Energy Efficiency Evaluation of Blower Heater Non-Purge Compressed Air Dryers

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Energy Efficiency Evaluation of Blower Heater Non-Purge Compressed Air Dryers

Alexandra Davis

Thesis submitted
to the Benjamin M. Statler College of Engineering and Mineral Resources at West Virginia University

in partial fulfillment of the requirements for the degree of Master of Science in
Industrial Engineering

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Morgantown, West Virginia 2019

Keywords: Compressed Air Dryers, Energy Efficiency, Desiccant Dryers, Energy Consumption, Energy Intensity, Compressed Air
Abstract

Energy Efficiency Evaluation of Blower Heater Non-Purge Compressed Air Dryers

Alexandra Davis

There are several compressed air dryers available for industrial use including, refrigerant, desiccant, and membrane. This research focuses on twin tower regenerative closed loop desiccant dryers, specifically: blower heater non-purge (BHNp) with and without cooling water pumps, Compressed-air Heater Purge (CHP), Blower Heater Purge (BHP), and Pressure Swing Heaterless (PSH). These styles of dryers are used mainly in industries that require extremely dry air such as, food manufacturing, medical air, or sensitive technology manufacturers. The research was conducted by collecting and analyzing real time current draw data on air compressors and associated dryers at eight different facilities (13 compressor systems) in terms of energy, power, and cost. A decision tool was developed to depict the operational characteristics (power, energy, cost) of each type of dryer if used in conjunction with the selected compressor system. Finally, this research, on an equivalent normalized basis, compared and contrasted the different types of dryers in terms of performance and cost. The research concluded that of the five types of desiccant dryer types observed the most energy efficient was the BHNp (with cooling water pump).
Dedication

This work is dedicated to my friends and family and to all those who have supported me through this journey. To my husband who encouraged me to pursue my dreams and to be better for the next generation. Finally, to my fearless grandmothers, who taught me the meaning of grit and perseverance.
Acknowledgement

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I would like to acknowledge my friends and family, past and present, who played a role in my development. To my parents, grandparents, and my siblings for their inspiration and encouragement throughout my journey. Finally, husband and rock, Damien Botts. I am grateful for his support, love, and reassurance throughout this adventure.
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Nomenclature

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>kWh</td>
<td>Kilo Watt Hour (Energy Usage)</td>
</tr>
<tr>
<td>kW</td>
<td>Kilo Watt (Power)</td>
</tr>
<tr>
<td>ACFM</td>
<td>Actual Cubic Feet per Minute (Volumetric Air Flow Rate)</td>
</tr>
<tr>
<td>HP</td>
<td>Output Horsepower for the motor</td>
</tr>
<tr>
<td>PF</td>
<td>Power Factor</td>
</tr>
<tr>
<td>LF</td>
<td>Load Factor</td>
</tr>
<tr>
<td>BHNCP</td>
<td>Blower Heater Non-Purge</td>
</tr>
<tr>
<td>BHP</td>
<td>Blower Heater Purge</td>
</tr>
<tr>
<td>PSH</td>
<td>Pressure Swing Heaterless</td>
</tr>
<tr>
<td>CHP</td>
<td>Compressed-air Heater Purge</td>
</tr>
</tbody>
</table>
1 Introduction

1.1 Introduction to Compressed Air System

Manufacturing in its simplest form can be defined as converting raw material into usable goods. Though the idea seems straightforward, the conversion process consists of layers of moving parts, planning, machinery, and labor. Throughout recent years reducing the energy consumption of the conversion process has been at the forefront of the manufacturing industry. One of the most widely used and inefficient processes in a manufacturing facility is compressing air for pneumatic controls, machines, and tools [1]. Compressed air is considered the most expensive resources used in manufacturing [2]. This has led the industry to taking a more systematic approach to reducing energy consumption in the compressed air process.

Compressed air systems can be broken down into two major categories, demand and supply. The demand side of the system consists of the equipment requiring compressed air. Equipment or uses being fed compressed air typically includes nozzles, tools, leaks, and pneumatic machinery. The supply side of a compressed air system includes receiver tanks, the compressors, air dryers as well as the supporting auxiliary equipment. The diagram in Figure 1.1 shows the topical view, flow, and setup of a compressed air system. A compressor takes in air, compresses it to the desired pressure set point. The compressed air then either goes into a receiver tank, known as wet storage, then to a dryer or directly to a dryer. Commonly, after the dryer the air will go to another receiver tank, dry storage, then to the end users on the demand side of the system. The terms wet storage and dry storage are dependent if the air has been dried via compressed air dryer to the desired dewpoint.
When ambient air is compressed to high pressures, water droplets are produced due to the moisture naturally present in the atmosphere. “At 75°F and 75% relative humidity, a 25-hp compressor will produce 20 gallons of water per day. This water vapor must be condensed and removed from the air system. Condensate is a contaminant that adversely affects end use applications” [3]. “Humidity is expressed in terms of pressure and dewpoint. Dewpoint is the temperature at which air is saturated with moisture, or in general, the temperature at which gas is saturated with respect to a condensable component. When the temperature of the air reduces to or below the dewpoint, condensation will occur.” [4] The lower the dewpoint, the less moisture present in the air. In general, compressed air dryers are controlled and operated via dewpoint settings and desired dewpoint is set by the industry type or by the ISO 8573.1 quality standard [4].

This research mainly focuses on twin tower regenerative closed loop desiccant dryers. This type of dryer is not used as commonly as other types due to capital cost and low dewpoints, which are only needed in select industries. Motivation for this research initiated due to the lack of reliable information in the academic field. Since most of the research and development is conducted by manufacturing facilities “results are rarely published” leading to a lack of advancement of information in the academic and industrial sectors [5].

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1 http://7.lokiu.treatymonitoring.de/data/compressed-air-diagram.html
Performance of the desiccant air dryers in this research were determined based on the energy intensity of the air production, that is, the amount of energy it takes to produce one acfm of dried compressed air (kWh/acfm). This intensity, along with similar metrics such as energy demand, were then used to determine the forecasted annual operational cost of the compressed air dryer system for a given facility.

1.2 Compressed Air Dryers

Concentrated moisture can be harmful to instrumentation, air system infrastructure, as well as the end product. Compressed air dryers are used to help reduce the amount of moisture in the compressed air and the system. There are several types of compressed air dryers on the market, all able to meet varying requirements depending on a facility’s operational needs and desired dewpoints.

One of the most common dryers, refrigerant dryers, use a liquid coolant to reduce the amount of moisture and for most processes this air is dry enough for general production. Warm compressed air will enter the refrigerant dryer where is cooled. Once cooled, the moisture in the compressed air will condense into water droplets which are then removed. This makes the air dryer than ambient air. This dryer type is common in the market due to the low operational and capital cost. Due to its commonality substantial research has been conducted on this type of dryer. A refrigerant dryer typically reaches dew points of 37°F. However, some processes require the compressed air to be extremely dry, such as, manufacturing of sensitive electronics, food manufacturing, pharmaceuticals, and hospital surgical air. Extremely dry air is accomplished using desiccant type dryers. Desiccant dryers typically range in minimum dewpoints from -40°F to -100°F. There are two main types of desiccant dryers, single tower deliquescent and twin tower regenerative. This research focused on the operational characteristics and limitations of twin tower desiccant air dryers.

1.2.1 Twin Tower Regenerative Desiccant Air Dryers

Twin tower regenerative desiccant dryers work by having two towers. Wet compressed air is routed through one tower (tower 1), passing through a porous desiccant type media, to dry the air before going to an end use. The porous structure, type, and design will depend on the dryer manufacturer and the minimum dewpoint that is trying to be accomplished. The type of desiccant
material used depends on the dryer manufacturer, however, the most common desiccant material type is silica gel. Once the first tower has become saturated with moisture, the air is then rerouted into the second tower (tower 2) while the first tower goes through a dormant regenerative cycle. A diagram of the process can be seen in Figure 1.2. These types of desiccant dryers differ by the regeneration method that is used. Methods include using selective auxiliary equipment. Some common equipment includes a blower to remove moisture, and heater to heat the tower, or heat recovery from the compressor, while some use purged compressed air to discard the moisture or a combination of the equipment. In general, the most common desiccant dryer types include, Blower Heater Non-Purge (BHNP), Compressed-air Heater Purge (CHP), Blower Heater Purge (BHP), and Pressure Swing Heaterless (PSH) [3]. The names given to each of the desiccant dryers is based on the method of drying the dormant tower.

**Figure 1.2: Twin Tower Regenerative Desiccant Dryer²**

### 1.3 BHNP Summary

Blower Heater Non-Purge (BHNP) dryers are named based on the supporting auxiliary equipment used for tower regeneration. This type of dryer contains a blower and heater which will heat the tower as it regenerates and uses the blower to discard moisture. A key aspect of this dryer design is the fact that the dryer does not utilize compressed air to remove moisture. BHNP dryers can differ in the method of tower cooling. Conventional BHNP dryers cool towers by utilizing ambient air. Newer technology on the market allows BHNP to cool tower via cooled water being pumped

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through the equipment. Both styles of tower cooling will be evaluated in this research. This design has also demonstrated a much lower dew point than other BHNP dryers commercially available. This pioneering technology patent could possibly have the most energy efficient advancements available to the industrial community.

1.4 Need for Research

As energy usage joins the frontier of the socioeconomic agenda, industrial consumers have become more aware of their consumption and the associated cost of operation, both environmental and financial. According to the U.S Energy Information Administration, in 2017 the industrial sector encompassed 32% of energy consumption in the United States. As shown in figure 1.3, industrial customers are the biggest consumers of energy in the US [6]. Major companies are establishing energy reduction standards and have started to reevaluate their operational practices. As one of the biggest consumers in a facility, compressed are systems are being reevaluated and enhanced to be more efficient. The West Virginia University’s Industrial Assessment Center (WVU IAC) reports compressed air systems at mid-sized manufacturing facilities can consume 15% - 50% of the total electricity usage of a facility depending on the industry. One example given by a 2019 WVU IAC report shows the compressed air system consuming over 5.2 million kWh, which roughly translates to $350,000 of the facility’s overhead budget [7].
Minimal research has been completed regarding the energy requirements with respect to the compressed air supply side characteristics of twin tower regenerative desiccant air dryers. This is leaving facilities to make their own conclusions without evidence of performance. This research will provide an unbiased source of knowledge for facilities considering the installation of desiccant dryer systems, allowing personnel to make informed decisions that could impact their operations and energy load for future years to come.

Without this research, the innovative desiccant dryer technology will stand to be an unknown and a misunderstood resource that could otherwise potentially improve industrial consumers energy efficiency efforts. This research stands to progress the understanding of energy requirements for twin tower regenerative desiccant dryer air systems for the academic community to later build upon with further investigative research. This analysis will offer a data driven comprehensive

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3 https://www.eia.gov/energyexplained/index.php?page=us_energy_use
comparison of desiccant air dryers such that the scientific community could identify enhancements that could be made to ultimately improve the energy efficiency of twin tower regenerative desiccant air dryers.

