fresh water from a hose connected to the sink and the water leaving the pump is either recirculated or dumped down a sink drain. Gauges will also be place in the system to verify that adequate water flow and pressure is maintained. This design can be seen in Figure A-1.

Figure A-1 A diagram of the cooling water system.
Once the conceptual design for the cooling water system is done then the design of the individual components can be done. Knowing the pipe size for the dynamometer is 1.5625” in diameter and that the reservoir consists of a water trough, then the only thing left to design is the pump. The design of the pump will be based on the dynamometer manufacturer’s water supply specifications. The water specifications are to be able to supply 45gal/min of water at 30psi for a maximum power condition. Knowing the specification then the next step is to figure out the correct size pump for the job. The following equations will govern the size of pump. The first equation is Bernoulli’s equation. ([5] all equations come from this resource)

\[
\frac{\text{Pr}_{\text{out}}}{\gamma} + \frac{V_{\text{out}}^2}{2g} + z_{\text{out}} = \frac{\text{Pr}_{\text{in}}}{\gamma} + \frac{V_{\text{in}}^2}{2g} + z_{\text{in}} + h_s - h_L
\]  

(A-1a)

Where:

- \(\text{Pr}\) = the pressure of the water
- \(V\) = velocity of the water
- \(z\) = height difference between the inlet/outlet with respect to a fixed datum
- \(h_s\) = the amount of head that a pump supplies
- \(h_L\) = the amount of head loss due to friction, bends, inlets/outlets, and etc.
- \(\gamma\) = specific weight of water
- \(g\) = gravity

Subscripts:

- Out = water conditions at the dynamometer inlet
- In = water conditions at the reservoir
Before plugging in the numbers a couple of things need and can be done. First thing, we can simplify Bernoulli's equation by making some assumptions. The first assumption is to make the \( z \) on both the out and in zero. This can be done by stating that the relative height between the datum point and the inlet or reservoir is insignificant and can be excluded. Another assumption that can be made is that the reservoir is a large open tank. By doing this we can say that the water pressure and velocity at the reservoir is zero, because it is atmospheric. Therefore equation A1-1 can be rewritten as.

\[
h_s = \frac{P_{\text{out}}}{\gamma} + \frac{V_{\text{out}}^2}{2g} + h_L
\]

The gravity is a constant (32.21bs/ft/\(^2\)) and specific weight can be looked up in tables. Assuming room temperature for water the specific weight is 62.4lbs/ft\(^3\). By the manufacturer's specification the required water pressure and velocity are also known. The pressure is 30psi (4320psf). The velocity is not given to us directly, but rather indirectly. We know the volumetric flow rate, and by knowing that we can calculate the water velocity by using the following equation.

\[
V_{\text{wa}} = \frac{Q}{\pi r^2}
\]

Where:

\( V_{\text{wa}} \) = water velocity

\( Q \) = volumetric flow rate (45gal/min or 0.1ft\(^3\)/s)

\( r_p \) = radius of the water line (0.78125in or 0.0651ft)

\[
V_{\text{wa}} = \frac{0.1 \text{ ft}^3/\text{s}}{\pi (0.0651 \text{ ft})^2}
\]
$V_{wa} = 7.51 \text{ ft/s}$

Now everything is known except for the head loss. The head loss is calculated by the following equation.

$$h_L = f \frac{l}{d} \frac{V_{wa}^2}{2g} + \sum i k_{L_i} \frac{V_{wa}^2}{2g}$$

(A-3)

Where:

- $h_L =$ head loss
- $f =$ roughness equivalent or smoothness of pipe surface
- $l =$ length of water line
- $d =$ diameter of water line
- $V_{wa} =$ velocity of water
- $g =$ gravity
- $k_{L_i} =$ loss coefficients for bend, inlets, outlets, and valves

To calculate the head loss we have to look up all the loss coefficient values for the different aspects of the piping. The $h_L$ values taken were considered for worst case. By doing this the different $h_L$ values are as follows: three water outlets at an $h_L$ of 0.5 each, two water inlets at an $h_L$ of 1.0 each, three 90° bends at an $h_L$ of 0.3 each, and ball valve at an $h_L$ of 0.05 at fully open. The water lines roughness equivalent is equal to 0.000005 with the pipes running 20ft and having a diameter of 1.5625”. Knowing this we can stick these numbers into equation A-3 to get the head loss.

$$h_L = 0.000005 \frac{20 \text{ ft}}{0.0651 \text{ ft}} \frac{7.51 \text{ ft/s}^2}{2(32.2 \text{ ft/s}^2)} + [3(0.5) + 2(1.0) + 3(0.3) + 0.5] \frac{7.51 \text{ ft/s}^2}{2(32.2 \text{ ft/s}^2)}$$
\[ h_L = 4.29 \text{ ft} \]

Now that all the values are known we can stick them into Bernoulli's equation to get the pump size.

\[ h_s = \frac{4320 \text{ lbs/ft}^2}{62.4 \text{ lbs/ft}^3} + \frac{7.51 \text{ ft/s}^2}{2(32.2 \text{ ft/s}^2)} + 4.29 \]

\[ h_s = 74.39 \text{ ft} \approx 75 \text{ ft} \]

To convert a pump size from head loss to feet to horsepower the following equation can be used.

\[ W = \frac{\lambda Q h_s}{550} \quad \text{(A-4)} \]

\[ W = \frac{62.4 \text{ lbs/ft}^3 (0.1 \text{ ft}^3/s)(75 \text{ ft})}{550 \frac{\text{lbs-ft/s}}{\text{Hp}}} \]

\[ W = 0.85 \text{ Hp} \]

Once these calculations were completed then the proper search for the proper size pump could be made. The pump that is needed for this application requires a minimum of 0.85 Hp with the ability to overcome 75 ft of head loss. Looking at most pumps the horsepower is rated in 0.25 Hp increments, so the actual pump selected was a 1\(\frac{1}{2}\) Hp pump from McMaster Carr order #4291K65.
A.2 Half Coupler Design

As stated earlier in the paper, only one half of the coupler that connects the dynamometer to the power transfer case was supplied by the dynamometer manufacturer (Digalog Inc.). After numerous attempts to find a company that would design and build a suitable coupler for our application failed, only one option remained. This option was to design our own and have a machine shop make the coupler. The design of the coupler was centered around the physical dimensions of the coupler supplied by Digalog Inc. This was done to insure proper fit and alignment with the two couplers. The physical dimensions of the coupler can be seen in Appendix C.

