ABSTRACT

CRANFORD, ELTON DALE. A Quantitative Investigation to Improve Industrial Reciprocating and Centrifugal Air Compressor Performance via Inlet Air Cooling with Heat Recovery to the Steam System. (Under the direction of Dr. Stephen Terry).

Industrial compressed air systems are a significant source of energy consumption at manufacturing facilities. Thermodynamic analysis of air compressors indicates that the amount of power consumed by the compressor is a direct function of the inlet air temperature. Previously, this relation has been utilized at manufacturing plants by ducting in outside air for compressors rather than inside air, with the exception of oil-flooded screw compressors. This benefit can be observed because air inside of interior compressor rooms is at a higher temperature due to waste heat as a result of compressor inefficiencies.

This study investigates the benefits of further cooling inlet air for industrial reciprocating and centrifugal compressors, and recovering the waste heat for use with the steam system at a subject facility in North Carolina. Data gathered from the subject facility included temperature, relative humidity, energy consumption, and air flow. Temperature and relative humidity data were collected using HOBO® U12-012 data loggers for approximately seven months. The temperature and relative humidity data for the remaining five months was gathered from local airport data available from the National Weather Service. Energy consumption and air flow data was collected by existing data logging equipment at the subject facility. These parameters were then utilized in the daily analysis of underground earth tubes, an industrial heat pump, and a chilled water coil as potential methods of cooling inlet air.
Once the daily analysis for each option was complete, energy and demand savings were computed for each month. In cases where there was a demand reduction, minimum demand was considered as demand savings. In cases of demand increase, maximum demand was considered. Once energy and demand savings were established for each month, energy rates from local utilities were applied to determine overall cost savings to the subject facility from implementing each option. Once the equipment necessary for each option was determined and designed, total implementation costs were determined for each option as well as simple payback period.

From the analysis, it was determined that the best option for cooling inlet air temperatures to improve compressor performance involves the installation of a 120 ton industrial heat pump. This method requires the greatest amount of capital up front at $110,405, but also presents the greatest overall energy savings of -36,323 kWh, 689 kW, 7,657 MMBTU energy, and 93 MMBTU demand. The overall anticipated cost savings with all energy savings and increases considered is $8,080 per year. Based on the implementation cost and energy cost saving of an industrial heat pump, the simple payback is 14 years.

An underground earth tube system presented modest energy savings of 85,158 kWh and 35 kW, which resulted in an overall cost savings of $4,471. This system consisted of four hundred Class 150 4 inch diameter PVC pipes. Each of these pipes was designed to a length of 50 feet. With an anticipated installation cost of $100,500, the simple payback period for underground earth tubes is 22.5 years.
The least attractive of the options analyzed was a chilled water coil. The installation of a chilled water coil would involve using the existing plant chilled water system to cool inlet air for the compressors. This option presented the least implementation cost of $9,718, but consumed 25,595 kWh more electricity and 694 kW more demand than the existing system, for an increase in energy costs of $11,395 per year. Since there is more energy consumed than saved with this option, this analysis resulted in an infinite payback period.
A Quantitative Investigation to Improve Industrial Reciprocating and Centrifugal Air Compressor Performance via Inlet Air Cooling with Heat Recovery to the Steam System

by
Elton Dale Cranford

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APPROVED BY:

Dr. Herbert M. Eckerlin
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DEDICATION

I would like to dedicate this work to several significant influences on my life. First of all, I would like to dedicate this work to my Lord and savior Jesus Christ for the sacrifice He made for me. I would also like to dedicate this work to my parents, Scott and Kenna Cranford for their love and support during my time as a student at North Carolina State University. I would also like to dedicate this work to Dr. Stephen Terry for all of the opportunities and support he has provided to me throughout this process.
Elton Dale Cranford was raised in the rural community of Seagrove, North Carolina. Cranford acquired an interest in engineering when working on his senior project in high school. This project involved design modifications of a gas grill, which involved the alteration of design dimensions in order to minimize waste while maintaining the same quality of product. Cranford also gained an appreciation for hands-on skills that are used in the manufacture of engineering designs by taking classes in metal fabrication and small engines with Mr. Everette Sheffield. These classes further enhanced his desire to become an engineer and learn more about mechanical systems in order to practice the skills learned at local and state Agricultural Mechanics competitions as a part of Future Farmers of America (FFA). Dale also learned and developed various invaluable leadership skills as a chapter officer with his high school FFA chapter under the advisement of Mr. Sheffield. After graduating from Southwestern Randolph High School in May 2008, Cranford sought further educational opportunities in engineering at North Carolina State University.

Cranford began attending North Carolina State University in August 2008 in the pursuit of a Bachelor of Science degree in engineering. After the first semester, Cranford decided that he wanted to focus his engineering degree in the mechanical department. During his sophomore and junior undergraduate years at North Carolina State University, Dale served as a Resident Advisor for a group of young men in Becton Residence Hall in the University Honors Village at Becton Residence Hall in the Quad. In this position, Cranford was able to further
develop and enhance his interpersonal and leadership skills. During his senior year, Dale left the Resident Advisor position to focus more time on senior design projects and to seek opportunities to gain engineering experience.

In the fall of his senior year, Cranford received an opportunity to gain real world engineering experience as an intern under the North Carolina State Energy Internship Program. As a part of this program, Cranford worked as a Student Energy Engineering Intern as part of a team of student engineers performing industrial energy surveys with the Industrial Assessment Center, and completing measurement and verification reports with the Energy Management Program. Working in these roles enabled Dale to discover an interest in becoming an energy engineer and assisting manufacturing facilities in decreasing energy usage and energy cost.

After the completion of his undergraduate degree, Cranford decided that it would be beneficial to further his education and pursue a thesis based Master of Science degree in mechanical engineering at North Carolina State University with a concentration on energy conversion and systems. Cranford completed his undergraduate degree in May 2012 and began his graduate program soon after. After working with Dr. Stephen Terry as an undergraduate in the State Energy Internship Program, Cranford joined the Industrial Assessment Center exclusively as a graduate student and worked with a team of fellow graduate and undergraduate energy engineers. During his participation in these programs, Cranford has been a part of approximately 30 industrial energy assessments. After
graduation, Cranford plans to pursue a career as an energy engineer or related position within the field.

Outside of his educational and work time, Dale enjoys attending North Carolina State football and men’s basketball games. Cranford has won the award as the most loyal student fan in these two sports for the past two years. Dale also enjoys playing pickup and intramural basketball with his family, friends, and colleagues at the IAC. Dale’s other interests include attending church with his family, fishing at local lakes with his dad and grandpa, hunting, bowling with his family and friends, and watching live sports on television including football, basketball, baseball, and NASCAR racing.
ACKNOWLEDGEMENTS

There are several people I would like to thank for their assistance in making this work possible. I would first like to thank God for all of the opportunities He has blessed me with and for always providing guidance in answering my prayers. I would like to thank my parents, Scott and Kenna Cranford for their love and support of all my educational endeavors as well as my brother Darren for his support. I would furthermore like to thank all other family and friends for all of your support throughout my undergraduate and graduate education at North Carolina State University.

I would also like to thank the subject facility for access to their compressed air systems throughout the duration of this project and their assistance in obtaining data.

I would like to thank Dr. Stephen Terry for providing me with the opportunity to make a Master’s Degree possible. I greatly appreciate the opportunity he gave me to learn and develop my educational background for applications in practical situations first as a Student Energy Engineer under the North Carolina State Energy Internship Program with North Carolina State University Energy Solutions during the senior year of my undergraduate education, and the opportunity to learn as a Research Assistant with the Industrial Assessment Center while working on my Master’s Degree. The experiences I have gained by performing approximately 30 total industrial energy assessments will be invaluable in my future career.
I would also like to thank Dr. Herbert Eckerlin and Dr. Kevin Lyons for serving on my graduate advisory committee. I appreciate your participation and assistance on this project.
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Chapter 1 Introduction

Industrial manufacturing facilities require a significant amount of energy to produce products that can be used for everyday applications. The energy used in industrial processes is consumed by a variety of end users including manufacturing equipment, mechanical equipment, facility lighting, and maintaining temperature and relative humidity in a particular plant. Nearly all of these devices use electricity or a fuel source (natural gas, fuel oil, wood waste, and others) for operation. This energy consumption incurs a significant cost to the facility because they must purchase this energy from local utilities unless product waste or heat can be reused. One of the largest consumers of energy at these facilities is compressed air systems. Since these systems consume a substantial amount of energy and constitute a significant cost in the manufacturing process, finding methods to reduce the amount of energy necessary for air compression at industrial manufacturing facilities is highly beneficial.

1.1. Energy Use in the United States Today

Energy constitutes one of the most significant challenges to the cost of operating industrial manufacturing facilities. Several different energy resources are used at these plants, but the most common energy sources used today are electricity and natural gas. Of all the electricity consumed in the United States today, nearly 26% is consumed by industrial facilities, which
includes “direct use” for equipment operation (1). This electricity is used to power numerous devices used in manufacturing, which includes air compressors, lighting, chillers, and manufacturing equipment. Natural gas consumption at these facilities also encompasses a significant amount of the total natural gas used in the United States at 28% (2). A chart which displays natural gas usage in other sectors is shown in the figure below

![Natural Gas Consumption Pie Chart](image)

**Figure 1.1: Natural Gas Consumption in 2011**

At 28% of the overall natural gas consumption in the United States in 2011, the total amount of natural gas consumed was 6.9 trillion cubic feet, or Tcf (2). Natural gas used at industrial facilities provides the energy needed to operate equipment such as boilers, ovens, and heating systems.
1.2. Compressed Air Energy Consumption

The primary system for observation in this experiment is compressed air systems. These systems consume electricity in both the form of electrical energy and demand. Though most residential consumers of electricity are only charged for the amount of electrical energy they consume during a given month, industrial manufacturing plants are charged for both the amount of energy they use as well as the maximum amount of power they require at peak operation during a given month due to their significant electricity needs. Compressed air systems contribute a significant load to the energy consumption and demand necessary for production.

Electricity, natural gas, and water are often considered to be the three largest utilities used at manufacturing facilities. Along with these three sources, compressed air is commonly accepted as the “fourth” largest utility at manufacturing plants today (3). This widely held belief occurs because of the amount of energy that is dedicated to these systems as well as their inefficiencies. Compressed air at manufacturing plants is often misconstrued as being “free” because one sees it as just air. However, this belief is not the case. Air is compressed through the use of an electric motor. When the compressed air system is not operating efficiently, this motor continues to work harder to provide the amount of pressure necessary for plant process.
There are several areas where compressed air production can be more efficient at manufacturing facilities, which saves energy and energy related costs. One area where compressed air systems are often inefficient is the development of air leaks in compressed air lines. It is widely known that with any compressed air system, air leaks are inevitable. However, if a facility establishes and maintains a strong air leak maintenance program, the motor driving the compressor will have decreased load from no longer having to supply air leaks on top the base production load.

Another area where compressors are also inefficient involves the amount of heat lost from compressors to the surrounding air (3). A general rule of thumb indicates that approximately 80% of all electricity used in the compression of air is lost to the surrounding air as heat due to compressor inefficiencies. If this heat is recovered, manufacturing plant can reap the benefits of decreased space heating loads or decreased fuel consumption in boilers. This heat added to the compressor space can also contribute to less efficient operation of the compressor and surrounding equipment if not recovered properly.

Additional measures can also be taken to reduce the cost of compressed air to a facility. These measures include taking in outside air for compressors, using refrigerated air dryers, and reducing the air pressure if the pressure is much higher than necessary for facility operation. Along with these commonly known measures, research into alternate methods of
increasing compressor efficiency and recovering waste heat is essential in reducing the cost of compressed air to industrial manufacturing plants.

1.3. **Inlet Air Temperatures for Industrial Air Compressors**

An energy conservation practice commonly used as a means of reducing the cost of compressed air involves moving the air intakes for compressors from the inside of the facility to the outside. As mentioned before, a significant amount of heat loss from compressors and neighboring equipment causes the temperature of the surrounding air to rise. In these cases, the temperature of air outside of the facility is often cooler than the air inside. Since the outside air is cooler, it is denser and can thus be compressed to the same pressure and flow by using less electrical energy at the compressor motor. Significant energy and cost savings can be observed from this type measure, which requires the small installation cost of ductwork from the compressor intake to the outside of the building as well as other necessary air filtration and miscellaneous equipment.

Since moving air intakes to the outside of the facility results in significant energy savings, cost savings, and typically has a reasonable payback on the initial investment, what would be the benefits of using an external source of heat removal to decrease the temperature even further? Based on the same principle as moving air intakes outside, it is believed that significant energy savings can be witnessed from using an external heat removal source, such as an underground air pipe, industrial heat pump, or chilled water coil to reduce the
temperature further. It is also believed that the cost savings from implementing such measures would be beneficial to offsetting any initial cost that these measures would require for implementation and operation. This experimental and quantitative data analysis of the effects of inlet air temperatures on industrial reciprocating and centrifugal air compressors also investigates the economics of the previously mentioned alternatives to current compressed air system operation for a subject facility in North Carolina in an effort to reduce energy usage and cost with a reasonable payback.
Chapter 2  Thermodynamic Properties

In order to better understand terminology associated with compressed air systems, background in various thermodynamic properties is beneficial. The following sections describe various properties that have a direct impact on the amount of energy and cost required to compress air for industrial purposes.

2.1. Temperature Scales

In order to quantify the temperature of solids, liquids, and gases, several different temperature scales have been developed. The two temperature scales commonly used today are the Celsius scale and the Fahrenheit scale. The Celsius scale is used as a means of quantifying temperature for the International System of Units (typically referred to as the abbreviation SI). The Fahrenheit scale is the temperature scale of choice for the English system.

Each of these systems used to place a measurement on the current temperature of a substance is based on properties of water. For the Celsius temperature scale, a temperature of 0°C represents the freezing point of water. This temperature corresponds to a temperature of 32°F on the Fahrenheit scale. The boiling point of water for the Celsius scale is 100°C, which corresponds to a temperature on the Fahrenheit scale of 212°F. These two scales are
commonly referred to as two-point scales due to the fact that their primary temperature values are set at two different points (ice point and steam point) (4).

Since these two parties measuring temperatures may use either one of these temperature scales, it is important to understand how to convert temperatures between the two scales. If temperatures are converted, they can be easily compared and evaluated for the situation analyzed. Converting temperature from the Celsius scale to the Fahrenheit scale can be easily accomplished by using the relation

\[ T(\degree F) = T(\degree C) \times \frac{9}{5} + 32 \]  

(2.1)

Similarly to the method used in Equation 2.1, temperature can be converted from the Fahrenheit scale to the Celsius scale by using the equation

\[ T(\degree C) = (T(\degree F) - 32) \times \frac{5}{9} \]  

(2.2)

Though using these two systems is an effective means of developing a temperature scale that can be used to relate the temperature of one substance to another, they are not ideal scales for thermodynamic analysis. In thermodynamic analysis of a substance, it is more desirable to use a temperature scale that is independent of any specific properties or states of the material, such as those used for water in the Celsius and Fahrenheit scales. For this reason, the use of absolute (thermodynamic) temperature scales is more commonly used for these analyses.

In order to have systems that relate to the common systems used in society, two temperature scales have been developed for the purpose of converting Celsius and Fahrenheit
temperatures to absolute temperatures. The thermodynamic temperature scale employed for
the SI system of measure is known as the Kelvin scale. It is important to note that the units
used for this temperature system are Kelvins, K, not °K. The degree symbol from this
notation was dropped in 1967, so earlier publications may use this designation (4). The
lowest possible temperature on the Kelvin scale is absolute zero, or 0 K. Then the scale
follows that only one nonzero point of reference is assigned to initiate the slope of this linear
scale (4). The Kelvin scale is able to be directly calculated from the Celsius scale temperature
of the substance by using the equation

\[ T(K) = T(\degree C) + 273.15 \]  \hfill (2.3)

It is typically acceptable to round 273.15 to 273 in Equation 2.3 to reduce the number of
significant figures used.

The thermodynamic scale that has been developed in conjunction with the English system of
units is known as the Rankine scale, which uses units designated by R much in the same way
the Kelvin scale uses K. The temperature of the substance being analyzed on the Rankine
temperature scale can be easily found from the temperature of the substance on the
Fahrenheit scale by using the relation

\[ T(R) = T(\degree F) + 459.67 \]  \hfill (2.4)

Just as with Equation 2.3 for the Kelvin scale, the additive term in Equation 2.4 is often
rounded to 460.
As was the case with the Celsius and Fahrenheit scales, relations have been developed to quickly convert from temperatures on the Kelvin scale to the Rankine scale and vice versa. Temperatures on the Kelvin scale can be easily converted to temperatures on the Rankine scale by using the equation

\[ T(R) = 1.8T(K) \]  

(2.5)

A comparison of various temperature scales is shown in Figure 2.1.

![Temperature Scale Comparison](image)

**Figure 2.1: Temperature Scale Comparison**  
(4)

It is important to note that the scale shown in Figure 2.1 displays the reference point of the Kelvin scale at 273.16 K instead of 273.15 K. Originally the Kelvin scale was based on the reference temperature being at 0°C. However, at the Tenth General Conference on Weights
and Measures held in 1954, it was determined that a more accurate reference point for the Kelvin scale would be to use the triple point of water which occurs at 0.01°C (4).

2.1.1. Absolute Temperatures in Thermodynamic Calculations

When performing thermodynamic calculations, it often not apparent whether the temperature required in completing a computation is from one of the common temperature scales or the thermodynamic temperature scales. In most cases, the temperature to be used in these calculations is the thermodynamic temperature. If temperatures from the Celsius or Fahrenheit temperature scales are used in thermodynamic calculations, an incorrect answer will result.

The only exception to the requirement of absolute temperatures involves calculations that require temperature differences. In the case of a temperature difference calculation, it does not matter whether the thermodynamic or common temperature scale is used. This exception occurs because the temperature difference of the SI system temperature scales are equated in the relation

\[ \Delta T(K) = \Delta T(^\circ C) \]  \hspace{1cm} (2.6)

Likewise, the temperature differences in the English system of measure are equated through the relation

\[ \Delta T(R) = \Delta T(^\circ F) \]  \hspace{1cm} (2.7)
Though Celsius and Fahrenheit temperatures can be used in temperature difference calculations, it is generally better to err on the side of caution and convert all temperatures to the absolute scale.

### 2.2. Understanding Pressure

Another important parameter to consider during thermodynamic analysis is pressure. Pressure is defined as a normal force that is exerted by a fluid per unit area (4). For thermodynamic purposes, pressure is used as a parameter to define liquids and gases. The equivalent of pressure when referring to solid mechanics is known as normal stress, which is created by a force acting perpendicular to a surface. Normal stress is calculated for solids by simply dividing the force by the area of the surface on which it is acting.

For the SI system of measure, pressure is defined by the force of the fluid acting on the surface in Newtons divided by the area of the surface on which it acts in square meters (N/m²). For simplicity, these units in combination have been simplified to the unit of a pascal (abbreviated at Pa). Often, the units of pascals are simplified for large pressures to kilopascals (1 kPa = 10³ Pa) or megapascals (1 MPa = 10⁶ Pa). For the English system of measure, pressure is often given in terms of pound per square inch, which is abbreviated as psi. For large quantities of pressure, this value may be represented in terms of kilopounds per square inch (1 ksi = 1,000 psi). Other pressure units commonly used in practice are the bar,
standard atmosphere, and kilogram-force per square centimeter (4). Relations for each of these pressures are shown in Table 2.1.

Table 2.1: Pressure Relations

<table>
<thead>
<tr>
<th>Original Pressure</th>
<th>1 bar (Pa)</th>
<th>1 atm</th>
<th>1 kgf/cm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Equivalent (Pa)</td>
<td>10⁵</td>
<td>101,325</td>
<td>9.807 x 10⁴</td>
</tr>
<tr>
<td>Pressure Equivalent (kPa)</td>
<td>100</td>
<td>101.325</td>
<td>98.07</td>
</tr>
<tr>
<td>Pressure Equivalent (MPa)</td>
<td>0.1</td>
<td>0.101325</td>
<td>0.09807</td>
</tr>
<tr>
<td>Pressure Equivalent (bars)</td>
<td>1</td>
<td>1.01325</td>
<td>0.9807</td>
</tr>
<tr>
<td>Pressure Equivalent (atm)</td>
<td>0.98692</td>
<td>1</td>
<td>0.9679</td>
</tr>
<tr>
<td>Pressure Equivalent (N/cm²)</td>
<td>10</td>
<td>10.1325</td>
<td>9.807</td>
</tr>
<tr>
<td>Pressure Equivalent (N/m²)</td>
<td>10⁵</td>
<td>101,325</td>
<td>9.807 x 10⁴</td>
</tr>
</tbody>
</table>

Once the units of the pressure being measured are understood, it is important to realize that there are several different types of pressure. The first type of pressure considered is known as the absolute pressure. The absolute pressure is the actual pressure at a given position in space and is measured relative to an absolute vacuum since an absolute vacuum is considered to be at zero absolute pressure (4). However, most gauges used to measure pressure in everyday practice measure the pressure relative to atmospheric pressure. This type of pressure is commonly known as gage pressure because it represents the difference between the absolute pressure and the local atmospheric pressure (4).
Pressures that are below the local atmospheric pressure are referred to as vacuum pressures and are measured by vacuum gages, which indicate the difference between atmospheric and absolute pressures (4). These pressures have been related to one another through the development of pressure relations. The gage pressure is related to the absolute and atmospheric pressures through the relation

\[ P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} \]  

(2.8)

where \( P_{\text{gage}} \) is the gage pressure, \( P_{\text{abs}} \) is the absolute pressure, and \( P_{\text{atm}} \) is the atmospheric pressure. Vacuum pressure is related to the atmospheric pressure and absolute pressure by using the equation

\[ P_{\text{vac}} = P_{\text{atm}} - P_{\text{abs}} \]  

(2.9)

The relation between these different pressures is further illustrated in Figure 2.2.

![Diagram of pressure relations](image)

Figure 2.2: Absolute, Gage, and Vacuum Pressure Relations (4)
Most pressure gages used in practice provide readings in measurements of gage pressure. For example, consider a typical air compressor gage. If the common reading of the gage is 100 psi, then it represents a pressure of 100 psi above the atmospheric pressure. If the local atmospheric pressure of that area is 15 psi, then the absolute pressure of the compressor is 115 psi (100 psi + 15 psi). In order to differentiate between these different scales, designations are often added to the end of the pressure unit. For example, gage pressure readings are often designated with the letter “g” (psig) and absolute pressure readings with an “a” (psia). An example of a pressure gage commonly used for industrial purposes is shown in Figure 2.3.

![Figure 2.3: Standard Pressure Gage](5)

Close observation of Figure 2.3 reveals that the notation for measurements of pressure on this pressure gage is in units of psi. For practical purposes, all gages that display units in psi are taken to be measures of gage pressure.
2.3. Ideal Gases

The most accurate method currently available for finding the properties of a substance is property tables. These tables have been developed over years of experimentation and mimic the behavior of a substance fairly precisely. In order to use property tables, two properties of a substance must be known. However, these tables are not always the most effective method of finding desired properties of a substance being analyzed. What if the known properties (such as temperature or pressure) are not exactly the same as the experimental temperatures used to develop the property tables? Linear interpolation can be used in this scenario to determine an approximation, but the behavior of the substance may not be linear. What if no property table exists for the substance being analyzed?

In order to find properties in the situations described above as well as other times when property tables are not as effective, equations of state can be used to find the desired property of a substance. An equation of state is an equation that relates the temperature, pressure, and specific volume of a substance being analyzed (4). These equations can be used as a means to find properties at intermediate temperatures and pressures as well as for substances without property tables that exhibit similar behavior to other substances. For the purposes of this analysis, air as an ideal gas is considered. These same ideal gas techniques can be used for the behavior of any gas as long as it is analyzed at low pressures.
In most applicable cases, air can be treated as an ideal gas for calculations. An ideal gas is described as a gas in which no attraction occurs between the molecules (6). Though most gases that occur in nature do have collisions amongst their respective particles, at low pressures these gases exhibit similar behavior to that of an ideal gases. Treating a gas as ideal as these pressures enables the use of the ideal gas equation of state, which is a simple relation used to find the pressure, temperature, or specific volume of a substance if two of the other properties are known. The ideal gas equation of state is typically given by the relation

\[ PV = R_{\text{gas}} T \]  \hspace{1cm} (2.10)

or

\[ PV = mR_{\text{gas}} T \]  \hspace{1cm} (2.11)

where \( P \) is the absolute pressure of the gas, \( v \) is the specific volume of the gas, constant of proportionality \( R \) is the gas constant, \( m \) is the mass, and \( T \) is the absolute temperature of the gas (4). The gas constant for the actual gas being analyzed can be found by a relation with the universal gas constant. This gas constant is found by using the equation

\[ R_{\text{gas}} = \frac{R_u}{M} \]  \hspace{1cm} (2.12)

where \( R_{\text{gas}} \) is the gas constant for the gas being analyzed, \( R_u \) is the universal gas constant, and \( M \) is the molar mass of the gas being analyzed. The universal gas constant is the same for all substances and its value in various SI and English system units is shown in Table 2.2.
The molar mass of a substance is defined as the mass of one mole of a substance in grams (also referred to as gram-mole, or kgmol) in kilograms (4). For the English system, the molar mass is the mass of 1 lbmol in lbm (4).

When looking at the periodic table of elements quick observation shows that no units are given for the molar mass of an element. This occurs because the method used to define a substance causes the numerical value of the molar mass to be the same for both systems of measurement. For example, the molar mass of helium (He) is 4.003. This means that the mass of 1 kmol of helium is 4.003 kg. For the English system of measure, it means that the mass of 1 lbmol of helium is 4.003 lbm. Therefore, the total mass of a substance is a product of the molar mass of the substance and the number of moles of that substance. The total mass of the system in this case can be found by using the equation

\[ m = MN \]  \hspace{1cm} (2.13)

where \( m \) is the total mass of the system and \( N \) is the total number of moles of the substance in the system. The molar mass and gas constant for various substances is shown in Table A.1 for SI units and Table A.2 for English units.
The ideal gas equation of state can also be represented in other forms. The first of these alternative forms involves the situation where the total volume of the system is known with or without the mass being defined. The specific volume of the substance to be analyzed can be found by using the relation

\[ V = m/v \]  
(2.14)

where \( V \) is the total volume of the system, \( m \) is the mass of the system, and \( v \) is the specific volume of the system. Substituting Equation 2.14 into Equation 2.10 to find the ideal gas equation of state as a function of total system volume and total system mass yields

\[ PV = mR_{\text{gas}}T \]  
(2.15)

Equation 2.15 is a useful form of the ideal gas equation of state if the total volume of the system is known and the system mass is desired.

The next alternative form of the ideal gas equation of state can be used to find the number of moles in the system or if the total volume and number of moles are given. If the relation in Equation 2.13 is substituted for the system mass, the resulting equation is

\[ mR_{\text{gas}} = (MN)R_{\text{gas}} \]  
(2.16)

Using the result of Equation 2.16 along with the relation from Equation 2.12 the resulting relation yields

\[ (MN)R = NR_{u} \]  
(2.17)

Substituting the result of Equation 2.17 into Equation 2.15 yields

\[ PV = NR_{u}T \]  
(2.18)
This relation is useful when the number of moles is known or needs to be determined because the universal gas constant can be used as opposed to needing the gas constant for the gas analyzed.

The next additional ideal gas relation makes use of a term known as \( \bar{v} \), or the molar specific volume. The total volume can be found by using the total number of moles along with this value in the relation

\[
V = \bar{v}N
\]  

(2.19)

The relation in Equation 2.19 can then be substituted into Equation 2.10 to find the relation

\[
P\bar{v} = R_\text{g}T
\]

(2.20)

This relation is beneficial when finding specific molar volume. Any other property denoted with a bar above it is a specific molar property.

In the event that a substance is analyzed over a process from an initial state (State 1) to a final state (State 2) and the mass remains constant, a relation can be developed by equating and simplifying Equation 2.15. This simplification yields the expression

\[
\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}
\]

(2.21)

where the subscripts denote the property at the state considered.

Many other gases also obey these same equations of state. Ideal gases are often observed as imaginary substances that obey Equation 2.10 (4). Though this relation represents an
imaginary substance, it has been determined to be a useful representation of gas behavior. Numerous observations from experiments show that the \( P-v-T \) behavior of real gases at low densities closely follows those of the ideal gas law (4). The density of a gas decreases at low pressures and high temperatures where the gas begins to show ideal gas behavior (4).

For the typical range of temperatures and pressures used in an industrial setting, many gases are often treated as ideal gases if property tables are unavailable. These gases include air, nitrogen, oxygen, hydrogen, helium, argon, neon, and krypton among others often with negligible error (often less than 1 percent) (4). For the purposes of engineering calculations, this minute amount of error is acceptable. However, this relation should not be applied to all substances. For instance, dense gases such as water vapor used in steam power plants and refrigerant vapor used in refrigerators should not be treated as ideal gases (4). For these substances, property tables should always be used.

If a desired value falls between table parameters for substances where the ideal gas equation of state is not applicable, linear interpolation should be used. A common equation used to linearly interpolate between values is

\[
d = d_1 + \frac{g - g_1}{g_2 - g_1} (d_2 - d_1)
\]

(2.22)

where \( g \) is the given value, \( g_1 \) and \( g_2 \) are the two closest approximations in the table, \( d \) is the desired value, and \( d_1 \) and \( d_2 \) are the corresponding values of the desired value to \( g_1 \) and \( g_2 \) (7).
2.3.1. Compressibility Factor

The ideal gas equation of state presented in Section 2.3 provides a simple method that can be used to quickly find properties of a substance. However, there is some error associated with using this equation of state. An example of the error involved in this assumption is illustrated for steam in Figure 2.4.

Figure 2.4: Percent Error in Treating Steam as an Ideal Gas (4)
The percentage of error shown in Figure 2.4 is found by using the relation

\[
% \text{ error} = \left( \frac{v_{\text{table}} - v_{\text{ideal}}}{v_{\text{table}}} \right) \times 100
\]  

(2.23)

where % error is the percentage error involved in the calculated value as compared to the actual value, \(v_{\text{table}}\) is the specific volume of the substance found in a property table, and \(v_{\text{ideal}}\) is the specific volume of a substance found by using the ideal gas law. Further analysis of Figure 2.4 reveals that gases deviate from ideal gas behavior much more at state residing near the saturation region and the critical point of the substance (4).

The variance of ideal gas values from table values can be accounted for by a correction factor \(Z\), also known as the compressibility factor, which has been developed. The relation used to find the compressibility factor is

\[
Z = \frac{PV}{R_\text{gas}T}
\]  

(2.24)

The reason why this relation can be used occurs because the compressibility factor for ideal gases is considered to be 1. Equation 2.24 can also be written as

\[
PV = Z R_\text{gas}T
\]  

(2.25)

Substituting the ideal gas relation from Equation 2.10 into Equation 2.24 simplifies the expression for the compressibility factor to

\[
Z = \frac{v_{\text{actual}}}{v_{\text{ideal}}}
\]  

(2.26)
where \( v_{\text{actual}} \) is the actual specific volume of the substance. The compressibility of real gases can be less than, equal to or greater than 1. The farther the compressibility factor is from 1, the less it behaves as an ideal gas (4). The value of the compressibility factor for several gases is shown in Figure 2.5.

Figure 2.5: Comparison of Compressibility Factor \( Z \) for Various Gases  (4)

As mentioned before, gases tend to mimic the behavior of an ideal gas at low pressures and high temperatures. However, determining whether a substance is at a high temperature and
low pressure is not as simple as stating that one temperature and pressure sets the standard for high pressure and low temperature. Instead, values that constitute high temperatures and low pressures are measured relative to the critical pressure and critical temperature of the substance.

At any given temperature and pressure combination, gases may behave much differently. However, these gases act in nature very similar to temperatures and pressures that are normalized with the critical temperature and critical pressure of the substance (4). The normalization for pressure is found by using the relation

\[ P_R = \frac{P}{P_{cr}} \]  

(2.27)

where \( P_R \) is referred to as the reduced pressure of the substance, \( P \) is the actual pressure of the substance, and \( P_{cr} \) is the critical pressure. Similarly, the reduced temperature can be determined by using the relation

\[ T_R = \frac{T}{T_{cr}} \]  

(2.28)

where \( T_R \) is the reduced temperature of the substance, \( T \) is the actual temperature of the substance, and \( T_{cr} \) is the critical temperature of the substance. The \( Z \) factor for all gases is approximately equivalent at the same reduced pressure and reduced temperature, which is referred to as the principle of corresponding states (4). Compressibility factor charts such as the one shown in Figure 2.5 above plot the compressibility factor as a function of the reduced pressure and reduced temperature. Since the gases obey the trends of the principle of
corresponding states relatively close, a generalized compressibility chart has been developed and is shown in charts Figure B.1, Figure B.2, Figure B.3, Figure B.4, and Figure B.5.

Based on the generalized compressibility charts presented in Figure B.1, Figure B.2, Figure B.3, Figure B.4, and Figure B.5, several observations can be made. These observations are as follows:

1. At relatively low pressures \( (P_R << 1) \), gases tend to behave as ideal gases regardless of their temperature (4)

2. At relatively high temperatures \( (T_R > 2) \), behavior of an ideal gas can be assumed with ample accuracy regardless of pressure (except for \( P_R >> 1 \)) (4)

3. The variation of a gas from the behavior of an ideal gas is highest in the vicinity of the critical point of the substance (4)

An illustration of the third observation is shown in Figure 2.6.

![Figure 2.6: Gases Deviate Most at Critical Point](4)
In most cases, the pressure and specific volume or the temperature and specific volume are given as a means to find the third parameter. For the situations, the compressibility charts can be used to find this property. However, in some cases pressure and temperature are given and the generalized compressibility charts are not useful and would require tedious trial and error to find the specific volume (4). As a result, it is important to have a relation established for when these two parameters are given. The equation that is therefore used to find what is known as the pseudo-reduced specific volume, \( v_R \), is

\[
   v_R = \frac{v_{\text{actual}}}{RT_{\text{cr}}P_{\text{cr}}}
\]  

An important item of note for the relation presented in Equation 2.29 is that instead of being directly related to a critical specific volume, the reduced specific volume is directly related to the critical pressure and critical temperature of the substance being analyzed. Lines of constant reduced specific volume can also be added to the compressibility charts so that \( T \) and \( P \) can be determined without having to resort to time consuming iterations (4).

2.4. Water Content of Air

Another important aspect to consider in the analysis of a compressed air system involves analysis of the amount of water vapor contained in the air being compressed. Since air that is brought into a compressor typically contains water vapor, the vapor is compressed during the process by the same means as the air. Understanding methods of quantifying the amount of
water vapor existent in air is an important component in a compressed air system analysis to prevent water vapor contained in the air from affecting the performance of compressors.

2.4.1. **Dry and Atmospheric Air**

Air exists in nature as a mixture of several different gases. The primary gases included in air are nitrogen and oxygen, along with several other gases in much smaller quantities. Air that exists in the atmosphere typically contains some amount of water vapor (otherwise known as moisture) and is commonly referred to as atmospheric air (4). Most air encountered for the purposes of compression in practical applications contains at least some amount of moisture, but there are a few cases where air does not contain water vapor. Air that does not contain any water vapor is typically referred to as dry air (4).

2.4.2. **Specific and Relative Humidity of Air**

The amount of water vapor present in air can be quantified by several different standards. The simplest method of quantifying the amount of water vapor present in air is to compare the mass of water vapor that resides within a unit mass of dry air. This type of measure used for water vapor content is known as the absolute or specific humidity (also given the name humidity ratio), $\omega$ (4). The absolute humidity can be found by using the relation
where \( m_v \) is the mass of the water vapor and \( m_{da} \) is the mass of dry air. Equation 2.30 can also be manipulated by using Equation 2.10 if parameters of the ideal gas law are available. Simplifying Equation 2.30 with Equation 2.10 yields

\[
\omega = \frac{m_v}{m_{da}} = \frac{P_v V / R_v T}{P_{da} V / R_{da} T} = \frac{P_v / R_v}{P_{da} / R_{da}}
\]  

(2.31)

where \( P_v \) is the absolute pressure of the water vapor, \( V \) is the volume, \( R_v \) is the gas constant of the vapor, \( T \) is the absolute temperature, \( P_a \) is the absolute pressure of the air, and \( R_a \) is the gas constant of the air. Since some of the values in Equation 2.31 are equivalent, it can be simplified to the expression

\[
\omega = \frac{P_v / R_v}{P_{da} / R_{da}}
\]  

(2.32)

Since the gas constants will remain the same regardless of pressure, Equation 2.32 can be further simplified to a function of the pressures of the vapor and air, which is found by using the expression

\[
\omega = 0.622 \frac{P_v}{P_{da}}
\]  

(2.33)

If the total pressure of the system is known as well as the vapor pressure and the pressure of the air is not, another relation can be used to find specific humidity. If these parameters are the only ones provided, absolute humidity can be found by using the equation

\[
\omega = \frac{0.622 P_v}{P_{tot} - P_v}
\]  

(2.34)
where $P_{tot}$ is the total pressure (4).

Though this method of quantifying the amount of water vapor present is air appears useful, it is important to notice that the mass of water vapor is compared to a unit mass of dry air. By the definition of dry air presented earlier, dry air represents air that has absolutely no water vapor content. In the case of pure dry air by definition, the specific humidity is zero. However, if water vapor is added to that unit mass of dry air, the specific humidity will steadily increase to a point at which the air can no longer hold more moisture. This point at which dry air can no longer hold moisture is known as saturated air (4). As a result, any moisture that exceeds the saturation point of air will condensate out as liquid. The amount of water vapor present in air that is determined to be saturated at a given temperature can be found by using Equation 2.34 by replacing $P_r$ with $P_g$, which is the saturation pressure of water at the given temperature (4).

Though the use of specific humidity provides a means of directly finding the amount of moisture content existent in a given mass of air, one may wonder how that relates to human perception of the amount of water vapor present in the air. During the summer months in the state of North Carolina, the amount of water vapor present in the surrounding air has a direct effect on human comfort levels. Air conditioning often is used as a means of not only decreasing the temperature of the air in a conditioned space, but also reduces the level of the
humidity in the room. If the humidity in the room is not controlled, people may still be uncomfortable and may perspire even if the temperature is reduced.

In order to quantify the level of water vapor present in air as it pertains to human perception, a parameter known as relative humidity is often used. The relative humidity of air at a given location determines the comfort level of the occupants of that space and is a ratio of the vapor pressure, $P_v$, and the saturation pressure, $P_g$ (4). The relative humidity, $\phi$, can be found by using the equation

$$\phi = \frac{P_v}{P_g}$$

(2.35)

where $P_g$ is found by using the equation

$$P_g = P_{\text{sat @}T}$$

(2.36)

Using combinations of Equation 2.34 and Equation 2.35, two further expressions can be derived to calculate humidity. The first of these equations relates the relative humidity as a function of the specific humidity and is given by the expression

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g}$$

(2.37)

The second equation derived from these relations presents specific humidity as a function of the relative humidity and is given by the expression

$$\omega = \frac{0.622\phi P_g}{P - \phi P_g}$$

(2.38)
Based on the amount of moisture content existing in the air, the relative humidity can range from as low as 0 for dry air to 1 for completely saturated air. However, the amount of water vapor that the air can hold varies depending on the temperature of the air. As a result, the relative humidity of air at a given locale changes with the temperature even in the event that the specific humidity remains the same (4).

One important relation when dealing with air that contains water vapor involves determining the enthalpy of the moist air. As with the humidity calculations, the amount of dry air present for these calculations does not change, but the amount of water vapor contained in that dry air does. Therefore, practical applications of finding the enthalpy of atmospheric air are typically carried out by being expressed per unit mass of dry air instead of the expected per unit mass of the air-water vapor mixture (4).

The first step in finding the enthalpy of atmospheric air involves finding the total enthalpy of the atmospheric air. The total enthalpy of atmospheric air is found by taking the sum of the enthalpy of the dry air and the enthalpy of the water vapor. This quantity can be found by using the expression

$$ H = H_{da} + H_v $$

(2.39)

where $H_{da}$ is the enthalpy of the dry air and $H_v$ is the enthalpy of the water vapor. Equation 2.39 can also be represented by the equation

$$ H = m_{da}h_{da} + m_vh_v $$

(2.40)
where \( m_{da} \) is the mass of the dry air, \( h_{da} \) is the specific enthalpy of the dry air, \( m_v \) is the mass of the water vapor, and \( h_v \) is the specific enthalpy of the water vapor. As stated previously, the typical convention of finding the enthalpy of atmospheric air involves taking the enthalpy per unit of dry air. Dividing Equation 2.40 by the mass of dry air in the system gives the specific enthalpy of the atmospheric air that is found by using the equation

\[
\frac{H}{m_a} = h = h_a + \frac{m_v}{m_a} h_v
\]  

(2.41)

Using the relation for absolute humidity from Equation 2.30 to simplify Equation 2.41 yields

\[
h = h_a + \omega h_v
\]  

(2.42)

Since \( h_v \approx h_g \), Equation 2.42 is often simplified to

\[
h = h_a + \omega h_g
\]  

(2.43)

It is also important to note that the ordinary temperature of the atmospheric air considered in these cases is often referred to as the dry-bulb temperature in order to differentiate it from the other forms of temperature considered for these type calculations (4).

Another method established by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) to find the enthalpy of air in English units commonly used in industrial applications involves using the expression

\[
h = 0.240T + \omega(1061 + 0.444T)
\]  

(2.44)

where 0.24 is the specific heat of air in BTU per lbm per °R, \( T \) is the temperature of the moist air in °F, \( \omega \) is the humidity ratio of the moist air in lbm water per lbm of dry air, 1061
represents the saturated vapor enthalpy of water at a temperature of 0°F, and 0.444 is the specific heat of water in BTU per lbm per °R (8).

### 2.4.3. Dew-Point Temperature

The dew-point temperature, $T_{dp}$, is defined as the temperature when condensation begins when air is cooled at constant pressure (4). The dew-point temperature is simply equivalent to the saturation temperature of the water at the vapor pressure and is represented by the equation

$$T_{dp} = T_{\text{sat} @ P_v}$$

(2.45)

Constant pressure cooling to the dew-point temperature is further illustrated in Figure 2.7.

![Figure 2.7: Constant Pressure Cooling and Dew-Point Temperature](image)

Close observation of Figure 2.7 shows that as air decreases in temperature at constant pressure, the pressure of the vapor also remains unchanged. As a result, the water vapor
present in the air (state 1) endures a constant-pressure cooling process until it intersects the saturated vapor line (state 2) (4). The point where these two lines intersect occurs at the dew-point temperature. Once the temperature drops below this point, vapor begins to condense out of the atmospheric air. Therefore, the accumulation of vapor in the air decreases, which in turn results in a reduction in $P_v$ (4). During the period of condensation, the air remains at 100 percent relative humidity. For the saturated air, the ordinary temperature and the dew-point temperature are equivalent (4).

