CHARACTERIZATION AND NONDIMENSIONAL ANALYSIS OF A
VARIABLE SPEED CENTRIFUGAL PUMP

A Thesis

presented to

the Faculty of the Graduate School

at the University of Missouri-Columbia

In Partial Fulfillment

of the Requirements for the Degree

Master of Science

Presented By

Michael Scheller

DECEMBER 2014

Graduate Advisor

Dr. A. Sherif El-Gizawy
The undersigned, appointed by the dean of the Graduate School, have examined the thesis entitled

CHARACTERIZATION AND NONDIMENSIONAL ANALYSIS OF A VARIABLE SPEED CENTRIFUGAL PUMP

presented by Michael Scheller,

a candidate for the degree of master of science

and hereby certify that, in their opinion, it is worthy of acceptance.

________________________________________
Professor Sherif El-Gizawy

________________________________________
Professor Hani Salim

________________________________________
Professor Yuyi Lin

________________________________________
Professor Billie Cunningham

________________________________________
Professor Roger Fales
ACKNOWLEDGEMENTS

Firstly, I would like to thank my entire family for the support they’ve given me throughout my entire graduate school process. I wasn’t sure they would be entirely supportive of my decision to attend graduate school instead of entering the work force. However, they stood behind my decision to enter graduate school, and supported me with anything I needed while completing my degree. I cannot thank them enough.

Next, I would like to thank all of the graduate and undergraduate students that have helped me on this project.

- Shane Corl, a Doctoral Candidate for Dr. El-Gizawy, coordinated practically everything that was done on this project. His countless hours programming and doing physical work on the system made this project doable. Without him, this project would have gone nowhere, and I am so thankful to have had his help.

- Bilal Hussain, an undergraduate student for Dr. El-Gizawy, helped me a great deal during the experimental process for this project. We spent full days in our lab running experiments and performing manual labor on the system, and without him, this project would have been much more difficult. Thank you so much.

-
- Brian Graybill, Jake Harris, Andy Gunn, Tony Adamson, Annemarie Hoyer, Zhentao Xie, Jay Shelby and Amer Krvavac. All of you have contributed something to this project, and I really appreciate you helping in one manner or another. I am so happy to call you all my friends. Thank you!

- My committee members, Dr. Yuyi Lin and Dr. Billie Cunningham, thank you for volunteering your time in order to aid me in my pursuit of higher learning. I am grateful for your thoughtful critique of my work, and I really do appreciate it.

Finally, I would like to thank Dr. El-Gizawy. I have worked with Dr. El-Gizawy for five years spanning across my undergraduate and graduate programs, and I couldn’t have asked for a better advisor. His resourcefulness when it comes to finding new projects is unparalleled. In addition, the experience that I’ve gained through working with industry officials and government employees is extremely valuable to me. I know that I would not have had these opportunities without Dr. El-Gizawy, and I am very thankful that he sponsored my learning for the past five years. Thank you so much.
TABLE OF CONTENTS

ACKNOWLEDGEMENTS..............................................................................................................................................ii

LIST OF ILLUSTRATIONS...........................................................................................................................................v

ABSTRACT.....................................................................................................................................................................vii

CHAPTER ONE – INTRODUCTION TO CENTRIFUGAL PUMPS AND LITERATURE REVIEW..............................1

CHAPTER TWO – EXPERIMENTAL TEST RIG DESIGN..........................................................................................12

CHAPTER THREE – STANDARD PUMP CHARACTERIZATION, RESULTS, AND ANALYSIS..........................27

CHAPTER FOUR – NONDIMENSIONAL PUMP CHARACTERIZATION, RESULTS, AND ANALYSIS...............52

CHAPTER FIVE – OVERALL PROJECT ANALYSIS AND CONCLUSIONS...............................................................66

REFERENCES..............................................................................................................................................................70

APPENDIX A – SYSTEM PICTURES..........................................................................................................................71

APPENDIX B – MATLAB CODE..............................................................................................................................77
# LIST OF ILLUSTRATIONS

<table>
<thead>
<tr>
<th>FIGURE</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Working Drawing of a Centrifugal Pump</td>
<td>2</td>
</tr>
<tr>
<td>2. Image of a modern centrifugal pump impeller</td>
<td>3</td>
</tr>
<tr>
<td>3. Comparison of a modern impeller to a vintage impeller</td>
<td>4</td>
</tr>
<tr>
<td>4. Diagram showing impeller and volute</td>
<td>5</td>
</tr>
<tr>
<td>5. Standard Single Speed Pump Characterization</td>
<td>7</td>
</tr>
<tr>
<td>6. Example Standard Variable Speed Pump Characterization</td>
<td>9</td>
</tr>
<tr>
<td>7. Initial Experimental Test Rig Design Proposal</td>
<td>13</td>
</tr>
<tr>
<td>8. Front View of Pump Canister</td>
<td>14</td>
</tr>
<tr>
<td>9. Top View of Pump Canister</td>
<td>15</td>
</tr>
<tr>
<td>10. Completed Test Rig</td>
<td>16</td>
</tr>
<tr>
<td>11. System Static Pressure Sensors</td>
<td>18</td>
</tr>
<tr>
<td>12. In-Line Flow Sensor</td>
<td>19</td>
</tr>
<tr>
<td>13. RTD (Resistance Thermometer Detector)</td>
<td>20</td>
</tr>
<tr>
<td>14. Accelerometers and Centrifugal Pump</td>
<td>21</td>
</tr>
<tr>
<td>15. Labview Interface</td>
<td>24</td>
</tr>
<tr>
<td>17. Initial Experimental Matrix</td>
<td>31</td>
</tr>
<tr>
<td>18. General Performance Plot Generated From First Experiments</td>
<td>33</td>
</tr>
<tr>
<td>19. Capacity-Efficiency Plot From Initial Experimental Matrix</td>
<td>34</td>
</tr>
<tr>
<td>20. Three Dimensional Plot of Capacity, Head, and Efficiency</td>
<td>38</td>
</tr>
<tr>
<td>21. Plot of Rotational Pump Speed vs. Pump Power Consumption</td>
<td>40</td>
</tr>
<tr>
<td>22. Plot of Capacity Versus Bulk Fluid Temperature</td>
<td>41</td>
</tr>
<tr>
<td>23. Plot of Capacity Versus Head, Using Valve Angle</td>
<td>44</td>
</tr>
<tr>
<td>24. Plot of Voltage and Efficiency for Various Valve Angles</td>
<td>45</td>
</tr>
<tr>
<td>25. Plot of Capacity and Efficiency for Various Valve Angles</td>
<td>47</td>
</tr>
<tr>
<td>26. Three Dimensional Surface Plot of Standard Pump Characterization</td>
<td>48</td>
</tr>
<tr>
<td>27. Plot of Pump Rotational Speed (RPM) Versus Pump Power Consumption</td>
<td>50</td>
</tr>
<tr>
<td>28. Chart of Specific Speed BEP Values</td>
<td>55</td>
</tr>
<tr>
<td>29. Plot of Specific Speed versus Efficiency (Constant Angle Lines)</td>
<td>57</td>
</tr>
<tr>
<td>30. Plot of Specific Speed versus Efficiency (Constant RPM Lines)</td>
<td>59</td>
</tr>
<tr>
<td>31. Chart Showing The Impellers Performance Versus Other Impeller Types</td>
<td>61</td>
</tr>
</tbody>
</table>
32. Image Showing The Centrifugal Pump Impeller and Vane Shape..........................62
33. Three Dimensional Surface Plot of Non-Dimensional Pump Characterization.........63
ABSTRACT

When dealing with centrifugal pumps in the aerospace field, current industry standards dictate the use of single speed drive pumps. The performance of these pumps is then modulated using simply an electro-mechanical valve, which is controlled by an aircraft’s onboard CPU. The French pump manufacturer, Intertechnique, has provided the University of Missouri with a variable speed pump for the purposes of this project.

The primary focus of this project is to create control logic which can keep the pump operating at its best efficiency point. This control software will be developed control the speed of the pump, as well as a control valve attached to the pump. By varying the speed of the pump, as well as the amount of flow being allowed by the valve, the pump should be operating most efficiently just through the use of the software being developed by this project. In order to create this software, however, the pump must be completely understood. That is, the pump must be fully characterized. The completion of this characterization will allow us to achieve and capacity and head within our allowable performance range. By doing this, we will be able to minimize power consumption of the pump, while simultaneously maximizing the life of the pump.

The purpose of this project is to show how this variable speed pump performs over a wide range of testing parameters. Plots showing capacity (volumetric flow rate), pump head (pressure), temperature, vibration and efficiency will be discussed. In addition, experiments
have been analyzed non-dimensionally, so that these results may be applied to any pump with
the same impeller shape. The experimental results were then analyzed against existing data in
the field of centrifugal pump performance. The significance of the experimental results to the
field of mechanical engineering is then discussed, showing the relative efficiencies of the testing
done on this pump, and where those efficiencies fall in comparison to other centrifugal pumps
Chapter One – Introduction and Literature Review

Before going into detailed analysis of characterizing a centrifugal pump, it becomes important to fully understand how a centrifugal pump works. In the field of mechanical engineering, a centrifugal pump is typically classified as a piece of turbomachinery, which is a term used to describe a machine that transfers energy between a motor and a fluid. The range of fluids and motor speeds for a centrifugal pump can vary significantly. For instance, the most common fluid used as a motive fluid in centrifugal pumps is water. Pumping water has many uses, such as in sewage systems, coal fired and nuclear power plants, and car radiators. In all of these cases, a centrifugal pump drives the motive flow of the fluid. Centrifugal pumps can also be used to pump other common fluids, such as petroleum or petrochemicals. In all of these cases, the flow demand by an engineer can be created by simply attaching a centrifugal pump to a line of fluid, and giving the centrifugal pump a source of power, whether that be electrical or gas generated.
Figure one shows a cross section of a centrifugal pump, and all of the important components of such a pump. As with any pump, a centrifugal pump has an inlet, outlet, impeller, and a shaft attached to the impeller. All four of these components are crucial to the operation of a pump; removing any of these four basic components would result in the failure of the pump.
Figure Two shows a pump impeller, which will be the focal point of this thesis. An impeller is a rotating disk which is the primary inducer of flow within the pump. It typically has many curved pieces of metal attached to it, which are typically referred to as “vanes”. The shape of a vane has a very significant impact on the type and velocity of the flow induced by the pump. An impeller typically gets its rotational energy from an electrical or gas powered source, which will then cause spinning on the shaft that runs through the center of the impeller. The method of flow generation for a centrifugal pump is through the conversion of the rotational energy of the impeller, into the kinetic energy of the flow of the fluid. This transfer of energy is normally described in engineering as ‘pushing’ or ‘lifting’ of the fluid, as the spinning vanes of the impeller are literally inducing the flow themselves.
The concept of a centrifugal pump has been around since the late 15th century, however it was not fully put into practice until an Italian engineer named Francesco di Giorgio Martini developed the first prototype of a centrifugal pump. However, for the two hundred years of the existence of a centrifugal pump, the impellers utilized straight vanes. These straight vanes caused extremely turbulent flow, as well as incredible amounts of cavitation within the pump.

