EXPERIMENTAL PERFORMANCE OF A VAPOR DELIVERY SYSTEM

A thesis submitted in partial fulfillment of the requirements
For the degree of Master of Science in
Mechanical Engineering

By

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Pones Rob

Jeffrey J. Bunting
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ABSTRACT

EXPERIMENTAL PERFORMANCE OF A

VAPOR DELIVERY SYSTEM

By

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Master of Science in
Mechanical Engineering

Engineering can be described as the real world application of theoretical concepts; it is the will of an engineer to form a practical means to accomplish the application of such concepts. This study sought to put engineering application to the test by designing, constructing, and testing an apparatus that would implement a concept envisioned by individuals in industry and ultimately prove its feasibility.

Pratt & Whitney, Rocketdyne originated the idea for the concept studied here and much of the problem statement and conditions comes from them. The concept analyzed was the notion of using a specific fluid close to its critical point and, through the use of excess thermal energy, vaporize the fluid to create a distinguishable and maintainable constant pressure vapor flow. This type of flow is commonly used to purge fuel tanks in propulsion systems, and is typically supplied by high pressure helium flowing through a
mechanical pressure regulator. The new concept would control the supplied vapor pressure by controlling the temperature of a liquid-vapor mixture. Rocketdyne defines ‘excess thermal’ energy as extra heat energy simply being wasted by being expelled to the surroundings, which is not unusual for propulsion systems. The goal of the experiments was to see if this high pressure vapor could purge a secondary fluid at a constant rate, i.e. the flow rates of both fluids and the pressure of the high pressure vapor could be constant over an appropriate time interval.

Fundamental thermal-fluids equations were used to design the experimental apparatus. The Bernoulli equation was used to choose orifice sizes that would produce flow rates in a desired range. A first law of thermodynamics analysis of the vaporization process in the working fluid tank was used to estimate the necessary heat transfer rate to produce a steady vapor flow. Heat transfer correlations for boiling heat transfer were used to predict the temperature (and pressure) drop in the tank to produce the necessary heat rate.

This study picked up where a previous CSUN student, Levon Gevorkyan, left off in determining appropriate fluids that could be used to prove this concept. For these experiments, carbon dioxide was chosen for its desirable density ratio at easily producible temperatures, its availability, and its low toxicity. The design of the experimental apparatus involved sizing the physical system, constructing the system, testing the system, and completing post data collection analysis.

This study was deemed a success based on the fact that a working fluid was selected and used in a system that showed that steady conditions were created within a certain range of operable flow rates. Only the lowest flow rates tested were capable of
producing steady conditions. It is recommended that further testing be completed at lower flow rates to more completely define the relationship between flow rate and accurate pressure control.
Chapter 1: Introduction

1.1. Purpose

The main objective of this study was to experimentally determine the validity of using a saturated fluid (hereafter referred to as working fluid) close to its critical point to act as a pressurized vapor delivery source. A pressurized vapor delivery source is defined as a physical system which utilizes thermal energy to produce mechanical energy via the production and delivery of a sustainable high pressure vapor flow. This concept was originally motivated by the desire to create an alternative to high pressure helium systems fitted with mechanical pressure regulators which are used to purge fuel tanks in propulsion systems. It is based on the notion that excess thermal energy, which is often available in propulsion systems, could be used to control the temperature of a saturated fluid close to its critical point in order to create a vapor source at a set pressure. This study outlines the steps taken to design, construct, and test this vaporization process, to see if the concept is feasible under appropriate conditions. The apparatus used in this study was designed to provide an uncomplicated means of gathering data regarding the vaporization process with the understanding that the physical process as modeled may need to be housed in a smaller and more efficient system to accommodate real world applications. The experimental apparatus was completely designed and constructed from the ground up in order to precisely fit the process requirements which are outlined in Sections 1.3 and 1.4.
1.2. **Background**

1.2.1. **Pratt & Whitney, Rocketdyne**

This study was funded and supported by Pratt & Whitney, Rocketdyne (PWR). PWR is one of the most famous names in the American space program; from hypersonic jet engines to the space shuttle’s main engines, Rocketdyne has been an integral part of many modern aerospace applications. As with any large engineering company, new concepts arise on a daily basis within Pratt & Whitney, Rocketdyne and for the ideas that have a high probability for being truly innovative, the company must complete rigorous testing to prove their validity. The concepts outlined in this study were brought to California State University, Northridge (CSUN) by PWR to utilize the university’s talent and infrastructure to perform and analyze experiments which would support the concept validation. Originated by a rather ingenious individual from Rocketdyne, this idea of using a fluid close to its critical point to create a high pressure vapor was originally brought to the Mechanical Engineering Department at CSUN in spring 2010. PWR provided research funding for the initial part of the project, which focused on identifying candidate working fluids which had favorable thermodynamic characteristics. Subsequently, they supplied additional funding during fall of 2010 and spring 2012 to physically test their concept, which is the work detailed in this thesis. It is important to note, however, that some information is Pratt & Whitney, Rocketdyne proprietary and has been omitted from this document.

1.2.2. **College of Engineering & Computer Science – Cal State University, Northridge**

The College of Engineering & Computer Science (CECS) at CSUN was an integral partner in this study. Like many engineering colleges, the CSUN CECS prides
itself in working closely with industry to not only provide learning experiences for students, but also a means for a student to make the transition from his/her college career to working career. The CECS has worked closely with PWR to bring this study to fruition by providing the personnel and facilities needed to house the testing of this concept. All of the testing was completed in the CSUN CECS Rocket Engine Test Cell (RETC). Figures are available in Appendix D.

The CECS provided the original student researcher who completed the bulk of the background research regarding potential working fluids and their thermodynamic properties. The first researcher provided all the details that lead to the research for this study and without his work, the experimental study would not have been possible.

1.3. **Problem Statement**

The overall goal of this study was to create a constant pressure vapor supply by taking thermal energy and using it to control the vaporizing process of a saturated liquid close to its critical point. The vapor supply was used to purge a secondary fluid (water) from a fuel tank borrowed from CSUN’s rocket engine test apparatus, in order to simulate the application of this concept in a propulsion system. The volumetric flow rate was controlled by an orifice plate located downstream of the secondary fluid tank. The specific operating conditions and the type of working fluid used were agreed upon by both the CECS and PWR personnel. Frequent meetings were held during the design phase to insure that the experimental results obtained would be consistent with PWR’s objectives.
1.4. **Problem Conditions**

It was important to determine the specific means by which this goal would be met. The proper parameters needed to be selected to size the system so that desired flow rates and run times could be achieved to demonstrate the ability of the system to operate at steady state. Some of the parameters which needed to be determined were the identity of the secondary fluid and its associated properties, the amount of secondary fluid to purge, the selection of the working fluid, the amount of working fluid, and its associated thermophysical properties.

It should be noted again that a study regarding applicable fluids for this type of process was completed prior to this work, and that a selection of a different working fluid for use in the experiments only arose due to proprietary and safety issues. Consequently, a suitable working fluid was selected to provide the required data. This selection process became part of the problem conditions and thus the remainder of the apparatus design was somewhat tailored to this particular fluid. A replacement fluid was deemed acceptable if it resembled the properties and qualities of the original group of fluids identified in the initial study. The working fluid was selected as carbon dioxide because it is easily available, safe to handle, and produced a desirable pressure at 32°F, which is easy to control and maintain with an ice bath. Another important characteristic was ratio of the saturated vapor and liquid densities at 32°F, as this determines how much volume can be purged from an initial liquid volume.

The secondary fluid was simply to act as a liquid to be purged, and so water was selected. Water was easy to come by, easy to handle, easy to dispose of, and non-reactive with the working fluid, making it a very attractive selection.
The purge volume was somewhat arbitrary since the system process could be sized to accommodate any reasonable value. Thus, the volume of the secondary fluid tank was over-sized to make sure the process did not run out of fluid to purge during a typical run. An existing fuel tank from CSUN’s rocket lab proved to be an ideal choice since it already had the appropriate internal plumbing for the purge process.

Once the purge volume was set, the working fluid tank size could be based on the requirement to either completely purge the secondary fluid or have secondary fluid left over. Note that Rocketdyne personnel had initially defined a purge amount of two liters as being typical for their applications, and thus, two liters was used for some of the design calculations to follow. Also relevant was the commercially available tank sizes for CO$_2$ use. A 10 lb tank seemed to be a fairly common size and, thus was selected partially based on availability. Preliminary predictive calculations verified that this size was compatible with the secondary fluid tank and appropriate run times.

The entirety of the predictive calculations was based on the following problem conditions in Table I. Note that these conditions were established for a saturation temperature of 32°F. The reason for using this saturation temperature was that it could be easily produced and maintained with an ice bath.
### Table I: Problem Conditions

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{\text{secondary fluid tank}}$</td>
<td>0.528</td>
<td>gallons</td>
</tr>
<tr>
<td><strong>Capacity</strong> $V_{\text{working fluid tank}}$</td>
<td>9.6</td>
<td>lb</td>
</tr>
<tr>
<td>$V_{\text{working fluid tank}}$</td>
<td>265.6</td>
<td>in$^3$</td>
</tr>
<tr>
<td>$h_{\text{working fluid tank}}$</td>
<td>14</td>
<td>in</td>
</tr>
<tr>
<td>$D_{\text{working fluid tank}}$</td>
<td>5.25</td>
<td>in</td>
</tr>
<tr>
<td>$\rho_{\text{secondary fluid}}$</td>
<td>1.94</td>
<td>slug/Ft$^3$</td>
</tr>
<tr>
<td>$P_{\text{sat working fluid}}$</td>
<td>505.56</td>
<td>psia</td>
</tr>
<tr>
<td>$\rho_{\text{liquid working fluid}}$</td>
<td>1.80</td>
<td>slug/Ft$^3$</td>
</tr>
<tr>
<td>$m_{\text{secondary fluid}}$</td>
<td>0.137</td>
<td>slugs</td>
</tr>
<tr>
<td>$m_{\text{working fluid}}$</td>
<td>0.316</td>
<td>slugs</td>
</tr>
<tr>
<td>$g_{CSUN}$</td>
<td>32.11</td>
<td>ft/sec$^2$</td>
</tr>
</tbody>
</table>

#### 1.5. Pressurized Vapor Delivery System Concept (PreVapS)

The name of this system was aptly coined as a Pressurized Vapor Delivery System or PreVapS for short. There were basically three phases of the physical system that set the tone of the type of process occurring in the system at any one given location. A schematic diagram is given below to show the order of the three phases.

**Figure I: Diagram of System Process**

The energy addition phase of the process provides the energy required to maintain the working fluid’s temperature at (or near) 32°F while the vapor is being produced from
the liquid. Once the working fluid is vaporized, it enters the Transport Phase which moves the vaporized working fluid to the secondary fluid tank to initiate the purge phase. The purge phase takes place in the secondary tank, and forces the water through an orifice plate and a flow meter, before it is collected in a catch tank.

The physical layout of the system apparatus can be examined on the next few pages in Figures II to V; the associated parts list can be examined in Table A-1. Figure II, III, IV, and V portray the system apparatus in slightly different forms for clarity. The table of parts includes the manufacturer of each component. Note that there is a removable secondary working fluid tank which is used to refill the system with CO₂. This tank can be quickly removed and refilled at a local CO₂ vendor when the primary working fluid tank is depleted.
Figure II: System Apparatus - Simplified
Figure III: System Apparatus Broken Down into Sections
Figure IV: Section View of the System Apparatus
Figure V: Detailed System Apparatus

Energy Addition Phase

Transport Phase

Purge Phase
Figure II includes information on the measurement points for pressure, temperature, and flow rate. As can be seen from Figure III, the system apparatus was broken down into three physical sections; the primary section, secondary section, and the process section. Figure III shows that the primary and secondary sections were isolated from each other by the separation valve. The entirety of the system was connected to a pressure relief valve designed to alleviate high pressure scenarios that could potentially arise in the system. More details will be provided below regarding the other key components of the system.

Figure IV shows the system broken down into sections outside of the three phases shown in Figure I. The purpose of this diagram is to clearly show how parts of the system were labeled.

In Figure V, the items labeled from one to thirteen were known as the primary working fluid section or primary section. Items one and two were classified as part of the energy addition phase. Note that item fourteen was specifically the separation section as defined by Figure III.

The items labeled from twenty-one to sixty-six comprised what was known as the process section. It was within this section that both the transport and the purge phase occurred. Specifically, the transport phase was associated with items twenty-one through thirty-six while the purge phase focused on the remainder. It was within the process section where most of the pressure losses due to friction in the flow of the working fluid occurred. This was because a majority of the fittings existed in this section which meant greater “minor” losses, which proved to be greater than losses along the tubing length.
The remainder of the items, sixty-seven through eighty-four, were part of the secondary working fluid section or the secondary section. As noted earlier, this section was designed to provide a safe and convenient way to refill the primary working fluid section after each test run.

A prime concern of this study was safety for students and faculty handling equipment under high pressures. The secondary section served to isolate the primary section from the user during the refill process, which was the most likely time for an occurrence of a safety issue. In accordance with safety design procedures, it was determined that the highest pressure to be attained in the system at any one time was approximately 1,000 psig. This pressure was only attainable if the working fluid’s temperature rose to approximately 84°F. As that was a real possibility, the system plumbing and parts were rated to a minimum pressure of 2,400 psig, providing a factor of safety of 2.4. In addition to system components being rated to a high pressure, an automatic pressure relief valve was installed on the system to release the high pressure working fluid in the event that the pressure rose above 1,000 psig.

One will notice that the secondary section did not have a transducer for pressure measurement, but rather an analog pressure gauge in addition to a thermocouple. The reason for using an analog pressure gauge rather than a pressure transducer was due to the fact that the user needed to be able to examine the pressure of the system during refilling. The refill process was based on the user’s judgment and was not automated. In actuality, the primary working fluid tank was held constant at 32°F and 500 psig, and the refill process was controlled to be slow enough so that the temperature and pressure in the secondary working fluid tank did not fall too low in order to maintain a sufficient
pressure difference between the tanks. Examining each of the significant components
presented in Table A-I and Figure V above will prove to be essential in understanding not
only the system as a whole, but the process itself.
Chapter 2: Theory

2.1. Working Fluid Research

The performance of this process depended heavily on the properties of the working fluid. A great deal of work regarding the determination of potential working fluids was completed prior to this particular study based on operating parameters specified by Pratt & Whitney, Rocketdyne. Although the working fluid (CO\textsubscript{2}) that was selected and tested in this study was not one of the fluids that were determined in the original study, the physical properties that make this working fluid desirable were similar among all eligible fluids. The original working fluid research was conducted by a senior mechanical engineering student who produced very detailed results on thermodynamic properties of potential working fluids and also on models for property prediction\textsuperscript{[23]}.

The working fluid research produced some guidelines for applicable fluids that could be used to drive this process. However, some of these fluids were not practical for these experiments, primarily for safety reasons.

The following list provides guidelines that shaped how the working fluid was selected. The working fluid had to\textsuperscript{[23]}:

- Be easily available
- Be inexpensive
- Be safe to handle
- Be nontoxic and nonflammable
- Have a large density ratio (saturated vapor / saturated liquid)
- Have a desirable saturation pressure near room temperature.
As noted in the previous chapter, the working fluid chosen was carbon dioxide, since it met all of the above criteria. Density ratio is defined as the ratio of saturated vapor density to saturated liquid density. A desirable density ratio was considered to be greater than approximately six. A higher density ratio means that a greater volume can be purged from a fixed liquid volume. The value of density ratio is related to how much lower the operating temperature is below the critical temperature. A greater difference would tend to increase the volume ratio, but it also increases the enthalpy of vaporization \( (h_{fg}) \). A higher enthalpy of vaporization means that more energy is required to maintain the working fluid at the desired operating temperature and pressure. Thus there tends to be an optimum working fluid critical temperature for any specific operating temperature. State diagrams illustrating these relationships are shown in the next section.

A key property of carbon dioxide was that its saturation temperature at the desired operating pressure of 500 psig was 32°F. Rather than using an expensive heating and temperature control system, the primary working fluid tank could be maintained at the desired temperature by submerging it in an ice water bath. Ice water was both safe and easy to handle and no matter the energy requirements of the working fluid, the system could be designed to provide enough ice water to maintain a 32°F temperature throughout the process.

2.2. The First Law of Thermodynamics

The fundamental concept involved in this process was transforming thermal energy into mechanical energy. The First Law of Thermodynamics quantifies how energy is manipulated within a system. The system process was modeled as an open system (see Section 2.3.5) and depending on how a region of interest was established
certain parameters in the First Law are known. Now, the First Law of Thermodynamics for a closed system is given as\textsuperscript{[6]}:

\[ T \text{h}
\]

\[ C \text{h}
\]

\[ a \text{i}
\]

\[ n \text{nt}
\]

\[ e \text{nl}
\]

\[ a \text{r}
\]

\[ e \text{n}
\]

\[ y
\]

\[ =\]

\[ H \text{e}
\]

\[ a \text{t}
\]

\[ e \text{d}
\]

\[ d\text{d}
\]

\[ t\text{o}
\]

\[ t\text{h}
\]

\[ e\text{m}
\]

\[ s\text{y}
\]

\[ s\text{t}
\]

\[ e\text{m}
\]

\[ y
\]

\[ n\text{w}
\]

\[ o\text{r}
\]

\[ d\text{U} = \delta Q - \delta W\]

Note that heat and work are counted when they cross the system boundary which leads to the descriptions used in the above equation. It should be noted that the heat and work terms are associated with being physically quantifiable forms of energy crossing the system boundary where the internal energy, noted as \( U \), accounts for the energy change of the system itself. For a quasi-static process work is equal to the following\textsuperscript{[12]}:

\[ -\delta W = -PdV\]

where \( P \) was defined as pressure and \( dV \) was defined as the change in volume.

Work is related to the pressure at the moving boundary of an arbitrary system. From a closed system point of view, this process does work by purging the secondary fluid. In this process, heat added to the system will produce a change in phase which will produce a large change in volume. Consequently, when considering the desirable properties of a working fluid, the thermodynamic properties related to the saturation curves are fundamentally important. By examining a pressure-specific volume curve, or \( P - v \) curve, for a fluid it can be shown that the specific volume of a substance will increase as the substance changes phases from liquid to vapor at a constant temperature and pressure. By examining Figure VI, one can see the changes in specific volume and
pressure among substance phases. (This figure is for water but the trends are similar for all common fluids).

![Diagram of the pressure-volume (P-v) diagram for water.](image)

**Figure VI: $P - v$ Diagram for Water**[41]

Figure VI shows the line that forms what is called a dome or pressure dome which defines the regions where different phases exist. For example, underneath the dome, one will have both liquid and vapor present; to the left of the curve, the substance will be completely liquid and to the right, completely vapor. For a “wider” dome, the volume difference which occurs during a vaporization process will be greater. Every known fluid exhibits these traits; however, the working fluid had to be able to establish a proper volume change while using a minimum amount of energy.

As noted earlier, the enthalpy difference between the saturated vapor and liquid states is an important property, and will vary as the temperature varies from the critical point. Figure VII shows a typical pressure-enthalpy diagram of a fluid. This curve
shows a similar pattern to that of the $P - v$ diagram, and illustrates how the enthalpy of vaporization varies with temperature.

**Figure VII: $P - h$ Diagram for a Refrigerant**\(^{[40]}\)

As can be seen in Figure VII, the energy content of the fluid increases as its phase changes from liquid to vapor. Note the enthalpy is a measure of the energy content of the substance and is related to the internal energy of the substance as well as the pressure and specific volume of the substance. This relationship is shown below and was recognized by Professor Richard Mollier\(^{[12]}\) to be of great importance in energy transformation in flow processes.

$$h = u + Pv \text{ or } H = U + PV$$

Here $h$ is defined as specific enthalpy and $u$ is defined as specific internal energy. Both have units of $kJ/kg$ or $Btu/lb_m$. Specific volume ($v$) is in units of $m^3/kg$ or $ft^3/lb_m$. The
capitalized parameters are defined as the total enthalpy, internal energy, and volume; they are the same units as listed for the first equation only not on a per mass unit basis.

Carbon dioxide proved to be a viable working fluid due to its desirable thermodynamic properties, consistent with the description of property trends in the previous paragraphs. Table II and Figure VIII & IX below show the saturation properties of carbon dioxide relevant to this process.

Table II: Saturation Properties of the Working Fluid

<table>
<thead>
<tr>
<th>$T$ (°F)</th>
<th>$v_f \left( \frac{ft^3}{lb_m} \right)$</th>
<th>$v_g \left( \frac{ft^3}{lb_m} \right)$</th>
<th>$\frac{v_g}{v_f}$</th>
<th>$P_{sat}$ (psia)</th>
</tr>
</thead>
<tbody>
<tr>
<td>32</td>
<td>0.0172</td>
<td>0.1643</td>
<td>9.5</td>
<td>506</td>
</tr>
<tr>
<td>35</td>
<td>0.0174</td>
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Figure VIII: Saturation Pressure of Working Fluid vs. Specific Volume

Figure IX: Saturation Temperature of Working Fluid vs. Specific Volume
The properties of carbon dioxide were determined from two sources. The first source was a thermophysical property calculator add-in for Microsoft Excel developed by three individuals. Dr. Larry Caretto, Dr. David McDaniel, and the late Dr. Tom Mincer developed VBA based software to work in conjunction with Microsoft Excel 2003 to provide easily attainable thermodynamic properties of specific fluids. Since both Dr. Mincer and Dr. Caretto have ties to the College of Engineering and Computer Science at CSUN, this particular resource was readily available. The second source for CO$_2$ properties came from a piece of literature translated from Russian to English by D. S. Grant under the directive of the United Kingdom Atomic Energy Authority in 1968. Written and compiled by M. P. Vukalovich and V. V. Altunin, *Thermophysical Properties of Carbon Dioxide* provided an accumulation of data and equations regarding the thermodynamic properties of carbon dioxide.