In order to illustrate this process, a common example of constant pressure cooling condensation occurs when a cold drink is consumed on a hot and humid day. While the drink sits out in the hot, humid air, dew begins to form on the outside of the container. The formation of this condensation on the side of the container indicates that the temperature of the beverage is below the dew-point temperature of the surrounding air (4). An example of this scenario is shown with the water bottle in Figure 2.8.
Figure 2.8: Dew-Point Temperature Effect on Cold Drink

A closer view of the water droplets formed when the cold beverage sweats is shown in Figure 2.9.

Figure 2.9: Water Droplets Formed from Condensation
Using this example, the dew-point temperature of air in an area can be found by taking a container and adding ice and stirring until condensation begins to form on the outside of the container. The point when the condensation begins to form represents the dew-point temperature.

2.4.4. Adiabatic Saturation and Wet-Bulb Temperatures

Though using the dew-point temperature to find the relative or specific humidity of a given mass of air is an effective method, the dew point temperature may not always be available. Another method commonly used to find the relative and specific humidity of a given mass of air is related to an adiabatic saturation process. The system used to represent an adiabatic saturation process is a long insulated channel that contains a pool of liquid water (4). An illustration of this process is shown in Figure 2.10.

![Figure 2.10: Adiabatic Saturation Process](4)
The adiabatic saturation process depicted in Figure 2.10 begins with unsaturated air with a specific humidity, \( \omega_1 \), relative humidity, \( \phi_1 \), and temperature \( T_1 \). This steady stream of air passes through the channel and gathers water vapor, which increases its moisture content. As a result, the temperature of the air decreases because a portion of the latent heat of vaporization of the evaporating water arrives from the air (4). The air then exits a specific humidity, \( \omega_2 \), relative humidity, \( \phi_2 \), and temperature \( T_2 \). If the channel that the water resides in is designed to be long enough, the airstream exits the channel as saturated air and \( T_2 \) is known as the adiabatic saturation temperature (4). The representation of the adiabatic saturation process is illustrated in Figure 2.11.

![Figure 2.11: Adiabatic Saturation Process on T-s diagram](image)

If makeup water is supplied to the reservoir at a rate equivalent to the rate of evaporation, the channel can be treated as a two inlet steady flow system. Since this process involves no heat or work interactions, changes in kinetic and potential energy can be neglected (4). Once these parameters are established, a mass and energy balance can be performed on the system. The
result of the mass and energy balance yields equations for the inlet and outlet specific humidity. The inlet specific humidity of the adiabatic saturation process can be found by using the equation

\[ \omega_1 = \frac{c_p (T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}} \]  

(2.46)

where \( c_p \) is the heat capacity of the air, \( T_2 \) is the outlet temperature of the air, \( T_1 \) is the inlet temperature of the air, \( h_{fg2} \) is the enthalpy of evaporation of the exit, \( h_{g1} \) is the enthalpy of saturated vapor at the inlet, and \( h_{f2} \) is the enthalpy of saturated liquid at the exit. The energy and mass balance also presents the relation for the outlet specific humidity that can be found by using the equation

\[ \omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g1}} \]  

(2.47)

where \( P_{g2} \) is the saturated vapor pressure at the exit and \( P_2 \) is the exit pressure. Equation 2.47 is only true in the event that the relative humidity at the exit is equal to 100 percent. Therefore, the specific and relative humidity of air can be determined from measuring the pressure and temperature of air at the inlet and exit of an adiabatic saturator using the aforementioned equations (4).

Though using an adiabatic separator is another effective method of determining the specific and relative humidity of a given quantity of air, it is not a practical way of to find these parameters. A more practical approach involves finding the wet bulb temperature, \( T_{wb} \). The wet bulb temperature is found by using a thermometer whose bulb is covered with a cotton
wick saturated with water and to blow air over the wick (4). An illustration of this method is shown in Figure 2.12.

![Figure 2.12: Measuring Wet-Bulb Temperature](image)

The method used to measure wet-bulb temperature uses a similar principle to the one that defines the adiabatic saturation process. Another method to measure the wet-bulb temperature involves placing the wet-wicked thermometer in a holder that is attached to a handle and then rotating the holder rapidly by moving the thermometer instead of the surrounding air (4). One instrument commonly used for this purpose is known as a sling psychrometer. An example of a sling psychrometer is shown in Figure 2.13.
Observation of a sling psychrometer quickly indicates that a sling psychrometer also often has a dry-bulb temperature thermometer. This extra thermometer is often attached so that dry-bulb and wet-bulb temperatures can be measured simultaneously.

Although these devices are still used in some applications, technological advancements will more than likely make these devices obsolete in future applications due to the advent of more electronic measuring devices. These instruments that measure humidity read moisture content based on the change in capacitance of a thin polyester film as it absorbs water vapor and are often accurate to within 1 percent (4).
For most practical purposes, the wet-bulb temperature and the adiabatic saturation temperature are not equivalent. Although for air-water vapor mixtures at atmospheric pressure, the wet-bulb temperature happens to be about the same as the adiabatic saturation temperature (4). As a result, the wet-bulb temperature can be used in Equation 2.46 instead of the outlet temperature if the outlet temperature is not available.

### 2.4.5. The Psychrometric Chart

In most practical applications when time is a limiting factor, engineers often want to use swift methods to find various parameters of air during an analysis. One of the most common ways that engineers use to quickly find parameters if two independent intensive properties are known is the use of tabulated charts and graphs. For the purpose of finding parameters related to moisture content in air, the psychrometric chart is often used.

Psychrometric charts are typically tabulated with values at an established pressure, such as 1 atm. A basic schematic of the layout of a typical psychrometric chart is shown in Figure 2.14.
The locations of several parameters discussed in preceding sections are shown in Figure 2.14. The specific humidity is located on the vertical axis, the dry-bulb temperature is shown on the horizontal axis, and the curved line bounding the upper left portion of the chart represents the saturation line (where the relative humidity is 100 percent). Lines following the same contour as the saturation line and to the right of the saturation line represent tabulated values of relative humidity. Diagonal lines with different slopes in the chart represent tabulated parameters such as the wet-bulb temperature, enthalpy, and specific volume of the air.

One important revelation of the psychrometric chart is that points along the line of saturation have the same dry-bulb, wet-bulb, and dew-point temperatures. This occurrence is shown in Figure 2.15.
ASHRAE Psychrometric Chart No. 1 in SI units at 1 atm is shown in Figure C.1. ASHRAE Psychrometric Chart No. 1 in English units at 1 atm is shown in Figure C.2.
Chapter 3  Thermodynamic Compressor Analysis

Once the different thermodynamic properties for consideration are known, they can be applied to the compressed air system to perform a thermodynamic analysis. The following sections detail the thermodynamic analysis of an air compressor, and various types of cycles associated with the process of compressing air.

3.1.  Compressor Background

Compressors are used in various applications as a method of increasing the pressure of a fluid. Unlike their turbine counterparts that produce work from the work of the fluid on the blades, compressors require work input to increase the pressure of the fluid. Work is typically transmitted to these devices by means of an external source, such as a rotating shaft (4). Although compressors are similar in nature to pumps and fans, they also have various differences. A fan increases the pressure of a gas slightly and is primarily used as a method of mobilizing a gas (4). In contrast, a compressor has the capability of increasing the pressure of the same gas to much higher pressures. A pump works much in the same way as a compressor, but instead of gases pumps use liquid as the operating medium (4).

For thermodynamic considerations of compressors, heat transfer is considered to be negligible except in the case of additional cooling required for certain applications. Another consideration in the typical thermodynamic analysis of a compressor involve the change in
potential and kinetic energy of the fluid being assumed as negligible (Δke=Δpe=0). The change in kinetic energy for compressors is considered negligible due to the fact that these devices typically operate with velocities that are too low to cause significant change (4).

3.2. **Compressor Energy Balance**

A typical approach to compressor analysis from a thermodynamic perspective involves modeling the compressor as a steady-flow system. For a compressor operating in a steady flow process, if there is no change in time at any point the change in mass of the control volume, Δm<sub>cv</sub>, is 0 as well as the change in energy of the control volume, ΔE<sub>cv</sub>. Air is assumed to be an ideal gas if it is at a high temperature and low pressure in relation to its critical point values. Mass flow rate of air is considered to be constant throughout the process. Energy balance for a compressor is given by

\[
\dot{E}_{IN} - \dot{E}_{OUT} = \frac{dE_{system}}{dt}
\]  

(3.1)

where \( \dot{E}_{IN} \) is the rate of energy into the system, \( \dot{E}_{OUT} \) is the rate of energy out of the system and \( \frac{dE_{system}}{dt} \) is the change in energy of the system of a period of time. In the case of a steady state compressed air process, the right hand side of the equation is 0. Simplification of the steady state solution yields

\[
\dot{E}_{IN} = \dot{E}_{OUT}
\]

(3.2)

The rate of energy into the system and the rate of energy out of the control volume are then substituted into Equation 3.2 to provide the relation
Simplifying Equation 3.3 to find the work input required for a certain compressor process yields

\[ \dot{W}_{IN} + \dot{m}h_1 = \dot{Q}_{OUT} + \dot{m}h_2 \]  

(3.3)

where \( \dot{W}_{IN} \) is the work input to the compressor, \( \dot{Q}_{OUT} \) is the heat output of the compressor, \( \dot{m} \) is the mass flow rate of the fluid, and \( h \) is the enthalpy of fluid at different states in the process.

### 3.3. Reversible Steady Flow Work

The work done by a thermodynamic process depends on several factors. These factors consist of the path followed by the process and the properties at the end states of the process (4). Reversible movable boundary work of a process is given by an expression of fluid properties in the equation

\[ W_b = \int_{1}^{2} PdV \]  

(3.5)

where \( W_b \) is the boundary work completed during the process, \( P \) is the pressure of the fluid at the end states of the process, and \( dV \) is the change in volume of the fluid during the process. These reversible, otherwise known as quasi-equilibrium, process work interactions lead to the maximum work output for devices that produce work as well as minimum work input required for devices that consume work (4).
Taking into consideration that the positive direction is work from the system (output work), the energy balance of a device assumed to be operating in a steady flow, internally reversible process can be found by using the differential equation

$$\delta q_{rev} - \delta w_{rev} = dh + dke + dpe$$  \hspace{1cm} (3.6)

where $\delta q_{rev}$ is the change in reversible heat transfer, $\delta w_{rev}$ is the change in reversible work, $dh$ is the change in enthalpy during the process, $dke$ is the change in kinetic energy during the process, and $dpe$ is the change in potential energy during the process. The change in reversible heat transfer during the process can be further simplified by using the relations

$$\delta q_{rev} = Tds$$  \hspace{1cm} (3.7)

$$Tds = dh - vdP$$  \hspace{1cm} (3.8)

where $T$ is the temperature of the fluid, $ds$ is the change in entropy of the system, $v$ is the specific volume of the fluid, and $dP$ is the change in pressure of the fluid. Substituting Equation 3.7 and Equation 3.8 into Equation 3.6 yields

$$dh - vdP - \delta w_{rev} = dh + dke + dpe$$  \hspace{1cm} (3.9)

Simplifying Equation 3.9 to find an expression for the change in reversible work of the system yields

$$- \delta w_{rev} = vdP + dke + dpe$$  \hspace{1cm} (3.10)

Integrating Equation 3.10 to find the reversible work of the system yields

$$w_{rev} = \frac{2}{1} vdP - \Delta ke - \Delta pe$$  \hspace{1cm} (3.11)
Since the changes in kinetic and potential energy are assumed to be negligible in these type processes, Equation 3.11 reduces to

\[ w_{rev} = -\int_{1}^{2} vdP \]  

(3.12)

Since Equations 3.11 and 3.12 are used to find the work output of a reversible, steady flow process Equation 3.12 can be rewritten as

\[ w_{rev, in} = \int_{1}^{2} vdP \]  

(3.13)

An important factor to not forget in these calculations involves the similarities between the quantities \( vdP \) and \( PdV \). However, these quantities must not be confused because \( PdV \) is a component of reversible boundary work in a closed system (4). An illustration of a steady-flow system similar to that of a compressor for the reversible case is shown in Figure 3.1.

![Figure 3.1: Steady-Flow System](4)

An example of a closed system used for reversible work analysis is a piston cylinder device as shown in Figure 3.2.
In order to perform the integration required by Equations 3.11, 3.12, and 3.13 for the reversible work of a system, the specific volume as a function of the fluid pressure must be known. One case to consider in this type of analysis is the case that the working fluid is incompressible, which means that the specific volume of the fluid remains constant. In this case, the term can be moved out from the integration (4). This simplification results in the equation

\[ w_{rev} = -v(P_f - P_i) - \Delta ke - \Delta pe \]  

(3.14)

3.4. **Proof that Steady-Flow Systems Deliver the Maximum and Consume the Minimum Work when the Process is Reversible**

Consider two devices operating in a state of steady flow, one of which is reversible and the other that is not. Next, consider that these processes operate with the same inlet and exit condition of the fluid. In the case that heat transfer to the system and work done by the
system are positive quantities, an energy balance for the reversible and irreversible processes are found by using Equation 3.6 above and Equation 3.15 shown below, respectively (4).

\[ \delta q_{\text{act}} - \delta w_{\text{act}} = dh + dke + dpe \]  

(3.15)

Since the right hand side of Equations 3.6 and 3.15 are equivalent, the two equations can be related through the expression

\[ \delta q_{\text{act}} - \delta w_{\text{act}} = \delta q_{\text{rev}} - \delta w_{\text{rev}} \]  

(3.16)

or the expression

\[ \delta w_{\text{rev}} - \delta w_{\text{act}} = \delta q_{\text{rev}} - \delta q_{\text{act}} \]  

(3.17)

However, from Equation 3.8 it is known that the change in reversible heat transfer is given by the relation \( Tds \). Substituting Equation 3.7 into Equation 3.17 and dividing both sides by \( T \) results in the relation

\[ \frac{\delta w_{\text{rev}} - \delta w_{\text{act}}}{T} = ds - \frac{\delta q_{\text{act}}}{T} \geq 0 \]  

(3.18)

Simplifying the relation in Equation 3.18 yields

\[ ds \geq \frac{\delta q_{\text{act}}}{T} \]  

(3.19)

Since \( T \) is the absolute temperature of the working fluid, it is always positive (4). Therefore, the change in reversible work is given by the relation

\[ \delta w_{\text{rev}} \geq \delta w_{\text{act}} \]  

(3.20)

and

\[ w_{\text{rev}} \geq w_{\text{act}} \]  

(3.21)
which proves that the reversible work of the system is the maximum work output or minimum work input that the system is capable of producing or using. An illustration of a reversible process for a turbine is illustrated in Figure 3.3.

![Figure 3.3: Reversible Turbine (4)](image)

The same holds true for compressors as turbines in this case except for the fact that $w_{rev}$ will be less than the $w_{act}$ that is required to complete the compression. An illustration of a reversible compressor is shown in Figure 3.4.

![Figure 3.4: Reversible Compressor (4)](image)
3.5. Minimizing Compressor Work

The reversible work required to power a compressor is given by Equation 3.12. An obvious method of minimizing the work input required for a compressor process would be to simulate the process as an internally reversible process as much as possible from thermodynamic limitations. This internally reversible process would be approached by minimizing the irreversibilities of the process, such as friction, turbulence, and nonquasi-equilibrium compression (4). One significant limiting factor in the pursuit of minimizing compressor work as much as possible is economic considerations. Considerable costs are associated with the research and development that would lead to minimized compressor work. These costs would further increase the cost of the end compressor to facilities using them. If the payback of the energy savings associated with such an efficiency grade is not in a realistic frame of time for the company, they will not be as inclined to make the effort to pursue maximum efficiency for compression processes.

Another method of reducing the work input required by a compressor is to maintain the specific volume of the gas as small as possible during the process of compression, which is a more practical approach (4). Since the specific volume of the gas is proportional to the temperature, the compressor work is minimized by maintaining the temperature of the gas as low as possible during combustion (4). As a result, the most practical approach to minimizing the work input to a compressor involves removing the heat of compression, thus decreasing the temperature of the fluid as it is compressed.
In order to completely understand the effects of decreasing the temperature of fluid during compression, the work requirement of several processes can be analyzed. The three processes commonly analyzed for this type of process are isentropic, polytropic, and isothermal. Assuming that all three of these processes take place with the same inlet and exit pressures \((P_1=P_2)\) in an internally reversible manner and the gas used in the compression process behaves as an ideal gas \((Pv = RT)\) with constant specific heats, the work required for compression is based on the integration detailed in Equation 3.12. Under these assumptions, expressions are developed to find compressor work. An isentropic process is a process which involves no cooling and is given by the equation

\[ P_0^k = \text{constant} \]  \hspace{1cm} (3.22)

where \(P\) is the pressure of the fluid, \(v\) is the specific volume of the fluid, and \(k\) is the ratio of specific heats of the fluid. Performing the integration of Equation 3.12 on this type process results in the compressor work equation for a isentropic process

\[ w_{\text{comp, in}} = \frac{kR(T_2 - T_1)}{k - 1} = \frac{kRT_1}{k - 1} \left[ \left( \frac{P_2}{P_1} \right)^{k-1} \right] - 1 \]  \hspace{1cm} (3.23)

A polytropic process is one in which compression involves some cooling and is given by the equation

\[ P_0^n = \text{constant} \]  \hspace{1cm} (3.24)

where \(n\) is a polytropic constant. The result of the integration from Equation 3.12 for a polytropic process is given by the equation
An isothermal process is one in which maximum cooling occurs and is given by the equation
\[ P_v = \text{constant} \]  
(3.26)

The result of integrating Equation 3.12 to find the compressor work for an isothermal process yields
\[ W_{\text{comp, in}} = RT \ln \frac{P_2}{P_1} \]  
(3.27)

An illustration of these three processes is shown on a \( P - v \) diagram in Figure 3.5.

![Diagram showing Compressor Processes](image)

**Figure 3.5: Compressor Processes**  (4)

The area to the left of each of these curves is the work done by the compressor process (result of the integral in Equation 3.12). Therefore, this area under the curve represents a measure of the steady-flow compression work done during the process. From this diagram, several observations can be made. Comparatively, the isentropic process requires the maximum
amount of work and the isothermal process requires the minimum amount of input work for the compressor. The work required for the polytropic process falls between these two values and depends on the value of the polytropic constant. The amount of work in this case decreases as the polytropic exponent decreases by increasing the amount of heat rejected during the compression process (4). If a sufficient amount of heat is removed during the compression process, the value of the polytropic constant approaches unity and it becomes an isothermal process (4). A common method of achieving the cooling of the fluid during the compression process is to use cooling jackets that envelop the casing of the compressors.

3.6. Multistage Compression with Intercooling

Based on the ideas presented in Section 3.5, it is apparent that the desired reduction in power input requirements for compressors is achieved best by cooling the working fluid of the compressor. Though the idea of simply reducing the temperature by removing heat from the compression process would seem to be simple, it is not often possible to generate an adequate amount of cooling of the fluid through the casing encompassing the compressor. As a result, additional processes are required to provide adequate cooling to the process. One technique often used in this type of process is referred to as multistage compression with intercooling. In this process, the gas is compressed over several stages and is cooled between each stage by passing it through a heat exchanger known as an intercooler. This cooling process preferably occurs in a state of constant pressure. At each of these intercoolers, the gas to be compressed is cooled to an initial temperature $T_1$. This process is highly desirable
when a gas needs to be compressed to very high pressures (4). This intercooling process has a profound effect on the work required for compression. An illustration of this process for a two-stage compressor is shown on a $P$-$v$ diagram in Figure 3.6.

![P-v Diagram for a Two-Stage Steady-Flow Compression Process](image)

**Figure 3.6: P-v Diagram for a Two-Stage Steady-Flow Compression Process**  (4)

An illustration of the process on a $T$-$s$ diagram is shown in Figure 3.7.

![T-s Diagram for a Two-Stage Steady-Flow Compression Process](image)

**Figure 3.7: T-s Diagram for a Two-Stage Steady-Flow Compression Process**  (4)
During this process, gas is compressed in the first stage from $P_1$ to an intermediate pressure labeled as $P_x$, cooled in a state of constant pressure to its initial temperature $T_1$, and then compressed in the second of the two stages to the final pressure designated $P_2$ (4). A common method used to model the compression processes involved in this cycle involves treating the process as one that is polytropic. The value of the polytropic constant, $n$, for this process typically varies between the ratio of constant specific heats, $k$, and 1. The work input required by the compressor that is saved during the process is outlined in Figure 3.6 by the shaded region. Figure 3.6 and Figure 3.7 also detail the process paths for single-stage isothermal and polytropic processes as a basis for comparison (4).

The actual amount of compressor work saved by this process varies as a direct result of any changes in the intermediate pressure, $P_x$. A significant challenge in the design of multistage compression with intercooling involves the exact design parameters necessary for the process in order to achieve the maximum possible amount of work input saved. A general expression for the total input required for a two-stage compressor is given by the equation

$$w_{\text{comp,in}} = w_{\text{comp,I,in}} + w_{\text{comp,II,in}}$$

(3.28)

where the inlet work required for the multistage process is broken down into the work input required by stage I and stage II. Substituting relations for this work from Equation 3.25 into Equation 3.28 to find the inlet work required for multistage compression yields the expression
59

The only varying parameter in Equation 3.29 that can be used to minimize the work input required for the compressor is the intermediate pressure, $P_x$. The $P_x$ value that minimizes the total amount of work can be found by differentiating Equation 3.29 with respect to $P_x$ and setting that expression equal to zero (4). Performing this operation yields the equation

$$P_x = (P_1P_2)^{1/2}$$

or

$$\frac{P_x}{P_1} = \frac{P_2}{P_x}$$

Therefore, in order to minimize the amount of compression work done during two-stage compression, the ratio of pressures across each stage of the compressor must be equivalent (4). In this case, the amount of input work required for each stage of compression is equivalent ($w_{\text{comp I,in}} = w_{\text{comp II,in}}$).

### 3.7. Isentropic Efficiencies of Compressors

An important consideration in the thermodynamic analysis of compressors is the isentropic efficiency. The isentropic efficiency of a compressor is defined as the ratio of required work input to raise the pressure of a gas to a specific amount using isentropic methods to the actual work input (4). This type of efficiency can be found by using the equation
\[ \eta_c = \frac{\text{Isentropic Compressor Work}}{\text{Actual Compressor Work}} = \frac{w_s}{w_a} \quad (3.32) \]

An important item to observe in this expression involves the input required for an isentropic compression process to be in the numerator rather than the denominator. This value is intentionally placed in the numerator due to the fact that \( w_s \) is always smaller than \( w_a \) for a compression process because input work is considered for compressors. The isentropic efficiency for a turbine is found by dividing actual work by isentropic work. These relations prevent the isentropic efficiency of the process from being greater than 100 percent. If this were not the case, then the expression would falsely imply that actual processes are more efficient than isentropic processes (4). It is also important to note that this type of efficiency uses the same inlet conditions as well as the exit pressure of the gas are equivalent for the actual and isentropic compressors considered.

As with many other cases in the thermodynamic analysis of closed, steady-state systems, the changes in kinetic and potential energies are considered to be negligible. Using this assumption, the work input of an adiabatic compressor reduces to the change in enthalpy of the compressor from the inlet to the exit (4). This relation can be used in Equation 3.32 to find the isentropic efficiency by using enthalpies with the expression

\[ \eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \quad (3.33) \]

where \( h_{2s} \) is the isentropic enthalpy at state 2 of the compressor process, \( h_{2a} \) is the actual enthalpy of the process at state 2 of the compressor process, and \( h_1 \) is the enthalpy at state 1.
of the compressor process. The values in Equation 3.33 are illustrated on the $h$-$s$ diagram shown in Figure 3.8.

![Diagram of Actual and Adiabatic Compressor Processes](image)

Figure 3.8: $h$-$s$ Diagram of Actual and Adiabatic Compressor Processes  (4)

Close observation of Figure 3.8 reveals once again that the value of isentropic efficiency is directly influenced by the design of the compressor. As a general rule of thumb, well-designed compressors have an isentropic efficiency of 80 to 90 percent (4).

### 3.8. Isothermal Efficiency of Compressors

In the case that no efforts are made to cool the gas as it is compressed, the actual compressor process is almost adiabatic. Therefore, the reversible adiabatic, or isentropic, process is a suitable representation of the ideal compressor process. However, there are often occurrences when compressors are cooled intentionally through the use of fins or a water jacket that may
be placed around the casing in an effort to reduce the work input required for the desired compression (4). This scenario is illustrated in Figure 3.9.

![Intentional Compressor Cooling](image)

**Figure 3.9: Intentional Compressor Cooling** (4)

In this scenario, the model of an ideal compression process as isentropic is not a suitable representation because the device no longer acts as adiabatic. Since the process cannot be modeled as isentropic, the isentropic efficiency defined by Equation 3.32 and Equation 3.33 becomes meaningless (4).

A more accurate representation for compressors that are intentionally cooled during the process involves one that is referred to as a reversible isothermal process. In this case, an isothermal efficiency can be easily defined by the equation

$$\eta_c = \frac{w_r}{w_a}$$  \hspace{1cm} (3.34)

where \(w_r\) is the work input required for compression in the isothermal case.
Various types of air compressors are used to meet the compressed air demand of industrial manufacturing plants. The most common types of air compressors used in industry today are reciprocating, rotary screw, and centrifugal. Each of these compressor types are chosen based on the application and size requirements of the facility it serves. These compressors generate and supply air for various processes and pneumatic equipment.

4.1. Types of Air Compressors

Several different types of air compressors are currently used for industrial purposes. These types of compressors are often generally categorized as either positive displacement or dynamic. Positive displacement compressors are identified by their ability to increase the pressure of a gas while decreasing the volume (9). Positive displacement compressors are further broken down into what are known as reciprocating or rotary screw compressors. Dynamic compressors typically encompass compressors that increase the kinetic energy of air, which in turn is converted to increased pressure energy (9). These compressors often refer to compressors that are also known in the industrial sector as centrifugal compressors. Air compressors are also further broken down into subcategories, which are illustrated in Figure 4.1.
Figure 4.1: Air Compressor Types
4.1.1. Reciprocating Air Compressors

Many common air compressors used for industrial applications are reciprocating compressors. Reciprocating compressors operate much in the same way as the pistons in an internal combustion engine and are the simplest method used in modern industrial practices to meet compressed air demands. Air is compressed in a piston cylinder device with one inlet and one exit. A cross section of a typical reciprocating air compressor used in industry is shown in Figure 4.2.

![Figure 4.2: Reciprocating Air Compressor](10)

The first step in this process involves the piston positioned at top dead center, otherwise known as TDC, where there is minimal volume between the top of the piston head and the top of the cylinder. However, there is a relatively small amount of air existing between the
The next stage of the compression process for a reciprocating compressor involves the expansion of the volume in the cylinder by means of the piston moving downward from the force caused by the rotation of the crankshaft. The expansion of the cylinder volume creates a vacuum inside of the cylinder. Air is taken in at the intake valve (referred to as the suction valve in Figure 4.3) where the atmospheric pressure acting on the top of the valve is greater than the gage pressure that resides inside of the cylinder. This difference in pressure between the cylinder and the top of the valve results in the valve opening and air to enter and fill the cylinder. An illustration of this second stage of compression in a reciprocating compressor is shown in Figure 4.4.

**Figure 4.3: Air Compressor Piston at TDC** (10)
The piston continues in a downward motion, expanding the volume of the air inside of the cylinder. This expansion end once the piston reaches the lowest point of its trajectory that is referred to as bottom dead center, or BDC. At BDC, the difference in volume between the total volume above the cylinder at BDC and the clearance volume is known as the displacement volume. The difference between the clearance volume and the displacement volume is shown in Figure 4.5 with the displacement volume illustrated on the left and the clearance volume illustrated on the right.
An illustration of the air compressor piston at BDC during this stage of the process is shown in Figure 4.6.

![Diagram of air compressor piston at BDC](image)

**Figure 4.6: Air Compressor Piston at BDC (10)**

Once the piston reaches BDC, the crankshaft continues to rotate and the ascension of the piston back towards TDC begins due to the force exerted on it by the crankshaft. Since the volume of the air in the cylinder decreases, the pressure of the air in the cylinder begins to increase as a result of Equation 2.10 if the air that is compressed in the cylinder is treated as an ideal gas. An illustration of this stage of the compression process is shown in Figure 4.7.

![Diagram of reciprocating air compressor compression](image)

**Figure 4.7: Reciprocating Air Compressor Compression (10)**
The final stage of compression results in the piston returning to TDC, where the compressed air is exhausted and delivered to the desired location. The air that is delivered from the exhaust valve (referred to as discharge valve in Figure 4.3) exits the cylinder when the proper difference in pressure is achieved as was the case with the intake valve. The pressure differential at which these valves open and close is often regulated by the installation of a spring that exerts a restoring force on the valves after they have opened in order to close them again. Alterations in how the springs open and close can be made for these compressors by simply changing the springs. An illustration of this final stage is shown in Figure 4.8.

![Final Stage of Reciprocating Compressor](image)

**Figure 4.8: Final Stage of Reciprocating Compressor**  (10)

Once the piston reaches TDC again, the same process is repeated as long as there exists a demand for compressed air from the compressor. The full cycle of a reciprocating compressor is shown in Figure 4.9.
Observation of Figure 4.9 reveals that these stages are often combined to form the two strokes of the compression process. The first of these two strokes is known as the suction strokes, which encompasses the original location of the piston at TDC, the expansion stage, and the arrival of the piston at BDC. Once the piston begins to move upward from BDC, the compression stroke occurs, which includes the compression stage and exhaust.

As shown in Figure 4.1, several different types of reciprocating compressors are used for industrial applications. The first of these reciprocating compressors is known as a reciprocating single acting compressor. Reciprocating single acting compressors are typically designed to act in either one or two stages. These compressors may use oil or another fluid to provide lubrication or are designed to not use fluids for lubrication. The single stage single acting compressor operates by drawing in air from the atmosphere and increasing its
pressures in a single stroke (11). An illustration of a single stage single acting reciprocating compressor is shown in Figure 4.10.

![Single Stage Single Acting Compressor](image)

**Figure 4.10: Single Stage Single Acting Compressor** (11)

Single stage single acting compressors such as the one shown in Figure 4.10 are generally used to generate pressures in the range of 30 psi to 70 psi.

The process of compression in a two stage single acting reciprocating compressor is slightly more complicated than the single stage reciprocating compressor. The two stage compressor uses the concept of multistage compression with intercooling presented in Section 3.6 by compressing the air to an intermediate pressure and passing the air through an intercooler before it is compressed to the final state. An illustration of a two stage single acting compressor is shown in Figure 4.11.
Single acting reciprocating compressors have a variety of practical applications. Common facilities that use these type compressors include automotive and truck repair shops, body shops, service businesses, and industrial plants (11). Most of these compressors that are used are lubricated by oil. However, some facilities such as hospitals and laboratories require the use of oil-free compressors as to reduce the possibility of oil contamination (11). An example of an oil-free single acting screw compressor used for these purposes is shown in Figure 4.12.
Another type of reciprocating compressor often used for industrial practices is referred to as rocking piston type compressors. These compressors are a slight variation from typical reciprocating compressors. These compressors generate pressure in air by using a single mounted piston and connecting rod where the piston head rocks as it reciprocates (11). Due to the motion of the piston, low friction rings are used and do not require oil for lubrication. These pistons are generally smaller in size than other reciprocating compressors are generally capable of relatively low pressures when compared to other types (11). An illustration of a rocking piston type compressor is shown in Figure 4.13.

![Figure 4.13: Rocking Piston Type Compressor](image1)

Diaphragm type reciprocating compressors can also be used to supply compressed air to industrial facilities. These compressors are another slight variation from the standard reciprocating compressor. This type of compressor generates pressure by means of a reciprocating or an oscillating action of a flexible disc that is actuated by an eccentric (11). This design also does not require lubrication because the moving parts of the diaphragm type compressor do not come into contact with one another. The type compressors are desired in
locations such as hospitals and laboratories where no contamination is allowed (11). An example of a diaphragm type compressor is shown in Figure 4.14.

Figure 4.14: Diaphragm Type Compressor  (11)

The current method used to provide power for reciprocating air compressors involves the use of electric motors. In the past, steam has also been used as a source of power to deliver the force needed to operate these compressors. The amount of electrical energy required to supply enough power to operate these motors is a significant factor in choosing a reciprocating compressor to meet small scale compressed air needs. As an example, a of 550 lbf be exerted over the course of a distance of one foot for every second would require a 1 hp electrical motor. This size motor is needed because the definition of a single horsepower is 550 foot pound force per second (550 ft-lbf/sec). The size of the motor needed for compressed air application varies directly with this factor.
There are also several disadvantages associated with reciprocating compressors. One of these disadvantages is that the up and down motion of the piston inside of the cylinder often causes a significant amount of vibration. As a result, reciprocating compressors must be used in locations where they can be stabilized. If they are not secured to a stable platform, premature compressor failure may occur. Another disadvantage of reciprocating compressors involves their limitation in size. Most reciprocating air compressors only exist in the 1 hp to 50 hp range, though some larger reciprocating compressors are available. Larger reciprocating compressors can be in the size range of 200 hp to 300 hp. These larger reciprocating compressors were more prevalent in industrial applications until the past 10 to 15 years. Due to their limitation in size, reciprocating compressors are not as widely used for larger scale applications.

### 4.1.2. Rotary Compressors

Another type of air compressor commonly used for industrial purposes is rotary compressors. These type compressors are desirable for use in manufacturing applications because they are compact, relatively inexpensive, and require a minimum amount of maintenance when compared to other types of compressors of the same size (10). For most small scale applications in industry today, a rotary compressor is preferred over a reciprocating compressor of the same rated capacity. Rotary compressors (primarily screw compressors) are preferable for these situations because rotary compressors occupy a fraction of the space.
and weight of a reciprocating device of equivalent size (10). Three general groups of rotary compressors are currently used. These groups are classified as slide vane-type, lobe-type, and liquid seal ring-type (10). Each of these types of rotary compressors has applications where one type may be preferred over another.

The first general type of rotary screw compressor used in industry is the slide vane-type compressor. These compressors have longitudinal vanes that slide radially in a slotted rotor, which is mounted eccentrically within a cylinder (10). Various types and sizes of these compressors are available. These include single and multi-stage units and versions that may or may not be oil lubricated (11). The choice of whether to use oil lubricated or oil free rotary slide vane type compressor relies primarily on the size and application required for compressor operation.

Oil-free compressors are often limited in their use to low pressure applications because of their high operating temperatures and sealing difficulties (11). However, the expense of oil and oil maintenance is alleviated by using oil-free air compressors. Another advantage to using oil lubricated slide vane-type compressors is the capability of achieving higher pressures (11). Although most industrial applications require approximately 90 psi to 100 psi to operate effectively, some specialized processes require much higher pressures. There are several advantages associated with using a sliding vane type rotary compressor. These
advantages include a smooth and pulse-free air output, compact size, relatively low noise levels, and low vibration levels (11).

An illustration of a rotary slide vane-type air compressor is shown in Figure 4.15.

Figure 4.15: Rotary Slide Vane-Type Compressor  (10)

Observation of Figure 4.15 shows that air enters a slide vane rotary compressor through an air inlet where it enters a cell created by the longitudinal vanes against the sides of the cylindrical case. As centrifugal force causes the rotor to rotate and approach the discharge point of the compressor, the size of the cells created by the longitudinal vanes decreases in size. As a result, the air is compressed and exhausted at the compressor outlet. Further illustration of this type air compressor is shown in Figure 4.16.
The next type of rotary compressor is the lobe-type rotary air compressor. This type of compressor operates through the mating of two lobe-type rotors that are mounted in a case. An illustration of a lobe-type air compressor is shown in Figure 4.17.

Figure 4.17 shows that air enters a rotary lobe compressor at the cavity created by the lobes that is greatest. Since the air enters at the larger cavity, it is compressed as it is moved by the lobes into the smaller cavity on the outlet end of the compressor. The lobes of a rotary lobe
air compressor are gear driven, but do not induce metal-to-metal contact between the lobes (10). Since the lobes do not come into direct contact with one another, a significant advantage is provided because the lobes will not wear out as quickly.

Another type of rotary air compressor commonly used in industrial applications is known as rotary liquid seal ring-type compressors. These type compressors feature a forward inclined open propeller in an oblong shaped cavity that is filled with liquid (10). An illustration of a rotary liquid seal ring-type compressor is shown in Figure 4.18.

![Rotary Liquid Seal Ring-Type Air Compressor](image)

**Figure 4.18: Rotary Liquid Seal Ring-Type Air Compressor** (10)

As the inclined propeller rotates in a rotary liquid seal ring-type compressor, the centrifugal force generated results in the liquid to seal the edge of the oblong cavity. Due to the oblong shape of the housing, the cells created by the impeller begin as large cells and reduce in size as the impeller rotates (10). Therefore, as is the case in design of other rotary air compressors, the air inlet is strategically placed at the point in rotation of the impeller where
the cells are the largest and the outlet where the cells are smallest. This type of design constraint results in maximum compression of the air brought into the compressor.

Rotary liquid seal ring-type air compressors are generally reserved for more specialized applications that their slide vane-type and lobe-type counterparts. These type compressors are more commonly utilized for the compression of extremely corrosive and exothermic gases, as well as in nuclear plants for the purpose of establishing an initial condenser vacuum (10).

As part of the three general categories of rotary air compressors, more specific designs are utilized for manufacturing purposes. Two common types currently used in industry are rotary screw compressors and rotary scroll compressors. Rotary helical screw compressors utilize the concept of using the shape typically associated as a screw for construction purpose as the rotors on a much large scale. This design uses two helical rotors that are oriented in intermeshing positions with one another. An illustration of the interior rotors of a helical rotary screw compressor is shown in Figure 4.19.

![Figure 4.19: Rotary Helical Screw Compressor](11)
As shown in Figure 4.19, the two rotors are designated as the male and female rotor. For a single-stage design such as the one shown in Figure 4.19, the air inlet is typically located at the top end of the compressor where the drive shafts are located. As with other types of rotary air compressors, the screw shaped rotors draw in air from the air inlet into a cavity created by the lobes of the rotors.

As the rotation of the rotors continues, the tips of the rotors pass the edges of the inlet ports and trap the air in a cell that is formed by the cavities of the rotor and the cylinder wall (11). The air continues to travel downward as a result of the moving cavity created by the lobes of the rotors, and the volume of the air cavities decrease. The volume of the cavities continues to decrease, and the pressure of the air in those cavities increases as a direct result of the ideal gas law (Equation 2.10). Oil is then injected after each closing of a cell. The oil that is injected aids in providing a seal for the cell and to provide a way to reduce the heat on the lobes (11). This process continues until the outlet of the compressor where the oil and compressed air mixture is discharged from the compressor.

Multi-stage versions of the compressors are also available. These compressors may also use water as a means of lubrication, or may have no lubrication at all. Once again, the amount and type of lubrication for the compressor depends on the size and application of the compressor. Several benefits are directly associated to using rotary helical screw compressors. These advantages include smooth and pulse-free output, compact size, higher
output volumes, low vibrations, prolonged maintenance intervals, and longer life of the compressor (11). Helical rotary screw compressors are typically the most common observed in industrial manufacturing applications.

Another type of rotary compressor used for small industrial applications are scroll-type compressors. The compression of air within a scroll takes place through the interaction of a fixed and an orbiting helical element that progressively compresses air (11). An illustration of a rotary scroll compressor is shown in Figure 4.20.

![Figure 4.20: Rotary Scroll Compressor](image)

The process of the interaction of the fixed and orbiting rotor continuously repeats to produce pulsation-free compressed air (11). One significant advantage of this compressor design over the rotary screw compressor and other types of rotary compressor is the decreased number of moving parts as a result of the fixed rotor. Fewer moving parts in this compressor results in
reduced maintenance costs. As with other rotary compressors, these compressors can be lubricated or lubrication free.

4.1.3. Centrifugal Compressors

Many large scale industrial applications require a much large compressed air capacity than reciprocating or rotary compressors can provide. For most of these scenarios, a centrifugal air compressor represents the preferred option. The original centrifugal compressors designed for larger operations were actually meant for the purpose of moving large volume of low pressure air (typically no more than 40 psi) (10). However, these air compressors have been developed to carry large volumes of air at high pressures through the use of multiple stages. These large volumes of air can be transported as pressures up to 3,500 psi (10). Even though centrifugal compressor can handle these high pressures, industrial applications do not require pressures that are this high in magnitude. Instead, most centrifugal compressors applied to an industrial setting are designed to generate medium volumes of air at medium pressures (10).

Centrifugal air compressors operate in a similar manner to centrifugal pumps. An illustration of a simplified centrifugal pump is shown in Figure 4.21.
As with a centrifugal pump, centrifugal air compressors utilize centrifugal force to generate motion. Instead of entering the compressor at an inlet to the lobes in the same way as rotary compressors, centrifugal compressors have an air intake located at the center, or eye, of the impeller. This location is designated as point $D$ in Figure 4.21. As centrifugal force causes the impeller to rotate, the air is thrown against the casing of the compressor (10). This motion of air is illustrated by the arrows that represent air movement in Figure 4.21. The compression of air in these type air compressors is accomplished as more and more air is thrown against the casing of the compressor by the impeller. One significant advantage associated with this design is the smooth delivery of compressed air to the application (10). The design of the impeller blades on these compressors varies based on the design of that compressor. Designs for these blades include forward and backward curves among many other designs and orientations (10). Other similarities that centrifugal compressors and pumps share include multiple stages that can be used in the production of higher pressures.
Centrifugal compressors are used for a variety of purposes. These applications include pneumatic control devices, pneumatic sensors, pneumatic valve operators, pneumatic motors, and starting air for diesel engines (10). A centrifugal compressor may be dedicated to individual equipment or may be used to provide compressed air to multiple areas.
Chapter 5 Reducing the Cost of Compressed Air

A variety of manufacturing processes in industrial facilities rely on compressors to provide air at high pressures. Compressed air at gage pressures typically in the range of 80 to 150 psig (550 to 1000 kPa) is widely used in these type facilities for numerous tasks including cleaning, operating pneumatic equipment, and even refrigeration (4). In comparison with other major utilities used for manufacturing processes, compressed air is often referred to as the fourth utility after electricity, water, and natural gas or oil (4).