1.5 Research Objectives

In this research, an energy assessment of compressed air systems will be utilized to acquire data from four types of desiccant dryers (BHNP, CHP, BHP, and PSH). It should be noted that only US units will be used in this research. The data will be utilized to develop a comparison matrix and a simulation model. The comparison matrix will focus on comparing the BHNP type dryer to the alternative types of desiccant dryers previously mentioned. The simulation model will be developed with a mindset of providing sufficient information to adequately estimate the cost of operating closed loop desiccant dryers based on process requirements; including energy intensity per acfm of compressed air and the cost associated with each acfm of compressed air. Simulation will allow for a variety of operating parameters to be entered according to a facility’s production requirements, including changes to the type of dryer being used. The simulation output will provide valuable information to the prospective users so that they can make informed decisions regarding the impact that a desiccant dryer (BHNP, CHP, BHP, and PSH) can have on their operations and energy load. The main objectives of this research are:

1) Analyze the data collected on air compressors and associated dryers at the eight facilities in terms of energy, power, and cost.
2) Develop a decision tool to depict the operational characteristics of the dryers based on seasonal and operational changes.
3) For a selected set of the eight facility’s compressor operational data, determine the operational characteristics (power, energy, cost) of each type of dryer if used in conjunction with the selected compressor system and verify the results.
4) On an equivalent normalized basis, compare and contrast the different types of dryers in terms of performance and cost.

1.6 Conclusion

Largely, manufacturing systems require compressed air to function optimally. Compressed air is seen as one of the safest, most reliable, but also most expensive resources traditionally used in
manufacturing and thus, needs further improvement [1]. As the energy load in the US continues to grow and the socioeconomic climate of the country changes manufactures are looking to reduce their energy load. To do so, manufacturers are systemically improving their processes. Compressed air dryers are a major component of a compressed air system and thus should be evaluated as such. Desiccant dryers are popular in tech companies, hospitals and food manufactures, but minimal research has been done to compare the energy consumption of the different forms of regenerative desiccant dryers. To broaden the academic and industrial sectors understanding, these dryers will be evaluated and compared in order to drive the energy efficiency standards and expectations further.
2 Literature Review

2.1 Compressed Air Systems Energy Efficiency Design

Seslija et. al [1] presents experiences within saving energy in a compressed air system. The article begins by stating that compressed air used throughout industry and is often seen as one of the more reliable and safe forms of energy used in production. The author approaches the issue of energy efficiency in compressed air systems with a systematic outline. This approach includes eight major steps; audit and system analysis, establishment of internal standards, minimization of losses, harmonization of production and consumption, identification of peak loads, automation and integration of peak loads, equipment maintenance and finally, performance monitoring. The authors continue by examining and explaining each of the eight steps of the systematic approach, which includes, the goal of the step and how to achieve said goal. Each step is individually identified and then examples are given to back up the proposed energy improvements if the step were to be followed. As an example, for the step “minimization of losses” the author explains the benefit of leak detection and the causes of pressure drop from one end of a facility to another. The paper is concluded by stating that special considerations should be well-thought-out in all aspects of the compressed air system, from design to management.

Mousavi et. al [8] discusses the overall efficiency of compressed air systems and the how a proper analysis can be conducted through simulation methods. The report dives a topical view on the modelling of energy consumption of a compressed air system, including inputs such as drive motors. The authors define the main energy loss points in a compressed air system as; compressors, dryers, filters, coolers, pipes, valves, nozzles, and controllers. The authors most prominent point, achieved through simulation, is the comparison of variable frequency drive systems to fixed frequency drive systems in terms of efficiency and energy consumption. The simulation resulted with the author choosing the optimal factor levels for a particular facility.

Vetal [9] discusses the general guidelines for designing a compressed air system. Its stated that the design of a compressed air system can affect the reliability and efficiency of compressors and ancillary equipment and to extend the life cycle cost of the system. The article continues by highlighting some of the more important considerations that should be made during the design phase of a compressed air system implementation. The author begins discussing impactful considerations by reviewing ambient conditions. Ambient conditions are those that cannot or that
are not easily changed in the surrounding environment. The author mentions several key ambient conditions that effect compressed air systems such as; elevation humidity, airborne dust particles, ambient process gases, etc. The author then continues to discuss further considerations such as centralized vs decentralized air systems, sizing and selection of a compressor, choosing the appropriate type of compressor, sizing receiver tanks both before and after the dryer and dryer selection. The author continues that, for dryers in particular, that facilities should avoid choosing a dryer straight out of a catalog or having a salesperson convince management to buy the best on the market. Instead, the author encourages facilities to understand the requirements of the system and the inlet conditions present before choosing a dryer. The author concludes the article by discussing general monitoring and control of compressed air systems. The author recommends that the system be evaluated from production to end uses before a system is installed to ensure that the optimal scheme is chosen for the facility.

Radgen et al. [5] presented at the ‘Energy efficiency in Motor Driven Systems Conference’ in 2017. The presented material included a paper where the author discusses a topical overview of compressed air systems and the importance of future work in the area. Future work that was suggested included the potential for energy efficiency improvements for compressed air systems and energy policy advancements needed improve the efficiency of the systems. The document separates the information into tree major portions, efficiency potentials, technology developments and policy. The paper highlights the importance of energy efficiency in compressed air systems around the globe by highlighting the substantial energy used in various countries for manufacturing. The author then focuses on efficiency measures that can be taken to reduce the load on compressed air systems. Suggestions include proper system sizing, compressed air leak repairs and appropriate controls and set points. Newer technology was then examined in terms of controls and smart sensors on the market or in development. Several case studies are then discussed where manufacturing companies implemented suggested recommendation and saw savings. Finally, the document discussed current policy that focuses on compressed air systems in various countries. Policy discussed included financial incentives that manufacturing companies could take advantage of as well as unbiased information programs available in the related countries. The document concludes by addressing the lack of measured information available to the general public. The authors claim that since most of the research and development are conducted by manufacturing facilities “results are rarely published” leading to a lack of advancement of
information in the academic and industrial sectors.

### 2.2 Compressed Air Systems Energy Assessments Efficiency Measures

Sheckler [10] conducted an energy assessment on compressed air systems within a metal wire manufacture. Sheckler discusses the importance of having compressed air systems analyzed systematically as to not have opinion overshadow raw, collected, data. The author claims that assessments must be based on fact-based information with an understanding in both the supply and demand side of the system. The author continues to claim without sufficient data a conclusion would be just an opinion. The research continues by taking a systematic approach throughout the auditing process. The author described the auditing process including meeting with plant personnel, having a detailed walkthrough, collecting preliminary data, attaching data loggers to pneumatic equipment and finally, analyzing the results. Data was logged for roughly seven days before the loggers were removed from the equipment, after which, the author made recommendations to the facility. Recommendations by the author included the following: installation of a variable frequency drive (VFD), added wet and dry storage to allow the compressor to have relief, and finally, implementation of an extensive air leak detection program. The recommendations would help stabilize system performance and improve the compressed air energy intensity. The new system went on to increase the overall efficiency by 14.7 percent. A cost saving of $68,446 per year in energy consumption was realized by the facility. The author closes the article by dictating the importance of compressed air system assessments in the industrial sector.

Li et al. [11] analyzes the current procedures and techniques in Chinese compressed air operations and assessments. The authors claim that industrial compressed air systems in China use more than 9% of the total energy consumed in the country. The authors conducted several compressed air assessments in China which were used to help analyze the current compressed air practices in the country. Through the assessments the authors were able to comprise a list of typical findings consistently seen. Some of the findings included; overestimating the potential savings realized from variable speed drives, poorly maintained after coolers, administration not understanding costs of air leakages, lack of isolation valves in systems, and lack of air regulation in the distribution system. After analysis of the assessments, recommendations were made to improve the compressed air system energy efficiency by improving the technology and management strategies,
repairing air leaks, and by optimizing system design based on the actual demand needed by the end users.

Saidur et al. [12] composed a report that analyzed the compressed air use and energy savings available through the energy auditing process. The report begins by explaining the contents of an energy audit. The contents included in this audit analysis included energy audit objective, process, types of energy audits for specific industry, tools required for an energy audit and data specifically needed for a compressed air energy audit. The authors continue by discussing various recommendations that can be found through the energy audit process to improve the compressed air system and the associated energy consumption. The authors present nine different recommendations including; use of high-efficiency motors, variable frequency drives, leak prevention, outside air intake for the compressor, pressure drops, heat recovery, energy efficient nozzles, variable displacement of the compressor, and keeping the mechanical workings of the compressor clean. Each recommendation is accompanied by mathematical equations and detailed explanations as to why the recommendation would save energy on a conceptual level. After the explanation of the nine recommendations, the author gives the mathematical formulations of the payback period and the emissions mitigation. The author then describes the two most common computer software used to analyze compressed air systems; AirMaster+ and AirSim. The author concludes by discussing the importance of education in the compressed air community. Without proper information and knowledge energy is unable to be saved. The author suggests that mass media and publicizing the benefit of energy savings may be the next step to increase energy savings in industry.

A case study [13] discusses how an aerospace facility was able to implement new controls and equipment to lower their compressed air cost, saving 86% over the initial suggested compressed air system design. This facility had an unusual situation and requirements compared to most facilities in the industry, the facility has two different type of compressed air scenarios that need to be addressed. The facility required both a low flow long-term air supply for general use as well as a very large and extremely dry but short duration of air demand for testing jet engines. The original proposal was to install one oversized system that could meet the requirements of both scenarios using a modulating control system then drying using a fixed-cycle desiccant air dryer. The facility engineers decided to implement an alternative system with two separate compressed
air sub-systems. A variable frequency drive was installed in place of the modulating system to avoid of exertion of the compressor for the general use compressor. A new controller was installed on the air-drying systems as to be controlled via dew point setting rather than fixed interval, this ensured high quality air without extra drying. Finally, instead of having a 250-hp compressor serve the engine testing the facility 2-3 times a day, the team installed a smaller 75-hp compressor with two large air storage tanks, this ensured that the volume of air was available when testing occurred. A few months after the new system was implemented the savings were verified and the facility was able to take advantage of incentives offered by the local utility. Once savings were realized a simple payback was calculated to be roughly 1.5 years.

2.3 Types of Compressed Air Dryers and Associated Energy Consumption

An informational manual [14] published by the Compressed Air and Gas Institute (CAIG) outlines the importance of drying compressed air as well as the consequences of improper drying methods. The manual starts by explaining the basics of air drying describing that compressed air needs to be dried to remove moisture to a specified dew point to avoid the damaging effects of water to equipment. The author then describes applications that would require clean, dry air including, plant air, valve and cylinder controls, air powered tools, instrument air, product preservation, test chambers, and breathing air. The author then moves on to discuss the different types of dryers on the market. The main types of dryers that are most commonly seen include refrigerant and desiccant type dryers. The author then goes into detail about the many available designs of desiccant dryers as well as the pros and cons of each. Some of the advantages and disadvantages include affects from ambient conditions and even a relative comparison of power consumption from one type of dryer to the next.

Marshal [15] discusses the cost on energy consumption that is sometimes not realized by facilities associated with the drying of compressed air. Marshall first explain the overall need for drying compressed air in industrial settings. Marshall explains that air coming out of a compressor is normally 100 percent saturated with water vapor and is typically at a temperature much higher than that of ambient temperature. When this air cools in facility distribution piping the vapor will condense into free water which can lead to sludge in the piping system, rust in pipe and equipment which can all ruin equipment or the compressor itself. For this reason, an air dryer is needed. Marshal states that the most common dryer seen in industry is refrigerant dryers, which cool the
air and separate the water vapor before sending the air to the end users or to dry storage. In some cases of freezing temperatures or a lower required dew point, desiccant dryers are used. Marshall states that the worst energy efficiency mistake made by facilities compressed air systems is not having dew point controls. The author notes that in twin tower desiccant dryers specifically use 15 – 20% of the nameplate capacity for the regeneration cycle. Some controllers will allow for the desiccant to become more saturated before entering the regeneration stage while other controls will allow the regeneration cycle to be moisture controlled rather than the commonly seen timer controls. Marshall states that installing energy efficient controls normally have a payback of 5 months, thus, are usually a good investment for facilities.