Figure Three – Comparison of a modern impeller to a vintage impeller

The late 19th century saw a very significant change in the shape of vanes in a centrifugal pump. Figure Three indicates this change very well, as it shows the innards of a modern
centrifugal pump, versus an impeller from the late 17\textsuperscript{th} century. The impeller on the right has straight vanes, which caused many issues with the lifespan of the pump, as well as the performance of the pump. However, the late 19\textsuperscript{th} century, as well as the British engineer John Appold, brought about the curved vane structure, which allowed for a vast increase in both of the aforementioned categories.

The spinning of the impeller, however, does not decide the direction of the flow. Referring back to Figure One, an indicator pointing to a volute is on the diagram. A volute is simply a swept portion of solid material that physically directs the fluid in the desired direction. In most cases, the volute directs the fluid in the direction of the outlet of the pump. For the purposes of this project, our volutes follow a similar pattern, which can be seen in Figure Four.

![Figure Four – Diagram showing impeller and volute](image)

The final necessary part to a working centrifugal pump is the motor, which drives the rotational motion of the impeller. When discussing motors that can be used to drive a centrifugal pump, there are many different classes that the motor can fall into. The vast
majority of pump motors will be powered with either a combustible, such as gasoline, or powered using electricity. Another way in which the motor has to be identified is whether the speed of the motor can be varied in some fashion. When discussing motor variation, most of the pumps are invariable in speed – that is, they are single speed pumps.

Since single speed pumps are the current industry standard for mass moving of fluids, it is important to discuss single speed pump properties, as well as why they are normally chosen for pumping demands. This is due to the fact that single speed pumps have been characterized in depth for every impeller size, shape, as well as motor speed. The characterizing of these pumps has been an ongoing engineering project since the early 20th century. Therefore, there is a massive amount of data which currently exists on nearly any type of pump conceived possible. In addition, the control of a single speed centrifugal pump is significantly easier to deal with than a variable speed pump.

Creating a control system for a single speed pump is very well understood, and can be done very easily, given that the engineer knows the general performance characteristics of the pump they are working with. In addition, the flow demand for a single speed pump can be lowered to the correct amount by attaching a valve to the pump, and turning the valve to throttle the flow to the correct amount. If the engineer wishes to create a greater amount of flow, a single speed pump is also much more simple to work with than a variable speed pump. A normal centrifugal pump will come with multiple impellers, all with different vane shapes, as well as overall impeller diameters. These different impellers can be easily installed on any existing single speed pump, given that the impeller is compatible with the existing motor. A
final factor which causes engineers to typically choose single speed centrifugal pumps over variable speed centrifugal pumps is that, in general, they are significantly cheaper. These reasons are all contributing factors as to why the current industry standard for pumping is to use a single speed pump.

![Single Line Pump Curve](image)

**Figure Five – Standard Single Speed Pump Characterization**

As mentioned above, the characterization for a single speed pump is generally much more simple than when dealing with a multiple speed drive pump. This is because a pump characterization curve, as seen in Figure Five, can be created with a very small amount of experiments. When purchasing a commercial centrifugal pump, part of the documentation that will come with the pump is a set of pump characterization curves. You will always receive a number of characterization sets equal to the number of impellers that came with your pump.
For example, if you purchase a new pump with three impellers, it will come with a set of characterization curves for each different impeller. This is due to the fact that different impeller diameters, as well as different impeller shapes, will have a major impact on the performance of a given pump. However, the general trends seen on these pump characterization curves will always been the same. Therefore, when designing the experiments in order to characterize the pump used in this research project, it became important to keep in mind exactly what experiments would allow us to generate similar curves.

Pump performance curves are always structured in the same way, in order to maintain consistency within the industry. A pump performance curve will always have capacity (flow rate) on the x-axis, and any of a number of other variables on the y-axis. Typical variables that can be plotted on the y-axis include head (pressure rise across the pump), pump efficiency, power consumption of the pump, heat generation of the pump, as well as the net positive suction head of the pump. As stated earlier, the shape of these variables should be generally the same, regardless of the impeller shape or size being used. Knowing this fact is a good way to check the accuracy of any pump testing, as a deviation from the general shape expected indicates an error somewhere in the experimental process.

Due to the fact that the vast majority of pumps are single speed driven, there is only a fair amount of literature currently available on variable speed centrifugal pumps. A variable speed pump is characterized in a similar fashion to a single speed pump, however, instead of different impeller shapes and diameters being used, pump speed is varied instead.
Figure Six shows an excellent example of a factory characterization for an existing variable speed pump. There are a couple interesting things to note here. The first thing that an engineer should notice is that the general setup for a variable speed pump performance graph is the same as in a single speed graph. That is, capacity (flow rate) is on the x-axis, and head (pressure rise) is on the y-axis. In this aspect, the characterization is more or less the same. The typical variables that are plotted are the same as a single speed pump as well – Figure Six shows head-flow curves for a variable speed pump. However, instead of one line per variable per plot,
there are multiple lines per variable per plot, with each line representing a different speed at which the pump can operate. In general, ‘speed 1’ is the lowest speed at which the pump can operate, while ‘speed n’ is the highest speed at which the pump can operate, where n is the number of different speeds that the pump is being tested at. For the purposes of the characterization being done on this project, the plots with lines representing different rotational velocities can also be interpreted as three dimensional surface plots, with capacity on the x-axis, head on the y-axis, and a third variable (efficiency, temperature, vibration parameters) on the z-axis.

Current literature suggests that the three dimensional plotting methods discussed above are unique to this project, as it is difficult to find anything similar to these types of plots in literature. There are some contour plots representing these three dimensional plots in two dimensional form, but using a three dimensional plot to characterize a pump would allow an engineer to have a greater sense of accuracy when attempting to identify which type of flow rate they would need for a desired efficiency output.

While there is a significant amount of literature on characterizing a variable speed centrifugal pump, there is currently very little of the literature that has been non-dimensionalized. Non-dimensionalizing data is a process which turns data that has some measurement associated with it (length, time, velocity, etc.) into a measurement which has no measurement associated with it, yet it still has a numerical value which represents something about the measurement. Turning a measurement into a non-dimensional number is useful for many reasons. The most obvious reason that an engineer would want to non-dimensionalize
data, is that it allows the engineer to apply data universally across similar types of projects. The most famous example of a non-dimensionalized unit is the Reynolds Number, which is used to describe fluid flow. The research in this project is much more useful when a non-dimensional speed is applied to the rotational velocity of the pump. Instead of having revolutions per minute as pump speed, it is much more useful to have the non-dimensional number “specific speed” represent the pump speed. In characterizing data this way, this allows anyone with a similar shaped impeller and vane to apply this data directly to that specific pump, regardless of the actual size of the pump. Representing data in a non-dimensional fashion, as well as the techniques that were used will be discussed at length in a later chapter.

While there is a very significant amount of work that has already been completed in characterizing single speed pumps, there is significantly less data available in characterizing variable speed pumps, simply due to the fact that they are not currently chosen in most aerospace and mechanical engineering projects. Therefore, the characterization done in this project will be used to show exactly how using a variable speed pump can use less energy, as well as providing a much more accurate flow demand than a single speed pump. In addition, the fact that the work on this project has been non-dimensionalized means that something meaningful will be added to the field of engineering, as any engineer with a similar shaped pump impeller will be able to apply the results from this project directly to another project.
Before the design of our test rig could be completed, it became important to address a few pressure questions. The French company Intertechnique provided very little information to go with the centrifugal pump used on this project. The information that was provided regarded the general size of the pump, in addition to some information regarding standard operating conditions and procedures for the pump. This was certainly not enough to design a full test rig. Instead, many important questions needed answered, such as what variables will need to be measured, what types of sensors will need to be purchased, and who will do the technical work in building system.

The important variables which will need to be measured include flow rate, pressure, temperature, and vibration. In addition, various data acquisition devices and programs needed to be identified well in advance of the actual design of the system, in order to insure that all of our hardware would be fully compatible with each other. After identification of the variables that were necessary to measure, selection of the actual hardware was the next task.

After selection of the sensors necessary for data collection, it became important to identify the actual programs and devices that were going to be used to collect the data. For data acquisition purposes for the experimental test rig, as well as data transferring, National Instruments devices were chosen. National Instruments DAQ devices are used currently the engineering standard for data collection and acquisition, so the choice was very simple.
Figure Seven – Initial Experimental Test Rig Design Proposal

Figure Seven shows the initial test rig design, which was submitted to the project leads for approval during May 2012. The first thing that should be examined is the top view of the experimental test system. Examination of the far most left drum shows a rectangle in the bottom of the tank. This rectangle represents the variable speed centrifugal pump, as the pump being used in this study actually operates under submersible conditions. That is, the pump has to be submersed for it to produce any flow whatsoever. This was a very important fact to keep in mind when designing our experimental test rig system.