2.3. **Predictive Calculations**

Once a working fluid was established and the properties of the working fluid were known, it was imperative to develop some preliminary calculations that sought to give some foresight into how the working fluid would react under the conditions present in the system apparatus. The starting point was to use Bernoulli’s equation to choose orifice sizes that would produce flow rates in a desired range. Then, a first law of thermodynamics analysis of the vaporization process in the working fluid tank was used to estimate the necessary heat transfer rate to produce a steady vapor flow. Heat transfer correlations for pool boiling heat transfer were used to predict the temperature drop in the tank to produce the necessary heat flux. And finally, that temperature drop between the
working fluid tank surface and the working fluid will be used to estimate an expected associated drop in pressure of the working fluid.

2.3.1. **Assumptions**

The first step in determining how to predict the overall system performance was to establish some assumptions regarding the entire process. These assumptions include the parameters established in the problem statement as well as any additional assumptions noted here. First and foremost, the system process was designed to be occur at 32°F. This meant that the working fluid and working fluid tank were homogeneously at 32°F at the start of the process and the temperature control fluid, or ice bath, supplied 32°F conditions throughout the process. In addition, the volumes of the working fluid tank and the secondary fluid tank were assumed fixed. Also, the secondary fluid tank was assumed to have a volume approximately ten times greater than that of the working fluid tank due to the fact that the working fluid would expand to near ten times its original volume. In retrospect, the volume of the secondary fluid tank was irrelevant so long as the system was designed to have a working fluid tank large enough to perform the desired work on the total mass of secondary fluid.

In conjunction with the working fluid remaining at 32°F, it was assumed to be at saturation conditions at this temperature. This was a reasonable assumption since the working fluid was allowed to reach equilibrium conditions at this temperature before beginning the process and ultimately this was verified by actual pressure measurements.
The gravitational acceleration at the location of the conducted tests was presumed to be \(32.11 \frac{Ft}{sec^2}\) or \(9.787 \frac{m}{sec^2}\). This was the appropriate gravitational acceleration determined for the California State University, Northridge campus[6].

It was assumed that the water flow from the secondary tank through the orifice plate had no friction loss, for the purpose of using Bernoulli’s equation to calculate the orifice plate sizes. Flow of the working fluid was assumed to be incompressible for the purpose of estimating tubing pressure losses.

Also, the working fluid and secondary fluid were assumed to not interact in any way and thus the volumetric flow rates would necessarily be equal. This meant that if the secondary fluid flow rate was measured, then the flow rate of the working fluid was known as well. If one knew flow rate, then run times for a particular purge volume could be estimated.

2.3.2. Flow Calculations

Using these assumptions, a set of calculations could be performed that would provide some basic insight into the system characteristics. A Bernoulli analysis of the secondary fluid tank established orifice sizes to produce flow rates that would establish reasonable run times given the fluid volumes involved. This analysis required the choice of two points within the secondary fluid tank to make the required calculations. The figure below shows the two points and the known conditions. Note that in the physical construction of the system, the output of the secondary fluid tank is not directly to atmosphere, but since downstream of the output, the fluid was expelled to atmosphere at
the catch tank, the pressure at the outlet of the secondary fluid tank was assumed to be sufficiently close to atmospheric pressure.

![Diagram of the secondary fluid tank and its modeled conditions](image)

**Figure X: Secondary Fluid Tank as Modeled for Calculations**

It should be noted how the pressures and velocities at these two points were justifiably assumed to be what is shown in the figure above. Point (2) was assumed to represent a free jet, which was the idea of pushing fluid through an orifice that exits to atmospheric conditions. Since the fluid was exiting to atmospheric conditions it can be
assumed that the pressure at this point was equal to atmospheric pressure or 0 psig. The velocity at point (2) was the velocity that was to be calculated; therefore, it was denoted as ‘x’ in Figure X.

Point (1) was defined as the point at the interface between the working fluid (carbon dioxide) and the secondary fluid (water). The velocity of the secondary fluid here could be assumed to be negligible because the interface area was large enough that any change in the fluid level with respect to time is very small. An example of this would be a large tank of fluid that is draining through a small orifice; the change in the amount of fluid leaving the tank, or the fluid surface level, would be so small that it can be neglected\[34]. This is classified as a quasi-steady process; it is the idea that the fluid surface level is changing so slowly that it can be assumed as a negligible change for instantaneous calculations.

Since the thermophysical properties of the working fluid were known for the temperatures and pressures expected to be seen in the system, it was a logical assumption to say that the pressure at the secondary fluid interface was the pressure of the working fluid at the start of the process. To give a general preview of the behavior of the process, this assumption was valid.

2.3.3. Bernoulli’s Equation

Bernoulli’s Equation represents the balancing of energy in a flow system. In fluid systems, Bernoulli’s Equation balances energy at two points along a streamline. Bernoulli’s Equation equates potential energy, kinetic energy, and flow energy at two points to determine a specific unknown flow parameter. The potential energy is the
energy the fluid possesses due to its elevation, the kinetic energy is the energy the fluid possesses due to the fact it is flowing, and flow energy is the energy a fluid possesses due to the pressure acting on it. Bernoulli’s equation is shown below[^19].

\[ E_{Total} = p_1 V_1 + mgz_1 + \frac{1}{2} mV_1^2 = p_2 V_2 + mgz_2 + \frac{1}{2} mV_2^2 \]

where \( p \) is defined as pressure, \( V \) as the volume of the fluid, \( m \) as mass, \( g \) as gravitational acceleration, \( z \) as elevation, and \( V \) as velocity.

Generally the following form of the Bernoulli Equation is used in engineering applications[^34].

\[ \frac{E_{Total}}{m} = \frac{p_1}{\rho_1} + gz_1 + \frac{1}{2} V_1^2 = \frac{p_2}{\rho_2} + gz_2 + \frac{1}{2} V_2^2 \]  

(1)

There are other ways to present Bernoulli’s Equation; one of the most common forms is the head form where all of the above terms are represented as energy heads in units of length rather than energy per unit mass as is the case for Equation (1).

### 2.3.4. Use of Bernoulli’s Equation

The orifice plate was ideal as a flow control device since it could be easily removed from the system and replaced with a different size orifice. The difficulty with high pressure systems is the fact that generally components are difficult to take apart or remove since the components are constructed to withstand high pressures. The orifice plate offered easy access to the system to change the outlet diameter. The orifice plate allows a known diameter to set the system flow rate which can be estimated via the Bernoulli Equation.
The equation for volumetric flow rate through a circular cross section is given as\[^{[34]}\]:

\[
Q = \dot{V} = AV = \left(\frac{\pi D^2}{4}\right)V
\]

where \(A\) is defined as the area, \(D\) is the diameter, \(\dot{V}\) was volumetric flow rate, and \(V\) is the velocity.

Starting at Equation (1), the downstream velocity can be found as:

\[
\frac{p_1}{\rho_1} + gz_1 + \frac{1}{2}V_1^2 = \frac{p_2}{\rho_2} + gz_2 + \frac{1}{2}V_2^2
\]

\[
V_2^2 = 2\left[\left(\frac{p_1}{\rho_1} - \frac{p_2}{\rho_2}\right) + g(z_1 - z_2) + \frac{1}{2}V_1^2\right]
\]

\[
V_2 = \sqrt{2\left[\left(\frac{p_1}{\rho_1} - \frac{p_2}{\rho_2}\right) + g(z_1 - z_2) + \frac{1}{2}V_1^2\right]} \quad (2)
\]

Also, if it is noted that the elevation change between points (1) and (2) was negligible then the following is produced:

\[
V_2 = \sqrt{\frac{1}{\rho_2} (p_1 - p_2)} \quad (3)
\]

By observing the equations above, it can be easily seen that if it was desirable to increase the flow rate of the secondary fluid exiting the tank, then simply increase the diameter of the exit area and vice versa. The calculations to determine the flow parameters of the system were completed for a range of outlet diameters spanning from \(\frac{1}{32}\)" to \(\frac{1}{4}\)" where zero discharge pressure was still assumed.
Also, calculations were completed that included the elevation change and the effect was indeed negligible.

Once the outlet velocity was obtained, the flow rate of the secondary fluid, as well as the time required to exhaust the secondary fluid could be determined. Returning to the flow rate equation shown above the outlet flow rate could be determined as:

\[
Q_{SF} = \dot{V}_{SF} = A_{SF} V_{SF} = \left( \frac{\pi D_{SFT0}^2}{4} \right) V_{SF}
\]  

(4)

where \(D_{SFT0}\) is defined as the outlet diameter of the secondary fluid tank or the orifice plate diameter. \(A_{SF}\) is the cross sectional area of the stream of secondary fluid exiting the tank and \(V_{SF}\) is the velocity of the secondary fluid exiting the tank.

Since the volume of the secondary fluid to exhaust was known and was defined in the problem statement as 0.528 gallons, or two liters, the time to exhaust the fluid could be determined as:

\[
t_{exhaust, SF} = \frac{V_{SF}}{Q_{SF}}
\]  

(5)

where \(V_{SF}\) is defined as the volume of the secondary fluid expelled.

If Equation (3) is substituted into Equation (4) and Equations (3) and (4) substituted into Equation (5) it can be shown that:

\[
Q_{SF} = \dot{V}_{SF} = A_{SF} V_{SF} = \left( \frac{\pi D_{SFT0}^2}{4} \right) \sqrt{\frac{1}{\rho \left( p_1 - p_2 \right)}}
\]  

(6)

\[
t_{exhaust, SF} = \frac{V_{SF}}{\left( \frac{\pi D_{SFT0}^2}{4} \right) \sqrt{\frac{1}{\rho \left( p_1 - p_2 \right)}}}
\]  

(7)
In addition since the flow rate was known as well as the volume of the working fluid, the time to exhaust the working fluid could then be determined as the following:

\[ t_{\text{exhaust}} = \frac{V_{WF}}{Q_{WF}} = \frac{V_{WF}}{Q_{SF} \left( \frac{\pi D^2_{SF} \rho}{4} \sqrt{\frac{L}{p}} \right)} \]

where \( Q_{WF} \) is defined as the working fluid volumetric flow rate and \( V_{WF} \) is the volume of the working fluid expelled.

Since the system took a high pressure fluid and pushed another fluid through an orifice, a force was created that acted upon the system oppositely to the direction of fluid flow. The thrust created determined the type of support system that would be potentially required to restrain the secondary fluid tank. In order to minimize the need for physical restraints, a fluid damper was created to slow the fluid flow down after the orifice plate; however, it still was important to determine the maximum force vector produced to ensure that standard restraints on the tank were sufficient.

From the conservation of momentum equation for fluid applications in the x-direction where travelling left to right was considered positive, the restraining force required to secure the output of the orifice flow could be calculated. If the control volume is taken as being around the orifice plate then a simplified form of the momentum equation can be shown as

\[ \sum_{\text{openings}} (\pm \dot{m})(\pm V_x) = \sum_{\text{openings}} (\pm PA)_x \pm R_x \pm W_x \]

where \( \dot{m} \) is the mass flow rate in and out of the control volume, \( V_x \) is defined the velocity in the x-direction, \( P \) is the pressure, \( A \) is the cross sectional area of control volume, \( R_x \) is
the restraining force required, and \( W_x \) is the body forces which was assumed negligible in this application.

If it is assumed that a negative sign represents a flow into the control volume and vice versa, as well as two openings (the upstream side of the orifice plate, and the downstream side of the orifice plate) the above momentum equation can be simplified as:

\[
-\dot{m}V_1 + \dot{m}V_2 = P_1A_1 - P_2A_2 + R_x
\]

\[
R_x = \dot{m}(V_2 - V_1) + P_2A_2 - P_1A_1 \tag{10}
\]

Assuming that the “1” above corresponds to the upstream side of orifice plate and “2” refers to the downstream side of the orifice, then Equation (10) can be solved and the restraining force calculated. Note that the restraining force must act opposite the direction of flow in order to be a success. Refer to Table V at the end of this chapter to see the results of the thrust calculations. Since the values were moderate for the flow rates tested, no special restraints were required on the tank.

Care needed to be taken with flow restrictions in the plumbing which transported the working fluid between the tanks. Orifice sizes in valves, for example, had to be large enough to prevent choking, which will occur if the local flow reaches acoustic velocity. If choking occurs then the overall flow rate will be limited and large pressure losses will result. Once the desired flow rates were known, it was important to determine the acoustic velocity of the working fluid so minimum orifice sizes could be determined. Generally the goal was to make sure that the working fluid did not flow over or close to a Mach number equal to 0.3 due to the fact that, in general, incompressibility is valid until this point and pressure losses would not be significant.
The calculations regarding these flow conditions were made on a worst case scenario basis. In order to determine the acoustic velocity, the following relationship was used\[^{[32]}\]:

\[
V_a = \sqrt{\gamma_{WF} R_{WF} T_{WF}}
\]

(11)

where \(V_a\) is defined as the acoustic velocity of the working fluid, \(\gamma_{WF}\) is the adiabatic index of the working fluid or more commonly known as the specific heat ratio, \(R_{WF}\) is the gas constant of the working fluid, and \(T_{WF}\) is the absolute temperature.

The adiabatic index for the working fluid was on the order of one with a value of 1.3 at 32°F. Since this was known, a conservative value for acoustic velocity could be calculated using the lowest temperature the working fluid was predicted to potentially achieve and the adiabatic index at 32°F. Using the previous arguments, it could be determined if the working fluid velocity was below this conservative acoustic velocity, then the flow would not choke.

The lowest possible temperature that could be attained within the system results from the vaporization process of the working fluid. Based on the predictive calculations, during vaporization, the temperature of the working fluid decreases and based on the flow requirement placed on the system, the lowest possible temperature calculated that the working fluid could potentially achieve was approximately \(-4^\circ F\) or \(456^\circ R\). With this known and the conditions noted above known, the acoustic velocity of the system could be conservatively calculated as:

\[
V_a = \sqrt{\gamma_{WF@32F} R_{WF} T_{WF}}
\]
\[ V_a = 810 \frac{ft}{sec} = 552 \text{ mph} \]

The acoustic velocity converted into SI Units is:

\[ V_a = 247 \frac{m}{sec} \]

Next, the cross sectional area was found for the various flow rates of the system which would produce a Mach number equal to 0.3. Sizing the calculations for this Mach number proved valuable since it would ultimately prove incompressibility and subsonic flow conditions. The following relation was used to determine the minimum cross sectional area \[ ^{34} \]

\[ A_{c\text{min}} = \frac{Q_{WF}}{V_{max}} = \frac{Q_{WF}}{V_a Ma} \]  \hspace{1cm} (12)

where \( V_a \) is the acoustic velocity, \( Ma \) is defined as the Mach number, \( V_{max} \) is the maximum velocity in the system, and \( A_{c\text{min}} \) is the minimum cross sectional area.

The minimum diameter, or minimum port diameter, that components in the system can have is calculated from Equation (13) as:

\[ D_{\text{port\text{min}}} = \sqrt{\frac{A_{c\text{min}}(4)}{\pi}} \]  \hspace{1cm} (13)

As long as all other cross sectional diameters in the system were above the minimum port diameter, the presumption that the flow would not choke was valid. It was verified that the minimum port diameter in all the valves used was larger than the smallest cross sectional diameter found from Equation (13).
2.3.5. **Heat Transfer Calculations**

The next step was to determine the heat transfer rate that the working fluid would require in order to successfully vaporize and maintain steady pressure. At the start of the process the working fluid was assumed to exist as a liquid at saturated conditions. In order for the working fluid to vaporize, heat energy must be transferred from the surroundings to the working fluid. Heat can only be transferred if a temperature differential is established between the working fluid tank surface and the working fluid. Since a temperature difference is created between the two mediums, an associated pressure drop occurs within the saturated working fluid. The goal of calculating the heat transfer in the system required by the vaporization of the working fluid was to determine the required temperature change of the working fluid to drive the required heat transfer rate. Then the pressure drop in the primary working fluid tank can also be estimated. The pressure of the working fluid is a key parameter for the system process since a steady vapor delivery pressure is desired. Since the vaporization process causes a reduction in system pressure, it was important to have a relative understanding of exactly how much the pressure would drop, as a function of system flow rate.

Determining the heat transfer and resulting pressure drop was one of the most complicated and dynamic calculations completed in this study. The study of convection heat transfer as a whole is largely an empirically based science and boiling heat transfer is no exception. Heat transfer coefficients can have significant uncertainty, especially in boiling conditions. This idea of uncertainty was best exemplified by a quote from Warren M. Rohsenow, a leader in the Boiling Heat Transfer field, which states the
premise of trying to determine just one parameter used in pool boiling heat transfer. The quote is taken from *Handbook of Heat Transfer* in which Rohsenow states\[^{40}\]:

“The pool-boiling case is unusual in that correlations and predictions are commonly based on mechanistic models...One may conclude, therefore, that prediction of critical heat flux in pool boiling is still surrounded by mechanistic uncertainties and that recourse must be had to the reasonably well-established correlations (notwithstanding their deficiencies as listed previously).”

The above quote shows that even for the most developed correlations they are indeed just predictions that have the potential to be inaccurate.

From initial estimates of the heat transfer rate, it became clear that the expected mode of heat transfer in the primary working fluid tank would be Nucleate Pool Boiling. The predictive calculations also assume the vaporization process will be under a constant pressure even though it is known this will not be the case.

The first step in trying to determine heat transfer characteristics of this system was to understand the different regimes of boiling heat transfer Figure XI shows a curve known as the Boiling Heat Transfer Curve for water. The trends in the boiling curve are similar for most fluids. The trick was to be able to either discover a boiling heat transfer curve for the working fluid (CO\(_2\)) or be able to generate the curve from correlations.
Figure XI: Boiling Curve for Water[29]

Figure XI shows that there are different regimes of boiling that can occur when vaporizing a fluid; each regime consists of its own characteristics and relationships. In brief, the regime characteristics should be outlined to provide a better understanding of the choices made in this study. Note that nucleate pool boiling was the prime concern but all the regimes played some role.

The first regime, known as natural (or free) convection, is characterized by the fact that the mode of heat transfer is governed by fluid motion driven by buoyancy. In
this regime the fluid is only slightly superheated with no indication of vapor bubbles forming. The temperature range (i.e. the difference between surface temperature and saturation temperature) covering each regime is different for each fluid; however, for water the natural convection range is 1°C to 5°C (2°F to 9°F).

The regime following natural convection is known as nucleate pool boiling and was the regime that was considered most likely for the working fluid heat transfer process. Nucleate pool boiling is characterized by the fact that vapor bubbles begin to form within the fluid at what are called nucleation sites. These initial nucleation sites are preferential sites on the heater surface that, because of their geometry and rate of heat supplied to the site, begin to form vapor bubbles first\[^5\]. Consequently, when the heat flux increases across the heater surface, so do the number of nucleation sites. One thing that is important to note regarding bubble theory, however, is that the way in which bubbles form and how bubbles leave the heater surface does affect the heat transfer rate.

There are essentially two sub-regimes within the nucleate boiling regime that are characterized by the rate at which formed bubbles are dissipated. When the bubble production rate increases from the natural convection regime, the bubbles are considered isolated and dissipate within the liquid shortly after the bubbles leave the heater surface. Note that when a bubble leaves the heater surface the space vacated is filled with liquid from the surrounding area. The movement of liquid into the available space causes stirring and agitation within the fluid which primarily contributes to the increase in heat flux within the nucleate boiling regime\[^8\]. Note that the range characterized by isolated bubbles (between A and B) is the most desirable one for controlling this process, since the heat rate changes by a factor of 10 with a relatively small temperature difference.
Still contained in the nucleate boiling regime is the next sub regime which is characterized by an increased number of nucleation sites and increased rate of bubble production. The rate of bubble production is so great at any one nucleation site, that continuous columns of vapor form that carry vapor bubbles to the free surface. As with the first sub-regime, this bubble production causes stirring and agitation of the fluid which in turn continues to increase the heat flux across the heater surface. The key thing to note here is that the liquid can still fill the vacant spaces created by bubble production thereby in essence still cooling the heater surface by evaporation\[29\].

The nucleate boiling regime continues to a maximum point known as the critical heat flux or burnout point. This is the point at which the nucleate boiling regime ends and transition boiling regime begins. The critical heat flux is a crucial point in the boiling process since it essentially represents the point at which the maximum heat flux can be attained without melting the heater material; hence, why it is called the burnout point. The boiling curve presents an interesting possibility of behaviors at this point which can either follow the boiling curve into the transition regime or go as the following arrowed lines in Figure XII below if under a constant heat flux constraint.
After the nucleate boiling regime, the process enters the transition regime which is denoted by a decrease in the heat flux caused by a blanketing effect on the heater surface. Essentially, the rate of bubble production increases so greatly, that the act of the liquid replacing the vacancies at the heater surface left by the vapor cannot occur as rapidly as in nucleate boiling and a layer of vapor film begins to form around the heater surface. In addition, since the amount of liquid coming into contact with the heater surface decreases the temperature difference of the system as a whole increases. The vapor film continues to form and the heat flux continues to decrease until the surface is completely covered in a vapor film at which point, the phenomenon of film boiling occurs\textsuperscript{[29]}.

The solution method was to use the First Law of Thermodynamics and then through boiling heat transfer methodologies, determine the temperature drop by using the
heat flux determined from the First Law. An iterative scheme could then be implemented to determine the correct temperature drop.