During the design of the half shaft several things had to be considered, including the material to be used, the dimensions, and how to attach the coupler to the power transfer case shaft. The use of a key was decided to attach the coupler to the power transfer case shaft. Therefore all that remained was the material selection and part dimensioning, which would come from the design calculations. To start the design calculations a few physical dimensions had to be guessed in order to do the calculations. If the physical dimensions were not large enough to withstand the stresses a change in the dimensions or the material or material properties could be made to allow the coupler to withstand the stress. The first thing that was looked at was the dimensions that were set by either existing equipment or the dynamometer mating coupler. Looking at these two criteria we find that the outside diameter had to be 8in, with a bore running through the center at 1 3/4” in diameter. Besides these two criteria everything else is flexible.
Therefore a neck down in the coupler from 8" to 4" in diameter with ½" fillet radius was selected. Once this was considered the design calculation could start.

The first thing that needs to be done is calculating the stresses in the coupler. A free body diagram will show the forces and how the forces are applied to this system. From the free body diagram only one force can be seen and it is torque through a rotating shaft. Therefore the shaft is in pure torsional shear and no other stresses will be assumed. ([10] all equation for this section will come from this resource)

\[
\tau_{\text{Max}} = \frac{T r_i}{J} \tag{A-5}
\]

Where:

\(\tau_{\text{max}}\) = the max torsional shear stress applied to the coupler

\(T\) = torque applied to the coupler

\(r_i\) = radius from the center of the specimen to stress location to be inspected

\(J\) = second mass moment of a specimen a geometric property of the shaft that can be calculated

\[
J = \pi \frac{D_o^4 - D_l^4}{32} \quad \text{for a round hollow specimen} \tag{A-6}
\]

\[
J = \pi \frac{D_o^4}{32} \quad \text{for a solid round specimen} \tag{A-7}
\]

Where:

\(D_o\) = outside diameter of specimen

\(D_l\) = inside diameter of specimen
In our case for the coupler the \( r = 2'' \), \( D_0 = 4'' \), \( D_1 = 1\frac{3}{4}'' \), and \( T = 10600\text{lb-in} \). The torque comes from the max torque absorption capabilities of the dynamometer.

\[
J = \pi \frac{4in^4 - 1.75in^4}{32}
\]

\[ J = 24.21\text{in}^4 \]

\[
\tau_{\text{Max}} = \frac{10600\text{in}\cdot\text{lbs}(2\text{in})}{24.21\text{in}^4}
\]

\[ \tau_{\text{Max}} = 876\text{psi} \]

Now that the shear stress is known, we can calculate the maximum effective stress from Von Mises effective stress calculation for pure shear.

\[
\sigma' = \sqrt{3} \lambda_{\text{Max}}
\]  \hspace{1cm} (A-8)

Where:

\[ \sigma' = \text{effective stress in the coupler} \]

\[ \tau_{\text{max}} = \text{the max torsional shear stress applied to a coupler} \]

The max effective stress in our case is.

\[
\sigma' = \sqrt{3}(876\text{psi})
\]

\[ \sigma' = 1517\text{psi} \]

Now that we know the max stress, we can also find the factor of safety with respect to max load failure by the following expression.

\[
\eta_{\text{Shear}} = \frac{Sy}{\sigma'}
\]  \hspace{1cm} (A-9)

Where:
\eta_{\text{shear}} = \text{factor of safety}

Sy = \text{yield strength of the material}

For our material we decided to use a hot rolled 1045 steel with a $Sy = 55000\psi$ and a $Su = 90000\psi$.

\[ \eta_{\text{shear}} = \frac{55000\psi}{1517\psi} \]

\[ \eta_{\text{shear}} = 36.25 \]

Besides this a stress concentration factor has to be taken into consideration for the neck down from 8" to 4" with a necking radius of 0.5".

\[ \sigma_{\text{max}} = K_T \sigma' \] \hspace{1cm} \text{(A-10)}

Where:

\[ \sigma_{\text{max}} = \text{maximum stress that the coupler will undergo during loading} \]

\[ K_T = \text{stress concentration factor} \]

\[ \sigma' = \text{Von Mises effective stress} \]

\[ K_T \cong A\left(\frac{R_n}{d}\right)^B \] \hspace{1cm} \text{(A-11)}

Where:

\[ K_T = \text{is the stress concentration factor} \]

\[ A & B = \text{are given values based on the neck down ratio} \]

\[ R_n = \text{radius of neck} \]

\[ d = \text{diameter of reduced section after being necked down} \]
A & B can be figured out by the solving for the following equation and then looking up the appropriate values off a table.

\[
A & B = \frac{D}{d}
\]  \hspace{2cm} (A-12)

Where:

\[D = \text{diameter of coupler before neck down}\]

\[d = \text{diameter after neck down}\]

\[A & B = \frac{8in}{4in} = 2\]

\[A = 0.86331\]

\[B = -0.23865\]

\[K_T \equiv 0.86331 \left(\frac{0.5in}{4in}\right)^{-0.23865}\]

\[K_T \equiv 1.4\]

\[\sigma_{max} = 1.4(1517) = 2123\text{psi}\]

Besides the max load another important mode of failure is fatigue, so therefore we also need to look at how fatigue may cause the coupler to fail. We will take two approaches to look at fatigue: first 1000 cycles (which is an estimation of the number of uses of the equipment over its lifetime) and second infinite life. The following equations will allow for the calculation of fatigue at 1000 cycles.

\[S_{m10^3} = 0.9Su\]  \hspace{2cm} (A-13)

Where:
\( S_{m10^3} \) = the allowable stress in a material to allow for 1000 cycles

\( Su \) = ultimate strength of material

\[ S_{m10^3} = 0.9(90000 \text{ psi}) = 81000 \text{ psi} \]

Then a correction factor is needed to take into account the effects of mean stress that the coupler will see.

\[
S_{m_{\text{corr}}} = \frac{-S_{m10^3}}{Su} \sigma'_{\text{mean}} + S_{m10^3}
\]

Where:

\( S_{m_{\text{corr}}} \) = the correct allowed stress for the number of required cycles

\( S_{m10^3} \) = the allowable stress in a material to allow for 1000 cycles

\( Su \) = ultimate strength of the material

\( \sigma'_{\text{mean}} \) = mean effective stress

For our case since we have repeated loading our \( \sigma'_{\text{mean}} = 0.5 \sigma_{\text{max}} - \sigma'_{\text{alt}} \).