The first challenge that the engineers tried to address was leakages throughout the system. The most obvious point of leakage will be where the pump is physically mounted to the
bottom of the left most tank. A very significant amount of time was spent trying to get the system to seal at this point.

Figure Eight – Front View of Pump Canister
Figure Nine – Top View of Pump Canister

The pump canister drawings seen in Figures Eight and Nine provide the details necessary for installation of this pump into a system. Figure Nine shows that there are 12 equidistant screws on the flange of the pump canister. Knowing this, 12 equidistant threaded holes were made on the bottom of our left most 55 gallon drum. The accuracy of the distance and diameter of these holes had to be extremely precise, as even a small error would make it impossible for the pump to install correctly into this machined area.

The initial attempt to get the pump to seal in this area was unsuccessful, as oil slowly leaked out of every screw hole. This initial approach was correct, however, and it was simply a case of needing to add another anti-leak mechanism. The method was chose was to machine a “ring” of metal, approximately 1 inch thick, that would be mounted on the bottom of the pump.
Then, instead of using short screws, longer bolts could be used. The additional threads on the bolts allowed for the connection between the system and the pump to finally correctly seal. In addition, this ring will allow the pump to be more stable, which will allow for more accurate accelerometer readings, which will be discussed at a later point.

*Figure Ten – Completed Test Rig*

Figure Ten shows what the test rig looks like after work was completed on it. The total construction time of the system was about one year from the beginning of the project, as the construction proved to be much more difficult than expected. Figure Ten shows that the fluid housing tanks are simply 55 gallon oil drums, which are set approximately 8 feet apart, and
connected using welding fittings on the outside of the tanks, and standard 1 inch internal
diameter PVC piping between the tanks. The physical placement of the sensors can be seen in
our diagram as well. The location of each sensor was carefully thought out before placing them
in our system. For example, our temperature sensors are placed in such a way that they
measure the inlet bulk temperature of the fluid, as well as the outlet temperature of the fluid
as soon as the fluid leaves the tank, for the most accurate reading of temperature rise across
the pump. The pressure sensors have been positioned in a similar manner, in order to get the
most accurate reading of pressure rise across the pump directly. If the pressure or temperature
sensors were placed further down stream, our data would be very susceptible to losses. For
example, the longer fluid travels through any section of piping, the natural skin friction
between the fluid and the inside of the pipe causes progressive pressure losses. Therefore,
sensor locations play a major role in the accuracy of the data that we collect off of the system,
which in turn will affect the accuracy of the characterizations performed on our pump.

Further examination of Figure Ten will confirm the diagram shown in Figure Seven. The
centrifugal pump is located in the bottom of the left hand tank, which will be discussed at
length later in this paper. The final test rig does have some deviations from the initial plan,
however. On Figure Ten, there are two vertical clear PVC pieces of piping attached to each 55
gallon drum. These are commonly referred to as viewing chambers, as they allow the engineers
to see how much fluid is located in each drum at any given time. The reason these viewing
chambers were added is to confirm the readings of the sensors which tell the engineers how
much fluid is located in each tank.
Another modification to the system is the added lab power supply, which can also been seen in Figure Ten. The pump uses a maximum of 32 volts, which initially lead to a power supply sizing estimate of 50 volts would be suffice for this project. However, the research team severely underestimated the amount of power that would be used from the various fluid property sensors. Therefore, a much large power supply had to be chosen. Figure Ten shows that the power supply is large not only in power output, but in size as well, as it takes up a significant amount of space in the lab.

![Image of the power supply in the lab](image)

**Figure Eleven – System Static Pressure Sensors**

In addition to the temperature and pressure sensors needing be located properly, the flow sensors also have to be placed in the correct locations in the system. In order for flow from a pressure source, such as the pump in this research, to be fully developed, any flow sensing element should be placed at least five pump diameters downstream from the pressure source.
Since this system uses 1 inch internal diameter PVC pipe, every flow sensor has to be at least five inches downstream from the pump, in order to ensure flow data accuracy. [1] This is indicated on the diagram very clearly, as both the flow sensor measuring the flow output of the pump, and the flow sensor measuring the flow back into the main tank are at least 10 inches away from the pressure source.

![Image of flow sensor](image)

*Figure Twelve – In-Line Flow Sensor*

When choosing the temperature devices that would be selected for the purposes of measuring the inlet and outlet temperature of the fluid, a number of attempts were needed to get an accurate temperature reading. The first issue that became apparent was leakage associated with taking a temperature measurement inside of a moving fluid. As with any pressurized fluid system, leakages are extremely likely to occur. Due to the “oily” nature of the
fluid being used in this system, leakages are even more likely to occur than if water was the
motive fluid being used in this project. The first attempt to take a temperature measurement
involved the use of simple thermocouples. Installing the thermocouple became the issue at
hand, however, as drilling into PVC pipe in order to install a thermocouple makes the pipe
almost impossible to seal. After multiple failed attempts to seal a PVC pipe with a thermocouple
installed in it, the engineers decided to find a better option. Ultimately, it was decided to use a
Resistance Temperature Detector (RTD) to measure the temperature of the fluid. An RTD
comes with a threaded nipple attached to it, in addition to a threaded female adapter. From
there, it became very easy to drill a hole in the PVC, attach the female adapter to the pipe, and
screw in the RTD. This option was significantly more expensive that using a thermocouple,
however, it was the only option that successfully prevented leaks.

Figure Thirteen – RTD (Resistance Thermometer Detector)
Another type of sensor was used on this project was an accelerometer. An accelerometer measures the amount of acceleration, typically in “g-forces”, in a single axial direction. In this project, an accelerometer was mounted on the sides of the pump in both the x and y direction, in order to see if there are any anomalies in the vibration of the pump.

*Figure Fourteen – Accelerometers and Centrifugal Pump*
Figure Fourteen shows how the accelerometers are oriented in the system, as well as showing the centrifugal pump they are attached to. The accelerometers in this project are used for two purposes. The first purpose is to measure how the pump itself is vibrating as the impeller is spinning. Any cantilevered structure, such as the pump in this project, will have different modes of vibration, and it is important to measure those modes properly, which is what the accelerometers help measure. The second purpose of the accelerometers is measure the revolutions per minute of the pump. The pump has no built in reading of how quickly the pump is operating. When vibration sensing devices are used, the comparison frequency source should be either line frequency, where stable, or a stable independent frequency and the speed determined by observation. [2] Relating this statement to the system, the best way to measure the RPM of the pump is to look at the vibration of the pump, and from there, use a Fast Fourier Transform to determine the rate at which the impeller is spinning. This Fast Fourier Transform analysis can be used to read which frequency has the most noise associated with it. The frequency with the most noise is the frequency at which the pump is spinning. The use of this Fast Fourier Transform analysis is a necessity in this project, as there isn’t another viable way to determine the RPM of the pump.

Figure Fourteen also shows copper tubing wrapped in a helical shape inside of the left most 55 gallon drum. This copper tubing is filled with water and attached to a heat exchanger outside of the system. The reason this feature was added to the experimental test rig is because the bulk fluid temperature needed to be the same at the start of every experiment. However, running an experiment dumps a large amount of heat into the fluid, causing its bulk temperature to rise very rapidly. The only way to cool the fluid back to room temperature in a
time efficient manner is to manually cool it off using the copper tubing heat exchanger. Before this heat exchanger was installed in the system, the fluid would take least 10 minutes between experiments for the fluid to cool down to the appropriate temperature. However, approximately 2 minutes of running this heat exchanger is suffice for cooling the fluid down to the appropriate temperature.

Returning to Figure Ten, there are a few modifications which were to the system made after the system was deemed in working order. Examination of the bottom of the system shows two plastic pallets, which were added after much of the testing had already been completed. These plastic pallets are used to catch any fluid that may leak from the system, or spill when adding additional fluid. In addition to these pallets, a metal stand was added underneath the left most tank. This tank is the one that holds the centrifugal pump, and the stand was added underneath it in order to allow easy access to the pump. The pump screws into the bottom of the tank, however, it was impossible to actually access the pump once the system had fluid in it. The left tank is usually filled with 55 gallons of 1010 mineral oil, commonly known as Brayco 460. The total weight of this fluid is approximately 400 pounds, making it impossible to actually lift the tank in order to access the pump. This metal stand alleviated the problem, and proved to be very helpful once it was installed.

The physical system construction proved to be the most difficult task for the purposes of this project. However, the completed construction of the system did not mean that it was ready to be tested, as a way to interface between the sensors and a computing system was still needed. For the purposes of this project, the engineers decided to use the “Labview 2014” for
data collection purposes. This program allows data to be collected from an outside source of hardware. In the case of this project, the hardware pieces are simply the various flow characteristic sensors, in addition the variable speed centrifugal pump. The program can be tuned to the exact needs of the engineer, and can be used either as a data collection resource, or a way to actually control hardware. For the purposes of this project, Labview 2014 was used in both manners. Initially, the program was used simply as a data collection tool. However, as the research progressed, Labview 2014 was used as a way to control flow demand, as well as the angle of valves in the system. Using Labview in this manner allowed for the creation of a large number of flow types, which allowed for a great range of variables to be used when designing the experiments.

*Figure Fifteen – Labview Interface*
Figure Fifteen shows a screenshot of the Labview program. This portion of the program is called the control panel, as many parts of the experimental test rig can be controlled from this portion of the program. On this figure, you will see buttons which open or close the primary control valve, control the angular velocity of the pump, control whether or not the pump is on or off, and control whether or not data is currently being collected. Having all of the experimental options located centrally allows the experimenters to change various flow and valve conditions very quickly, allowing for a large number of unique experiments to be completed in a relatively short amount of time. In addition to this portion of the program being able to control various parts of the system, it is also used to read the data being outputted by the sensors. Examination of Figure Fifteen will show multiple plots for temperature, pressure, flowrate, and vibration data. All of this data is analyzed in real time, and plotted to the control panel graphs in Labview 2014. Having all of the graphs centrally located allows the researchers on this project to quickly determine if there is a problem with the system. In addition, having the plots organized in this fashion allows the researchers to determine general flow trends of the system, just from watching these graphs change during testing.