The advantage to being able to determine the heat flux from a thermodynamic perspective is that if one knows the heat flux beforehand, the temperature can be backed out of the boiling heat transfer relationships. If one derives the First Law of Thermodynamics for a transient system based on the system control volume, using the Uniform State Uniform Flow assumptions, the result is\[^7\] :

\[
Q_{CV} + \sum \left[ \dot{m}_i \left( h_i + \frac{v_i^2}{2g_c} + z_i \left( \frac{g}{g_c} \right) \right) \right] = \sum \left[ \dot{m}_e \left( h_e + \frac{v_e^2}{2g_c} + z_e \left( \frac{g}{g_c} \right) \right) \right] + \\
\left[ m_2 \left( u_2 + \frac{v_2^2}{2g_c} + z_2 \left( \frac{g}{g_c} \right) \right) - m_1 \left( u_1 + \frac{v_1^2}{2g_c} + z_1 \left( \frac{g}{g_c} \right) \right) \right] + W_{CV} \tag{14}
\]

where \( Q_{CV} \) is defined as the heat transferred into the control volume., \( \dot{m}_i \) is defined as inlet mass, \( h_i \) is inlet enthalpy, \( v_i \) is inlet velocity, \( z_i \) is inlet elevation, \( \dot{m}_e \) is outlet mass, \( h_e \) is outlet enthalpy, \( v_e \) is outlet velocity, \( z_e \) is outlet elevation, \( m_2 \) is final state mass, \( u_2 \) is final state internal energy, \( v_2 \) is final state velocity, \( z_2 \) is final state elevation, \( m_1 \) is initial state mass, \( u_1 \) is initial state internal energy, \( v_1 \) is initial state velocity, \( z_1 \) is initial state elevation, and \( W_{CV} \) is work energy into or out of the control volume.

Figure XIII shows the control volume being examined for equation (14); it contains the primary working fluid tank and is indicated by the dotted region, or region of interest (ROI).
Simplifying Equation (14) by noting that there is no inlet, one exit, no elevation change, minimal velocity change, and no work to get the following:

\[
Q_{CV} = m_e h_e + (m_2 u_2 - m_1 u_1)
\]  

Assuming that the tank is initially filled with saturated liquid then becomes a saturated vapor at the end of the process, the masses can be found by:

\[
m_1 = m_f = \frac{\nu_{WF_{Tank}}}{\nu_{f_{WF}}}
\]

where \(\nu_{f_{WF}}\) is the specific volume of the saturated liquid working fluid.

\[
m_2 = m_g = \frac{\nu_{WF_{Tank}}}{\nu_{g_{WF}}}
\]
where $V_{WF,Tank}$ is the volume of the working fluid tank and $v_{gWF}$ is the specific volume of the saturated vapor working fluid.

$$u_1 = u_{fWF}$$

where $u_{fWF}$ is defined as the internal energy of the saturated liquid working fluid.

$$u_2 = u_{gWF}$$

where $u_{gWF}$ is defined as the internal energy of the saturated vapor working fluid.

$$h_e = h_{gWF}$$

where $h_{gWF}$ is defined as the enthalpy of the saturated vapor working fluid that left the tank.

Also note that the mass that exited the primary working fluid tank was equal to the starting mass less the ending mass, as conservation of mass permits, or as:

$$m_e = m_1 - m_2 = \frac{V_{WF,Tank}}{v_{fWF}} - \frac{V_{WF,Tank}}{v_{gWF}} = V_{WF,Tank} \left( \frac{1}{v_{fWF}} - \frac{1}{v_{gWF}} \right)$$

Recall that the temperature at which these thermodynamic states are defined at is 32°F, or the temperature of the ice bath surrounding the tank. With all of the above noted Equation (15) becomes:

$$Q_{CV} = \left[ V_{WF,Tank} \left( \frac{1}{v_{fWF}} - \frac{1}{v_{gWF}} \right) \right] h_{gWF} + \left( \frac{V_{WF,Tank}}{v_{gWF}} u_{gWF} - \frac{V_{WF,Tank}}{v_{fWF}} u_{fWF} \right)$$

(16)
All of the parameters in Equation (16) can be determined from physical measurements, assumptions, and/or thermodynamic property tables; therefore, the overall heat into the control volume could be determined.

The heat value determined in Equation (17) was defined as the heat that entered the system from the beginning state to the end state. To obtain the heat transfer rate, the heat calculated in Equation (17) needs to be divided by the time to exhaust the working fluid per run as calculated in Equation (8):

\[
\dot{Q} = \frac{Q_{CV}}{t_{exhaustWF}}
\]

\[
\dot{Q} = \frac{\nu_{WFTank}(\frac{1}{v_fWF} - \frac{1}{v_gWF})}{} \frac{h_gWF}{\nu_{gWF}} \frac{\nu_{WFTankbf}}{} \frac{\nu_{WFTankbf}}{} (\frac{\pi D^2}{4}) \left( \frac{1}{2}(P_1 - P_2) \right)
\]

With the above said, the next step in these calculations was use heat transfer methods to compare heat fluxes. Recall, it was assumed that the boiling regime was nucleate pool boiling and this assumption was made so that the nucleate pool boiling relations could be used to back out the temperature difference between the working fluid tank wall and the working fluid saturation temperature. The best way to confirm this was to examine the value of critical heat flux (CHF). The critical heat flux is dependent on geometry and fluid properties\[4\]. Most of the calculations and experiments conducted for determining the CHF were for other fluids other than the working fluid and geometries other than the system apparatus. Much of the work regarding critical heat flux was conducted for wires, small diameter horizontal cylinders, and flat plates; some researchers developed discussions for vertical surfaces as well, however\[20\]. The
approach taken for this study was to compile the current research and tailor it to the application at hand. The working fluid tanks were cylinders and as such, for heat transfer purposes were treated as a combination of a flat horizontal plate (bottom), and a vertical flat plate (sidewalls).

The following steps were used to calculate the CHF for the system and determine exactly what kind of range would be considered valid for nucleate pool boiling. Since at least part of our system could be considered as a flat horizontal plate, the empirical relationship determined by Rohsenow for flat horizontal plates could be used. The relationship used is \(^{[29]}\):

\[
CHF = C h_{fg} \rho_v \left[ \frac{\sigma \rho_l (\rho_l - \rho_v)}{\rho_v} \right]^{\frac{1}{4}}
\]  

(18)

The constant \(C\) is determined by the geometry of the heating surface which is equal to 0.149 for flat horizontal plates. The density is defined as \(\rho\) and \(h_{fg}\) was defined as the enthalpy of vaporization of the working fluid. Finally, \(\sigma\) was defined as the surface tension.

Now, as mentioned previously the CHF separates the nucleate pool boiling regime from the transition boiling regime indicating the maximum heat flux that can be achieved before the system would exit the nucleate pool boiling range. By varying the difference between the temperature of the heater surface and saturation temperature of the working fluid, one could calculate the associated thermophysical properties and then the resulting CHF.
A second correlation that is available for estimating CHF is based on the work of Kandlikar. Equation (19) was obtained from the December 2001 volume of ASME’s Journal of Heat Transfer wherein, Satish G. Kandlikar, developed a model to predict the CHF that incorporated the effects of contact angle and orientation of the heating surface. Kandlikar’s relation was used above and is:\[^{[31]}\]

\[
CHF = h_f \rho_g \left(\frac{1 + \cos \beta}{16}\right) \left[\frac{2}{\pi} + \frac{\pi}{4} (1 + \cos \beta) \cos \phi \right]^{\frac{1}{2}} \left[\frac{\sigma g (\rho_l - \rho_g)}{\rho_g \rho_l \cos \phi}\right]^{\frac{1}{4}}
\] (19)

As compared to Equation (18), Equation (19) was similar in nature except for the cosine terms. \(\beta\) represents the angle which vapor bubbles come off the heater surface as depicted by Figure XIV. In general, Kandlikar suggests that a vapor bubble angle of 45° was a good assumption to make for most boiling processes; therefore, that angle was assumed for the system process. Additionally, \(\phi\) was defined as the angle of the boiling surface and was set at 90° to ensure the most conservative result. In the figure below, the other term can be neglected since they are not relevant to the study at hand. Note that this depiction is in fact for a vertical surface, and bubble deformation from gravity does not appear to be taken into effect.
Recall from above that the assumption was that the entirety of the primary working fluid tank was to be treated as a flat horizontal plate; however, the calculated results shown here take surface orientation into account. This flat plate assumption was made since prior to the discovery of the Kandlikar relationship, the relationship used to determine CHF was for a flat plate. Note that both sets of calculations were completed, and are shown here. Also recall, however, that CHF for vertical plates was noted to produce conservative results; therefore, using Kandlikar’s relationship was more appropriate for this study. The following table shows the CHF’s for a range of temperature changes, based on Kandlikar’s correlation.
### Table III: CHF Values for the Working Fluid

<table>
<thead>
<tr>
<th>Temperature Difference (°F)</th>
<th>CHF Kandlikar ($\text{BTU sec ft}^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>41</td>
<td>74.99</td>
</tr>
<tr>
<td>50</td>
<td>76.71</td>
</tr>
<tr>
<td>68</td>
<td>79.84</td>
</tr>
<tr>
<td>86</td>
<td>81.29</td>
</tr>
<tr>
<td>90.5</td>
<td>81.45</td>
</tr>
<tr>
<td>92.75</td>
<td>81.50</td>
</tr>
<tr>
<td>94.1</td>
<td>81.52</td>
</tr>
<tr>
<td>95</td>
<td>81.52</td>
</tr>
<tr>
<td>95.9</td>
<td>81.53</td>
</tr>
<tr>
<td>96.08</td>
<td>81.53</td>
</tr>
<tr>
<td>96.125</td>
<td>81.53</td>
</tr>
<tr>
<td>96.17</td>
<td>81.53</td>
</tr>
<tr>
<td>96.26</td>
<td>81.53</td>
</tr>
<tr>
<td>96.35</td>
<td>81.53</td>
</tr>
<tr>
<td>97.25</td>
<td>81.52</td>
</tr>
<tr>
<td>99.5</td>
<td>81.50</td>
</tr>
<tr>
<td>104</td>
<td>81.37</td>
</tr>
<tr>
<td>122</td>
<td>80.67</td>
</tr>
<tr>
<td>140</td>
<td>79.28</td>
</tr>
<tr>
<td>158</td>
<td>77.33</td>
</tr>
</tbody>
</table>

Note that the CHF can be seen to be approximately 81.53 $\text{BTU sec ft}^2$ or 921 $\text{KW m}^2$.

For comparison, CHF values using Rohsenow’s correlation are shown in Table IV. These values are about 108.3 $\text{BTU sec ft}^2$ or 1,224 $\text{KW m}^2$. 
Table IV: CHF Values for the Working Fluid

<table>
<thead>
<tr>
<th>Temperature Difference (°F)</th>
<th>CHF Rohsenow (BTU/sec ft²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>149</td>
<td>107.77</td>
</tr>
<tr>
<td>151.25</td>
<td>108.32</td>
</tr>
<tr>
<td>153.5</td>
<td>108.33</td>
</tr>
<tr>
<td>155.75</td>
<td>108.30</td>
</tr>
<tr>
<td>158</td>
<td>108.24</td>
</tr>
<tr>
<td>167</td>
<td>107.49</td>
</tr>
<tr>
<td>176</td>
<td>106.86</td>
</tr>
<tr>
<td>194</td>
<td>104.84</td>
</tr>
</tbody>
</table>

Now, since the CHF was known, the boiling regime can be assumed to be nucleate pool boiling for heat fluxes less than $81.53 \text{ BTU/sec ft}^2$ and the heat flux was then calculated and compared against this assumption. Based on the relationships for nucleate pool boiling, Equation (16) & (21), in order to determine the associated temperature change, one had to use an iterative scheme. The problem with this process was the fact that with each iteration, new thermophysical properties of the fluid had to be calculated since the relationships were dependent on said properties which were dependent on temperature.

To determine the heat flux into the control volume, or ROI, the area through which the heat must transfer must be known. The area of the control volume was simply the surface area of the primary working fluid tank. For the predictive calculations completed in this study, it was presumed that the area was the bottom of the working fluid tank as well as the tank walls. This assumes that the working fluid tank is initially full and doesn’t significantly drop during the run. The relationship between the heat flux and the heat transfer rate found from Equation (17) is\[^7\]:

48
\[ q'' = \frac{\dot{Q}_{CV}}{A_{CV}} \]  

(20)

where \( A_{CV} \) is defined as the area of the control volume based on the assumptions stated above.

The heat flux determined from Equation (20) was key to develop the predictions for the temperature change associated with the system process. Now, the relationship shown in Çengel that was developed by Rohsenow in 1952 for nucleate pool boiling is shown below and was the basis for comparison against the results produced by Equation (20). Rohsenow’s equation is\(^{[12]}\):

\[ q''_{nucleate} = \mu_l h_f g \left[ \frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[ \frac{c_p_l(T_s-T_{sat})}{C_s f h_f g \Pr_l} \right]^3 \]  

(21)

where \( q''_{nucleate} \left( \frac{BTU}{ft^2 \cdot sec} \right) \) is defined as the nucleate pool boiling heat flux, \( \mu_l \left( \frac{lbm}{ft \cdot s} \right) \) is the viscosity of the liquid working fluid, \( h_f g \left( \frac{BTU}{lbm} \right) \) is the enthalpy of vaporization, \( g \left( \frac{ft}{s^2} \right) \) is the gravitational acceleration, \( \rho_l \left( \frac{lbm}{ft^3} \right) \) is the density of the liquid working fluid, \( \rho_v \left( \frac{lbm}{ft^3} \right) \) is the vapor density of the working fluid, \( \sigma \left( \frac{lb}{ft} \right) \) is the surface tension of the liquid-vapor interface, \( C_p_l \left( \frac{BTU}{lbm \cdot R} \right) \) is the specific heat of the liquid working fluid, \( T_s \left( R \right) \) is the temperature of the heater surface, \( T_{sat} \left( R \right) \) is the saturation temperature of the liquid, \( C_s f \) is the heater surface to liquid combination constant, \( \Pr_l \) is the Prandtl number of the liquid, and \( n \) is an experimental constant that equaled one for this study.

Now, the iteration process stems from the fact that the temperature change was not known and since all of the thermodynamic properties were based on the temperature
change, the thermodynamic properties were not known. The thermodynamic properties as well as the temperature were treated as dynamic variables, therefore every time temperature changed the other properties had to as well. Since this was the case, iterations had to be made based on an initial guess, however, with each guess, the associated thermophysical properties had to be calculated again. A Microsoft Excel workbook was created to complete the iterations and necessary property calculations for each secondary fluid outlet diameter.

For clarity, the overall steps the workbook took to calculate the temperature changes will be discussed as follows. To determine the temperature change based on Equation (21), the varying temperature associated with a heat flux was calculated and compared to the heat flux determined from Equation (20) and when the difference between the heat fluxes differed by approximately $0.00001$, the iterations were stopped. It should be noted that Excel’s built in Goal Seek function was utilized heavily for these calculations. The equations of the workbook were created with interdependencies that allowed Goal Seek to change the temperature, calculate a heat flux, and converge towards a solution via comparative means.

It should be noted that since the properties were based on temperature, interpolative methods were used to calculate the exact properties. This meant that the proper temperature range and the final calculated temperature would fall in had to be known prior to beginning the calculations. Guesses were used for the property temperature ranges, and if the final temperature fell outside the initial temperature range, the ranges were updated, and the calculation redone. Although the program was important, any further details shall be omitted since the program was simply established...
to make repetitive calculations. Note that once an initial temperature drop was determined, a new heat energy term could be calculated and thus a new heat flux could be determined. With the new heat flux, the iteration scheme could be utilized once again to calculate a new temperature drop. Once the temperature change was calculated, then the associated pressure change, among other properties, of the working fluid could be calculated.

The pressure drop calculations were completed by using an interpolation scheme for the properties of the working fluid at saturation conditions. For example, if one calculated a 5\(^\circ\)F drop in temperature from 32\(^\circ\)F then since the saturation properties were known the pressure drop could be interpolated.

The temperature change and associated pressure drop were the parameters sought after in these calculations, thus the predictive calculations were complete at this point. Recall that the calculations were completed for a range of secondary fluid orifice diameters; therefore, the values of the parameters shown in the Table V were for the range of diameters.

Refer to Table V to see the results of the calculations and to Chapter 6: Results & Discussions to witness the differences in the predictive calculations and the actual results. Note that the heat fluxes found for the two largest diameters appear to exceed the value of CHF, which would invalidate the nucleate pool boiling assumption. The experimental results will support the fact that the CHF was not really reached in practice.
Table V: Results of Predictive Calculations

<table>
<thead>
<tr>
<th>Orifice Diameter (in)</th>
<th>$F_{Jet}$ (lbf)</th>
<th>$t_{exhaust \ WF}$ (sec)</th>
<th>$Q_{SF \ Out}$ (gpm)</th>
<th>$Q \left( \frac{BTU}{sec} \right)$</th>
<th>$q'' \left( \frac{BTU}{sec ft^2} \right)$</th>
<th>$\Delta T_{WF} \ (^\circ F)$</th>
<th>$\Delta P_{WF} \ (psig)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.03125</td>
<td>276</td>
<td>106.89</td>
<td>0.645</td>
<td>8.42</td>
<td>4.82</td>
<td>37.4</td>
<td>39.12</td>
</tr>
<tr>
<td>0.0625</td>
<td>274</td>
<td>26.72</td>
<td>2.581</td>
<td>33.67</td>
<td>19.29</td>
<td>40.7</td>
<td>61.52</td>
</tr>
<tr>
<td>0.125</td>
<td>265</td>
<td>6.68</td>
<td>10.325</td>
<td>134.66</td>
<td>77.16</td>
<td>46.0</td>
<td>96.71</td>
</tr>
<tr>
<td>0.1875</td>
<td>272</td>
<td>2.97</td>
<td>23.232</td>
<td>302.97</td>
<td>173.62</td>
<td>50.8</td>
<td>126.04</td>
</tr>
<tr>
<td>0.25</td>
<td>233</td>
<td>1.67</td>
<td>41.301</td>
<td>538.62</td>
<td>308.65</td>
<td>55.1</td>
<td>151.80</td>
</tr>
</tbody>
</table>

Note that the time required to exhaust the working fluid was what sized the system as a whole; all instrumentation and plumbing was sized to meet the above criteria.

All orifice sizes stated in Table V were tested except for the $\frac{1}{4}$" diameter orifice.

The calculations indicate the high heat fluxes required to create the necessary vapor for the secondary fluid purge process. This meant that the process would be limited by the heat transfer rate in the working fluid tank, and in practice steady conditions would probably not be realizable. While these calculations are approximate due to the assumptions made, they offer the trends that would be seen during testing.
Chapter 3: System Components

3.1. Measurement Equipment

The behavior of the process was quantified using specific measurement devices that recorded vital parameters such as temperature, pressure, and fluid flow. By measuring these characteristics at specific locations within the system, an accurate depiction of the process could be portrayed.

3.1.1. Omegadyne Type J Stainless Steel Thermocouples

Figure V shows that diagram numbers corresponding to the four independent type J thermocouples used in the system apparatus are: thirteen, eighteen, thirty-three, and seventy-nine. These thermocouples, otherwise known as TC #1, TC #2, TC #3, and TC #4 were placed at key locations in the system to monitor temperature of the working fluid and how it changes throughout the process. The first thermocouple measures the temperature at the exit of the working fluid tank and before the entrance of the 3-way manifold valve. The placement of TC #1 was for the sake of measuring any temperature variations that may have occurred from the working fluid travelling from the tank to the valve manifold. This thermocouple also monitored when the primary section of the system reached equilibrium conditions prior to system activation. Basically, TC #1 was able to tell when the initial part of the system reached an equilibrium temperature by comparing data pulled from the multilevel thermocouple in the primary tank.

TC #2 was positioned across the 3-way valve manifold and provided temperature readings of the working fluid as it travelled from the initial part of the system through the valve manifold. The main purpose of gathering temperature data at the exit of the valve manifold was to provide temperature of the working fluid before entering the process.
section of the system and account for any temperature variations. The process section was where the most distance was travelled by the working fluid and, therefore, the greatest potential for heat transfer between the working fluid and the surroundings. Next, TC #3 is located at the end of the process section before the secondary fluid tank. Thermocouple #3 was a key component in the system because it quantified the amount of heat transfer through the main section that occurred during the process.

TC #4 was placed in the secondary working fluid section of the system. The purpose of this fourth thermocouple was to monitor and log the temperature data of the working fluid during the refill process. The data collected here gave insight into the proper methodology of how to recharge the system with the working fluid.

All four of these thermocouples work in conjunction to clarify exactly what was occurring during the transition of the working fluid through the system. The thermocouples were selected based on what conditions needed to be met. These conditions include time response as well as compatibility with the multilevel thermocouple and the data acquisition system. The specifications of thermocouples #1 through #4 are outlined below in Table VI. Images of the thermocouples are shown in Figure XV.

<table>
<thead>
<tr>
<th>Table VI: Specifications of Thermocouples #1 - #4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple Type</td>
</tr>
<tr>
<td>Thermocouple Conductors</td>
</tr>
<tr>
<td>Thermocouple Sheath</td>
</tr>
<tr>
<td>Temperature Range</td>
</tr>
<tr>
<td>Thermocouple Diameter</td>
</tr>
<tr>
<td>Thermocouple Connector</td>
</tr>
<tr>
<td>Thermocouple Sensitivity</td>
</tr>
</tbody>
</table>
3.1.2. **Omegadyne 1,000 PSIG Pressure Transducers**

Figure V shows that diagram numbers corresponding to the pressure transducers were eight and twenty-eight. Like the thermocouples, the transducers were placed at key locations in the system so as to determine an accurate representation of the system process. The first pressure transducer, also known as PT #1, was placed in the primary section of the system apparatus for several reasons. Like TC #1, PT #1 gave inclinations into what the working fluid was doing at this stage of the process, i.e. as the vapor left the primary tank. PT #1 also served another important because it provided pressure data during the refill process. A second pressure transducer was also placed in a downstream position to assess pressure losses in the system. Pressure Transducer #2 was placed at the entrance of the secondary fluid tank, at the exit of the process section, along with TC #3. The reason for this was to get the complete pressure differential in the working fluid from the outlet of the primary tank to where the fluid must accomplish work by purging the secondary fluid.
Although there is some pressure loss going from where pressure transducer #2 took its measurement to where the working fluid enters the secondary fluid tank, this was considered negligible in reference to the pressure loss created by passing through the process section. The pressure transducers were special ordered from Omega and the technical specifications as well as Figure XVI of the transducers can be examined below.