\[
S_{m_{\text{corr}}} = \frac{-81000 \text{ psi}}{90000 \text{ psi}} \cdot 1062 \text{ psi} + 81000 \text{ psi}
\]

\( S_{m_{\text{corr}}} = 80045 \text{ psi} \)

Now the safety factor can be calculated for cycle of life 1000.

\[
\eta_{10^3} = \frac{S_{m_{\text{corr}}}}{\sigma'_{\text{alt}}}
\]

Where:

\( \eta_{10^3} \) = factor of safety

\( S_{m_{\text{corr}}} \) = the corrected allowed stress for the number of required cycles
\( \sigma'_{\text{Alt}} = \text{alternating effective stress} \)

\[ \eta_{10} = \frac{80045 \text{ psi}}{1062 \text{ psi}} = 75.4 \]

Now we need to do the same thing for the infinite life. Infinite life calculations are similar to the calculations for 1000 cycles, but it is different by taking other effects into account such as environmental and manufacturing.

\[ S_{e_{\infty}} = 0.5 S_u C_{\text{load}} C_{\text{size}} C_{\text{surface}} C_{\text{temp}} C_{\text{reliab}} \]  \hspace{1cm} (A-16)

Where:

- \( S_{e_{\infty}} \) = the correct allowed stress for the infinite life
- \( S_u \) = ultimate strength of material
- \( C_{\text{load}} \) = correction factor for the type of load
- \( C_{\text{size}} \) = correction factor for the specimen size
- \( C_{\text{surface}} \) = correction factor for the couplers surface
- \( C_{\text{temp}} \) = correction factor for the temperature operation of the coupler
- \( C_{\text{reliab}} \) = correction factor for the reliability of the coupler

The correction factors are determined by many different means[10]. Some of the correction values are obtained by equation, others off of graphs or table, and others by value depending on a classification. For example \( C_{\text{load}} \) and \( C_{\text{temp}} \) = 1 due to the classification they fall in. For \( C_{\text{load}} \) it is one because it is in torsion and the critical stress is only the surface, while \( C_{\text{temp}} \) is one because it works in temperatures less than 450°C. The others have to be either calculated or looked up off of graphs.

Since the specimen is 4”, than the equation to calculate the \( C_{\text{size}} \) is as follows.
\[ C_{size} = 0.869D^{-0.097} \quad (A-17) \]

Where:

\[ D = \text{diameter of the coupler} \]

\[ C_{size} = 0.869(4)^{-0.097} = 0.76 \]

Next we need to find the \( C_{\text{surf}} \) off a chart and is found to be 0.9 for machined surfaces. The only correction factor remaining is \( C_{\text{reliab}} \), which is found off a chart to be 0.659 for a 99.999% reliability. Now having all the correction data we can solve for \( Se \).

\[ Se_{\infty} = 0.5(90000 \text{ psi})(1)(0.76)(0.9)(1)(0.659) \]

\[ Se_{\infty} = 20275 \text{ psi} \]

Now we need to make a correction to this calculation (based on the modified Goodman diagram to account for the mean stress), same as before.

\[ Se_{\text{corr}} = \frac{-Se_{\infty}}{Su} \sigma'_{\text{mean}} + Se_{\infty} \quad (A-18) \]

\[ Se_{\text{corr}} = \frac{-18957 \text{ psi}}{90000 \text{ psi}} -1062 \text{ psi} + 18957 \text{ psi} \]

\[ Se_{\text{corr}} = 200036 \text{ psi} \]

The factor of safety for infinite life is as follows.

\[ \eta_{\infty} = \frac{Se_{\text{corr}}}{\sigma'_{\text{Alt}}} \quad (A-19) \]

Where:

\[ \eta_{\infty} = \text{factor of safety} \]
Now that the calculations for the coupler are done, calculations for a key need to be done to size a key to join the power transfer shaft and the coupler together. Due to the extreme over sizing of the coupler the keyway will not play much of an effect on the coupler, but the coupler calculation does need to be redone to verify that the keyway will not affect the coupler. The recalculation will not be done in this section, but will be done in the computer program. The only change in the calculations will be the inside diameter, instead of being 1¼" will be 1.875".

The sizing of the key is based on a lot of the same issues as for the coupler. The key material and size all have an effect on the design. For the key we started out with a cold rolled 1020 steel with a \( \text{Sy} = 57000 \text{psi}, \) \( \text{Su} = 68000 \text{psi}, \) and a safety of factor of 1.25. This safety of factor was chosen because we wanted the key to fail before anything else, since they are easiest to replace and are inexpensive compared to the coupler and shaft. The first thing we have to calculate in designing a proper key is the max stress that will be applied to the key. The force being applied to the key needs to be known.

\[
F_k = \frac{T}{0.5D_s} \quad (A-20)
\]

Where:

\( F_k = \text{force applied to the key} \)

\( T = \text{is the torque on the shaft} \)

\( D_s = \text{diameter of shaft} \)
The shaft diameter is $1\frac{3}{4}''$ and the maximum torque is the same as the torque on the coupler 10600 lb-in.

$$F = \frac{10600 \text{in} \cdot \text{lbs}}{0.5(1.75\text{in})} = 12144 \text{lbs}$$

As in the coupler, we also need to check for fatigue in the key. Since we are not designing the key for infinite life, we will only have to calculate for 1000 cycles. Fatigue loading will occur in the key when there is a change in angular velocity. This change will cause a change in loading between the key and the keyway grooves, and if the angular velocity is at a steady state there will be no fluctuating stress to cause fatigue. The equations to calculate fatigue will be equations A13-A15.

$$S_{n10^4} = 0.9S_u \quad (A-13)$$

$$S_{m10^4} = 0.9(6800 \text{psi}) = 61200 \text{psi}$$

$$S_{m_{corr}} = \frac{-S_{m10^4}}{S_u} \sigma'_{\text{mean}} + S_{m10^4} \quad (A-14)$$

$$S_{m_{corr}} = \frac{-61200 \text{psi}}{6800 \text{psi}} 22800 \text{psi} + 61200 \text{psi} = 40680 \text{psi}$$

$$\eta_{10^4} = \frac{S_{m_{corr}}}{\sigma'_{\text{Alt}}} \quad (A-15)$$

$$\eta_{10^4} = \frac{40680 \text{psi}}{22800 \text{psi}} = 1.8$$

We can start to calculate the size of the key. The key is undergoing two types of stress. The first one is shear and the second is a bearing stress. Shear stress acts on the length and width plane of the key, while the bearing stress act on the length and half
height plane of the key. See Figure A-2. To solve for the key size we will first need to solve for the shear stress by assuming that the key has a width of 0.375".