The article [16] published by the Compressed Air Best Practices outline types of compressed air dryers including both refrigerant and regenerative desiccant dryers. The article outlines that there are four main types of dryers, each with a subcategory of operation parameters. The four main types listed in the article includes, refrigerant, regenerative desiccant, single tower, and membrane however, only two types are discussed in this article, refrigerant and regenerative desiccant. The author stated that the two types of refrigerant include cycling and non-cycling. In the non-cycling refrigerant dryer, the refrigerant continuously circulates through the system, providing rapid response to changes in operating loads. The cycling type of refrigerant dryer uses the refrigerant to chill a passage in a heat exchanger. The compressed air is cooled via heat sink in this scenario. When compared in terms of energy, the cycling type can realize energy saving at partial or zero air flow while the non-cycling does not have this advantage, however, the cycling type has a higher initial capital cost. The second type of dryer discussed is the regenerative desiccant. The three main categories of desiccant outlined by the author include heatless, heated and heat of compression. The author gives a quick description of each type and then summarizes the category as a whole, describing the advantages and disadvantages seen with regenerative desiccant. Advantages include very low dew point and moderate cost of operation. The disadvantages include high capital cost, periodic replacement of desiccant belly, negative effects seen due to oil aerosols, and the loss of purge air.

Ursillo [17] published an article with the Compressed Air Best Practices discussing the direct energy savings associated with selecting an appropriate compressed air dryer. The author begins by stating that every facility has different needs and applications with their compressed air, thus,
a compressed air dryer should be chosen based on that situation. The dryers most widely used in industry include refrigerant and desiccant. In most situations refrigerant dryers can deliver the specification for an application. A refrigerant dryer can reach dew points between 38-degrees and 50-degrees Fahrenheit, which is typically suitable for most industries. The author explains that refrigerant dryer operates by reducing the temperature of the air (removing moisture) by putting the compressed air in contact with a cold medium. The resulting moisture is removed and discarded through a drain system. Refrigerant dryers are relatively inexpensive compared to other options on the market and are more commonly seen in operation. The author continues to explain the second most commonly seen dryer, regenerative desiccant. Instead of cooling the compressed air to remove moisture, desiccant dryers use porous desiccant beads to absorb the moisture form untreated air. Dew points for desiccant dryers typically range from -35-degrees to -100-degrees Fahrenheit. The author explains that the energy savings can be realized is desiccant dryers by the type of regenerative processed used. The author describes that desiccant dryers that use heat typically also use more compressed air during the regeneration cycle, thus, heatless desiccant dryers are more efficient. The author concludes stating that optimizing the dryer section can lead to energy savings but should be accompanied my system optimization and controls throughout the system.

Fozcz [3] discusses the pros and cons of various types of compressed air dryer and consideration that need to be made with each when choosing a dryer for a system. The author stresses the importance of drying compressed and gives the statistic, “Ambient air entering an air compressor always contains water vapor. At 75°F and 75% relative humidity, a 25-hp compressor will produce 20 gal. of water per day. This water vapor must be condensed and removed from the air system. Condensate is a contaminant that adversely affects end use applications.” This is a staggering figure that allows the reader to imagine the amount of vapor that could ultimately be inserted into compressed air lines and possible product. The author also explains that when ambient air is compressed it is heated to higher temperatures, thus, vaporizing the moisture naturally found in the air, a basic thermodynamics concept. When the heated compressed air enters the system, the moisture is vapor but when the air cools in the lines the vapor will condense and lead to water droplets in the lines, thus, leading to damage to the system and the product. The remainder of the document the author discusses pros and cons of various dryer types. Finally, Fozcz discusses the
importance of dryer selection. Dryers must be selected for the appropriate uses. The wrong dryer selection can lead to air that is too wet and damaging to the system or lead to air that is too dry which accrues unnecessary energy and capital expense.

2.3 Desiccant Type Compressed Air Dryers

Marshall [18] discusses how to improve the efficiency of heatless desiccant compressed air dryers. The author shares that compressors in general are not efficient machines, only 10-15% of the energy put into a compressor actually converts to mechanical energy output. The author notes that this efficiency is even worse for systems with desiccant dryers that use compressed air to purge during the regeneration cycle. The author notes that when purging, this type of dryer consumes 15% of the name plate capacity regardless of the air demand. The purge air alone could cost $30,000/yr. Marshall suggests that the first step towards a more efficient compressed air system is to check if a desiccant dryer is required, most processes don’t need the level of dryness a desiccant dryer produces. Once it has been confirmed that a desiccant dryer is needed Marshall suggests checking to make sure the right size of dryer is purchased, a dryer larger than needed will just continuously draw more power than required. Marshall then suggests engineering controls such as dew point controllers. Savings is attainable by using dew-point-dependent switching. A sensor will turn off the dryer purge when it is not required based on the quality of air on the output of the dryer. The author says that most modern desiccant have dew point sensors installed at time of purchased but need to be activated. If a desiccant dryer does not have the appropriate sensor then one may be retro fitted with a simple payback less than a year. Marshall suggests that desiccant protection should be considered as a saving method. Sometimes the desiccant beads can become contaminated and work improperly causing for there to be more purge air. Protection can come from inlet filter monitoring along with other routine maintenance such as checking for leaks in the drying system.

White [19] published an article in Compressed Air Best Practices describing the inner workings of a desiccant material and how it is used in desiccant dryers. White also explains the different types of desiccant dryers on the market and the pros and cons of each. Desiccant dryers use solid absorbents in granulated form to reduce moisture in compressed air. These absorbents have a plethora of nanopore cavities that are used to capture the moisture in the compressed air. White says there are three commonly used synthetic absorbents including activated alumina, silica gel,
and molecular sieve. Regenerative desiccant dryers consist of two towers with one of the listed absorbents. One tower is used to dry the air while the other is simultaneously regenerated, and the moisture is discarded. The towers will then switch to alternative cycles. The distinguishing characteristic of desiccant dryers is the method used to regenerate the saturated tower. The author states that desiccant dryers are separated into two main classes; pressure-swing and externally heated regenerative dryers. The author then continues to describe the different types of regenerative desiccant dryers on the market as well as their respective operating parameters. The author encourages facilities to stringently review operating requirements before a dryer is chosen. White also encourages facilities to run the proper calculations to ensure an oversized dryer is not purchased.

Marshall [20] reviews personal experience in the field to compile a list of heuristics and common misconceptions regard regenerative desiccant compressed air dryers. Marshall begins by describing the general desiccant air dryer. A dryer that used a desiccant material to absorb moisture in the compressed air, in some cases these dryers can use more than one type of desiccant. Marshall claims the most common dew point for desiccant dryers is -40-degrees Fahrenheit, though not usually needed in general manufacturing facilities unless pipes are exposed to freezing temperatures. Marshall’s first lesson is; purge is based on nameplate rating. That is, the dryer will consume the same amount of compressed air regardless of the end user demand. The second lesson is; sometimes the purge continues when the compressor is off. Without the proper controls the dryer will continue to operate as if there is still water to be removed from the air. Lessons 3,4, and 5 all are associated with purge air; air dryers are one of the biggest consumers of compressed air, purge flow can change, and pressure effects purge. Lessons 6 and 7 discussed sensors that desiccant dryers should be equipped with to avoid over-drying of the air. Marshall then discusses in lesson 8 than purgeless doesn’t always mean purgeless. Marshall describes a situation in which a “purgeless” dryer was unable to cool fast enough naturally before the next cycle phase. The dryer was redesigned to cool using compressed air. Though technically purgeless, the dryer still used compressed air in the regeneration cycle. Marshall’s finally lesson shared is, filter differential costs energy. Marshall discusses the sensitivity that desiccants have to oil and free water contamination, this requires the dryers to have a series of filters before the air is dry. The filters cause a drop from 5 to 7 psi which is used by the compressed to overcome the resistance provided by the filters, in turn, losing energy. Marshall states with the right choice of filter this loss can be minimized.
Marshall [21] discusses the benefits of having controls on desiccant type compressed air dryers but the downfall when these sensors and controls don’t work properly. The author then gives case studies that show the unfortunate effects of dryer controls working improperly. Marshall says that currently, thousands of dollars are saved annual by installing controls on compressed air dryers. The controls reduce the amount of purge air required in the regeneration stage of the dryer’s operation, thus, saving expensive compressed air. Some heated desiccant dryers use sensors and electric elements to heat the air before it comes to the desiccant. This measure increases the effectiveness of the purge and can cut the amount of purge air required to half. Though the electric heating will negate some of the savings it is still an improvement from the alternative. Another popular energy saving control the author discusses is the dew point dependent switching. This style of control allows the desiccant towers to become fully saturated before entering the regeneration cycle, this assists in reducing the purge air required subsequently saving energy. Marshall continues by reminding facilities that probes are sensitive equipment and should be calibrated appropriately. Marshall shares an experience at a manufacturer where the probes experienced calibration drift that cost the company $6,500. The facility was experience calibration drift and was able to make the correction after a consultant was hired. Marshall says it is foolish to rely absolutely on the accuracy of an installed meter. The efficiency of a system relies on probe working properly and retrieving accurate data.

Billet [4] reviews the theoretical workings and common issues of compressed air desiccant dryers that lead to loss of performance. Billet discusses applications where humidity level is specified and that must meet set fourth standards, ISO 8573.1 quality standard. The need for desiccant air dryers and the common applications where desiccant dryers are seen were then identified; pharmaceuticals, dental, medical, electronics, telecommunications, etc. To better educate the reader, Billet explains humidity and dewpoint. “Humidity is expressed in terms of pressure dewpoint. Dewpoint is ‘the temperature at which air is saturated with moisture, or in general, the temperature at which gas is saturated with respect to a condensable component’. When the temperature of the air reduces to or below the dewpoint, condensation will occur.” To narrow the document’s focus, Billet then discusses the general inner workings of desiccant compressed air dryers. The author continues to explain that the porous absorbent in the desiccant towers have specific structures depending on the dewpoint capabilities of a dryer. Billet then claims, “The rate of adsorption is affected by several factors which ultimately determine the adsorption isotherm.
profile and thus the size of the packed bed.” The author continues by discussing the principles of operation of desiccant dryers that require purge air for regeneration. The remaining discussion in the paper is highlighting the purge air used for regeneration, mainly, reviewing factors that can affect the purge air and operation of the dryer. The fundamental claims from the author is that dryers must be sized based on their outflow and that, “low pressure increases volumetric flow and reduces purge air leading to incomplete regeneration, reduced performance and possible failure.” Billet emphasizes the importance of proper compressed air dryer operation and consequently the importance of the cost associated with drying compressed air within the manufacturing and medical industry.