**Table VII: Specifications of Pressure Transducers #1 & #2**

<table>
<thead>
<tr>
<th>Transducer Type</th>
<th>Gauge</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transducer Range</td>
<td>0-1,000 PSI</td>
</tr>
<tr>
<td>Transducer Output</td>
<td>10 mV/V</td>
</tr>
<tr>
<td>Transducer Accuracy</td>
<td>± 0.20%</td>
</tr>
<tr>
<td>Transducer Process Fitting</td>
<td>1/4-18 Female NPT</td>
</tr>
<tr>
<td>Temperature Compensation</td>
<td>-7°C to 60°C (20°F to 140°F)</td>
</tr>
</tbody>
</table>

**Figure XVI: Pressure Transducers Used in Study**

3.1.3. **ARI Industries Type J Multilevel Thermocouple Probe**

One of the most important as well as unique measurement devices used in this study was the multilevel thermocouple probe. The multilevel thermocouple probe allows for multiple temperature readings to be taken along the physical length of a probe. The true benefit to a multilevel thermocouple in this particular application was the fact that the probe could be used to take multiple temperature readings along the length of the working fluid tank. Like the other measurement devices, the multilevel thermocouple
served to establish an accurate depiction of the process; however, unlike the other
devices, the multilevel thermocouple probe was designed to specifically gather data
regarding the energy addition phase of the process.

As noted in the previous section, the high heat fluxes required to supply the vapor
delivery process causes a temperature drop in the primary tank. The multilevel
thermocouple probe allowed the working fluid temperature to be monitored at different
heights as a function of time during the run. The multilevel thermocouple probe was also
used as a possible method to track the vaporization rate of the working fluid by
attempting to measure the fluid level in the working fluid tank. If there was a
distinguishable temperature difference between the liquid and vapor, the probe could
potentially provide the approximate location of the liquid level as it dropped throughout
the process.

The multilevel thermocouple was also beneficial from a geometric standpoint.
The notion to be able to measure temperature at multiple points inside the working fluid
tank along its height was proposed early on in the study and many ideas were pitched
regarding how the temperature could be measured in such a manner. Ultimately, the
result was to custom order a thermocouple device that consisted of multiple
thermocouples that could take the required measurements. The multilevel thermocouple
probe was engineered to have as small as possible packaging and eliminated the second
best solution of custom building a probe in house. The technical specifications as well as
an image of the multilevel thermocouple are shown below.
Table VIII: Specifications of Multilevel Thermocouple

<table>
<thead>
<tr>
<th>Thermocouple Type</th>
<th>Type J</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple Conductors</td>
<td>Iron- Constantan</td>
</tr>
<tr>
<td>Thermocouple Sheath</td>
<td>304 Stainless Steel</td>
</tr>
<tr>
<td>Temperature Range</td>
<td>-184°C to 540°C</td>
</tr>
<tr>
<td>Thermocouple Probe Diameter</td>
<td>0.125 Inches</td>
</tr>
<tr>
<td>Thermocouple Connector</td>
<td>Molded Connector</td>
</tr>
<tr>
<td>Thermocouple Junction</td>
<td>Grounded</td>
</tr>
<tr>
<td>Thermocouple Probe Length</td>
<td>21”</td>
</tr>
<tr>
<td>Start of Thermocouples from Probe Tip</td>
<td>1”</td>
</tr>
<tr>
<td>Thermocouple Separation Distance</td>
<td>5”</td>
</tr>
<tr>
<td>Number of Thermocouples in Probe</td>
<td>4</td>
</tr>
</tbody>
</table>

Figure XVII: Depiction of Multilevel Thermocouple Probe[2]

Figure XVIII: Multilevel Thermocouple Probe

3.1.4. Kobold Flow Meter

The final parameter to measure was the flow rate of secondary fluid expelled from the system. The reason that the output secondary fluid rate was measured as opposed to
directly measuring the flow rate of the working fluid was as follows. The system was
designed to limit the output fluid flow via diagram item forty-two from Figure V, the
orifice plate. As one may expect, the more conditions that must be satisfied by a
particular measurement device ultimately increases the price. Using the orifice plate to
limit the outgoing secondary fluid’s pressure and volumetric flow rate meant that the
system could use a flow meter not rated to the high pressure levels that the working fluid
would be operating at; therefore a less costly flow meter was required for the secondary
fluid. Also, it is important to note again that the flow meter was placed downstream of
the secondary fluid tank to measure its flow rate and not the working fluid. Although this
seems an obvious notion, placing the flow meter before the secondary fluid tank meant
that a vapor flow rate rather than liquid flow rate would have to be measured.

Another important reason for placing the flow meter downstream of the secondary
was due to the promotion of safety. The safety of the system operators was of the utmost
importance, especially since the system operated under high pressures. Therefore, the
system was designed to have the fewest amount of high pressure connections. Although
moving the flow meter downstream doesn’t seem like a significant reduction in the
amount of connections, it was reduction in at least two high pressure connections which
was deemed appropriate for promoting safety.

The technical specifications as well as an image of the Kobold Flow Meter are in
Table IX and Figure XIX.
Table IX: Specifications of Kobold Flow Meter

<table>
<thead>
<tr>
<th>Flow Meter Type</th>
<th>Magnetic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Power</td>
<td>24 VDC ±20%, 80 mA max</td>
</tr>
<tr>
<td>Electrical Connection</td>
<td>Micro-DC plug, 5 pin male</td>
</tr>
<tr>
<td>Display Type</td>
<td>3 Digit LED 0.5” Characters</td>
</tr>
<tr>
<td>Housing Material</td>
<td>304 Stainless Steel</td>
</tr>
<tr>
<td>Analog Output</td>
<td>0/4-20 mA into a max. loop load of 500 Ω</td>
</tr>
<tr>
<td>Flow Range</td>
<td>1.3-26 gpm</td>
</tr>
</tbody>
</table>

Figure XIX: Kobold Flow Meter

3.1.5. Airgas Analog Pressure Gauge

The analog pressure gauge was an important measurement component used primarily during the refill process of the working fluid tank. It should be noted that the analog pressure gauge, along with the rest of the refill system components, were isolated from the rest of the system via the valve manifold or the separation valve during system operation. The analog pressure gauge was placed in the system for the sole purpose of monitoring the working fluid pressure in the secondary tank during the refill process. The pressure that was measured was correlated to the amount of working fluid in the tank as well as its temperature, which tended to drop during the refill process. Monitoring the
The analog gauge was useful in setting a refill flow rate that was sustainable. The weight of the secondary tank was also measured to assess the amount of transmitted CO$_2$.

The technical specifications for the analog pressure gauge are shown below including an image of the gauge similar to the one used in the experiment.

Table X: Specifications of Airgas Pressure Gauge

<table>
<thead>
<tr>
<th>Specification</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Housing Material</td>
<td>316 Stainless Steel</td>
</tr>
<tr>
<td>Output Connector</td>
<td>¼” Male NPT</td>
</tr>
<tr>
<td>Pressure Range</td>
<td>0-1,000 psig</td>
</tr>
<tr>
<td>Accuracy</td>
<td>±1% full scale</td>
</tr>
</tbody>
</table>

![Figure XX: Airgas Pressure Gauge](image_url)

3.2. Key Plumbing Equipment

An interesting thing to note is that this study was not only concerned with gathering data regarding the process, but rather how to construct the system to house the process. The measurement equipment was just a small part of the physical construction of the system apparatus where a majority of the concern and effort was spent on selecting specific plumbing components rated for the expected operating pressures. Now, the following section goes into detail regarding plumbing components of the apparatus; however, only the most significant items are examined in detail here.
3.2.1. \( \frac{1}{2} \) \textit{FNPT SS Union Ball Joint}

The union ball joint served a specific purpose for the system apparatus and more than anything it was implemented into the overall design for reducing the amount of labor required to assemble and disassemble the system. An image of the union all joint is shown in Figure XXI for clarity into the design of the item.

![Figure XXI: Union Ball Joint\(^{[45]}\)](image)

As can be inferred from the figure above, the union ball joint is a rather ingenious component for plumbing and piping systems. Obviously, when components are threaded, they must be rotated to be connected properly. The fact that components need to be rotated in order to secure a connection was problematic when said components were large, cumbersome, or connected to a stationary system; thus the invention of swivel and ball joints. The union ball joint allows parts of the system to be connected while remaining stationary because both halves of the ball joint can rotate while maintaining a solid connection.
Essentially, the union ball joints were placed at various positions in the system so that certain sections could be isolated if need be for maintenance or new component implementation. By examining the system diagram, one will notice that union ball joints allowed the isolation of the primary working fluid tank, the secondary working fluid tank, and the secondary fluid tank from the process section of the system. In addition the process section could be broken into two parts isolating the 3-Way valve manifold. The union ball joints were a necessary component to the system apparatus; without the ball joints, labor required to assemble and disassemble the system would have increased tenfold. It should be noted that a union ball joint was not placed beyond the secondary fluid tank because the orifice plate essentially served as a union and could be used to isolate the outgoing section of the system in the same manner as a ball joint could. Note that the union ball joints used in this study were rated to handle a pressure up to 6,000 psig.

3.2.2. High Pressure $\frac{1}{8}$" TC Probe Adapter

A rather unique and interesting component was the high pressure $\frac{1}{8}$" TC probe adapter, which allowed for the adaptation of the individual thermocouple probes into the system. These adapters worked by using a compression fitting type connection to secure a thermocouple into the adapter and thusly into the main piping. The adapter created a successful seal between the thermocouple probe and the adapter wall capable of withstanding pressures up to 15,000 psig. The versatility of the high pressure adapter was in the fact that the thermocouple probe did not have to be modified by securing a collar to the probe via an adhesive compound; the collar simply slid over the probe then the assembly was secured by tightening the adapter cap. Note that in each place a
thermocouple was placed, a thermocouple adapter such as the one described here was used except for the multilevel thermocouple probe which used an NPT fitting directly. Figure XXII shows the section view of the thermocouple adapter.

![Figure XXII: High Pressure Thermocouple Adapter](image)

3.2.3. **SS 3-Way \( \frac{1}{2}'' \) FNPT Instrument Valve**

An important component in the system apparatus was the 3-way valve. This manifold, like the union ball joints, made isolating part of the system easy. The true purpose of the manifold was to allow the secondary section of the system to always be connected to the primary section of the system. This was a useful concept since keeping the secondary section connected to the primary section meant that the refill line did not have to be purged repeatedly. Essentially the idea was to be able to refill the system with as little time as possible in between running the experiments. This was compatible with the idea that potentially the primary working fluid tank could be refilled multiple times with the same secondary working fluid tank. If the line had to be broken in between
refills, then the working fluid blown to atmosphere would be useless and the system operators would be wasting CO$_2$. An added benefit of using the manifold and keeping the secondary working fluid tank continually connected to the system, was that the working fluid could be diverted back to it in the event of a problem. For example, in the event that a component failed and caused the working fluid to be trapped in the system, the secondary working fluid tank could be opened up to capture the working fluid. In addition, the constant connection to the valve manifold gave the secondary section access to the systems pressure relief valve. Figure XXIII shows the manifold.

![Figure XXIII: 3-Way Instrument Valve][45]

3.2.4. **90° 120V Full Port Solenoid Valve with 1/4" NPT Connections**

The solenoid valve facilitated automation of the system by successfully creating remote access to control the isolation between the working fluid and the secondary fluid. It is important to note that when the solenoid valve was activated, the process initiated due to the working fluid becoming exposed to low pressure conditions. The solenoid valve facilitated near instantaneous exposure to low pressure conditions, which was impossible to establish via a manual valve. Similarly, it was desired that the system be shut down in the same near instantaneous manner. For the following two reasons, the
quicker the system could be shut down, the less inapplicable data was obtained during the time the system returned to equilibrium conditions. The second advantage of a fast shut down was that if the system needed to be stopped half-way through a run, it was desired to save as much of the working fluid as possible from exiting the working fluid tank.

The solenoid valve also created safer working conditions that were desired by system operators. Since the system was entirely new and had never been tested, it was important to be able to run the system while the operators were out of shrapnel distance in the event of catastrophic failure; the solenoid facilitated this desired remote operation. Properties and an image of solenoid valve used can be examined below in Table XI and Figure XXIV.

### Table XI: Solenoid Valve Properties

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve Body</td>
<td>316 Stainless Steel</td>
</tr>
<tr>
<td>Piston</td>
<td>303 Stainless Steel</td>
</tr>
<tr>
<td>Seal</td>
<td>Teflon</td>
</tr>
<tr>
<td>O Ring Seal</td>
<td>Teflon</td>
</tr>
<tr>
<td>Spring</td>
<td>302 Stainless Steel</td>
</tr>
<tr>
<td>Plunger</td>
<td>430 Stainless Steel</td>
</tr>
<tr>
<td>Bonnet</td>
<td>316/430 Stainless Steel</td>
</tr>
<tr>
<td>Bonnet Retainer</td>
<td>430 Stainless Steel</td>
</tr>
<tr>
<td>Fluid Temperature</td>
<td>-350°F to 400°C</td>
</tr>
<tr>
<td>Power</td>
<td>22 Watts</td>
</tr>
<tr>
<td>Volts</td>
<td>120 VAC</td>
</tr>
<tr>
<td>AC Inrush</td>
<td>2.5A @ 120 VAC</td>
</tr>
<tr>
<td>AC Holding</td>
<td>0.2A @ 120 VAC</td>
</tr>
<tr>
<td>Duty</td>
<td>Continuous</td>
</tr>
</tbody>
</table>
3.2.5. O’Keefe Controls Co. Precision Orifice w/Changeable Orifice Plate

The orifice plate controlled the output flow rate of the secondary fluid which, in turn, controlled the output flow rate of the working fluid. The orifice not only limited the flow rate of the system, but also increased the pressure differential from the secondary fluid to atmosphere. This allowed the pressure of the secondary fluid to decrease to a pressure that the flow meter could physically handle. In addition, the orifice plate offered the ability to run the process at different flow rates so as to gain a solid understanding of the processes occurring in the system. It was important to run at different conditions to determine the vapor flow rate, which could be produced at a steady pressure given the constraint imposed by the heat addition process. With this particular orifice plate, different orifice plates could be easily swapped out to gain the desired differences in flow rates desired.

The orifice plate fitting was constructed in the same manner as the union ball joints; therefore, the system could be separated at this particular point with ease. The orifice plate diameter determined the secondary fluid flow rate, which was estimated from the Bernoulli Equation as explained in Chapter 2. If the orifice diameter changed,
thus did every other parameter in the system. An image of the orifice plate can be examined below in Figure XXV.

![Orifice Plate Image]

**Figure XXV: Orifice Plate**

3.2.6. **Manual Override**

The manual relief valve was designed into the system for two specific reasons. Firstly, the manual relief valve provides a failsafe for the system in the event that the system becomes over pressurized. If the system reached a pressure that was above desired conditions, the relief valve was designed to vent the working fluid in the system to atmosphere before the pressure reached the maximum allowable operating pressure of the components. Secondly, a relief valve with a manual override was specifically selected for this apparatus so that, if need be, the system operators could relieve pressure in the system manually. This was most beneficial after the system refill process when the secondary working fluid tank needed to be removed from the system and be refilled itself. If the secondary section of the system still contained the working fluid, the operators could relieve the fluid to atmosphere in a safe manner before physically separating the system.
It should be noted that the relief valve selected was rated to 1,000 psig. This pressure was sufficient due to the fact that the working fluid was never meant to exceed approximately 500 psig during operation and the only time the working fluid would be able to reach 1,000 psig was if the temperature of the fluid rose to nearly 72°F. Now, this was possible; however, due to the fact that the system was contained in the rocket test cell, this was deemed satisfactory for safety if the relief valve was tripped. An image of the relief valve can be examined below in Figure XXVI.

![Figure XXVI: Manual Relief Valve](image)

3.2.7. **Stainless Steel 1/2” Tubing**

Stainless steel tubing was used in some locations to connect different parts of the system. The tubing was convenient, because it gave greater flexibility in the placement of system sections. For example, the process section did not have to be placed directly next to the primary section which proved to be a safety benefit since if either section failed catastrophically during the process, they were far enough apart so as not to affect
each other. The additional benefit to using tubing at specific locations was the fact that the tubing adapters used compression type fitting which could be easily undone if needed. Also since the tubing did not have to be orientated a specific way, the adapters could be used like the union ball joints, thus creating two additional easy access points into the system. An image of a tubing adapter can be examined below in Figure XXVII.

![Tubing Adapter](image)

**Figure XXVII: Tubing Adapter[^45]**

3.2.8. **Stainless Steel Braided Hose w/Swivel Fittings**

An eighteen inch steel braided hose was used to connect the secondary section to the valve manifold and the fact that not only was it flexible, but had swivel connections proved most excellent. For example, since the hose was flexible, the secondary working fluid tank could be easily and accurately weighed during the refill process. Tubing was initially considered for use at this point, but ultimately the flexibility the hose could provide outweighed any potential benefit to using tubing.

3.2.9. **5/8” Garden Hose Y-Valve (Back Fill Valve)**

Prior to each test run, the system must be primed with the secondary fluid (water). Priming the system with secondary fluid affirmed the fact that all of the air would be removed from the secondary fluid tank and its associated components, in addition to maintaining a healthy volume of secondary fluid in the system. The back fill valve, as
depicted in the system diagram, was essentially a three-way valve that allowed for the secondary fluid to flow reversely into the secondary fluid tank while being isolated from the atmosphere via another isolation valve. When the system apparatus was flowing, the valve isolating the flow meter was opened and the valve to the secondary fluid source was isolated. It should be noted that in order to achieve a successful back fill, the orifice plate had to be removed, otherwise the fill process would take an enormous amount of time. Note that this was a manually operated valve cluster.

3.2.10. **Fluid Damper (Diagram # 44-53)**

The idea behind this set of components was to provide a damper, or fluid settling chamber, for the exiting high velocity secondary fluid jet. As was shown in the previous section, the force that could potentially be created from the secondary fluid passing through the orifice plate was significant and therefore needed to be reduced. This was facilitated by allowing the secondary fluid to exit the orifice into a large diameter manometer-like device charged with secondary fluid before testing. This damper was successful since the increased diameter and the viscous forces created by injecting the secondary fluid jet into a nearly stationary fluid were high, forced the velocity to drop in a controlled manner.

3.3. **Ancillary Equipment Used in the System Apparatus**

In addition to the equipment shown in Figure V, other ancillary equipment was used to conduct the study; generally this included custom equipment that had to be manufactured in house. This equipment was usually supportive equipment that was used to physically support the system apparatus.
3.3.1. **Temperature Control Fluid Tanks**

This piece of equipment was a custom built tank that contained the temperature control medium (i.e. the ice bath) that was used as a temperature controller for the primary working fluid tank. This tank was made from $\frac{1}{16}$" steel sheet metal manufactured to house part of the primary section of the system so that the temperature control medium could be used to surround this section and control the working fluid temperature. The sheet metal was welded together and leak tested to assure it was going to function properly during testing. It should be noted that care had to be taken when designing the tank so as to make sure the proper distance between the working fluid tank and the temperature control fluid tank walls was attained. The idea was to make sure a great enough distance existed between the two surfaces so that a thick layer of temperature control medium could be created and maintained throughout the process. The volume of the temperature control medium had to be large enough to act as a temperature ‘reservoir’ while the working fluid was being discharged. The tank also included measures to essentially suspend the working fluid tank so that the bottom was exposed to the control medium. A model of the tank is shown in Figures XXVIII through XXX with different views to see each aspect of the tank.
Figure XXVIII: Side View of Temperature Control Fluid Tank

Figure XXIX: Top View of Temperature Control Fluid Tank
3.3.2. **Secondary Section Tank**

A custom temperature control fluid tank needed to be created for the secondary section as well, since the working fluid needed to be refilled into the system at a desired temperature to maintain a pressure difference between the primary and secondary tanks. Also, another reason for completing the refill process at a desired control temperature was to make sure the system relief valve was not activated if the secondary system got too warm due to ambient conditions. The temperature control fluid tank for the secondary section was actually an exact replica of the temperature control fluid tank for the primary section as shown in Figure XXX.

The remaining components of the system apparatus that are not discussed in detail here are simply standard pipe fittings and do not require any significant remarks;
however, the specifications as well as figures of the various pipe fittings used in the system can be examined in the appendix.

Images of the completed system can be examined in Appendix D. It should be noted that the system constructed here really served as means to measure the process being studied. The construction of this system was not designed for space optimization or anything of that particular nature; the system was simply fashioned to gather data and house a process.

Now that the system components have been described in detailed it is important to reflect on the procedures and details regarding the calibration of the measurement equipment in the system apparatus. The devices calibrated include four type J thermocouples, one multilevel thermocouple consisting of four type J thermocouples, two high pressure transducers, and one magneto-inductive flow meter.
Chapter 4: Measurement Equipment Calibration

The following section describes in detail the calibration of the measurement equipment and all associated data acquisition components.