Figure A-2 The shear and bearing planes on a key.

From a derived version of Von Mises equation A-8 we get the following equation.

\[
\sigma' = \sqrt[3]{\left( \frac{F}{A_{shear}} \right)^2} \tag{3-21}
\]

Where:

\( A_{shear} = \text{shear area on the key} \)

By rewriting this equation and setting \( \sigma' \) equal to the maximum allowable stress for a maximum load safety factor equal to 1.25 (a 1000 cycle fatigue factor safety of 1.8), we can get the following expression.

\[
A_{shear} = \frac{F}{\sqrt[3]{\frac{\sigma_{Max}^2}{3}}}.
\]
Since we know what area shear acts on the key, we can solve for the length \( l \) of the key.

\[
l = \frac{A_{\text{shear}}}{w}
\]  

(3-22)

\[
l = \frac{0.460in^2}{0.375in} = 1.23in
\]

Making the key a length of 2.25”, which is greater than 1.23” because 1.23” is a minimum length that the key has to be, and plugging this number into the factor of safety equation we come up with the following result.

\[
\eta_{\text{shear}} = \frac{Sy}{\sqrt{\frac{F^2}{hw}}}
\]  

(3-23)

Where:

\( \eta_{\text{shear}} \) = factor of safety in shear

\( Sy \) = yield stress of the material

\( F \) = the max shear force

\( w \) = width of key

\[
\eta_{\text{shear}} = \frac{57000 \text{ psi}}{\sqrt{\frac{12114lbs}{2.25in(0.3875in)}}^2} = 2.37
\]
Now that we know the length, we can solve area of the bearing stress to solve for
the height of the key.

\[ A_{\text{Bearing}} = \frac{F_{\text{max}}}{\sigma_{\text{max}}} \]  \hspace{1cm} (3-24)

Where:

\( A_{\text{Bearing}} \) = the area in which the bearing force is being applied to

\[ A_{\text{Bearing}} = \frac{12114\text{lbs}}{45600\text{psi}} = 0.266\text{in}^2 \]

The area is comprised of the width and height.

\[ h = \frac{A_{\text{Bearing}}}{l} \]  \hspace{1cm} (3-25)

Where:

\( h \) = height of the key

\( l \) = length of the key

\[ A_{\text{Bearing}} = \] the area in which the bearing force is being applied to

\[ h = \frac{0.266\text{in}^2}{2.25\text{in}} = 0.118\text{in} \]

The height value calculated is only half the height and needs to be multiplied by 2
to get the total height of the key, which is 0.236". Also now we can solve for the factor
of safety in bearing.

\[ \eta_{\text{bearing}} = \frac{S_y}{F} \frac{1}{lh} \]  \hspace{1cm} (3-26)

Where:
\( \eta_{bearing} = \text{factor of safety in bearing} \)

\[
\eta_{bearing} = \frac{57000 \text{psi}}{12114 \text{lbs} \times \frac{2.25\text{in}(0.118\text{in})}{\text{lb}}} = 1.25
\]

Now the key is calculated the Power transfer case shaft needs to be evaluated. Unlike the coupler and key the shaft is already purchased and installed. Therefore all we need to do is check to see the factor of safety for the shaft with the given loads. For the shaft calculations we will look at the shaft after it has been notched out for the key. The shaft calculation will look very similar to those of the coupler, but before we can start the calculations we have to give you the background on the shaft. The shaft is 1045 steel. The actual type of 1045 steel is unknown, so the lowest values for 1045 will be used, which are \( S_y = 55000 \text{psi} \) and \( S_u = 90000 \text{psi} \) at a diameter of \( 1\frac{3}{4}'' \). The keyway will reduce the diameter to from \( 1\frac{3}{4}'' \) to \( 1\frac{1}{2}'' \) for calculation purposes.

First thing we need to find is the second mass moment for the shaft, equation (A-7).

\[
J = \pi \frac{D^4}{32}
\]

\[
J = \pi \frac{1.5^4}{32} = 0.497 \text{in}^4
\]

Then the maximum unconcentrated shear stress can be found by equation A-5 and the effective stress can be found from equation A-8. The factor of safety can be found from equation A-12.

\[
\tau_{\text{Max}} = \frac{Tr}{J}
\]

(A-5)
As in the coupler, fatigue also has to be considered for the shaft. The shaft will be looked at both for 1000 cycles and at infinite life. The equation will be governed once again by the same equation that governed the coupler with a few additions.

\[ \tau_{\text{Max}} = \frac{10600 \text{in} \cdot \text{lbs}(0.75\text{in})}{0.497\text{in}^4} = 15996 \text{psi} \]

\[ \sigma' = \sqrt{3}\lambda_{\text{Max}} \]

\[ \sigma' = \sqrt{3}(15996) = 27705 \text{ psi} \]

\[ \eta_{\text{shear}} = \frac{S_y}{\sigma_{\text{Al}}}, \]

\[ \eta_{\text{shear}} = \frac{55000 \text{ psi}}{27705 \text{ psi}} = 1.98 \]

As in the coupler, fatigue also has to be considered for the shaft. The shaft will be looked at both for 1000 cycles and at infinite life. The equation will be governed once again by the same equation that governed the coupler with a few additions.

\[ S_{m_{10^3}} = 0.9S_u \]

\[ S_{m_{10^3}} = 0.9(90000 \text{ psi}) = 81000 \text{ psi} \]

\[ S_{m_{\text{corr}}} = \frac{-S_{m_{10^3}}}{S_u} K_{fs} \sigma'_{\text{mean}} + S_{m_{10^3}} \]

Where:

\[ K_{fs} = \text{correction factor for the key way} \]

Equation A-27 is basically the same equation as 3-14, except that it has a correction factor for the keyway being place in the shaft. The correction factor looks at many different aspects of the key way, from the groove radius to diameter of the shaft. To do this we first state that the end mill will leave a 0.02in radius in the groove. Once we know the groove left in the shaft, we can start to calculate the $K_{fs}$ factor.
First thing we need to do is find the ratio between the groove radius and the shaft 
diameter to look up a $K_{ts}$ correction value off a graph [10].

$$c = \frac{r_s}{D_s}$$  \hspace{1cm} (A-28)

Where:

$c =$ ratio

$r_s =$ groove radius

$D_s =$ outside diameter of the shaft

$$c = \frac{0.02in}{1.75in} = 0.011$$

For this $c$ value we find the $K_{ts}$ value to be 3. Now we can look up Neubler’s 
constant. Neubler’s constant is dependant on the ultimate strength of the steel, and for 
our steel Neubler’s constant $= 0.07in^{0.5}$. Now that the groove radius and Neubler’s 
constant is known, we can calculate the notch sensitivity by Kunn-Hardrath formula in 
terms of Neubler’s constant.