The need for compressed air at these high pressures presents a multitude of energy saving opportunities because there is a prevalent waste of energy through compressed air. Employees at these facilities are also often unaware of the potential ramifications of wasting small amounts of compressed air during the course of their individual shift. These workers often turn on additional compressors when more compressed air is needed for plant processes. Although operating more than the minimum number of compressors provides sufficient air to meet this demand, turning on additional compressors results in significant energy costs. Plant management is also often unaware of these issues as well as the significant savings potential possible with compressed air systems.

A considerable amount of the energy wasted in the process of compressing air for industrial purposes can be avoided simply through commonsense methods.
5.1. Repairing Air Leaks in Compressed Air Lines

The most typical form of energy losses observed in energy assessments of manufacturing facilities involves air leaking from lines used to deliver compressed air to plant processes. All air that escapes compressed air lines represents energy losses proportional to the amount of energy required to compress the air to the desired pressure. In order to recuperate the compressed air lost through leaks in lines, the compressor requires more input work to provide the same amount of compressed air required to complete manufacturing processes. Studies indicate that up to 40 percent of the compressed air used at manufacturing facilities is lost in the form of air leaks in extreme cases (4). However, the reduction of air leaks in compressed air lines does not come without a cost. Air leaks appear constantly in these lines and would require constant monitoring of the system in order to eliminate these losses entirely. Due to the economic considerations of implementing such a program, regular maintenance of compressed air leaks is a more practical approach to this issue. Therefore, an air leakage rate of approximately 10 percent within manufacturing facilities is considered acceptable (4). An illustration of a compressed air leak is shown in Figure 5.1.

![Compressed Air Leak Illustration](image)

Figure 5.1: General Compressed Air Leak (4)
As shown in Figure 5.1, air leaks in compressed air lines most commonly occur at joints and connections where lines come together. Other common areas for air leaks to occur include flange connections, elbows, reducing bushes, sudden expansions, valve systems, filters, hoses, check valves, relief valves, extensions, and other equipment that is connected to compressed air lines (4).

Maintenance personnel responsible for checking air leaks should be familiar with these type connections. This knowledge prepares those performing a compressed air leaks survey for common trouble areas. The first of these connections mentioned is a flanged connection. A flange connection (inlet or discharge) refers to a method of connecting the casing to piping intended for inlet or discharge piping by using ends that can be bolted (flanged) (12). An example of a flanged connection kit commonly used for air compressors is shown in Figure 5.2.

![Flanged Connection](image)

**Figure 5.2: Flanged Connection**  (13)

Another common fitting where airleaks may occurs are referred to as elbows. Elbows are connections often used to connect pipe that requires the piping change direction. These
elbows must also be able to receive the type of pipe connected to it. An illustration of a 90° elbow with one male end is shown in Figure 5.3.

![90° Elbow with One Male End](image)

**Figure 5.3: 90° Elbow with One Male End**  (14)

An illustration of a 90° elbow union is shown in Figure 5.4.

![90° Elbow Pipe Union](image)

**Figure 5.4: 90° Elbow Pipe Union**  (14)

Another fitting commonly used for compressed air lines are reducing bushes. An example of a reducing bush is shown in Figure 5.5.

![Reducing Bush](image)

**Figure 5.5: Reducing Bush**  (15)
Sudden expansions in piping occur when the diameter of piping increase substantially. Sudden expansions may occur where two pipes of varying size or joined, or at the inlet of the compressor or a compressed air storage tank. An illustration of a sudden expansion is shown in Figure 5.6.

![Figure 5.6: Sudden Expansion](image)

Valve systems are often used as a means of regulating the flow of compressed air through a system. An example of a compressed air valve is shown in Figure 5.7.

![Figure 5.7: Compressed Air Valve](image)
As part of an air leak maintenance program, it important to monitor filters where leaks may occur. One method recognized by the Compressed Air and Gas Institute (CAGI), is to replace filters systematically rather than as they fail. CAGI recommends that filters should be inspected and replaced at the same time to ensure the quality of air being compressed as well as preventing pressure drops in the compressed air lines (17). CAGI also mentions the importance of replacing all filters associated with compressed air processes. These filters include air-line and point-of-use filters within the facility, which are equally as important to maintain as filters on the compressor and in the compressor room (17). An example of a compressed air filter is shown in Figure 5.8.

![Compressed Air Filter](image)

**Figure 5.8: Compressed Air Filter** (18)

Leaks in compressed air hoses also contribute to energy loses in manufacturing facilities. An example of an air hose used for compressed air is shown in Figure 5.9.
These places where air leaks typically occur can be the result of a variety of factors. For instance, expansion and contraction as a product of thermodynamic cycling and vibration represent common causes of loosening that takes place at the junctions of air lines (4). An effective method of preventative maintenance for the loosening of these connections is to periodically monitor the joints for tightness. Another good practice as part of this measure involves tightening the lines periodically.

Another important consideration when surveying compressed air systems for leaks is the end point of the compressed air line where the air is used. Air leaks commonly occur at these points or where lines are connected to equipment. These areas often experience air leaks more frequently because of the persistent opening and closing of the lines at these points, which wears out the gaskets quickly (4). These gaskets must be changed often to decrease the effects of air leaks at the point of use.

Numerous methods are commonly used in the detection of air leaks. The simplest form of detection of air leaks occurs when the observer listens to air lines and connections for the
sound of air escaping. This method typically reveals most large leaks in compressed air lines by listening for a hissing sound produced by the compressed air. The primary limitation involved with listening for air leaks is that leaks may be difficult to hear in facilities that have loud process equipment operating in close proximity to the compressor.

For smaller air leaks, a commonly used practice is to use soapy water on the compressed air line where the leaks are suspected to occur. If an air leak exists at the point where the soapy water is applied, bubbles will form. However, this practice becomes difficult for large compressed air systems that would require significant labor costs.

One method commonly used to detect air leaks in loud areas or large systems involves using an acoustic leak detector. An acoustic leak detector consists of a directional microphone, amplifiers, audio filters, and digital indicators (4). An example of an acoustic leak detector commonly used for compressed air systems is shown in Figure 5.10.

![Acoustic Leak Detector](image)

**Figure 5.10: Acoustic Leak Detector** (20)
For practical air leak survey applications, an acoustic leak detector is often considered the most efficient method. Acoustic leak detectors possess the ability to recognize the ultrasonic frequency hissing noises emitted by air leaks (21). This piece of equipment is simple to operate, but can vary significantly in cost and sensitivity. It is often recommended that these systems be tested before one is purchased for use at a facility.

Another component of implementing an air leak maintenance program by means of an acoustic leak detector involves the cost to the facility to train maintenance employees on the operation of the device. Once one maintenance worker has been trained on the operation of this device and a device is chosen, that worker can train other plant employees.

Several methods have also been developed for quantifying the total amount of air leak energy loss present in a compressed air system. One of these methods is to conduct a procedure known as a pressure drop test. This test is conducted by stopping all of the operations taking place in the facility that require compressed air and by shutting down the air compressors and closing the pressure relief valve (4). Closing the pressure relief valve is helpful in this procedure because it releases the pressure of the compressor automatically. If this process is followed, then any drop in pressure in the compressed air lines is directly associated with the cumulative effects of air leaks (4). Pressure drop in the compressed air lines is monitored over a set amount of time until the pressure reaches a specified value, which is usually half of the system pressure. Using this method enables the pressure to be kept above 30 – 40 psig.
Keeping the pressure above this range enables the flow to remain choked. The duration of time that passes while the air leaves the system is measured and recorded. Next, the decay of pressure as a function of the time duration is recorded (4). As a result, the flow rate of compressed air leaks in standard cubic feet per minute (scfm) can be found by using the equation

\[
m_{\text{leaks}} = \frac{V \times \Delta P}{t \times 14.7 \text{ psia}} \times 1.25
\]

(5.1)

where \(V\) is the volume in cubic feet, \(\Delta P\) is the change in pressure in psia, \(t\) is time in minutes, 14.7 psia is atmospheric pressure, and 1.25 is a constant that considers the discharge coefficient.

Once the pressure drop over time of the system is measured and recorded, the size of the compressed air system must be considered. The total volume, \(V\), of the compressed air system analyzed will consist of the compressed air tanks, headers, accumulators, and the primary compressed air lines (4). In order to provide a more conservative estimate of compressed air leaks, small lines are typically neglected. Leaving these lines out of the calculations also makes the calculations simpler, which saves plant employees’ time. Once all of these calculations are completed, the rate of an air leak can be found by using the ideal gas equation of state under the assumption that air behaves as an ideal gas. Another method that can be used to calculate air leaks involves using computer software. An example of commonly used software available for free download from the United States Department of energy is AIRMaster+. AIRMaster+ uses baseline data collected on a compressed air system,
as well as various current system parameters to predict the amount of energy savings that can be realized from several common energy saving projects including repair of compressed air leaks (22). This program allows a facility to perform air leak calculations easily. Training courses are available that can teach facility personnel how to implement and use the program to better enhance the efficiency of their compressed air system.

After the rate of the air leak is calculated, the amount of mechanical work wasted through air leaks can be determined. This amount of energy wasted as a unit mass of air escapes through leaks is equal to the actual amount of energy required to compress it (4). This work is determined as a form of Equation 3.25 and is modified to give the equation

\[
W_{\text{comp,in}} = \frac{W_{\text{reversible,comp,in}}}{\eta_{\text{comp}}} \quad (5.2)
\]

Simplifying Equation 5.2 with the second relation in Equation 3.25 yields

\[
W_{\text{comp,in}} = \frac{nRT_1}{\eta_{\text{comp}}(n-1)} \left[ \left( \frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] \quad (5.3)
\]

where \( n \) is referred to as the polytropic compression exponent. The value of \( n \) is equal to 1.4 for air when the compression is isentropic and varies between 1 and 1.4 when there is intercooling (4). The other term shown in Equation 5.2 and Equation 5.3, \( \eta_{\text{comp}} \), is known as the compressor efficiency. The compressor efficiency typically ranges between the values of 0.7 and 0.9 (4).
Next, principles of compressible flow theory can be used to analyze parameters of air leaks. Based on this theory it can be shown that whenever the pressure of the line is above 2 atm, which is typically the case, the velocity at air leak site must be equivalent to the local speed of sound (4). When this theory is applied, the mass flow rate of air escaping through air leaks of a minimum cross sectional area designated as $A$ can be found by using the equation

$$ \dot{m}_{\text{air}} = C_{\text{discharge}} \left( \frac{2}{k+1} \right)^{(k-1)} \frac{P_{\text{line}}}{RT_{\text{line}}} A \sqrt{kR \left( \frac{2}{k+1} \right) T_{\text{line}}} \quad (5.4) $$

where $k$ is the ratio of specific heats ($k = 1.4$ for air) and $C_{\text{discharge}}$ is a term used as a discharge or loss coefficient. This coefficient is used as a means of accounting for any imperfections that may affect the flow of air at the site of the leak. The value of the discharge coefficient varies in the range of about 0.60 for an orifice with sharp edges to 0.97 for a circular hole that is well rounded (4). In general, the site of an air leak is imperfect in shape and is difficult to model without complicated analysis and equipment that is too costly for facilities to justify. In the case that there is an absence of actual data for the air leak, the discharge coefficient can be taken as 0.65 for practical use (4). Using this value keeps the mass flow rate of air from being estimated as too high, and keeps the estimated air leak savings conservative. The other constants in Equation 5.4, $P_{\text{line}}$ and $T_{\text{line}}$, represent the pressure and temperature of the compressed air line.

Once the mass flow rate and inlet work required for the compressor have been calculated, the power that is wasted by the compressed air leak can be found by using the equation

$$ \text{Power Saved} = \text{Power wasted} = \dot{m}_{\text{air}} \, w_{\text{comp,in}} \quad (5.5) $$
It is important to note that the amount of power saved shown in Equation 5.5 only holds true if the compressor operates at full load all of the time, the compressor is an on-off compressor, or the compressor is a variable frequency drive (VFD) compressor. However, compressors used in most industrial applications do not follow this behavior. Instead, compressors used for industrial applications can operate unloaded for extended periods of time until compressed air is used. In this case, the unloaded power must be considered as well as the full load power in order to obtain accurate savings. Determining the amount of actual demand savings for an industrial compressed air system requires that a factor be used, which is typically denoted as $F$. Therefore, the actual demand reduction of compressors used for industrial applications can be found by using the power saved value from Equation 5.5 and the relation

$$DR = L \times CF \times (1 - F)$$  \hspace{1cm} (5.6)

where $DR$ is the demand reduction in kW, $L$ is the power saved in hp, $CF$ is a conversion factor (0.746 kW/hp), and $F$ is the fraction of full load power used while the compressor operates unloaded. Once the demand reduction is calculated, the amount of energy savings from reducing compressed air leaks for an industrial air compressor can be found by using the equation

$$ES = DR \times H$$  \hspace{1cm} (5.7)

where $ES$ represents air leak energy savings and $H$ represents hours of compressor operation per year. Energy cost savings are then determined by considering the demand cost savings and energy costs savings. These values will vary based on the energy provider of the facility.
5.2. Installing High Efficiency Motors

Nearly all air compressors used in industry today acquire the work input necessary by means of an electric motor. The electrical energy drawn by this motor for a specified power output is inversely proportional to the efficiency of the motor (4). Just as the compressor itself, the electric motor used to supply the input work is not capable of converting all of the electrical energy it consumes into mechanical input work used to drive the compressor. The ratio of this mechanical power that is supplied to the electrical power consumed during the operation of the electric motor is referred to as the motor efficiency, $\eta_{\text{motor}}$ (4). As a result, a relation of these parameters can be used to find the electric power consumed by the motor through the equation

$$W_{\text{electric}} = \frac{W_{\text{comp}}}{\eta_{\text{motor}}} \quad (5.8)$$

An illustration of an electric motor used to provide input work for compressors is shown in Figure 5.11.

![Electric Compressor Motor](image)

**Figure 5.11: Electric Compressor Motor** (4)
Table 5.1 shows the electrical power consumed per kW of mechanical, or shaft, power output at various motor efficiencies.

Table 5.1: Electrical Power Consumed at Various Efficiencies

<table>
<thead>
<tr>
<th>Motor Efficiency, $\eta_{\text{motor}}$ (%)</th>
<th>Electrical Power Consumed Per kW of Mechanical (Shaft) Power Output, $\dot{W}<em>{\text{electric}} = \dot{W}</em>{\text{shaft}}/\eta_{\text{motor}}$ (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>1.00</td>
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<tr>
<td>90</td>
<td>1.11</td>
</tr>
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<td>10</td>
<td>10.00</td>
</tr>
</tbody>
</table>

For example, consider a motor operating at an efficiency of 90%. The electric power consumed for this motor per kW of mechanical power output is $1/0.9$, or 1.11 kW of electric power is required for each kW of shaft power used by the compressor. Based on this information, high efficiency motors are far superior to their less efficient counterparts available on the market. However, the increase in efficiency that these motors provide to the compressed air process generally is much more expensive in initial purchase price to the facility. Typical motor efficiencies, rewind costs, and replacement cost are available from the MotorMaster Plus 4.0 database, which is a free program available from the U.S. Department of Energy (23). Motor efficiencies, rewind costs, and replacement costs for motors ranging in size from 5 hp to 125 hp are shown in the table below.
Table 5.2: Cost and Efficiency of Electric Motors

<table>
<thead>
<tr>
<th></th>
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In order to justify the purchase of higher efficiency motors at industrial manufacturing facilities, the electrical energy saved by using higher efficiency motors must offset the additional initial investment in a short period of time. This simple payback of initial expenditures typically takes place within a few years. Facilities also often continue to have their current lower efficiency motors rewound when this maintenance is required. However, each time the motor is rewound the efficiency of that motor decreases by 1 to 2 percent for a standard rewind and 3 to 5 percent for a rush job. A rush job occurs when the facility needs the motor to be rewound quickly. Due to the quick rewind of a rush job the magnets inside the motor experience higher temperatures, which results in a greater efficiency decrease. This decrease in efficiency further emphasizes the importance of high efficiency motors for
compressed air processes. These savings are especially evident for larger compressors that are used over multiple production shifts per day.

In order to justify the savings from replacing motors with power sources that are more efficient, the amount of electric power saved by the replacement must be determined. This electric power saved by replacing a motor that has an existing efficiency $\eta_{\text{standard}}$ with a new motor that has efficiency $\eta_{\text{efficient}}$ is found by using the equation

$$\dot{W}_{\text{electric,saved}} = \dot{W}_{\text{electric,standard}} - \dot{W}_{\text{electric,efficient}}$$  \hspace{1cm} (5.9)

Equation 5.9 can also be expressed in the form

$$\dot{W}_{\text{electric,saved}} = \dot{W}_{\text{compsaved}} \left(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}} \right)$$  \hspace{1cm} (5.10)

Another expression commonly used to find the electric work saved other than Equation 5.9 and Equation 5.10 is the equation

$$\dot{W}_{\text{electric,saved}} = (\text{Rated Power})(\text{Load Factor})(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}} )$$  \hspace{1cm} (5.11)

where the rated power is the nominal power of the motor that is listed on the label (power delivered by the motor at full load) and the load factor is the fraction of the rated power where the motor nominally operates (4). Once the electric work saved is found, then the total energy savings for installing high efficiency motors is found. The total energy savings can be determined by using the equation

$$\text{Energy Savings} = \dot{W}_{\text{electric,saved}} \times \text{Annual Operating Hours}$$  \hspace{1cm} (5.12)

where $\text{Annual Operating Hours}$ is the total number of hours per year that the facility is in operation. If the facility operates during all hours of the year, this value will be 8,760 hours.
The typical efficiency of electric motors used to provide power to air compressors varies from about 70 percent to over 96 percent (4). As with many other mechanical processes, the remaining percentage of power not converted to compressor work is lost to the surroundings as heat. Due to the wide range of electric motor efficiencies, the amount of energy lost from these motors in the form of heat varies proportionally to the efficiency of the motor at full load. When the compressor is not operating at full capacity, the amount of heat generated during the compressor process increases to higher levels. If this heat is not dispersed properly, system overheating may occur.

Another result of the heat generated by compressor electric motors is increased temperatures within the area where the compressor is located. This increased temperature can have both positive and negative effects if it is located within the confines of a facility. If a compressor is located in spaces of a facility that are conditioned throughout the course of the year, the higher temperatures are a welcome addition during winter months when space heating of the facility is needed. This advantage allows the facility to save small amounts of energy that would otherwise be necessary to heat the facility to the desired temperature. Still, the heat provided by the compressor is much less efficient ($\eta = 100\%$) than boiler heat or a heat pump.

However, a consequence of this benefit involves the temperature increase still taking place during the summer months when the conditioned space requires the removal of heat from
these spaces to decrease the temperature. In this case, the amount of air conditioning load required for these areas increases to account for the extra heat provided to the space from the compressor motors. This extra air conditioning requires more electrical energy to be used for the air conditioning system used at a facility.

As an example, consider a 75 kW (100 hp) rated electric compressor motor operating at an efficiency of 90 percent. Since this motor operates at 90 percent efficiency, it will consume 83.3 kW of electric power. 8.3 kW provided to the motor is lost in the form of heat. This heat generation is the equivalent of an 8.3 kW electric strip heater in the compressor room. The resulting 8.3 kW of electric heat produced into the area of the compressor room would raise the temperature to undesirable levels during warmer months of the year. If this higher temperature air is not vented properly and the air drawn into the compressor comes from the compressor room itself, the efficiency of the compressor itself will decrease (4).

Several important factors must be taken into consideration when selecting electric motors chosen for the purpose of air compression. These considerations include the operating profile of the compressor (the variation of compressor load with time) and the efficiency of the motor at partial load conditions (4). The partial load efficiency of the motor is often as important as the full load efficiency of the motor. This aspect is especially evident if the compressor is expected to operate at partial load for significant amounts of time (4). Typical compressor motors have a nearly flat efficiency curve from half load to full load, and the
peak efficiencies of these motors are typically designed for approximately 75% load (4). Below an operating point of about half load, compressor motor efficiency sharply decreases. Therefore, operating compressor motors above 50% compressor load should be a top priority for facilities using compressed air.

As an example, consider the motor efficiency curve shown in Figure 5.12.

![Figure 5.12: Efficiency of Electric Motor at Various Compressor Loads](image)

The efficiency of the motor shown in Figure 5.12 varies greatly from approximately 96% at 96% load to approximately 88% at half load. Notice in Figure 5.12 that once the compressor is operated at levels below half load, the motor efficiency begins to decrease exponentially. The motor efficiency for the compressor stays within an approximate 8% range between 50% load and 100% load, while the efficiency between 20% load and 50% load is within an efficiency range of approximately 38%.
However, these ranges vary for each different motor available on the market. For example, another motor may only drop from an efficiency of 85% at 96% load to an efficiency of 75% at 20% load. Though this motor is not as efficient at high load as the motor presented in Figure 5.11, this motor is more useful in applications where the motor operates at 20% load more often. These lightly loaded motors still draw a significant amount of current, and the power factor is low. One method that has been developed to improve the efficiency of these motors at partial load involves installing a variable frequency controller if it is economical (4).

A common engineering approach to designing systems to be used for various mechanical applications involves oversizing systems. For example, solid mechanics design typically involves the use of a factor of safety in calculations to oversize items to be used for construction such as steel beams and frames. This form of approach enables the engineer to account for any unforeseen circumstances where the force exerted on these structural members may far exceed the expected loads to be experienced by that structural member. However, when choosing a compressor to be used for industrial applications this method of caution cannot be used.

Oversizing a motor in a similar fashion for a compressor to be conservative for power input expectations seems like a desirable idea. However, this approach used to keep the motor from becoming overloaded from unforeseen circumstances can have numerous negative
ramifications. Using this practice to design a compressor motor will ensure that the motor almost always operates at lower loads, and hence lower motor operating efficiency (4). Another item to consider with this method of thinking is that an oversized system will bring with it an increase in initial system cost. Though, as long as oversized motors are operate at half load or higher where they operate in smaller ranges and higher efficiencies they waste only small amounts of energy.

Often, compressors motors are manufactured with a significant service factor. The service factor for these motors can range anywhere from 1.1 to 1.25. The service factor for and electric compressor motor can be found by using the expression

\[ SF = \frac{\text{Max Safe Capacity}}{\text{Rated Capacity}} \]  

(5.13)

where \( SF \) denotes the service factor and the ratio to find the service factor involves the relationship between the maximum safe capacity of the motor and the rated capacity of the motor.

### 5.2.1. Using a Smaller Motor at Higher Capacity

As described in Section 5.2 above, manufacturing plants tend to oversize the anticipated need of equipment at their facility. Larger equipment than necessary for the required processes is chosen for reasons including safety, or anticipated future business expansion (4). Compressors are not exempt to this method of thinking. One of the most influential components that influence this method of thinking at manufacturing facilities involves
uncertainty in operation. Many facilities are not sure when they purchase compressor motors about the production demands that will be required of them over time. In many of these scenarios, facilities opt to install the maximum amount of capacity they think they may need because they feel that it is better to pursue more capacity in the event they need it rather than meeting the current compressed air process needs of the facility.

Several factors contribute to this approach of thinking. If additional compressor capacity is needed immediately, the facility will already have the extra capacity available and will not lose the potential productivity increase while new compressor capacity is installed. Plant managers may determine that it is more economical of an investment if they purchase more capacity in the present if they feel that the implementation cost of compressor capacity will increase substantially in the near future.

The result of these economic considerations is that facilities often install a considerable amount of excess compressed air capacity. Sometimes, the compressor capacity at a facility is several times the required capacity of a facility with the perception that additional compressor capacity may be needed in the future (4). As a result, the compressor often operates only occasionally at full load or always at reduced loads.

Since an oversized motor operates at lower loads for a majority of the required operating time, the efficiency of the motor will decrease as the load of the compressor decreases as
shown in Figure 5.12. The result is a motor that consumes higher amounts of electricity per unit of power delivered to the compressor, which results in more expensive compressed air operations (4). The most significant savings that can be realized from these changes results from the unloaded power of the compressor. This unloaded power can be as much as 70% of the full load power of the compressor as shown previously in Figure 5.12. In order to reduce these operating costs, it is recommended that manufacturing facilities smaller compressor motors operating at full load in an effort to increase efficiency.

5.3. Using Outside Air for Compressor Intake

Another common method to reduce the cost of compressed air use at a facility consists of using outside air for the compressor intake. As mentioned before, the power consumed by a compressor is directly proportional to specific volume, which in turn is proportional to the absolute temperature of the gas (in the case of air compressors, air) at a given pressure (4). Equation 5.3 shows that the compressor work is also directly influenced by the inlet temperature of air supplied to the compressor. The resulting lower input air temperatures to the compressor will result in lower input work required to operate the compressor at the desired load.

As a result, a power reduction factor, \( f_{\text{reduction}} \), has been developed to determine how much input power is required for compression as the temperature decreases. If the intake air for a compressor is currently being taken from the inside of the facility, the intake can be ducted in
from outside of the facility where the air is typically cooler. The resulting relation for power factor reduction for moving the air intake of a compressor from the inside of the facility to the outside of the facility is represented by the equation

$$f_{\text{reduction}} = \frac{W_{\text{comp,inside}} - W_{\text{comp,outside}}}{W_{\text{comp,inside}}}$$  (5.14)

where the power reduction factor is determined by the compressor work required by inlet and outlet conditions. As noted above, since the compressor work required in Equation 5.3 is directly proportional to the inlet temperature of the compressor, Equation 5.14 can be directly reduced to a function of the inlet temperatures inside and outside of the building in the equation

$$f_{\text{reduction}} = \frac{T_{\text{inside}} - T_{\text{outside}}}{T_{\text{inside}}}$$  (5.15)

Equation 5.15 can be further simplified by dividing all terms on the right hand side of the equation to obtain the relation

$$f_{\text{reduction}} = 1 - \frac{T_{\text{outside}}}{T_{\text{inside}}}$$  (5.16)

It is important to note that when using the relations presented in Equation 5.14 and Equation 5.15 that absolute temperatures (K or R) are used or inaccurate results may occur. For example, reducing the absolute inlet air temperature by 5 percent will reduce the power input required for the compressor power input by 5 percent (4). One rule of thumb commonly used for a specified amount of compressed air is that the power consumption decreases (or for a fixed power input the amount of compressed air increases) by 1 percent for each 3°C (5.4°F)
drop in temperature of inlet air to the compressor (4). This rule of thumb comes from a base
temperature of 80°F, or 540°R.

Compressors used at manufacturing facilities are often located in an enclosed area. This area
may be in production areas of the facility, in a separate room within the main building, or in a
small outdoor building dedicated to housing the compressor alone. In most cases, the
compressor takes in air from the space in which it is located. These spaces are often at higher
temperatures than outside of the facility or small building due to a variety of heat sources.

If the compressor is located in production areas or a separate room within the facility, heat
from other process equipment used like boilers causes the indoor temperature to rise because
the other equipment operate with a mechanical efficiency as the motors that power the
compressors do. This method of heat gain also occurs in all of these compressor locations
due to the fact that the compressor itself produces heat from energy lost due to electric motor
and compressor efficiencies. Other potential sources of higher indoor temperatures than
outdoor temperatures are space heaters used to make workspaces more comfortable for
employees during winter months and boilers used to produce steam and heating for
manufacturing processes.

The air outside of the facility or compressor house generally provides an alternative to these
inlet conditions. Due to the factors described, the outside air is typically cooler and denser
than the air in the compressor room even on the hottest days of summer months (4). Outside air can be drawn in to the area where the compressor is located by means of an intake duct. The intake duct is then connected to the air intake of the compressor where outside air can be used instead of air inside of the area. An illustration of this method is shown in Figure 5.13.

![Figure 5.13: Outside Air Compressor Intake (4)](image)

Installing this type of system will reduce the amount of energy consumed by the compressor because less energy is required to compress cooler air than the same amount of air at a higher temperature. Using outside air for compressor intakes is also advantageous because during the winter months, energy used to heat the air within a facility is not wasted by being compressed in addition to the increased compressor efficiency.

Though drawing in outside air for compressors can result in significant energy savings, a facility should be aware that not all types of compressors experience these reductions in compressor energy consumption. This method of energy conservation does not hold true in the case of flooded oil screw compressors. These compressors are not affected by the
decrease in inlet air temperature because compression temperatures for oil flooded screw compressors are more a function of the inlet temperature of the oil providing lubrication rather than the temperature of the air itself.

5.4. Reducing the Air Pressure Setting

Another common area in compressed air systems of energy loss involves the desired compressed air pressure for use in the facility. A similar mindset to that of oversizing compressor capacity is that of establishing a compressor pressure setting that is much higher than necessary for manufacturing processes. A higher pressure setting may be chosen to be sure that the pressure delivered to the end process meets the requirements of that process. This fear may be a result of losses from different sources, such as compressed air leaks or head losses in the piping delivering the compressed air. However, if the compressed air piping system is sized large enough, the pressure drop in the compressed air lines will be minimal. Another explanation to higher pressures being used than necessary is that many plants combine compressed air lines to different processes that may have different compressed air needs. If this is the case, they may determine a single pressure to operate all of the compressed air at rather than purchase and install valves at each process to regulate the pressure.

Manufacturing facilities that operate with pressures much higher than needed greatly benefit from a compressed air analysis. Pressures may be set much higher than necessary because of
a change in production requirements and may not be performed often due to financial considerations. A compressed air analysis to determine the minimum operating pressure enables these facilities to reduce plant operating pressures to these levels and observe considerable energy savings (4). For both screw-type and reciprocating compressors, the operating pressure can be reduced by simply adjusting the pressure setting on the compressor to match the minimum pressure required.

In order to quantify the amount of energy saved by implementing this type of practice, the amount of energy required to compress a unit mass of air must be determined. This value can be found by using Equation 5.3. From this relation, it is important to note that the higher the pressure at the compressor exit ($P_2$), the larger the work will be that is required for compression (4). For the scenario of reducing the air pressure setting, a new pressure at the compressor exit, $P_{2,\text{reduced}}$, is used. This new outlet compressor pressure will result in a power reduction factor for the compressor as with using outside air that can be found by using the equation

$$f_{\text{reduction}} = \frac{W_{\text{comp, current}} - W_{\text{comp, reduced}}}{W_{\text{comp, current}}}$$  \hspace{1cm} (5.17)

where the power reduction factor is a function of the current work required by the compressor and the work required for compression if the pressure setting is reduced. Substituting the relation from Equation 5.3 into Equation 5.17 to find the power reduction factor as a function of the reduced pressure yields
\[ f_{\text{reduction}} = 1 - \frac{\left( \frac{P_{2,\text{reduced}}}{P_1} \right)^{\frac{n-1}{n}} - 1}{\left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1} \]  

(5.18)

For example, a power reduction (or savings) factor of \( f_{\text{reduction}} = 0.08 \) indicates that the amount of power consumed by the compressor is reduced by 8 percent by simply reducing the power setting of the compressor (4).

### 5.5. Replace Low Pressure Compressed Air Needs with a Blower

Low pressure (<15 psig) compressed air is required to meet the needs of some uses at manufacturing facilities. For these situations, a blower can be used to meet the same application. Low pressure blowers are capable of producing pressures up to 15-20 psig. A considerable amount of energy can be saved in this manner because a blower only requires a small fraction of the power that is needed by a compressor for a specified flow rate of air (4).

For example, consider an air knife used as a means of blowing air across a part produced as it enters or leaves a process. The air knife is used as an additional means of drying parts or cleaning debris from them before they are to be processes, painted, or powder coated.

### 5.6. Turning off Compressors When Not in Use

One simple measure to take that can save energy and money involves turning off the compressors when they are not in use. Plant employees at manufacturing facilities often do not realize the ramifications of leaving on equipment when unattended. This activity often
takes place during breaks, lunch hours, nights, or even over the duration of a weekend. A substantial amount of energy is wasted during these periods of time if the compressors are left running while no production takes place. These losses are especially evident in the case of screw-type compressors that are known to consume up to 85 percent of their rated power while unloaded (4). Reciprocating compressors are not as affected as their screw-type counterparts. However, reciprocating compressors are not immune to these types of power losses.

Since the compressors are still operating during times when they are left on and not used, power is still lost in reciprocating compressors based on the amount of compressed air lost due to compressed air leaks. There are two common methods used to prevent these losses from occurring. The first of these methods would be for someone at the facility to have the responsibility of manually powering down the compressors during times that production does not take place. The other method involves installing a controller and timer to shut off the compressors automatically since it is human nature to postpone responsibilities when the benefits are not obvious or instantaneous (4). Using the controller is generally the preferred method for these reasons. Timers are used when the compressor has operated long enough unloaded. Although, the timer should also have a manual override that enables the individual responsible for the system to turn it on in the case of compressor maintenance.
5.7. Use Refrigerated Dryers for Compressed Air

Refrigerated air dryers are also used as a method of reducing the cost of compressed air. These dryers are often used as a means of cooling the compressed air considerably below its dew point in an effort to condense and remove a significant fraction of the water vapor present in the air in addition to other incondensable gases such as oil vapors (4). As air is compressed, its temperature rises significantly. Sometimes the temperatures of compressed air may exceed 250°C (482°F) at the compressor exit when it is compressed adiabatically in a single stage to a pressure of just 700 kPa (102 psia) (4). As a result, it is recommended to cool air after it is compressed.

Cooling air after it is compressed minimizes the amount of power that is consumed by the refrigeration system (4). This concept is similar to that of allowing food to cool in a pan to the ambient temperature before it is placed in a refrigerator (4). This cooling can be performed by either the ambient air or by using water. Once this heat is drawn out of the compressed air, it can be used for a variety of other purposes, such as space heating during the winter, preheating of feed water to the boiler, or any other process related heating.

5.8. Use Waste Heat for Space Heating During Winter Months

Compressors typically are cooled by an external means in order to increase compressor efficiency and reducing input power requirements. This can be achieved by the circulation of
air or a liquid that acts as a heat exchanger. The heat that is drawn off of the compressor is then translated to the circulating fluid. The heat that is transferred to the fluid is then typically rejected to the ambient in a liquid-to-air heat exchanger (4). The heat that is rejected typically amounts to approximately 60 to 90 percent of the required input power, and thus represents a significant amount of energy that can be utilized for heating needs of the facility (4).

This heat rejected can be used for a variety of purposes including space heating for the facility during winter months, preheating air or water for a furnace, or other areas where fluids or gases require preheating for the process (such as preheating boiler feed water). An example of using the rejected heat to provide space heating during the winter months is shown in Figure 5.14.

![Diagram](image)

**Figure 5.14: Compressor Waste Heat** (4)
For example, under the assumption that 80 percent of the power input is converted to heat energy, a 150-hp compressor is capable of rejecting as much heat as a 90-kW electric resistance heater or a 400,000-Btu/h natural gas heater when the compressor is operating at full load (4). Therefore, appropriate usage of compressor waste heat is an excellent method of saving energy and finances.
Chapter 6  Compressor Air Inlet Temperature Analysis

Compressor air inlet temperature analysis involves a variety of components. Data gathered using data logging equipment as well as data provided by the subject facility was considered for this experiment. This data was then analyzed to determine its validity and applicability to this study on the effects of inlet air temperatures on industrial reciprocating and centrifugal air compressors.

6.1. Hypothesis and Experimental Overview

One objective of this experiment is to prove theoretically and experimentally that the amount of power necessary to operate reciprocating and centrifugal air compressors is directly impacted by the inlet air temperature. Since the compressed air demand of industrial manufacturing facilities remains relatively constant, the cost of producing compressed air can be a significant expense. Air drawn into the compressor at lower temperatures is denser than air at higher temperatures. As a result, less power is necessary to produce the same amount of compressed air.

The most common method of decreasing inlet air temperatures for reciprocating compressors involves moving air intakes from inside of an enclosed compressor room or building at a facility to the outside of the facility. This method is used because outside air temperature is typically cooler than the air present in the compressor room that is heated due to compressor
inefficiencies. All electricity not used for the compression of air is lost to the surrounding air as heat. These cooler outdoor temperatures result in less work needed by the compressor to produce the same amount of air. Using this method is also effective for applications involving centrifugal or oil free screw compressors. Oil-flooded screw compressors are affected minimally by a change in inlet air temperature because the oil has a greater effect on the operation of the compressor than the air temperature does.

Though taking in outside air is an effective method of decreasing the amount of energy used in reciprocating compressors, the temperature still varies based on the outdoor air temperature. During the course of a year, these compressor intakes have air temperatures that can vary from as low as in the teens during the winter to over 100°F during some of the hottest summer days. These differences in temperature present another challenge in the cost of compressed air. Since compressed air is still directly affected by the temperature, decreasing these temperatures to a lower more consistent temperature through use of an external system will provide additional savings. The ideal temperature for inlet air temperature depends on a variety of factors. One factor that must be considered is the freezing point because water vapor will be present in the air. Another factor to consider is the capability of equipment available to decrease the air temperature and the economic viability of the different options available.
The amount of energy and energy costs expected to be saved by cooling inlet air temperatures to reciprocating compressors can be modeled by deriving a relationship between the air temperature and the boundary work performed by the compressor. Beginning with Equation 3.5, air can be treated as an ideal gas. The result of the ideal gas relation presented in Equation 2.15 for pressure can then be substituted into the equation above, which yields

\[ W_{\text{comp, in}} = \int_{V_1}^{V_2} \frac{m_{\text{air}} R_{\text{air}} T_{\text{air}}}{V} dV \]  \hspace{1cm} (6.1)

Since the mass of air, gas constant of air, and temperature of air remain constant for a reciprocating compressor, the relation above simplifies to

\[ W_{\text{comp, in}} = m_{\text{air}} R_{\text{air}} T_{\text{air}} \left[ \frac{2}{1} \right] \frac{1}{V} dV \]  \hspace{1cm} (6.2)

Integration of the above expression with respect to the volume of air in the compressor yields

\[ W_{\text{comp, in}} = m_{\text{air}} R_{\text{air}} T_{\text{air}} \left[ \ln(V) \right]^{V_2}_{V_1} \]  \hspace{1cm} (6.3)

\[ W_{\text{comp, in}} = m_{\text{air}} R_{\text{air}} T_{\text{air}} \left[ \ln(V_2) - \ln(V_1) \right] \]  \hspace{1cm} (6.4)

\[ W_{\text{comp, in}} = m_{\text{air}} R_{\text{air}} T_{\text{air}} \ln \left( \frac{V_2}{V_1} \right) \]  \hspace{1cm} (6.5)

Depending on the temperature of air compressed, the relation above will produce different results. In order to compare the difference in power consumption between two different air inlet temperatures, the expression for power reduction factor outlined in Equation 5.14 can be used. In this case, the temperatures high and low will be used for comparison in the relation...
Substituting the results of Equation 6.5 into Equation 6.6 yields

\[ f_{\text{reduction}} = \frac{W_{\text{comp},T_h} - W_{\text{comp},T_i}}{W_{\text{comp},T_h}} \]  

(6.6)

Since the mass of air, gas constant of air, initial volumes, and final volumes of the two processes are the same regardless of air temperature, Equation 6.7 simplifies to

\[ f_{\text{reduction}} = \frac{m_{\text{air}} R_{\text{air}} T_{\text{air}} \ln\left(\frac{V_2}{V_1}\right)}{T_h} - \frac{m_{\text{air}} R_{\text{air}} T_{\text{air}} \ln\left(\frac{V_2}{V_1}\right)}{T_i} \]  

(6.7)

Cancelling out like terms yields

\[ f_{\text{reduction}} = \frac{m_{\text{air}} R_{\text{air}} \ln\left(\frac{V_2}{V_1}\right)(T_h - T_i)}{m_{\text{air}} R_{\text{air}} \ln\left(\frac{V_2}{V_1}\right)T_h} \]

(6.8)

As previously mentioned, only absolute temperatures can be used with this relation, or inaccurate results may occur. If data collected for an experiment involving this relation is not in absolute temperatures (Rankine or Kelvin), another version of Equation 6.9 can be generated for use with large data sets not collected in absolute units. For the English system, conversion of the temperature values to Rankine yields

\[ f_{\text{reduction}} = \frac{T_h - T_i}{T_g} \]  

(6.9)

\[ f_{\text{reduction}} = \frac{(T_h + 459.67) - (T_i + 459.67)}{T_h + 459.67} \]  

(6.10)
An important observation in the above relation yields that the temperature conversion factor will cancel out since the numerator is a temperature difference. The simplification of Equation 6.10 yields

\[
\text{f}_{\text{reduction}} = \frac{T_{h,F} - T_{i,F}}{T_{h,F} + 459.67} \tag{6.11}
\]

A similar conversion can be carried out if data is collected in SI units for Celsius temperatures and results in the relation

\[
\text{f}_{\text{reduction}} = \frac{T_{h,C} - T_{i,C}}{T_{h,C} + 273.15} \tag{6.12}
\]

In order to visualize the effects of air temperature on the amount of power consumed by compressors, a plot of power reduction versus inlet air temperature for a desired air temperature of 45°F is shown in Figure 6.1.

![Figure 6.1: Power Reduction versus Air Temperature before Cooling](image-url)
Based on the theoretical data, the amount of power consumed by reciprocating compressors is expected to increase by 0.18% per °F increase in temperature if a constant decreased inlet temperature is achieved.

6.2. Current Compressor Operation

A manufacturing facility located in the southeastern region of North Carolina agreed to a request to monitor the inlet air conditions of their existing compressed air system. The system currently used to provide compressed air for facility processes consists of 10 air compressors. However, only 5 of these compressors are used as a primary source of compressed air. The remaining 5 air compressors are used for backup purposes or when maintenance is performed on the primary compressors.

The current primary compressors used at this facility range in size from 600 hp to 700 hp. All 10 compressors are monitored by the facility in their power consumption and are denoted by numbers from 1 to 10. The primary compressors used are compressors 1, 2, 4, 7, and 8. These compressors are detailed in Table 6.1.
Table 6.1: Primary Compressors

<table>
<thead>
<tr>
<th>Compressor Number</th>
<th>Make</th>
<th>Size (hp)</th>
<th>CFM Delivery</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Worthington</td>
<td>600</td>
<td>3,172</td>
</tr>
<tr>
<td>2</td>
<td>Worthington</td>
<td>600</td>
<td>3,172</td>
</tr>
<tr>
<td>4</td>
<td>IR Centac</td>
<td>700</td>
<td>3,000</td>
</tr>
<tr>
<td>7</td>
<td>IR XLE</td>
<td>600</td>
<td>2,850</td>
</tr>
<tr>
<td>8</td>
<td>Joy WN</td>
<td>600</td>
<td>3,000</td>
</tr>
</tbody>
</table>

6.3. Measurement Procedure

In order to monitor the current operation of the compressed air system, several parameters were considered for measurements. These parameters were air temperature, relative humidity, compressor power consumption, and system air flow. Each of these parameters was collected over an extended period of time in order to observe changes in compressor power consumption with changes in temperature with consistent air flow demand.