4.1. Data Acquisition (DAQ) and LabVIEW 6

Data acquisition was a large component of this study. For example, a data acquisition system not only had to be assembled to take the final measurements of the system process, but had to be used to calibrate each measurement device. In order to accomplish this goal, the proper type of data acquisition hardware had to be selected. In this study, a Measurement Computing, Inc. PCI-DAS08 Analog Input & Digital I/O data acquisition board was used in conjunction with a CIO EXP 32 multiplexer board.

The PCI-DAS08 is an analog to digital data acquisition board designed to read and convert an analog measurement signal from a measurement device. The specifications of the PCI-DAS08 are listed below:

- Eight single-ended 12 bit analog signals
- 12 bit A/D resolution
- Sample rates up to 40 Hz
- ±5 Volts
- Three 16 bit counters
- Seven digital I/O bits

The CIO EXP 32 multiplexer board is an expander board designed to increase the number of available analog input channels for measurement signals. The multiplexer board worked in conjunction with the PCI-DAS08 DAQ board to expand available
channels from 8 channels to 32 channels. Note that the gain of half of the 32 channels on the multiplexer can be adjusted separately from the other half of the expanded channels; each gain controller can be set to have a gain of 1, 10, 100, 200, and 500.

Additionally, a separate smaller interface board was used to specifically measure the process signal coming from the flow meter. The reason for the additional expander board stemmed from the fact that the gain settings for the thermocouples and pressure transducers were not compatible with the flow meter.

In conjunction with these boards, a simple LabVIEW 6 code was used to display, log, and export the data to Microsoft Office Excel. This singular code, known as VI (virtual instrument) file was used for both the calibration of the measurement devices as well as the actual data collection system for the process.

The subtle things that should be noted about the code lie in the subVI’s embedded in the VI. Each channel contains a subVI that essentially tells the VI what type of signal should be read and output. For example, if the channel is a temperature measurement device, then the subVI converts the analog signal to a digital temperature signal wherein instead of outputting a voltage to the front panel or file, the program output a temperature value. If the signal were to be output as a voltage, the conversion would simply be a straight A/D conversion with no manipulation embedded to convert said signal to a different unit.

The LabVIEW code was used to determine the calibration constants for each measurement device. This was done by comparing the digital signal measured via the DAQ system and a physical measurement of that same signal, then extracting a linear
relationship between the two. Each device’s calibration could be completed separately using the same code or simultaneously if the measurement devices being calibrated were the same and measuring the same physical phenomenon.

The subVI code was similar in that the user is required to specify the Board Number and Channel Number for each measurement device. By using a subVI to combine several functions into one icon, the block diagram for the main VI is simplified the output signal as well as the time is displayed on the front panel so that the user can monitor the data acquisition results as the program is running. The Number of Points and the Sampling Rate can be specified on the front panel so as to easily allow for changes in the data acquisition process. Below Figure XXXI of the front panel and Figure XXXII block diagram of the LabVIEW code are shown.

![Figure XXXI: Front Panel of PreVapS.VI](image-url)
Figure XXXII: Block Diagram of PreVapS.VI

The channel numbers for each measurement device are listed in Table XII.

Table XII: DAQ Channel Numbers

<table>
<thead>
<tr>
<th>Measurement Device</th>
<th>Channel Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouple #1</td>
<td>17</td>
</tr>
<tr>
<td>Thermocouple #2</td>
<td>18</td>
</tr>
<tr>
<td>Thermocouple #3</td>
<td>19</td>
</tr>
<tr>
<td>Thermocouple #4</td>
<td>20</td>
</tr>
<tr>
<td>Thermocouple #5</td>
<td>21</td>
</tr>
<tr>
<td>Thermocouple #6</td>
<td>22</td>
</tr>
<tr>
<td>Thermocouple #7</td>
<td>23</td>
</tr>
<tr>
<td>Thermocouple #8</td>
<td>24</td>
</tr>
<tr>
<td>Pressure Transducer #1</td>
<td>35</td>
</tr>
<tr>
<td>Pressure Transducer #2</td>
<td>34</td>
</tr>
<tr>
<td>Flow meter</td>
<td>5</td>
</tr>
</tbody>
</table>

The description of the procedures for calibrating each measurement device is outlined below.
The channel numbers for the thermocouples and pressure transducers are in a higher range since they are connected to the multiplexer board. The description of the procedures for calibrating each measurement device is outlined below.

4.2. **Omegadyne Type J Stainless Steel Thermocouples (TC #1 – TC#4)**

The thermocouples were a vital measurement device to this study due to the fact that the entire process being studied was a temperature dependent process. Because of this realization, thermocouples were used in excess to make sure temperature was captured at any point deemed reasonable. Therefore, four Type J thermocouples were used to measure the working fluid’s temperature at points scattered throughout the system and as such all four of these thermocouples had to be calibrated accordingly.

An important concept to note is that thermocouple signals were linearized by the temperature reading sub VI in LabVIEW, therefore, it was deemed as a reasonable engineering approximation that even though the estimated temperature range that could be seen by any one thermocouple could be −58°F to 32°F, calibration data would be valid if taken from 32°F to nearly 212°F. Therefore a simple experiment was set up to do so.

The first step was to wire the thermocouples from the multiplexer board in the control room to the test cell on the other side of the concrete blast wall. For clarity, Figure XXXIII below shows how the thermocouples were wired to the multiplexer board. Once the thermocouples were wired up, the following procedures were used to calibrate the thermocouples.
4.2.1. **Thermocouple Calibration Procedures**

-The ice, water, and the four thermocouple probes were placed in a 500 mL beaker.

-Using a conventional thermometer, the temperature of the ice bath was measured until it reached a constant 32°F throughout the entire ice bath.

-The PreVapS.VI LabVIEW file was set up to measure the appropriate channel(s) from the DAQ system and the number of data points to be collected. For the calibrations completed, the amount of data that was recorded for each temperature reading varied. Nonetheless, an appropriate amount of data was taken to complete the calibration for each device. Another important thing to note is that the gain on the multiplexer board was set to 100 for the thermocouples.

-As the program collects data continually the ice bath is stirred to make sure the temperature stays consistent.
- Once the program finishes collecting data, the file is saved. The data is then opened in Microsoft Excel and manipulated to determine the calibration coefficients.

- The ice was removed from the water and the water was heated. At various temperatures from zero to one hundred, data was collected using the LabVIEW program. The temperatures were recorded in intervals of 50°F. The data that was collected for thermocouples #1 through #4 is shown below in Table XIII.

**Table XIII: Thermocouple Calibration Data for TC #1-#4**

<table>
<thead>
<tr>
<th>Actual Temp. (°C)</th>
<th>TC #1 Measured Temp. (°C)</th>
<th>TC #2 Measured Temp. (°C)</th>
<th>TC #3 Measured Temp. (°C)</th>
<th>TC #4 Measured Temp. (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>2.52</td>
<td>2.49</td>
<td>2.40</td>
<td>3.04</td>
</tr>
<tr>
<td>25.5</td>
<td>27.10</td>
<td>27.13</td>
<td>27.45</td>
<td>27.75</td>
</tr>
<tr>
<td>38.5</td>
<td>40.96</td>
<td>41.49</td>
<td>41.34</td>
<td>41.75</td>
</tr>
<tr>
<td>50</td>
<td>51.43</td>
<td>53.92</td>
<td>55.41</td>
<td>52.62</td>
</tr>
<tr>
<td>59</td>
<td>60.11</td>
<td>62.97</td>
<td>63.61</td>
<td>61.20</td>
</tr>
<tr>
<td>68</td>
<td>68.61</td>
<td>69.74</td>
<td>71.03</td>
<td>69.62</td>
</tr>
<tr>
<td>78</td>
<td>77.08</td>
<td>77.68</td>
<td>77.67</td>
<td>77.77</td>
</tr>
<tr>
<td>88.5</td>
<td>88.10</td>
<td>90.98</td>
<td>94.79</td>
<td>89.22</td>
</tr>
</tbody>
</table>
Figure XXXIV: Calibration Scatter of Thermocouples #1 through #4

As can be inferred from the Figure XXXIV, the output behavior of the thermocouples are indeed linear. It is also important to note that as was described in the section above, the output of the LabVIEW code is temperature and not voltage. Table XIV below shows the calibration coefficients that were determined for thermocouples #1 through #4. Since the output responses of the thermocouples are linear and the coefficients are used in the following manner:

\[ \text{Actual Temperature} = (\text{Measured Temperature}) \times A + B \]

where \( A \) and \( B \) are the thermocouple calibration coefficients.
Table XIV: Calibration Coefficients for Thermocouples #1 through #4

<table>
<thead>
<tr>
<th>TC</th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>1.04</td>
<td>-3.05</td>
</tr>
<tr>
<td>#2</td>
<td>1.01</td>
<td>-2.82</td>
</tr>
<tr>
<td>#3</td>
<td>0.98</td>
<td>-2.08</td>
</tr>
<tr>
<td>#4</td>
<td>1.02</td>
<td>-3.04</td>
</tr>
</tbody>
</table>

4.2.2. ARI Industries Type J Multilevel Thermocouple Probe (TC #5 – TC#8)

As was described in a previous section, one of the most unique and important measurement devices that was used in this study was the multilevel thermocouple. The multilevel thermocouple allows for multiple temperature measurements to be taken along the length of a probe. This facilitates temperature measurements at different vertical locations in the primary working fluid tank which provides an insight to how the working fluid is behaving during the vapor production process. The calibration process regarding the multilevel thermocouple was the same as for the four other single thermocouples save for the fact that a 1000mL graduated cylinder was used instead of a 500mL beaker. The graduated cylinder was used in order to accommodate the length of the probe. One thing to note regarding the difference in the calibration process between the two types of thermocouples is that water could not be boiled in the graduated cylinder due to geometry concerns; therefore, the water was heated in a beaker and poured into the cylinder. The calibration data for the multilevel thermocouple is shown in Table XV and Figure XXXII below.
Table XV: Thermocouple Calibration Data for TC #5-#8

<table>
<thead>
<tr>
<th>TC #5</th>
<th>TC #6</th>
<th>TC #7</th>
<th>TC #8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actual Temp (°C)</td>
<td>Measured Temp (°C)</td>
<td>Actual Temp (°C)</td>
<td>Measured Temp (°C)</td>
</tr>
<tr>
<td>51.5</td>
<td>50.87</td>
<td>57</td>
<td>57.35</td>
</tr>
<tr>
<td>25.5</td>
<td>25.17</td>
<td>51.5</td>
<td>52.20</td>
</tr>
<tr>
<td>1</td>
<td>5.31</td>
<td>25.5</td>
<td>25.96</td>
</tr>
<tr>
<td>0</td>
<td>3.59</td>
<td>0</td>
<td>3.59</td>
</tr>
</tbody>
</table>

Figure XXXV: Calibration Scatter of Thermocouples #5 through #8

Table XVI: Calibration Coefficients for Thermocouples #5 through #8

<table>
<thead>
<tr>
<th>TC</th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>#5</td>
<td>1.10</td>
<td>-3.93</td>
</tr>
<tr>
<td>#6</td>
<td>1.05</td>
<td>-3.07</td>
</tr>
<tr>
<td>#7</td>
<td>1.07</td>
<td>-3.24</td>
</tr>
<tr>
<td>#8</td>
<td>1.12</td>
<td>-4.23</td>
</tr>
</tbody>
</table>

As can be seen from comparing the calibration data from the two sets of thermocouples, more data points were taken for the individual thermocouples than for the
multilevel probe. This was in part due to the geometry of the multilevel probe hindering the calibration procedures; however, despite this fact, the thermocouple coefficients were successfully determined. Note that there were different calibration temperatures used for the thermocouples since not all temperatures tested produced viable results.

4.3. **Omegadyne 1,000 PSIG Pressure Transducers**

As was stated in previous sections, this study used two custom built Omega pressure transducers to measure the pressure at two points in the system during operation. The first transducer measured the pressure of the working fluid at the primary working fluid tank exit. The second transducer was placed near the entrance of the secondary fluid tank. It should be noted that to gather calibration data for the pressure transducers, the same PreVapS.VI code was used.

4.3.1. **Pressure Transducer Calibration Procedures**

The primary tool used to calibrate the pressure transducer was a deadweight tester. The deadweight tester allows the user to create a known artificial pressure differential that can be measured with a pressure transducer. The deadweight tester creates this artificial pressure using calibrated weights to act upon a piston of known area in a hydraulic system. Figure XXXVI below shows the deadweight tester used in the calibration process.
As with the thermocouples, the first step to calibrating the pressure transducers was to make the electrical connections from the control room to the test cell. Figure XXXVII below shows how the transducers are connected to the multiplexer board. It should be noted that the transducers require an excitation voltage; therefore, the transducers were wired to a 12V power supply as well as the multiplexer board.
The DAQ board can only handle signals that were plus or minus five volts. Because of this, it was necessary to make sure that the measurement devices did not output a signal greater than five volts, while maintaining a fine enough resolution to acquire accurate data. Obviously, this is why the multiplexer board allows for adjustable gain for measurement devices. To maximize resolution of the pressure transducer signals, a gain of 10, in conjunction with a potentiometer was used to manipulate the incoming signal to the DAQ board. The potentiometer was used to adjust the pressure transducer signal to zero volts at zero gauge pressure. Table XVII below shows the potentiometer settings for each pressure transducer.

**Table XVII: Pressure Transducer Potentiometer Settings**

<table>
<thead>
<tr>
<th>Transducer</th>
<th>Potentiometer Setting (Ω)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>3.61 kΩ</td>
</tr>
<tr>
<td>#2</td>
<td>3.614 kΩ</td>
</tr>
</tbody>
</table>

The transducers had to be connected to the deadweight tester via the appropriate NPT pipe adapters. The transducers were female $\frac{1}{4}$" NPT; therefore an adapter needed to be used to go from the transducer to the tester.

The deadweight tester could create an artificial pressure up to 680 psig; therefore, pressure measurements were taken in intervals of 100 psig until the maximum pressure was reached. Once the maximum pressure was achieved, the pressure was reduced in intervals of 100 psig. This meant that for any one pressure, two data points were taken for each transducer.

Once the deadweight tester was set up at a specific pressure, the LabVIEW program was executed to collect the needed data. This process was repeated for the
second transducer and the data for each was imported into MS Excel where the calibration coefficients were determined. The pressure transducers were linear in the output response therefore the calibration coefficients determined were for a linear function of the form:

\[ Actual\text{Pressure} = (\text{MeasuredVoltage}) \times A + B \]

where \( A \) and \( B \) are the pressure transducer calibration coefficients.

The data for the calibration of each transducer can be examined in Figure XXXVIII & XXXIX and Table XVII & XVIII below.

<table>
<thead>
<tr>
<th>Pressure (psig)</th>
<th>PT #1 Output (V)</th>
<th>PT #1 Output (V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-6E-05</td>
<td>-0.00373</td>
</tr>
<tr>
<td>100</td>
<td>0.114</td>
<td>0.104</td>
</tr>
<tr>
<td>200</td>
<td>0.223</td>
<td>0.213</td>
</tr>
<tr>
<td>300</td>
<td>0.333</td>
<td>0.321</td>
</tr>
<tr>
<td>400</td>
<td>0.441</td>
<td>0.431</td>
</tr>
<tr>
<td>500</td>
<td>0.550</td>
<td>0.540</td>
</tr>
<tr>
<td>600</td>
<td>0.657</td>
<td>0.647</td>
</tr>
<tr>
<td>680</td>
<td>0.742</td>
<td>0.735</td>
</tr>
</tbody>
</table>
Figure XXXVIII: Pressure Transducer #1 Calibration Curve

Figure XXXIX: Pressure Transducer #2 Calibration Curve
Table XIX: Pressure Transducer Calibration Coefficients

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>914.38</td>
<td>-2.88</td>
</tr>
<tr>
<td>#2</td>
<td>920.02</td>
<td>3.95</td>
</tr>
</tbody>
</table>

As can be seen from Figures XXXVIII and XXXIX, the pressure transducers’ output responses were indeed linear.

4.4. Kobold Flow Meter Calibration

In order to accurately determine the overall performance of the system, the output flow rate had to be measured as a function of time. A magnetic type flow meter with a current output was used for this principle. The flow meter, like the other measurement devices, had an expected linear output response and thus followed the following relationship regarding the calibration coefficients:

$$ActualFlowRate = (MeasuredVoltage) \times A + B$$

where $A$ and $B$ are the flow meter calibration coefficients.

The flow meter, unlike the other measurement devices, did not connect directly to the multiplexer board but rather to an expansion board which connected directly to the DAS08, since no amplification of the signal was necessary. A resistance of 234.6 $\Omega$ was placed across the output signal, seated the output current to output voltages from zero to five volts across the range of operation. Also required for the flow meter was an excitation voltage of 24 V, which was supplied by an Omega power supply. The manner in which the flow meter was connected to the multiplexer board is shown in Figure XL.
The procedures for calibrating the flow meter are straightforward and are outlined below.

**Figure XL: Flow Meter Connection to the Multiplexer Board**

### 4.4.1. Kobold Flow Meter Calibration Procedures

Once the flow meter was wired properly, as well as connected to a water supply source, the power supply was turned on. Upon first power up, the flow meter entered a short set up session where when finished the flow meter’s default settings were set up.

The calibration of the flow meter was conducted by setting a specific flow rate then measuring the amount of time required to fill a certain volume. The time to fill a specified volume, along with the digital flow rate display in gallons per minute from the flow meter itself, and the output voltage to the DAQ was used to determine the calibration coefficients which are outlined in Table XX below. The waterline in the test cell was the water supply source for the flow meter and as such the maximum flow rate that could be attained was approximately thirteen gallons per minute. Thus, the flow rates used for calibration ranged from approximately two gallons per minute to thirteen.
Table XX: Flow meter Calibration Data

<table>
<thead>
<tr>
<th>Kolbold Flow Rate (gpm)</th>
<th>Volume (L)</th>
<th>Time (sec)</th>
<th>Output (volts)</th>
<th>Measured Flow Rate (gpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.17</td>
<td>6</td>
<td>47.6</td>
<td>0.38</td>
<td>2.00</td>
</tr>
<tr>
<td>3.75</td>
<td>6</td>
<td>26.7</td>
<td>0.67</td>
<td>3.56</td>
</tr>
<tr>
<td>4.4</td>
<td>6</td>
<td>21.8</td>
<td>0.81</td>
<td>4.36</td>
</tr>
<tr>
<td>6.82</td>
<td>6</td>
<td>14.5</td>
<td>1.28</td>
<td>6.56</td>
</tr>
<tr>
<td>7.2</td>
<td>6</td>
<td>13.5</td>
<td>1.31</td>
<td>7.05</td>
</tr>
<tr>
<td>9.5</td>
<td>6</td>
<td>10.3</td>
<td>1.78</td>
<td>9.23</td>
</tr>
<tr>
<td>10.3</td>
<td>6</td>
<td>9.5</td>
<td>1.95</td>
<td>10.01</td>
</tr>
<tr>
<td>12.8</td>
<td>6</td>
<td>7.6</td>
<td>2.34</td>
<td>12.51</td>
</tr>
</tbody>
</table>

In Table XX, the “Kobold Flow Rate” refers to the flow rate indicated on the flow meter itself. Calibration coefficients were determined for both the measured flow rate and the Kobold flow rate.

Figure XLI: Flow Rate Calibration Scatters
Table XXI: Calibration Coefficients for Kobold Flow Meter

<table>
<thead>
<tr>
<th>Flow Rate</th>
<th>A</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured</td>
<td>5.23</td>
<td>0.029</td>
</tr>
<tr>
<td>Kobold</td>
<td>5.32</td>
<td>0.117</td>
</tr>
</tbody>
</table>

Note the “measured” coefficients were used in the final VI.

4.5. Airgas Analog Pressure Gauge

As was described in a previous section, the analog pressure gauge was a vital component to the system apparatus during the refill process. When the primary working fluid tank needed to be refilled, the valve was opened to the secondary section that facilitated the refill process. The analog pressure gauge was vital to the refill process because it allowed the user to visually determine the pressure in the secondary tank. If the refill process was allowed to occur too rapidly, the secondary tank’s temperature (and pressure) would drop, aborting the refill process.

The method in which the analog pressure gauge was calibrated was exactly the same as for the pressure transducers, except the data was not being taken using LabVIEW but rather manually. The intervals for the dead weight tester were the same for the analog gauge as for the transducers. Unlike the pressure transducers where the pressure was correlated to a voltage, for the analog gauge, the pressure created by the dead weight tester was correlated to a pressure reading in psig. The data taken as well as the calibration curve and calibration coefficients are shown in Table XXII.
Table XXII: Calibration Data for Pressure Gauge

<table>
<thead>
<tr>
<th>Pressure (psig)</th>
<th>PG Output (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>200</td>
<td>200</td>
</tr>
<tr>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>400</td>
<td>400</td>
</tr>
<tr>
<td>500</td>
<td>500</td>
</tr>
<tr>
<td>600</td>
<td>600</td>
</tr>
<tr>
<td>680</td>
<td>680</td>
</tr>
</tbody>
</table>

Overall, the objective of calibrating the measurement devices was a success.

Figure XLII: Pressure Transducer #2 Calibration Curve

Table XXIII: Pressure Transducer Calibration Coefficients

<table>
<thead>
<tr>
<th>Pressure Gauge</th>
<th>$A$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PG</td>
<td>1</td>
<td>0</td>
</tr>
</tbody>
</table>

Overall, the objective of calibrating the measurement devices was a success.
Chapter 5: Experimental Procedure and Safety Precautions

Now, it is important to emphasize the notion of safety since the experiment involved the handling of high pressure gas cylinders and equipment. This Chapter discusses the experiment procedure and everything related to safety regarding the safe handling of the high pressure equipment used in this study. In addition, the specific procedures for the refilling of the working fluid will be discussed since, as mentioned previously, this was the most likely point at which an incident may occur.