Though the construction of the experimental test rig was a long and difficult process, the researchers on this project are extremely happy with the finished product of the system. Overall, the system meets the strict requirements of quality set for us by the advisor. In addition, the system allows us to collect very accurate data, due to the fact that the sensors are located in very close proximity to the DAQ devices, and the computer collecting the data. Setting up the system in this manner means that only a small amount of wiring was used, which means that there is significantly less signal loss than there normally would be in an
experimental set up. In general, however, the system produces very accurate data in a simple
time saving manner. The overall design of the system makes running an experiment extremely
simple, as just about everything can be controlled from the local computer, which has allowed
the results gathered in this project to be very accurate.
Chapter Three – Standard Pump Characterization, Results and Analysis

Characterizing a pump's performance has been an area of scientific study since the centrifugal pump was introduced into modern society. The earliest attempts at pump characterization appear to be linked to the first attempts to create water and sewage systems in major cities. This is logical, as engineers in this situation would need to understand how much fluid can be moved in a certain amount of time. The easiest way to understand the performance of the pumps in this situation would be to characterize the pump in a useful manner. Performance analysis requires repeatable measurements of process parameters such as temperature, pressure, flow, displacement, speed, power and time. [3]

Figure Sixteen – Standard Single Speed Pump Characterization
Figure Sixteen shows an example of a standard pump characterization for a single speed pump. Any centrifugal pump manufacturer will include a characterization similar to the one seen in Figure Sixteen in the package with the pump itself. The layout of this characterization is standard across the entire field of mechanical engineering, meaning that any purchased pump will have the same variables plotted on the graph, such as pump efficiency and head capacity. The first thing to notice on Figure Sixteen is that the X-axis is labeled “Flow Per Minute”. Many manufacturers will refer to volumetric flow rate as “flow per minute”. However, for the purposes of this paper, volumetric flow rate will be referred to as “capacity”. Capacity is simply a measurement of how much fluid can be moved over a standard unit of time, either in seconds or minutes. The standard measurement of capacity in the English system is gallons per minute, usually referred to in centrifugal pump literature as “GPM”.

Next, the Y-axis should be examined. On Figure Sixteen, the Y-axis is labeled as “total dynamic head”. Most engineering textbooks have the y-axis of a centrifugal pump characterization graph labeled as simply “head”. Head is an engineering term used to describe the amount of pressure being generated from the pump. The measurement of head is simply the pressure at the outlet of the pump minus the pressure at the inlet of the pump. When advertising the “size” of a centrifugal pump, any centrifugal pump manufacturer will “rate” their pump to the maximum head and capacity. In doing this, any person that knows their flow demand will be easily able to select a pump that fits their needs.

In order to perform the characterization for a single speed pump, a valve must be attached to the outlet of the pump. From here, varying the valve position creates a change in
flow rate. For example, when the valve is fully open, the pump will operate at its maximum capacity. However, closing the valve will change the amount of fluid being moved by the pump, simply due to the obstruction of the fluid’s path. In addition to the valve being used to change the capacity of the pump, changing how intrusive the obstruction is will change the head rise across the pump as well. As the valve is closed, the pump will naturally see a drastic rise in head, as fluid being pumped will become heavily pressurized as it moves through a closing orifice. This pressurization of the fluid must be counterbalanced by the pump in order to maintain flow in the direction it was already travelling. As such, the head rise across the pump will increase during this phenomena, which can be seen as the “System Head Curve” plot on Figure Sixteen.

The other variables plotted on a typical centrifugal pump characterization curve are pump power consumption, and pump efficiency. Both of these variables are represented on Figure Sixteen as “Motor Input” and “Pump Efficiency”, respectively. The general trends of the plotted variables shown on the example pump characterization are another important thing to note from this plot. Any time a pump is being characterized, the variables should follow the same shapes as the plots shown in the example. For instance, the shape of the pump efficiency curve should always be an upside down parabola. If the plot is not shaped in this manner, there is an issue with the way the characterization was performed. Either there was a problem with the experiments themselves, or a problem with the data that was collected off the pump. Either way, there is clearly an issue that needs to be addressed if the plots do not manifest in a similar manner to the one shown above. This fact was very helpful when performing the analysis on
the experimental centrifugal pump. If the data from the pump did not match the general shapes shown in the example plot, a mistake was clearly made.

The pump being used in this study is, as mentioned previously, a variable speed centrifugal pump. A variable speed pump is a pump that increases its rotational speed, in revolutions per minute, as a greater input power is supplied to the pump. As discussed above, a single speed pump is generally characterized by attaching a valve to the pump, and varying the valve position. In doing this, a great range of capacities and head values can be generated. A variable speed pump is characterized in the exact same manner. The difference is that pump speed is a second independent variable that has been introduced into the characterization. Therefore, in the case of variable speed pumps, the independent variables in the characterization are control valve angle and the rotational speed of the pump.

There are two independent variables in the system, which causes a performance plot to shift from a standard two-dimensional plot into a three-dimensional plot, with two independent variables and one dependent variable. Generally, it is hard for an engineer to get anything useful from a three-dimensional plot, so most three dimensional plots are typically turned into a two-dimensional contour plot. A contour plot simply turns a three dimensional plot into a two dimensional plot using level contour lines, which are placed at user defined increments.

As discussed above, the two independent variables in the system are the valve position and the rotational speed of the centrifugal pump. Logically, it was determined that varying these two independent variables is an excellent spot to begin designing experiments. The
centrifugal pump operates on a rotational speed between six thousand and eleven thousand revolutions per minute. It was decided for the purposes of this project that varying the rotational speed of the pump every one thousand revolutions per minute would be an acceptable resolution for this project. As far as the control valve is concerned, obviously any valve has two extreme positions of fully open and fully closed. For the purposes of this project, a position of zero degrees is considered to be fully closed, while a position of ninety degrees is considered to be fully open. For the purposes of this project, it was decided that an appropriate resolution for the experiments would be varying the valve angle in ten degree increments.

<table>
<thead>
<tr>
<th>Angle</th>
<th>0</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
<th>60</th>
<th>70</th>
<th>80</th>
<th>90</th>
</tr>
</thead>
<tbody>
<tr>
<td>RPM</td>
<td>6000</td>
<td>7000</td>
<td>8000</td>
<td>9000</td>
<td>10000</td>
<td>11000</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>1</td>
<td>11</td>
<td>21</td>
<td>31</td>
<td>41</td>
<td>51</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>2</td>
<td>12</td>
<td>22</td>
<td>32</td>
<td>42</td>
<td>52</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>3</td>
<td>13</td>
<td>23</td>
<td>33</td>
<td>43</td>
<td>53</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>4</td>
<td>14</td>
<td>24</td>
<td>34</td>
<td>44</td>
<td>54</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>5</td>
<td>15</td>
<td>25</td>
<td>35</td>
<td>45</td>
<td>55</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>6</td>
<td>16</td>
<td>26</td>
<td>36</td>
<td>46</td>
<td>56</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>60</td>
<td>7</td>
<td>17</td>
<td>27</td>
<td>37</td>
<td>47</td>
<td>57</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>70</td>
<td>8</td>
<td>18</td>
<td>28</td>
<td>38</td>
<td>48</td>
<td>58</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>80</td>
<td>9</td>
<td>19</td>
<td>29</td>
<td>39</td>
<td>49</td>
<td>59</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>90</td>
<td>10</td>
<td>20</td>
<td>30</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Figure Seventeen – Initial Experimental Matrix*

Figure Seventeen shows the initial experimental matrix that was created, based on the resolutions stated above. This matrix is very simple, and systematically goes through all of the different combinations of experiments. This initial experimental matrix has a total of sixty different experiments that were run, which is quite a large number of unique experiments.
These initial experiments were intended to give a good base line for what should be expected from the pump performance wise.

After designing this experimental matrix, it became important to determine a method of testing that would ensure accuracy of the data. When testing any type of fluid flow, any variation in testing will create a very large error in the accuracy of the data collected. In the past, fluid flow experimental data has been ruined due to changes in the environment around the system. For example, a change in the weather can cause the air pressure in the room to change up to as much as half of a pound per square inch. In order to negate this problem, an extra pressure sensor was added to measure the ambient air pressure in the room. Then, the difference between this ambient pressure and the pressure in the system is measured as the “gauge” pressure of the system. In measuring “gauge” pressure, the risk of any environmental parameters affecting the system is eliminated.

Another way in which the accuracy of the data was insured is that the experiments were completed in a relatively short amount of time. The entire experimental matrix shown in Figure Seventeen was completed in the span of a week. Completing the experiments in this small time frame allowed us to establish a routine for the most efficient testing. For example, the engineers were able to determine that the fluid needed approximately five minutes of cooling time between experimental runs. During these five minutes, the settings of the speed of the pump and the angle of the pump were set to the appropriate values. In addition, the data collected on the previous experiment was examined for any anomalies. If a problem was detected, the previous experiment was run again instead of moving onto the next experiment.
Figure Eighteen is the first plot generated from the initial experimental test matrix. It shows the performance of the centrifugal pump based on standard centrifugal pump parameters, with capacity on the X-axis and head on the Y-axis. As mentioned previously in this paper, the variation in the capacity of the system comes from the variation of the angle of the control valve attached to the centrifugal pump. Referring back to Figure Sixteen, the user of this chart can clearly see that the trend represented by the head-capacity plot is matched by the trend seen in Figure Eighteen. As stated earlier, if the shape of the plot did not match the shape of the plot represented in various literature, then a problem with the data has occurred and would have to be rerun. However, this is not the case, and the data is indeed usable. A few
things should be examined on Figure Eighteen. The first is that a lower capacity, along any given RPM value, always results in a higher head value. This is due to the fact that the lower capacity is driven by the control valve. A small capacity means that the valve is choking down the amount of flow being generated by the pump, which in turn creates a greater amount of stress on the pump. In order to counteract this stress, the pump will produce a greater amount of head. This phenomenon is well documented in literature, and will be seen throughout the presentation of this research.