6.3.1. Temperature and Relative Humidity

The first step involved in the experimental analysis of the effects of inlet air temperatures on reciprocating and centrifugal air compressors involved gathering temperature data over the course of several months. An initial meeting was arranged with plant personnel to become familiar with the compressed air system and current intake air system. During the tour of the system, it was observed that the facility already was using outside air for their compressors through intakes installed on the roof. These intakes are large pipes that extend out of the roof...
and then curve back down to prevent contamination from rainwater or any other object that may fall into them. A computer generated model of a compressor intake is shown in the figure below.

![Model of Compressor Intake](image)

**Figure 6.2: Model of Compressor Intake**

In order to evaluate the temperature of the air drawn into these intakes for compression, it was determined that two of these intakes would be used as points of data collection.

The first intake chosen (referred to in the data analysis as Compressor Intake #1) is a smaller intake located close to the edge of the roof above the compressor house and the second intake chosen (referred to in the data analysis as Compressor Intake #2) is a large intake toward the opposite edge of the roof of the compressor house adjacent to the main plant. Since the area
where air was drawn into the compressors is above the roof, it was determined that the data
logging equipment to be used would need to be affixed inside of the compressed air intakes
in order to generate the most accurate results.

The data loggers chosen for this application of gathering temperature and relative humidity
were HOBO® U12 temp/RH/light/ext channel data loggers similar to the one shown in Figure
6.3.

![Figure 6.3: Temperature and Relative Humidity Meter](image)

Before the loggers could be installed inside of the designated compressor intakes, they were
programmed to collect proper data at appropriate intervals. The software used with the
HOBO® U12 data loggers used in this experiment is Onset HOBOware Pro. Once the
software is opened, the data logger is connected to the computer with Onset HOBOware Pro
installed by means of a USB cable as shown in Figure 6.4.
After the program is opened and the data logger is connected to the computer, the program should recognize the logger with device identification in the bottom left corner of the screen as shown in Figure 6.5.

If the device is not recognized, make sure that the battery used by the logger is not dead or that there are no other issues with the logger, such as moisture. If the device is read properly, the device can be launched for data collection by clicking on Device and the Launch as shown in Figure 6.6.
Once at the launch screen, several options appear for data acquisition and the current status of the logger such as the battery level. For the purposes of this analysis, temperature and relative humidity were set to be recorded by the logger in five minute increments on a delayed launch similar to the options chosen in Figure 6.7 shown below.
Once the data loggers were programmed for the data desired, they were to be prepared for installation inside of the compressor intakes.

Placing these loggers inside of the compressor intakes required some ingenuity. Since the intakes vibrated, the magnets that can be used with these loggers such as the one shown in the figure below were not strong enough to keep the loggers from falling out of the compressor intakes and being rendered useless.

![Magnet Used to Affix HOBO Data Logger to Magnetic Surfaces](image)

**Figure 6.8: Magnet Used to Affix HOBO Data Logger to Magnetic Surfaces**

Instead, it was determined that the best means of affixing the loggers inside of the compressor intakes would require the use of zip ties such as the one shown in the figure below.

![Zip Tie for Data Logger Installation](image)

**Figure 6.9: Zip Tie for Data Logger Installation**
In order to install the zip tie inside of the data logger, the back of the logger must be removed by extracting the screws as shown in Figure 6.10.

**Figure 6.10: Temperature and Relative Humidity Metter with Back Removed**

Once the back of the data logger is removed, the zip tie is run through the openings on the removed portion of the logger as shown in Figure 6.11.

**Figure 6.11: Installation of Zip Tie**

After the zip tie has been installed, the back of the data logger can be replaced and the resulting data logger for use inside of the air intake is complete as shown in Figure 6.12.
Once the loggers were preassembled with zip ties, they were taken to the two predetermined intakes where they were installed by fastening the zip ties around the framing members of the interior of the compressor intake as modeled in Figure 6.13.
Once these loggers were installed inside of the compressor intakes, a location was chosen on the ground outside of the compressor house for comparison to the temperatures taken inside of the compressor intakes. This location was chosen as a point of reference to see if the roof was the best location for the outdoor compressor intakes as well as to confirm that the temperature values acquired on the roof were reasonable. The first location chosen for this reference point was on top of a small concrete wall that sees direct sunlight for most of the day. Later on, a point was chosen under an overhanging vent to gain a different perspective that is discussed in more detail later. The outside temperature data collected was completed by using a HOBO® U12 Outdoor/Industrial data logger similar to the one shown in Figure 6.14.

![Outdoor Temperature Meter](image)

**Figure 6.14: Outdoor Temperature Meter**
This logger was chosen for this application rather than the ones used in the compressor intakes due to exposure to the weather that the outdoor temperature logger would experience. In order to record the outdoor temperature, this data logger uses a temperature probe similar to the one shown in Figure 6.15.

![Temperature Probe](image)

**Figure 6.15: Temperature Probe (24)**

This probe is installed on the data logger by removing the faceplate using an Allen wrench such as the one shown below, and removing one of the channel ports as shown in Figure 6.17.

![Allen Wrench](image)

**Figure 6.16: Allen Wrench for Removing Outside Logger Faceplate**

![Installing Temperature Probe](image)

**Figure 6.17: Installing Temperature Probe**
Once the probe connection is run through the port for the data collection channel shown in the figure above, the faceplate and side port seals must be replaced. Then the logger can be placed in the location where the desired data is to be collected. This data logger is programmed using the same method as with the other loggers through the removable USB port shown in the following figure.

![USB Port on Outdoor Temperature Logger for Launch and Data Retrieval](image)

**Figure 6.18: USB Port on Outdoor Temperature Logger for Launch and Data Retrieval**

Once all of the data loggers were installed in the desired locations, they were initially left for one week. After that week was over, data was collected from the installed data loggers. Since the loggers were placed in equipment locations that are difficult to take a laptop computer with to directly download data to, an Onset HOBO® U-Shuttle was used to gather the data, and then transfer the data to a computer later. The Onset HOBO® U-Shuttle (Model # U-DT-1) used for this experiment is shown in Figure 6.19.
After these data files were downloaded to the computer with Onset HOBOware installed, they were viewed to ensure that the data loggers were still collecting data and that the data being collected was reasonable. The shuttle uses USB technology with ports compatible with either end of a standard USB data transfer cable. The connection of this cable to the Shuttle in preparation for extraction of data from a data logger is shown in the figure below.
Once the USB cable is connected to the Shuttle, the smaller end is connected to the data logger in the compressor intake as modeled in the figure below.
Once the shuttle is connected to the logger and turned on, a screen will appear that asks whether the user desires to find a device or acquire Shuttle information. For this application, the Find Device function is used. Once the data logger is found by the Shuttle, the user indicates that they desire for the data to be downloaded to the Shuttle. Once the download of data is complete, the Shuttle can be used to relaunch the data logger with the existing parameters set forth when the logger was initially launched. Then, the Shuttle will relay important information to the user such as logger battery level to make the user aware of how much battery life the logger has remaining. The user can then indicate whether they desire to change the battery or not. Once this process is complete, the Shuttle indicates that it can be safely removed from the logger.

After the data was downloaded from each logger to the Shuttle, the Shuttle was connected to the laptop computer the HOBOware in the same manner as the data logger shown previously. Once the Shuttle is connected to the USB port of the computer with the proper software, the Shuttle must be turned on in order to communicate with the computer. Once this process is complete, the indicator at the bottom left of the HOBOware program will look similar to the figure below.

![Figure 6.22: Onset HOBOware Shuttle Device Identification](image-url)
Once the Shuttle is recognized by the HOBOware program, the data can be downloaded by going to Device and Manage Shuttle as shown in the figure below.

![Manage Shuttle for Data Download](image1)

**Figure 6.23: Manage Shuttle for Data Download**

After navigating to the Manage Shuttle option, the screen displayed in the following figure should appear.

![U-Shuttle Management](image2)

**Figure 6.24: U-Shuttle Management**
From the screen shown in the figure above, the user can choose which downloaded data on the Shuttle is desired for download to the computer. Once the files for extraction are chosen by clicking or unclicking the boxes to the left hand side of the U-Shuttle Management pane, the user will click Offload Checked for the data extraction process to begin. Once these files are offloaded, a save destination on the computer is designated by the user where the data files can be easily retrieved. These files appear as a .hobo extension, and can be only opened in the HOBOware program. Once this file is located in the location where the user designated it be saved, HOBOware opens the datafile and provides options to plot the data logged as shown in the figure below.
From the Plot Setup screen, the user can designate which of the sets of data they desire to be best for a data plot as well as various parameters for the plot. Once these parameters are set, the user will click on the Plot button to produce a plot of the data similar to the one shown in the figure below for a sample week of temperature and relative humidity data collected for this experiment.
The next step involved in data acquisition involved converting the files to a more user friendly format. Data collected by the data loggers and uploaded to the HOBOware program place the data in a .hobo file extension, which already has predetermined formatting of data. In order to make the data compatible with Microsoft Excel for more detailed analysis, the files were exported to .csv file extensions that can be opened in Microsoft Excel. This export is achieved by selecting File and then Export Table Data as shown in the figure below.
Once the Export Table Data option has been chosen from the File menu, the screen depicted in the following figure appears.
From this screen, users have options of the type of export they desire. For this experiment, all data files converted were exported to a single file. Once this option was chosen, the export button was clicked and the .csv file was saved to the directory designated by the user. Once the file format has been converted, it was opened in Microsoft Excel for further analysis.

Once the data files were collected and converted to an acceptable Excel format, the data was used for various analyses. After the first data set was deemed to be adequate and reasonable, the data loggers were left at the facility for several weeks between collections of data sets.

As another parameter to ensure that data collected was reasonable, temperature data was collected from the National Weather Service for Fayetteville Regional Airport (25). However, the data available online for these values is only given in hourly increments as opposed to the 5 minute increments taken by the data loggers. Since the data loggers were programmed to begin taking data at even hourly measurements and the weather data available was given at the 53rd minute of each hour, the basis of comparison for these values is the one provided by the national weather service and the value taken by the data loggers at the 50th minute of each hour. An example of the data available from the National Weather Service is shown in Figure 6.29.
6.3.2. Compressor Power Consumption and System Flow

The next important parameter in this analysis is the amount of power consumed by the compressors. The original idea of obtaining these values involved using a current transformer and data logger to measure the amount of current consumed by the compressors over an extended interval of time such as the one shown in Figure 6.30.
However, the compressors at this facility draw power at 2,400 volts. This high voltage prevented us from directly monitoring the power consumption of the compressors because our equipment is rated for 600 volts or less. The potential hazards associated with this endeavor resulted in seeking another method of gathering this data. Once it was determined that using current transformers in the panels delivering electricity to the compressors was unsafe while they were in operation, plant personnel offered to provide us their measured power of the compressors that was monitored over 15 minute intervals in watt-hours. These values provide an accurate representation of the amount of power consumed by the compressors in their current state of operation as well as the amount of electrical demand they require. The amount of demand used by the compressors indicates how much of the rated capacity of the air compressor is being used over a given amount of time as long as the compressors are running at full load all of the time. An example of the data provided by the facility for power used at an individual data point for each of the ten compressors is shown in Table 6.2.

### Table 6.2: Power Consumption Data

<table>
<thead>
<tr>
<th>Device</th>
<th>Description</th>
<th>Time Logged</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>32</td>
<td>Powhouse Cmp #1</td>
<td>11/2/2012 0:00</td>
<td>40,045,704,000</td>
</tr>
<tr>
<td>33</td>
<td>Powhouse Cmp #2</td>
<td>11/2/2012 0:00</td>
<td>41,953,076,000</td>
</tr>
<tr>
<td>34</td>
<td>Pwrhouse Cmp #3</td>
<td>11/2/2012 0:00</td>
<td>154,035,000</td>
</tr>
<tr>
<td>35</td>
<td>Pwrhouse CMP #4</td>
<td>11/2/2012 0:00</td>
<td>5,878,968,000</td>
</tr>
<tr>
<td>36</td>
<td>Powhouse Cmp #5</td>
<td>11/2/2012 0:00</td>
<td>0</td>
</tr>
<tr>
<td>37</td>
<td>Powhouse Cmp #6</td>
<td>11/2/2012 0:00</td>
<td>10,346,006,000</td>
</tr>
<tr>
<td>38</td>
<td>Powhouse Cmp #7</td>
<td>11/2/2012 0:00</td>
<td>43,479,631,000</td>
</tr>
<tr>
<td>39</td>
<td>Powhouse Cmp #8</td>
<td>11/2/2012 0:00</td>
<td>42,020,788,000</td>
</tr>
<tr>
<td>40</td>
<td>Powhouse Cmp #9</td>
<td>11/2/2012 0:00</td>
<td>1,119,462,000</td>
</tr>
<tr>
<td>41</td>
<td>Powhouse Cmp #10</td>
<td>11/2/2012 0:00</td>
<td>9,505,325,000</td>
</tr>
</tbody>
</table>
Once these values were collected, they were sorted by Time Logged and then filtered by Device number into the watt hour readings for each individual compressor. The watt hour readings (Value) given were instantaneous totalized readings of the energy used by that compressor since the meter was installed. Therefore, the difference in these values for each compressor from one point in time to another shows the amount of energy consumed by the compressors during that time interval. For example, daily energy analysis consisted of taking the midnight reading from one day and subtracting it from the midnight reading of the next day to determine the energy consumed by the compressor for that day. In cases where midnight compressor energy data was not available, the first available time was taken. Since the energy was calculated on a daily basis, the compressor demand was found by taking the total daily energy use and dividing by 24 hours. These values were then converted to kWh and kW for the final comparison.

The final parameter for consideration in this analysis was the amount of compressed air flow generated by the system on a given time period. Another challenge was posed with this data acquisition because recording flow would require purchasing equipment and then having to leave that equipment in the system because the system is only down a few times per year. Fortunately, the facility was also monitoring compressed air flow. However, the only flow measurements that the facility had available were total daily flow for the entire compressed air system. As a result, the final analysis of the data would involve only comparing daily average temperatures, total daily power consumption, and the total daily flow that the facility
could provide. An example of the flow measurements provided by the facility over a period of a few days in thousand standard cubic feet (mscf) is shown in Table 6.3.

### Table 6.3: Compressed Air System Daily Air Flows

<table>
<thead>
<tr>
<th>Date/Time</th>
<th>A1 MSCF</th>
<th>A2 MSCF</th>
<th>B1 MSCF</th>
<th>Mod 1 MSCF</th>
<th>Mod 2 MSCF</th>
<th>Mod 3 MSCF</th>
<th>B2F MSCF</th>
<th>Total MSCF</th>
</tr>
</thead>
<tbody>
<tr>
<td>5/18/12 11:00 PM</td>
<td>4,639</td>
<td>1,204</td>
<td>8,717</td>
<td>700</td>
<td>1,550</td>
<td>2,030</td>
<td>3,710</td>
<td>22,550</td>
</tr>
<tr>
<td>5/19/12 11:00 PM</td>
<td>4,931</td>
<td>1,307</td>
<td>9,301</td>
<td>730</td>
<td>1,560</td>
<td>2,080</td>
<td>4,181</td>
<td>24,090</td>
</tr>
<tr>
<td>5/20/12 11:00 PM</td>
<td>4,656</td>
<td>1,324</td>
<td>8,807</td>
<td>690</td>
<td>1,490</td>
<td>2,040</td>
<td>3,683</td>
<td>22,690</td>
</tr>
<tr>
<td>5/21/12 11:00 PM</td>
<td>4,797</td>
<td>1,292</td>
<td>9,277</td>
<td>710</td>
<td>1,530</td>
<td>2,070</td>
<td>4,104</td>
<td>23,780</td>
</tr>
<tr>
<td>5/22/12 11:00 PM</td>
<td>4,721</td>
<td>1,258</td>
<td>9,246</td>
<td>730</td>
<td>1,610</td>
<td>2,060</td>
<td>4,135</td>
<td>23,760</td>
</tr>
<tr>
<td>5/23/12 11:00 PM</td>
<td>4,668</td>
<td>1,245</td>
<td>9,268</td>
<td>720</td>
<td>1,620</td>
<td>2,050</td>
<td>4,129</td>
<td>23,700</td>
</tr>
<tr>
<td>5/24/12 11:00 PM</td>
<td>4,402</td>
<td>1,251</td>
<td>8,998</td>
<td>740</td>
<td>1,550</td>
<td>2,050</td>
<td>4,109</td>
<td>23,100</td>
</tr>
</tbody>
</table>

### 6.4. Data

As the data sets were collected, they were analyzed individually before being combined into the final product. The following sections detail the raw data collected from the data loggers and facility personnel.

#### 6.4.1. Temperature and Relative Humidity

The first data collected and analyzed for this experiment was the temperature and relative humidity data collected from the compressor intakes and the area outside of the compressor house. In order to make data easier to manage, it was determined that data sets would be
broken up into one week sets. These sets were broken up by dates that data was collected into sets of six to eight days of data, and later sets were combined to make sets exactly seven days long starting at midnight of Friday of one week through midnight of Thursday on the next week. The first process in this analysis involved plotting the original data that was taken at five minute intervals and compared to each other. An example of raw temperature data for a typical week during the summer months when the temperature is expected to be higher is shown in Figure 6.31.

![Temperature Plot](image)

**Figure 6.31: Typical Summer Week Raw Temperature Data (June 22, 2012 – June 29, 2012)**

Close observation of the temperature plot above reveals that each of the three data loggers used at the facility provided nearly equivalent values for temperature data. This revelation is important for several reasons. For example, since the temperature values are very close throughout the data collection period, it was determined that the data loggers were producing
reasonable results. Also, since the temperature values are extremely close there is no benefit from moving the air intakes from the roof of the compressor house closer to the ground. There are several reasons why this result may be the case. One reason that the temperatures on the roof are consistent with those on the ground may be the white membrane roofing utilized above the compressor house. White membrane roofing tends to be much cooler than black roofing, which would absorb more radiation from the sun and increase the temperature of the roof. Also, the curvature shape of the intakes creates natural shading from direct sunlight that may make a small difference in temperature.

The other data collected in conjunction with the temperature was the relative humidity. An example of raw data collected for the relative humidity at the two compressor intakes for that same typical summer week is shown in Figure 6.32.

![Figure 6.32: Typical Summer Week Raw Relative Humidity Data (June 22, 2012 – June 29, 2012)](image-url)
Observation of the relative humidity plot above reveals that the values obtained by the data loggers were not nearly as consistent as the temperature data. One reason that this may be case involves the relative humidity sensor on the data loggers not being as reliable as the temperature meters. Also, the location of the air intakes may have been a factor in the relative humidity meters being much different. A wet data logger may have been another source of error. However, it is difficult to determine the exact cause. In order to determine which of the data loggers produced more accurate results, these values would be later compared to relative humidity data from the local airport.

In order to have the most complete data possible, these values were taken over a period of several months to show a wide variety of temperatures. An example of raw data acquired during a typical week in the winter months is shown in Figure 6.33.

![Figure 6.33: Typical Winter Week Raw Temperature Data (December 28, 2012 – January 3, 2013)](image-url)
An example of raw data collected for the relative humidity at the two compressor intakes for that same typical winter week is shown in Figure 6.34.

**Figure 6.34: Typical Winter Week Raw Relative Humidity Data (December 28, 2012 – January 3, 2013)**

The next step involved in the temperature and relative humidity analysis involved comparing instantaneous hourly values acquired from the data loggers with corresponding points from the national weather service. An example of an instantaneous hourly temperature data set from the typical summer week used before is shown in Figure 6.35.
The temperature plot shown above further confirms that the temperature data acquired from the data loggers is reasonable. An example of an instantaneous hourly relative humidity data set from the typical summer week used before is shown in Figure 6.36.
Observation of the relative humidity plot above reveals the compressor 2 intake more closely follows the relative humidity readings from the example week than compressor intake 1 does.

The same data was also compared for winter months. For the typical winter week previously discussed, the hourly instantaneous temperature data is shown in the figure below.
The hourly instantaneous relative humidity data for the typical winter week is shown in the following figure.
Once the analysis of the temperature and relative humidity values collected was complete, daily average temperatures and relative humidity were taken for the overall analysis. The overall analysis is covered in more detail in Section 7.2. Temperature and relative humidity data collected for all other weeks in the same formats presented above are available in Appendix D. Due to limitations in time available for data collection, data from the National Weather Service was collected for temperature and relative humidity to be considered in this analysis. Another important item to note is the substitution of Compressor Intake 1 temperature and relative humidity data for Compressor Intake 2 during the time period when the logger at Compressor Intake 2 fell from the intake (May 22, 2012 – June 7, 2012) for completeness in energy calculations.

### 6.4.2. Compressor Power Consumption and System Flow

The next two essential components to observe the effects of inlet air temperatures on reciprocating and centrifugal compressors were compressor energy usage and the amount of compressed air flow generated by the compressor over a set time interval. Due to high voltages at the plant for power consumption data collection, compressor power data monitored by the subject facility at fifteen minute intervals was provided. This data was then broken up into one week sets that matched the temperature data analysis for ease of comparison. An example of the energy consumption data provided for an operating
compressor (Powerhouse Compressor #1) for the same typical summer week of plant operation from the previous section for temperature data is shown in the figure below.

Figure 6.39: Typical Summer Week Powerhouse Compressor #1 Energy Readings (June 23, 2012 – June 28, 2012)

Close observation of the typical summer week compressor energy data reveals that some points were not available or were removed as outliers that were not consistent with increasing energy usage. This plot also reveals that the energy usage of the compressor shown is linear over the course of each day. Due to this revelation, it is assumed that the compressors operate at constant load throughout the day. Since energy data was not provided for shorter intervals of time, loading and unloading or modulation of the compressors cannot be observed.
For comparison, the energy readings taken for the same compressor during the typical winter week discussed before are shown in the following figure.

![Figure 6.40: Typical Winter Week Powerhouse Compressor #1 Readings (December 28, 2012 – January 3, 2013)](image)

Observation of the typical winter week reveals the same linear characteristic more or less in the progression of energy readings with a high correlation. An interesting aspect of this linear relationship is the decrease of the slope from summer to winter. The slope between these two typical weeks decreases by 1,914 kWh/day (9,120 kWh/day – 7,206 kWh/day). This change in slope reveals a trend in decreasing energy usage from summer to winter for this compressor.
The compressors that operate at the facility follow this same trend during all weeks of production. The alternate compressors are typically reserved for backup use or maintenance when the main compressors are serviced. An example of compressor energy readings acquired during a maintenance week for a maintenance compressor (Powerhouse Compressor #6) is displayed in the plot below.

![Figure 6.41: Typical Maintenance Week Maintenance Powerhouse Compressor #6 (June 30, 2012 – July 6, 2012)](image)

The maintenance week data shows that the maintenance compressor was used at different points during the week at lower loads, resulting in a weaker correlation.

Once the operation of the compressors were confirmed based on the fifteen minute interval energy data, the energy usage of the compressors for each day was calculated by subtracting
An example of the energy consumption of the compressors broken down individually over the typical summer week is shown in the figure below.

![Typical Summer Week Compressor Energy Usage Breakdown](image)

**Figure 6.42: Typical Summer Week Compressor Energy Usage Breakdown (June 22, 2012 – June 29, 2012)**

The figure above confirms that the five main compressors used at the plant are Powerhouse #1, #2, #4, #7, and #8. The remaining five compressors have minimal contribution to the overall energy usage. Compressor power consumption over the typical winter week chosen is shown in the figure below.
During the typical winter week, Powerhouse compressors #1, #2, and #4 consumed the majority of the energy used for plant compressed air. Powerhouse compressors #7 and #8 were not used much until the last two days of the week. This compressor inactivity may have been the result of the plant just starting back after Christmas, or plant production levels being decreased. It is also important to notice with the typical winter week that the compressors used relatively the same amount of energy, which indicates that the compressors that are not needed for a particular day are not running unloaded.
The next set of data collected from the compressed air system consisted of air flow generated. Due to limitations of installing a flow meter inside of the compressed air system, the only values available for comparison to the temperature, relative humidity, and energy data collected were total daily air flow values. These values were provided in units of thousand standard cubic feet (mscf). This data was available for the seven areas that comprised the compressed air system. These areas are denoted as A1, A2, B1, Mod 1, Mod 2, Mod 3, and B2F. These areas combined to show the total amount of compressed air generated at the facility for any given day. For example, the compressed air generated each day during the typical summer week from before in the figure below.

![Figure 6.44: Typical Summer Week Daily Compressed Air Flow (June 22, 2012 – June 29, 2012)](image)
Observation of the typical summer week data reveals that the amount of compressed air flow generated daily is between 20,000 mscf and 25,000 mscf, and remains fairly consistent in the amount of air generated for each individual area. Area B1 tends to require the most compressed air each day, while Mod 1 requires the least. Since data was not collected from the facility on June 26, the data may not have been present due to data logging error. A similar analysis for the typical winter week mentioned previously is shown in the figure below.

![Figure 6.45: Typical Winter Week Daily Compressed Air Flow (December 28, 2012 – January 3, 2013)](image)

Observation of the typical winter week reveals that the amount of compressed air demand is considerably less than the typical summer week. However, it is important to realize that the same week for energy showed that two of the five primary compressors were not in operation
during these days. During most other weeks and indicated during the final day of the typical week, system daily compressed air flow ranges from 20,000 mscf to 25,000 mscf when the plant is in operation.

The complete set of energy and flow data collected from the facility for the same time period that temperature measurements were collected can be found in Appendix E. The charts provided in this Appendix display totalized compressed air energy usage and flow data since the breakdown of compressor energy usage and flow for each area remained consistent with the data shown above for typical weeks. For weeks that data was unable to be collected, energy consumption for each day was determined by choosing a similar day with the closest temperature due to the direct effect of inlet air temperature on performance. Tables detailing which days were equated for this analysis in energy consumption are provided in Appendix E. Since compressed air flow is expected to remain relatively constant, the daily flow considered for these weeks was based on an average of typical production days.

6.5. Data Analysis

Once all of the data was collected from the data loggers installed at the facility and other necessary data provided by plant personnel, the data was converted to forms that could be easily compared. Since flow data presented the most significant obstacle to time constraints on data, the daily values were chosen as the basis for comparison. In order to achieve daily
values for comparison, it was determined that daily average temperatures and daily energy usage would be needed to compare to the total daily flows.

Since the energy data provided by the facility was already instantaneous values, the daily energy for each compressor was found by subtracting the beginning previous day energy reading from the beginning of the current day energy reading and performing a sum for each day over all ten compressors. These values were also converted from watt hours to kilowatt hours by simply dividing the provided value by 1,000. Converting to kilowatt hours enabled the energy values to have a standard basis of comparison to the units typically used to measure electrical energy in industry. These values were then plotted and compared to daily flow.

In order to obtain a daily average temperature, a method was needed to convert five minute instantaneous temperature readings into a twenty four hour average. The method chosen to perform these calculations for this experiment involved the use of an Excel based program developed by the North Carolina State University Industrial Assessment Center referred to as Data Cruncher. A screenshot of the program is shown in the figure below.
The Data Cruncher program shown in the figure above works by using user entered data to generate averages over certain time intervals from data collected at shorter time intervals. This average is generated from the use of a crunch factor, which represents the number of $x$ periods of time of data within the longer desired average time interval. For example, if data is collected at five minute intervals from a data logger and the user desires to shorten that data to fifteen minute averages, the a crunch factor of three is chosen since there are three 5 minute periods in 15 minutes. Therefore, the crunch factor used for daily averages from 5 minute temperature and relative humidity data was 288. These averages were taken for the entire set of temperature data and consolidated with the daily energy and flow data for comparison.
Once daily values were calculated for each of the three data sets, they were combined in an Excel spreadsheet for daily energy analysis. The first step in this energy analysis involved calculating the amount of daily average power consumption of the compressors in kW. These values were obtained by taking the total daily kWh calculated from plant provided data and dividing by 24 hours per day. This calculation was performed based on the previous analysis of the 15 minute compressor data, which showed that the compressors consuming energy at the facility operate at constant capacity throughout the day.

Once these demand values were calculated, the efficiency of the compressors could be computed as a function of the amount of power draw and compressed air flow produced for that day. If the expectation that decreased inlet air temperatures result in higher efficiency air compression and thus significant energy savings, then the amount of air produced per kW of energy consumed should be higher at lower temperatures. From the data collected in this experiment, the efficiency of the compressed air system through the data collection period at a range of average daily inlet air temperature of approximately 35°F to 95°F was found. These values were then plotted for comparison as shown in the following figure for the intake designated as Compressor Intake 1.
Figure 6.47: Compressor Intake 1 Efficiency during Data Collection Period (May 18, 2012 – January 10, 2013)
Observation of the plot above reveals that as temperature decreases, the amount of compressed air that is generated per kW of energy consumed does in fact increase. For example, at a daily average temperature of 90°F, the efficiency of the compressed air system is 11 mscf per kW consumed. At a temperature of 45°F, the efficiency of the compressed air system is approximately 12 mscf per kW consumed. The improvement in efficiency over this temperature range can be calculated as

\[
\text{Efficiency Increase} = \frac{\left| \frac{11 \text{ mscf}}{\text{kW}} - \frac{12 \text{ mscf}}{\text{kW}} \right|}{11 \text{ mscf}} = 0.09
\]

which means that the efficiency of the compressor increases by 9% over a 45°F temperature range, or 1.1% improvement for every 5.5°F that the temperature is decreased. This efficiency increase is consistent with a theoretical 1% improvement for every 5.5°F at a temperature of 90°F (550°F), or 0.18% per degree Fahrenheit.

Another important consideration in this analysis is the outlying compressor efficiency values where efficiency was in the range of 10 mscf/kW. These values occurred most often during times when the facility was down for maintenance or closed for holidays. If the plant was at the same operation level throughout the year, it is expected that compressor efficiency for these days would be higher and more consistent with typical production efficiency.

The same analysis was performed at Compressor Intake 2 and is shown in the figure below.
Figure 6.48: Compressor Intake 2 Efficiency during Data Collection Period
Observation of the efficiency plot for Compressor Intake 2 reveals many of the same tendencies as the plot for Compressor Intake 1. Along with displaying similar results, this additional data set confirms that the amount of compressed air produced per kW of energy consumed increases as the temperature of the inlet air decreases. This data also confirms the efficiency increase of 1.1% improvement for each 5.5°F using the same calculation performed for Compressor Intake 1.
Chapter 7  Methods of Reducing Inlet Air Temperature

Another objective of this experiment involves the investigation of methods that can be used to decrease the inlet air temperatures to reciprocating and centrifugal industrial air compressors. From the data presented in the previous chapter, it was determined that decreasing the inlet air temperatures to these type compressors does result in increased efficiency. This increase in efficiency results in energy and demand savings in relation to the electricity used to operate the compressors. The amount of savings possible from decreasing inlet air temperature depends upon the established baseline data chosen for energy consumption analysis. For example, many industrial plants take in air from outside the building for the compressors to avoid waste heat from the compressor increasing inlet air temperatures. Energy savings in this case are a direct result from the efficiency increase associated with the temperature difference from the inside to the outside.

However, what if a facility has already began taking in outside air to their compressors? Though the inlet air temperatures are now less than they were inside of the facility, the inlet air temperature is now greatly affected by climate and weather. One question to consider in this scenario is what happens to the efficiency of the compressor during summer months, when the inlet air temperature is often significantly higher than during winter months? The results of the data analysis presented in the previous chapter for this experiment show that even when the compressor intakes are moved outside, additional savings can be realized from lower air temperatures during the year.
Since decreasing inlet air temperatures has been found to be beneficial for plant operation from the data acquired, an external source of heat removal or temperature reduction is required to achieve these energy savings. Several methods exist that can be used for this application, a few of which will be covered in more detail in the following energy and cost savings analyses.

7.1. Underground Air Piping

One method that can be used to decrease the inlet air temperature for industrial air compressors involves the use of an underground air pipe. An underground air pipe is based on the idea that the temperature at a certain distance under the surface of the earth remains at a relatively constant temperature. However, it is important to realize that the temperature attainable from an underground pipe is no lower than the temperature of the ground itself. The most significant advantage that this method offers is the ability to decrease the air temperature without the need of an energy consuming system.

7.1.1. Underground Air Piping Background

One method that can be used to reduce the temperature of inlet air to industrial reciprocating and centrifugal air compressors involves the use of an underground air pipe. By placing a
pipe or multiple pipes underground, it is believed that the temperature of the air can be reduced to a lower, more consistent air temperature. This idea comes from the use of geothermal heat pumps to increase efficiency by using warmer air underground to draw heat from in the winter rather than much cooler outside air. For this application, the advantages will be observed during the summer months, when the temperature above the surface of the earth is much higher than the temperature below the surface. However, there is a limit to the temperature that can be achieved with this type of installation.

Currently, the most common application of underground air pipes, otherwise known as earth tubes, is the conditioning of a home. In certain regions of the United States where the climate tends to be very cold in the winter, preheating outside air by using an underground air pipe is expected to raise the temperature of the incoming air to a fairly constant 55 to 60 degrees (the actual temperature will vary by latitude location) (26). Once the temperature has been raised after being drawn through an underground air pipe, it is much easier to provide heating or cooling before the outside air enters the home. Another application similar to the earth tube is a geothermal heat pump, which uses the same principle with a set of underground coils such as in the image of a horizontal geothermal heat pump depicted in the figure below.
Geothermal heat pumps such as the one shown in the previous figure use the underground air temperature as the exchange medium rather than outside air (27). The most significant savings with these types of systems occurs in locations with the most extreme climates. However, could this technology have an application in altering inlet air temperatures for industrial compressors? Since the temperatures are expected to decrease to the same level as the underground air temperature, energy and cost savings are expected from this measure. However, these savings may or may not be enough to offset the cost of installing an underground air pipe. This analysis investigates the possibilities of potential savings and payback of an underground air pipe system.

**Figure 7.1: Closed Loop Horizontal Geothermal Heat Pump**  (27)
7.1.2. **Underground Air Pipe Energy Savings**

The amount of energy savings that can be realized from installing an underground air pipe for cooling inlet air temperatures for industrial reciprocating and centrifugal air compressors is based on the ground temperature at the expected depth. The factors that determine how deep the pipe should be placed underground include the ground temperature itself and the cost to excavate the ground for installation of an underground air pipe system.

For example, the temperature of soil at a depth of 4 inches varies considerably in much the same way that outdoor temperature does, and offers little energy saving opportunities. Data to confirm these variances in temperature was made available from the State Climate Office of North Carolina (28). A plot of data provided from the State Climate Office of North Carolina that details soil temperature at a depth of 4 inches for the closest data acquisition station to the subject facility of this analysis is shown in the figure below.
Therefore, the underground air pipe must be installed at a depth of a few feet below the ground in order to take advantage of the constant ground temperature. The average ground temperature to be considered for several feet below the surface of the earth has been recorded for various regions in the United States of America, as shown in the figure below.
It is important to note that the temperatures shown in the figure above are taken at various substations denoted by the small circles. Once enough data points were acquired from these stations, the lines drawn on the mean ground temperature map were superimposed along well-water contours (29). Using this map, it was determined that the design ground temperature to be used for this analysis would be 64°F based on proximity to the location of the subject facility and the contours of neighboring well-water lines.

Once the baseline temperature of 64°F was established from national soil temperature data, it was determined that the amount of energy savings available from the installation of an underground air pipe would only be from decreased electrical energy and demand of the compressor. Since the compressors operate at constant load and throughout the day as
indicated in compressor energy usage data, it was determined the method used to calculate potential energy savings from the installation of an underground air pipe would consist of finding the power reduction factor derived as a function of temperature in Equation 6.11 for Fahrenheit units. This temperature difference was based on the measured or National Weather Service data, and the established baseline temperature. If the daily average temperature for a given day fell below the established baseline temperature of 64°F, then the temperature from that day was used instead. It is expected in these cases that a damper will be used with the existing intake air system that will be used when the outside air temperature is below the ground temperature. Once this power reduction factor was found, it was applied to the total energy and demand used by the compressors for a given day.

For example, the power reduction factor considering data collected at Compressor Intake 1 for the first day of temperature data acquisition (May 18, 2012) was found to be

\[ f_{\text{reduction}} = \frac{68.5°F - 64°F}{68.5°F + 459.67} \]

\[ f_{\text{reduction}} = 0.00845 \]

Applied to the energy consumption of the compressors on this day, the expected energy savings was found to be

\[ E_{\text{S\_UAP\_Comp1}} = 0.00845 \times 45,701 \text{kWh} \]

\[ E_{\text{S\_UAP\_Comp1}} = 386 \text{kWh} \]
However, it is assumed that all of the compressors at this facility are two stage air compressors with intercooling. Considering this type of compression, it is believed that the actual amount of energy savings will only be half of the expected energy savings calculated from the power reduction factor, because it is believed that the temperature of the air after the intercooler will be the same regardless of cooling inlet air to the compressors. Considering this assumption, the actual energy saved by implementing an underground air pipe from the data collected at Compressor Intake 1 for May 18, 2012 yields

\[
ES_{\text{UAP,Comp1}} = \frac{386 \text{kWh}}{2}
\]

\[
ES_{\text{UAP,Comp1}} = 193 \text{kWh}
\]

The same analysis was performed for energy savings for each day of the year, and then totaled into monthly and annual savings. Monthly and annual energy savings are provided in the underground air pipe energy savings summaries for both compressor intakes at the end of this section.

Once the energy savings were calculated for this measure, the demand reduction was found. This calculation was performed by using the same method employed for energy savings calculations by using the daily average power consumption of the compressors. For example, the demand reduction for May 18, 2012 was found to be

\[
DR_{\text{UAP,Comp1}} = 0.00845 \times 1,904 \text{ kW}
\]

\[
DR_{\text{UAP,Comp1}} = 16.1 \text{ kW}
\]
Considering two stage compressors with intercooling yields

\[
\text{DR}_{\text{UAP,Comp}1} = \frac{16.1 \text{ kW}}{2}
\]

\[
\text{DR}_{\text{UAP,Comp}1} = 8.05 \text{ kW}
\]

The same analysis was performed for each day of the year for demand reduction. Since electrical utilities charge demand by maximum demand consumption during the month, the demand savings for each month were found by taking the day with the minimum demand reduction and designating that value as the demand savings for the month. The minimum demand reduction for each month is chosen as the demand savings because it will result the new established demand for the compressors for that month. This demand is also expected to be coincident when the facility has the highest energy needs during a given month, which is when the electrical demand would be established by the utility. This assumption is made due to the fact that the compressors operate at the same load throughout each day of the month based on the energy consumption data discussed previously.

The results of this energy savings analysis for Compressor Intake 1 are shown in the table below.
### Table 7.1: Compressor Intake 1 Underground Air Pipe Electrical Energy Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>February</td>
<td>178</td>
<td>0.00</td>
</tr>
<tr>
<td>March</td>
<td>3,501</td>
<td>0.00</td>
</tr>
<tr>
<td>April</td>
<td>3,094</td>
<td>0.00</td>
</tr>
<tr>
<td>May</td>
<td>11,265</td>
<td>1.38</td>
</tr>
<tr>
<td>June</td>
<td>16,454</td>
<td>5.74</td>
</tr>
<tr>
<td>July</td>
<td>23,166</td>
<td>17.14</td>
</tr>
<tr>
<td>August</td>
<td>17,476</td>
<td>11.01</td>
</tr>
<tr>
<td>September</td>
<td>13,738</td>
<td>0.00</td>
</tr>
<tr>
<td>October</td>
<td>2,818</td>
<td>0.00</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>December</td>
<td>159</td>
<td>0.00</td>
</tr>
<tr>
<td>Total</td>
<td>91,848</td>
<td>35.26</td>
</tr>
</tbody>
</table>

The results of the energy savings analysis for the installation of an underground air pipe from the data collected at Compressor Intake 2 are shown in the table below.

### Table 7.2: Compressor Intake 2 Underground Air Pipe Electrical Energy Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>February</td>
<td>178</td>
<td>0.00</td>
</tr>
<tr>
<td>March</td>
<td>3,501</td>
<td>0.00</td>
</tr>
<tr>
<td>April</td>
<td>3,094</td>
<td>0.00</td>
</tr>
<tr>
<td>May</td>
<td>11,795</td>
<td>1.38</td>
</tr>
<tr>
<td>June</td>
<td>16,578</td>
<td>5.74</td>
</tr>
<tr>
<td>July</td>
<td>22,781</td>
<td>15.89</td>
</tr>
<tr>
<td>August</td>
<td>17,339</td>
<td>13.85</td>
</tr>
<tr>
<td>September</td>
<td>11,528</td>
<td>0.00</td>
</tr>
<tr>
<td>October</td>
<td>3,138</td>
<td>0.00</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>December</td>
<td>138</td>
<td>0.00</td>
</tr>
<tr>
<td>Total</td>
<td>90,070</td>
<td>36.86</td>
</tr>
</tbody>
</table>
Though these are the expected energy savings for this project based on a best case scenario, it is actually the maximum amount of energy that can be removed if the design temperature of 64°F could be achieved. Instead, not all of the heat removal required to achieve this temperature is possible while still maintaining a reasonable implementation cost. For the purposes of this design, it will be considered that 4 inch pipe will be needed in order to obtain adequate heat removal to approach the desired design temperature. For these 4 inch tubes, the design velocity will be set at 500 feet per minute. Based on the experimental data from the facility, it is expected that the maximum flow rate per day that this system will need to be designed for is 25,000,000 scf. Therefore, the total amount of area needed for air flow in the tubes is found to be

\[
\dot{V} = \frac{25,000,000 \text{ ft}^3}{\text{day}}
\]

\[
\dot{V} = \frac{14,400 \text{ min}}{\text{day}}
\]

\[
\dot{V} = 17,361 \frac{\text{ft}^3}{\text{min}}
\]

At a design velocity of 500 feet per minute, the resulting cross sectional area needed for air flow is

\[
A = \frac{17,361 \text{ ft}^3}{500 \text{ ft}}
\]

\[
A = 34.7 \text{ ft}^2
\]
The cross sectional area of each individual tube is

\[ A_{\text{tube}} = \pi \left( \frac{\text{2 in}}{\text{12 in/ft}} \right)^2 \]

\[ A_{\text{tube}} = 0.0873 \text{ ft}^2 \]

Therefore, the number of tubes needed for this installation will be

\[ N_{\text{tubes}} = \frac{34.7 \text{ ft}^2}{0.0873 \text{ ft}^2} \]

\[ N_{\text{tubes}} = 398 \text{ tubes} \]

For simplicity in installation, the number of tubes will be rounded up to 400 total tubes. The next parameter to consider for this project involves the length of the tubes in order to find total heat transfer area. For cost considerations, only 50 feet of pipe length will be considered for this application.