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r_s}}}$$  \hspace{1cm} (A-29)

Where:

$q =$ notch sensitivity

$\sqrt{a} =$ Neubler’s constant

$$q = \frac{1}{1 + \frac{0.08}{\sqrt{0.02}}} = 0.64$$
Now we can solve for $K_{fs}$ by using the following stress concentration equation.

$$K_{fs} = 1 + q(K_{ts} - 1)$$  \hfill (A-30)

Where:

- $K_{fs}$ = final notch correction factor
- $q$ = notch sensitivity
- $K_{ts}$ = first notch correction factor

$$K_{fs} = 1 + 0.64(3 - 1) = 2.28$$

Now we can use equation A-27 to solve for the corrected 1000 cycle value, and using a modified equation A-15 the factor of safety for 1000 cycles of life can be found.

$$Sm_{corr} = \frac{-Sm_{10^3}}{Su} K_{fs} \sigma'_{mean} + Sm_{10^3}$$ \hfill (A-27)

$$Sm_{corr} = \frac{-81000 \text{ psi}}{90000 \text{ psi}} \left(2.28(13852 \text{ psi}) + 81000 \text{ psi}\right) = 52605 \text{ psi}$$

$$\eta_{10^3} = \frac{Sm_{corr}}{K_{fs} \sigma'_{Alt}}$$ \hfill (A-28)

$$\eta_{10^3} = \frac{52605}{2.28(13852)} = 1.7$$

Using the same criteria for the shaft as we did for the coupler, we can solve for the factor of safety for infinite life. The C values will be the same as for the coupler except that the $C_{size}$ will be 0.823 instead of 0.71.

$$Se_{\infty} = 0.5SuC_{load}C_{size}C_{surface}C_{temp}C_{rehab}$$ \hfill (A-16)

$$Se_{\infty} = 0.5(90000 \text{ psi})(1)(0.835)(0.9)(1)(0.659)$$
This means that the shaft does not have infinite fatigue life with respect to the maximum load capabilities of the dynamometer load system. In Appendix C the drawing of the coupler, key, and shaft can be seen.
Automotive Lab Test Equipment Selection Report

Dr. Greg Kremer
Ohio University Mechanical Engineering Department
740-593-1561, kremer@ohiou.edu

39.250. Background

As a result of several student design projects we have constructed a “frame and roller system” as the first step in constructing a chassis dynamometer system (see Figure 1). We have a computer system available adjacent to the dynamometer with some student-designed data acquisition software. An RFQ (dated 1/23/2001) was distributed soliciting bids for a load system (and any associated control systems and support systems) that could be attached to the existing “frame and roller system” to create a working chassis dynamometer system. Specifications for the load systems were established to allow testing of motor and transmission systems in a vehicle test bed run on the chassis dynamometer. Current plans are to test electric and hybrid-electric vehicle systems at motor powers of 100 HP and higher (300 HP or more is desirable).

This report reviews the major activities related to the evaluation and selection of the load system, control system, and associated hardware for completing the chassis dynamometer system.

Figure 1: Frame and Roller System
2.0 Analysis of Alternatives

As previously mentioned, this dynamometer design and construction project has been the focus of numerous student projects. The first group of students (C. Brian Hintz and Marc B. Morgan) designed and built the frame and roller system and hypothesized that an electric generator would be linked to the rollers and the power would be dissipated using large wet cell resistors. However, no evaluation of this power dissipation design concept was documented. A second group of students (Nathan Haag, Robert Wilson III and Matt Hanagan) picked up the project and focused on the power absorption system. They designed and built a “power transfer carriage”, essentially a counter shaft belt-connected to the main roller shaft and positioned to allow connection to a load system. A secondary purpose of the “power transfer carriage” was to provide an option for directly connecting a motor or a motor/transmission system to the load system, thereby creating a dynamometer system for vehicle and component testing. This group of students evaluated numerous alternatives for load systems, including AC/DC Electric Motor/Generators, Eddy Current Brakes, hydraulic motors, and water brakes. At the time of their analysis the budget was very small so they dismissed several alternatives such as the eddy current brake and the motor/generators because of price alone and they recommended a water brake dyno even though it could not meet our desired load specifications. Water brakes are generally the least expensive load systems and have a good power-to-size ratio and a good power-to-cost ratio, but they do not provide the torque at low speed necessary for testing electric and hybrid-electric vehicle systems.

Since the time of the students' analysis of load system alternatives we have been awarded grant money from the Ohio Board of Regents Hayes Investment Fund for the development of a dynamometer system capable of testing transmission systems for future vehicles. This grant opened up the possibility of eddy current brakes or motor/generator type load systems that better match the load specifications for electric and hybrid-electric vehicle systems. After further discussions with dynamometer system vendors we solicited quotes for various types of systems and again analyzed them relative to our load/power specifications and the following additional criteria as listed in the formal request for quote document.

Alternative dynamometer systems will be evaluated based on

- Power dissipation requirements (with respect to desired load envelope)
- Cost (systems with a total cost less than $50000 are preferred)
- Size/Space requirements (lab space is severely limited)
- Operating requirements (water supply system, electrical connections, etc.)
- Other system performance characteristics (ease of use, accuracy, speed of response, drift (ability to hold constant torque), etc.)
- Quality of data acquisition
39.250. **2.1 Load System Evaluation**

We reviewed quotes and specifications from three water brake dynamometer vendors, Hydra-Brake, Stuska, and Land and Sea, Inc. Although the water brakes do have acceptable maximum power levels, their limitations in terms of low-speed torque, water supply, speed of response, and controllability eliminated them from serious consideration. We also received a quote on an AC motor/generator dynamometer system from Dyna Systems that met all of our specifications but was well beyond our budget at a cost of $158000. Similarly, we received a quote on a reconditioned DC motor/generator dynamometer system from Dyne Systems that met all of our specifications but was also beyond our budget at a cost of $70300.