Van Ormer [22] wrote an article specifically analyzing heatless desiccant compressed air dryers. Van Ormer introduces myths that were commonly seen at the time of publication. The main myth that is debunked is that low inlet moisture load to a heatless desiccant dryer can have adverse effects on a dryer system. However, Van Ormer explains that this myth is misleading and untrue. The author claims, “that subjecting heatless desiccant dryers to low-moisture inlet air actually delivers lower dewpoints because there’s less moisture to remove from the desiccant bed” and therefor less energy intensive. Van Ormer continues to discuss a case study where a plant in Illinois saw improvements when the inlet air was adjusted. The claim was also made that the primary driver that would remove moisture from the desiccant want in fact not the heat absorption, but rather the moisture content in the purge air. The author then explains that the information provided in the article is not just important for system and dryer designers, but also key for facility personnel to be aware of during operation. The more maintenance personnel that are aware of the dryer operation the better decision making will be made, and thus, energy demand will be optimized for the system.

Thirugnanasambandam [23] et al. discuss the over whelming cost of compressed air systems. The authors claim that the compressed air systems seen in the manufacturing industry, in many cases, should be seen as a company’s fourth utility falling shortly after electricity, water, and fuel. The authors highlight that efficiency measures in compressed air systems usually fall short of examining the system as a whole. Typically, the focus is solely on the operation of the compressor itself and the end users and, in many cases, neglect the air treatment equipment. The authors continue to then focus on desiccant dryers specifically. The authors claim, “a detailed study on
these [desiccant] dryers revealed that the desiccant type dryers cause energy loss of more than 70% over refrigerated dryer.” The authors then go into more detail describing the energy loss to purging compressed air in twin tower desiccant dryers during their regeneration cycles. The document cites, “The annual energy loss due to compressed air purging is estimated at 176 MWh and for the same capacity, refrigerated dryer consume only 46 MWh per annum.” Finally, the authors highlight the importance of studying desiccant air dryers in order to find an improved method of reaching such low desired dewpoints.

2.4 Conclusion

The literature review conveys the message that compressed air is expensive and is often considered its own utility in the industrial sector. The review shows that there are several different methods used for treating the compressed air and shows that each method has its place in the industry. It is apparent that, though not always required, extremely dry air can only be produced by regenerative desiccant type compressed air dryers. These dryers are considered some of the most energy intensive pieces of equipment used for drying air, but the quality is incomparable. The literature has a plethora of information about the inner workings and the available styles of desiccant air dryers. However, the literature is lacking energy efficiency comparison of these desiccant style air dryers based on their regeneration methods. Thus, research in this area comparing regeneration methods and the simultaneous energy consumption would prove to be beneficial to the scientific community and to the industrial sector. It is important for facilities to be aware of all their options and in turn make an educated decision when installing or improving compressed air systems. With this research, facilities will be more aware of the compressed air energy intensity and be able to make educated decisions, not only regarding the compressor and dryers, but also when procuring new equipment that may draw compressed air for operations. The research can provide economic benefit across the manufacturing community by involving energy efficiency in the administrative decisions ultimately impacting profits and the bottom line.
3 Research Approach

To determine the energy requirements of the compressed air dryers real time data was needed to be collected and analyzed in order to calculate the expected energy consumption and intensity. This is done by using real time data logging equipment to measure the current draw of a particular piece of equipment as well as instantaneous data readings and air pressure as seen in the system. With this information, alongside operating parameters provided by facility personal, the data can then be used with department of energy software and heuristics to adequately forecast the annual energy consumption of the system and the associated cost as related to the volume of air produced, $/acfm.

3.1 Overview of Data Logging and Collection

Data was collected on system compressors as well as the desiccant dryers and regeneration equipment. This data allows for the comparison of twin tower desiccant dryer types by evaluating the current drawn from the compressors and dryers’ auxiliary equipment. The current, in turn, allows for the energy intensity comparison and access to a snapshot of the systems overall efficiency health. Data was collected using current transducers (CTs) that were attached to one leg of a three-phase connection in the electrical panels for the respective pieces of equipment. The CTs collected and recorded the electrical current drawn on a set interval, generally a 3 second interval. At the time of installation, an instantaneous power factor (PF), voltage (V) and current (Amps) were recorded along with nameplate information which was used for the energy consumption calculations. Instantaneous data was collated via clamp on multimeter while the equipment was operational. Collection consisted of the use of ‘onset HOBO®’ data loggers to collect consumption data for each compressed air system observed. Loggers collected the current drawn by various components of the air system including, compressors, dryers, and auxiliary equipment. For the preliminary data, data collection occurred on compressors and the dryer’s equipment including, blower, heater, and cooling water pump. Pressure data was also collected when possible by using pressure transducers (PTs) placed in strategic locations to gather air pressure from the compressor as well as operational air that is being consumed by the dryer. Once the data had been logged, the stored data was uploaded to the accompanied software, HOBOware®, producing a graph and data points ready for analysis.
3.2 First Principle Data Analysis

The electricity consumption was found by applying first principle and foundational energy laws. The equations are used to convert the collected current draw into power then convert the power into electrical energy consumption. Utilizing the extensive database in the US Department of Energy’s software, AIRMaster+, the flow from the compressors were then estimated and the energy intensity was be found for a system.

3.2.1 Energy Consumption

Balanced circuits that use three phase power are considered either a delta (Δ) connection or a wye (Y) connection. The type of connection depends on how the circuit is constructed, a delta connection resembles that of the Greek letter Δ and the wye connection resembles a Y. This research focused on connections with a delta connection. Using instantaneous data and nameplate information the three-phase input power (kW) of a delta connection for a motor is calculated by,

\[ kW = \sqrt{3} \times V \times I \times \cos \phi / 1000 \]  

Where,

- kW = Power consumption in kW
- V = Voltage
- I = Amperage
- \( \cos \phi \) = Power Factor

The average kW calculated was then calculated for active operational periods for 15 minute and one-hour intervals, this assisted in finding the flow of the compressor as well as the power demand for the system. The energy consumption (kWh) is calculated as,

\[ kWh = \Sigma (\text{Avg kW for operating time x Operating time in hours}) \]  

This process was completed for each piece of dryer auxiliary equipment as well as the compressor motors themselves.
3.2.2 Compressor Flow

The flow from each compressor was determined based on the performance profile of that particular compressor, as obtained from the Department of Energy’s AirMaster+ software\(^4\). Regression equations were developed based on performance profiles and utilized to estimate the air flow correlated to the electrical load that was calculated using the data collected by the CTs. The performance profile is in terms of load factor on the vertical axis, and airflow on the horizontal axis. The load factor (LF) of a motor is calculated as follows,

\[
LF = \left( \frac{P_o}{P_r} \right) \times 100\%
\]

(3)

Where,

- \(LF\) = Output power as a % of rated power
- \(P_o\) = Measured input three-phase power in kW
- \(P_r\) = Input power at full-rated load in kW (Output Power/Efficiency)

Then, by utilizing the regression equation as formulated from the AIRMaster+ performance profile, the estimated air flow was then determined by inputting the calculated LF at a given point.

3.3 Preliminary Data

For a manufacturing facility, the data was acquired on the current drawn by the rotary screw oil free air compressors, Blower Heater Non-Purge (BHNP) compressed air dryer heater, BHNP dryer blower, and the cooling water pump that provides cooling to the three air compressors and the dryer. A sample schematic of a typical BHNP dryer, without the cooling water, can be seen in Figure 3.1. The blower connected to the dryer supplies forced air to the heater through the opened valves. The heater heats the air which is then forced through the tower that is regenerating and assists in removing the moisture from the desiccant bed. Data was collected on the pressure variations over time at points near the discharge from the compressors and the dryer as well as within the plant. Data logged continuously on a 3 second interval resulting in over 28,000 data points. With the use of AIRMaster+, simulation, and first principle energy equations, the annual energy consumption for the system was forecasted and the energy intensity of compressed air was projected.

\(^4\) [https://www.energy.gov/eere/amo/articles/airmaster](https://www.energy.gov/eere/amo/articles/airmaster)
3.3.1 Preliminary Data Collection

At the manufacturing facility, data was collected on six pieces of equipment. Data collection consisted of the three compressors in the facility, two 270 kW compressors and one 150 kW compressor; as well as three pieces of auxiliary equipment used by the BHNP air dryer, a blower, heater, and a cooling water pump. The electrical connections were three phase delta for all of the equipment. The compressors showed a steady level of operation and power consumption throughout the assessment. The dewpoint attained by the BHNP dryer was –87°C on the day of the assessment. Based on research and observed data, the BHNP air dryer may be able to operate at stable conditions with dew point of –110°C. Plant personnel noted that the operational hours for the compressed air system was 24 hours a day, 7 days a week. It was also noted that the heater in the dryer operated on a timed cycle and would be manually switched depending on the season, colder months would require longer regeneration operation.
3.3.2 Preliminary Analysis

Initially, data was uploaded into the HOBOware® software then exported into Microsoft Excel® where the rest of the analysis is conducted. The voltage is known for the facility and the power factor was recorded during the logger installation using a clamp on multimeter. Using the 3 second current draw data, the power (kW) and load factor (LF) can be calculated. As an example, power consumed by the 150-kW compressor at 12:00 am is as follows,

\[
\text{kW} = \sqrt{3} \times V \times I \times \cos \phi / 1000 \\
= \sqrt{3} \times 380 \times 159 \times .92 / 1000 \\
= 96.71 \text{ kW}
\]

The load factor at the data point is calculated as,

\[
\text{LF} = (P_o / P_r) \times 100 \% \\
= [96.71 / (150/92)] \times 100 \\
= 59.31 \%
\]

Using the known parameters of the compressor, capacity, type of compressor, operating conditions; a performance profile can be acquired through AIRMaster+. The performance profile for the 150-kW compressor can be seen in figure 3.2. The regression line obtained from figure 3.2 was found to be,

\[
A = (\text{LF} - 18) / 70 \times 100
\]

Where,

A = % of total air flow capacity of the compressor
LF = Load Factor, %
Therefore, the percent airflow utilization (A) for the 150-kW compressor at 12:00am can be calculated as,

\[
A = \frac{(LF - 18)}{70} \times 100 \\
= \frac{(59.31 - 18)}{70} \times 100 \\
= 59.01\%
\]

Finally, the anticipated air flow (acfm) from the compressor can be calculated by multiplying the percent airflow (A) by the full capacity airflow available from the compressor. This full load flow can be found on the compressor’s nameplate or estimated using the AIRMaster+ database. For the 150-kW compressor, the flow at 12:00am can be calculated as,

\[
\text{Flow} = A \times \text{Full Load flow} \tag{3} \\
= .5901 \times 724 \text{ acfm} \\
= 428 \text{ acfm}
\]

---

5 https://www.energy.gov/eere/amo/articles/airmaster
This process was repeated for each compressor and each data point collected from the manufacturing facility. The flow profile for each compressor in the preliminary can be seen in Appendix I.

3.3.3 Annual Extrapolation

After the 3-second flow and power data were found, the 15-minute and 1-hour averages were calculated to determine the estimated operating costs. Several assumptions were made in order to extrapolate the findings. It is assumed that the 2-week data collection period would act as a representative “snapshot” of the facility’s annual operation with little variation in compressed air system requirements. During the two-week period, it is assumed that the maximum power requirement from the compressed air system occurred at least once. It was also assumed that this peak power consumption would occur at least once each month of the year. Finally, according to the manufacturer, 18% of the cooling water pump’s power and energy consumption can be attributed to the BHNP dryer and is reflected in this analysis. The hourly average data for the power in the system for one day can be seen in Table 3.1. The data was then extrapolated to encapsulate the annual usage of the compressed air system. This included simulating the expected consumption of the dryer and forecasted the annual operational cost.