![Plot of Capacity Versus Efficiency for Various Pump RPMs](image)

*Figure Nineteen – Capacity-Efficiency Plot From Initial Experimental Matrix*

Figure Nineteen is the capacity versus efficiency plot from the first round of testing. Again, referring back to Figure Sixteen, the user of the graph can see that the general shaping
and trends of the experimentally generated plot match the trends expected by literature. Although the shaping seen in this figure is what the expected shape, the actual results of the experimental data are shifted downward from the expected results. Further examination of Figure Sixteen will lead the user to see that a general expected value of a peak efficiency to be around 50 percent. However, the centrifugal pump reaches a maximum of approximately 30 percent. This reduced efficiency is not a sign of experimental failure, however. Instead, it is simply the result of a small capacity pump being used. The example plot in Figure Sixteen is generated from a pump with a much higher capacity than the one being used in this project. This phenomenon can be understood by examining the equation used to calculate pump efficiency in this project. As with any efficiency, this pump’s efficiency is measured as a ratio of the output power to the input power. In addition, pump efficiency can only be derived from the power input required. \[\text{Pump Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{\text{Pump Work}}{\text{Horsepower}}\]

\[
\text{Pump Work} = \frac{\text{(Specific Gravity of Fluid)}(\text{Flow Rate})(\text{Head})}{3960}
\]

and

\[
\text{Horsepower} = \frac{(\text{Wattage Input})(\text{Pump Motor Efficiency})}{746}
\]

The equations shown above are all based on the fact that any efficiency is always a ratio of the output power to the input power. From this initial ratio, it becomes important to identify exactly what represents the input and output for a centrifugal pump. Literature suggests that
the output of a pump is always given defined in terms of “work”, while the input to a pump is always some form of horsepower, which is electrical horsepower for the purposes of this project. Examination of the “pump work” equation will show the user that the two measurements of “flow rate” and “head” are variables which are being directly measured in the experimental test rig, which allows for easy calculation of the amount of work being done by the pump at any given time. The final variable which must be determined in order to make the calculation of pump work is the “specific gravity of fluid”. Specific gravity is a ratio of the density of the fluid to the density of water. A specific gravity below 1 means that the fluid is less dense than water, while a specific gravity above 1 means that the fluid is more dense than water. The specific gravity for any fluid can be looked up in any fluid mechanics textbook, or by contacting the manufacturer. Brayco 460, which is more common known as 1010 mineral oil, is being used in this project, meaning that this fluid does not occur naturally. Therefore, consultation with the manufacturer became necessary, which resulted in a specific gravity of 0.87 being used as the specific gravity value for this project.

The final equation which had to be satisfied in order to create a satisfactory value of pump efficiency is the horsepower equation. The wattage input is whatever the users of the test rig determine using the lab power supply. The pump operates on a range of 14 to 32 volts, which can be converted to watts by multiplying the voltage value by the current value of the system. Finally, the motor efficiency needs to be determined. Determining the motor efficiency proved to be a difficult challenge for the research group. After running a couple of experiments to determine this number, the researchers determined that a viable value for this variable could not be determined. Instead, it became much easier to contact the manufacturer and ask
about the horsepower rating of the pump. As it turns out, the horsepower value for the pump is exactly 1 horsepower, as determined through technical data provided to us from Intertechnique.

Examination of these 3 equations can be used to understand why an efficiency curve has an inverse parabolic shape. First, an efficiency curve has a parabolic shape due to the value of motor efficiency. A motor always operates at its most efficient point in the center of its operating range. This fact is why cars get a higher amount of miles per gallon in the 40 to 50 mile per hour range, and get significantly less miles per gallon above or below this range. The shape of a motor’s efficiency is always parabolic in nature, as well. Being that the motor efficiency is in the denominator of the pump efficiency equation, the shape of any pump’s efficiency will always be an inverted parabola. This phenomenon can be seen in the example pump characterization in Figure Sixteen, as well as the results generated in Figure Nineteen.
Figure Twenty – Three Dimensional Plot of Capacity, Head, and Efficiency
Figure Twenty shows a three dimensional plot of the three most significant pump performance characteristics, which are capacity, head, and efficiency. In most pump characterizations, these variables are kept separate from each other, plotting only two of these three variables against each other at a given time. In doing this, most pump characterizations have three distinct plots which are looked at; capacity versus head, capacity versus efficiency, and head versus efficiency. However, it would be even more useful to plot all of these variables against each other at the same time. In doing this, a three dimensional plot is created, something which seems to be unique to this project.

The benefits of presenting pump data in this fashion are numerous. However, properly being able to read this graph requires a plotting software such as MATLAB, in order to correctly find a desired location on the graph. A very distinct benefit of presenting results in this fashion is that the pump’s maximum efficiency can be very easily determined when looking at the results in this manner. Utilizing MATLAB or other software, a simple maximum command can be run on the data generated from the experiments. This maximum command will find the global maximum for pump efficiency. Once this global value has been determined, the values of capacity and head which cause this maximum can then be linked to this point. Only the level of fluid in the system, as well as the speed at which the pump is operating, would need to be adjusted.
Figure Twenty One – Plot of Rotational Pump Speed vs. Pump Power Consumption

Figure Twenty One shows a plot of the power consumption of the pump, based on various capacities. This plot is extremely useful for an engineer for a number of reasons. For example, a pump this size could be installed on a non-crucial system on a plane, such as waste removal. If a situation were to occur that would require the fuel pumps to the plane’s engine to provide more fuel, then it would be possible to lower the power consumption of this non-crucial pump to a level which would allow the plane’s fuel pump to receive the power it needs to maintain the operation desired by the pilots. An effective way to maximize energy cost savings is to use the well known 80-20 rule. Applied to pumps, this means that 80% of power consumption savings can be achieved from 20% of the pump systems. [4] Meaning, the top 20%
of pumps (in terms of power consumption) represent 80% of the cost reduction potential for that facility. This should illustrate that small changes in power consumption for 1/5 of the pumps currently in operation at a facility can mean major savings for that facility.

Figure Twenty One shows us that the power differential between 11000 RPM and 6000 RPM is, at a maximum, about 700 watts. This is amount of power which could be used in other aircraft systems during times of high power demand from the aircraft, such as takeoff or landing. In addition, if an emergency situation were to occur which would require the plane to accelerate very quickly, this pump could be completely turned off by the aircrafts onboard computer, in order to provide up to an additional 770 watts of power to other systems on the aircraft.

![Figure Twenty Two – Plot of Capacity Versus Bulk Fluid Temperature](image)
Figure Twenty Two is useful in determining heat generation of the variable speed centrifugal pump. This plot shows capacity versus bulk fluid temperature of the pump, with the thicker lines representing the temperature of the fluid at the inlet of the pump, and the thinner lines representing the temperature of the fluid after the fluid has passed through the pump, shortly after the outlet of the pump. In addition, it was noted that the bulk fluid temperature does not increase throughout the length of a given experiment. Knowing this means that the pump will not cause excessive change in fluid temperature for bulk amounts of fluid. This fact is especially helpful when dealing with fluids that have a flammable temperature point, commonly known as a “flash point”. The fluid on this project, Brayco 460, has a flash point of 133 degrees Celsius. From Figure Twenty Two, it is possible to see that the fluid temperature on this project does not exceed 23 degrees Celsius, which is well within the safe temperature operating range for this project. This pump, however, can and will be used with any type of aerospace lubricating fluid. This includes but is not limited to brake fluid and various coolants, both of which generally have a flash point lower than that of Brayco 460. The results shown indicate that unless the pump is operated for an extremely long amount of time continuously (on the order of days), the pump will not generate a large enough amount of heat to cause the operating fluid to reach a critical level.

Figures Eighteen through Twenty Two represent the relevant results generated from the first experimental test matrix. The results from this first matrix are in the opinion very useful for applications using this specific pump. However, it was decided that more data could be generated which would give further insight into the performance of the variable speed
centrifugal pump. Since the first batch of data was plotted based on the rotational speed of the pump, it was determined that it would be an excellent idea to have the next batch of data be based on the angle of the control valve attached to the pump. From here, a batch of experiments was designed to generate the data needed for this type of analysis. The experiments involve simply setting the control valve to an angle between fully open, and 60 degrees closed. Closing the valve any further past 60 degrees closed results in a very low amount of fluid actually passing through the orifice, which creates wildly inaccurate data. In order to create data over the full range of angle values possible, it was decided to use valve increments of every 10 degrees. In order to generate a change in data among every degree increment, the LabView program was set up to vary the rotational speed of the pump from 6000 RPM to 11000 RPM over a time of two minutes. This experiment was run for every degree increment described above.
Figure Twenty Three shows a standard capacity-head plot for the variable speed centrifugal pump, using valve angle instead of pump rotational speed as seen previously. The first thing that must be noted when using this chart is that the 90 degree increment indicates the valve being “fully open”. In the same vein, the 30 degree increment indicates that the valve is 60 degrees closed. Moving on to the actual data, this chart shows the engineers that as capacity increases over a given time, the head generated by the pump increases as well. This positive slope is maintained for every degree increment tested. However, the slope increases drastically for the situations in which the valve is more closed than it is open. The phenomenon of head increasing more quickly when the control valve is more closed is very easily explainable.
and understood. With a smaller orifice to flow through, the pump must work harder to move
the same amount of fluid through a smaller hole than normal. In order for the pump to do this,
it must create higher amounts of head pressure. The converse of this statement is also true.
With a fully sized orifice, the head pressure required to move the same fluid is significantly less
than with a small orifice, which is illustrated quite clearly in Figure Twenty Three.
Unsurprisingly, this chart also clearly shows that the maximum capacity for a given valve angle
is much higher for situations in which the valve is fully open, as opposed to more closed.