The processes and procedures outlined here applied to all individuals involved with this experiment, including faculty, staff, and students[^22].

All persons on campus property in the position to handle compressed gas equipment should only partake if he/she is familiar with the hazards associated with compressed gas equipment as well as are trained in the proper handling techniques. Compressed gas equipment is often heavy and difficult to transport. Improperly handling of compressed gas equipment can lead to serious physical injury to the user and other individuals that may be in the vicinity. Equipment containing pressurized gas poses a serious risk to health due to the fact that said equipment has a high potential to explode if improperly handled. In addition, depending on the nature of the compressed gas contained, sudden exposure of gas to atmospheric conditions can cause blindness, deafness, disfigurement, or death. Following all safe handling processes is necessary and procedures to maintain safe working conditions.

[^22]:
The following provides a brief overview of the experimental procedures completed for each test run. More detailed procedures, guidelines, and tips can be examined in Appendix E.

1. Leak Test/Nitrogen Purge
   a. Acquire & use appropriate safety equipment.
   b. Properly transport nitrogen tank to test cell and connect to system apparatus at the secondary section.
   c. Pressurize system with nitrogen and apply soap-water solution to connections to locate any leaks present.
   d. Depressurize system, address leaks, and retest system. Note to store nitrogen appropriately when not in use.

2. System Prime
   a. Properly transport secondary working fluid tank to test cell and connect to system apparatus at secondary section.
   b. Pressurize secondary section & separation section with working fluid, allowing thermal equilibrium to be established before priming primary section.
   c. Maintain a pressure below the system cracking pressure (1,000 psig); however, above operating pressure (500 psig) using the control fluid (i.e. to control temperature/pressure of secondary working fluid tank).
   d. Disconnect tubing to secondary fluid tank and fill the secondary fluid tank with water via the back fill valve until water comes out of tank.
   e. Close valve to secondary fluid source and open valve to flow meter.
f. Pressurize primary section then isolate from the other sections of the system (separation and secondary section), allowing thermal equilibrium to be established.

g. Open process section valve. Slowly crack primary section valve to allow the working fluid to purge the process section of air.

h. Reconnect tubing to secondary fluid tank while simultaneously closing primary section valve.

i. Close solenoid valve.

j. Bleed excess working fluid from secondary section and part of separation section.

3. System Activation

a. Open primary section valve

b. From the control room, activate measurement system and verify measurements are being taken properly.

c. Activate process section valve.

d. Monitor characteristics of system process throughout test run.

e. Close process section valve when the predicted time has been reached for expelling the specified volume of secondary fluid.

f. System characteristics should reveal when the secondary fluid has been exhausted.

4. System Conclusion

a. Deactivate measurement system.

b. Reenter test cell and close primary section valves open separation valve.
c. Vent residual vapor in system.

d. Make appropriate measurements & refill secondary fluid tank.

e. If enough working fluid remains, repeat experiment, otherwise, vent residual vapor in entire system.

f. Disconnect secondary working fluid tank from secondary section and replace with replenished tank.

g. If test runs are complete, disconnect and store system apparatus appropriately per safe handling processes and procedures.
Chapter 6: Results & Discussion

The following section presents the results of the experiment and subsequently provides some discussion giving explanation to what was observed.

Recall the ultimate goal of this study was determine the feasibility of using temperature to regulate the pressure of a two phase fluid close to its critical point to create a maintainable pressure differential. This was done by regulating the output flow of the secondary fluid by varying the size of an orifice the secondary fluid must pass through.

The orifice diameters tested were sized at $\frac{1}{32}$", $\frac{1}{16}$", $\frac{1}{8}$", and $\frac{3}{16}$". These diameters were selected based on the predictive calculations which indicated that these sizes were deemed the most likely to prove the feasibility of this temperature controlled process.

Each orifice was tested two times except the $\frac{1}{16}$" orifice diameter, which was tested five times. This orifice was tested more than twice in an attempt to establish steady state conditions where the pressure of the delivered vapor could be considered constant or changed very little over time.

In total, eleven tests were conducted and all tests had an operating time between twenty-five and one hundred and twenty seconds. The duration of the test depended mostly on the trends in the data observed by the operators. If it was estimated that a constant output pressure was attainable, the tests were extended. After each test was completed, it was verified that the pressure in the working fluid tank returned to its starting pressure of 505.56 psig, or the saturation pressure of carbon dioxide at 32°F. This return to initial pressure confirmed that the liquid still remained in the working fluid tank. Below, in Table XXIV, the durations of each test can be examined.
### Table XXIV: Test Durations

<table>
<thead>
<tr>
<th>Test Run</th>
<th>Test Duration (seconds)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test #1 (\frac{1}{32})&quot;</td>
<td>115</td>
</tr>
<tr>
<td>Test #2 (\frac{1}{32})&quot;</td>
<td>120</td>
</tr>
<tr>
<td>Test #1 (\frac{1}{16})&quot;</td>
<td>30</td>
</tr>
<tr>
<td>Test #2 (\frac{1}{16})&quot;</td>
<td>30</td>
</tr>
<tr>
<td>Test #3 (\frac{1}{16})&quot;</td>
<td>30</td>
</tr>
<tr>
<td>Test #4 (\frac{1}{16})&quot;</td>
<td>140</td>
</tr>
<tr>
<td>Tests #5 (\frac{1}{16})&quot;</td>
<td>115</td>
</tr>
<tr>
<td>Test #1 (\frac{1}{8})&quot;</td>
<td>25</td>
</tr>
<tr>
<td>Test #2 (\frac{1}{8})&quot;</td>
<td>40</td>
</tr>
<tr>
<td>Test #1 (\frac{3}{16})&quot;</td>
<td>25</td>
</tr>
<tr>
<td>Test #2 (\frac{3}{16})&quot;</td>
<td>25</td>
</tr>
</tbody>
</table>

The goal of the experiment was to gather specific data to show a constant vapor pressure could be maintained during the purge process. The following measurements were made to support this goal.

- Pressure in the working fluid tank
- Pressure in the secondary fluid tank
- Temperature along the height of the working fluid tank (tank bottom, 5-inches from bottom, 10-inches from bottom, tank top).
- Temperature directly outside of working fluid tank (before valve manifold)
- Temperature directly downstream of valve manifold.
• Temperature directly outside of secondary fluid tank.
• Flow rate of secondary fluid (downstream of orifice plate).

Note that pressure and flow rate were the most important parameters to be measured in these experiments. The temperatures in the working fluid tank were also important for understanding the vaporization process within in the tank.

Recall that the predicted operating flow results were estimated based on a Bernoulli equation analysis of the secondary fluid tank assuming that the CO₂ saturation pressure at 32°F was imposed on the tank. The measured flow rates were significantly less than predicted values. This was due to two effects.

First, the highest value of secondary fluid tank pressure, which occurred shortly after the test run was initialized by opening the solenoid valve, was never near the saturation pressure at 32°F (506 psig). Values of maximum secondary fluid tank pressure are shown in Table XXV. The difference between this maximum and the initial working fluid tank pressure is due to a sudden drop in working fluid tank pressure at the start of the run, as well as pressure drops in the manifold and tubing. These phenomena will be shown in figures presented subsequently.
Table XXV: Starting Pressure Comparison

<table>
<thead>
<tr>
<th>Test Run</th>
<th>Predicted Pressure (psig)</th>
<th>Actual Pressure (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test#1 $\frac{1}{32}$&quot;</td>
<td>505.56</td>
<td>468.94</td>
</tr>
<tr>
<td>Test#2 $\frac{1}{32}$&quot;</td>
<td>505.56</td>
<td>468.94</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{16}$&quot;</td>
<td>505.56</td>
<td>480.17</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{16}$&quot;</td>
<td>505.56</td>
<td>462.2</td>
</tr>
<tr>
<td>Test #3 $\frac{1}{16}$&quot;</td>
<td>505.56</td>
<td>457.71</td>
</tr>
<tr>
<td>Test #4 $\frac{1}{16}$&quot;</td>
<td>505.56</td>
<td>460</td>
</tr>
<tr>
<td>Tests #5 $\frac{1}{16}$&quot;</td>
<td>505.56</td>
<td>467.7</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{8}$&quot;</td>
<td>505.56</td>
<td>448.73</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{8}$&quot;</td>
<td>505.56</td>
<td>446.48</td>
</tr>
<tr>
<td>Test #1 $\frac{3}{16}$&quot;</td>
<td>505.56</td>
<td>435.25</td>
</tr>
<tr>
<td>Test #2 $\frac{3}{16}$&quot;</td>
<td>505.56</td>
<td>417.28</td>
</tr>
</tbody>
</table>

The second effect is due to the frictional losses in the secondary tank plumbing, especially the orifice plate. A comparison of predicted and maximum measured flow rates (at the beginning of the test run) are shown in Table XXVI. The reduction in actual flow rates from predicted flow rate values also affects other predicted quantities, such as heat flux and tubing pressure losses. It also required the use of a more sensitive flow meter for the smallest orifice size, since the lowest flows were below the operable range of the meter shown in Figure XIX.
Table XXVI: Starting Flow rate Comparison

<table>
<thead>
<tr>
<th>Test Run</th>
<th>Predicted Flow rate (gpm)</th>
<th>Actual Flow rate (gpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test#1 $\frac{1}{32}$&quot;</td>
<td>0.645</td>
<td>0.45</td>
</tr>
<tr>
<td>Test#2 $\frac{1}{32}$&quot;</td>
<td>0.645</td>
<td>0.45</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{16}$&quot;</td>
<td>2.581</td>
<td>1.31</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{16}$&quot;</td>
<td>2.581</td>
<td>1.33</td>
</tr>
<tr>
<td>Test #3 $\frac{1}{16}$&quot;</td>
<td>2.581</td>
<td>1.3</td>
</tr>
<tr>
<td>Test #4 $\frac{1}{16}$&quot;</td>
<td>2.581</td>
<td>1.32</td>
</tr>
<tr>
<td>Tests #5 $\frac{1}{16}$&quot;</td>
<td>2.581</td>
<td>1.6</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{8}$&quot;</td>
<td>10.325</td>
<td>4.58</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{8}$&quot;</td>
<td>10.325</td>
<td>6.14</td>
</tr>
<tr>
<td>Test #1 $\frac{3}{16}$&quot;</td>
<td>23.323</td>
<td>10.78</td>
</tr>
<tr>
<td>Test #2 $\frac{3}{16}$&quot;</td>
<td>23.323</td>
<td>10.78</td>
</tr>
</tbody>
</table>

Figures XLIII to XLVI show the measured pressure and flow rates for the four orifice diameters. Note that only one plot is shown for each orifice diameter, however the remainder of the plots can be examined in Appendix B. The two pressures shown in each plot were measured at the working fluid tank outlet and the secondary fluid tank inlet.
Figure XLIII: System Pressures & Flow for $\frac{1}{32}$" Orifice Plate

Figure XLIV: System Pressures & Flow for $\frac{1}{16}$" Orifice Plate
Figure XLV: Pressures & Flow for $\frac{1}{8}$" Orifice Plate

Figure XLVI: Pressures & Flow for $\frac{3}{16}$" Orifice Plate
Figure XLIII above depicts the result for the $\frac{1}{32}$" orifice. After the initial transient is over, it is clear that the flow rate is nearly constant for the 120 second duration of the run. The two pressures are nearly the same, indicating very little pressure drop due to tubing and valve losses. The drop with pressure with time is fairly small, approximately 40 psig over the run duration. Note the separation of the two pressure curves towards the end of the test. This point just represents the moment at which the solenoid (process) valve was closed and conditions began to return to starting conditions. Measurements after this point represent the gradual blow down of the system and are of lesser importance.

It is interesting to note the trend of the pressure curves at the initial activation of the flow. As Figure XLIII shows, the pressure of the working fluid quickly dips down as soon as the solenoid valve closes. The pressure of the secondary tank quickly rises to meet the falling working fluid tank pressure. Notice then that the pressure in both tanks climbs to a local maximum then begins to settle at a slightly lower value. The flow rate rapidly stabilizes after the solenoid valve opens and remains steady until the valve closes. It should be noted that a more sensitive flow meter was used for this orifice size then for the remainder of the tests.

Figure XLIV depicts the data for the $\frac{1}{16}$" diameter orifice, and it can be seen that the flow rate has increased to about 1.3 gpm. Figure XLIV reveals the increased frictional losses associated with the system as the flow rate increases, as there is a discernible pressure difference between the two tanks for this orifice size. The overall pressure drop with time across the duration of the test is nearly double that of the $\frac{1}{32}$"
case, although the flow rate is quite steady. It is apparent that a steady pressure is not maintained for this test, and this trend will be shown to worsen as the orifice size increases. As mentioned previously, this is the orifice size that was tested more than twice in an attempt to gain more steady results. Despite the number of tests, or duration, steady results could not be produced and the trends shown in Figure XLIV remained similar.

The delay in the rise of the measured flow rate with respect to the valve opening time was pronounced for this orifice. This was due to the fact that the flow meter used for the orifice diameters equal and greater to \( \frac{1}{16} \)" could only read flow rates greater than approximately 1.3 gpm. Thus the flow rate for the \( \frac{1}{16} \)" tests were right at the minimum of the operable range of the flow meter.

The trends shown in the \( \frac{1}{16} \)" results become more pronounced at larger orifice sizes and higher flow rates. Figures XLV and XLVI show the results for the \( \frac{1}{8} \)" and \( \frac{3}{16} \)" orifices. These cases show the increased pressure drop with time, an inability to maintain a constant flow rate, and increased pressure drop between the two tanks. It is clear that the required heat flux to vaporize the working fluid requires a significant temperature difference between the constant temperature ice bath and the working fluid.

It seemed interesting to look at the pressure drop associated with flowing through the approximate ten foot section of half inch tubing connecting the process valve and the secondary fluid tank and compare that to the pressure drop seen in the Figures XLIII – XLVI. The calculation was completed for the flow rates determined by the predictive
calculations as well as for the flow rates seen in the actual tests. The known parameters were constant for both cases despite the fact that upstream pressure of the actual case was far less than that of the predicted case in some of the tests. This had a minimal effect on the pressure calculations; however, since the pressure loss calculated was small compared to the pressure loss seen in the test. Table XXVII shows the known parameters used in the calculation of the pressure loss.

**Table XXVII: Known Parameters used for Pressure Loss in Pipe Calculation**

<table>
<thead>
<tr>
<th>Name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upstream Pressure</td>
<td>505.56</td>
<td>psig</td>
</tr>
<tr>
<td>Diameter</td>
<td>0.5</td>
<td>in</td>
</tr>
<tr>
<td>Roughness[^34^]</td>
<td>0.00000467</td>
<td>in/in</td>
</tr>
<tr>
<td>Length</td>
<td>10</td>
<td>ft</td>
</tr>
<tr>
<td>Density[^12^]</td>
<td>6.09</td>
<td>lb/m³</td>
</tr>
<tr>
<td>Dynamic Viscosity[^51^]</td>
<td>2.15×10⁻⁹</td>
<td>lb/ft²</td>
</tr>
</tbody>
</table>

Note that this table depicts the estimated values meant for calculating the pressure loss associated with the working fluid travelling through the tubing in the system, or the major losses. Table XXVIII shows the results of the calculations for both the predicted flow rates and adjusted values for the actual measured flows. Also shown is the average measured pressure difference between the two tanks, which included losses across the valve manifold and the solenoid valve. It is clear that the pressure drop in the tubing was negligible compared to the valve losses.
Table XXVIII: Pressure Drop Calculation in Tubing

<table>
<thead>
<tr>
<th>Orifice Diameter (in)</th>
<th>$\Delta P_{tube}$(psig)</th>
<th>$\Delta P_{tube\ adjusted}$(psig)</th>
<th>$\Delta P_{actual}$(psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.03125</td>
<td>0.0948</td>
<td>0.0657</td>
<td>10.5</td>
</tr>
<tr>
<td>0.0625</td>
<td>0.113</td>
<td>0.0786</td>
<td>12.4</td>
</tr>
<tr>
<td>0.125</td>
<td>0.818</td>
<td>0.249</td>
<td>25</td>
</tr>
<tr>
<td>0.1875</td>
<td>2.89</td>
<td>0.723</td>
<td>61.5</td>
</tr>
</tbody>
</table>

It is clear from the preceding results for tank pressure versus time that the ability to vaporize at a constant pressure is limited by the heat transfer process in the working fluid tank. Based on the predictive calculations it was expected that nucleate pool boiling would be the mode of heat transfer, possibly reaching critical heat flux at higher flow rates. The experimental results establish a value of heat flux that can be achieved while maintaining a controllable temperature (and pressure) in the working fluid tank.

Table XXIX below shows the predictive and actual heat flux as compared to the CHF used in this study.

Table XXIX: Heat Transfer Comparison

<table>
<thead>
<tr>
<th>Orifice Diameter (in)</th>
<th>$q''$ Predicted ($\frac{BTU}{sec\cdot ft^2}$)</th>
<th>$q''$ Actual ($\frac{BTU}{sec\cdot ft^2}$)</th>
<th>CHF ($\frac{BTU}{sec\cdot ft^2}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.03125</td>
<td>4.82</td>
<td>4.43</td>
<td></td>
</tr>
<tr>
<td>0.0625</td>
<td>19.29</td>
<td>13.72</td>
<td></td>
</tr>
<tr>
<td>0.125</td>
<td>77.16</td>
<td>17.57</td>
<td></td>
</tr>
<tr>
<td>0.1875</td>
<td>173.62</td>
<td>22.28</td>
<td>81.53</td>
</tr>
</tbody>
</table>
Table XXIX shows the original predicted values of heat flux, based on the first law balance explained in Chapter 2, as well as corrected values which account for actual state values and measured flow rates from the experimental data. As expected, due to the lower flow rates, the heat fluxes are somewhat reduced. Based on the estimate of critical heat flux, these conditions should all fall in the nucleate pool boiling range, although the temperature difference required to drive the heat transfer is too high for effective control of the delivered vapor pressure, except at the lowest flow rates tested.

Additional insight into the vaporization process can be seen in Figures XLVII to L. These figures show temperature at four vertical locations in the working fluid tank, for the four different orifice sizes. Details on the temperature probe are given in Chapter 3, Table VIII. Also shown on the plots is the saturation temperature (labeled as ideal temperature) evaluated at the measured tank outlet pressure.
Figure XLVII: Working Fluid Tank Temperatures for the $\frac{1}{32}$" Orifice Plate

Figure XLVIII: Working Fluid Tank Temperatures for the $\frac{1}{16}$" Orifice Plate
Figure XLIX: Working Fluid Tank Temperatures for the \( \frac{1}{8} '' \) Orifice Plate

Figure L: Working Fluid Tank Temperatures for the \( \frac{3}{16} '' \) Orifice Plate
The following general observations can be made:

- The temperatures drop with time, but approach a steady value for the two smallest orifice sizes
- The temperatures are lower at the top of the tank than at the bottom.
- The temperature near the bottom of the tank is very noisy, making it difficult to interpret.
- The calculated ideal temperature closely tracks the trends of the measured temperatures. It tends to become lower than the measured values at higher orifice sizes.

The first observation, of course, is consistent with the falling pressures seen in the preceding plots. For the smaller orifice sizes, one can conclude that the temperature has dropped sufficiently to support the heat transfer rate required to maintain a nearly steady downstream vapor pressure.

The second observation is consistent with expectations for a pool boiling situation where heat is added at the bottom. Some of the bubbles that are created at the surface will condense as they move upward, which along with the natural convection currents in the liquid phase, will maintain a slightly negative temperature gradient in the vertical direction. In this case, heat is also being added at the sides, so one would expect an overall temperature variation similar to that shown in Figure XLI. However, all of the measured temperatures are along the centerline of the tank, so there is no data to quantify radial differences in temperature.
Figure LI: Temperature Variations in Working Fluid Tank

It was originally thought that these temperature measurements might give an indication regarding the changing liquid level in the working fluid tank, i.e. there might be a sharp change with time as the liquid level passed a measurement location. However, there are no obvious trends in the data to support this notion.

The third observation may be due to a couple of causes. One is that this thermocouple is surrounded by turbulent vapor traffic since it is close to the bottom of the tank. Another is that it may be touching the bottom of the tank, and may be picking up thermal or electrical noise due to this contact. This may be easily confirmed by additional testing, but due to time limitations this was not explored further.

The ideal temperature, based on the measured outlet pressure of the tank, was calculated and plotted as a check on the measured values. The close matching of the trends with time supports the accuracy of the measured values, as well as the lack of any time delay due to time constant effects. It is interesting that the ideal temperature tends
to become lower, relative to the other values, as orifice size and flow rate increase. It is believed that this might be due to the increased pressure drop between the inside of the tank and the outlet point at which pressure is measured. If the pressure being used to calculate the temperature is in fact too low, then the resulting temperature will tend to be lower as well.

Now that the major results have been presented, one can attempt to see if the values could be estimated from thermodynamic and heat transfer relationships. If estimates are fairly accurate, it would support the notion that a two phase vapor production system could be designed to meet desired specifications.