The most appropriate load system for our application seems to be an eddy current dynamometer. Two full-service dynamometer companies that offer a range of dynamometer systems both proposed eddy current systems in response to our request for quote. Eddy current brakes have excellent torque at low speeds, excellent controllability and stability, are well suited for computer controlled simulation tests, and have excellent repeatability and accuracy. The major decisions with respect to eddy current dynamometers involve selecting the maximum power capability, the cooling method (dry gap versus wet gap for water cooled systems), and the manufacturer/vendor. We received quotes for a range of eddy current systems from both Dyne Systems and Digalog. A comparison plot of load capability versus load requirements for the three leading eddy current brake alternatives is shown in Figure 2. The symbols represent desirable test points, and the lines represent the load envelopes for the load systems. A system can handle the load requirements for all test points underneath its line, and a system that can handle more test points is a better match for the requirements than one that can handle fewer test points. As the maximum power capacity increases, the load system can handle more of the desirable test conditions. Therefore, the Digalog AE250 eddy current brake, rated for 250 Kw (333 HP), is the best of the three systems shown in terms of satisfying the load requirements. The Digalog AE150 (150 Kw, 200 HP) is acceptable for speeds above 2000 RPM but not at lower speeds. The Digalog AE400 (536 HP) would handle even more test conditions but is eliminated from consideration because of its cost ($44000 base cost plus shipping and taxes) and because of size and space considerations. The Midwest Dynamometer Company Model 1014WHS is a 250 HP system which uses wet gap water cooling to get significantly more power capability than the model 1014AHS dry gap system (175 HP). It is acceptable above 1700 RPM but not at lower speeds.

In our transmission research application it is important to have load capability across the entire speed range to support tests such as:
1. designing electronic shifting systems for manual transmissions
2. testing continuously variable transmissions in high torque applications
3. developing hybrid vehicle control strategies for optimum efficiency
4. developing shift schedules for electric race vehicles
5. estimating transmission and driveline losses under various conditions
6. running computer-controlled vehicle simulation tests

Figure 2: Comparison of load system torque requirements with the capabilities of the three leading eddy current brake alternatives.

The other considerations in selecting a specific eddy current system are size/space requirements and required support systems (cooling water, etc.). The size ranges of the three leading candidate systems are similar and they all fit into the available space. However, the Midwest Dynamometer Company Model 1014WHS has two disadvantages in that the input shaft is very high relative to the base and a coolant catch basin is required under the unit for wet gap operation. The height of the input shaft practically eliminates the possibility of a direct connection between the frame/roller output shaft and the dynamometer, necessitating an additional belt system and the associated belt losses. Also, it will be very difficult to find space for the catch basin. The AE250 is larger than the AE150 but it still fits in the available space and the input shaft matches reasonably well with the frame/roller output shaft (see Figure 3) and should be able to be directly
connected via a flexible coupling. All three eddy current systems have similar support system requirements, with the amount of cooling water flow directly proportional to the power dissipation level (approximately 0.1 GPM for each HP absorbed). The specifications for the Digalog AE250 are reduced from this level (45GPM minimum at a pressure of 1.6 atmospheres) and the cooling system for the Digalog eddy current brakes has a technically advanced design and provides thermocouple-based feedback of loss plate temperature for cooling water flow control.

![Image](image.png)

**Figure 3:** Top view and front view of AE250 Dyno along with roller system and “power transfer carriage” showing that it fits within allowable space limits

One additional consideration is the current government emphasis on research with large vehicle platforms such as large SUVs, transit busses, and long haul trucks. To test hybrid power trains and transmission concepts for these vehicles requires high power capability and high torque capability, so a system with these capabilities will make the research facility more competitive in seeking external funding.

The base costs of the three leading candidate systems are $22065 for a reconditioned 1014WHS with 90 day warranty, $23100 for the Digalog AE150 with 12 month warranty, and $31550 for the Digalog AE250 with a 12 month warranty. Overall, the Digalog AE250 appears to be a good compromise between cost and power capability based on our present and anticipated future testing needs. Additionally, the Digalog AE250 has advantages in terms of size, configuration, durability, cooling system, and warranty. Therefore, the Digalog AE250 is selected for purchase.
2.2 Control System

The load control system and data acquisition system are a critical part of the test system but are sold independently of the system. Manual control is the standard with many water brakes but to take full advantage of the AE250 eddy current brake a digital control system is necessary. Using a manual control or a simple closed-loop control system would eliminate the ability to do automated tests, sweep tests, and simulations, and would require a separate data acquisition system for accuracy and a high sample rate. The Digalog TESTMATE digital controller is an excellent system but has more functionality than we currently need (and more than we will likely need in the future) and its total cost including installation is over $23700. Dyne Systems Dyn-Loc IV digital dynamometer controller has all of the functionality we need, is a well proven system with high accuracy control and data acquisition capability, includes Tune/Term software for PC control and data logging, and has a reasonable base cost of $7429. The Dyn-Loc IV control system is compact and will fit easily with the test system, and the Dyne Systems company has over 25 years of experience with dynamometer control. Therefore, the Dyne Systems Dyn-Loc IV is selected for purchase.

Additional Torque Sensors  (efficiency tests require multiple torque measurements)

In-line torquemeter is an interesting option but is very expensive ($10-15K for some models). Eliminates “inertia error” that could be present and would allow much easier transmission and component efficiency tests.

Strain gage telemetry systems are an interesting alternative.
Dynamometer System – Request for Quote
1/23/2001

Dr. Greg Kremer
Ohio University Mechanical Engineering Department
251 Stocker Center
Athens, OH. 45701
740-593-1561, kremer@ohiou.edu

Ohio University had received a $50000 grant from the Ohio Board of Regents to assist in the purchase of a dynamometer system. A semi-formal bid process must be followed prior to release of these funds for purchase of the dynamometer system components. This request for quote solicits written bids from any qualified vendor interested in supplying a load system and a load control system that satisfy the specifications detailed below. Individual vendors are encouraged to submit multiple bids if they have multiple systems that meet the specifications. The bids may be sent via email, as an attachment to an email, or mailed to the address given above. The review of bids will begin 2/1/2001 and will continue until an acceptable system has been identified and purchased. Please direct all questions regarding this RFQ to Dr. Kremer at the phone number or email address given above.