Figure Twenty Four – Plot of Voltage and Efficiency for Various Valve Angles
Figure Twenty Four is another standard pump characterization plot generated from the second round of experiments. In this chart, pump voltage is plotted against efficiency, which should give the user of the pump a good idea of the operating voltage which is associated with efficient power consumption. The first trend that should be noted on this figure is that the operating efficiency of the pump generally increases when the valve is more closed than it is open. Remember, as stated during the analysis done on the previous figure, the line representing 90 degrees is when the control valve is fully open. Therefore, the most efficient control valve angle is actually more closed than it is open.

The next valve position that should be analyzed is the line representing 30 degrees on this chart. Interestingly, the slope of this line is significantly (to the point of being noticeably different) different from the lines representing the other control valve angles. Due to the fact that the derivative of this line has larger absolutely value that the lines around it, this line spans a greater amount of efficiency values than the lines around it. It is believed that this line represents accurate data, and at the same time, leads to some conclusions as to why this data is the way it is. The explanation arrived at is that the valve being 60 degrees closed creates a very small orifice for the fluid to flow through. Generally, when fluid with laminar (smooth) flow passes through a small orifice, that flow is then transitioned to turbulent flow. Turbulent flow is fundamentally different from laminar flow, on a particle physics and mathematical level. Turbulent flow is generally difficult to predict the behavior of, which should explain why the plot for this specific valve angle plot is different than the ones around it.
Figure Twenty Five shows another standard pump characterization plot, this time with capacity on the x-axis and efficiency on the y-axis. The general trends shown on Figure Twenty are again visible on Figure Twenty Five, with the fully open valve position (90 degrees) generating the lowest pump efficiencies. Conversely, the control valve positions that are the most closed (30 and 40 degrees) generate the highest pump efficiencies. Again, the derivatives of the most closed valve positions have the highest absolute values for this chart. In addition, it is possible to see that the 30 and 40 degree plots only span approximately 7 gallons per minute of flow (in the x-direction). This is easily explainable, as a more closed control valve limits the amount of flow through the valve orifice.
Figure Twenty Six – Three Dimensional Surface Plot of Standard Pump Characterization
Figure Twenty Six is the culmination of the plots shown so far from the second round of experiments. This plot shows a standard centrifugal pump characterization in three dimensional form, plotting the three most relevant variables together all at the same time. As you can see from the chart, capacity is on the x-axis, with head on the y-axis, and efficiency plotted on the z-axis. As you can see, the chart has what appears to be “folds” on it. These folds are nothing more than the various valve angles seen in the other figures up until this chart. The further left “fold” represents a valve angle of 30 degrees, while the furthest right “fold” represents the control valve in a fully open position. This chart would be extremely useful to any engineer working on a centrifugal pump similar to this one, provided they had a reliable software program that could read this chart. With MATLAB or another plotting aid, it would be easy to determine the efficiency level that is required for a given project. From there, it would be possible to look up the efficiency that is desired on Figure Twenty Six. Note that a given efficiency likely occurs a large number of times on this Figure Twenty Six. Therefore, if the experimental conditions dictated that an independent variable be a certain value, that value would likely have the desired efficiency associated with it.
The final plot that will be discussed in this chapter is Figure Twenty Seven, which shows the rotational speed of the pump versus the amount of power the pump is consuming. The first thing that the user of this chart may note is that the general trend, or shaping, of the different angle lines on this plot is essentially the same. Each angle line has a positive first derivative, which can be seen as the lines slope from the bottom left of the graph to the top right of the graph. In addition, the graphs have a slight positive second derivative associated with them. This can be seen in the slightly concave shaping of each line. Unsurprisingly, the pump consumes more power when the rotational pump speed is higher for every case. The only
somewhat shocking trend that can be seen on this chart is that the pump utilizes less power when the fluid is being pumped through a smaller orifice, which can be seen by the red line, representing “30 degrees”, being on the lower end of power consumption for every pump rotational speed. Initially, it was expected the pump to utilize more power when pumping into a smaller orifice, as it seems like the pump would have to exert more work to move fluid through a smaller space. However, this is not the case, as it can be seen that the pump actually consumes more power when pumping the fluid through a completely open control valve.

The figures seen and discussed in this chapter provide the complete characterization for the performance of this pump under any given circumstance. A normal single speed pump characterization would be performed by the manufacturer before sale. The analysis done on the variable speed pump, however, is significantly more complicated than a standard single speed pump analysis, as a single speed pump analysis requires the creation of only a single graph. The results discussed in this chapter show that a very large number of plots were created in order to fully describe and characterize the performance of the variable speed centrifugal pump. However, it was decided that non-dimensionalizing the results would bring significantly more academic meaning to the research completed on this project.
Chapter Four – Non-Dimensional Pump Characterization, Results and Analysis

While any centrifugal pump will certainly go through extensive testing in order to arrive at an acceptable pump characterization, very few of these pumps are characterized in a non-dimensional fashion. Firstly, it becomes important to define non-dimensional characterization, and next, it is important to understand the reasons and benefits from characterizing a pump without dimensions.

When designing a centrifugal pump, a number of design variables need to be determined. They are impeller rotational speed, impeller inlet or suction dimensions, impeller outlet diameter, impeller blade number, impeller blade passage geometry, and impeller position relative to the casing. [8] Surely, the shape and size of the impeller in a centrifugal pump is extremely vital to determining the pump’s performance. However, these variables are much more useful when determined in a non-dimensional format.

The development of the specific speed concept was a most important one in the history of centrifugal pump development. At once, it became possible to utilize test and design data on existing pumps to develop new designs of dimensionally similar pumps but of larger or smaller size because the specific speed of a pump remains independent of size. [6] As the “non-dimensional” clause suggests, non-dimensional characterization involves using variables that do not have a unit associated with them. A non-dimensional variable is always created by using an equation which causes all of the units to cancel each other out, leaving only a unitless number.
as the result from the equation. These non-dimensional variables hold a special power mathematically, as their lack of a unit allows for these numbers to be applied universally across different applications. Firstly, for a counter-example of a non-dimensional variable, time is a variable that has a standard dimension. It is easy to see that time is a dimensional variable, as time always has a unit associated with it, which is usually “seconds” in the scientific community.

Unfortunately, there are no extremely common non-dimensional variables that are used in every day life. The most famous non-dimensional variable is undoubtedly the Reynolds number. The Reynolds number is a very famous fluid mechanics equation, which takes a number of standard fluid mechanics variables into account. These variables are inserted into the Reynolds number equation, and all of their dimensional qualities cancel each other out. Running a calculation with the Reynolds number equation leaves a dimensionless quantity, as suggested from the discussion thus far. However, just because this number does not have a dimension associated with it, does not make this number useless. In fact, it is significantly more useful than a dimensional quantity. A Reynolds number of below 2300 indicates that the flow being studied is a smooth, laminar flow, while a Reynolds number of over 4000 indicates that the flow is turbulent. This holds true for an astonishingly high number of fluid types, as well as pipe materials and diameters. In addition to this number being able to explain the type of flow expected from certain parameters, this number is also able to be used in scaling of projects. Take for example, an aircraft wing. The flow over the wing will have a certain Reynolds number associated with it. If the engineers suddenly decided to make the wing 10% bigger, the easiest way to scale the wing while keeping the same flow performance is to use the Reynolds number. Using the initial Reynolds number, a desired length of the wing could be set to 1.1 times the
original length of the wing, while keeping the desired Reynolds number the same. From here, the Reynolds number equation could be rearranged in order to solve for the width or height of the wing. From here, the width and height would be simply determined using basic algebra. In this sense, the Reynolds number is extremely powerful, and without a doubt the most commonly used dimensionless variable used in the field Mechanical Engineering.

While the Reynolds number is an excellent example of the power of a dimensionless value, it is not very applicable to the research on a variable speed centrifugal pump. Instead, centrifugal pump literature suggests that a value called “specific speed” is much more common to research in this area. Specific speed is a value which is commonly used in characterizing the operating speed of any piece of turbomachinery. A normal operating speed is typically defined in terms of revolutions per minute (RPM), or angular velocity (radians per second). Obviously, both of these quantities have dimensions associated with them, which greatly limits the ways they can be used mathematically. However, the addition of specific speed as a variable into the analysis allows for this research to be applied universally across all types of centrifugal pumps. The catch with this type of non-dimensional number, however, is that the analysis performed can only be applied to centrifugal pumps with the same shape of impeller. However, if a pump does have the same shape of impeller as the one being used in this research, the use of specific speed allows this research to be applied directly to the performance of that pump as well, regardless of the size of the pump. The pump can be micro-sized, or very large, and the performance of the pump should be the exact same when using specific speed as a unit of analysis.
The above equation is the unitless equation used to calculate specific speed, which is denoted as $N_s$ in centrifugal pump literature. In this equation, $n$ is pump rotational speed (in radians per second), $Q$ is flowrate (in US gallons per minute for this project), $g$ is the acceleration due to gravity (32.17 ft/s$^2$) and $H$ is total head pressure (in pounds per square inch). Using simple algebra, conversion of all of these variables into English Imperial units will allow all of the units of each variable to cancel out in the final specific speed equation. As suggested previously, the result of this equation will be a dimensionless number.