### Table XXX: Temperature Drop Comparison

<table>
<thead>
<tr>
<th>Orifice Dia. (in)</th>
<th>$\Delta T_{\text{meas}}(R)$ ($T_{\text{wall}} - T_{\text{sat}}$)</th>
<th>$\Delta T_{\text{wall_conduction}}(R)$ ($T_{s,\text{outside}} - T_{s,\text{inside}}$)</th>
<th>$\Delta T_{\text{meas._Corrected}}(R)$</th>
<th>$\Delta T_{\text{rohsenow}}(R)$ ($T_{\text{wall}} - T_{\text{sat}}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.03125</td>
<td>13.4</td>
<td>0.4</td>
<td>13.0</td>
<td>5.2</td>
</tr>
<tr>
<td>0.0625</td>
<td>19.0</td>
<td>1.0</td>
<td>17.9</td>
<td>7.7</td>
</tr>
</tbody>
</table>

Knowing the heat flux based on measured parameters, Rohsenow’s nucleate boiling correlation can be used to estimate a saturation temperature, based on a wall temperature of zero degrees Celsius, or thirty two degrees Fahrenheit. This delta temperature can be compared to the measured value of working fluid temperature. This calculation is only presented for the two smallest orifice sizes, since they are the only cases that approached a steady condition. The results are presented in Table XXX. The values show that the predicted temperature difference is significantly lower than the measured value. One possible difference between the values is the neglect of the
temperature drop across the \(\frac{1}{4}\)" thick aluminum tank wall. In reality the outside of the wall is at 32°F and thus the wall temperature exposed to the fluid is somewhat lower. Using the values of heat flux and assuming steady conduction across the tank wall, the temperature drop across the wall for the two cases is calculated and is also shown in Table XXX. The temperature change due to conduction does not appear to have significant effects on the actual temperature indicating that the difference seen above result from the uncertainties present in Rohsenow’s correlation.

One of the last trends examined in this experiment was the system temperatures as a function of time. The system temperatures refer to the temperatures measured outside of the working fluid tank; recall, this is the temperature at the exit the working fluid tank, the temperature just downstream of the valve manifold, and the temperature at the entrance to the secondary fluid tank. Figures LII-LV show the trends seen for the system temperatures for the four orifice sizes.
Figure LII: System Temperatures for $\frac{1}{32}$" Orifice Plate

Figure LIII: System Temperatures for $\frac{1}{16}$" Orifice Plate
Figure LIv: System Temperatures for $\frac{1}{8}$" Orifice Plate

Figure LV: System Temperatures for $\frac{3}{16}$" Orifice Plate
Examining Figure LII shows that steady temperatures were produced and maintained for a majority of the operation time of the system. This was expected since Figure XLIII had shown steady conditions for this orifice size. Figure XLIII and LII solidify the fact that producing a steady vapor supply with a two phase system is possible, at sufficiently low flow rates.

A somewhat unexpected result was that, by the time it travels to the secondary fluid tank, the temperature of the working fluid is nearly at room temperature \(71.6^\circ F \) \(22^\circ C\). In fact the temperature of the working fluid increases to values much higher than the tank temperature right outside of the tank and even higher just beyond the valve manifold.

The general trends for the system temperatures were similar for all of the flow rates. Examination of Figures LII to LV reveals the following:

- As the flow rate increases, the rate of drop in temperatures with respect to time increases, and steady values are not achieved during the run
- As the flow rate increases, the temperature change across the valve manifold decreases
- As the flow rate increases, the temperature of the fluid entering the secondary fluid tank decreases

All of these trends are consistent with heat transfer from the environment to the fluid along the tubing and in the valve manifold and associated fittings. Based on these results, it appears that more emphasis should have been placed on insulating the tubing,
as well as pre-chilling the plumbing before each run. On the other hand, the relative change in absolute temperature is small on a percentage basis and has a relatively small effect on fluid density and pressure. While insulation will be considered for future experiments, the effect of heat transfer along the tubing does not influence any of the key conclusions of this study.
Chapter 7: Conclusions

An experimental study was performed to prove the feasibility of a vapor production process. The goal of the process is to produce a vapor flow at a pressure which is controlled by the working fluid tank temperature, rather than by using traditional mechanical pressure regulators. The primary purpose of such a vapor flow would be to purge liquid from a tank by displacing it, as was done in this experiment. However, there are numerous other applications that might be suitable.

Carbon dioxide was used as the working fluid, due to its favorable thermodynamic properties, availability, and safe handling. An experimental apparatus was designed and built in the rocket test cell at California State University, Northridge. The temperature of the working fluid tank was controlled to be 32°F by submerging it in an ice bath. The fuel tank for CSUN’s old rocket engine experiment was adapted to use as the secondary fluid tank, which held water to be purged by the CO₂ vapor flow. A plumbing system was designed to handle the high pressure CO₂ vapor flow from the working fluid tank to the secondary fluid tank, and accommodate sensors for pressure and temperature measurement. Downstream of the secondary fluid tank, a flow meter was located to measure the water flow rate as it is purged from the tank. The flow rate was controlled with interchangeable orifice plates of different sizes, located at the exit of the secondary fluid tank.

The system was sized and designed using basic equations from thermodynamics, fluid mechanics, and heat transfer. The Bernoulli equation was used to choose orifice sizes that would produce flow rates in the desired range. A first law analysis of the vaporization process in the working fluid tank was used to estimate the necessary heat
transfer rate to produce a steady vapor flow. Heat transfer correlations for boiling heat transfer were used to predict the temperature drop in the tank to produce the necessary heat flux. These predictions were used as a benchmark for comparison with the experimental results. It was hoped that good agreement between predicted and measured results would support the notion that performance of this type of vapor production system could be determined from the analysis methodology presented here.

Four different orifice sizes were used for the experiments. Measured flows were roughly half of the estimated values, because of the frictional losses which were unaccounted for in the Bernoulli equation. It was found that vapor flows at a steady pressure could not be produced with the two largest orifice sizes, due to the inability of the heat transfer rate to maintain a steady working fluid temperature. Measured temperature drops of the working fluid were found to significantly exceed values predicted by Rohsenow’s pool boiling correlation. More work is required to resolve differences between predicted and measured results.

On the positive side, good results were obtained for the two smallest orifice sizes. This is particularly true for the $\frac{1}{32}$" orifice, which was able to produce approximately 0.45 gpm of vapor flow at a nearly steady pressure of 425 psi. Keeping in mind that the saturation pressure at the controlled temperature of 32°F is approximately 500 psi, there is still a significant drop between the pressure produced and the “set” pressure based on temperature.

This fact leads to the key recommendation based on the results of this study. The lowest flow rates examined here are at the upper limit of the system’s ability to regulate
properly. A more complete understanding of the operable range of this system should be determined by using smaller orifice sizes to collect data at lower flow rates. Experiments run at lower flow rates should allow a true steady state to be achieved, and this data should help to resolve the current disagreement between theoretical predictions and experimental data.

It is also recommended to insulate the tubing and plumbing connecting the two tanks. Measurements of temperature indicate that significant heat transfer was occurring, particularly at the lower flow rates. Although this heat transfer did not affect the working fluid tank conditions, and thus the pressure of produced vapor, insulation would provide more consistent and repeatable conditions and should be used for future experiments.

The design of the experiment apparatus and the instrumentation system was a success and provided the required data. All of the instrumentation was successfully calibrated and worked well throughout the experiments, with one exception. The thermocouple located near the bottom of the working fluid tank produced data that was significantly noisier than the other channels. Investigation of this problem should be completed before running additional tests.
References


Appendix A

The following section simply shows the component summary as it correlates to the system diagram as shown in figure II.

Table A-I: System Component Summary

<table>
<thead>
<tr>
<th>Diagram #</th>
<th>Part Description</th>
<th>Manufacturer</th>
<th>Part #</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Primary Control Fluid Tank</td>
<td>CSUN Fab Shop</td>
<td>N/A</td>
</tr>
<tr>
<td>2</td>
<td>Primary Working Fluid Tank</td>
<td>McMaster-Carr</td>
<td>7822A11</td>
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<td>3</td>
<td>3/4&quot; MNPT to 1/2&quot; MNPT Adapter</td>
<td>Swagelok</td>
<td>SS-12-HRN-8</td>
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<tr>
<td>4</td>
<td>1/2&quot; FNPT SS Tee</td>
<td>Swagelok</td>
<td>SS-8-T</td>
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<td>5</td>
<td>Multilevel Type J Thermocouple</td>
<td>ARI Industries, Inc.</td>
<td>T-50780.4-21DJ8A-1,6,11,16</td>
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<td>Swagelok</td>
<td>SS-8-MT</td>
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<tr>
<td>7</td>
<td>1/2&quot; FNPT to 1/4&quot; MNPT SS Reducing Adapter</td>
<td>Swagelok</td>
<td>SS-8-RA-4</td>
</tr>
<tr>
<td>8</td>
<td>1,000 PSI Pressure Transducer</td>
<td>Omega Engineering, Inc.</td>
<td>MMG1.0KV1P4D0T3A5CE</td>
</tr>
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<td>9</td>
<td>1/2&quot; FNPT SS Union Ball Joint</td>
<td>Swagelok</td>
<td>SS-8-UBJ</td>
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<td>10</td>
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<td>Swagelok</td>
<td>SS-8-MT</td>
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<td>Swagelok</td>
<td>SS-8-HCG</td>
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<td>High Pressure 1/8&quot; TC Probe Adapter (15,000 psi)</td>
<td>High Pressure Equipment Company</td>
<td>15-21AF2NMD-T</td>
</tr>
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<td>Type J TC with Molded Connector</td>
<td>Omega Engineering, Inc.</td>
<td>JQSS-18G-6</td>
</tr>
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<td>3-Way 1/2&quot; FNPT SS Instrument Valve (6000 psi)</td>
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<td>SS-V3NBFl</td>
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<td>Swagelok</td>
<td>SS-8-MT</td>
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<td>16</td>
<td>1/2&quot; FNPT SS Hex Coupler</td>
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<td>SS-8-HCG</td>
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<tr>
<td>17</td>
<td>High Pressure 1/8&quot; TC Probe Adapter (15,000 psi)</td>
<td>High Pressure Equipment Company</td>
<td>15-21AF2NMD-T</td>
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<td>Type J TC with Molded Connector</td>
<td>Omega Engineering, Inc.</td>
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<td>SS-8-UBJ</td>
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<td>SS-8-HRN-4</td>
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<td>90° Full Port Solenoid Valve w/ 1/4&quot; FNPT Connections</td>
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<td>1/4&quot; MNPT - FNPT SS Elbow</td>
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<td>Description</td>
<td>Manufacturer/Model</td>
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<td>-----</td>
<td>-----------------------------------------------------------------------------</td>
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<td>Swagelok SS-T8-S-035-6ME</td>
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<td>36</td>
<td>1/2&quot; FJIC SS Braided Hose</td>
<td>OEM</td>
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<td>37</td>
<td>JIC Siphon</td>
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<td>1/2&quot; FNPT SS Restriction Orifice Plate Union w/ 1/32&quot; Plate (3000 psig)</td>
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<td>Model/Part No.</td>
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<td>1 1/2&quot; PVC Tubing Coupler</td>
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<td>N/A</td>
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<td>3.5' of 1 1/2&quot; PVC Tubing</td>
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<td>1 1/2&quot; PVC 90° Elbow</td>
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<td>1' of 1 1/2&quot; PVC Tubing</td>
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<td>1 1/2&quot; PVC 90° Elbow</td>
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<tr>
<td>53</td>
<td>1/2&quot; MNPT to 3/8&quot; Garden Hose Barb PVC Adapter</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>54</td>
<td>10' of 3/8&quot; Garden Hose</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>55</td>
<td>3/8&quot; Garden Hose Y-Valve</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>56</td>
<td>10' of 3/8&quot; Garden Hose from Secondary Fluid Source</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>57</td>
<td>10' of 3/8&quot; Garden Hose</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>58</td>
<td>1/2&quot; MNPT to 3/8&quot; Garden Hose Barb PVC Adapter</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>59</td>
<td>1/2&quot; FNPT to 1&quot; PVC Tubing Adapter</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>60</td>
<td>1' of 1&quot; PVC Tubing</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>61</td>
<td>Flow meter 1.3 - 26 gpm</td>
<td>Kobald</td>
<td>MIK-SNAUBMC34P</td>
</tr>
<tr>
<td>62</td>
<td>1' of 1&quot; PVC Tubing</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>63</td>
<td>1/2&quot; FNPT to 1&quot; PVC Tubing Adapter</td>
<td>Home Depot</td>
<td>N/A</td>
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<tr>
<td>64</td>
<td>1/2&quot; MNPT to 3/8&quot; Garden Hose Barb PVC Adapter</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>65</td>
<td>5' of 3/8&quot; Garden Hose</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>66</td>
<td>Secondary Fluid Catch Tank</td>
<td>Home Depot</td>
<td>N/A</td>
</tr>
<tr>
<td>67</td>
<td>Secondary Control Fluid Tank</td>
<td>CSUN Fab Shop</td>
<td>N/A</td>
</tr>
<tr>
<td>68</td>
<td>Secondary Working Fluid Tank</td>
<td>McMaster-Carr</td>
<td>7822A11</td>
</tr>
<tr>
<td>69</td>
<td>CGA 580 Gate Valve</td>
<td>McMaster-Carr</td>
<td>As Part of 7822A11</td>
</tr>
<tr>
<td>70</td>
<td>CGA 580 to 1/4&quot; MNPT Brass Nipple</td>
<td>Cramer-Decker</td>
<td>NP188</td>
</tr>
<tr>
<td>71</td>
<td>1/4&quot; FNPT to 1/2&quot; FNPT SS Coupler</td>
<td>Swagelok</td>
<td>SS-8-HRCG-4</td>
</tr>
<tr>
<td>72</td>
<td>1/2&quot; MNPT SS Tee</td>
<td>Swagelok</td>
<td>SS-8-MT</td>
</tr>
<tr>
<td>73</td>
<td>1/2&quot; FNPT to 1/4&quot; FNPT SS Reducing Adapter</td>
<td>Swagelok</td>
<td>SS-8-HRCG-4</td>
</tr>
<tr>
<td>74</td>
<td>Airgas® 2 1/2&quot; X 1/4&quot; Male NPT Connector 0 - 1000 PSI S.S. Gauge</td>
<td>Airgas</td>
<td>Y19425D10</td>
</tr>
</tbody>
</table>
### Table A-I Concluded: System Component Summary

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
<th>Manufacturer</th>
<th>Part Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>75</td>
<td>1/2” FNPT SS Union Ball Joint</td>
<td>Swagelok</td>
<td>SS-8-UBJ</td>
</tr>
<tr>
<td>76</td>
<td>1/2” MNPT SS Tee</td>
<td>Swagelok</td>
<td>SS-8-MT</td>
</tr>
<tr>
<td>77</td>
<td>1/2” FNPT SS Hex Coupler</td>
<td>Swagelok</td>
<td>SS-8-HCG</td>
</tr>
<tr>
<td>78</td>
<td>High Pressure 1/8” TC Probe Adapter (15,000 psi)</td>
<td>High Pressure Equipment Company</td>
<td>15-21AF2NMD-T</td>
</tr>
<tr>
<td>79</td>
<td>Type J TC with Molded Connector</td>
<td>Omega Engineering, Inc.</td>
<td>JQSS-18G-6</td>
</tr>
<tr>
<td>80</td>
<td>1/2” FNPT to 1/4” MNPT SS Reducing Adapter</td>
<td>Swagelok</td>
<td>SS-8-HRCG-4</td>
</tr>
<tr>
<td>81</td>
<td>18” of 1/4” FNPT SS Braided Hose</td>
<td>Cramer-Decker</td>
<td>CFHP18</td>
</tr>
<tr>
<td>82</td>
<td>1/2” MNPT to 1/4” MNPT Hex Adapter</td>
<td>Swagelok</td>
<td>SS-8-HRN-4</td>
</tr>
<tr>
<td>83</td>
<td>Stainless Steel 1/2” Mal NPT to 1/4” Female NPT Adapter</td>
<td>Swagelok</td>
<td>SS-8-RB-4</td>
</tr>
<tr>
<td>84</td>
<td>Manual Override 1/4” M/FNPT PRV</td>
<td>Swagelok</td>
<td>SS-4R3A5-MO-SETC</td>
</tr>
</tbody>
</table>
Appendix B

The following section simply shows the remaining results for the study. These were not included in the report for sake of brevity and relativity.

**Test#1: W.F. Tank Temperatures for 1/32" Orifice**

![Graph showing temperature changes over time for different tank temperatures](image)

**Figure B-I: Temperatures in the W.F.T. for Test #1 of 1/32" Orifice**

133
Test #1: System Pressure & Flow for 1/32" Orifice

Figure B-II: System Pressure & Flow for Test #1 of 1/32" Orifice

Test#1: System Temperatures for 1/32" Orifice

Figure B-III: System Temperatures for Test #1 of 1/32" Orifice
Figure B-IV: Temperatures in the W.F.T. for Test #1 of $\frac{1}{16}$" Orifice

Test#1: W.F. Tank Temperatures for 1/16" Orifice

Test #1: System Pressure & Flow for 1/16" Orifice

Figure B-V: System Pressure & Flow for Test #1 of $\frac{1}{16}$" Orifice
Test#1: System Temperatures for 1/16" Orifice

Figure B-VI: System Temperatures for Test #1 of $\frac{1}{16}$" Orifice

Test#2: W.F. Tank Temperatures for 1/16" Orifice

Figure B-VII: Temperatures in the W.F.T. for Test #2 of $\frac{1}{16}$" Orifice
Test #2: System Pressure & Flow for 1/16" Orifice

Figure B-VIII: System Pressure & Flow for Test #2 of $\frac{1}{16}$" Orifice

Test#2: System Temperatures for 1/16" Orifice

Figure B-IX: System Temperatures for Test #2 of $\frac{1}{16}$" Orifice
Test#4: W.F. Tank Temperatures for 1/16" Orifice

Figure B-X: Temperatures in the W.F.T. for Test #4 of 1/16" Orifice

Test #4: System Pressure & Flow for 1/16" Orifice

Figure B-XI: System Pressure & Flow for Test #4 of 1/16" Orifice
Figure B-XII: System Temperatures for Test #4 of $\frac{1}{16}$" Orifice

Figure B-XIII: Temperatures in the W.F.T. for Test #5 of $\frac{1}{16}$" Orifice

Test#4: System Temperatures for 1/16" Orifice

Test#5: W.F. Tank Temperatures for 1/16" Orifice
Figure B-XIV: System Temperatures for Test #5 of $\frac{1}{16}$" Orifice

Figure B-XV: System Temperatures for Test #5 of $\frac{1}{16}$" Orifice
Test#1: W.F. Tank Temperatures for 1/8"
Orifice

Figure B-XVI: Temperatures in the W.F.T. for Test #1 of \( \frac{1}{8}'' \) Orifice

Test #1: System Pressure & Flow for 1/8"

Figure B-XVII: System Pressure & Flow for Test #1 of \( \frac{1}{8}'' \) Orifice
Figure B-XVIII: System Temperatures for Test #1 of 1/8" Orifice

Figure B-XIX: System Temperatures for Test #2 of 3/16" Orifice
Figure B-XX: System Pressure & Flow for Test #2 of \( \frac{3}{16}'' \) Orifice

Test#2: System Temperatures for 3/16'' Orifice

Figure B-XXI: System Temperatures for Test #2 of \( \frac{3}{16}'' \) Orifice
Aside from the remaining charts, some additional data should be presented that was important to the results of this study.