Since the total number, diameter, and length of PVC pipe is determined from these design parameters, the amount of heat transfer possible between the ground and the air can be calculated. The first parameter needed for this calculation is the Reynolds number for the air traveling through the pipes. The Reynolds number, \( Re \), of a fluid is found by using the equation

\[ Re_{\text{air}} = \frac{\rho UD}{\mu} \quad (7.1) \]
where $\rho$ is the density of the fluid, $U$ is the velocity of the fluid, $D$ is the diameter of the pipe, and $\mu$ is the dynamic viscosity of the working fluid. For this application, the density of air is considered to be 0.075 pounds per cubic foot, the air velocity is 8.333 feet per second, the diameter of the pipe is 4 inches, and the dynamic viscosity is $3.778 \times 10^{-7}$ pound seconds per square foot. Substituting these values into Equation 7.1 and considering gravity for unit conversion to find the Reynolds number for air in the design case yields

$$
\text{Re}_{\text{air}} = \left( \frac{0.075 \text{ lb}}{\text{ft}^3} \cdot \frac{8.333 \text{ ft}}{\text{sec}} \cdot 0.333 \text{ ft} \right) \div \left( \frac{3.778 \times 10^{-7} \text{ lb} \cdot \text{sec}}{\text{ft}^2} \cdot \frac{32.2 \text{ ft}}{\text{sec}^2} \right)
$$

which indicates turbulent flow in the pipes.

Once the Reynolds number for air was found, the Prandtl number is needed to find the heat transfer coefficient. The Prandtl number for air at various temperatures is provided in *Fundamentals of Heat and Mass Transfer* by Incropera and DeWitt. However, an important consideration when using data from these tables is that this heat transfer textbook only provides values in SI units. For the design temperature of 64°F, the design temperature in absolute SI units is 290.9 K. Since the data provided in the Incropera and DeWitt textbook is only given in increments of 50 K and 100 K, a linear interpolation was necessary to find properties at the design temperature. From the textbook, the Prandtl number of air at 300 K is
0.707 and the Prandtl number for air at 250 K is 0.720 (30). Using a linear interpolation to find the Prandtl number at 64°F (290.2 K) yields

\[
\Pr_{64^\circ F} = 0.720 + \left(0.707 - 0.720\right)\frac{(290.2 - 250)}{(300 - 250)}
\]

\[
\Pr_{64^\circ F} = 0.709
\]

Once the Reynolds number and Prandtl number are found for this design, the Nusselt number can be found by using the Colburn equation for turbulent flow in circular tubes, which is given as the expression

\[
\text{Nu}_D = 0.023 \text{Re}^{\frac{4}{5}} \Pr^{\frac{1}{5}}
\]

Substituting the previously calculated values for Reynolds number and Prandtl number into this expression yields

\[
\text{Nu}_D = 0.023(17,125)^{\frac{4}{5}} (0.709)^{\frac{1}{5}}
\]

\[
\text{Nu}_D = 50.0
\]

Once the Nusselt number was found for this design, the convective heat transfer coefficient, \(h\), can be found by rearranging the Nusselt number expression

\[
\text{Nu}_D = \frac{hD}{k}
\]

where \(D\) is the diameter of the pipe and \(k\) is the thermal conductivity of air. Thermal conductivity values for air at various temperatures and also be found in the Incropera and DeWitt textbook. For this design, the thermal conductivity of air at a temperature of 300 K is 0.02673 Watts per meter per Kelvin, and the thermal conductivity of air at a temperature of
250 K is 0.0223 Watts per meter per Kelvin (30). Performing a linear interpolation of these values to find the thermal conductivity at 64°F (290.2 K) in SI units is

\[
k = 0.0223 \frac{W}{mK} + \left( 0.0263 \frac{W}{mK} - 0.0223 \frac{W}{mK} \right) \frac{(290.9 K - 250 K)}{(300 K - 250 K)}
\]

\[
k = 0.0256 \frac{W}{mK}
\]

For this experiment, the units of the thermal conductivity must be English for compatibility with other data. As a result, a conversion factor of 0.57779 BTU per hour per foot per degree Fahrenheit for each Watt per meter per Kelvin was used to find the thermal conductivity of air at 64°F (30). Using this relation resulted in a thermal conductivity of

\[
k = 0.0256 \frac{W}{mK} \times \frac{0.57779 \text{ BTU}}{\text{hr} \cdot \text{ft} \cdot ^\circ \text{F}} \frac{W}{mK}
\]

\[
k = 0.0148 \frac{\text{BTU}}{\text{hr} \cdot \text{ft} \cdot ^\circ \text{F}}
\]

After the thermal conductivity of air at the design temperature was found, the convective heat transfer coefficient was found by rearranging Equation 7.3. The resulting convective heat transfer coefficient for this design was found to be

\[
h = \frac{(50.0) \left( 0.0148 \frac{\text{BTU}}{\text{hr} \cdot \text{ft} \cdot ^\circ \text{F}} \right)}{0.333 \text{ ft}}
\]

\[
h = 2.22 \frac{\text{BTU}}{\text{hr} \cdot \text{ft}^2 \cdot ^\circ \text{F}}
\]
Once the convective heat transfer coefficient was found, the overall heat transfer coefficient of the design was found. Typically for an unfinned, tubular heat exchanger such as the one in this design, the overall heat transfer coefficient is found by using the equation

\[
\frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{R^n_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R^n_{f,o}}{A_o} + \frac{1}{h_o A_o}
\]  

(7.4)

However, for the purposes of this design, it is considered that all terms other than the convective heat transfer term for air are much smaller and that air is the most influential in driving the heat transfer of the design. Therefore, all other terms are deemed negligible and the convective term for air is all that remains. As a result, the areas cancel and the overall heat transfer coefficient is the same as the convective heat transfer coefficient calculated previously (2.22 BTU per hour per square foot per degree Fahrenheit).

Once the overall heat transfer coefficient was found, the surface area of the pipes used was found by using the equation

\[
A_s = \pi D_o L
\]

(7.5)

where \(D_o\) is the outer diameter of the pipe, and \(L\) is the length of the pipe. Considering a pipe length of 50 feet and an estimated outer diameter for 4” PVC pipe, Class 150 at 4.6”, the surface area per tube used in this application is

\[
A_s = \pi \left( 4.6 \text{ in} \times \frac{1 \text{ ft}}{12 \text{ in}} \right) (50 \text{ ft})
\]

\[
A_s = 60.2 \text{ ft}^2
\]

Thus, the total surface area for heat transfer considering all 400 tubes is 24,086 ft².
Once the overall heat transfer coefficient and surface area of the tubes was found, the effectiveness-NTU method was used to find the heat transfer capabilities of this design for energy and demand savings. The first parameter to consider in the evaluation of the effectiveness-NTU method for this design was the hot and cold fluid heat capacity rates of air and the ground, $C_h$ and $C_c$, respectively. Since the ground is treated as an infinite heat source for this application the fluid heat capacity rate is considered to be infinite. The hot fluid heat capacity rate for air can be found by using the relation

$$C_h = \dot{m}_h c_{p,h}$$

(7.6)

where $\dot{m}_h$ is the mass flow rate of air and $c_{p,h}$ is the specific heat of air. For this design, the mass flow rate of air considered is 1,302 pounds per minute. The design heat capacity of air is 0.24 BTU per pound per degree Fahrenheit (8). Substituting these values into Equation 7.6 for the air heat capacity rate yields

$$C_h = \left(1302 \frac{lb}{min}\right) \left(0.24 \frac{BTU}{lb \cdot ^\circ F}\right)$$

$$C_h = 312.5 \frac{BTU}{min \cdot ^\circ F}$$

Converting to an hourly heat capacity rate yields a hot fluid heat capacity rate of 18,750 BTU per hour per degree Fahrenheit. Since the cold fluid heat capacity rate is infinite, it is considered to be $C_{\text{max}}$, which leaves the air heat capacity rate as $C_{\text{min}}$. The next parameter to find as part of the effectiveness-NTU method is the heat capacity ratio, $C_r$, which was found by using the equation
\[
C_r = \frac{C_{\text{min}}}{C_{\text{max}}} 
\] (7.7)

Substituting the values previously determined for \(C_{\text{min}}\) and \(C_{\text{max}}\) for this design yields

\[
C_r = \frac{18,750 \text{ BTU} \cdot \text{hr} \cdot \text{°F}^{-1}}{\infty \text{ BTU} \cdot \text{hr} \cdot \text{°F}^{-1}} = 0
\]

The next parameter to find for the effectiveness-NTU method involved finding the NTU. The NTU for a given design is found by using the parameters of overall heat transfer coefficient, surface area, \(C_{\text{min}}\) calculated previously, and the relation

\[
\text{NTU} \equiv \frac{UA}{C_{\text{min}}} 
\] (7.8)

Substituting the previously calculated parameters into this expression yields

\[
\text{NTU} = \left( \frac{2.22 \text{ BTU} \cdot \text{hr}^{-1} \cdot \text{ft}^2 \cdot \text{°F}^{-1}}{24,085 \text{ ft}^2} \right) \frac{18,750 \text{ BTU} \cdot \text{hr} \cdot \text{°F}^{-1}}{\infty \text{ BTU} \cdot \text{hr} \cdot \text{°F}^{-1}} = 2.85
\]

Now that the NTU is known, the effectiveness, \(\varepsilon\), can be calculated. For all heat exchanger types with \(C_r = 0\), the effectiveness is found by using the expression

\[
\varepsilon = 1 - e^{-NTU} 
\] (7.9)

Substituting the NTU value determined above into this expression yields

\[
\varepsilon = 1 - e^{-2.85} = 0.942
\]
Once the effectiveness of the ground earth tube heat exchanger was found, the actual power reduction factor could be found by determining the actual inlet temperature to the compressor. This temperature was found by relating the final temperature at the compressor inlet to the parameters previously calculated. First, the effectiveness was related to the amount of heat transfer through the relation

\[ \varepsilon = \frac{q_{\text{actual}}}{q_{\text{max}}} \]  \hspace{1cm} (7.10)

where \( q_{\text{actual}} \) is the actual heat removed by the heat exchanger, and \( q_{\text{max}} \) is the amount of heat removal if the heat exchanger were 100 percent effective. For this application, the actual heat removed by the underground air pipe system is found by using the expression

\[ q_{\text{actual}} = C_h(T_i - T_f) \]  \hspace{1cm} (7.11)

where \( T_i \) is the air temperature at the inlet of the air pipe and \( T_f \) is the final temperature of the air entering the compressor at the end of the air pipe system. The maximum heat that can be removed from the air through the use of underground earth tubes is found by using the expression

\[ q_{\text{max}} = C_{\text{min}}(T_i - T_g) \]  \hspace{1cm} (7.12)

where \( T_g \) is the design ground temperature of 64°F. Substituting Equation 7.11 and Equation 7.12 into Equation 7.10 and considering that \( C_h \) equals \( C_{\text{min}} \) yields

\[ \varepsilon = \frac{C_h(T_i - T_f)}{C_h(T_i - T_g)} \]  \hspace{1cm} (7.13)
Canceling the $C_h$ terms and solving for the final temperature at the compressor yields the new relation

$$T_f = T_i - e(T_i - T_g)$$  \hspace{1cm} (7.14)

This relation was then applied to the average daily temperature data collected to find the actual final temperature of the air entering the compressors after being drawn through an underground air pipe. For example, the final temperature at the compressor for the data collected at Compressor Intake 1 for March 18, 2012 was found to be

$$T_f = 68.5°F - 0.942(68.5°F - 64°F)$$

$$T_f = 64.2°F$$

The same calculation was then performed to find the new final temperature as long as the temperature for the day was above 64°F. For days below 64°F, the temperature is expected to remain the same through the use of a damper control with the existing intake system. Once the new final temperature was determined for each day, a new power reduction factor was found. The power reduction for Compressor Intake 1 on March 18, 2012 was found to be

$$f_{\text{reduction, } T_f} = \frac{68.5°F - 64.2°F}{68.5°F + 459.67}$$

$$f_{\text{reduction, } T_f} = 0.00799$$

After the new power reduction factor was determined, it was applied to the energy and demand savings as shown previously while considering two stage compression with intercooling. The results of this analysis for Compressor Intake 1 data energy savings on March 18, 2012 was found to be
The demand reduction for this sample day was found to be

\[
\text{DR}_{\text{UAP,Comp1}, T_f} = \frac{0.00799 \times 1,904 \text{ kW}}{2} = 7.6 \text{ kW}
\]

Once these calculations were performed for each day of the year, the results were combined by monthly values as done in the theoretical case to represent expected savings that would be observed on an energy bill for the subject facility. The results of this analysis for Compressor Intake 1 Data are shown in the following table.

**Table 7.3: Compressor Intake 1 Underground Air Pipe Actual Energy Savings**

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>February</td>
<td>168</td>
<td>0.00</td>
</tr>
<tr>
<td>March</td>
<td>3,310</td>
<td>0.00</td>
</tr>
<tr>
<td>April</td>
<td>2,926</td>
<td>0.00</td>
</tr>
<tr>
<td>May</td>
<td>10,651</td>
<td>1.30</td>
</tr>
<tr>
<td>June</td>
<td>15,557</td>
<td>5.43</td>
</tr>
<tr>
<td>July</td>
<td>21,903</td>
<td>16.20</td>
</tr>
<tr>
<td>August</td>
<td>16,523</td>
<td>10.41</td>
</tr>
<tr>
<td>September</td>
<td>12,989</td>
<td>0.00</td>
</tr>
<tr>
<td>October</td>
<td>2,664</td>
<td>0.00</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>December</td>
<td>150</td>
<td>0.00</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>86,840</td>
<td>33.34</td>
</tr>
</tbody>
</table>
The results at Compressor Intake 2 are shown in the following table.

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>February</td>
<td>168</td>
<td>0.00</td>
</tr>
<tr>
<td>March</td>
<td>3,310</td>
<td>0.00</td>
</tr>
<tr>
<td>April</td>
<td>2,926</td>
<td>0.00</td>
</tr>
<tr>
<td>May</td>
<td>11,152</td>
<td>1.30</td>
</tr>
<tr>
<td>June</td>
<td>15,674</td>
<td>5.43</td>
</tr>
<tr>
<td>July</td>
<td>21,539</td>
<td>15.02</td>
</tr>
<tr>
<td>August</td>
<td>16,394</td>
<td>13.10</td>
</tr>
<tr>
<td>September</td>
<td>10,899</td>
<td>0.00</td>
</tr>
<tr>
<td>October</td>
<td>2,967</td>
<td>0.00</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>December</td>
<td>130</td>
<td>0.00</td>
</tr>
<tr>
<td>Total</td>
<td>85,158</td>
<td>34.85</td>
</tr>
</tbody>
</table>

In order to remain conservative in savings estimates, the energy savings at Compressor Intake 2 will be considered for the purposes of this analysis. Therefore, the anticipated energy savings from installing underground earth tubes at this facility is 85,158 kWh and the anticipated demand savings is 34.9 kW.

### 7.1.3. Underground Air Pipe Cost Savings

In order to determine the amount of cost savings realized through the installation of an underground air pipe, energy rates for electrical demand and electrical energy usage are required. In order to protect facility confidentiality, it was determined that the basis for
electric energy rates for this facility can be based on the Progress Energy Large General Service Time-of-Use (LGS-TOU) rate schedule. The details of this rate schedule are provided in Appendix F. This rate schedule is commonly applied to facilities possessing significant electrical demand needs and facilities that operate on a 3 shift, 24 hour schedule as the subject facility for this analysis does.

The first rate to determine in order to calculate energy cost savings is the electrical demand charge. Though the demand rates provided in the LGS-TOU rate schedule are tiered based on various levels of electrical demand used in a given month, the actual demand cost savings that can be realized for this facility lies in the highest tier that the plant falls under until the plant reduces enough electrical demand for their charges to fall back into a lower tier. For this facility, it is assumed that the facility falls in the above 5,000 kW tier and that the total amount of demand savings from this system upgrade will not reduce the electrical demand at the facility below 5,000 kW. This tier of demand rates requires that different rates be paid depending on the time of year. On this rate schedule, months designated as summer months (June through September) carry a demand charge of $18.56 per kW used, and other months (October through May) are charged at a rate of $13.25 per kW. Since the demand savings are calculated on a per month basis in the previous section, these rates will be applied to their respective months. In cases where only annual savings may be considered, a combined annual demand rate of $15.02 will be used as calculated below.

\[
LGS-TOU_{\text{demand}} = \frac{18.56 \times 4 \text{ months}}{12 \text{ months}} + \frac{13.25 \times 8 \text{ months}}{12 \text{ months}}
\]
The next rate required for energy cost savings calculations involves the cost for overall energy usage. From the LGS-TOU rate schedule provided, the on-peak energy rate for electricity is $0.0482 per kWh and the rate for off-peak energy usage is $0.0432 per kWh. These rates based on time of usage are used as a way for the utility to encourage facilities that are operating 24 hours per day to focus more of their energy usage at times when the utility does not have the electrical demands of primary work shifts. This incentive enables the utility to require fewer power stations to be built to provide power for industrial facilities. A quick observation of the times designated as on-peak and off-peak reveal that even though the time designations are different, the total amount of on-peak and off-peak time each day is 12 hours each, excluding weekends that are completely off-peak.

Therefore, the combined energy rate charge for on peak and off-peak is solely based on the number of hours each week that the plant is on-peak or off-peak due to 24 hour production. As a result of this revelation, the number of hours per week for on-peak energy is expected to be 60 hours (12 hours per day on-peak, 5 days per week), and 108 hours of off-peak (60 hours for weekdays just as with on-peak, and an additional 48 hours from Saturday and Sunday). These hours can then be used as ratios in order to determine the combined electrical energy rate. Based on these parameters, the combined rate for this facility is found to be

\[
LGS - TOU_{\text{energy}} = \frac{0.0482 \times 60 \text{ hours}}{168 \text{ hours}} + \frac{0.0432 \times 108 \text{ hours}}{168 \text{ hours}}
\]

\[
LGS - TOU_{\text{energy}} = 0.044999
\]
The result of applying these energy rates to the energy and demand savings observed from each month and the total cost savings for implementing an underground air pipe based on data collected at Compressor Intake 1 and Compressor Intake 2 are detailed in the tables below.

**Table 7.5: Compressor Intake 1 Underground Air Pipe Energy Cost Savings**

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Overall Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>February</td>
<td>178</td>
<td>$8</td>
<td>0.0</td>
<td>$0</td>
<td>$8</td>
</tr>
<tr>
<td>March</td>
<td>3,501</td>
<td>$157</td>
<td>0.0</td>
<td>$0</td>
<td>$157</td>
</tr>
<tr>
<td>April</td>
<td>3,094</td>
<td>$139</td>
<td>0.0</td>
<td>$0</td>
<td>$139</td>
</tr>
<tr>
<td>May</td>
<td>11,265</td>
<td>$507</td>
<td>1.4</td>
<td>$18</td>
<td>$525</td>
</tr>
<tr>
<td>June</td>
<td>16,454</td>
<td>$740</td>
<td>5.7</td>
<td>$107</td>
<td>$847</td>
</tr>
<tr>
<td>July</td>
<td>23,166</td>
<td>$1,042</td>
<td>17.1</td>
<td>$318</td>
<td>$1,360</td>
</tr>
<tr>
<td>August</td>
<td>17,476</td>
<td>$786</td>
<td>11.0</td>
<td>$204</td>
<td>$990</td>
</tr>
<tr>
<td>September</td>
<td>13,738</td>
<td>$618</td>
<td>0.0</td>
<td>$0</td>
<td>$618</td>
</tr>
<tr>
<td>October</td>
<td>2,818</td>
<td>$127</td>
<td>0.0</td>
<td>$0</td>
<td>$127</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>December</td>
<td>159</td>
<td>$7</td>
<td>0.0</td>
<td>$0</td>
<td>$7</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>91,848</td>
<td><strong>$4,132</strong></td>
<td><strong>35.3</strong></td>
<td><strong>$647</strong></td>
<td><strong>$4,779</strong></td>
</tr>
</tbody>
</table>
Table 7.6: Compressor Intake 2 Underground Air Pipe Energy Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Overall Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>February</td>
<td>178</td>
<td>$8</td>
<td>0.0</td>
<td>$0</td>
<td>$8</td>
</tr>
<tr>
<td>March</td>
<td>3,501</td>
<td>$157</td>
<td>0.0</td>
<td>$0</td>
<td>$157</td>
</tr>
<tr>
<td>April</td>
<td>3,094</td>
<td>$139</td>
<td>0.0</td>
<td>$0</td>
<td>$139</td>
</tr>
<tr>
<td>May</td>
<td>11,795</td>
<td>$531</td>
<td>1.4</td>
<td>$18</td>
<td>$549</td>
</tr>
<tr>
<td>June</td>
<td>16,578</td>
<td>$746</td>
<td>5.7</td>
<td>$107</td>
<td>$852</td>
</tr>
<tr>
<td>July</td>
<td>22,781</td>
<td>$1,025</td>
<td>15.9</td>
<td>$295</td>
<td>$1,320</td>
</tr>
<tr>
<td>August</td>
<td>17,339</td>
<td>$780</td>
<td>13.9</td>
<td>$257</td>
<td>$1,037</td>
</tr>
<tr>
<td>September</td>
<td>11,528</td>
<td>$519</td>
<td>0.0</td>
<td>$0</td>
<td>$519</td>
</tr>
<tr>
<td>October</td>
<td>3,138</td>
<td>$141</td>
<td>0.0</td>
<td>$0</td>
<td>$141</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>December</td>
<td>138</td>
<td>$6</td>
<td>0.0</td>
<td>$0</td>
<td>$6</td>
</tr>
<tr>
<td>Total</td>
<td>90,070</td>
<td>$4,052</td>
<td>36.9</td>
<td>$677</td>
<td>$4,729</td>
</tr>
</tbody>
</table>

The cost savings in the previous case considered the best case scenario where all heat is removed by the ground from the air to achieve a constant underground air temperature of 64°F when the temperature is above that threshold. Otherwise, air would be taken in at the existing intakes at lower temperatures and there would be no power reduction. The same electricity rates were used to find cost savings for the actual case as the ones used to determine cost savings for the theoretical case. The results of this cost savings analysis for the data collected at Compressor Intakes 1 and 2 are shown in the tables below.
Table 7.7: Compressor Intake 1 Underground Air Pipe Actual Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Overall Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>February</td>
<td>168</td>
<td>$8</td>
<td>0.0</td>
<td>$0</td>
<td>$8</td>
</tr>
<tr>
<td>March</td>
<td>3,310</td>
<td>$149</td>
<td>0.0</td>
<td>$0</td>
<td>$149</td>
</tr>
<tr>
<td>April</td>
<td>2,926</td>
<td>$132</td>
<td>0.0</td>
<td>$0</td>
<td>$132</td>
</tr>
<tr>
<td>May</td>
<td>10,651</td>
<td>$479</td>
<td>1.3</td>
<td>$17</td>
<td>$496</td>
</tr>
<tr>
<td>June</td>
<td>15,557</td>
<td>$700</td>
<td>5.4</td>
<td>$101</td>
<td>$801</td>
</tr>
<tr>
<td>July</td>
<td>21,903</td>
<td>$985</td>
<td>16.2</td>
<td>$301</td>
<td>$1,286</td>
</tr>
<tr>
<td>August</td>
<td>16,523</td>
<td>$743</td>
<td>10.4</td>
<td>$193</td>
<td>$936</td>
</tr>
<tr>
<td>September</td>
<td>12,989</td>
<td>$584</td>
<td>0.0</td>
<td>$0</td>
<td>$584</td>
</tr>
<tr>
<td>October</td>
<td>2,664</td>
<td>$120</td>
<td>0.0</td>
<td>$0</td>
<td>$120</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>December</td>
<td>150</td>
<td>$7</td>
<td>0.0</td>
<td>$0</td>
<td>$7</td>
</tr>
<tr>
<td>Total</td>
<td>86,840</td>
<td>$3,907</td>
<td>33.3</td>
<td>$612</td>
<td>$4,518</td>
</tr>
</tbody>
</table>

Table 7.8: Compressor Intake 2 Underground Air Pipe Actual Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Overall Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>February</td>
<td>168</td>
<td>$8</td>
<td>0.0</td>
<td>$0</td>
<td>$8</td>
</tr>
<tr>
<td>March</td>
<td>3,310</td>
<td>$149</td>
<td>0.0</td>
<td>$0</td>
<td>$149</td>
</tr>
<tr>
<td>April</td>
<td>2,926</td>
<td>$132</td>
<td>0.0</td>
<td>$0</td>
<td>$132</td>
</tr>
<tr>
<td>May</td>
<td>11,152</td>
<td>$502</td>
<td>1.3</td>
<td>$17</td>
<td>$519</td>
</tr>
<tr>
<td>June</td>
<td>15,674</td>
<td>$705</td>
<td>5.4</td>
<td>$101</td>
<td>$806</td>
</tr>
<tr>
<td>July</td>
<td>21,539</td>
<td>$969</td>
<td>15.0</td>
<td>$279</td>
<td>$1,248</td>
</tr>
<tr>
<td>August</td>
<td>16,394</td>
<td>$737</td>
<td>13.1</td>
<td>$243</td>
<td>$981</td>
</tr>
<tr>
<td>September</td>
<td>10,899</td>
<td>$490</td>
<td>0.0</td>
<td>$0</td>
<td>$490</td>
</tr>
<tr>
<td>October</td>
<td>2,967</td>
<td>$133</td>
<td>0.0</td>
<td>$0</td>
<td>$133</td>
</tr>
<tr>
<td>November</td>
<td>0</td>
<td>$0</td>
<td>0.0</td>
<td>$0</td>
<td>$0</td>
</tr>
<tr>
<td>December</td>
<td>130</td>
<td>$6</td>
<td>0.0</td>
<td>$0</td>
<td>$6</td>
</tr>
<tr>
<td>Total</td>
<td>85,158</td>
<td>$3,831</td>
<td>34.9</td>
<td>$640</td>
<td>$4,471</td>
</tr>
</tbody>
</table>
Therefore, based on the data collected at Compressor Intake 2 as chosen previously, the total expected cost savings from installing and underground air pipe system at the subject facility is $4,471 per year. It is important to note that the amount of cost savings that can be realized from implementing this measure depends significantly on weather conditions, which can be variable from year to year. Another important consideration in the cost savings involves considering the differences in electric rates from various providers. If the plant were considered to be receiving electricity from a different utility or electric rate schedule, cost savings will increase or decrease accordingly.

7.1.4. Underground Air Pipe Implementation and Payback

The implementation of the underground air pipe system as mentioned in previous sections will require the installation of four hundred 4 inch tubes. These tubes are chosen to be of PVC pipe, Class 150 and 50 feet each. These pipes will be arranged underground in a similar configuration to the illustration model shown in the figure below.

Figure 7.4: Underground Earth Tube Configuration Model
Based on the design length of the tubes at 50 feet in length, the total linear feet of tubing needed for this application is 20,000 feet. The estimated material cost for PVC pipe, Class 150 is $3.52 per linear foot, including labor (all labor is considered to be half of the national values for this project) (31).

The other costs associated from this installation arise from the need to excavate a trench in the ground where the tubes will be placed and surrounded by soil. It is expected that this trench will need to be excavated at a minimum of 3 feet deep and 50 feet in length from the compressor house to achieve the expected results, and for the pipes to be deep enough to experience the benefits of constant ground temperature. The width of the trench will be based on the desired orientation of the tubes by the subject facility. It is estimated that the cost to excavate this trench will be $10,000. Other additional costs associated with this project will be assumed as 25% of the excavation and tube installation cost. These costs include the installation of a damper so that the existing compressed air intakes can be used when the outside air temperature is below 64°F. An overall breakdown of the expected installation costs for an underground earth tube is shown in the table below.
As shown in the table above, the total implementation cost for this measure is expected to be $100,500. With an overall cost savings of $4,471 per year, the simple payback period for this system upgrade is 22.5 years, or 270 months.

### 7.2. Industrial Heat Pumps

Another option available to reduce the inlet air temperatures for reciprocating and centrifugal air compressors involves the use of an industrial heat pump. Industrial heat pumps operate in the same manner as standard heat pumps. However, the industrial heat pump is used at a manufacturing facility as a means of heat recovery in addition to heat removal. Rather than taking or removing air from the outdoors to a conditioned space for occupant comfort, an industrial heat pump, or water cooled air conditioning system, removes heat to improve efficiency from one application in an industrial process. Then, the heat that is removed is utilized in another component of the manufacturing process. This analysis investigates the
case where heat is removed from compressor intake air to improve compressed air efficiency. The heat removed is then used to preheat makeup water for facility boilers.

7.2.1. Heat Pump Background

A heat pump is a device designed to transfer energy in the form of heat from a low temperature reservoir to a high temperature reservoir. Since the second law of thermodynamics dictates that heat can be transferred from areas of high temperature to low temperature naturally, work input is needed to achieve this transfer of energy against the temperature gradient. A depiction of this process is shown in the figure below.

![Heat Pump Diagram](image)

**Figure 7.5: Heat Pump Diagram (4)**

The most common application for heat pumps is the case shown in the previous figure. In this case, a home or building is desired to be maintained at a higher temperature than the outdoor temperature. The heat pump is primarily needed for this application during winter months when the temperature outdoors is cooler than the comfort level for most people.
The efficiency of a heat pump is determined by calculating a parameter referred to as the coefficient of performance (COP). The COP for a heat pump can be found by using the relation

$$COP_{HP} = \frac{\text{Desired Output}}{\text{Required Input}} = \frac{Q_H}{W_{\text{net,in}}}$$ (7.15)

where $Q_H$ is the amount of heat energy transferred by the heat pump to the conditioned space, and $W_{\text{net,in}}$ is the amount of work required to move the heat. Equation 7.15 can also be expressed in the form

$$COP_{HP} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - \frac{Q_L}{Q_H}}$$ (7.16)

where $Q_L$ is the amount of heat removed from the low temperature reservoir. The coefficient of performance of a heat pump can also be related to the coefficient of performance of a refrigerator through the use of the expression

$$COP_{HP} = COP_R + 1$$ (7.17)

One important realization that can be made from the preceding formulas concerning the coefficient of performance of heat pumps involves the fact that the COP of a heat pump will always be greater than or equal to unity. This benefit occurs because the COP of a refrigerator is always positive. However, this realization is only true in theory. In fact, some of the heat transferred to the conditioned space will be lost back to the ambient air due to piping and other devices, which may cause COP for the heat pump to fall below unity if outdoor air temperature is extremely low (4). These type heat pumps are known as air-source
Heat pumps, and are commonly used for conditioning air in residential, industrial, and commercial buildings.

However, some regions of the United States have temperatures too low for an air source heat pump to be economically viable from an energy perspective. In these regions, it is recommended to use geothermal heat pumps. Geothermal heat pumps draw their heat for energy transfer from underground piping excavated several feet below the ground as mentioned previously. At a depth of several feet, the temperature of the ground is warmer in the winter months than the ambient air, and the temperature remains relatively constant. These higher temperatures result in higher efficiency of the heat pump (up to 45% higher efficiency than air-source heat pumps) due to less work needed from the compressor (4). Since there will be greater energy savings in areas of lower temperatures, the amount of time needed for the savings to offset the additional installation cost of the geothermal heat pump will be more economically attractive.

The performance of a heat pump is often referred to by a term known as the energy efficiency ratio (EER) or seasonal energy efficiency ratio (SEER). These parameters are determined by an established set of testing standards. SEER can be determined by finding the ratio of the total amount of heat removed by the heat pump to the total amount of electrical work consumed (4). This parameter is known to be a measure of seasonal performance of standard cooling equipment. EER is found by taking a measure of the instantaneous energy
efficiency, and is the ratio comprised of the rate of heat removal from a cooled space and the rate of electrical consumption for typical operating conditions (4). In order to promote widespread energy efficiency initiative in society, governments globally have established COP, SEER, and EER standards for performance of heat pumps. For example, most air conditioners and heat pumps available in the market have SEER values from 13 to 21, which are equivalent to COP values of 3.8 to 6.2 (4).

Another important parameter for the understanding of heat pumps involves the units in which refrigeration is measured. The cooling capacity of a refrigeration system, such as a heat pump, is typically given in terms of tons of refrigeration. For example, capacity of a refrigeration system that can freeze 1 ton (2000 lbm) of water in the liquid state at 32°F (0°C) in a time duration of 24 hours is said to be one ton of cooling (4). One ton of refrigeration is equivalent to 12,000 BTU per hour of heat removal. Another method to find tons of cooling involves using the efficiency of a refrigeration system if the energy demand for removal is given in units of kW. In this case, air cooled refrigeration units typically have an efficiency in the range of 1.0 to 1.2 kW/ton and water cooled units have an efficiency of approximately 0.6 kW/ton.

Another important consideration with heat pumps involves the conditioning of air in a space during summer months. Though heat pumps have a high initial cost, their high efficiency enables them to have long term energy cost benefits. However, heat pumps in theory are only
designed to condition air in one direction. As a result, one may ask what happens to condition air during the summer months, when heat removal from the conditioned space is desired. In this case, a heat pump can be modified through the installation of a reversing valve in the heat pump system to enable the heat pump to operate in both directions (4). An illustration of this application is shown in the figure below.

Figure 7.6: Modified Heat Pump for Year-Round Air Conditioning  (4)

This analysis investigates the benefits of installing an industrial heat pump to remove heat from inlet air for the air compressors and recovering that heat to preheat boiler makeup
water. Though this recommendation will result in the most significant energy and cost savings, will the total amount of savings justify the additional implementation cost?

7.2.2. **Industrial Heat Pump Heat Energy Savings Analysis**

The energy savings associated with the installation of an industrial heat pump come as a result of three changes in energy consumption at the facility. These changes are decreased fuel consumption in the boilers from waste heat recovery, increased electrical usage from the installed heat pump, and a decrease in electrical energy usage from increased compressor efficiency. The analysis described in this section refers specifically to energy savings realized from decreased fuel consumption in the boilers. Although, this analysis also directly affects the other two areas of energy change.

The first step in determining the amount of energy savings available at the boilers by using heat removed from compressor intake air to preheat boiler makeup water involves a thermodynamic analysis of the system. Since an industrial heat pump (water cooled air conditioning system) is chosen for this application, a vapor compression cycle is used to model the heat removed by the heat pump as well as the heat received by the boiler makeup water. An illustration of the ideal vapor compression refrigeration cycle is shown in the figure below.
Observation of the ideal vapor-compression refrigeration cycle reveals that work input will be required to a compressor in the heat pump system, heat will be removed from the compressor intake air (cold refrigerated space in the figure) at the evaporator, and the heat will be transferred to the boilers (warm environment in the figure) at the condenser. The $T$-$s$ diagram for this cycle is shown in the figure below.
The $P-h$ diagram for the ideal vapor-compression refrigeration cycle is shown in the figure below.

![P-h Diagram](image)

**Figure 7.9: Ideal Vapor-Compression Refrigeration Cycle $P-h$ Diagram**

Based on these diagrams, the amount of heat removed by the heat pump and rejected by the heat pump to the higher temperature reservoir can be found by knowing the mass flow rate of the fluid flowing and the change in enthalpy. An interesting observation from the $P-h$ diagram involves the fact that the enthalpy change in the heat supplied to the boilers is more than the amount of heat removed from the compressor intake air. This additional heat can be accounted for by performing an energy balance on the vapor-compression cycle and realizing that the work input to the compressor is also transferred to the heat available to preheat boiler makeup water. As a result, the compressor work must also be considered in the heat energy savings.

The first step in the calculation of heat energy savings at the boilers involved calculating the enthalpy at each end of the evaporator where the heat would be removed from the
compressor inlet air based on the temperature and relative humidity values acquired from the data loggers. In order to establish a baseline for the energy calculations, it was determined that air above 45°F would be cooled to that temperature, and air cooled to that temperature would have a relative humidity of 100% at the end of the evaporator. A diagram of the process over the evaporator is shown in the figure below.

![Diagram of heat removal from moist compressor intake air](image)

**Figure 7.10: Heat Removal from Moist Compressor Intake Air Schematic**

The analysis above begins by determining the enthalpy at state 1, which is the enthalpy of the atmospheric air originally entering the compressor intake. The first step involved in this process utilizes the relative humidity relation given by Equation 2.35. In order to use this relation, the saturation pressure of the water vapor must be known at the compressor intake inlet temperature. This temperature corresponds to the temperature data values recorded by the data loggers. Finding these values without interpolating hundreds of values from steam
tables or reading hundreds of values from a standard psychrometric chart required the use of computer software. The program used to obtain this data is an Excel based program developed by Magnus Holmgren called X Steam (available in English and SI units) that uses macros to calculate various properties in the thermodynamic steam tables. Each of these properties has a call function that can calculate the property given a certain property.

For example, the saturation pressure found for the first day of data collection (May 18, 2012) was found by using the function

\[ P_{\text{sat},T} = (7.18) \]

where the value in parentheses corresponds to the average inlet air temperature recorded for that day. As a result of this function, the saturation pressure for the water vapor on this day was found to be 0.345 psia. Using this value and the relative humidity ratio mentioned before, the vapor pressure for this day was found to be

\[ P_{v_i} = P_{\text{sat}} (68.5^\circ F) \times 57.1\% \]

\[ P_{v_i} = 0.197 \text{ psia} \]

Next, since the vapor pressure was found from the inlet temperature and relative humidity, and atmospheric pressure is known, the humidity ratio for this day can be found by using Equation 2.34. Completing this analysis for May 18, 2012 yields

\[ \omega_i = 0.622 \frac{0.197 \text{ psia}}{14.696 \text{ psia} - 0.197 \text{ psia}} \]
\[ \omega_i = 0.00845 \frac{\text{lb}_v}{\text{lb}_{da}} \]

Since the humidity ratio and temperature are now known, the ASHRAE formula provided in Equation 2.44 can be used to find the enthalpy for the day desired. The enthalpy for May 18, 2012 for Compressor Intake 1 is found to be

\[ h_i = 0.24 \times 68.5^\circ F + 0.00845 \frac{\text{lb}_v}{\text{lb}_{da}} (1061 + 0.444 \times 68.5^\circ F) \]

\[ h_i = 25.7 \frac{\text{BTU}}{\text{lb}} \]

The same method of calculation was used for state 2 in this process. The only exceptions were that any temperatures below the established set point of 45°F were left as the initial temperature along with their corresponding original relative humidity. It is expected that at these times the heat pump will not be in operation, thus resulting in no energy savings.

The next step involved in this calculation involved the determination of a flow rate of air to determine the amount of air drawn into the compressor at a given point in time. Volumetric flow in terms of thousand standard cubic feet per day was provided by the facility. However, in order for these values to be useful in heat energy removal calculations, the volumetric flow needed to be converted to a mass flow rate. For these calculations, a standard air density of 0.075 pounds per cubic foot was used. As an example, the mass flow rate per day on May 22, 2012 was determined to be

\[ \dot{m} = 22,550,000 \frac{\text{scf}}{\text{day}} \times 0.075 \frac{\text{lb}}{\text{scf}} \]
\[ \dot{m} = 1691.250 \frac{\text{lb}}{\text{day}} \]

These values were then used along with the enthalpy difference in order to find the amount of heat removed from the air each day. The amount of heat that could be removed by decreasing inlet air temperature to the set point established previously on May 12, 2012 was found to be

\[ Q_{\text{removed}} = 1691.250 \frac{\text{lb}}{\text{day}} \times \left( 25.7 \frac{\text{BTU}}{\text{lb}} - 17.6 \frac{\text{BTU}}{\text{lb}} \right) \]

\[ Q_{\text{removed}} = 13,582,192 \frac{\text{BTU}}{\text{day}} \]

Once the heat removed from the inlet air is found, it is important to remember that the amount of heat available to preheat boiler makeup water is also a function of the work input to the compressor. It is assumed that a heat pump with a COP of 5 will be adequate for this application. In this case, the amount of energy needed from the compressor of the heat pump to remove this heat is one fifth of the heat removed calculated above, or 20%. In this case, the total heat available to the boilers for the preheat of makeup water can be found to be

\[ Q_{\text{available}} = 13.6 \frac{\text{MMBTU}}{\text{day}} \times 1.2 \]

\[ Q_{\text{available}} = 16.3 \frac{\text{MMBTU}}{\text{day}} \]

This heat is heat energy that otherwise would have been generated by the boilers. Therefore, the efficiency of the boilers must be considered to calculate the actual amount of fuel energy saved. For the purposes of this analysis, a typical industrial boiler efficiency of 80% is considered. Therefore, the fuel energy savings observed for May 18, 2012 are found to be
Another component needed for the overall fuel savings involves finding the fuel demand for a given month for cost savings. This analysis included taking the calculated fuel energy savings values and finding the minimum daily average fuel savings for each month. Minimum demand reduction is chosen again for demand savings because these are the points where the new demand will be established. This minimum value is also expected to be savings from when the most fuel would be needed for the boilers, which occurs when the fuel demand would be established for the given month.

The amount of fuel energy and demand savings observed from data collected at Compressor Intake 1 and Compressor Intake 2 are shown in the tables below.

\[
Q_{fuel} = \frac{16.3 \text{ MMBTU}}{\text{day}}
\]

\[
Q_{fuel} = 20.4 \frac{\text{MMBTU}}{\text{day}}
\]
Table 7.10: Compressor Intake 1 Heat Pump Fuel Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Natural Gas Energy Savings (MMBTU)</th>
<th>Natural Gas Demand Savings (MMBTU)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>111</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>123</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>596</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>533</td>
<td>3.1</td>
</tr>
<tr>
<td>May</td>
<td>1,028</td>
<td>12.8</td>
</tr>
<tr>
<td>June</td>
<td>1,100</td>
<td>19.3</td>
</tr>
<tr>
<td>July</td>
<td>1,252</td>
<td>20.9</td>
</tr>
<tr>
<td>August</td>
<td>1,228</td>
<td>19.9</td>
</tr>
<tr>
<td>September</td>
<td>916</td>
<td>14.4</td>
</tr>
<tr>
<td>October</td>
<td>554</td>
<td>2.2</td>
</tr>
<tr>
<td>November</td>
<td>62</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>153</td>
<td>0.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>7,657</strong></td>
<td><strong>92.5</strong></td>
</tr>
</tbody>
</table>

Table 7.11: Compressor Intake 2 Heat Pump Fuel Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Natural Gas Energy Savings (MMBTU)</th>
<th>Natural Gas Demand Savings (MMBTU)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>116</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>123</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>596</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>533</td>
<td>2.5</td>
</tr>
<tr>
<td>May</td>
<td>1,033</td>
<td>10.3</td>
</tr>
<tr>
<td>June</td>
<td>1,169</td>
<td>20.0</td>
</tr>
<tr>
<td>July</td>
<td>1,525</td>
<td>17.7</td>
</tr>
<tr>
<td>August</td>
<td>1,317</td>
<td>19.2</td>
</tr>
<tr>
<td>September</td>
<td>1,078</td>
<td>13.6</td>
</tr>
<tr>
<td>October</td>
<td>625</td>
<td>1.9</td>
</tr>
<tr>
<td>November</td>
<td>85</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>203</td>
<td>0.0</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>8,403</strong></td>
<td><strong>85.2</strong></td>
</tr>
</tbody>
</table>
In order to find annual estimated fuel energy and demand savings, days outside of the collection period were determined from daily averages obtained from National Weather Service Data and values for enthalpy found from this data. Flow rates for these days were found by taking an average flow value over times with similar weather patterns. Calculated values and parameters considered for all days in this analysis are shown in Appendix D.