![Figure Twenty Eight – Chart of Specific Speed BEP Values](image)

Figure Twenty Eight shows a plot with the maximum range of specific speed values on the number line above the images of various impeller shapes. A common type of analysis performed using specific speed involves replacing $H$ (head pressure) with $H(\text{BEP})$ (head pressure at the best efficiency point). In doing this, a singular value of specific speed is generated, which...
is then described as the specific speed BEP. This specific speed BEP value will then be forever associated with that shape of impeller. Optimization of pump hydraulic geometry in terms of the BEP specific speed has taken place empirically and analytically throughout the history of pump development. [7] As stated earlier, it can be clearly seen that the maximum value of specific speed is 15000, while the minimum is around 500. It is important to note that the impellers shown on Figure Twenty Eight represent the impellers specific speed when operating at its best efficiency point (BEP). Obviously, the non-dimensional analysis that will be performed on this project will be analyzed over the centrifugal pump’s full range of specific speed values. From here, it will be possible to determine the specific speed value which causes the pump to operate most efficiently, which will then allow us to determine where the impeller type falls on Figure Twenty Eight. This type of analysis is very important in determining the “area” in which the pump operates. Once the area in which this pump operates has been determined, it will allow other pump users to universally apply this research to other pumps that fall within the same “area”.
Figure Twenty Nine shows a non-dimensional plot of specific speed versus pump efficiency. As you can see from the plot, the specific speed variable does not have a unit associated with it. Because of this, any pump with the same shaped impeller would be able to apply the data shown in Figure Twenty Nine directly to the performance of that pump. One thing that should be noted on this figure is that the range of specific speed values for the centrifugal pump being used in this project is between 6000 and 8500. Referencing Figure Twenty Eight, it is possible to see that the maximum range of values for a pump’s specific speed is anywhere from 500 to about 15000. It is possible from Figure Twenty Nine to see that the more closed the control valve is, the higher the resultant efficiency is. The only anomaly is the
plot representing the “30 degree” valve position. This plot is interestingly below the 40 degree valve position in terms of efficiency. It has been theorized the behavior of the pump for this specific control valve angle is due to a transition from laminar (smooth) flow during the “40 degree” plot, to a turbulent flow during the “30 degree” plot. The turbulence of the fluid in this 30 degree plot will create increased friction between the fluid and the inside of the PVC piping wall. In doing so, the efficiency at which the pump is operating will be lower than the efficiencies seen on the “40 degree” plot.
Figure Thirty shows a plot of the same two variables seen in the previous figure, which are specific speed being plotted versus efficiency. A quick comparison of the graphs shows that the figures are indeed quite different when being plotted based on different trendlines. As you can see, Figure Twenty Nine was plotted on a constant angle trendline basis. For Figure Thirty, the same plot was generated, using constant RPM trendlines instead. The results shown in Figure Thirty tell us a few key pieces of information about the variable speed centrifugal pump. First, it is important to note that this type of analysis would generally be very difficult to do with a single speed centrifugal pump. Examination of the specific speed equation should allow the
user to see that the “n” value represents pump rotational speed. In a single speed centrifugal pump, the “n” value is completely fixed, which would make it impossible to create a matrix with a range of specific speed values. However, the pump is variable speed, allowing for this “n” value to be changed.

Moving on to the actual analysis of Figure Thirty, it is possible to see that the greatest efficiencies occur around a specific speed value of 5000 for every RPM value. This result is a very important step to completing the non-dimensional analysis of the experimental centrifugal pump. The specific speed value of 5000 clearly holds a special place in fully understanding the inner workings of the pump. It is important to remember, however, that the specific speed in general tells engineers more about the performance of the shape of the impeller, and less about the pump itself.
Since the impeller has a BEP (best efficiency point) Specific Speed value of 5000, it becomes important to determine what flow region this corresponds to. Figure Thirty One shows that the impeller falls in the region between “mixed flow” and pure “axial flow”. “Mixed flow” generally refers to a type of flow that start out in the radial direction (outwards from the center of the impeller) and ends in an axial direction, which is perpendicular to the direction of flow as fluid comes off of the impeller. Conversely, “Axial Flow” refers to a type of flow that heads in the direction of the axis of the impeller. This generally can only occur if the vanes are pointing in the same direction of the axis of the impeller.
In an attempt to fully understand why the pump’s impeller falls in between the “mixed flow” and “axial flow” regions on Figure Thirty One, Figure Thirty Two is called upon. The important thing to examine on this figure is the shape of the vanes on the impeller. As stated previously, it is important to remember that radial flow will occur if the vanes are straight vertical (into the page). Conversely, axial flow will occur if the vanes lie in the same plane as the face of the impeller (horizontally). Figure Thirty Two shows us that the vanes are almost completely vertical. However, close examination will show the engineer that they indeed do have a slight upward tilt. This realization allows the engineer to determine that this impeller shape does indeed belong in the “Mixed Flow” region shown on Figure Thirty One. This result is very exciting, as it helps us determine that the data pulled from the experiments, as well as the analysis, has been accurate up to this point.
Figure Thirty Three – Three Dimensional Surface Plot of Non-Dimensional Pump Characterization
The final figure to be discussed in this chapter is Figure Thirty Three, which compiles the results shown so far on non-dimensional analysis into a three dimensional plot. This plot is the culmination of the non-dimensional analysis which has been performed on the centrifugal pump, as the variables plotted in this chart are the 3 variables vital to fully characterizing a pump. As this chart has been analyzed non-dimensionally, it becomes important to replace the rotational speed of the pump (RPM) with Specific Speed, as it will allow the results of this analysis to be applied to other centrifugal pumps with the same shape impeller. The “creases” or “folds” shown on Figure Thirty three represent different valve positions, with the far left crease representing the valve being 60 degrees closed, while the far right crease represents the valve being fully open. It is important to keep in mind that this analysis would not be feasible on a single speed centrifugal pump, as the variation of the specific speed variable would not be possible on a pump with only one speed. Examination of Figure Thirty Three shows the user highest efficiency operating point for this pump occurs at a specific speed value of 5000, as stated previously, and a head value of approximately 25 PSIG.

Again, anyone with a pump that has the same shape of impeller, regardless of the size of the pump, impeller, or vanes, could utilize the data presented in this chapter. The process of utilizing this data with another pump would be extremely simple, as well. Since the results seen in this chapter have been non-dimensionalized using the Specific Speed value, another pump with the same shape impeller and vanes would need to be characterized non-dimensionally as well. Once this has been done, however, the results seen here could directly applied to the new pump, regardless of the actual size of the pump. Meaning, the pump could be on a micro or macro scale, but the performance of the pump would be the same as long as the Specific Speed
values of the new pump matches the specific speed values discussed in this chapter. This example illustrates the full power of utilizing a non-dimensional value, such as Specific Speed, as data characterized in this manner can be applied to other research projects globally. The classification of a variable speed centrifugal pump in a non-dimensional manner is novel to this research project, specifically with three dimensional charts and contour plots being used.
Chapter Five – Overall Project Analysis and Conclusions

As with any graduate research project, the overall goal of the project is always to add something unique and useful to the chosen field of study. It is felt that the work presented in this paper furthers the understanding of variable speed centrifugal pumps in a measurable way. Firstly, the construction of the physical system for experimental testing shows that the students on this project possess a level of technical ability rarely showcased by graduate students. The work done on building this system, while not novel to the field of mechanical engineering, is not to be underestimated. A very large amount of time was invested in preparing the system for data collection, and along the way, many technical were learned.

The study of the inner workings of a centrifugal pump is another key element to the research completed on this project as well. Through the work completed on the physical construction of the system, centrifugal pumps as a mechanical device were studied in a very detailed manner, as test rig and experimental design would be heavily influenced through knowledge of how a centrifugal pump works. Information such as vane shape and size, impeller shape and size, volute shape, pitch, and size played a great role in determining how the system was physically constructed, as well as in determining the battery of experiments. The way this knowledge is presented is common across all centrifugal pumps, whether they are single speed or variable speed.

In addition to the specifications of single speed pumps, a great deal of information was required to be learned about variable speed pumps. While it turns out that the inner workings of a variable speed pump are very similar to that of a single speed pump, that does not mean
that working with a variable speed pump would be equally as convenient. In fact, the addition of a variable speed motor to a centrifugal pump made this project significantly more challenging than it would have been otherwise. Mapping voltages to angular operating velocities of the pump was certainly one of the most challenging aspects of this whole project. Through experimentation, it turned out that there was no way to initially determine the speed at which the pump was operating. The solution to this problem was very complex, as it became necessary to measure the vibration of the pump. In using a fast fourier transform (FFT) on the vibration data generated from the pump, it became possible to measure different resonant frequencies that occur during pump operation. From here, it became possible to isolate the resonant frequency responsible for the spinning motion of the impeller. Once this frequency was determined, the amplitude of this frequency was then mapped to the speed of the impeller, allowing a determination of exactly what speed the pump was operating at. This breakthrough allowed for the beginnings of designing experiments based off pump speed, which is one of the primary variables in which pumps are characterized with.

The experiments used to collect data were designed in a manner that allowed the most common centrifugal pump variables to be the independent variables in the experiments. The variables on this project that were the most controllable were determined to be pump rotational speed, most commonly measured in revolutions per minute, and head, most commonly measured in inches of fluid. As it turns out, these two variables are the variables which play the largest roles in pump characterization. In fact, when purchasing an off the shelf pump, the way of determining if the pump will meet specific flow demands is by examining the head-flow curve. This type of plot is the most commonly cited figure in the field of centrifugal
pumps, as it will tell you the operating speeds of the pump, as well as the pressures that it generates at these speeds. For a single speed pump, the head-flow curves generated are presented in a two dimensional plot. However, a variable speed pump produces head-flow curves in a three dimensional plot.