**Table B-I: Start Temperature Data for Each Test**

<table>
<thead>
<tr>
<th>Test</th>
<th>TC #5 (°F)</th>
<th>TC #6 (°F)</th>
<th>TC #7 (°F)</th>
<th>TC #8 (°F)</th>
<th>T&lt;sub&gt;avg,start&lt;/sub&gt; (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test #1</td>
<td>28.31</td>
<td>29.57</td>
<td>29.696</td>
<td>31.622</td>
<td>29.804</td>
</tr>
<tr>
<td>Test #2</td>
<td>26.348</td>
<td>29.48</td>
<td>29.57</td>
<td>46.922</td>
<td>33.08</td>
</tr>
<tr>
<td>Test #3</td>
<td>25.286</td>
<td>28.49</td>
<td>30.992</td>
<td>43.052</td>
<td>31.964</td>
</tr>
<tr>
<td>Test #1</td>
<td>25.754</td>
<td>28.004</td>
<td>28.256</td>
<td>32.936</td>
<td>28.742</td>
</tr>
<tr>
<td>Test #2</td>
<td>24.872</td>
<td>27.716</td>
<td>28.472</td>
<td>31.784</td>
<td>28.22</td>
</tr>
<tr>
<td>Test #1</td>
<td>24.998</td>
<td>28.868</td>
<td>28.544</td>
<td>33.206</td>
<td>28.904</td>
</tr>
<tr>
<td>Test #4</td>
<td>24.386</td>
<td>28.328</td>
<td>28.202</td>
<td>31.01</td>
<td>27.986</td>
</tr>
<tr>
<td>Test #5</td>
<td>27.104</td>
<td>29.642</td>
<td>29.39</td>
<td>32.882</td>
<td>29.75</td>
</tr>
<tr>
<td>Test #1</td>
<td>24.728</td>
<td>28.13</td>
<td>28.112</td>
<td>32.162</td>
<td>28.274</td>
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<tr>
<td>Test #2</td>
<td>24.98</td>
<td>28.382</td>
<td>29.102</td>
<td>32.45</td>
<td>28.724</td>
</tr>
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</table>

**Table B-II: End Temperature Data for Each Test**

<table>
<thead>
<tr>
<th>Test</th>
<th>TC #5 (°F)</th>
<th>TC #6 (°F)</th>
<th>TC #7 (°F)</th>
<th>TC #8 (°F)</th>
<th>T&lt;sub&gt;avg,End&lt;/sub&gt; (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test #1</td>
<td>17.4</td>
<td>22.6</td>
<td>23.9</td>
<td>24.4</td>
<td>17.4</td>
</tr>
<tr>
<td>Test #2</td>
<td>18.3</td>
<td>23.3</td>
<td>25.0</td>
<td>26.6</td>
<td>18.3</td>
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<td>Test #3</td>
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<td>24.6</td>
<td>27.1</td>
<td>15.9</td>
</tr>
<tr>
<td>Test #1</td>
<td>10.3</td>
<td>16.6</td>
<td>21.1</td>
<td>17.9</td>
<td>10.3</td>
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<td>Test #2</td>
<td>-0.3</td>
<td>10.2</td>
<td>13.9</td>
<td>9.9</td>
<td>-0.3</td>
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<td>9.2</td>
<td>17.8</td>
<td>9.8</td>
<td>0.4</td>
</tr>
<tr>
<td>Test #2</td>
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<td>13.3</td>
<td>20.5</td>
<td>13.7</td>
<td>4.1</td>
</tr>
<tr>
<td>Test #4</td>
<td>2.9</td>
<td>7.6</td>
<td>12.1</td>
<td>19.8</td>
<td>2.9</td>
</tr>
<tr>
<td>Test #5</td>
<td>10.7</td>
<td>13.6</td>
<td>16.2</td>
<td>20.5</td>
<td>10.7</td>
</tr>
<tr>
<td>Test #1</td>
<td>18.2</td>
<td>20.8</td>
<td>22.2</td>
<td>26.5</td>
<td>18.2</td>
</tr>
<tr>
<td>Test #2</td>
<td>19.1</td>
<td>21.5</td>
<td>22.9</td>
<td>26.5</td>
<td>19.1</td>
</tr>
</tbody>
</table>
### Table B-III: Volumes & Flow rates

<table>
<thead>
<tr>
<th>Test</th>
<th>Measured S.F. Expelled Volume (gal)</th>
<th>Average Flow rate (gpm)</th>
<th>Calculated S.F. Expelled Volume (gal)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test #1 $\frac{1}{16}$</td>
<td>2.06</td>
<td>1.31</td>
<td>0.46</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{16}$</td>
<td>1.65</td>
<td>1.34</td>
<td>0.43</td>
</tr>
<tr>
<td>Test #3 $\frac{1}{16}$</td>
<td>1.60</td>
<td>1.31</td>
<td>0.54</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{8}$</td>
<td>3.70</td>
<td>5.00</td>
<td>1.47</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{8}$</td>
<td>4.94</td>
<td>5.08</td>
<td>2.91</td>
</tr>
<tr>
<td>Test #1 $\frac{3}{16}$</td>
<td>4.99</td>
<td>10.58</td>
<td>3.60</td>
</tr>
<tr>
<td>Test #2 $\frac{3}{16}$</td>
<td>4.99</td>
<td>10.59</td>
<td>2.77</td>
</tr>
<tr>
<td>Test #4 $\frac{1}{16}$</td>
<td>4.22</td>
<td>1.26</td>
<td>2.69</td>
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<tr>
<td>Test #5 $\frac{1}{16}$</td>
<td>2.51</td>
<td>1.47</td>
<td>2.72</td>
</tr>
<tr>
<td>Test #1 $\frac{1}{32}$</td>
<td>0.90</td>
<td>0.43</td>
<td>0.82</td>
</tr>
<tr>
<td>Test #2 $\frac{1}{32}$</td>
<td>0.90</td>
<td>0.43</td>
<td>0.80</td>
</tr>
</tbody>
</table>

### Table B-IV: Acoustic Velocity Considerations

<table>
<thead>
<tr>
<th>Diameter (in)</th>
<th>Ma #</th>
<th>$V_{max}$ ($\frac{ft}{sec}$)</th>
<th>$A_{min}$ (ft$^2$)</th>
<th>$D_{min}$ (ft$^2$)</th>
<th>$D_{min}$ (in$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.03125</td>
<td>0.3</td>
<td>243</td>
<td>0.00000592</td>
<td>0.0027</td>
<td>0.0329</td>
</tr>
<tr>
<td>0.0625</td>
<td>0.3</td>
<td>243</td>
<td>0.00002367</td>
<td>0.0055</td>
<td>0.0659</td>
</tr>
<tr>
<td>0.125</td>
<td>0.3</td>
<td>243</td>
<td>0.00009468</td>
<td>0.0110</td>
<td>0.1318</td>
</tr>
<tr>
<td>0.1875</td>
<td>0.3</td>
<td>243</td>
<td>0.000213</td>
<td>0.0165</td>
<td>0.1976</td>
</tr>
<tr>
<td>0.25</td>
<td>0.3</td>
<td>243</td>
<td>0.000379</td>
<td>0.0220</td>
<td>0.2635</td>
</tr>
</tbody>
</table>
Appendix C

The following section slightly expands the discussions regarding the results of this study. First and foremost, it should be shown at this point the equation used to calculate the heat transfer for the actual results of the study. This heat transfer equation differed slightly from the one used in the predictive calculations since the following included the effects of kinetic energy. Although these effects were minimal, the equation is still presented. Starting with the First Law of Thermodynamics, the equation can be simplified as follows.

\[
(Q_{in} + W_{in} + \sum_{in} m(h + KE + PE)) - (Q_{out} + W_{out} + \sum_{out} m(h + KE + PE)) = \Delta m (u + \dot{KE} + \dot{PE}) - m_1 (u + KE + PE)
\]

where \(KE\) is the kinetic energy and \(PE\) is the potential energy.

If the same assumptions are made that were made for Equation (18) above as well as:

\[
KE = \frac{1}{2} m V^2
\]

where \(V\) is defined as the velocity at specific point in the system and the velocity can be simplified by noting:

\[
\dot{Q} = \dot{\mathcal{L}} = V \cdot A \rightarrow V = \frac{\dot{\mathcal{L}}}{A}
\]

Therefore it can be seen that kinetic energy on a per mass unit basis is:

\[
KE = \frac{1}{2} \left( \frac{\dot{\mathcal{L}}}{A} \right)^2
\]

where \(A\) is the cross sectional area of where the kinetic energy is being measured.
Recall that Equation (C-1) can be simplified further by noting the locations in which points (1) and (2) define in Figure VIII as well as the assumptions made to reduce the First Law to Equation (18).

\[
\therefore Q_{\text{Total}} - (\sum_{\text{in}} m(h + KE + PE)) = W_{\text{Total}} + (\sum_{\text{out}} m(h + KE + PE)) + (m_2(u + KE + PE) - m_1(u + KE + PE)).
\]

Presuming no mass in, only one outlet, no potential energy changes, no kinetic energy at the beginning and end of vaporization, and no mechanical work, the above reduces too:

\[
Q_{\text{Total}} = m_{\text{out}}(h + KE) + (m_2(u_2) - m_1(u_1)).
\]

Therefore:

\[
Q_{\text{Total}} = m_{\text{out}} \left( h_{\text{out}} + \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 \right) + (m_2(u_2) - m_1(u_1)).
\]

Since the mass out was equal to the difference in the starting and ending mass, which, recall, was related to the specific volume of the associated states, the above equation could be simplified to:

\[
Q_{\text{Total}} = (m_1 - m_2) \left( h_{\text{out}} + \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 \right) + (m_2(u_2) - m_1(u_1)).
\]

Multiply out and combine like terms to get:

\[
Q_{\text{Total}} = m_1 \left( h_{\text{out}} + \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 - u_1 \right) + m_2 \left( u_2 - h_{\text{out}} - \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 \right).
\]

Where the above then simplifies to:

\[
Q_{\text{Total}} = m_{\text{start}} \left( h_{g@ \text{out}} + \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 - u_f @ \text{Start} \right) + m_{\text{End}} \left( u_{g@ \text{End}} - h_{g@ \text{out}} - \frac{1}{2} \left( \frac{\dot{\nu}_{CO_2}}{A_{\text{out}}} \right)^2 \right).
\]

(C-2)
The above equation was the simplified version used to determine the heat transfer from the data taken from the system. Note that the temperatures for the ending and starting states were taken as averages of the data collected from the study; therefore, in a sense, uniform state - uniform flow presumptions were taken into consideration. As mentioned previously, the predictive calculations did not include the kinetic energy as Equation (C-2) does; however, the effect of kinetic energy was minimum at best. The kinetic energy was included here since it could be determined from the study, wherein the predictive calculations could not presume an accurate enough kinetic energy term to support using kinetic energy.

The mass was known via the mass balance conducted at the end of each test run and was correlated through the volume ratios shown previously in the report. Now, one could argue that if the system pack really occurred and the ending mass was in error, that the heat transfer determined was also in error. On the contrary, since the masses being dealt with were on an order of magnitude less than that of the energy terms seen. Packing, although not completely negligible, was not enough to slay the goal of this study which was to prove feasibility over everything. As provided in the recommendations, this accountability was posed necessary in the next round of testing.

Once the heat energy was determined, the heat transfer and resulting heat transfer coefficient could be determined as follows:

$$h_{avg} = \frac{\frac{Q_{Total}}{\Delta t}}{A_{surface}\Delta T_{avg}}$$

(\text{C-3})

where \(\Delta t\) is the total time the system operated and \(\Delta T_{avg}\) is the average temperature change seen during operation of the system. Both of these values were looked at as averages of the raw data. In accordance with this notion, the start temperature was
averaged among the four thermocouples in the tank since they all were uniformly measuring the same bulk of fluid. For the end temperature, however, the temperature was determined using the average of just thermocouple five since it measured just the vapor that was present in the tank.

It should be noted that the volume of secondary fluid that exited the system was determined by looking at the average flow rate for the duration of the operation of the system and divided by the time of operation. In addition, the initial ramp up of the flow rate, as seen in the figures in Appendix B, was factored into the final volume. This calculated volume was then compared to the physical measurement of the secondary fluid captured after each test run. The calculated volume from the flow rate curves is what was used in the calculations of the results.
Appendix D

The following section contains images of the system as a whole in addition to the location of where the tests were conducted; these images were omitted from the actual report since they were deemed as superfluous to the content of the report.

Figure D-I: CSUN CECS RETC Entrance

Figure D-II: CSUN CECS RETC

Figure D-III: RETC Control Room Entrance

Figure D-IV: RETC Control Room
Figure D-V: System Apparatus I

Figure D-VI: System Apparatus II

Figure D-VII: System Apparatus III

Figure D-VIII: System Apparatus IV
Appendix E

Transmission of Compressed Gas & Associated Equipment under Pressure

The following list comprises of some, but not all, of the improper actions to take when transporting compressed gas equipment. *Never do the following:*

- Drag equipment across the ground or other abrasive surface.
- Tamper, disassemble, or attempt to disassemble equipment; specifically valves and pressure relief devices.
- Damage valves via some form of mechanical stress such as blunt force.
- Use equipment for any other task than is specified by the equipment’s original design specifications.
- Allow any combustible materials to be stored on or near compressed gas equipment.
- Transport equipment without the proper safety devices in place which may include caps or locks.
- Lift any equipment without assistive devices when said equipment is greater than fifty pounds.
- Remove identification tags or marks associated with equipment.
In addition to the various things to avoid when transporting compressed gases or compressed gas equipment, the following should *always be remembered*:

- Transport equipment using the proper assistive devices including dollies, carts, hand trucks, or forklifts when applicable.
- Secure compressed gas equipment properly when transporting to reduce the risk or injury from equipment coming loose during transportation.
- Wear proper protective gear when transporting compressed gas or compressed gas equipment.

**Storage of Compressed Gas & Associated Equipment under Pressure**

The following list comprises of some, but not all, of the improper actions to take when storing compressed gas equipment. *Never do the following:*

- Permit compressed gas equipment containing compressed gas to reach temperatures greater than the allowable maximum temperature specified in the design criteria.
- Allow equipment to come into contact with corrosive or explosive environments.
- Store equipment with open valves and/or ports to atmosphere.
- Store equipment containing compressed gas for periods longer than one year.
In addition to the various things to avoid when storing compressed gases or compressed gas equipment, the following should *always be remembered*:

- Store equipment in designated secure areas which should be areas with little traffic, are away from emergency exits, are properly vented, and allow for proper room to maneuver equipment safely.
- Conduct routine inspections of stored equipment to ensure leaks or hazardous conditions have not arisen.
- Storage areas should have limited access and be monitored with regards to which individuals enter and exit storage areas.
- Properly inventory equipment and gas quantities being stored so as to not create a hazardous condition by failing to remember how much and what type of gas may be present in equipment.
- Store equipment in the proper orientation as specified by design criteria.
- Compressed gas equipment should be stored in the order that said equipment was received.
- Equipment containing compressed gas should be stored separately from equipment not containing compressed gas.

**Usage of Compressed Gas & Associated Equipment under Pressure**

The following list comprises of some, but not all, of the improper actions to take when using compressed gas equipment. *Never do the following*:

- Mix different types of compressed gases in equipment.
- Use equipment containing compressed gas to be used in an electric circuit.
• Vary the temperature of the equipment to manipulate pressure of compressed gas unless using the proper approved method.

• Introduce compressed gas from equipment to the atmosphere in the direction of an individual.

• Use equipment in any way that is not specified in the design specifications.

• Introduce different compressed gases into equipment that is not intended to house said gas or would result in a change of service of said equipment from original gas class.

• Pry or force equipment to operate under improper conditions or with improper tools.

• Reduce the equipment pressure below that of the specified design pressure.

In addition to the various things to avoid when storing compressed gases or compressed gas equipment, the following should always be remembered:

• Properly understand the equipment being used in addition to the gas being used.

• Operate valves slowly and cautiously.

• Leak test equipment with the proper methodology including using inert gases and appropriate indicator solutions.

• Equipment valves should not be difficult to operate; if unable to operate valves by hand notify the individuals for proper valve replacement.

• Use proper regulatory devices when operating compressed gas equipment.

• All components should be inspected routinely for damages and assess the need to replace them.
• The use of check valves should be initiated when appropriate to avoid accidental mixing or contamination of compressed gas equipment.

Leaks

In the event that a leak may develop on the equipment, proper procedures should be taken to address the condition of the leaking component and its eventual repair. Leaks can be categorized into two specifications of which both entail different steps to take to address the leak. The classifications of the leaks are in accordance with campus regulations; however, falls under the judgment of the user him/herself. All leaks, minor or major, should be reported to campus safety department.

Minor Leaks

These leaks are generally classified as being nonhazardous and of no immediate threat to persons or property. Generally, these leaks are inaudible and impossible to sense alone. The leak should be identified using the proper detector which may include equipment or soap-water solution. Once the leak is identified, take appropriate steps to blow down equipment and repair leak. Equipment should not be transported, stored, or used when knowledge of leak has already been established.

Major Leaks

Major leaks are classified as leaks being hazardous in nature and requiring immediate attention so as to avoid damage to persons or property. In the event that personal protective equipment (PPE) is not readily available when a major leak has been identified, emergency services should be called immediately. In addition, the building’s fire alarm system should be activated so as to evacuate all persons to safety. Make sure
to assist those that need assistance to the exits securing all entrances and exits concurrently. When emergency personnel have arrived, provide them with details regarding the situation so that they may handle it properly.

**Personal Injury**

When utilizing compressed gases and compressed gas equipment, the potential for injury is always present. In the event of serious injury to personnel as a result of the failure of a compressed gas system, the following steps should be taken:

- If the injury poses a serious medical threat, emergency services should be contacted.
- Assist individuals injured and if possible administer aid which may include utilizing safety showers and eye wash stations, or removal of contaminated clothing, or administration of artificial respiration.
- Notify surrounding areas of incident and, if need be, activate building alarms.
- Only to prevent further exposure to harm should the injured person be moved.

**Final Note**

Safety is the responsibility of the user and all safety procedures should be read and understood thoroughly before operating any compressed gas equipment. Failure to properly adhere to safety procedures will result in personal injury to the user.

**High Level Procedures for Operating the PreVapS**

The following section summarizes a high level overview of the basic procedures used when operating a system that uses a compressed gas and essentially expels it to the
atmosphere under controlled conditions. Note that all of the above conditions were satisfied during the operation of the system.

The start of these procedures assumes that the compressed gas cylinders have already been procured and are in the laboratory.

1. Discuss emergency procedures amongst the individuals participating in the experiment.
2. Conduct a dry run of the procedures noted here prior to actually completing the experiment.
3. Properly put on safety glasses and mechanic's gloves before handling any materials.
4. Verify that all system apparatus valves are securely closed.
5. Properly transport the Nitrogen bottle from the loading platform to the holding tank using a dolly.
6. Secure the nitrogen tank in the holding tank using the proper lifting and handling techniques.
7. Connect the nitrogen tank to the system apparatus at the appropriate union connection and with the proper tools.
8. Verify once more that the valves separating the system from the atmosphere are closed.
9. Slowly crack the valve on the nitrogen tank to allow the gas to fill the system.
10. Verify any leaks in the system by applying a soap-water mixture to the joints of the piping network comprising the system apparatus.
11. Note any leaks in the system and record them properly including location, severity, and course of action for fixing the leak.
12. When all leaks have been recorded, close the valve on the nitrogen bottle.

13. Activate the manual override relief valve in the system and vent the nitrogen to the atmosphere.

14. Vacate the laboratory and allow the facility vents to properly vent the nitrogen to the outside.

15. Once returned to the laboratory, open the remaining system valves and allow any residual gas to escape the system.

16. Close all system valves.

17. Disconnect the nitrogen tank from the system using the proper tools.

18. Properly remove the nitrogen tank from the holding tank and store according to the process discussed previously.

19. Repair any leaks that were detected from the nitrogen testing appropriately.

20. Repeat nitrogen leak testing until no leaks remain in the system.

21. Make sure all measurement devices are working properly and are calibrated correctly.

22. Once again, verify that all system apparatus valves are securely closed.

23. Inspect all insulation and verify that it is intact and no piping is exposed.

24. Properly transport the empty primary working fluid tank from the loading platform to the primary temperature control tank using a dolly.

25. Secure the primary working fluid tank in the primary temperature control tank using the proper lifting and handling techniques.

26. Connect the primary working fluid tank to the system apparatus at the appropriate connection and with the proper tools.
27. Verify once more that the valves separating the system from the atmosphere are closed.

28. Properly transport the filled secondary working fluid tank from the loading platform to the secondary temperature control tank using a dolly.

29. Secure the secondary working fluid tank in the secondary temperature control tank using the proper lifting and handling techniques.

30. Connect the secondary working fluid tank to the system apparatus at the appropriate connection and with the proper tools.

31. Using the temperature control fluid, control the temperature of the working fluid within the secondary working fluid tank so that pressure within the secondary working fluid tank remains below that of cracking pressure of the system.

32. Using the temperature control fluid, control the temperature of the primary working fluid tank so that the temperature of the primary always remains below the secondary.

33. Open the valve at the manifold isolating the primary section of the system from the secondary and process sections of the system.

34. Open the valve isolating the primary working fluid tank from the primary section of the system.

35. Verify that the process section is indeed isolated from the primary and secondary sections via the solenoid valve.

36. Verify the manual override valve is indeed closed.

37. Open the separation valve that isolates the primary and secondary sections of the system.

38. Procure hearing protection and properly install.
39. Slowly open the secondary working fluid tank valve and allow the working fluid to fill the secondary section making sure not to open the valve all the way.

40. Monitor the pressure and temperature of the working fluid in the secondary Section using measurement devices.

41. Allow the equilibrium conditions to be established in the secondary section making sure the pressure remains above the system operating pressure, but below the system cracking pressure. Add temperature control fluid as is necessary.

42. Verify the primary working fluid tank valve and primary section valve is fully open; in addition, verify the process section is still isolated from the primary and secondary sections.

43. Crack the separation valve and monitor the working fluid as it is transferred from the secondary section to the primary section.

44. When the pressure in the secondary section drops to slightly above system operating pressure, close the separation valve.

45. Close the secondary working fluid tank valve.

46. Using the manual relief valve, vent the residual working fluid in the secondary Section to atmosphere.

47. Vacate the laboratory and allow the facility vents to properly vent the working fluid to the outside.

48. Once returned to the laboratory, close the valve that isolates the secondary Section from the rest of the system.

49. Open the separation valve slowly allowing the working fluid to reach equilibrium within the valve manifold.
50. Monitor the temperature in the primary section maintaining operating conditions in the primary section of the system; adding temperature control fluid as needed.

51. Properly remove the secondary working fluid tank from the secondary temperature control tank and store according to the process discussed previously.

52. Inspect all components in the process section of the system and verify working and safe conditions exist.

53. Complete the pretest procedures which include following through all the motions of executing the experiment without actually activating the system apparatus. Note this is a partial repeat of step 2; however, is necessary to promote a safe lab environment.

54. Verify the secondary fluid catch tank is properly installed and secure.

55. Verify all system components are secure and ready to conduct the experiment.

56. Return to the control room making sure to secure the system apparatus behind both blast doors of the test cell.

57. Run the validation check of the system and Measurement System. If any errors are returned, the experiment should be immediately stopped and the working fluid vented to atmosphere.

58. Correct any errors that may have arisen and repeat the previous steps.

59. Once the validation check is cleared, verify that all personnel are cleared from the test area and all emergency equipment is accounted for.

60. Initiate the Measurement System.

61. Energize the system process valve, which will begin the process by essentially joining the primary section with the process section.

62. Monitor the temperature and pressure of the system as the process is executed.
63. When the pressure in the primary section drops to approximately 10 pounds above atmospheric conditions, close the system process valve.

64. Terminate the Measurement System and save the data collected.

65. Wait approximately 10-15 minutes to allow the test cell vents to properly vent all of the working fluid.

66. Examine the pressure and temperature of the system to verify conditions are safe for personnel to reenter the test cell.

67. Close the primary working fluid tank valve.

68. Operate the manual override relief valve and vent the remaining working fluid to atmosphere.

69. Close the override valve as well as the separation valve.

70. Drain and dispose of the temperature control fluid properly and appropriately.

Store the system apparatus properly as outlined in the process steps above.