7.2.3. Heat Pump Energy Consumption

The next step involved in finding the overall energy savings of installing an industrial heat pump at this facility involves the amount of additional energy that will be consumed by the heat pump. Since the heat pump will be installed at an industrial facility, the additional energy usage will be a combination of the energy in kWh used by the heat pump over the course of a year as well as the peak energy demand used by the heat pump in kW.

Finding the electrical energy necessary to remove heat from the intake air requires a simple unit conversion from heat removal in BTU to kWh. For this analysis, monthly energy values were calculated for electrical energy from total MMBTU of heat removed from the air. For example, total heat removed from the inlet air calculated for the month of June was calculated as 696 MMBTU. Converting this heat energy into electrical energy yields

$$ W_{hp} = 696 \text{ MMBTU} \times 10^6 \frac{\text{BTU}}{\text{MMBTU}} \times \frac{1 \text{ kWh}}{3,413 \text{ BTU}} $$

$$ W_{hp} = 203,895 \text{ kWh} $$
The same analysis was performed for each month of data collection as well as months determined from National Weather Service Data. These values are shown in an overall increased energy summary at the end of this section.

The next step to finding the increase in energy associated with this system involves the increase in electrical demand. For this analysis, it was determined that the best way to find the expected increase in demand involved taking the maximum enthalpy for cooling and comparing it to a baseline value. The baseline value chosen for this case is the design conditions of 45°F and 100% relative humidity, which corresponds to an enthalpy of 17.6 BTU per pound. Next, it was determined that the method to be used for establishing a maximum enthalpy would be to use TMYcalc data. TMYcalc is a computer program developed by the North Carolina State University Industrial Assessment Center that calculates degree heating hours, degree cooling hours, and degree enthalpy hours for various locations in North Carolina where meteorological data is available. This program also provides minimum and maximum temperature or enthalpy, time above the set point value, time below the set point value, and time at the set point value. A screenshot of TMYcalc as an example for enthalpy data in Raleigh, NC for the month of June is shown in the figure below.
The city of Raleigh, NC was chosen for this calculation because it is the only location where enthalpy data is available and is reasonably close to the weather at the subject facility. This screenshot also shows that the maximum enthalpy seen at this location for the month of June is 39 BTU per pound. The maximum and base enthalpies for each month during the year found by using this program are shown in the table below.
Once these enthalpy values were found, it was determined that the most conservative method of calculating demand increase would occur when the flow of the system was at a maximum, which is expected to occur after this project is implemented. Thus, the maximum daily flow rate was determined and used with the difference in enthalpies from above to find the maximum rate of heat removal from the air. For the experimental month of June, the expected maximum rate of heat removal was found to be

\[
\dot{Q}_{\text{removed}} = 1,861,500 \text{ lb/day} \times \frac{1 \text{ day}}{24 \text{ hours}} \times 21.4 \frac{\text{ BTU}}{\text{ lb}}
\]

\[
\dot{Q}_{\text{removed}} = 1,658,271 \frac{\text{ BTU}}{\text{ hr}}
\]

Using a conversion factor of 12,000 BTU per hour per ton of cooling yields a demand of
\[ \dot{Q}_{\text{removed}} = 1,658,271 \text{ BTU/hr} \times \frac{1 \text{ Ton}}{12,000 \text{ BTU/hr}} \]

\[ \dot{Q}_{\text{removed}} = 138 \text{ Tons} \]

A heat pump with a COP of 5 has an efficiency of approximately 0.7 kW per Ton. Therefore, the billing demand increase for the month of June was found to be

\[ \dot{Q}_{\text{removed}} = 138 \text{ Tons} \times 0.7 \frac{\text{kW}}{\text{Ton}} \]

\[ \dot{Q}_{\text{removed}} = 96.7 \text{ kW} \]

The same analysis was performed to find the increase in demand for each month during the year. The following table summarizes the results of expected energy and demand increase from installing an industrial heat pump at this location based on Compressor Intake 1 data.

**Table 7.13: Compressor Intake 1 Heat Pump Monthly Increased Energy and Demand**

<table>
<thead>
<tr>
<th>Month</th>
<th>Heat Pump Energy Consumption (kWh)</th>
<th>Heat Pump Demand Consumption (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>4,341</td>
<td>43.5</td>
</tr>
<tr>
<td>February</td>
<td>4,818</td>
<td>47.7</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>56.1</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>60.3</td>
</tr>
<tr>
<td>May</td>
<td>40,161</td>
<td>89.5</td>
</tr>
<tr>
<td>June</td>
<td>42,977</td>
<td>96.7</td>
</tr>
<tr>
<td>July</td>
<td>48,926</td>
<td>106</td>
</tr>
<tr>
<td>August</td>
<td>47,991</td>
<td>98.2</td>
</tr>
<tr>
<td>September</td>
<td>35,779</td>
<td>91.0</td>
</tr>
<tr>
<td>October</td>
<td>21,645</td>
<td>81.9</td>
</tr>
<tr>
<td>November</td>
<td>2,427</td>
<td>64.4</td>
</tr>
<tr>
<td>December</td>
<td>5,963</td>
<td>53.9</td>
</tr>
<tr>
<td>Total</td>
<td>299,122</td>
<td>889.4</td>
</tr>
</tbody>
</table>
The increase in energy and demand from installing an industrial heat pump at this facility based on Compressor Intake 2 data is shown in the table below.

Table 7.14: Compressor Intake 2 Heat Pump Monthly Increased Energy and Demand

<table>
<thead>
<tr>
<th>Month</th>
<th>Heat Pump Energy Consumption (kWh)</th>
<th>Heat Pump Demand Consumption (kW)</th>
</tr>
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<tbody>
<tr>
<td>January</td>
<td>4,549</td>
<td>43.5</td>
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<tr>
<td>February</td>
<td>4,818</td>
<td>47.7</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>56.1</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>60.3</td>
</tr>
<tr>
<td>May</td>
<td>40,348</td>
<td>89.5</td>
</tr>
<tr>
<td>June</td>
<td>45,665</td>
<td>96.7</td>
</tr>
<tr>
<td>July</td>
<td>59,587</td>
<td>106</td>
</tr>
<tr>
<td>August</td>
<td>51,470</td>
<td>98.2</td>
</tr>
<tr>
<td>September</td>
<td>42,112</td>
<td>91.0</td>
</tr>
<tr>
<td>October</td>
<td>24,405</td>
<td>81.9</td>
</tr>
<tr>
<td>November</td>
<td>3,304</td>
<td>64.4</td>
</tr>
<tr>
<td>December</td>
<td>7,929</td>
<td>53.9</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>328,281</strong></td>
<td><strong>889.4</strong></td>
</tr>
</tbody>
</table>

7.2.4. Compressed Air Energy Savings

The final point of energy change as a result of installing an industrial heat pump at this facility involves the electrical energy and demand savings resulting from increased air compressor efficiency. As mentioned before, the power reduction associated with a decrease in inlet air temperatures is directly affected by the change in inlet air temperature. From this analysis, the expected energy and demand savings were found by calculating a power reduction factor for each day of the year.
The energy savings that can be realized from the installation of an industrial heat pump are found directly by taking the energy values provided by the facility and converted to daily usage. The power reduction factor for each day was found by using Equation 6.11. For example, the power reduction factor for May 18, 2012 was found to be

\[
f_{\text{reduction}} = \frac{68.5^\circ F - 45^\circ F}{68.5^\circ F + 460}
\]

\[
f_{\text{reduction}} = 0.0444
\]

which represents a 4.44% reduction in energy usage. Applied to the energy usage provided by the facility for May 18, 2012 yields an energy savings of

\[
Q_{\text{compressor savings}} = 45,701 \text{kWh} \times 0.0444
\]

\[
Q_{\text{compressor savings}} = 2,029 \frac{\text{kWh}}{\text{day}}
\]

Once these values were calculated for each individual day, they were added up by month for overall energy analysis. The values for each month are shown in the energy savings summary at the end of this section.

The average daily demand reduction for this analysis was found by taking the power reduction factor and multiplying it by the average daily demand for the compressors since 15 minute data indicated that compressed air power usage is constant. The daily demand for May 18, 2012 is found by dividing the energy savings by 24 hours. Therefore, the demand reduction for May 18, 2012 if an industrial heat pump were installed is estimated to be 84.6 kW. Once each of these daily values were calculated, the maximum demand for each month
was found and is used in the energy savings summary and for cost savings purposes. The overall energy summary for this change in energy based on data collected at Compressor Intake 1 is shown in the table below.

**Table 7.15: Compressor Intake 1 Heat Pump Energy and Demand Savings**

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>5,985</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>10.0</td>
</tr>
<tr>
<td>May</td>
<td>36,880</td>
<td>30.2</td>
</tr>
<tr>
<td>June</td>
<td>42,445</td>
<td>44.6</td>
</tr>
<tr>
<td>July</td>
<td>46,974</td>
<td>32.8</td>
</tr>
<tr>
<td>August</td>
<td>40,662</td>
<td>38.9</td>
</tr>
<tr>
<td>September</td>
<td>38,896</td>
<td>32.0</td>
</tr>
<tr>
<td>October</td>
<td>23,009</td>
<td>6.7</td>
</tr>
<tr>
<td>November</td>
<td>3,303</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>8,770</td>
<td>0.0</td>
</tr>
<tr>
<td>Total</td>
<td>301,964</td>
<td>195.2</td>
</tr>
</tbody>
</table>

The compressed air energy savings for this application based on Compressor Intake 2 data is displayed in the table below.
Table 7.16: Compressor Intake 2 Heat Pump Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>6,133</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>10.0</td>
</tr>
<tr>
<td>May</td>
<td>37,390</td>
<td>30.2</td>
</tr>
<tr>
<td>June</td>
<td>42,564</td>
<td>44.6</td>
</tr>
<tr>
<td>July</td>
<td>46,602</td>
<td>31.6</td>
</tr>
<tr>
<td>August</td>
<td>40,531</td>
<td>36.6</td>
</tr>
<tr>
<td>September</td>
<td>36,887</td>
<td>35.0</td>
</tr>
<tr>
<td>October</td>
<td>24,231</td>
<td>7.4</td>
</tr>
<tr>
<td>November</td>
<td>4,353</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>8,954</td>
<td>0.0</td>
</tr>
<tr>
<td>Total</td>
<td>302,687</td>
<td>195.3</td>
</tr>
</tbody>
</table>

7.2.5. Industrial Heat Pump Overall Energy and Cost Savings Analysis

Once all of the energy and demand savings and increases were found as the result of installing an industrial heat pump, cost savings were needed to quantify the financial benefit of the equipment upgrade for the subject facility. For electrical energy and demand savings, the same electric rates computed for the underground air pipe can be used from the Progress Energy LGS-TOU rate schedule ($0.04499 per kWh, $18.56 per kW summer demand, and $13.25 per kW other months demand). However, since there is also natural gas savings associated with this upgrade, a natural gas cost is needed.
In order to protect facility confidentiality, the natural gas rates for the subject facility will be assumed as supplied by Piedmont Natural Gas, a large natural gas provider in the region that serves many facilities of the same size. For this analysis, the rate schedule that natural gas savings will be based on is Piedmont Natural Gas 103 – Large General Sales Service. The facility is a large user of natural gas, so it is expected that the incremental price for gas is over the 600,000 Therms per month rate. This rate has a charge of $2.93 per MMBTU for the months of November through March, and a charge of $2.84 per MMBTU for the months of April through October. These rates were applied to the natural gas energy savings for each month in the analysis. This rate schedule also carries a fuel demand charge of $19.26 per MMBTU, which is applied to the fuel demand savings computed for each month. To see other fuel rates available from Piedmont Natural Gas, refer to Appendix G. Once these rates were determined, they were applied to the natural gas energy and demand savings found for installing and industrial heat pump mentioned previously. The results of this cost savings analysis for Compressor Intake 1 and Compressor Intake 2 data are shown in the tables below.
Table 7.17: Compressor Intake 1 Heat Pump Natural Gas Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Natural Gas Energy Savings (MMBTU)</th>
<th>Natural Gas Cost Savings</th>
<th>Natural Gas Demand Savings (MMBTU)</th>
<th>Natural Gas Demand Cost Savings</th>
<th>Total Natural Gas Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>111</td>
<td>$326</td>
<td>0.0</td>
<td>$0</td>
<td>$326</td>
</tr>
<tr>
<td>February</td>
<td>123</td>
<td>$361</td>
<td>0.0</td>
<td>$0</td>
<td>$361</td>
</tr>
<tr>
<td>March</td>
<td>596</td>
<td>$1,746</td>
<td>0.0</td>
<td>$0</td>
<td>$1,746</td>
</tr>
<tr>
<td>April</td>
<td>533</td>
<td>$1,516</td>
<td>3.1</td>
<td>$60</td>
<td>$1,575</td>
</tr>
<tr>
<td>May</td>
<td>1,028</td>
<td>$2,924</td>
<td>12.8</td>
<td>$247</td>
<td>$3,171</td>
</tr>
<tr>
<td>June</td>
<td>1,100</td>
<td>$3,129</td>
<td>19.3</td>
<td>$371</td>
<td>$3,500</td>
</tr>
<tr>
<td>July</td>
<td>1,252</td>
<td>$3,562</td>
<td>20.9</td>
<td>$382</td>
<td>$3,944</td>
</tr>
<tr>
<td>August</td>
<td>1,228</td>
<td>$3,494</td>
<td>19.9</td>
<td>$382</td>
<td>$3,876</td>
</tr>
<tr>
<td>September</td>
<td>916</td>
<td>$2,605</td>
<td>14.4</td>
<td>$277</td>
<td>$2,882</td>
</tr>
<tr>
<td>October</td>
<td>554</td>
<td>$1,576</td>
<td>2.2</td>
<td>$42</td>
<td>$1,618</td>
</tr>
<tr>
<td>November</td>
<td>62</td>
<td>$182</td>
<td>0.0</td>
<td>$0</td>
<td>$182</td>
</tr>
<tr>
<td>December</td>
<td>153</td>
<td>$447</td>
<td>0.0</td>
<td>$0</td>
<td>$447</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>7,657</strong></td>
<td><strong>$21,867</strong></td>
<td><strong>92.5</strong></td>
<td><strong>$1,782</strong></td>
<td><strong>$23,649</strong></td>
</tr>
</tbody>
</table>

Table 7.18: Compressor Intake 2 Heat Pump Natural Gas Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Natural Gas Energy Savings (MMBTU)</th>
<th>Natural Gas Cost Savings</th>
<th>Natural Gas Demand Savings (MMBTU)</th>
<th>Natural Gas Demand Cost Savings</th>
<th>Total Natural Gas Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>116</td>
<td>$341</td>
<td>0.0</td>
<td>$0</td>
<td>$341</td>
</tr>
<tr>
<td>February</td>
<td>123</td>
<td>$361</td>
<td>0.0</td>
<td>$0</td>
<td>$361</td>
</tr>
<tr>
<td>March</td>
<td>596</td>
<td>$1,746</td>
<td>0.0</td>
<td>$0</td>
<td>$1,746</td>
</tr>
<tr>
<td>April</td>
<td>533</td>
<td>$1,516</td>
<td>2.5</td>
<td>$60</td>
<td>$1,575</td>
</tr>
<tr>
<td>May</td>
<td>1,033</td>
<td>$2,938</td>
<td>10.3</td>
<td>$247</td>
<td>$3,185</td>
</tr>
<tr>
<td>June</td>
<td>1,169</td>
<td>$3,325</td>
<td>20.0</td>
<td>$482</td>
<td>$3,807</td>
</tr>
<tr>
<td>July</td>
<td>1,525</td>
<td>$4,338</td>
<td>17.7</td>
<td>$425</td>
<td>$4,763</td>
</tr>
<tr>
<td>August</td>
<td>1,317</td>
<td>$3,747</td>
<td>19.2</td>
<td>$462</td>
<td>$4,210</td>
</tr>
<tr>
<td>September</td>
<td>1,078</td>
<td>$3,066</td>
<td>13.6</td>
<td>$327</td>
<td>$3,393</td>
</tr>
<tr>
<td>October</td>
<td>625</td>
<td>$1,777</td>
<td>1.9</td>
<td>$46</td>
<td>$1,823</td>
</tr>
<tr>
<td>November</td>
<td>85</td>
<td>$248</td>
<td>0.0</td>
<td>$0</td>
<td>$248</td>
</tr>
<tr>
<td>December</td>
<td>203</td>
<td>$595</td>
<td>0.0</td>
<td>$0</td>
<td>$595</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>8,403</strong></td>
<td><strong>$23,997</strong></td>
<td><strong>85.2</strong></td>
<td><strong>$2,050</strong></td>
<td><strong>$26,046</strong></td>
</tr>
</tbody>
</table>
Next, the electrical energy and demand costs incurred from operation of the heat pump were found and are shown for the data collected for Compressor Intake 1 in the table below.

**Table 7.19: Compressor Intake 1 Heat Pump Electrical Energy and Demand Costs**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>4,341</td>
<td>$195</td>
<td>43.5</td>
<td>$576</td>
<td>$772</td>
</tr>
<tr>
<td>February</td>
<td>4,818</td>
<td>$217</td>
<td>47.7</td>
<td>$632</td>
<td>$849</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>$1,047</td>
<td>56.1</td>
<td>$743</td>
<td>$1,790</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>$936</td>
<td>60.3</td>
<td>$799</td>
<td>$1,735</td>
</tr>
<tr>
<td>May</td>
<td>40,161</td>
<td>$1,807</td>
<td>89.5</td>
<td>$1,186</td>
<td>$2,993</td>
</tr>
<tr>
<td>June</td>
<td>42,977</td>
<td>$1,933</td>
<td>96.7</td>
<td>$1,795</td>
<td>$3,729</td>
</tr>
<tr>
<td>July</td>
<td>48,926</td>
<td>$2,201</td>
<td>106.4</td>
<td>$1,974</td>
<td>$4,175</td>
</tr>
<tr>
<td>August</td>
<td>47,991</td>
<td>$2,159</td>
<td>98.2</td>
<td>$1,822</td>
<td>$3,981</td>
</tr>
<tr>
<td>September</td>
<td>35,779</td>
<td>$1,610</td>
<td>91.0</td>
<td>$1,689</td>
<td>$3,298</td>
</tr>
<tr>
<td>October</td>
<td>21,645</td>
<td>$974</td>
<td>81.9</td>
<td>$1,085</td>
<td>$2,058</td>
</tr>
<tr>
<td>November</td>
<td>2,427</td>
<td>$109</td>
<td>64.4</td>
<td>$853</td>
<td>$962</td>
</tr>
<tr>
<td>December</td>
<td>5,963</td>
<td>$268</td>
<td>53.9</td>
<td>$715</td>
<td>$983</td>
</tr>
<tr>
<td>Total</td>
<td>299,122</td>
<td>$13,456</td>
<td>889.4</td>
<td>$13,868</td>
<td>$27,324</td>
</tr>
</tbody>
</table>

The additional energy and demand costs associated with installing an industrial heat pump are shown in the table below for the data collected at Compressor Intake 2.
Table 7.20: Compressor Intake 2 Heat Pump Electrical Energy and Demand Costs

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>4,549</td>
<td>$205</td>
<td>43.5</td>
<td>$576</td>
<td>$781</td>
</tr>
<tr>
<td>February</td>
<td>4,818</td>
<td>$217</td>
<td>47.7</td>
<td>$632</td>
<td>$849</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>$1,047</td>
<td>56.1</td>
<td>$743</td>
<td>$1,790</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>$936</td>
<td>60.3</td>
<td>$799</td>
<td>$1,735</td>
</tr>
<tr>
<td>May</td>
<td>40,348</td>
<td>$1,815</td>
<td>89.5</td>
<td>$1,186</td>
<td>$3,001</td>
</tr>
<tr>
<td>June</td>
<td>45,665</td>
<td>$2,054</td>
<td>96.7</td>
<td>$1,795</td>
<td>$3,850</td>
</tr>
<tr>
<td>July</td>
<td>59,587</td>
<td>$2,681</td>
<td>106.4</td>
<td>$1,974</td>
<td>$4,654</td>
</tr>
<tr>
<td>August</td>
<td>51,470</td>
<td>$2,315</td>
<td>98.2</td>
<td>$1,822</td>
<td>$4,137</td>
</tr>
<tr>
<td>September</td>
<td>42,112</td>
<td>$1,894</td>
<td>91.0</td>
<td>$1,689</td>
<td>$3,583</td>
</tr>
<tr>
<td>October</td>
<td>24,405</td>
<td>$1,098</td>
<td>81.9</td>
<td>$1,085</td>
<td>$2,183</td>
</tr>
<tr>
<td>November</td>
<td>3,304</td>
<td>$149</td>
<td>64.4</td>
<td>$853</td>
<td>$1,002</td>
</tr>
<tr>
<td>December</td>
<td>7,929</td>
<td>$357</td>
<td>53.9</td>
<td>$715</td>
<td>$1,071</td>
</tr>
<tr>
<td>Total</td>
<td>328,281</td>
<td>$14,768</td>
<td>889.4</td>
<td>$13,868</td>
<td>$28,636</td>
</tr>
</tbody>
</table>

Using these same rates, the amount of compressor electrical energy and demand cost savings were computed and are shown in the tables below for Compressor Intake 1 and Compressor Intake 2 data.
Table 7.21: Compressor Intake 1 Air Compressor Energy and Demand Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Total Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>5,985</td>
<td>$269</td>
<td>0.0</td>
<td>$0</td>
<td>$269</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>$301</td>
<td>0.0</td>
<td>$0</td>
<td>$301</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>$1,113</td>
<td>0.0</td>
<td>$0</td>
<td>$1,113</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>$1,062</td>
<td>10.0</td>
<td>$132</td>
<td>$1,194</td>
</tr>
<tr>
<td>May</td>
<td>36,880</td>
<td>$1,659</td>
<td>30.2</td>
<td>$401</td>
<td>$2,060</td>
</tr>
<tr>
<td>June</td>
<td>42,445</td>
<td>$1,909</td>
<td>44.6</td>
<td>$828</td>
<td>$2,737</td>
</tr>
<tr>
<td>July</td>
<td>46,974</td>
<td>$2,113</td>
<td>32.8</td>
<td>$609</td>
<td>$2,722</td>
</tr>
<tr>
<td>August</td>
<td>40,662</td>
<td>$1,829</td>
<td>38.9</td>
<td>$722</td>
<td>$2,552</td>
</tr>
<tr>
<td>September</td>
<td>38,896</td>
<td>$1,750</td>
<td>32.0</td>
<td>$595</td>
<td>$2,344</td>
</tr>
<tr>
<td>October</td>
<td>23,009</td>
<td>$1,035</td>
<td>6.7</td>
<td>$88</td>
<td>$1,124</td>
</tr>
<tr>
<td>November</td>
<td>3,303</td>
<td>$149</td>
<td>0.0</td>
<td>$0</td>
<td>$149</td>
</tr>
<tr>
<td>December</td>
<td>8,770</td>
<td>$395</td>
<td>0.0</td>
<td>$0</td>
<td>$395</td>
</tr>
<tr>
<td>Total</td>
<td>301,964</td>
<td>$13,584</td>
<td>195.2</td>
<td>$3,375</td>
<td>$16,959</td>
</tr>
</tbody>
</table>

Table 7.22: Compressor Intake 2 Air Compressor Energy and Demand Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Energy Cost Savings</th>
<th>Compressor Demand Savings (kW)</th>
<th>Compressor Demand Cost Savings</th>
<th>Total Compressor Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>6,133</td>
<td>$276</td>
<td>0.0</td>
<td>$0</td>
<td>$276</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>$301</td>
<td>0.0</td>
<td>$0</td>
<td>$301</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>$1,113</td>
<td>0.0</td>
<td>$0</td>
<td>$1,113</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>$1,062</td>
<td>10.0</td>
<td>$185</td>
<td>$1,247</td>
</tr>
<tr>
<td>May</td>
<td>37,390</td>
<td>$1,682</td>
<td>30.2</td>
<td>$561</td>
<td>$2,243</td>
</tr>
<tr>
<td>June</td>
<td>42,564</td>
<td>$1,915</td>
<td>44.6</td>
<td>$828</td>
<td>$2,743</td>
</tr>
<tr>
<td>July</td>
<td>46,602</td>
<td>$2,096</td>
<td>31.6</td>
<td>$586</td>
<td>$2,683</td>
</tr>
<tr>
<td>August</td>
<td>40,531</td>
<td>$1,823</td>
<td>36.6</td>
<td>$679</td>
<td>$2,502</td>
</tr>
<tr>
<td>September</td>
<td>36,887</td>
<td>$1,659</td>
<td>35.0</td>
<td>$649</td>
<td>$2,308</td>
</tr>
<tr>
<td>October</td>
<td>24,231</td>
<td>$1,090</td>
<td>7.3</td>
<td>$136</td>
<td>$1,226</td>
</tr>
<tr>
<td>November</td>
<td>4,353</td>
<td>$196</td>
<td>0.0</td>
<td>$0</td>
<td>$196</td>
</tr>
<tr>
<td>December</td>
<td>8,954</td>
<td>$403</td>
<td>0.0</td>
<td>$0</td>
<td>$403</td>
</tr>
<tr>
<td>Total</td>
<td>302,687</td>
<td>$13,617</td>
<td>195.3</td>
<td>$3,624</td>
<td>$17,241</td>
</tr>
</tbody>
</table>
Once all of these values were calculated, the overall energy, demand and cost savings were determined. The results of this analysis based on the data collected at Compressor Intake 1 are shown in the table below.

**Table 7.23: Compressor Intake 1 Overall Energy and Demand Cost Savings**

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Electrical Energy Savings (kWh)</th>
<th>Overall Electrical Demand Savings (kW)</th>
<th>Overall Natural Gas Energy Savings (MMBTU)</th>
<th>Overall Natural Gas Demand Savings (MMBTU)</th>
<th>Overall Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,644</td>
<td>-43.5</td>
<td>111</td>
<td>0.0</td>
<td>-$177</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>-47.7</td>
<td>123</td>
<td>0.0</td>
<td>-$186</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>-56.1</td>
<td>596</td>
<td>0.0</td>
<td>$1,069</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>-50.3</td>
<td>533</td>
<td>3.1</td>
<td>$1,035</td>
</tr>
<tr>
<td>May</td>
<td>-3,281</td>
<td>-59.3</td>
<td>1,028</td>
<td>12.8</td>
<td>$2,238</td>
</tr>
<tr>
<td>June</td>
<td>-532</td>
<td>-52.1</td>
<td>1,100</td>
<td>19.3</td>
<td>$2,508</td>
</tr>
<tr>
<td>July</td>
<td>-1,952</td>
<td>-73.6</td>
<td>1,252</td>
<td>20.9</td>
<td>$2,511</td>
</tr>
<tr>
<td>August</td>
<td>-7,329</td>
<td>-59.2</td>
<td>1,228</td>
<td>19.9</td>
<td>$2,447</td>
</tr>
<tr>
<td>September</td>
<td>3,117</td>
<td>-58.9</td>
<td>916</td>
<td>14.4</td>
<td>$1,928</td>
</tr>
<tr>
<td>October</td>
<td>1,364</td>
<td>-75.2</td>
<td>554</td>
<td>2.2</td>
<td>$683</td>
</tr>
<tr>
<td>November</td>
<td>876</td>
<td>-64.4</td>
<td>62</td>
<td>0.0</td>
<td>-$632</td>
</tr>
<tr>
<td>December</td>
<td>2,807</td>
<td>-53.9</td>
<td>153</td>
<td>0.0</td>
<td>-$141</td>
</tr>
<tr>
<td>Total</td>
<td>2,842</td>
<td>-694.2</td>
<td>7,657</td>
<td>92.5</td>
<td>$13,284</td>
</tr>
</tbody>
</table>

The overall energy, demand, and cost savings for this application based on the data collected for Compressor Intake 2 are shown in the table below.
Table 7.24: Compressor Intake 2 Overall Energy and Demand Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Electrical Energy Savings (kWh)</th>
<th>Overall Electrical Demand Savings (kW)</th>
<th>Overall Natural Gas Energy Savings (MMBTU)</th>
<th>Overall Natural Gas Demand Savings (MMBTU)</th>
<th>Overall Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,584</td>
<td>-43.5</td>
<td>116</td>
<td>0.0</td>
<td>-$164</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>-47.7</td>
<td>123</td>
<td>0.0</td>
<td>-$186</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>-56.1</td>
<td>596</td>
<td>0.0</td>
<td>$1,069</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>-50.3</td>
<td>533</td>
<td>2.5</td>
<td>$1,088</td>
</tr>
<tr>
<td>May</td>
<td>-2,958</td>
<td>-59.3</td>
<td>1,033</td>
<td>10.3</td>
<td>$2,427</td>
</tr>
<tr>
<td>June</td>
<td>-3,101</td>
<td>-52.1</td>
<td>1,169</td>
<td>20.0</td>
<td>$2,699</td>
</tr>
<tr>
<td>July</td>
<td>-12,985</td>
<td>-74.8</td>
<td>1,525</td>
<td>17.7</td>
<td>$2,791</td>
</tr>
<tr>
<td>August</td>
<td>-10,939</td>
<td>-61.6</td>
<td>1,317</td>
<td>19.2</td>
<td>$2,574</td>
</tr>
<tr>
<td>September</td>
<td>-5,225</td>
<td>-56.0</td>
<td>1,078</td>
<td>13.6</td>
<td>$2,118</td>
</tr>
<tr>
<td>October</td>
<td>-174</td>
<td>-74.5</td>
<td>625</td>
<td>1.9</td>
<td>$867</td>
</tr>
<tr>
<td>November</td>
<td>1,049</td>
<td>-64.4</td>
<td>85</td>
<td>0.0</td>
<td>-$558</td>
</tr>
<tr>
<td>December</td>
<td>1,025</td>
<td>-53.9</td>
<td>203</td>
<td>0.0</td>
<td>-$74</td>
</tr>
<tr>
<td>Total</td>
<td>-25,595</td>
<td>-694.2</td>
<td>8,403</td>
<td>85.2</td>
<td>$14,652</td>
</tr>
</tbody>
</table>

Based on the data represented in the tables above and to remain conservative in expected energy, demand, and cost savings, the expected savings for this upgrade will be based on the results of this assessment at Compressor Intake 1. Therefore, the overall expected energy savings from this upgrade are 7,657 MMBTU of natural gas and a savings of 2,842 kWh in electricity. Natural gas demand is expected to decrease by 93 MMBTU per year and electrical demand is expected to increase by 694 kW per year. As a result of these energy savings, the expected overall cost savings of this measure is $13,284 per year. This value will
vary based on the price of natural gas, which is currently very low. If the price of natural gas increases soon, the overall cost savings of this recommendation may increase substantially.

7.2.6. Industrial Heat Pump Implementation and Payback

The implementation of an industrial heat pump system will consist of the installation of flanged air coils, a storage tank, a pump, piping, and a water cooled chiller (industrial heat pump). This system will be configured in a manner as shown in the figure below.
Figure 7.12: Industrial Heat Pump Configuration
The design parameters for this system will be a face velocity on the flanged air coils of 500 feet per minute, and a pipe velocity of 8 feet per second. The design airflow designated for this analysis is the same as considered before at 25,000,000 scf per day. Converting this value to volumetric flow as done previously for the underground air pipe system yields a volumetric flow of 17,361 cubic feet per minute. At a face velocity of 500 feet per minute as calculated previously, the amount of area needed for the flanged air coil is 34.7 ft$^2$. In order to obtain this area, several flanged air coils will be needed.

The basis for the cost estimate of the flanged air coils for this application are based on a cost per square foot of area found from a standard size available. Since the standard dimensions are given in terms of inches, the total area needed for the design face velocity is 4,997 in$^2$. For this application, the chosen standard flanged air coil is a basic water, DX, or condenser coil. This coil has copper tubes, aluminum fins, and galvanized end sheets. The tube diameter chosen for the design in the flanged air coil is 1/2” 0.017 tube with 0.0065 aluminum fins. The air coil chosen has a design of 6 rows and 8 fins per inch. The dimensions of the face of this flanged air coil for establishing a unit area cost is 35” in finned height and 90” in finned length. The face area of this coil is found to be

\[ A_f = 35" \times 90" \]

\[ A_f = 3,150 \text{ in}^2 \]
Therefore, the decimal number of the amount of flanged air coils needed for this application is

\[ N_c = \frac{4,997 \text{ in}^2}{3,150 \text{ in}^2} = 1.586 \text{ coils} \]

The cost of one of these coils for two passes is $2,433 including labor, and the price increase for 6 rows is 65% (31). As a result, the total cost for this total face area is expected to be

\[ TC_c = (1.586 \text{ coils}) \left( \frac{$2,433}{\text{coil}} \right) (1.65) \]

\[ TC_c = $6,367 \]

Based on this expected cost, the estimated cost per square foot of coil face area is found to be

\[ UC_c = \frac{$6,367}{4,997 \text{ in}^2} \times \frac{144 \text{ in}^2}{\text{ft}^2} = $183.48 \]

The next item for consideration is the size of the pipe that is needed for this application. Based on a rule of thumb of 2.4 gpm per ton of cooling needed, the total flow rate of water will be 288 gpm. Converting this value to volumetric flow rate yields

\[ \dot{V}_w = 288 \frac{\text{gal}}{\text{min}} \times \frac{0.133681 \text{ ft}^3}{\text{gal}} = 38.5 \frac{\text{ft}^3}{\text{min}} \times \frac{1 \text{ min}}{60 \text{ sec}} = 0.642 \frac{\text{ft}^3}{\text{s}} \]

Based on a design flow velocity of 8 feet per second, the total cross sectional area of pipe needed is

\[ A_p = \frac{0.642 \frac{\text{ft}^3}{\text{s}}}{8 \frac{\text{ft}}{\text{s}}} = 0.0802 \text{ ft}^2 \]
The radius of the pipe needed for this area is

\[ r_p = \sqrt{\frac{0.0802 \text{ ft}^2}{\pi}} = 0.1598 \text{ ft} \times \frac{12 \text{ in}}{\text{ft}} = 1.92 \text{ in} \]

The resulting diameter of the pipe is 3.84 inches. Therefore, 4 inch pipe is believed to provide adequate flow velocity for the design parameters. The specifications for the pipe chosen for this design are pressure pipe Class 150, SDR 18, AWWA C900, 4” diameter (31). The estimated cost of this pipe including labor is $3.52 per linear foot installed.

Based on the required flow rate, it is believed that a 5 hp pump can provide adequate flow for the system. The pump chosen for this application is a base mounted, bronze impeller, coupling guard in-line centrifugal 4” size 5 hp hydronic pump which can deliver flows up to 350 gpm with an initial cost of $8,758 including labor (31). However, it is important to note that the amount of revolutions per minute (RPM) and the amount of head will affect the capacity in gallons per minute that the pump can deliver. This pump capacity is based on 1800 RPM at a head of 60 feet (31). These parameters are expected to be adequate for this application.

Along with the pump and piping, a storage tank will be needed to collect the makeup water and condensate return for preheat of makeup water for the boilers. It is believed that a 5,000 gallon steel storage, above ground will be adequate for these purposes. This tank includes cradles, coating, fittings (not including the foundation, pumps, or piping), and is single walled with an expected cost of $6,333 including labor (31). The chiller chosen for this
application is 120 tons and is based on the maximum annual industrial heat pump demand discussed previously. The chiller chosen for this application is water cooled with dual compressors, semi-hermetic, and a tower is not included with an installation cost of $56,700 including labor (31). Each of these systems and total costs are shown in the table below as well as an anticipated 25% cost for additional expenses.

### Table 7.25: Industrial Heat Pump Implementation Costs

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Quantity</th>
<th>Unit Type</th>
<th>Cost Per Unit</th>
<th>Total Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flanged Air Coils</td>
<td>34.7</td>
<td>ft²</td>
<td>$183</td>
<td>$6,367</td>
</tr>
<tr>
<td>5,000 gallon storage tank</td>
<td>1</td>
<td>Tank</td>
<td>$6,333</td>
<td>$6,333</td>
</tr>
<tr>
<td>5 hp in-line centrifugal hydronic pump</td>
<td>2</td>
<td>Pump</td>
<td>$8,758</td>
<td>$17,516</td>
</tr>
<tr>
<td>Class 150 4” diameter PVC pipe</td>
<td>400</td>
<td>Linear Feet</td>
<td>$3.52</td>
<td>$1,408</td>
</tr>
<tr>
<td>120 ton water cooled chiller</td>
<td>1</td>
<td>Chiller</td>
<td>$56,700</td>
<td>$56,700</td>
</tr>
<tr>
<td>Subtotal</td>
<td></td>
<td></td>
<td></td>
<td>$88,324</td>
</tr>
<tr>
<td>Anticipated Additional Costs (25% of subtotal)</td>
<td></td>
<td></td>
<td></td>
<td>$22,081</td>
</tr>
<tr>
<td>Total Cost</td>
<td></td>
<td></td>
<td></td>
<td>$110,405</td>
</tr>
</tbody>
</table>

As shown in the table above, the total anticipated cost for this measure is $110,405. Since a pump is also included as part of the installation, its operating energy and cost will need to be considered in the energy and cost savings calculated previously. Based on a conversion factor of 0.746 kW per hp and an expected pump efficiency of 80%, the electrical demand added from the pump will be

\[
D_p = \frac{5 \text{ hp} \times 0.746 \text{ kW}}{0.8} = 4.66 \text{ kW}
\]
Since expected plant operation is 24 hours per day, 7 days per week, and 50 weeks per year (excluding weeks of July 4 and December 25), this pump is expected to be in operation 8,400 hours per year. Therefore, the amount of energy consumed by the pump is

\[ E_p = 4.66 \text{ kW} \times 8,400 \frac{\text{hours}}{\text{yr}} = 39,165 \frac{\text{kWh}}{\text{yr}}. \]

The combined demand cost based on the LGS-TOU rate is $15.02 based on the number of months at each demand rate, which results in a demand cost of

\[ DC_p = 4.66 \text{ kW} \times \frac{\$15.02}{\text{kW}} \times 12 \text{ months} = \frac{\$840}{\text{yr}}. \]

The increase in energy cost from pump operation will be

\[ EC_p = 39,165 \frac{\text{kWh}}{\text{yr}} \times \frac{\$0.04499}{\text{kWh}} = \frac{\$1,762}{\text{yr}}. \]

Therefore, the total increase in energy cost per year due to the pump is $2,602. With a second pump needed at the flanged air coil, the total cost to operate two pumps is $5,204. Based on the previous $13,284 considered for total savings, the actual savings when the pump energy consumption is considered is $8,080 per year. As a result, the simple payback period for this measure is 13.7 years, or 164 months.

### 7.3. Chilled Water Coil

The next option that can be used to decrease the temperature of inlet air for the air compressors involves the use of a chilled water coil. In this application, chilled water is used with a heat exchanger to cool the inlet air of the compressors to the desired temperature. This
analysis considers the benefits of using a chilled water coil as opposed to the industrial heat pump just considered. Though this recommendation carries a lower initial cost, will the loss of the natural gas savings present an option that is still economically viable?

7.3.1. **Chilled Water Coil Energy Savings**

If we only look at cooling the compressor intake air and not recovering heat to the boiler makeup water, we can use the existing chilled water system. This greatly reduces the implementation cost, but also reduces the cost savings. The energy savings associated with the installation of a chilled water coil will be the same as the energy savings from the heat pump installation without the ability to recover heat to the boilers. Therefore, these savings will consist of additional energy needed to make the chilled water and the energy saved by the compressors. Since it is expected that the temperature of the inlet air for the compressors as it exits the heat exchanger can be cooled to 45°F, this same method applies. The energy and demand savings for the air compressors observed by installing a chilled water coil based on the data collected from Compressor Intake 1 and Compressor Intake 2 are shown in the tables below.
Table 7.26: Compressor Intake 1 Chilled Water Coil Air Compressor Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>5,985</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>10.0</td>
</tr>
<tr>
<td>May</td>
<td>36,880</td>
<td>30.2</td>
</tr>
<tr>
<td>June</td>
<td>42,445</td>
<td>44.6</td>
</tr>
<tr>
<td>July</td>
<td>46,974</td>
<td>32.8</td>
</tr>
<tr>
<td>August</td>
<td>40,662</td>
<td>38.9</td>
</tr>
<tr>
<td>September</td>
<td>38,896</td>
<td>32.0</td>
</tr>
<tr>
<td>October</td>
<td>23,009</td>
<td>6.7</td>
</tr>
<tr>
<td>November</td>
<td>3,303</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>8,770</td>
<td>0.0</td>
</tr>
<tr>
<td>Total</td>
<td>301,964</td>
<td>195.2</td>
</tr>
</tbody>
</table>

Table 7.27: Compressor Intake 2 Chilled Water Coil Air Compressor Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Compressor Energy Savings (kWh)</th>
<th>Compressor Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>6,133</td>
<td>0.0</td>
</tr>
<tr>
<td>February</td>
<td>6,697</td>
<td>0.0</td>
</tr>
<tr>
<td>March</td>
<td>24,734</td>
<td>0.0</td>
</tr>
<tr>
<td>April</td>
<td>23,611</td>
<td>10.0</td>
</tr>
<tr>
<td>May</td>
<td>37,390</td>
<td>30.2</td>
</tr>
<tr>
<td>June</td>
<td>42,564</td>
<td>44.6</td>
</tr>
<tr>
<td>July</td>
<td>46,602</td>
<td>31.6</td>
</tr>
<tr>
<td>August</td>
<td>40,531</td>
<td>36.6</td>
</tr>
<tr>
<td>September</td>
<td>36,887</td>
<td>35.0</td>
</tr>
<tr>
<td>October</td>
<td>24,231</td>
<td>7.4</td>
</tr>
<tr>
<td>November</td>
<td>4,353</td>
<td>0.0</td>
</tr>
<tr>
<td>December</td>
<td>8,954</td>
<td>0.0</td>
</tr>
<tr>
<td>Total</td>
<td>302,687</td>
<td>195.3</td>
</tr>
</tbody>
</table>
Next, the additional energy needed to produce the chilled water was calculated. The additional energy needed to operate the chiller and produce chilled water to reduce the inlet air temperature to 45°F based on the data from Compressor Intake 1 and Compressor Intake 2 are shown in the tables below.