Thank you, and I look forward to receiving your bids.

Background

As a result of several student design projects we have constructed a “frame and roller system” as the first step in constructing a chassis dynamometer system. The 1.75 inch diameter output shaft where the load system will be connected is approximately 13 inches above ground level. We have a computer system available adjacent to the dynamometer with some student-designed data acquisition software. A standard water supply (approximately 14 GPM) and standard electrical connections (110V and 220V) are available in the lab area.

The primary vehicle types that will be tested on the system are electric and hybrid-electric vehicles. The electric motors that we are currently running in our electric race vehicle and our electric test bed vehicle have a maximum power of 120 HP, but it is likely that in the future higher power systems will be tested.

We are currently searching for a load system (and any associated control systems and support systems) that can be attached to the existing “frame and roller system” to create a working chassis dynamometer system.
Specifications and key considerations in the decision process

Power dissipation requirements: Two load envelopes are shown in both Figure 1 (Load System Power Requirements) and Figure 2 (Load System Torque Requirements) below. The solid line represents the minimum acceptable load envelope (based on the current 120 HP motors) and the dashed line represents the desirable load envelope (based on a 200 HP motor). The curves shown assume direct connection to the output shaft of the roller system. Load systems that include a speed-multiplying gearbox between the output shaft and the load system to shift the peak power point are possible (assuming the dynamometer system has sufficient high-speed capability); however the additional cost and complexity of this arrangement will be considered as part of the load system in the final system selection.

Alternative dynamometer systems will be evaluated based on
- Degree of satisfaction of the load requirements
- Cost (systems with a total cost less than $50000 are preferred)
- Size/Space requirements (lab space is severely limited)
- Operating requirements (requirements for water supply system, electrical connections, etc.)
- Other system performance characteristics (ease of use, accuracy, speed of response, drift)
- Data acquisition system
Figure 1: Load Systems Power Requirements

Figure 2: Load System Torque Requirements
To: Stephen Riesbeck, Grants & Contracts Manager, ORSP
Linda Shapiro, Grants Accounting

From: Greg Kremer, Mechanical Engineering Department, RCENT

Subject: Request to initiate the purchase of dynamometer system components using grant money from OBOR Hayes Investment Fund

This letter briefly describes the vendor/bid evaluation procedure followed for the purchase of dynamometer system components using funds from the “Propulsion Systems for Future Vehicles” Hayes Investment Award Grant competition for which Ohio University was awarded $50000. The original Ohio University budget request for this project was $100000 for a research quality transmission dynamometer and lab improvements necessary to support the equipment. The proposal was funded at a 50% level, so original plans had to be scaled back relative to the new budget.

Our only realistic option at this price level was to develop and construct a transmission dynamometer system ourselves, using existing vehicle components and computer hardware where possible and using the grant money to purchase several key components (load system, control system, measurement sensors) necessary to complete the dynamometer system. The automotive research program at Ohio University has a functioning spare electric vehicle complete with electric motor and driveline that is suitable for use as a “test bed” for transmission system testing on a chassis dynamometer. This test bed concept saves the large expense of a separate motor/controller to provide the input power for a stand alone transmission dynamometer. A chassis dynamometer was already being constructed for the Electric Bobcat Racing Team – it is currently a “frame and roller” setup with a computer available for data acquisition. Therefore, to develop a research facility capable of transmission testing our main tasks were:

1) To identify and purchase a load system, a control system, and torque sensors necessary for completing the construction of the system
2) To identify the lab improvements necessary for system operation (i.e. to provide the necessary cooling water for the load system)
3) To complete the lab improvements and complete the construction, installation and calibration of the dynamometer system within the budget.

After extensive searches of all load system options and talking with many dynamometer system vendors, we created a targeted request for quote (RFQ) defining our specific requirements and requesting quotes on components and systems that met our requirements. The RFQ is attached as Appendix 1. In response to the RFQ we received competitive bids from two full-service dynamometer companies (Digalog and Dyne Systems) and price quotes for individual components from several other vendors. Copies of the bids are attached as Appendix 2.

The bids were evaluated using the criteria listed in the RFQ and with the overall budget as a controlling factor. The decision procedure is documented in a “Test Equipment Selection Report” attached as Appendix 3, and the purchases recommended in that report are detailed in Table 1. Please initiate the paperwork to purchase the Phase 1 items listed in Table 1 immediately.

Table 1: Vendor Information for Dynamometer System Component Purchases

<table>
<thead>
<tr>
<th>Phase 1 – Critical Components to be purchased immediately</th>
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</thead>
<tbody>
<tr>
<td>Item</td>
</tr>
</tbody>
</table>


### Load System

<table>
<thead>
<tr>
<th>Items 4.1</th>
<th>Items 4.2</th>
<th>Item 4.1, $30800</th>
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</thead>
<tbody>
<tr>
<td>(AE250 Eddy Current Dynamometer)</td>
<td>(calibration weights) in quotation #</td>
<td>$750</td>
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<tr>
<td>and 4.2</td>
<td>042400-2608-MS</td>
<td>Total $31550 plus taxes and costs for</td>
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<td></td>
<td></td>
<td>shipping from Detroit</td>
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</table>

### Control System

<table>
<thead>
<tr>
<th>Item C (Dyn-Loc IV digital controller)</th>
<th>Item D (OS/US-EMS module) in quotation # Q010125</th>
<th>Item C, $7125</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td>Item D, $304</td>
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<tr>
<td></td>
<td></td>
<td>Item E, $350</td>
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<td></td>
<td></td>
<td>Total $7779 plus shipping and taxes</td>
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**Digalog**
P.O. Box 3315
Ventura, CA 93006-3315
805-644-5000

**Dyne Systems**
N 114 W 19049 Clinton Drive
Germantown, WI 53022
262-250-2700

Additional components and services for the dynamometer system will need to be purchased and/or contracted during final construction and calibration testing. Final quotes for these components and services are not yet available because the actual needs will not be known until the load system has been placed. Initial quotes and estimates for these costs are included for informational purposes in the table below.