### Table 3.1: Hourly Power Averages for Manufacturing Facility

<table>
<thead>
<tr>
<th>Hour</th>
<th>Cooling Water pump (kW)</th>
<th>Heater (kW)</th>
<th>Blower (kW)</th>
<th>150 kW Compressor Power (kW)</th>
<th>Compressor B Power (kW)</th>
<th>Compressor A Power (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12:00 AM</td>
<td>5.5</td>
<td>37.0</td>
<td>3.7</td>
<td>100</td>
<td>156</td>
<td>172</td>
</tr>
<tr>
<td>1:00 AM</td>
<td>5.5</td>
<td>0.3</td>
<td>3.8</td>
<td>103</td>
<td>158</td>
<td>176</td>
</tr>
<tr>
<td>2:00 AM</td>
<td>5.5</td>
<td>60.3</td>
<td>3.8</td>
<td>106</td>
<td>159</td>
<td>177</td>
</tr>
<tr>
<td>3:00 AM</td>
<td>5.5</td>
<td>70.3</td>
<td>3.8</td>
<td>107</td>
<td>159</td>
<td>175</td>
</tr>
<tr>
<td>4:00 AM</td>
<td>5.5</td>
<td>44.6</td>
<td>3.8</td>
<td>106</td>
<td>156</td>
<td>172</td>
</tr>
<tr>
<td>5:00 AM</td>
<td>5.4</td>
<td>0.3</td>
<td>3.8</td>
<td>107</td>
<td>155</td>
<td>172</td>
</tr>
<tr>
<td>6:00 AM</td>
<td>5.5</td>
<td>58.5</td>
<td>3.8</td>
<td>107</td>
<td>156</td>
<td>172</td>
</tr>
<tr>
<td>7:00 AM</td>
<td>5.5</td>
<td>69.1</td>
<td>3.8</td>
<td>106</td>
<td>156</td>
<td>173</td>
</tr>
<tr>
<td>8:00 AM</td>
<td>5.4</td>
<td>45.2</td>
<td>3.8</td>
<td>106</td>
<td>154</td>
<td>171</td>
</tr>
<tr>
<td>Hour</td>
<td>Cooling Water pump (kW)</td>
<td>Heater (kW)</td>
<td>Blower (kW)</td>
<td>150 kW Compressor Power (kW)</td>
<td>Compressor B Power (kW)</td>
<td>Compressor A Power (kW)</td>
</tr>
<tr>
<td>------------</td>
<td>-------------------------</td>
<td>-------------</td>
<td>-------------</td>
<td>------------------------------</td>
<td>------------------------</td>
<td>------------------------</td>
</tr>
<tr>
<td>9:00 AM</td>
<td>5.5</td>
<td>0.3</td>
<td>3.8</td>
<td>107</td>
<td>155</td>
<td>171</td>
</tr>
<tr>
<td>10:00 AM</td>
<td>5.5</td>
<td>58.7</td>
<td>3.8</td>
<td>106</td>
<td>152</td>
<td>169</td>
</tr>
<tr>
<td>11:00 AM</td>
<td>5.4</td>
<td>71.3</td>
<td>3.9</td>
<td>105</td>
<td>152</td>
<td>168</td>
</tr>
<tr>
<td>12:00 PM</td>
<td>5.4</td>
<td>47.9</td>
<td>3.8</td>
<td>105</td>
<td>153</td>
<td>170</td>
</tr>
<tr>
<td>1:00 PM</td>
<td>5.4</td>
<td>0.4</td>
<td>3.8</td>
<td>105</td>
<td>151</td>
<td>167</td>
</tr>
<tr>
<td>2:00 PM</td>
<td>5.4</td>
<td>57.7</td>
<td>3.8</td>
<td>105</td>
<td>154</td>
<td>170</td>
</tr>
<tr>
<td>3:00 PM</td>
<td>5.4</td>
<td>71.7</td>
<td>3.8</td>
<td>105</td>
<td>155</td>
<td>173</td>
</tr>
<tr>
<td>4:00 PM</td>
<td>5.4</td>
<td>49.0</td>
<td>3.8</td>
<td>105</td>
<td>155</td>
<td>172</td>
</tr>
<tr>
<td>5:00 PM</td>
<td>5.5</td>
<td>0.4</td>
<td>3.8</td>
<td>106</td>
<td>155</td>
<td>171</td>
</tr>
<tr>
<td>6:00 PM</td>
<td>5.5</td>
<td>57.2</td>
<td>3.8</td>
<td>107</td>
<td>157</td>
<td>172</td>
</tr>
<tr>
<td>7:00 PM</td>
<td>5.5</td>
<td>71.7</td>
<td>3.9</td>
<td>108</td>
<td>161</td>
<td>178</td>
</tr>
<tr>
<td>8:00 PM</td>
<td>5.5</td>
<td>50.0</td>
<td>3.9</td>
<td>108</td>
<td>161</td>
<td>176</td>
</tr>
<tr>
<td>9:00 PM</td>
<td>5.5</td>
<td>0.4</td>
<td>3.8</td>
<td>108</td>
<td>159</td>
<td>174</td>
</tr>
<tr>
<td>10:00 PM</td>
<td>5.5</td>
<td>55.9</td>
<td>3.8</td>
<td>108</td>
<td>159</td>
<td>175</td>
</tr>
<tr>
<td>11:00 PM</td>
<td>5.5</td>
<td>71.6</td>
<td>3.8</td>
<td>107</td>
<td>158</td>
<td>175</td>
</tr>
</tbody>
</table>

### 3.3.3.1 Simulation

Several methods of simulation were explored before an adequate, successful, method was discovered. The initial approach considered end users found at a standard manufacturing company. The intention was to calculate the flow (acfm) demand in the plant and thus the dryer’s operational characteristics based on the production schedule proposed by the facility. This method led to many unknown variables with unusable results.

The second method used included using the frequency statistics of the data to determine the statistical distribution. As an example, for the BHNP type dryer, the data was sorted into regenerating periods and non-regenerating periods. The data collected during regeneration was then sorted into bins and a histogram was produced. The histogram for the BHNP blower can be
seen in figure 3.3. As seen in the histogram, the data for the “on” periods followed a classic normal distribution.

![Histogram of BHNP blower current draw](image)

*Figure 3.3: Histogram of BHNP blower current draw*

This method of evaluation was carried out for the BHNP type dryer and for the auxiliary equipment. It was exposed that the electrical profile of this BHNP type of dryer does not depend on the compressed air flow from the compressors or the dew point, it requires manual control to determine the regeneration cycle time based on the seasons. Temperatures during the season determine the regeneration time, winter months require longer heater operation due to the lower temperatures in the season. An 8-hr regeneration cycle is used in the winter and a 4-hour regeneration cycle is used in the summer. Simulation is used to mimic the expected heater operation for each season. The seasonal variations in electrical profile for the BHNP dryer were based upon its current operational characteristics which have been determined based on frequency distribution of the data points. Figure 3.4 depicts the cyclical current drawn during the assessment; Figure 3.5 shows the actual data collected compared to the simulated data using the frequency probability method.
However, upon further evaluation, it was determined to be unnecessary to drill down on the data to such an extreme. As more data points from different dryers were gathered and evaluated, it became evident that a binomial distribution was consistent and reoccurring throughout the data. When the frequency of the current was plotted for the remaining dryers it was evident that the binomial method would be more appropriate. As seen in the original histogram (Figure 3.5), the
range of the data is less than 2 amps (roughly 1.5 kW) which results in a minimal adjustment in the results, there was a 2\% energy consumption difference when comparing the normal and binomial method using the preliminary data. The range was even smaller for the remaining dryer observations. This led to the third and final method used for the annual extrapolation.

For the remaining dryers, the data was sorted to account for when the dryer was on/off. The probability that a dryer component was running (probability of success) was then calculated along with the average “on” power and the peak power that was consumed. This information was then extrapolated to find the estimated annual energy consumption. To account for the preliminary dryer being the only dryer using a cyclical timer, the binomial method of extrapolation made it possible to account for dryers that use Dewpoint Demand Switching (DDS). Dewpoint Demand Switching controls the dryer by *dewpoint hunting*, the dryer will only operate when required by the setpoint. This control system displays an energy profile with varying regeneration times, the binomial method of extrapolation assists in mimicking the sporadic profile. The estimated annual proposed energy usage (PEU) for a piece of equipment was then calculated as,

$$ \text{PEU} = \% \times \text{OH} \times \text{AkW} \quad (3) $$

Where,

\% = Percentage of time running  
OH = Operating Hours  
\text{AkW} = \text{Average Power while Running}

### 3.3.4 Initial Results

Based on the binomial data analysis, the kW peak power consumption and the average air flow produced from the compressors was determined. The operating conditions found for the compressors are shown in Table 3.2.

<table>
<thead>
<tr>
<th>Compressor</th>
<th>Peak power consumed (kW)</th>
<th>Average air flow produced (acfm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>150 kW compressor</td>
<td>108</td>
<td>486</td>
</tr>
<tr>
<td>270 kW compressor (A)</td>
<td>178</td>
<td>667</td>
</tr>
<tr>
<td>270 kW compressor (B)</td>
<td>161</td>
<td>577</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>447</strong></td>
<td><strong>1,730</strong></td>
</tr>
</tbody>
</table>
Based on the data analysis in table 3.2, the efficiency (flow intensity) and health of the compressor system as a whole was determined to be 0.2584 kW/acfm. The flow intensity (FI) is calculated as,

\[
FI = \frac{C_{kW}}{AF}
\]  

Where,

\[C_{kW} = \text{Compressor System Peak Power, kW}\]

\[AF = \text{Average Flow During Active Compressor Periods, acfm}\]

It should be noted that, the average flow during active compressor periods can differ from the average system flow. However, for this manufacturer, the average system flow and average active compressor flow is equal. As an example, the flow intensity for the manufacturing facility is calculated as,

\[
FI = \frac{C_{kW}}{AF} = \frac{447 \text{ kW}}{1,730 \text{ acfm}} = \frac{447 \text{ kW}}{1,730 \text{ acfm}} = 0.2584 \text{ kW/acfm}
\]

After simulation and annual extrapolation, the estimated annual consumption was found. Table 3.3 summarizes the expected annual consumption for the dryer. Assuming that the cost of energy is $0.04/kWh and the peak demand cost is $10/kW-month, the total estimated operating cost for the BHNP dryer is estimated to be $28,229. Table 3.4 summarizes the expected cost to operate each component of the BHNP dryer.