**Table 7.28: Compressor Intake 1 Chilled Water Coil Chiller Energy and Demand**

<table>
<thead>
<tr>
<th>Month</th>
<th>Chiller Energy Consumption (kWh)</th>
<th>Chiller Demand Consumption (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>4,341</td>
<td>43.5</td>
</tr>
<tr>
<td>February</td>
<td>4,818</td>
<td>47.7</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>56.1</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>60.3</td>
</tr>
<tr>
<td>May</td>
<td>40,161</td>
<td>89.5</td>
</tr>
<tr>
<td>June</td>
<td>42,977</td>
<td>96.7</td>
</tr>
<tr>
<td>July</td>
<td>48,926</td>
<td>106.4</td>
</tr>
<tr>
<td>August</td>
<td>47,991</td>
<td>98.2</td>
</tr>
<tr>
<td>September</td>
<td>35,779</td>
<td>91.0</td>
</tr>
<tr>
<td>October</td>
<td>21,645</td>
<td>81.9</td>
</tr>
<tr>
<td>November</td>
<td>2,427</td>
<td>64.4</td>
</tr>
<tr>
<td>December</td>
<td>5,963</td>
<td>53.9</td>
</tr>
<tr>
<td>Total</td>
<td>299,122</td>
<td>889.4</td>
</tr>
</tbody>
</table>
Table 7.29: Compressor Intake 2 Chilled Water Coil Chiller Energy and Demand

<table>
<thead>
<tr>
<th>Month</th>
<th>Chiller Energy Consumption (kWh)</th>
<th>Chiller Demand Consumption (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>4,549</td>
<td>43.5</td>
</tr>
<tr>
<td>February</td>
<td>4,818</td>
<td>47.7</td>
</tr>
<tr>
<td>March</td>
<td>23,279</td>
<td>56.1</td>
</tr>
<tr>
<td>April</td>
<td>20,817</td>
<td>60.3</td>
</tr>
<tr>
<td>May</td>
<td>40,348</td>
<td>89.5</td>
</tr>
<tr>
<td>June</td>
<td>45,665</td>
<td>96.7</td>
</tr>
<tr>
<td>July</td>
<td>59,587</td>
<td>106.4</td>
</tr>
<tr>
<td>August</td>
<td>51,470</td>
<td>98.2</td>
</tr>
<tr>
<td>September</td>
<td>42,112</td>
<td>91.0</td>
</tr>
<tr>
<td>October</td>
<td>24,405</td>
<td>81.9</td>
</tr>
<tr>
<td>November</td>
<td>3,304</td>
<td>64.4</td>
</tr>
<tr>
<td>December</td>
<td>7,929</td>
<td>53.9</td>
</tr>
<tr>
<td>Total</td>
<td>328,281</td>
<td>889.4</td>
</tr>
</tbody>
</table>

As a result of these analyses, the overall energy and demand savings from installing a chilled water coil to reduce inlet air temperatures for these compressors was calculated. The results of this analysis for Compressor Intake 1 data are shown in the table below.
Table 7.30: Compressor Intake 1 Chilled Water Coil Overall Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Energy Savings (kWh)</th>
<th>Overall Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,644</td>
<td>-43.5</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>-47.7</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>-56.1</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>-50.3</td>
</tr>
<tr>
<td>May</td>
<td>-3,281</td>
<td>-59.3</td>
</tr>
<tr>
<td>June</td>
<td>-532</td>
<td>-52.1</td>
</tr>
<tr>
<td>July</td>
<td>-1,952</td>
<td>-73.6</td>
</tr>
<tr>
<td>August</td>
<td>-7,329</td>
<td>-59.2</td>
</tr>
<tr>
<td>September</td>
<td>3,117</td>
<td>-58.9</td>
</tr>
<tr>
<td>October</td>
<td>1,364</td>
<td>-75.2</td>
</tr>
<tr>
<td>November</td>
<td>876</td>
<td>-64.4</td>
</tr>
<tr>
<td>December</td>
<td>2,807</td>
<td>-53.9</td>
</tr>
<tr>
<td>Total</td>
<td>2,842</td>
<td>-694.2</td>
</tr>
</tbody>
</table>

The overall energy and demand savings from installing a chilled water coil based on data collected at Compressor Intake 2 are displayed in the table below.
Table 7.31: Compressor Intake 2 Chilled Water Coil Overall Energy and Demand Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Energy Savings (kWh)</th>
<th>Overall Demand Savings (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,584</td>
<td>-43.5</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>-47.7</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>-56.1</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>-50.3</td>
</tr>
<tr>
<td>May</td>
<td>-2,958</td>
<td>-59.3</td>
</tr>
<tr>
<td>June</td>
<td>-3,101</td>
<td>-52.1</td>
</tr>
<tr>
<td>July</td>
<td>-12,985</td>
<td>-74.8</td>
</tr>
<tr>
<td>August</td>
<td>-10,939</td>
<td>-61.6</td>
</tr>
<tr>
<td>September</td>
<td>-5,225</td>
<td>-56.0</td>
</tr>
<tr>
<td>October</td>
<td>-174</td>
<td>-74.5</td>
</tr>
<tr>
<td>November</td>
<td>1,049</td>
<td>-64.4</td>
</tr>
<tr>
<td>December</td>
<td>1,025</td>
<td>-53.9</td>
</tr>
<tr>
<td>Total</td>
<td>-25,595</td>
<td>-694.2</td>
</tr>
</tbody>
</table>

7.3.2. Chilled Water Coil Cost Savings

Based on the energy savings calculated from installing a chilled water coil, the cost savings for each compressor intake was found. The electrical energy rates used to find the cost savings for this option are the same rates used previously (Progress Energy LGS-TOU). The energy and demand cost savings for installing a chilled water coil based on the data collected from Compressor Intake 1 and Compressor Intake 2 are shown in the tables below.
### Table 7.32: Compressor Intake 1 Chilled Water Overall Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Energy Savings (kWh)</th>
<th>Overall Energy Cost Savings</th>
<th>Overall Demand Savings (kW)</th>
<th>Overall Demand Cost Savings</th>
<th>Total Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,644</td>
<td>$74</td>
<td>-43.5</td>
<td>-$576</td>
<td>-$502</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>$85</td>
<td>-47.7</td>
<td>-$632</td>
<td>-$547</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>$65</td>
<td>-56.1</td>
<td>-$743</td>
<td>-$678</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>$126</td>
<td>-50.3</td>
<td>-$666</td>
<td>-$541</td>
</tr>
<tr>
<td>May</td>
<td>-3,281</td>
<td>-$148</td>
<td>-59.3</td>
<td>-$785</td>
<td>-$933</td>
</tr>
<tr>
<td>June</td>
<td>-532</td>
<td>-$24</td>
<td>-52.1</td>
<td>-$968</td>
<td>-$992</td>
</tr>
<tr>
<td>July</td>
<td>-1,952</td>
<td>-$88</td>
<td>-73.6</td>
<td>-$1,365</td>
<td>-$1,453</td>
</tr>
<tr>
<td>August</td>
<td>-7,329</td>
<td>-$330</td>
<td>-59.2</td>
<td>-$1,099</td>
<td>-$1,429</td>
</tr>
<tr>
<td>September</td>
<td>3,117</td>
<td>$140</td>
<td>-58.9</td>
<td>-$1,094</td>
<td>-$954</td>
</tr>
<tr>
<td>October</td>
<td>1,364</td>
<td>$61</td>
<td>-75.2</td>
<td>-$996</td>
<td>-$935</td>
</tr>
<tr>
<td>November</td>
<td>876</td>
<td>$39</td>
<td>-64.4</td>
<td>-$853</td>
<td>-$814</td>
</tr>
<tr>
<td>December</td>
<td>2,807</td>
<td>$126</td>
<td>-53.9</td>
<td>-$715</td>
<td>-$588</td>
</tr>
<tr>
<td>Total</td>
<td>2,842</td>
<td>$128</td>
<td>-694.2</td>
<td>-$10,493</td>
<td>-$10,365</td>
</tr>
</tbody>
</table>

### Table 7.33: Compressor Intake 2 Chilled Water Overall Cost Savings

<table>
<thead>
<tr>
<th>Month</th>
<th>Overall Energy Savings (kWh)</th>
<th>Overall Energy Cost Savings</th>
<th>Overall Demand Savings (kW)</th>
<th>Overall Demand Cost Savings</th>
<th>Total Cost Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>January</td>
<td>1,584</td>
<td>$71</td>
<td>-43.5</td>
<td>-$576</td>
<td>-$505</td>
</tr>
<tr>
<td>February</td>
<td>1,879</td>
<td>$85</td>
<td>-47.7</td>
<td>-$632</td>
<td>-$547</td>
</tr>
<tr>
<td>March</td>
<td>1,456</td>
<td>$65</td>
<td>-56.1</td>
<td>-$743</td>
<td>-$678</td>
</tr>
<tr>
<td>April</td>
<td>2,794</td>
<td>$126</td>
<td>-50.3</td>
<td>-$613</td>
<td>-$488</td>
</tr>
<tr>
<td>May</td>
<td>-2,958</td>
<td>-$133</td>
<td>-59.3</td>
<td>-$625</td>
<td>-$758</td>
</tr>
<tr>
<td>June</td>
<td>-3,101</td>
<td>-$140</td>
<td>-52.1</td>
<td>-$968</td>
<td>-$1,107</td>
</tr>
<tr>
<td>July</td>
<td>-12,985</td>
<td>-$584</td>
<td>-74.8</td>
<td>-$1,388</td>
<td>-$1,972</td>
</tr>
<tr>
<td>August</td>
<td>-10,939</td>
<td>-$492</td>
<td>-61.6</td>
<td>-$1,143</td>
<td>-$1,635</td>
</tr>
<tr>
<td>September</td>
<td>-5,225</td>
<td>-$235</td>
<td>-56.0</td>
<td>-$1,040</td>
<td>-$1,275</td>
</tr>
<tr>
<td>October</td>
<td>-174</td>
<td>-$8</td>
<td>-74.5</td>
<td>-$948</td>
<td>-$956</td>
</tr>
<tr>
<td>November</td>
<td>1,049</td>
<td>$47</td>
<td>-64.4</td>
<td>-$853</td>
<td>-$806</td>
</tr>
<tr>
<td>December</td>
<td>1,025</td>
<td>$46</td>
<td>-53.9</td>
<td>-$715</td>
<td>-$669</td>
</tr>
<tr>
<td>Total</td>
<td>-25,595</td>
<td>-$1,151</td>
<td>-694.2</td>
<td>-$10,244</td>
<td>-$11,395</td>
</tr>
</tbody>
</table>
In order to remain more conservative in savings estimates, the cost savings for Compressor Intake 2 are considered for this analysis. As a result, it would require 25,595 kWh more of electrical energy, 694 kW more of electrical demand, and an additional $11,395 per year to generate enough chilled water to produce inlet air for the compressors at or below the desired 45°F.

### 7.3.3. Chilled Water Coil Implementation and Payback

The installation of a chilled water coil will require that flanged air coils be installed along with piping from the existing chilled water system as illustrated in the figure below.

![Chilled Water Coil Installation Diagram](image)

**Figure 7.13: Chilled Water Coil Installation**
It is assumed that the existing pumps will be sufficient to send chilled water to the flanged air coil for this application. The costs of these components are the same as those outlined in the industrial heat pump analysis. The cost breakdown of each of these components for a chilled water coil is shown in the table below.

### Table 7.34: Chilled Water Coil Implementation Costs

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Quantity</th>
<th>Unit Type</th>
<th>Cost Per Unit</th>
<th>Total Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flanged Air Coils</td>
<td>34.7</td>
<td>ft^2</td>
<td>$183</td>
<td>$6,367</td>
</tr>
<tr>
<td>Class 150 4” diameter PVC pipe</td>
<td>400</td>
<td>Linear Feet</td>
<td>$3.52</td>
<td>$1,408</td>
</tr>
<tr>
<td>Subtotal</td>
<td></td>
<td></td>
<td></td>
<td>$7,775</td>
</tr>
<tr>
<td>Anticipated Additional Costs (25% of subtotal)</td>
<td></td>
<td></td>
<td></td>
<td>$1,944</td>
</tr>
<tr>
<td>Total Cost</td>
<td></td>
<td></td>
<td></td>
<td>$9,718</td>
</tr>
</tbody>
</table>

As shown in the table above, the total expected implementation cost for a chilled water coil is $9,718. Since the electricity required in the production of adequate chilled water for heat removal costs more than the amount of electricity saved at the compressors, this method possesses an infinite simple payback period.
Chapter 8  Conclusions and Recommendations

The purpose of this experiment was to improve industrial reciprocating and centrifugal air compressors performance by cooling inlet air temperatures as well as analyzing the energy and costs benefits of heat recovery to the steam system. The first step in this process involved acquiring temperature, relative humidity, energy, and compressed air flow data for comparison of compressor efficiency at various temperatures. Once these values were collected and analyzed, the energy and cost savings were found for an underground earth tube system, an industrial heat pump, and a chilled water coil. Once the savings were computed, they were compared against the initial implementation cost of the project in order to determine the economic feasibility of each option.

8.1.  Conclusions

As a result of this experiment, it is clear that inlet air temperature does directly affect the performance of electrically driven reciprocating and centrifugal air compressors. As the temperature of air decreases, the power consumption of these compressors decreases by a power reduction factor of approximately 0.0012 per degree Fahrenheit drop in temperature. This value is also consistent with the theoretical power reduction of 0.0018 per degree Fahrenheit. These values were determined through analysis of experimental data obtained from data logging equipment and weather data obtained from the National Weather Service. One apparent limitation of this project is the consideration of two stage compression with
intercooling, which negates half of the potential energy and demand savings since the temperature after the intercooler is expected to be the same regardless of inlet air cooling.

After applying this analysis to the underground earth tube option, it was determined that a system of 400 four inch diameter pipes 50 feet in length would be needed to provide a reasonable implementation cost. Once the actual heat removed by the ground was determined for comparison to the theoretical best case, the final compressor demand savings were determined to be 35 kW, the final energy savings were determined to be 85,158 kWh, and the overall cost savings was found to be $4,471 per year. With an initial anticipated implementation cost of $100,500, the simple payback period for this recommendation is 22.5 years.

Due to the high implementation cost and low cost savings, an underground earth tube system is not economically attractive for the initial investment. It is also important to note that these energy and cost savings do not consider the rise in ground temperature due to the heat removal from the inlet air. Though the ground is considered to be an infinite heat source in theory, most of the heat removed will remain in the vicinity of the air pipe system. Though the effectiveness does account for actual heat removed while the air is in the heat exchanger, it does not account for this heat addition to the ground since the ground is treated as an infinite heat source.
The next option considered in this analysis was the use of an industrial heat pump system for heat removal from the air and supply of that waste heat for the preheating of boiler makeup water. This analysis involved considerations of heat removed from the air, heat added from the compressor work of the heat pump, and power reduction from the compressors. As a result of this analysis, it was determined that possible energy savings from this recommendation would be -36,323 kWh in electricity, 689 kW savings in electrical demand, 7,657 MMBTU savings in natural gas consumption, and 93 MMBTU savings in natural gas demand. The overall cost savings when all of these factors are considered is $8,080 per year. With an expected implementation cost of $110,405, the simple payback period of this recommendation is expected to be 13.7 years.

Though the implementation cost of this project is significant, it provides better economic feasibility to the facility in regard to payback period. The greatest limitation to the economic feasibility of this option outside of the two stage compression discussed previously is the current low price of natural gas. For this recommendation to be a more attractive option, the price of natural gas would need to increase significantly. It is recommended that before a project such as this one is implemented, natural gas prices should be monitored to find a time when the fuel cost savings would be much greater.

The final option considered in this analysis was the use of a chilled water coil with existing plant chilled water. This analysis involved the calculation of energy savings at the
compressors and the amount of energy consumed by the chiller to make enough chilled water to remove the proper amount of heat from the air. The results of this analysis show that the demand increase associated with this option would be 694 kW, the increase in energy consumption was found to be 25,595 kWh, and the cost increase was found to be $11,395 per year. With an expected implementation cost of $9,718, the simple payback period for this measure is expected to be infinite.

Since this project is not currently cost beneficial, it is not recommended for implementation at this time. Though this option has the smallest implementation cost, it costs more to generate than the savings would be. The most significant factor in this loss of revenue is the demand savings. Since the new demand from the savings is established when the demand reduction is at a minimum and the demand increase in chilled water by the maximum, the demand to generate the necessary chilled water consistently outweighs the potential savings. This issue also prevented the industrial heat pump from being a better option since the compressor demand savings and boiler natural gas demand savings were based on a demand reduction minimum.

8.2. Recommendations

The analyses of these three systems indicate that the driving force behind cost savings for cooling inlet temperatures to the compressors is the natural gas savings from heat recovery. If one of these systems were to be implemented in this facility, it is recommended to pursue the
industrial heat pump option because it possesses the best payback period. However, this system should only be pursued if natural gas prices rise. It is also recommended that the new system be maintained properly for the most efficient operation.

8.3. **Suggestions for Future Work**

In order to further determine the impact of cooling inlet air temperatures to improve performance of industrial reciprocating and centrifugal air compressors, more widespread projects should be pursued to determine economic feasibility. This project only considered the potential for these systems at a subject facility in North Carolina. Another study that would be beneficial to this area of research would be to investigate the feasibility of these systems in various other locations in the United States and in the world. If these same types of measurements were taken at a location where temperatures are much higher, the amount of savings that can be realized would be much greater and thus reduce the payback period. However, due to time and logistical restraints such a study would more than likely require the use of meteorological data rather than experimental data.

One of the most significant limitations of this study involved the increments of data available. A suggestion for future projects like this one would be to find a way to gather all data at smaller increments to gather more accurate results than the daily averages and totals that were used in this experiment. In order to achieve this higher accuracy, starting the project further in advance would be desirable. Starting earlier would allow ample time to
have complete annual data and enable data logging equipment to be programmed for shorter intervals. Another method of increasing accuracy would be to find a method of gathering energy consumption data before and after the intercoolers of two stage compressors.
REFERENCES


APPENDIX
# Appendix A  Molar Mass, Gas Constant, and Critical-Point Properties

**Table A.1: Molar Mass, Gas Constant, and Critical-Point Properties (SI)**

<table>
<thead>
<tr>
<th>Substance</th>
<th>Formula</th>
<th>Molar mass, $M$ kg/kmol</th>
<th>Gas constant, $R$ kJ/kg·K&lt;sup&gt;*&lt;/sup&gt;</th>
<th>Temperature, $T$ K</th>
<th>Pressure, $P$ MPa</th>
<th>Volume, $V$ m&lt;sup&gt;3&lt;/sup&gt;/kmol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>$-$</td>
<td>28.97</td>
<td>0.2870</td>
<td>132.5</td>
<td>3.77</td>
<td>0.0883</td>
</tr>
<tr>
<td>Ammonia</td>
<td>$\text{NH}_3$</td>
<td>17.03</td>
<td>0.4882</td>
<td>405.5</td>
<td>11.28</td>
<td>0.0724</td>
</tr>
<tr>
<td>Argon</td>
<td>$\text{Ar}$</td>
<td>39.948</td>
<td>0.2081</td>
<td>151</td>
<td>4.86</td>
<td>0.0749</td>
</tr>
<tr>
<td>Benzene</td>
<td>$\text{C}_6\text{H}_6$</td>
<td>78.115</td>
<td>0.1064</td>
<td>562</td>
<td>4.92</td>
<td>0.2603</td>
</tr>
<tr>
<td>Bromine</td>
<td>$\text{Br}_2$</td>
<td>159.808</td>
<td>0.0520</td>
<td>584</td>
<td>10.34</td>
<td>0.1355</td>
</tr>
<tr>
<td>n-Butane</td>
<td>$\text{C}<em>4\text{H}</em>{10}$</td>
<td>58.124</td>
<td>0.1430</td>
<td>425.2</td>
<td>3.80</td>
<td>0.2547</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>$\text{CO}_2$</td>
<td>44.01</td>
<td>0.1889</td>
<td>304.2</td>
<td>7.39</td>
<td>0.0943</td>
</tr>
<tr>
<td>Carbon monoxide</td>
<td>$\text{CO}$</td>
<td>28.011</td>
<td>0.2968</td>
<td>133</td>
<td>3.50</td>
<td>0.0930</td>
</tr>
<tr>
<td>Carbon tetrachloride</td>
<td>$\text{CCl}_4$</td>
<td>153.82</td>
<td>0.05405</td>
<td>556.4</td>
<td>4.56</td>
<td>0.2759</td>
</tr>
<tr>
<td>Chlorine</td>
<td>$\text{Cl}_2$</td>
<td>70.906</td>
<td>0.1173</td>
<td>417</td>
<td>7.71</td>
<td>0.1242</td>
</tr>
<tr>
<td>Chloroform</td>
<td>$\text{CHCl}_3$</td>
<td>119.38</td>
<td>0.06964</td>
<td>536.6</td>
<td>5.47</td>
<td>0.2403</td>
</tr>
<tr>
<td>Dichlorodifluoromethane (R-12)</td>
<td>$\text{CCl}_2\text{F}_2$</td>
<td>120.91</td>
<td>0.06876</td>
<td>384.7</td>
<td>4.01</td>
<td>0.2179</td>
</tr>
<tr>
<td>Dichlorotrifluoromethane (R-21)</td>
<td>$\text{CHCl}_3\text{F}$</td>
<td>102.92</td>
<td>0.08078</td>
<td>451.7</td>
<td>5.17</td>
<td>0.1973</td>
</tr>
<tr>
<td>Ethane</td>
<td>$\text{C}_2\text{H}_6$</td>
<td>30.070</td>
<td>0.2765</td>
<td>305.5</td>
<td>4.48</td>
<td>0.1480</td>
</tr>
<tr>
<td>Ethyl alcohol</td>
<td>$\text{C}_2\text{H}_5\text{OH}$</td>
<td>46.07</td>
<td>0.1805</td>
<td>516</td>
<td>6.38</td>
<td>0.1673</td>
</tr>
<tr>
<td>Ethylene</td>
<td>$\text{C}_2\text{H}_4$</td>
<td>28.054</td>
<td>0.2964</td>
<td>282.4</td>
<td>5.12</td>
<td>0.1242</td>
</tr>
<tr>
<td>Helium</td>
<td>$\text{He}$</td>
<td>4.003</td>
<td>2.0769</td>
<td>5.3</td>
<td>0.23</td>
<td>0.0578</td>
</tr>
<tr>
<td>n-Hexane</td>
<td>$\text{C}<em>6\text{H}</em>{14}$</td>
<td>86.179</td>
<td>0.09647</td>
<td>507.9</td>
<td>3.03</td>
<td>0.3677</td>
</tr>
<tr>
<td>Hydrogen (normal)</td>
<td>$\text{H}_2$</td>
<td>2.016</td>
<td>4.1240</td>
<td>33.3</td>
<td>1.30</td>
<td>0.0649</td>
</tr>
<tr>
<td>Krypton</td>
<td>$\text{Kr}$</td>
<td>83.80</td>
<td>0.09921</td>
<td>209.4</td>
<td>5.50</td>
<td>0.0924</td>
</tr>
<tr>
<td>Methane</td>
<td>$\text{CH}_4$</td>
<td>16.043</td>
<td>0.5182</td>
<td>191.1</td>
<td>4.64</td>
<td>0.0993</td>
</tr>
<tr>
<td>Methyl alcohol</td>
<td>$\text{CH}_3\text{OH}$</td>
<td>32.042</td>
<td>0.2595</td>
<td>513.2</td>
<td>7.95</td>
<td>0.1180</td>
</tr>
<tr>
<td>Methyl chloride</td>
<td>$\text{CH}_3\text{Cl}$</td>
<td>50.488</td>
<td>0.1647</td>
<td>416.3</td>
<td>6.68</td>
<td>0.1430</td>
</tr>
<tr>
<td>Neon</td>
<td>$\text{Ne}$</td>
<td>20.183</td>
<td>0.4119</td>
<td>44.5</td>
<td>2.73</td>
<td>0.0417</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>$\text{N}_2$</td>
<td>28.013</td>
<td>0.2968</td>
<td>126.2</td>
<td>3.39</td>
<td>0.0899</td>
</tr>
<tr>
<td>Nitrous oxide</td>
<td>$\text{N}_2\text{O}$</td>
<td>44.013</td>
<td>0.1889</td>
<td>309.7</td>
<td>7.27</td>
<td>0.0961</td>
</tr>
<tr>
<td>Oxygen</td>
<td>$\text{O}_2$</td>
<td>31.999</td>
<td>0.2598</td>
<td>154.8</td>
<td>5.08</td>
<td>0.0780</td>
</tr>
<tr>
<td>Propane</td>
<td>$\text{C}_3\text{H}_8$</td>
<td>44.097</td>
<td>0.1885</td>
<td>370</td>
<td>4.26</td>
<td>0.1998</td>
</tr>
<tr>
<td>Propanol</td>
<td>$\text{C}_3\text{H}_6$</td>
<td>42.081</td>
<td>0.1976</td>
<td>365</td>
<td>4.62</td>
<td>0.1810</td>
</tr>
<tr>
<td>Sulfur dioxide</td>
<td>$\text{SO}_2$</td>
<td>64.063</td>
<td>0.1298</td>
<td>430.7</td>
<td>7.88</td>
<td>0.1217</td>
</tr>
<tr>
<td>Tetrafluoroethane (R-134a)</td>
<td>$\text{CF}_2\text{CH}_2\text{F}$</td>
<td>102.03</td>
<td>0.08149</td>
<td>374.2</td>
<td>4.059</td>
<td>0.1993</td>
</tr>
<tr>
<td>Trichlorofluoromethane (R-11)</td>
<td>$\text{CCl}_3\text{F}$</td>
<td>137.37</td>
<td>0.06052</td>
<td>471.2</td>
<td>4.38</td>
<td>0.2478</td>
</tr>
<tr>
<td>Water</td>
<td>$\text{H}_2\text{O}$</td>
<td>18.015</td>
<td>0.4615</td>
<td>647.1</td>
<td>22.06</td>
<td>0.0560</td>
</tr>
<tr>
<td>Xenon</td>
<td>$\text{Xe}$</td>
<td>131.30</td>
<td>0.06332</td>
<td>289.8</td>
<td>5.88</td>
<td>0.1186</td>
</tr>
</tbody>
</table>

*The unit kJ/kg·K is equivalent to kPa·m<sup>3</sup>/kg·K<sup>*</sup>. The gas constant is calculated from $R = R_\gamma/M$, where $R_\gamma = 8.31447$ kJ/kmol·K and $M$ is the molar mass.*
Table A.2: Molar Mass, Gas Constant, and Critical-Point Properties (English)

<table>
<thead>
<tr>
<th>Substance</th>
<th>Formula</th>
<th>Molar mass, $M$ (lbm/lbmol)</th>
<th>Btu/lbm-R°</th>
<th>psia-ft°/lbm-R°</th>
<th>Temperature, $T_c$ (R)</th>
<th>Pressure, $P_c$ (psia)</th>
<th>Volume, $V_c$ (ft°/lbm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>—</td>
<td>28.97</td>
<td>0.06855</td>
<td>0.3704</td>
<td>238.5</td>
<td>547</td>
<td>1.41</td>
</tr>
<tr>
<td>Ammonia</td>
<td>NH₃</td>
<td>17.03</td>
<td>0.1166</td>
<td>0.6301</td>
<td>729.8</td>
<td>1636</td>
<td>1.16</td>
</tr>
<tr>
<td>Argon</td>
<td>Ar</td>
<td>39.948</td>
<td>0.04971</td>
<td>0.2686</td>
<td>272</td>
<td>705</td>
<td>1.20</td>
</tr>
<tr>
<td>Benzene</td>
<td>C₆H₆</td>
<td>78.115</td>
<td>0.02542</td>
<td>0.1374</td>
<td>1012</td>
<td>714</td>
<td>4.17</td>
</tr>
<tr>
<td>Bromine</td>
<td>Br₂</td>
<td>159.908</td>
<td>0.01243</td>
<td>0.06714</td>
<td>1052</td>
<td>1500</td>
<td>2.17</td>
</tr>
<tr>
<td>n-Butane</td>
<td>C₄H₁₀</td>
<td>58.124</td>
<td>0.03417</td>
<td>0.1846</td>
<td>765.2</td>
<td>551</td>
<td>4.08</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>CO₂</td>
<td>44.01</td>
<td>0.04513</td>
<td>0.2438</td>
<td>547.5</td>
<td>1071</td>
<td>1.51</td>
</tr>
<tr>
<td>Carbon monoxide</td>
<td>CO</td>
<td>28.011</td>
<td>0.07090</td>
<td>0.3831</td>
<td>240</td>
<td>507</td>
<td>1.49</td>
</tr>
<tr>
<td>Carbon tetrachloride</td>
<td>CCl₄</td>
<td>153.82</td>
<td>0.01291</td>
<td>0.06976</td>
<td>1001.5</td>
<td>661</td>
<td>4.42</td>
</tr>
<tr>
<td>Chlorine</td>
<td>Cl₂</td>
<td>70.906</td>
<td>0.02801</td>
<td>0.1517</td>
<td>751</td>
<td>1120</td>
<td>1.99</td>
</tr>
<tr>
<td>Chloroform</td>
<td>CHCl₃</td>
<td>119.38</td>
<td>0.01664</td>
<td>0.08988</td>
<td>965.8</td>
<td>794</td>
<td>3.85</td>
</tr>
<tr>
<td>Dichlorodifluoromethane (R-12)</td>
<td>CCl₂F₂</td>
<td>120.91</td>
<td>0.01643</td>
<td>0.08874</td>
<td>692.4</td>
<td>582</td>
<td>3.49</td>
</tr>
<tr>
<td>Dichlorotrifluoromethane (R-21)</td>
<td>CHCl₃F</td>
<td>102.92</td>
<td>0.01930</td>
<td>0.1043</td>
<td>813.0</td>
<td>749</td>
<td>3.16</td>
</tr>
<tr>
<td>Ethane</td>
<td>C₂H₆</td>
<td>30.020</td>
<td>0.06616</td>
<td>0.3574</td>
<td>549.8</td>
<td>708</td>
<td>2.37</td>
</tr>
<tr>
<td>Ethyl alcohol</td>
<td>C₃H₇OH</td>
<td>46.07</td>
<td>0.04311</td>
<td>0.2329</td>
<td>929.0</td>
<td>926</td>
<td>2.68</td>
</tr>
<tr>
<td>Ethylene</td>
<td>C₂H₄</td>
<td>28.054</td>
<td>0.07079</td>
<td>0.3825</td>
<td>508.3</td>
<td>742</td>
<td>1.99</td>
</tr>
<tr>
<td>Helium</td>
<td>He</td>
<td>4.003</td>
<td>0.4961</td>
<td>2.6809</td>
<td>9.5</td>
<td>33.2</td>
<td>0.926</td>
</tr>
<tr>
<td>n-Hexane</td>
<td>C₆H₁₄</td>
<td>86.178</td>
<td>0.02305</td>
<td>0.1245</td>
<td>914.2</td>
<td>439</td>
<td>5.89</td>
</tr>
<tr>
<td>Hydrogen (normal)</td>
<td>H₂</td>
<td>2.016</td>
<td>0.9851</td>
<td>5.3224</td>
<td>59.9</td>
<td>188.1</td>
<td>1.04</td>
</tr>
<tr>
<td>Krypton</td>
<td>Kr</td>
<td>83.80</td>
<td>0.02370</td>
<td>0.1280</td>
<td>376.9</td>
<td>798</td>
<td>1.48</td>
</tr>
<tr>
<td>Methane</td>
<td>CH₄</td>
<td>16.043</td>
<td>0.1238</td>
<td>0.6688</td>
<td>343.9</td>
<td>673</td>
<td>1.59</td>
</tr>
<tr>
<td>Methyl alcohol</td>
<td>CH₃OH</td>
<td>32.042</td>
<td>0.06198</td>
<td>0.3349</td>
<td>923.7</td>
<td>1154</td>
<td>1.89</td>
</tr>
<tr>
<td>Methyl chloride</td>
<td>CH₃Cl</td>
<td>50.488</td>
<td>0.03934</td>
<td>0.2125</td>
<td>749.3</td>
<td>968</td>
<td>2.29</td>
</tr>
<tr>
<td>Neon</td>
<td>Ne</td>
<td>20.183</td>
<td>0.09840</td>
<td>0.5316</td>
<td>80.1</td>
<td>395</td>
<td>0.668</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N₂</td>
<td>28.013</td>
<td>0.07090</td>
<td>0.3830</td>
<td>227.1</td>
<td>492</td>
<td>1.44</td>
</tr>
<tr>
<td>Nitrous oxide</td>
<td>N₂O</td>
<td>44.013</td>
<td>0.04512</td>
<td>0.2438</td>
<td>557.4</td>
<td>1054</td>
<td>1.54</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O₂</td>
<td>31.999</td>
<td>0.06206</td>
<td>0.3353</td>
<td>278.6</td>
<td>736</td>
<td>1.25</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>44.097</td>
<td>0.04504</td>
<td>0.2433</td>
<td>665.9</td>
<td>617</td>
<td>3.20</td>
</tr>
<tr>
<td>Propylene</td>
<td>C₃H₆</td>
<td>42.081</td>
<td>0.04719</td>
<td>0.2550</td>
<td>656.9</td>
<td>670</td>
<td>2.90</td>
</tr>
<tr>
<td>Sulfur dioxide</td>
<td>SO₂</td>
<td>64.063</td>
<td>0.03100</td>
<td>1.1675</td>
<td>775.2</td>
<td>1143</td>
<td>1.96</td>
</tr>
<tr>
<td>Tetrafluoroethane (R-134a)</td>
<td>CF₃CH₂F</td>
<td>102.03</td>
<td>0.01946</td>
<td>0.1052</td>
<td>673.6</td>
<td>588.7</td>
<td>3.19</td>
</tr>
<tr>
<td>Trichlorofluoromethane (R-11)</td>
<td>CCl₃F</td>
<td>137.37</td>
<td>0.01446</td>
<td>0.07811</td>
<td>848.1</td>
<td>635</td>
<td>3.97</td>
</tr>
<tr>
<td>Water</td>
<td>H₂O</td>
<td>18.015</td>
<td>0.1102</td>
<td>0.5956</td>
<td>1164.8</td>
<td>3200</td>
<td>0.90</td>
</tr>
<tr>
<td>Xenon</td>
<td>Xe</td>
<td>131.30</td>
<td>0.01513</td>
<td>0.08172</td>
<td>521.55</td>
<td>852</td>
<td>1.90</td>
</tr>
</tbody>
</table>

*Calculated from $R = R_c/M$, where $R_c = 1.98588$ Btu/lbm-R° = 10.7316 psia-ft°/lbmol-R° and $M$ is the molar mass.
Appendix B  Generalized Compressibility Charts

Figure B.1: Generalized Compressibility Chart at Low Pressures ($0 < P_R < 1.0$)  (4)
Figure B.2: Generalized Compressibility Chart at Intermediate Pressures

\(0<P_{R}<7\)
Figure B.3: Compressibility Factor $Z$ for $P_r \leq 1$ (32)
Figure B.4: Compressibility Factor $Z$ for $P_r \leq 10$ (32)
Figure B.5: Compressibility Factor $Z$ for $10 \leq P_r \leq 40$  \hfill (32)
Appendix C  Psychrometric Charts

Psychrometric charts for the SI and English unit systems are shown on the following pages.
Figure C.1: Psychrometric Chart at 1 atm (SI Units)
Figure C.2: Psychromeric Chart at 1atm (English Units)
Appendix D   Temperature and Relative Humidity Data

The following data was collected from data loggers placed inside of the compressor intakes and outside of the compressor house. Each week is broken down into segments of approximately seven days. Earlier data sets may be slightly shorter or longer since data was broken apart based on the data collection date. Any important occurrence with a particular set of data is described with that particular data set.

D.1. Week 1 Temperature and Relative Humidity Data

The time period chosen for week 1 data occurred between Thursday, May 17, 2012 and Thursday, May 24, 2012. Raw temperature data collected at five minute intervals for week 1 are shown in the graph below.

![Graph showing temperature data for Week 1 with labels: Compressor 1, Compressor 2, Outside Air.]

Figure D.1: Week 1 Raw Temperature Data
Week 1 raw relative humidity data taken at five minute intervals is shown in the plot below.

![Graph showing relative humidity data](image)

**Figure D.2: Week 1 Raw Relative Humidity Data**

An interesting observation of the raw data collected involves data from compressor intake 2 not being present past May 22, 2012. It is important to note that during the first data collection, it was discovered that the zip tie fastening the data logger to the framing member of the compressor intake failed. When the zip tie failed, the logger fell from the compressor intake where it was exposed to the weather. Water was then able to get inside of the data logger and caused it to finish logging. Since no extra data loggers were taken up to the roof during this data collection, this logger was not able to be replaced until the next data
collection was scheduled a few weeks later. Also, it is important to note that the relative humidity points collected during this time period for each of the compressor intakes closely followed each other. This revelation suggests that future data sets where the relative humidity data did not follow as closely was more than likely the result of a defective relative humidity sensor on the data logger.

The next data comparison performed with the data set collected for week 1 involved an instantaneous comparison of values with those obtained from the National Weather Service. The results of this temperature analysis for week 1 are shown in the graph below.

![Figure D.3: Week 1 Hourly Instantaneous Temperature Data](image-url)
The results of the hourly instantaneous relative humidity analysis are shown in the figure below.

![Figure D.4: Week 1 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.4: Week 1 Hourly Instantaneous Relative Humidity Data**

### D.2. Week 2 Temperature and Relative Humidity Data

The time period chosen for week 2 began on Thursday, May 24, 2012 and ended Thursday, May 31, 2012. Data from compressor intake 2 was absent from this data set because the data logger had not yet been replaced. The raw temperature data taken at five minute intervals taken during this time period are shown in the figure below.
Relative humidity data taken at five minute intervals during week 2 are shown in the plot below.

Figure D.5: Week 2 Raw Temperature Data

Figure D.6: Week 2 Raw Relative Humidity Data
Hourly instantaneous temperature data collected for this week compared to data from the National Weather Service is shown in the figure below.

![Figure D.7: Week 2 Hourly Instantaneous Temperature Data](image)

The hourly instantaneous relative humidity data collected for week 2 is shown in the plot below.

![Figure D.8: Week 2 Hourly Instantaneous Relative Humidity Data](image)
D.3. Week 3 Temperature and Relative Humidity Data

The time period chosen for week 3 data began on Thursday, May 31, 2012 and ended on Thursday, June 7, 2012. The data from compressor intake 2 was still absent from this data set because the logger had not yet been replaced. Temperature data taken at five minute increments for this week are shown in the graph below.

![Figure D.9: Week 3 Raw Temperature Data](image)

Relative humidity data collected at five minute increments for week 3 are shown in the figure below.
Figure D.10: Week 3 Raw Relative Humidity Data

Hourly instantaneous temperature data for week 3 is shown in the plot below.

Figure D.11: Week 3 Hourly Instantaneous Temperature Data
Hourly instantaneous relative humidity data collected during week 3 is shown in the graph below.

Figure D.12: Week 3 Hourly Instantaneous Relative Humidity Data

D.4. Week 4 Temperature and Relative Humidity Data

The time period designated as week 4 for this analysis began on Thursday, June 7, 2012 and ended on Thursday, June 14, 2012. The Compressor Intake 2 data logger was replaced in time to record data for the duration of this week. The five minute temperature data collected for this time period is shown in the figure below.
Relative humidity data collected at five minute intervals during week 4 is shown in the plot below.
Hourly instantaneous temperature data collected during week 4 is shown in the graph below.

![Graph showing temperature data for week 4](image)

**Figure D.15: Week 4 Hourly Instantaneous Temperature Data**

Week 4 hourly instantaneous relative humidity data is shown in the figure below.

![Graph showing relative humidity data for week 4](image)

**Figure D.16: Week 4 Hourly Instantaneous Relative Humidity Data**
D.5. Week 5 Temperature and Relative Humidity Data

The time period chosen for week 5 data began on Friday, June 15, 2012 and ended on Friday, June 22, 2012. The temperature data taken at five minute intervals for week 5 is shown in the figure below.

![Figure D.17: Week 5 Raw Temperature Data](image)

Raw relative humidity data collected for week 5 is shown in the plot below.
Week 5 hourly instantaneous temperature data is shown in the figure below.

Figure D.19: Week 5 Hourly Instantaneous Temperature Data
Week 5 hourly instantaneous relative humidity data is shown in the plot below.

![Week 5 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.20: Week 5 Hourly Instantaneous Relative Humidity Data**

D.6. **Week 6 Temperature and Relative Humidity Data**

The time period designated for week 6 data began on Friday, June 22, 2012 and ended on Friday, June 29, 2012. Raw temperature data collected for this week from the data loggers is shown in the figure below.
Raw relative humidity data collected during week 6 is shown in the graph below.
Hourly instantaneous temperature data collected during week 6 is shown in the figure below.
Hourly instantaneous relative humidity data collected during week 6 is shown in the figure below.

![Week 6 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.24**: Week 6 Hourly Instantaneous Relative Humidity Data

### D.7. Week 7 Temperature and Relative Humidity Data

The time period chosen for week 7 data began on Saturday, June 30, 2012 and ended on Friday, July 6, 2012. The raw temperature data collected during this week is shown in the figure below.
Relative humidity data collected at five minute intervals during week 7 is shown in the figure below.
Hourly instantaneous temperature data collected during week 7 is shown in the plot below.

Figure D.27: Week 7 Hourly Instantaneous Temperature Data

Hourly instantaneous relative humidity data collected during week 7 is shown in the graph below.

Figure D.28: Week 7 Hourly Instantaneous Relative Humidity Data
D.8. Week 8 Temperature and Relative Humidity Data

The time period chosen for week 8 began on Saturday, July 7, 2012 and ended on Friday, July 13, 2012. The temperature data collected during this time period is shown in the figure below.

![Figure D.29: Week 8 Raw Temperature Data](image)

The relative humidity data collected during week 8 is shown in the graph below.
Hourly instantaneous temperature data collected during week 8 is shown in the figure below.
Hourly instantaneous relative humidity data for week 8 is shown in the figure below.