<table>
<thead>
<tr>
<th>Phase 2 - Additional Items to be purchased/contracted as required</th>
<th>Item</th>
<th>Dollar Amount</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling Water System: Extend ⅜ inch water line to the EC dynamometer and make the connection to the dynamometer inlet. Install water line from dynamometer outlet to drain.</td>
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<td>$500</td>
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<tr>
<td>Additional Hardware: Flexible couplings, jack shafts, mounting bases, etc. for final assembly of system</td>
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<td>$1000 - $5000</td>
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<tr>
<td>Dyne Systems technician services: start-up assistance, if required</td>
<td></td>
<td>($2700 for 2 days)</td>
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<tr>
<td>Additional torque Sensors for increasing capability of system</td>
<td></td>
<td>$2395 each</td>
</tr>
<tr>
<td>(TorqueTrak 9000 Digital Radio Telemetry System)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Please contact Dr. Greg Kremer (740-593-1561, kremer@ohiou.edu) if you have any questions concerning this request.

Thank you for your assistance.

Gregory G. Kremer
DETAIL A

- be machined with a tolerance of ±0.0001 and a finish of 125 ± 0.0001

- Do not invert the interfering section and important

All dimensions are in inches, and ±0.001 unless otherwise noted.

<table>
<thead>
<tr>
<th>Rev.</th>
<th>Part Number</th>
<th>Date Rev.</th>
<th>Date</th>
<th>Rev.</th>
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<tr>
<td>J</td>
<td>123456</td>
<td>1/23/2023</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Drawn by: [Name]

Checked by: [Name]

Approved by: [Name]
D.1 Calibration Procedure

1) Turn electrical bus above computer to on.

2) Read the Torque value displayed on LED. If value is not between +/-2 then a calibration is required.

3) Fill the water reservoir with water, and while it is filling up turn on the Dyne-Loc IV to warm up for 30 minutes. While the water reservoir is filling go to step 4.

4) To calibrate the load system the frame and roller system must be moved to allow room to install the calibration arms. To move the frame and rollers disconnect the 4 anchor angles holding the frame to the floor by removing the top bolt and loosening the masonry bolt. Disconnect the power transfer case from the roller carriage by removing the 2 attachment bolts (2 additional bolts must be removed due to interference) and remove the L bracket holding the tensioner. Remove the sprocket from the power transfer case shaft and slide off the belt so that it hangs loose around the shaft.

5) Push/Pull the Roller carriage out of the way (as far as belt will allow) to make room for the attaching of the calibration arms to the dynamometer.

6) Plug pump into GFCI outlet on inside wall and let run for 60 seconds.

7) Press the auto zero button on the Dyne-Loc IV with a pencil tip (see Figure D-1) and all values should go to 00000.

8) Connect the calibration arms to the dynamometer by removing the large nuts and then sliding the small locating hole over the locating pin and the large hole over the bolt and tighten the nut with a pip wrench.
9) Place the slider weights onto each arm and adjust them until the Dyne-Loc IV torque displays reads zero.

10) Starting on the left side add one plate at a time to the weight pan. As you add each plate press the torque button and adjust active lever switches to correct torque reading of 200Nm for each plate and hit the Auto Span button to get the Led to update to the same value as the lever switch. After all six plates are loaded begin removing the plates one by one and verify that the torque reading being displayed is the correct reading. Repeat this step as many times as necessary to achieve calibration.

11) Repeat step 10 for the other side.

12) Now the system is calibrated and can be reconnected to test. To reconnect the system the roller carriage frame will have to be push/pulled back into position. Once the roller carriage frame is placed into position reconnect the four anchor angles and bolt, but do not tighten the bolts. Reconnect the two attachment bolts between the power transfer case and the roller carriage system, but do not tighten the bolts. Once all the bolts are in place tighten them. Now reconnect the belt around the power transfer case. The reconnect the belt tensioner and apply adequate tensioning.

Note: the system maintains calibration al long as the Dyne-Loc IV is turned on. Once the Dyne-Loc IV is turned off it will forget calibration settings so make sure you leave system on during test period.
Figure D-1 A front view of the Dyne-Loc IV data acquisition controller.
D.2 Operation Procedure

1) Ensure that the system is calibrated (see D1).

2) Fill the water reservoir with water, and while it is filling up turn on the Dyne-Loc IV to warm up for 30 minutes.

3) Load car without batteries by placing the car onto the chassis dynamometer and securing it to the roller chassis by using the rachet straps attached to the eye bolts. Wrap straps multiple times around top chassis members and crank as tight as possible. Run car on rollers slowly to let it settle and retighten straps.

4) Press the mode of operation that you want to run the car in, rpm or torque. If you press rpm you will run the car to a selected speed and see how much torque is being produced, while the torque mode allows you to select a torque to see how fast you can go.

5) Once you select the mode of operation, select the speed or torque you want to test using the active lever switches.

6) Start the car to get the rollers moving slowly and make sure the Dyne on button is lit and hit the OS/US reset button on the Dyne-Loc IV and then go full throttle.

7) Watch the output values and once steady state is reached, record the numbers for this test setting and let off the accelerator.

8) Repeat steps 3-6 until testing is completed.

9) Unsecure the car and take it off the chassis dynamometer.

10) Leave power on to maintain calibration if additional tests will be run in near future.

11) Drain water system (using pump and diverted or siphon).
Note: Unit are in ftlbs, Hp, and rpm, but can be changed internally using instructions in the Dyne-Loc IV manual.
<table>
<thead>
<tr>
<th>Item</th>
<th>Vendor</th>
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<td>Roller Carriage System</td>
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<td><strong>Dyne-Loc IV</strong></td>
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<td>Dyne-Loc IV Controller Dyne Systems</td>
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<td>OS/US Relay Dyne Systems</td>
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<td>Systems Dyne</td>
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<td><strong>G20 Couplers</strong></td>
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<td>Applied Industrial Technology</td>
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<td><strong>Water Cooling System</strong></td>
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<td>150gal Hi-Density Polyethylene Though</td>
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### Tensioner

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<td>SO6 Idler Shaft</td>
<td>Motion Industries</td>
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<td>P4F Flange Pulley</td>
<td>Motion Industries</td>
<td>$110.17</td>
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<td>Slater Builders Supply</td>
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**Misc.**

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<td>5/8&quot; X 4-1/4&quot; Anchor Bolts</td>
<td>C&amp;E Hardware</td>
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**Shipping Costs**

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<th>Price</th>
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<td>Dyne-Loc IV</td>
<td>Dyne Systems</td>
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<td>Tensioner</td>
<td>Motion Industries</td>
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**Total** $46,114.01