**Table 3.3: BHNP Dryer Operating conditions for Manufacturing Facility**

<table>
<thead>
<tr>
<th>BHNP dryer equipment</th>
<th>Peak power consumed (kW)</th>
<th>Projected kWh per year (Energy)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heater</td>
<td>72</td>
<td>378,466</td>
</tr>
<tr>
<td>Blower</td>
<td>4</td>
<td>33,410</td>
</tr>
<tr>
<td>Cooling water pump</td>
<td>6</td>
<td>47,852</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>82</strong></td>
<td><strong>459,728</strong></td>
</tr>
</tbody>
</table>

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### Table 3.4: Projected Annual Operating Costs for BHNP Dryer for the Facility

<table>
<thead>
<tr>
<th>BHNP dryer equipment</th>
<th>Demand cost per year ($/year)</th>
<th>Energy cost per year ($/year)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heater</td>
<td>8,640</td>
<td>15,139</td>
</tr>
<tr>
<td>Blower</td>
<td>480</td>
<td>1,336</td>
</tr>
<tr>
<td>Cooling water pump</td>
<td>720</td>
<td>1,914</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>9,840</strong></td>
<td><strong>18,389</strong></td>
</tr>
</tbody>
</table>

### 3.4 Other Data Points

After the preliminary data was analyzed the remaining data points were gathered. Observations, including the preliminary data, included five different styles of desiccant dryers, seven companies (some with multiple plants), and 13 dryer observations. Table 3.5 shows each observation with the corresponding dryer type and the identifying name. Note, the preliminary observation will henceforth be referred to as “Manufacturer E.” Observations with similar notation may have been in the same plant, on a different compressed air header, or a different plant within the same company. The recorded current profiles for each of the facilities can be seen in the appendix of this document.

### Table 3.5: Dryer Observation Types

<table>
<thead>
<tr>
<th>Location</th>
<th>Observed Dryer Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>Heatless (PSH)</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>Heatless (PSH)</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>Heatless (PSH)</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>Heatless (PSH)</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>Heater Purge (CHP)</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>Heater Purge (CHP)</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>Heater Purge (CHP)</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>Blower Heater Non-Purge (BHNP) - no cooling pump</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>Blower Heater Purge (BHP)</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>Blower Heater Purge (BHP)</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>Heater Purge (CHP)</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>Blower Heater Non-Purge (BHNP) - no cooling pump</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>Blower Heater Non-Purge (BHNP) - with cooling pump</td>
</tr>
</tbody>
</table>
4 Discussion and Results

4.1 Original Operating Characteristics

Similar to the method described for the preliminary data, the standard operating characteristics were calculated for all of the dryer observations. Table 4.1 shows the standard operating characteristics for each facility. Except for the dewpoint of Manufacturer E, these values will remain constant throughout the analysis. Table 4.1 has been organized by order of type of dryer shown in table 3.5.

<table>
<thead>
<tr>
<th>Location</th>
<th>Original Dryer Type</th>
<th>Max Design Compressor acfm</th>
<th>Max acfm Recorded</th>
<th>Average System acfm</th>
<th>Dew point (°C)</th>
<th>Flow Intensity kW/acf m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>PSH</td>
<td>140</td>
<td>76</td>
<td>3.9</td>
<td>-36</td>
<td>7.6122</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>PSH</td>
<td>136</td>
<td>80</td>
<td>5.4</td>
<td>-10</td>
<td>2.8064</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>PSH</td>
<td>184</td>
<td>102</td>
<td>7.6</td>
<td>-22</td>
<td>4.2320</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>PSH</td>
<td>272</td>
<td>172</td>
<td>22.6</td>
<td>-14</td>
<td>1.8140</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>CHP</td>
<td>1,881</td>
<td>1,250</td>
<td>344</td>
<td>-40</td>
<td>0.2710</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>CHP</td>
<td>3,111</td>
<td>2,625</td>
<td>2,041</td>
<td>-40</td>
<td>0.2339</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>CHP</td>
<td>1,569</td>
<td>1,388</td>
<td>623</td>
<td>-24</td>
<td>0.3041</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>CHP</td>
<td>311</td>
<td>353</td>
<td>206</td>
<td>-40</td>
<td>0.2378</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>BHP</td>
<td>1,455</td>
<td>545</td>
<td>454</td>
<td>-40</td>
<td>0.1842</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>BHP</td>
<td>1,455</td>
<td>545</td>
<td>454</td>
<td>-40</td>
<td>0.1842</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>BHNP- no pump</td>
<td>2,063</td>
<td>1,145</td>
<td>534</td>
<td>-40</td>
<td>0.2403</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>BHNP- no pump</td>
<td>2,886</td>
<td>261</td>
<td>165</td>
<td>-40</td>
<td>0.3634</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>BHNP-with pump</td>
<td>3,276</td>
<td>2,670</td>
<td>1,730</td>
<td>-87</td>
<td>0.2584</td>
</tr>
</tbody>
</table>

Similarly, the compressed air dryer characteristics were calculated and recorded. The operating parameters for the dryers and auxiliary equipment can be seen in Table 4.2. These characteristics include the peak power for the equipment, the average run power of the equipment and the percentage of the time each piece of equipment runs.
<table>
<thead>
<tr>
<th>Location</th>
<th>Original Dryer Type</th>
<th>Blower avg on (kW)</th>
<th>Peak Blower kW</th>
<th>Blower time</th>
<th>Heater avg on (kW)</th>
<th>Peak Heater kW</th>
<th>Heater time</th>
<th>Cooling Pump Avg (kW)</th>
<th>Cooling Pump Peak (kW)</th>
<th>Cooling Pump Time</th>
<th>Purge Rate</th>
<th>Dry Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>PSH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>17%</td>
<td>6%</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>PSH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>17%</td>
<td>100%</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>PSH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>17%</td>
<td>17%</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>PSH</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>17%</td>
<td>37%</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>CHP</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>31.5</td>
<td>34.0</td>
<td>21%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>CHP</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>27.3</td>
<td>29.0</td>
<td>40%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>CHP</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>16.3</td>
<td>18.0</td>
<td>12%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>CHP</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>10.4</td>
<td>10.5</td>
<td>57%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>BHP</td>
<td>9.1</td>
<td>9.6</td>
<td>97%</td>
<td>13.4</td>
<td>13.6</td>
<td>69%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>BHP</td>
<td>9.2</td>
<td>9.5</td>
<td>98%</td>
<td>11.7</td>
<td>11.8</td>
<td>80%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12%</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>BHNP - no pump</td>
<td>7.5</td>
<td>14.0</td>
<td>35%</td>
<td>37.6</td>
<td>40.0</td>
<td>25%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>BHNP - no pump</td>
<td>15.1</td>
<td>16.1</td>
<td>41%</td>
<td>18.7</td>
<td>18.8</td>
<td>38%</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>BHNP- with pump</td>
<td>3.8</td>
<td>3.9</td>
<td>100%</td>
<td>70.8</td>
<td>71.8</td>
<td>61%</td>
<td>5.5</td>
<td>5.5</td>
<td>100%</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
4.2 Normalization

In order to compare and contrast the different types of dryers in terms of performance and expense, data for the facilities needed to be normalized on an unbiased bases in terms of performance, system health, and cost. Operational elements that will need to be adjusted or considered include the power (kW) consumption, the percentage of run time for each piece of auxiliary equipment, the volumetric flow (acfm) going into the dryer, the dryer’s dew setpoint, and the purge rate. During the normalization process, if there were more than one observation for a dryer type the initial system used as the normalization basepoint was determined by the average flow into the compressor.

Literature has shown that there is a relationship between the dryer’s dew set point and the run time of the dryer. Most desiccant dryers determine run time via a dewpoint setpoint using Dewpoint Demand Switching (DDS), meaning that the dewpoint is directly correlated to the run time of the dryer. The other control method for desiccant dryers is using a cyclical timer with set regeneration times. [24] Only one of the dryers observed, Manufacturer E, used a cyclical timer. For this research, a linear normalization is used between the dew point and the run time. The equation used to calculate the runtime is,

\[
\%_2 = (\%_1 \times \text{dew}_2)/\text{dew}_1 \tag{5}
\]

Where,
\[
\% = \text{Percentage of Runtime} \\
\text{dew} = \text{Dewpoint Setpoint for the Dryer}
\]

As an example, the BHNP – with pump dryer (from manufacturer E) will be normalized for Manufacture A2. The run time for the heater would be calculated as,

\[
\%_2 = (\%_1 \times \text{dew}_2)/\text{dew}_1 \tag{5}
\]

\[
= (61\% \times -40^\circ C) / -87^\circ C
\]

\[
= 28\%
\]
There is also a relationship between the flow of air (acfm) and the power demand (kW) from the auxiliary equipment. This relationship was established to account for the varying load on the dryers. This assumption includes load consideration and negates the need for an additional load factors for the equipment as well as the need to consider a multiplying factor for the dryer that would have been considered at the time of installation. This relationship will be applied to both the average running power supplied to the equipment as well as the peak power consumed by a piece of equipment. The relationship between the flow and the power consumption is assumed to be linear and is given as,

\[ kW_2 = \frac{(kW_1 \times acfm_2)}{acfm_1} \]  

Where,
\[ kW = \text{Power Requirement} \]
\[ acfm = \text{Average Compressed Air Flow into the Dryer} \]

As an example, the BHNP – with pump dryer (from manufacturer E) will be normalized for Manufacture A2. The average power intensity of the heater would be calculated as,

\[ kW_2 = \frac{(kW_1 \times acfm_2)}{acfm_1} \]
\[ = \frac{(70.8 \text{ kW} \times 2,041 \text{ acfm})}{1,730 \text{ acfm}} \]
\[ = 83.6 \text{ kW} \]

The estimated proposed energy usage (PEU) for the heater is then calculated as,

\[ \text{PEU}_1 = \% \times \text{OH} \times \text{AkW} \]
\[ = 28\% \times 8,760 \text{ hr/yr} \times 83.6 \text{ kW} \]
\[ = 205,054 \text{ kWh/yr} \]

Similarly, the run time and power for all the BHNP – with pump auxiliary equipment was calculated. The total energy consumption by auxiliary equipment was calculated for Manufacture A2 as 249,366 kWh/yr and the peak demand was calculated as 89 kW - month.
Other considerations during the normalization process included examining a dryer’s limitations and standard operating practices. The standard operating practice that stayed constant with the dryer type was the purge rate. The purge rates for the dryers were found in the operating manuals provided by the manufacturers or in publicly available from Compressed Air and Gas Institute (CAGI) sheets. The amount of purge air is found by,

\[ P = PR \times \text{acfm} \]  \hspace{1cm} (7)

Where,

- \( P \) = Purged Air, acfm
- \( PR \) = Purge Rate
- \( \text{acfm} \) = Average Compressed Air Flow into the Dryer

The purge rate cost (PRC) is calculated using the flow intensity output from the compressors. The flow rate intensity for the corresponding facility can be seen in table 4.1. The purge rate cost is calculated as,

\[ \text{PRC} = (FI \times P \times $/kW \times 12 \text{ months/yr}) + (FI \times P \times \text{OH} \times $/kWh) \]  \hspace{1cm} (8)

Where,

- \( FI \) = Flow Intensity, kW/acf m
- \( P \) = Purged Air, acfm
- \$/kW = Compressed Air Flow into the Dryer
- \( \text{OH} \) = Operating Hours, hr/yr
- \$/kWh = Unit Cost of Energy, $/kWh

The purge rate cost played a key role in finding the optimal dryer selection. The dryers who used compressed air to purge were seen as the highest cost dryers. The BHNP dryer did not require purge air, thus, there is not purge rate cost associated with the dryer. The total annual cost (TAC) is given by,

\[ \text{TAC} = ($/kW \times kW \times 12 \text{ months}) + (PEU \times $/kWh) + \text{PRC} \]  \hspace{1cm} (9)

Where,

- \( \text{TAC} \) = Total Annual Cost, $/yr
- \$/kW = Unit Cost of Power, $/kW
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kW = Peak Demand of the Dryer, kW
PEU = Proposed Energy Usage, kWh/yr
$/kWh = Unit Cost of Energy, $/kWh
PRC = Purge Rate Cost, $/yr

The total estimated annual cost for a BHNP – with pump dryer when normalized for Manufacture A2 is calculated as,

\[
TAC_1 = ($/kW \times kW \times 12 \text{ months/yr}) + (PEU \times $/kWh) + PRC
\]

\[
= ($10/kW \times 89.32 \text{ kW-month} \times 12 \text{ months/yr}) + (249,366 \text{ kWh/yr} \times $0.04/kWh)
\]

\[
= $10,718 + $9,975 + $0
\]

\[
= $20,693/yr
\]

There were only two dryer types that could be normalized for all facilities where data was collected, the BHP dryer and the CHP dryer. The manuals for these particular dryers showed the minimum and maximum flow that could be supported through similar models. One of the dryer types was unable to sustain the airflow from alternative facilities, the PSH dryer could only operate with extremely low flow, thus, only the hospitals used this category of dryer. The other three types of dryers were unable to service the flow from all of the facilities.