Figure D.32: Week 8 Hourly Instantaneous Relative Humidity Data

D.9. Week 9 Temperature and Relative Humidity Data

The time period chosen for week 9 data began on Friday, July 13, 2012 and ended Friday, July 20, 2012. The raw temperature data collected for this time period is shown in the figure below. It is important to note with the outside temperature that the logger was found to be 6 hours out of time phase compared to the other loggers. The data presented in the following plot reflects this change. It is believed that the reason for the in discrepancy between the
compressor intake data and the outdoor temperature data occurred because some water got into the outdoor temperature logger.

![Figure D.33: Week 9 Raw Temperature Data](image)

The relative humidity data collected during week 9 is shown in the graph below.

![Figure D.34: Week 9 Raw Relative Humidity Data](image)
The hourly instantaneous temperature data for week 9 is shown in the figure below.

![Figure D.35: Week 9 Hourly Instantaneous Temperature Data](image)

The hourly instantaneous relative humidity data for week 9 is shown in the figure below.

![Figure D.36: Week 9 Hourly Instantaneous Relative Humidity Data](image)
D.10. Week 10 Temperature and Relative Humidity Data

The time period chosen for week 10 data analysis began on Saturday, July 21, 2012 and ended on Thursday, July 26, 2012. The temperature data collected at the compressor intakes and outside of the compressor house for week 10 is shown in the figure below. This data is also part of the data set of outside temperature that was found to be out of phase by six hours. The graph below reflects the changes.

![Figure D.37: Week 10 Raw Temperature Data](image)

Relative humidity data collected during week 10 is shown in the plot below.
Hourly instantaneous temperature data for week 10 is shown in the figure below.

Figure D.38: Week 10 Raw Relative Humidity Data

Figure D.39: Week 10 Hourly Instantaneous Temperature Data
Hourly instantaneous relative humidity data for week 10 is shown in the plot below.

![Figure D.40: Week 10 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.40: Week 10 Hourly Instantaneous Relative Humidity Data**

**D.11. Week 11 Temperature and Relative Humidity Data**

The time period chosen as week 11 for data analysis began on Thursday, July 26, 2012 and ended on Thursday, August 2, 2012. The temperature data collected during this time period is shown in the figure below. This week was also included in the data collection where the outside temperature logger was out of phase by six hours. The data in the plot below reflects these changes.
Figure D.41: Week 11 Raw Temperature Data

Raw relative humidity data for week 11 is shown in the figure below.

Figure D.42: Week 11 Raw Relative Humidity Data
Hourly instantaneous temperature data for week 11 is shown in the plot below.

![Hourly Instantaneous Temperature Data](image1)

**Figure D.43: Week 11 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity data for week 11 is shown in the graph below.

![Hourly Instantaneous Relative Humidity Data](image2)

**Figure D.44: Week 11 Hourly Instantaneous Relative Humidity Data**
D.12. Week 12 Temperature and Relative Humidity Data

The time period chosen for week 12 temperature analysis began on Thursday, August 2, 2012 and ended on Wednesday, August 8, 2012. Data in this set has also been changed to reflect the time phase issue with the outdoor temperature logger. The temperature data collected for week 12 is shown in the figure below.

![Temperature Graph](image)

Figure D.45: Week 12 Raw Temperature Data

The relative humidity data collected by the data loggers for week 12 is shown in the figure below.
Hourly instantaneous temperature data for week 12 is shown in the figure below.

---

**Figure D.46: Week 12 Raw Relative Humidity Data**

**Figure D.47: Week 12 Hourly Instantaneous Temperature Data**
Hourly instantaneous relative humidity data for week 12 is shown in the figure below.

![Figure D.48: Week 12 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.48: Week 12 Hourly Instantaneous Relative Humidity Data**

D.13. **Week 13 Temperature and Relative Humidity Data**

The time period chosen for week 13 data analysis began on Thursday, August 9, 2012 and ended on Friday, August 17, 2012. Data in this set has also been changed to reflect the time phase issue with the outdoor temperature logger.
Week 13 raw relative humidity data is shown in the plot below.
Week 13 hourly instantaneous temperature data is shown in the graph below.

![Temperature Data](image1.png)

**Figure D.51: Week 13 Hourly Instantaneous Temperature Data**

Week 13 hourly instantaneous relative humidity data is shown in the figure below.

![Relative Humidity Data](image2.png)

**Figure D.52: Week 13 Hourly Instantaneous Relative Humidity Data**
D.14. Week 14 Temperature and Relative Humidity Data

The time period designated as week 14 began on Friday, August 17, 2012 and ended Thursday, August 23, 2012. The temperature data collected at five minute increments on the data loggers is shown in the figure below.

![Week 14 Raw Temperature Data](image)

*Figure D.53: Week 14 Raw Temperature Data*

Week 14 raw relative humidity data is shown in the plot below.
Week 14 hourly instantaneous temperature data is shown in the figure below.
The hourly instantaneous relative humidity data for week 14 is shown in the figure below.

Figure D.56: Week 14 Hourly Instantaneous Relative Humidity Data

D.15. Week 15 Temperature and Relative Humidity Data

The time period chosen for week 15 started on Friday, August 24, 2012 and ended on Friday, August 31, 2012. The temperature data collected for this week is shown in the graph below.
Week 15 raw relative humidity data is shown in the graph below.

Figure D.57: Week 15 Raw Temperature Data

Figure D.58: Week 15 Raw Relative Humidity Data
The figure below shows hourly instantaneous temperature data for week 15.

![Temperature Data Chart](image)

**Figure D.59: Week 15 Hourly Instantaneous Temperature Data**

The figure below shows hourly instantaneous relative humidity data for week 15.

![Relative Humidity Data Chart](image)

**Figure D.60: Week 15 Hourly Instantaneous Relative Humidity Data**
D.16. Week 16 Temperature and Relative Humidity Data

The time period chosen for week 16 began on Friday, August 31, 2012 and ended on Friday, September 7, 2012. The temperature data collected during this time period is shown in the figure below.

![Temperature Data Graph](image)

**Figure D.61: Week 16 Raw Temperature Data**

Raw relative humidity data collected during week 16 is shown in the graph below.
Hourly instantaneous temperature data collected for week 16 is shown in the figure below.
Week 16 hourly instantaneous relative humidity data is shown in the plot below.

![Week 16 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.64: Week 16 Hourly Instantaneous Relative Humidity Data**

**D.17. Week 17 Temperature and Relative Humidity Data**

The time period designated as week 17 in the data analysis began on Saturday, September 8, 2012 and ended on Friday, September 14, 2012. The temperature data collected for this analysis is shown in the figure below.
Raw relative humidity data for week 17 is shown in the plot below.
Hourly instantaneous temperature data for week 17 is shown in the figure below.

![Temperature Data](image)

**Figure D.67: Week 17 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity data for week 17 is shown in the figure below.

![Relative Humidity Data](image)

**Figure D.68: Week 17 Hourly Instantaneous Relative Humidity Data**
D.18. Week 18 Temperature and Relative Humidity Data

The time period chosen for week 18 began on Saturday, September 15, 2012 and ended on Friday, September 21, 2012. The temperature data collected during this time period is shown in the figure below.

Figure D.69: Week 18 Raw Temperature Data

The relative humidity data logged during week 18 is shown in the figure below.
Hourly instantaneous temperature data for week 18 is shown in the graph below.
Hourly instantaneous relative humidity data is shown in the figure below.

Figure D.72: Week 18 Hourly Instantaneous Relative Humidity Data

D.19. Week 19 Temperature and Relative Humidity Data

The time period designated as week 19 for data analysis began on Friday, September 21, 2012 and ended on Thursday, September 27, 2012. The temperature data collected for this week is shown in the figure below.
Week 19 relative humidity data is shown in the figure below.
Week 19 hourly instantaneous temperature data is shown in the figure below.

![Temperature Data](image)

**Figure D.75: Week 19 Hourly Instantaneous Temperature Data**

Week 19 hourly instantaneous relative humidity data is shown in the figure below.

![Relative Humidity Data](image)

**Figure D.76: Week 19 Hourly Instantaneous Relative Humidity Data**
**D.20. Week 20 Temperature and relative Humidity Data**

The time period chosen for week 20 began on Friday, September 28, 2012 and ended on Thursday, October 4, 2012. The temperature data collected during this week is shown in the plot below.

![Week 20 Raw Temperature Data](image)

**Figure D.77: Week 20 Raw Temperature Data**

The relative humidity data collected during week 20 is shown in the figure below.
Figure D.78: Week 20 Raw Relative Humidity Data

The hourly instantaneous temperature data for week 20 is shown in the figure below.

Figure D.79: Week 20 Hourly Instantaneous Temperature Data
The hourly instantaneous relative humidity data for week 20 is shown in the plot below.

Figure D.80: Week 20 Hourly Instantaneous Relative Humidity Data

D.21. Week 21 Temperature and Relative Humidity Data

Week 21 consists of the time period beginning Friday, October 5, 2012 and ending Thursday October 11, 2012. The temperature data collected during this week is shown in the figure below.
The relative humidity data logged during week 21 is shown in the graph below.
The hourly instantaneous temperature data for week 21 is shown in the figure below.

![Figure D.83: Week 21 Hourly Instantaneous Temperature Data](image1)

The hourly instantaneous relative humidity data for week 21 is shown in the figure below.

![Figure D.84: Week 21 Hourly Instantaneous Relative Humidity Data](image2)
D.22. Week 22 Temperature and Relative Humidity Data

The time period selected for week 22 began on Friday, October 12, 2012 and ended on Thursday, October 18, 2012. The temperature data collected during this time period is shown in the figure below.

![Week 22 Raw Temperature Data](image)

**Figure D.85: Week 22 Raw Temperature Data**

Relative humidity data collected during week 22 is shown in the figure below.
Hourly instantaneous temperature data for week 22 is shown in the figure below.
Hourly instantaneous relative humidity data for week 22 is shown in the figure below.

![Figure D.88: Week 22 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.88: Week 22 Hourly Instantaneous Relative Humidity Data**

### D.23. Week 23 Temperature and Relative Humidity Data

The time period chosen for week 23 analysis began on Friday, October 19, 2012 and ended on Thursday October 25, 2012. The temperature data collected for week 23 is shown in the figure below.
The relative humidity data logged for week 23 is shown in the plot below.

![Week 23 Raw Temperature Data](image1)

**Figure D.89: Week 23 Raw Temperature Data**

![Week 23 Raw Relative Humidity Data](image2)

**Figure D.90: Week 23 Raw Relative Humidity Data**
The hourly instantaneous temperature data for week 23 is shown in the figure below.

![Figure D.91: Week 23 Hourly Instantaneous Temperature Data](image1)

The hourly instantaneous relative humidity data for week 23 is shown in the figure below.

![Figure D.92: Week 23 Hourly Instantaneous Relative Humidity Data](image2)
D.24. Week 24 Temperature and Relative Humidity Data

The time period chosen for week 24 data analysis began on Friday, October 26, 2012 and ended on Thursday, November 1, 2012. The temperature data logged during this week is shown in the graph below.

![Week 24 Raw Temperature Data](image)

**Figure D.93: Week 24 Raw Temperature Data**

The relative humidity data collected during week 24 is shown in the figure below.
The hourly instantaneous temperature data for week 24 is shown in the plot below.

**Figure D.95: Week 24 Hourly Instantaneous Temperature Data**
The hourly instantaneous relative humidity data for week 24 is shown in the plot below.

Figure D.96: Week 24 Hourly Instantaneous Relative Humidity Data

**D.25. Week 25 Temperature and Relative Humidity Data**

The days selected for week 25 data analysis began on Friday, November 2, 2012 and ended on Thursday, November 8, 2012. The temperature data collected during this week is shown in the following figure.
Relative humidity data for week 25 is shown in the graph below.
Week 25 hourly instantaneous temperature data is shown in the figure below.

![Week 25 Hourly Instantaneous Temperature Data](image)

**Figure D.99: Week 25 Hourly Instantaneous Temperature Data**

Week 25 hourly instantaneous relative humidity data is shown in the figure below.

![Week 25 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.100: Week 25 Hourly Instantaneous Relative Humidity Data**
D.26. Week 26 Temperature and Relative Humidity Data

The time frame chosen for week 26 data analysis began on Friday, November 9, 2012 and ended Thursday, November 15, 2012. The temperature data logged during week 26 is shown in the plot below.

![Temperature Data](image)

**Figure D.101: Week 26 Raw Temperature Data**

The relative humidity data for week 26 is shown in the figure below.
Hourly instantaneous temperature data for week 26 is shown in the following figure.
Hourly instantaneous relative humidity data for week 26 is shown in the following figure.

![Week 26 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.104: Week 26 Hourly Instantaneous Relative Humidity Data**

### D.27. Week 27 Temperature and Relative Humidity Data

The time period chosen for week 27 temperature and relative humidity analysis began on Friday, November 16, 2012 and ended on Thursday, November 22, 2012. The temperature data collected for this week is shown in the plot below.
The relative humidity data for week 27 is shown in the following figure.
Week 27 hourly instantaneous temperature data is shown in the figure below.

*Figure D.107: Week 27 Hourly Instantaneous Temperature Data*

Week 27 hourly instantaneous relative humidity data is shown in the figure below.

*Figure D.108: Week 27 Hourly Instantaneous Relative Humidity Data*
D.28. Week 28 Temperature and Relative Humidity Data

The time period designated as week 28 for temperature and relative humidity data analysis began on Friday, November 23, 2012 and ended on Thursday, November 29, 2012. The temperature data collected during this time period is shown in the figure below.

![Figure D.109: Week 28 Raw Temperature Data](image_url)

The relative humidity data collected for week 27 is shown in the following figure.
Hourly instantaneous temperature data for week 28 is shown in the figure below.
Hourly instantaneous relative humidity data is shown in the figure below.

![Figure D.112: Week 28 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.112: Week 28 Hourly Instantaneous Relative Humidity Data**

**D.29. Week 29 Temperature and Relative Humidity Data**

The period of time chosen as week 29 for data analysis began on Friday, November 30, 2012 and ended on Thursday, December 6, 2012. The temperature data collected during this time period is shown in the figure below.
Relative humidity data collected during week 29 is shown in the figure below.
Hourly instantaneous temperature data gathered during week 29 is shown in the figure below.

Figure D.115: Week 29 Hourly Instantaneous Temperature Data

Hourly instantaneous relative humidity data for week 29 is shown in the figure below.

Figure D.116: Week 29 Hourly Instantaneous Relative Humidity Data
D.30. Week 30 Temperature and Relative Humidity Data

The seven day period chosen as week 30 for data analysis began on Friday, December 7, 2012 and ended on Thursday, December 8, 2012. The temperature data collected during this week is shown in the figure below.

![Figure D.117: Week 30 Raw Temperature Data](image)

The relative humidity data collected during this time period is shown in the figure below.
Figure D.118: Week 30 Raw Relative Humidity Data

Hourly instantaneous temperature data for week 30 is shown in the figure below.

Figure D.119: Week 30 Hourly Instantaneous Temperature Data
Hourly instantaneous relative humidity data collected for week 30 is shown in the figure below.

![Figure D.120: Week 30 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.120: Week 30 Hourly Instantaneous Relative Humidity Data**

**D.31. Week 31 Temperature and Relative Humidity Data**

The time period designated as week 31 for data analysis began on Friday, December 14, 2012 and ended on Thursday, December 20, 2012. The temperature data collected for this week is shown in the table below.
Relative humidity data collected during week 31 is displayed in the figure below.
Hourly instantaneous temperature data for week 31 is shown in the figure below.

Figure D.123: Week 31 Hourly Instantaneous Temperature Data

Hourly instantaneous relative humidity data for week 31 is displayed in the figure below.

Figure D.124: Week 31 Hourly Instantaneous Relative Humidity Data
D.32. Week 32 Temperature and Relative Humidity Data

The time period designate as week 32 for data analysis began on Friday, December 21, 2012 and ended on Thursday, December 27, 2012. The temperature data collected during this week is displayed in the figure below.

![Figure D.125: Week 32 Raw Temperature Data](image)

Relative humidity data collected during week 32 is shown in the figure below.
Hourly instantaneous temperature data for week 32 is shown in the figure below.
Hourly instantaneous relative humidity data for week 32 is shown in the figure below.

Figure D.128: Week 32 Hourly Instantaneous Relative Humidity Data

D.33. Week 33 Temperature and Relative Humidity Data

The time period designated as week 33 for data analysis began on Friday, December 28, 2012 and ended on Thursday, January 3, 2013. The temperature data collected during this time period is displayed in the following figure.
Relative humidity logged during week 33 is shown in the following figure.
Hourly instantaneous temperature data for week 33 is shown in the figure below.

Figure D.131: Week 33 Hourly Instantaneous Temperature Data

Hourly instantaneous relative humidity data for week 33 is shown in the figure below.

Figure D.132: Week 33 Hourly Instantaneous Relative Humidity Data
D.34.  **Week 34 Temperature and Relative Humidity Data**

The time period chosen as week 34 for data analysis began on Friday, January 4, 2013 and ended on Thursday, January 10, 2013. The temperature data collected during this week is shown in the following figure.

![Temperature Data](image)

**Figure D.133: Week 34 Raw Temperature Data**

Relative humidity data collected during week 34 is shown in the figure below.
Hourly instantaneous temperature data collected for week 34 is shown in the figure below.
Hourly instantaneous relative humidity data for week 34 is shown in the following figure.

![Figure D.136: Week 34 Hourly Instantaneous Relative Humidity Data](image)

**Figure D.136: Week 34 Hourly Instantaneous Relative Humidity Data**

**D.35. Week 35 Temperature and Relative Humidity Data**

Week 35 for temperature and relative humidity data collection begins the data sets collected from National Weather Service Data due to time constraints on this experiment. Since the data values collected for the previous four weeks mirrored the hourly values taken at the local airport to the subject facility, these values were considered to be appropriate to provide the remainder of an annual analysis of the effects of inlet air temperature on industrial reciprocating and centrifugal compressors. The time period designated for week 35 data began on Wednesday, January 11, 2012 and ended on Tuesday, January 17, 2012. The hourly
instantaneous temperature data collected from the National Weather Service for this week is illustrated in the plot below.

![Figure D.137: Week 35 Hourly Instantaneous Temperature Data](image1)

Hourly instantaneous relative humidity data for week 35 is shown in the figure below.

![Figure D.138: Week 35 Hourly Instantaneous Relative Humidity Data](image2)
D.36. Week 36 Temperature and Relative Humidity Data

The week of National Weather Service Data designated as week 36 began on Wednesday, January 18, 2012 and ended on Tuesday, January 24, 2012. The hourly instantaneous temperature data collected for this time period is displayed in the graph below.

![Temperature Graph](image)

**Figure D.139: Week 36 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity for week 36 is shown in the figure below.
The time period designated as week 37 for temperature and relative humidity analysis began on Wednesday, January 25, 2012 and ended on Tuesday, January 31, 2012. The results of this data collection for hourly instantaneous temperatures are shown in the figure below.
Figure D.141: Week 37 Hourly Instantaneous Temperature Data

The relative humidity data collected for this time period is shown in the plot below.

Figure D.142: Week 37 Hourly Instantaneous Relative Humidity Data
D.38. Week 38 Temperature and Relative Humidity Data

The time period designated as week 38 for temperature and relative humidity data analysis began on Wednesday, February 1, 2012 and ended on Tuesday, February 7, 2012. The temperature data collected from the National Weather Service for this week is illustrated in the figure below.

![Week 38 Hourly Instantaneous Temperature Data](image)

**Figure D.143: Week 38 Hourly Instantaneous Temperature Data**

The hourly instantaneous relative humidity data collected for this week is displayed in the graph below.
Figure D.144: Week 38 Hourly Instantaneous Relative Humidity Data

D.39. Week 39 Temperature and Relative Humidity Data

The week chosen as week 39 for data analysis began on Wednesday, February 8, 2012 and ended on Tuesday, February 14, 2012. The hourly instantaneous temperature data collected from this time period is shown in the figure below.
Hourly instantaneous relative humidity data for week 39 is shown in the figure below.
D.40. **Week 40 Temperature and Relative Humidity Data**

The time period designated as week 40 for data analysis began on Wednesday, February 15, 2012 and ended on Tuesday, February 21, 2012. The hourly instantaneous temperature data collected for this time period is shown in the figure below.

![Temperature Data Graph](image)

**Figure D.147: Week 40 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity for week 40 is displayed in the plot below.
D.41. Week 41 Temperature and Relative Humidity Data

The days chosen for week 41 data analysis began on Wednesday, February 22, 2012 and ended on Tuesday, February 28, 2012. The hourly instantaneous temperature data collected for this week is shown in the figure below.
Figure D.149: Week 41 Hourly Instantaneous Temperature Data

The hourly instantaneous relative humidity data for week 41 is shown in the figure below.

Figure D.150: Week 41 Hourly Instantaneous Relative Humidity Data
D.42. Week 42 Temperature and Relative Humidity Data

The time period designated as week 42 for data analysis began Thursday, March 1, 2012 and ended on Wednesday, March 7, 2012. Though 2012 was a leap year, data for February 29, 2012 is not considered in this analysis to better represent a more typical year. The hourly instantaneous temperature data collected for this week is shown in the figure below.

![Figure D.151: Week 42 Hourly Instantaneous Temperature Data](image)

Hourly instantaneous relative humidity data for week 42 is shown in the figure below.
The time period designated as week 43 for data analysis began on Thursday, March 8, 2012 and ended on Wednesday, March 14, 2012. The hourly instantaneous temperature data collected for this week from the National Weather Service is displayed in the figure below.
Hourly instantaneous relative humidity data for week 43 is shown in the plot below.

Figure D.153: Week 43 Hourly Instantaneous Temperature Data

Figure D.154: Week 43 Hourly Instantaneous Relative Humidity Data
Week 44 Temperature and Relative Humidity Data

The days chosen as week 44 for data analysis began on Thursday, March 15, 2012 and ended on Wednesday, March 21, 2012. The hourly instantaneous temperature data for week 44 is shown in the graph below.

**Figure D.155: Week 44 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity data for week 44 is shown in the table below.
Figure D.156: Week 44 Hourly Instantaneous Relative Humidity Data

D.45. Week 45 Temperature and Relative Humidity Data

The time period designated as week 45 for temperature and relative humidity data analysis began on Thursday, March 22, 2012 and ended on Wednesday March 28, 2012. Hourly instantaneous temperature data collected for week 45 is shown in the figure below.
Hourly instantaneous relative humidity data for week 45 is displayed in the figure below.
**D.46. Week 46 Temperature and Relative Humidity Data**

The days chosen as week 46 for data analysis began on Thursday, March 29, 2012 and ended on Wednesday, April 4, 2012. Hourly instantaneous temperature data for this time period is shown in the figure below.

![Temperature Data](image)

**Figure D.159: Week 46 Hourly Instantaneous Temperature Data**

Hourly instantaneous relative humidity data for week 46 is depicted in the plot below.

![Humidity Data](image)
Figure D.160: Week 46 Hourly Instantaneous Relative Humidity Data

D.47. Week 47 Temperature and Relative Humidity Data

Week 47 consists of the time period beginning Thursday, April 5, 2012 and ending Wednesday, April 11, 2012. Hourly instantaneous temperature data gathered for week 47 is illustrated in the graph below.
Hourly instantaneous relative humidity data for week 47 is shown in the figure below.
D.48.  Week 48 Temperature and Relative Humidity Data

The time period designated as week 48 for data analysis began on Thursday, April 12, 2012 and ended on Wednesday, April 18, 2012. Hourly instantaneous temperature data for this week is shown in the plot below.

![Figure D.163: Week 48 Hourly Instantaneous Temperature Data](image)

Hourly instantaneous relative humidity data for week 48 is shown in the figure below.
D.49. Week 49 Temperature and Relative Humidity Data

The time period designated as week 49 for data collection began on Thursday, April 19, 2012 and ended on Wednesday, April 25, 2012. The hourly instantaneous temperature data for week 49 is shown in the figure below.
Hourly instantaneous relative humidity data for week 49 is shown in the figure below.
D.50. Week 50 Temperature and Relative Humidity Data

The time period designated as week 50 for data analysis began on Thursday, April 26, 2012 and ended on Wednesday, May 2, 2012. Hourly instantaneous temperature data for week 50 is shown in the figure below.

![Figure D.167: Week 50 Hourly Instantaneous Temperature Data](image)

Hourly instantaneous relative humidity data for week 50 is shown in the figure below.
Figure D.168: Week 50 Hourly Instantaneous Relative Humidity Data

D.51. Week 51 Temperature and Relative Humidity Data

The days chosen for week 51 data analysis began on Thursday, May 3, 2012 and ended on Wednesday, May 9, 2012. Hourly instantaneous temperature data for week 51 is shown in the plot below.
Figure D.169: Week 51 Hourly Instantaneous Temperature Data

Hourly instantaneous relative humidity data for week 51 is displayed in the plot below.

Figure D.170: Week 51 Hourly Instantaneous Relative Humidity Data
D.52. Week 52 Temperature and Relative Humidity Data

The time period designated as week 52 for data analysis began on Thursday, May 10, 2012 and ended on Thursday, May 17, 2012. Hourly instantaneous temperature data for week 52 is displayed in the figure below.

![Figure D.171: Week 52 Hourly Instantaneous Temperature Data](image)

Hourly instantaneous relative humidity data for week 52 is shown in the figure below.
Figure D.172: Week 52 Hourly Instantaneous Relative Humidity Data
Appendix E   Compressor Energy Usage and Flow Data

The following sections represent compressor energy usage and flow data provided by the subject facility. The weeks chosen for periods of data analysis correspond to the weeks chosen for temperature and relative humidity data analysis.

E.1.   Week 1 Compressor Energy Usage and Flow Data

Overall compressor energy usage and total system compressed air flow for week 1 is shown in the figure below.

Figure E.1: Week 1 Compressor Energy Usage and Compressed Air Flow
E.2. Week 2 Compressor Energy Usage and Flow Data

Energy usage for all compressors and total system air flow for week 2 are shown in the following figure.

![Figure E.2: Week 2 Compressor Energy Usage and Compressed Air Flow](image_url)
E.3. Week 3 Compressor Energy Usage and Flow Data

Total energy usage for compressors at the facility and total compressed air flow for week 2 is shown in the figure below.

![Figure E.3: Week 3 Compressor Energy Usage and Compressed Air Flow](image-url)
E.4. Week 4 Compressor Energy Usage and Flow Data

Total compressor energy usage and total daily air flows for the compressed air system used at the facility are shown in the figure below.

Figure E.4: Week 4 Compressor Energy Usage and Compressed Air Flow
E.5. Week 5 Compressor Energy Usage and Flow Data

Energy usage of all compressors and compressed air flows for the overall compressed air system for week 5 is shown in the figure below. An important item to mention with this data set involves the fact that flow data was not available for June 18, 2012. This missing data may be the result of the data being collected on that day, which would result in daily totals not being present.

Figure E.5: Week 5 Compressor Energy Usage and Compressed Air Flow
E.6. Week 6 Compressor Energy Usage and Flow Data

Total compressed air energy usage and total daily flows for week 6 are shown in the figure below. Just as with week 6, one of the daily total flows is missing. This data may be missing from a data collection day, or may be the result of bad data.

Figure E.6: Week 6 Compressor Energy Usage and Compressed Air Flow
E.7. Week 7 Compressor Energy Usage and Flow Data

Overall compressed air energy consumption and total daily air flows for week 7 are shown in the following figure.

![Figure E.7: Week 7 Compressor Energy Usage and Compressed Air Flow](image-url)
E.8. Week 8 Compressor Energy Usage and Flow Data

Overall compressor energy usage and total daily system flows for week 8 are shown in the figure below.

Figure E.8: Week 8 Compressor Energy Usage and Compressed Air Flow
E.9. **Week 9 Compressor Energy Usage and Flow Data**

Week 9 overall compressed air system energy usage and total daily air flow is shown in the figure below.

![Figure E.9: Week 9 Compressor Energy Usage and Compressed Air Flow](image)
E.10. Week 10 Compressor Energy Usage and Flow Data

Compresses air energy usage and total daily compressed air flows for the facility for week 10 are shown in the figure below.

![Graph showing energy usage and compressed air flow over a week](image)

**Figure E.10: Week 10 Compressor Energy Usage and Compressed Air Flow**
E.11. Week 11 Compressor Energy Usage and Flow Data

Week 11 compressed air energy usage and total daily air flows are shown in the following figure.

![Figure E.11: Week 11 Compressor Energy Usage and Compressed Air Flow](image)

Figure E.11: Week 11 Compressor Energy Usage and Compressed Air Flow
E.12. Week 12 Compressor Energy Usage and Flow Data

Overall compressed air energy usage and total daily air flows for week 12 are shown in the following figure.

Figure E.12: Week 12 Compressor Energy Usage and Compressed Air Flow
E.13.  

Week 13 Compressor Energy Usage and Flow Data

Week 13 overall compressed air energy usage and total daily airflow are shown in the figure below.

Figure E.13: Week 13 Compressor Energy Usage and Compressed Air Flow
E.14. Week 14 Compressor Energy Usage and Flow Data

Total compressed air energy usage and daily air flows at the facility for week 14 are shown in the figure below.

Figure E.14: Week 14 Compressor Energy Usage and Compressed Air Flow
E.15. Week 15 Compressor Energy Usage and Flow Data

Compressor energy usage and daily system air flows for week 15 are shown in the following figure.

Figure E.15: Week 15 Compressor Energy Usage and Compressed Air Flow
**E.16. Week 16 Compressor Energy Usage and Flow Data**

Week 16 compressed air energy usage and total daily air flows collected are shown in the figure below.

![Energy Usage and Compressed Air Flow](image.png)

**Figure E.16: Week 16 Compressor Energy Usage and Compressed Air Flow**
E.17. Week 17 Compressor Energy Usage and Flow Data

Facility compressed air energy usage and daily air flow for week 17 are shown in the figure below.

Figure E.17: Week 17 Compressor Energy Usage and Compressed Air Flow
E.18. Week 18 Compressor Energy Usage and Flow Data

Overall compressor energy usage and total daily air flows for week 18 are shown in the figure below.

![Figure E.18: Week 18 Compressor Energy Usage and Compressed Air Flow](image-url)
E.19.  Week 19 Compressor Energy Usage and Flow Data

Compressed air energy consumption and daily air flows for week 19 are shown in the figure below.

![Week 19 Compressor Energy Usage and Compressed Air Flow](image)

**Figure E.19: Week 19 Compressor Energy Usage and Compressed Air Flow**
E.20. Week 20 Compressor Energy Usage and Flow Data

Week 20 compressed air energy consumption and compressed air flows are shown in the figure below.

![Figure E.20: Week 20 Compressor Energy Usage and Compressed Air Flow](image)
E.21. Week 21 Compressor Energy Usage and Flow Data

Compressed air energy usage and total daily system air flows for week 21 are shown in the figure below.

Figure E.21: Week 21 Compressor Energy Usage and Compressed Air Flow
E.22. Week 22 Compressor Energy Usage and Flow Data

Week 22 compressed air energy usage and total daily flows for figure 22 are shown in the figure below.

Figure E.22: Week 22 Compressor Energy Usage and Compressed Air Flow
E.23. Week 23 Compressor Energy Usage and Flow Data

Compressed air energy usage and total daily flows for week 23 are shown in the following figure.

Figure E.23: Week 23 Compressor Energy Usage and Compressed Air Flow
E.24. Week 24 Compressor Energy Usage and Flow Data

Week 24 overall compressor energy consumption and total daily flows are shown in the figure below.

Figure E.24: Week 24 Compressor Energy Usage and Compressed Air Flow
E.25. Week 25 Compressor Energy Usage and Flow Data

Compressed air energy usage and total daily flows for week 25 are shown in the figure below.

![Figure E.25: Week 25 Compressor Energy Usage and Compressed Air Flow](image-url)
E.26. **Week 26 Compressor Energy Usage and Flow Data**

The total energy used by all of the air compressors and total daily air flows for week 26 are shown in the figure below.

![Week 26 Compressor Energy Usage and Compressed Air Flow](image)

**Figure E.26: Week 26 Compressor Energy Usage and Compressed Air Flow**
E.27. Week 27 Compressor Energy Usage and Flow Data

Week 27 overall compressed air energy usage and total daily air flows for week 27 are shown in the figure below.

Figure E.27: Week 27 Compressor Energy Usage and Compressed Air Flow
E.28. **Week 28 Compressor Energy Usage and Flow Data**

Overall compressed air system energy usage and total daily air flows are displayed in the following figure.

**Figure E.28: Week 28 Compressor Energy Usage and Compressed Air Flow**
E.29. Week 29 Compressor Energy Usage and Flow Data

Week 29 compressed air power consumption and total daily air flows are shown in the figure below.

![Figure E.29: Week 29 Compressor Energy Usage and Compressed Air Flow](image-url)
E.30. Week 30 Compressor Energy Usage and Flow Data

Compressed air energy consumption and daily air flow produced for week 30 is shown in the following figure.

Figure E.30: Week 30 Compressor Energy Usage and Compressed Air Flow
E.31. **Week 31 Compressor Energy Usage and Flow Data**

Total energy usage for facility compressors and daily air flows for week 31 are shown in the following figure.

![Figure E.31: Week 31 Compressor Energy Usage and Compressed Air Flow](image-url)
E.32. Week 32 Compressor Energy Usage and Flow Data

Overall energy usage by the facility air compressors as well as daily air flows are shown in the figure below. It is important to note that energy readings early in week 32 are not consistent with data from any other week due to power consumption readings are were given at increments of more than one day rather than the typical 15 minutes, probably as a result of winter shutdown for the holidays.

Figure E.32: Week 32 Compressor Energy Usage and Compressed Air Flow
E.33. Week 33 Compressor Energy Usage and Flow Data

Energy consumed by facility air compressors and daily total flows produced during week 33 are shown in the figure below.

Figure E.33: Week 33 Compressor Energy Usage and Compressed Air Flow
E.34. Week 34 Compressor Energy Usage and Flow Data

Week 34 compressor energy usage and total daily air flows for week 34 are shown in the figure below.

Figure E.34: Week 34 Compressor Energy Usage and Compressed Air Flow
E.35.  Weeks 35 – 52 Compressor Energy Usage and Flow Data

Since the compressed air flow is expected to be relatively constant, all remaining weeks employed an average production (greater than 20,000 mscf) air flow of 22,991 mscf. Due to energy consumption by the compressors being directly affected by inlet air temperatures, energy consumption of the compressed air system was equated for each day’s temperature and relative humidity readings to a similar day in the experimental set with an emphasis placed on choosing closest temperatures due to the direct impact inlet temperature has on compressor power. The following tables summarize the similar days chosen for this analysis.

<table>
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<th>Date</th>
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### Table E.2: February Chosen Similar Days for Compressor Energy Consumption

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### Table E.3: March Chosen Similar Days for Compressor Energy Consumption

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### Table E.4: April Chosen Similar Days for Compressor Energy Consumption

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### Table E.5: May Chosen Similar Days for Compressor Energy Consumption

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<td></td>
<td></td>
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</tbody>
</table>
Appendix F    Progress Energy LGS-TOU Rate Schedule

The Progress Energy LGS-TOU rate schedule that electrical energy cost savings in this
eperiment are calculated from is shown in the following figures.

Figure F.1: Progress Energy LGS-TOU Page 1  (33)
III. kWh Energy Charge:

- 4.820¢ per on-peak kWh
- 4.320¢ per off-peak kWh

IV. Renewable Energy Portfolio Standard (REPS) Adjustment:

The monthly bill shall include a REPS Adjustment based upon the revenue classification:

- Commercial/Governmental Classification - $7.28/month
- Industrial/Public Authority Classification - $34.32/month

Upon written request, only one REPS Adjustment shall apply to each premise serving the same customer for all accounts of the same revenue classification. If a customer has accounts which serve in an auxiliary role to a main account on the same premise, no REPS charge should apply to the auxiliary accounts regardless of their revenue classification (see Annual Billing Adjustments Rider BA).

V. Transformation Discounts:

When Customer owns the step-down transformation and all other facilities beyond the transformation which Company would normally own, except Company's metering equipment, the charge per kW of on-peak Billing Demand and per kWh will be reduced in accordance with the following:

<table>
<thead>
<tr>
<th>Transmission Service Transformations</th>
<th>Distribution Service Transformation Discount</th>
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</thead>
<tbody>
<tr>
<td>Transmission Discount</td>
<td>Distribution Discount</td>
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<tr>
<td>$0.48/kW</td>
<td>$0.75/kW</td>
</tr>
<tr>
<td>$0.0008/kWh</td>
<td>$0.0001/kWh</td>
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</tbody>
</table>

Transmission: For Customer to qualify for the Transmission Service Transformation Discount, Customer must own the step-down transformation and all other facilities beyond the transformation which Company would normally own, except Company's metering equipment, necessary to take service at the voltage of the 69 kV, 115 kV, or 230 kV transmission line from which Customer received service.

Distribution: For Customer to qualify for the Distribution Service Transformation Discount, Customer must own the step-down transformation and all other facilities beyond the transformation which Company would normally own, except Company's metering equipment, necessary to take service from the distribution line of 12.47 kV or higher from which Customer receives service. The distribution service source must be from a general distribution line and must be from other than a transmission-to-distribution substation built primarily for Customer's use in order to qualify for the Distribution Service Transformation Discount. A general distribution line is a 12.47 kV or higher voltage distribution line built to serve the general area and not built primarily to serve a specific customer.

Company shall have the option to install high-side metering equipment or low-side metering equipment compensated for Customer-owned transformer and line losses.
Any facilities which Company provides above those which Company would normally have utilized to service Customer's Contract Demand shall be considered as Additional Facilities. Any Company-owned protection system installed when service is directly from Company's 69 kV, 115 kV, or 230 kV transmission system or a distribution line of 12.47 kV or higher shall be considered Additional Facilities.

If changing conditions on Company's electrical system make continuation of the current delivery voltage impractical, Customer shall be responsible for all costs for the conversion beyond the point of delivery except any Company-owned metering equipment. At the time of the conversion, Company reserves the right to provide service at one of its available voltages.

If subsequent changes in the use of Company's facilities occur which cause the reclassification of either transformers or lines, Customer's entitlement to the discount may be changed.

VI. Minimum Bill

The minimum monthly charge shall be the Basic Customer Charge plus the REPS Adjustment plus a charge for 1,000 kW at the off-peak excess demand rate.

BILLING DEMANDS

I. The on-peak Billing Demand shall be the maximum demand registered or computed by or from Company's metering facilities used in the on-peak hours of the current month during any 15-minute interval.

II. The off-peak excess Billing Demand is the maximum demand registered or computed by or from Company's metering facilities used during any 15-minute interval in the off-peak hours of the current month less the on-peak Billing Demand.

DETERMINATION OF ON-PEAK AND OFF-PEAK HOURS

I. On-Peak Hours:

A. Service used beginning at 12:00 midnight March 31 and ending at 12:00 midnight September 30:

The on-peak hours are defined as the hours between 10:00 a.m. and 10:00 p.m., Monday through Friday, excluding holidays considered as off-peak.

B. Service used beginning at 12:00 midnight September 30 and ending at 12:00 midnight March 31:

The on-peak hours are defined as those hours between 6:00 a.m. and 1:00 p.m., plus 4:00 p.m. through 9:00 p.m., Monday through Friday, excluding holidays considered as off-peak.

II. Off-Peak Hours:

The off-peak hours in any month are defined as all hours not specified above as on-peak hours. All hours for the following holidays will be considered off-peak: New Year's Day, Good Friday, Memorial Day, Independence Day, Labor Day, Thanksgiving Day and the day after, and Christmas Day. When one of the above holidays falls on a Saturday, the Friday before the holiday will be considered off-peak; when the holiday falls on a Sunday, the following Monday will be considered off-peak.
POWER FACTOR ADJUSTMENT

When the power factor in the current billing month is less than 85%, the monthly bill will be increased by a sum equal to $0.40 multiplied by the difference between the maximum reactive kilovolt-amperes (kVAr) registered by a demand meter suitable for measuring the demand used during a 15-minute interval and 62% of the maximum kW demand registered in the current billing month.

SALES TAX

To the above charges will be added any applicable North Carolina Sales Tax.

PAYMENTS

Bills are due when rendered and are payable within 15 days from the date of the bill. If any bill is not so paid, Company has the right to suspend service in accordance with its Service Regulations. In addition, any bill not paid on or before the expiration of twenty-five (25) days from the date of the bill is subject to an additional charge of 1% per month as provided in Rule R12-9 of the Rules and Regulations of the North Carolina Utilities Commission.

RIDER APPLICATIONS

When this Schedule is used in conjunction with any applicable rider, the charges, if any, as stated in the rider will be adjusted to reflect the on-peak and off-peak periods and on-peak and off-peak charges in this Schedule unless specific and different on-peak and off-peak periods and charges are stated in the rider.

CONTRACT PERIOD

The Contract Period shall not be less than one year.

GENERAL

Service rendered under this Schedule is subject to the provisions of the Service Regulations and any changes therein, substitutions therefore, or additions thereto lawfully made.

ADDITIONAL CHARGES

The Monthly Rate, shown above, includes approved decremental rates for Merger Fuel-Related Savings Rider MFS and Merger Capacity Mitigation Rider MCM and the following approved charges as set forth in Annual Billing Adjustments Rider BA:

a) Fuel Adjustment Rate
b) Fuel Adjustment Experience Modification Factor
c) DSM/EE Rate
d) DSM/EE Experience Modification Factor

Supersedes Schedule LGS-TOU-21
Effective for service rendered on and after December 1, 2012
NCUC Docket No. E-2, Subs 1018, 1019 and 1020
## Appendix G  Piedmont Natural Gas Rates

Rates obtained from Piedmont Natural Gas for fuel rates used in energy, demand, and cost savings recommendations for this application are shown in the following figure.

![Figure G.1: Piedmont Natural Gas Rates](image)

### Table G.1: Piedmont Natural Gas Rates

<table>
<thead>
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<th>Service Type</th>
<th>Monthly Charge</th>
<th>Rate/Therm</th>
<th>Rate/Therm</th>
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<td>April-October</td>
<td>10.00</td>
<td>0.984717 *</td>
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<tr>
<td><strong>102 - Small General Service</strong></td>
<td>November-March</td>
<td>75.00</td>
<td>0.70911 *</td>
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<tr>
<td></td>
<td>April-October</td>
<td>75.00</td>
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<td><strong>103 - Medium General Service</strong></td>
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<td>142.00</td>
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<td>April-October</td>
<td>142.00</td>
<td>0.72437 *</td>
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<td><strong>104 - Natural Gas Vehicle Fuel</strong></td>
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<td>150.00</td>
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<td>150.00</td>
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<td><strong>105 - Large General Sales Service</strong></td>
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<td>500.00</td>
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(34)