Over 120 experiments were run in order to fully map the performance characteristics of this specific pump, and it was important to describe the pump in a way which allows this research to be applied to other projects. In order to do this, the help of a non-dimensional variable called specific speed was enlisted. Non-dimensional numbers allow equations and data to be applied universally across projects, as long as certain conditions are met. The condition that needs to be met in order to use the non-dimensional data presented in this paper is that the shape of the vanes need to be the same in another pump. If another pump has similar shaped vanes, then the research presented in this paper can be applied directly to the performance of that pump. The most powerful part of a non-dimensional number is that it allows the data to be applied to another pump regardless of the size of the pump. The pump could be very large or very small, and the data would still be directly applicable to the performance of the pump. In addition, three dimensional plots of pump performance were created using this specific speed value. Presenting the research completed on this project in a three dimensional manner, through the use of non-dimensional variables, is novel to this project. The creation of the plots in this manner will hopefully influence other engineers around the world to present their research in this manner, as it is a more robust way of presenting data than traditional methods. A three dimensional plot using non-dimensional independent variables will allow the user of such a plot to quickly determine the exact flow demand needed.
From here, the user of a plot could determine various other dependent variable values, such as efficiency, temperature, or power consumption at a given flow demand. These graphs are extremely powerful, and present the research done on this project in a clear and concise manner.
References


[9] Vachez, Mr. Jean-Francois. "Intertechnique Correspondence." Interview.

Appendix A – Variable Speed Centrifugal Pump Images, Solidworks Renders and Test Rig Images
**Appendix B – MATLAB Code**

Note: Raw data is needed for MATLAB programs to produce figures.

**Constant RPM Experiment Analysis Program**

```matlab
function data_compile_crp

rpm = [6,7,8,9,10,11];

rpm2 = rpm*10^3;
cal_d_p_m = 5.9994;
cal_d_p_a = -0.0385;
cal_m_p_m = 6.0018;
cal_m_p_a = -0.0425;
cal_p_p_m = 9.987;
cal_p_p_a = -0.089;
cal_a_p_m = 6.0072;
cal_a_p_a = -0.024;
cal_f_m = 6.4076;
cal_f_a = -1.2904;
cal_v_m = 4.981469;
cal_v_a = 0;
cal_c_m = 5.9427;
cal_c_a = 0.0432;
v_m = 4.98;
i_m = 5.9427;
i_a = 0.0432;

for ii = 1:6
    filename = [num2str(rpm(ii)),'000rpm.tdf'];
    output = tdfread(filename);
    dtp = output.Drain_Tank_Pressure*cal_d_p_m+cal_d_p_a;
    mtp = output.Main_Tank_Pressure*cal_m_p_m+cal_m_p_a;
    pp = output.Pump_Pressure*cal_p_p_m+cal_p_p_a;
    ap = output.Ambient_Pressure*cal_a_p_m+cal_a_p_a;
    flow = output.Flowrate*cal_f_m+cal_f_a;
    t_in = output.Inlet_Temperature*cal_tin_m-cal_tin_a;
    t_out = output.Outlet_Temperature*cal_tout_m-cal_tout_a;
    tin(ii,1:430) = t_in(1:430);
    tout(ii,1:430) = t_out(1:430);
```

drain_p(ii,1:430) = dtp(1:430);
tank_p(ii,1:430) = mtp(1:430);
pressure_rise(ii,1:430) = pp(1:430)-mtp(1:430);
actual_v(ii,1:430) = output.Voltage(1:430)*v_m;
actual_i(ii,1:430) = output.Current(1:430)*i_m+i_a;
gpm(ii,1:430) = flow(1:430);

efficiency(ii,1:430) =
(0.435*actual_v(ii,1:430).*actual_i(ii,1:430))./watts(ii,1:430);

ss(ii,1:430) =
rpm2(ii)*(gpm(ii,1:430).^.5)./(pressure_rise(ii,1:430).^0.75);

figure(1);plot(gpm(ii,1:430),pressure_rise(ii,1:430),'Color',colors(ii),'LineWidth',2);
xlabel('Capacity (Gallons Per Minute)');ylabel('Head (Pounds Per
Square Inch)');
legend('6000 RPM','7000 RPM','8000 RPM','9000 RPM','10000 RPM','11000
RPM')
title('Plot of Capacity and Head for Various Pump RPMs');hold on

figure(2);plot(gpm(ii,1:430),efficency(ii,1:430),'Color',colors(ii),'LineWidth',2);
figure(3);plot(ss(ii,1:430),efficency(ii,1:430),'Color',colors(ii),'LineWidth',2);

figure(6);plot(gpm(ii,1:430),tin(ii,1:430),'Color',colors(ii),'LineWidth',1);
hold on
xaxis('Capacity (GPM)');ylabel('Bulk Fluid Temperature (Degrees
 Celcius)');
title('Plot of Inlet and Outlet Temperature Versus Capacity');hold on

plot(gpm(ii,1:430),tout(ii,1:430),'Color',colors(ii),'LineWidth',3);
figure(9);plot(gpm(ii,1:430),watts(ii,1:430),'Color',colors(ii),'LineWidth',2);

xlabel('Capacity (GPM)');ylabel('Power Consumption (Watts)');
Constant Position Analysis Program

```matlab
function data_compile_cpos
    close all;
    rpm = [6,7,8,9,10,11];
    deg = [30,40,50,60,70,80,90];
    rpm2 = rpm*10^3;
    cal_d_p_m = 5.9994;
    cal_d_p_a = -0.0385;
    cal_m_p_m = 6.0018;
    cal_m_p_a = -0.0425;
    cal_p_p_m = 9.987;
    cal_p_p_a = -0.089;
    cal_a_p_m = 6.0072;
    cal_a_p_a = -0.024;
    cal_f_m = 6.4076;
    cal_f_a = -1.2904;
    cal_v_m = 4.981469;
    cal_v_a = 0;
    cal_c_m = 5.9427;
    cal_c_a = 0.0432;
    v_m = 4.98;
    i_m = 5.9427;
    i_a = 0.0432;
    eff_cal = 0.435;
    colors = ['r','g','k','b','c','m','y'];
    vol_m = 1103.3; vol_a = 1735.2;
    for ii = 1:7
        filename = [num2str(deg(ii)),'.tdf'];
        output = tdfread(filename);
        dtp = output.Drain_Tank_Pressure*cal_d_p_m+cal_d_p_a;
        % Further processing...
    end
```
```
mtp = output.Main_Tank_Pressure*cal_m_p_m+cal_m_p_a;
pp = output.Pump_Pressure*cal_p_p_m+cal_p_p_a;
ap = output.Ambient_Pressure*cal_a_p_m+cal_a_p_a;
flow = output.Flowrate*cal_f_m+cal_f_a;
drain_p(ii,200:648) = dtp(200:648);
tank_p(ii,200:648) = mtp(200:648);
pressure_rise(ii,200:648) = pp(200:648)-mtp(200:648);
actual_v(ii,200:648) = output.Voltage(200:648)*v_m;
actual_i(ii,200:648) = output.Current(200:648)*i_m+i_a;
watts(ii,200:648) = actual_v(ii,200:648).*actual_i(ii,200:648);
gpm(ii,200:648) = flow(200:648);
rpm3=(output.Voltage*vol_m+vol_a)';
rpm4(ii,200:648)=rpm3(200:648);

efficency(ii,200:648) =
(0.435*gpm(ii,200:648).*pressure_rise(ii,200:648))./watts(ii,200:648);
efficency(efficency >= 1|efficency <= 0)=0;
ss(ii,200:648) =
rpm4(ii,200:648).*(gpm(ii,200:648).^.5)./(pressure_rise(ii,200:648).^0.75);
ss(imag(ss)~=0)=0;

figure(1);plot(gpm(ii,200:648),pressure_rise(ii,200:648),'
'Color',colors(ii),'LineWidth',2);
xlabel('Capacity (Gallons Per Minute)');ylabel('Head (Pounds Per
Square Inch)');
legend('30 Degrees','40 Degrees','50 Degrees','60 Degrees','70
Degrees','80 Degrees','90 Degrees')
title('Plot of Capacity and Head for Various Valve Angles');hold on

figure(2);plot(actual_v(ii,200:648),efficency(ii,200:648),'
'Color',colors(ii),'
'LineWidth',2);hold on
xlabel('Pump Voltage');ylabel('Efficency');
legend('30 Degrees','40 Degrees','50 Degrees','60 Degrees','70
Degrees','80 Degrees','90 Degrees')
title('Plot of Voltage Versus Flow Rate for Various Valve
Angles');hold on

figure(3);plot(gpm(ii,200:648),efficency(ii,200:648),'
'Color',colors(ii),'
'LineWidth',2);hold on
xlabel('Capacity (Gallons Per Minute)');ylabel('Efficency');
legend('30 Degrees','40 Degrees','50 Degrees','60 Degrees','70
Degrees','80 Degrees','90 Degrees')
title('Plot of Capacity Versus Efficency for Various Pump RPMs');hold on

figure(7);plot(rpm4(ii,200:648),efficency(ii,200:648),'
'Color',colors(ii),'
'LineWidth',2);hold on
xlabel('Specific Speed');ylabel('Efficency');
legend('30 Degrees','40 Degrees','50 Degrees','60 Degrees','70
Degrees','80 Degrees','90 Degrees')
title('Plot of Capacity Versus Efficency for Various Valve Positions');hold on
figure(9); plot(rpm4(ii,200:648),watts(ii,200:648),'Color',colors(ii),'LineWidth',2); hold on
    xlabel('Rotational Pump Speed (RPM)'); ylabel('Power Consumption (Watts)');
    legend('30 Degrees','40 Degrees','50 Degrees','60 Degrees','70 Degrees','80 Degrees','90 Degrees')
    title('Plot of Capacity Versus Efficiency for Various Valve Positions'); hold on
end

figure(4)
surf(gpm(1:7,250:end),pressure_rise(1:7,250:end),efficiency(1:7,250:end))
xlabel('Capacity (Gallons Per Minute)'); ylabel('Head (Pounds Per Square Inch)');
zlabel('Efficiency'); title('Surface Plot of Pump Performance');
figure(5)
surf(ss(1:7,250:end),pressure_rise(1:7,250:end),efficiency(1:7,250:end))
xlabel('Specific Speed (Unitless)'); ylabel('Head (Pounds Per Square Inch)');
zlabel('Efficiency'); title('Non-Dimensional Surface Plot of Pump Performance');
end