Finally, one facility, Manufacturer E, had an extreme dew point of -87°C which could only be supported by the original dryer type (BHNP – with pump). In order to normalize the operating parameters, the dew point for this facility was brought up to the -40°C, this is the temperature limit for the remaining dryer models. Thus, this facility could not realistically be supported by the other dryer types unless the facility was able to adjust the dewpoint requirements within the constraints set forth by the dryer manufacturers.

4.3 Results

After the normalization process was conducted for each facility, where applicable, an energy usage, peak demand, cost of purge air, and total annual cost matrices were composed. The cells marked with “NA” were unable to be normalized to the limitations previously mentioned. Major assumptions that were made while composing the matrices included the annual operating hours, the operating hours for each facility that was visited was consistent, 8,760 hr/yr. The cost of energy and power were assumed to be $0.04/kWh for the energy consumption and $10/kW-month for the
peak demand cost. The normalized characteristics for each facility and corresponding dryer type can be seen in the appendix of this document.
### 4.3.1 Energy Consumption

The energy consumption for the selected dryer with the corresponding facility was calculated and can be seen in table 4.3.

#### Table 4.3: Energy Consumption Matrix

<table>
<thead>
<tr>
<th>Facility</th>
<th>BHNP - no pump</th>
<th>BHNP - with pump</th>
<th>BHP</th>
<th>PSH</th>
<th>CHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>NA</td>
<td>NA</td>
<td>1,236</td>
<td>-</td>
<td>885</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>NA</td>
<td>NA</td>
<td>473</td>
<td>-</td>
<td>338</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>NA</td>
<td>NA</td>
<td>1,479</td>
<td>-</td>
<td>1,058</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>NA</td>
<td>NA</td>
<td>2,783</td>
<td>-</td>
<td>1,992</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>68,588</td>
<td>42,029</td>
<td>121,144</td>
<td>NA</td>
<td>76,428</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>406,945</td>
<td>249,366</td>
<td>718,767</td>
<td>NA</td>
<td>76,428</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>74,530</td>
<td>45,670</td>
<td>131,639</td>
<td>NA</td>
<td>49,350</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>NA</td>
<td>NA</td>
<td>72,546</td>
<td>NA</td>
<td>41,594</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>90,521</td>
<td>55,469</td>
<td>158,506</td>
<td>NA</td>
<td>76,428</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>90,521</td>
<td>55,469</td>
<td>160,416</td>
<td>NA</td>
<td>76,428</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>106,472</td>
<td>65,243</td>
<td>188,056</td>
<td>NA</td>
<td>49,350</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>116,537</td>
<td>20,159</td>
<td>58,107</td>
<td>NA</td>
<td>41,594</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>344,936</td>
<td>459,727</td>
<td>609,244</td>
<td>NA</td>
<td>81,806</td>
</tr>
</tbody>
</table>
4.3.2 Demand Requirements

The peak demand for the selected dryer with the corresponding facility was calculated and can be seen in table 4.4.

**Table 4.4: Power Requirement Matrix**

<table>
<thead>
<tr>
<th>Facility</th>
<th>BHNP - no pump</th>
<th>BHNP - with pump</th>
<th>BHP</th>
<th>PSH</th>
<th>CHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>NA</td>
<td>NA</td>
<td>1</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>NA</td>
<td>NA</td>
<td>1</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>NA</td>
<td>NA</td>
<td>1</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>NA</td>
<td>NA</td>
<td>1</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>35</td>
<td>15</td>
<td>17</td>
<td>NA</td>
<td>34</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>206</td>
<td>89</td>
<td>100</td>
<td>NA</td>
<td>29</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>63</td>
<td>27</td>
<td>31</td>
<td>NA</td>
<td>18</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>NA</td>
<td>NA</td>
<td>10</td>
<td>NA</td>
<td>11</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>46</td>
<td>20</td>
<td>23</td>
<td>NA</td>
<td>45</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>46</td>
<td>20</td>
<td>21</td>
<td>NA</td>
<td>45</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>54</td>
<td>23</td>
<td>26</td>
<td>NA</td>
<td>15</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>35</td>
<td>7</td>
<td>8</td>
<td>NA</td>
<td>8</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>175</td>
<td>81</td>
<td>85</td>
<td>NA</td>
<td>25</td>
</tr>
</tbody>
</table>
4.3.3 Cost of Purge Air

The cost of purge air for the selected dryer with the corresponding facility was calculated and can be seen in table 4.5. It should be noted that the PSH style of dryer uses only purge air for the regeneration method, hence, the cost will only be reflected in the purge air.

Table 4.5: Annual Cost of Purged Air Matrix

<table>
<thead>
<tr>
<th>Facility</th>
<th>BHNP - no pump</th>
<th>BHNP - with pump</th>
<th>BHP</th>
<th>PSH</th>
<th>CHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>NA</td>
<td>NA</td>
<td>1,676</td>
<td>2,374</td>
<td>1,676</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>NA</td>
<td>NA</td>
<td>850</td>
<td>1,205</td>
<td>850</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>NA</td>
<td>NA</td>
<td>1,824</td>
<td>2,584</td>
<td>1,824</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>NA</td>
<td>NA</td>
<td>2,312</td>
<td>3,275</td>
<td>2,312</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>-</td>
<td>-</td>
<td>5,262</td>
<td>NA</td>
<td>5,262</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>-</td>
<td>-</td>
<td>26,948</td>
<td>NA</td>
<td>26,948</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>-</td>
<td>-</td>
<td>10,694</td>
<td>NA</td>
<td>10,694</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>-</td>
<td>-</td>
<td>2,765</td>
<td>NA</td>
<td>2,765</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>-</td>
<td>-</td>
<td>4,721</td>
<td>NA</td>
<td>4,721</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>-</td>
<td>-</td>
<td>4,721</td>
<td>NA</td>
<td>4,721</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>-</td>
<td>-</td>
<td>7,243</td>
<td>NA</td>
<td>7,243</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>-</td>
<td>-</td>
<td>3,385</td>
<td>NA</td>
<td>3,385</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>-</td>
<td>-</td>
<td>25,234</td>
<td>NA</td>
<td>25,234</td>
</tr>
</tbody>
</table>
4.3.4 Annual Dryer Cost

Once the energy consumption, required power, and cost of purged air were found the total estimated cost of dryer operation was determined. The total estimated annual cost for the dryer operation with given facility can be seen in table 4.6.

**Table 4.6: Total Estimated Cost of Dryer Operation**

<table>
<thead>
<tr>
<th>Facility</th>
<th>BHN - no pump</th>
<th>BHN - with pump</th>
<th>BHP</th>
<th>PSH</th>
<th>CHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hospital A</td>
<td>NA</td>
<td>NA</td>
<td>1,845</td>
<td>2,374</td>
<td>1,831</td>
</tr>
<tr>
<td>Hospital B1</td>
<td>NA</td>
<td>NA</td>
<td>989</td>
<td>1,205</td>
<td>984</td>
</tr>
<tr>
<td>Hospital B2</td>
<td>NA</td>
<td>NA</td>
<td>2,003</td>
<td>2,584</td>
<td>1,986</td>
</tr>
<tr>
<td>Hospital B3</td>
<td>NA</td>
<td>NA</td>
<td>2,556</td>
<td>3,275</td>
<td>2,530</td>
</tr>
<tr>
<td>Manufacturer A1</td>
<td>6,918</td>
<td>3,488</td>
<td>12,130</td>
<td>NA</td>
<td>12,399</td>
</tr>
<tr>
<td>Manufacturer A2</td>
<td>41,045</td>
<td>20,693</td>
<td>67,696</td>
<td>NA</td>
<td>33,485</td>
</tr>
<tr>
<td>Manufacturer B1</td>
<td>10,541</td>
<td>5,098</td>
<td>19,622</td>
<td>NA</td>
<td>14,828</td>
</tr>
<tr>
<td>Manufacturer D1</td>
<td>NA</td>
<td>NA</td>
<td>6,878</td>
<td>NA</td>
<td>5,689</td>
</tr>
<tr>
<td>Manufacturer C1</td>
<td>9,130</td>
<td>4,603</td>
<td>13,839</td>
<td>NA</td>
<td>13,162</td>
</tr>
<tr>
<td>Manufacturer C2</td>
<td>9,130</td>
<td>4,603</td>
<td>13,697</td>
<td>NA</td>
<td>13,162</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>10,739</td>
<td>5,414</td>
<td>17,905</td>
<td>NA</td>
<td>11,069</td>
</tr>
<tr>
<td>Manufacturer D2</td>
<td>8,852</td>
<td>1,673</td>
<td>6,679</td>
<td>NA</td>
<td>6,058</td>
</tr>
<tr>
<td>Manufacturer E</td>
<td>34,791</td>
<td>28,137</td>
<td>59,773</td>
<td>NA</td>
<td>31,456</td>
</tr>
</tbody>
</table>
4.4 Comparing Dryer Types

Though not all dryers were compatible with all facilities there is an evident dryer that has the preferred annual cost. At the original unit cost settings, the BHNP – with pump was the lowest cost dryer in any of the facilities it was compatible with. Figure 4.1 through figure 4.13 represent the estimated annual cost for each facility with a given dryer type. The highlighted bar in the figures represent the original dryer type observed at the given facility.

Hospital A originally had a PSH dryer. Due to the low flow at this facility the only dryers that were able to be normalized for this facility where the BHP and CHP type. For this hospital, this was the sole source of air for medical use. The air is used for surgery air and medical pneumatic tools only. For the original dryer type, PSH, the estimated annual cost was calculated as $2,374/yr. This facility had the lowest average flow (acfm) recorded. In turn, this caused Hospital A to have the highest flow intensity cost of 7.6 kW/acf m. The facility was normalized for a BHP dryer, which resulted in an estimated cost of $1,845/yr, and the CHP dryer, which had an estimated annual cost of $1,831/yr. Figure 4.1 illustrates the normalization findings.

Hospital B originally had PSH dryers on each of its three compressed air systems. This hospital was much larger than hospital A, hence, this hospital required three separate systems to serve the entire hospital. Hospital system B1 has the lowest flow amongst this facility. The compressor system was calculated to have a flow intensity of 2.8 kW/acf m. The original annual consumption
estimation for hospital B1 is $1,205/yr. Once normalized, the estimated cost for the BHP was $989/yr and the cost for CHP was estimated as $984/yr. The normalization results are shown in figure 4.2.

The second system at Hospital B, Hospital B2, had the second highest flow at the facility. This compressor system, again, had an extremely low flow serving only medical purposes at the facility. The compressor system was calculated to have a flow intensity of 4.2 kW/acfm. The estimated annual consumption for the PSH dryer was calculated as $2,584/yr. The normalization process resulted with the BHP costing $2,003 and the CHP dryer costing $1,986/yr. The normalization results are shown in figure 4.3.
The final hospital air system was hospital B3, this is the third and final system at this facility. This air system had the highest average flow (acfm) of any of the hospital systems observed. The average flow was calculated to be 22.6 acfm. This flow, however, was too small to normalize for either of the BHNP style dryers. Similar to the other hospital systems, this system utilized the PSH style dryer. The estimated annual cost for the PSH style dryer was $3,275/yr. After normalization, the annual estimated cost for the BHP dryer was $2,556 and the annual cost for the CHP dryer was estimated as $2,530/yr. The results of the normalization is illustrated in figure 4.4.

![Figure 4.4: Hospital B3 Dryer Analysis](image)

Manufacturer A had two separate compressed air systems that served two different process lines (A1 and A2). Both of the compressed air systems at this facility had the CHP style dryer. The A1 air system was able to be normalized for both of the BHNP style dryers and the BHP style dryer. The estimated flow intensity for this system was 0.2710 kW/acfm and the average system flow was shown to be 344 acfm. The estimated cost for the original dryer type (CHP) was estimated to be $12,399/yr. The BHP style dry was estimated to cost $12,130/yr. The options that cost the least were the BHNP style dryers. The BHNP – no pump was estimated to cost $6,918/yr. The optimal option in this scenario was the BHNP – with pump. The estimated cost of the BHNP – with pump was $3,488/yr. The results of the normalizations are shown in figure 4.5.
The second line in manufacture A, A2, had a much higher utilization than the first line. The calculated average flow for this line was 2,041 acfm. This observed system had the highest average flow of any of the systems observed in this research. The flow intensity for this system was calculated to be 0.2339 kW/acfm. The original dryer cost for the CHP dryer was estimated as $33,485/yr. The most expensive dryer normalized for this system was the BHP dryer at $67,696/yr. The BHNP dryers were estimated to be the lowest cost options. The BHNP – no pump was estimated to cost $41,045/yr. The BHNP – with pump was estimated to cost $20,693/yr. The results of the normalization are illustrated in figure 4.6.
Manufacturer B had two different air systems that were observed. These systems were located at different plants but were owned and operated by the same company. The manufacturer B1 system had the third highest flow of any of the systems observed. The system was calculated to have a flow intensity of 0.3041 kW/acfm. The original dryer type, CHP, was estimated to have an annual cost of $14,828/yr. The highest cost system was the BHP system at $19,622. The BHNP – no pump was estimated to have an annual cost of $10,541/yr. The optimal cost dryer was the BHNP – with pump at an estimated annual cost of $5,098/yr. The results of the normalization are illustrated in figure 4.7.

![Figure 4.7: Manufacturer B1 Dryer Analysis](image)

The second plant for Manufacturer B had a similar process to the first, however, the original dryer type was different. The average flow for the plant was calculated as 1,145 acfm. The flow intensity was calculated as 0.2403 kW/acfm. The original dryer type, BHNP – no pump, was estimated to have an annual cost of $10,739/yr. After the normalization process, the BHP style dryer had the highest cost at $17,905/yr. The CHP style dryer had the annual estimated cost of $11,069/yr. The lowest cost dryer style was the BHNP – with pump. The BHNP – with pump had the estimated annual cost of $5,414, almost half of the cost of the original dryer type. The results of the normalization process are shown in figure 4.8.
Manufacture C had two separate desiccant air dryers on the same header. The dryers operated at the same time and had similar operational characteristics and current profiles. It was assumed that since the dryers were on the same header and were fed by the same compressor that roughly half of the flow volume (acfm) was fed to each dryer. Thus, the results are almost identical. The original dryer for the facility was the BHP dryer. For the manufacturer C1 system, the original dryer was estimated to consume $13,839/yr. After normalization, the CHP dryer was estimated to cost $13,162/yr. The BHNP – no pump was estimated to cost $9,130/yr and the BHNP – with pump was estimated to cost $4,603/yr. The results of the normalization are illustrated in figure 4.9.
Similar to the manufacturer C1 results, the optimal choice for the manufacturer C2 system was estimated to be the BHNP – with pump at $4,603/yr. The original dryer, BHP, was slightly less than the first system at, $13,697/yr. The results for the other dryers were the same as the manufacturer C1. The BHNP – no pump was estimated to cost $9,130/yr and the estimated cost for the CHP dryer was $13,162/yr. The results for the normalization are shown in figure 4.10.

![Manufacturer C2](image)

*Figure 4.10: Manufacturer C2 Dryer Analysis*

Manufacturer D had two separate plants at different locations that were owned and operated by the same company. The first facility, manufacturer D1, had the smallest compressor design flow of any of the manufacturers, 311 acfm. This made the system unable to be normalized for the BHNP type dryers. However, the max flow was still too large to normalize for the PSH style dryer. The only dryer that the system could be normalized for was the BHP style dryer. The original dryer, CHP, was estimated to cost $5,689/yr. The only dryer the system could be normalized for was the BHP style dryer. The annual cost for the BHP style dryer was estimated to cost $6,878/yr. This system was one of two that was calculated to have the optimal dryer choice as the actual, original, dryer in the facility. The results are shown in figure 4.11.
The second system at manufacturer D was located at a separate plant than the first system. The process at the manufacture D2 was also different than at the first plant. This system has a BHNP – no pump style dryer with an estimated annual cost of $8,852/yr. This is the only facility where the BHNP – with pump dryer was the most expensive option. The average flow of this system was calculated to be 261 acfm. The flow intensity of this system was calculated to be roughly 0.2378 kW/acfm. After the normalization process, the BHP dryer was estimated to cost $6,679/yr and the CHP style dryer was estimated to cost $6,058/yr. The optimal choice for this system was the BHNP – with pump dryer. The estimated annual cost for the BHNP – with pump dryer was $1,673/yr. The results of the normalization can be seen in figure 4.12.
Manufacturer E was the only facility with the BHNP – with pump style of dryer. This compressed air system was one of two whose original dryer type was the optimal selection. It should be noted that manufacturer E had a dewpoint requirement of \(-87^\circ C\), meaning, that no other dryer type observed could realistically serve this facility unless the process could withstand a higher dewpoint. For the normalization process it was assumed that for the alternative dryer types the facility could operate with at least a \(-40^\circ C\) dewpoint. The average flow of this facility was calculated to be 2,670 acfm, this was the highest average flow observed in this research. The flow intensity for this facility was calculated to be 0.26 kW/acfm. The original dryer type (with \(-87^\circ C\)) was estimated to have an annual cost of $28,137/yr. The highest cost dryer was the BHP style dryer with an annual estimated cost of $59,773. The CHP and BHNP – no pump style dryers had simial costs. The BHNP – no pump dryer was estimated to cost $34,791/yr and the CHP style dryer was estimated to cost $31,456/yr. The results of the normalization are shown in figure 4.13.

![Figure 4.13: Manufacturer D2 Dryer Analysis](image)

### 4.5 Decision Tool

Using the results of the normalization, a decision tool was able to be constructed to evaluate alternative scenarios. The basic user interface is shown in figure 4.14. The user is able to select the dryer type and facility of from a drop-down list. On this page the user is also able to change the demand cost ($/kW) as well as the energy cost ($/kWh). This input area is constructed such that the user can only use values from the dropdown menu, the user is unable to type the name of the dryer or the name of the facility.
Once the input selections are made, the tool will retrieve the operational costs from matrices similar to those seen in tables 4.3 through 4.5. The matrices will update automatically depending on the users demand and energy costs. The output on this page will reflect the peak demand, estimated energy consumption, reflect the average flow for the facility, show the air loss due to purge etc. The output also discloses the annual dryer cost of operation and energy intensity ($/acfm) for drying the air. The other portion of this page shows the bar graphs for each facility similar to figures 4.1 through 4.13. The bar graphs will update automatically as the user changes the input values.

On the second page of the decision tool, “Characteristics”, the user is also able to adjust some of the original operating parameters for a specific facility. These parameters include compressor health statistics (kW/acfm), operating hours, and compressor flow details. Figure 4.15 shows the available information to change. It is assumed that if the user changes anything on this page it is within the limits of this research and dryer capability. Note, the base flow used for normalization will not change with this page.
### Facility Info

<table>
<thead>
<tr>
<th>Facility</th>
<th>Original Dryer Type</th>
<th>Max design acfm</th>
<th>max acfm at given point</th>
<th>Average acfm</th>
<th>Hours/yr</th>
<th>kW/acfm</th>
</tr>
</thead>
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<td>Heatless (PSH)</td>
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<td>8,760</td>
<td>7.6122</td>
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<td>Heatless (PSH)</td>
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<td>80.0</td>
<td>5.4</td>
<td>8,760</td>
<td>2.8064</td>
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<td>Heatless (PSH)</td>
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<td>8,760</td>
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<td>1.8140</td>
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<td>454.0</td>
<td>8,760</td>
<td>0.1842</td>
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<td>Blower Heater Purge (BHP)</td>
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<td>454.0</td>
<td>8,760</td>
<td>0.1842</td>
</tr>
<tr>
<td>Manufacturer B2</td>
<td>Blower Heater Non-Purge (BHNP) - no cooling pump</td>
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<td>Blower Heater Non-Purge (BHNP) - no cooling pump</td>
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<tr>
<td>Manufacturer E</td>
<td>Blower Heater Non-Purge (BHNP) - with cooling pump</td>
<td>3,276.0</td>
<td>2,670.0</td>
<td>1,730.0</td>
<td>8,760</td>
<td>0.2584</td>
</tr>
</tbody>
</table>

*Figure 4.15: Decision Tool - Facility Info*
The remainder of the pages are separated based on dryer type. The dryer pages reflect the normalization results. The dryers that we able to be normalized are color coded similar to the coding in figure 4.15, i.e. if the facility was able to be normalized with a CHP dryer, the facility info on the CHP dryer page would be yellow. The results reflect the run time for equipment as well as the power required that was calculated during the normalization process. The user is unable to change the cells or equations on these pages.

4.6 Discussion

Over the course of this research several aspects about the type dryers observed have been exposed. The CHP and BHP were the most versatile dryers examined. These dryers are designed for flows from 150 scfm up to 8,000 scfm. These dryers were the most common observed (6 of 13 observations) and were able to be normalized for each facility.

Through the normalization process it was discovered that the PSH type dryer is used primarily for low flow applications. The manufacturer of the PSH dryers used in this analysis disclosed that the maximum flow any of their models could support is 200 acfm. It was also mentioned in the manual that the typical application for PSH is high hazard, or low tolerance for heat and maintenance shutdowns, hence, why these types of desiccant dryers were found in the hospitals. The analysis showed that the PSH was not the optimal dryer for the hospitals is terms of cost, however, the other style of dryers would include the auxiliary equipment that was likely trying to be avoided.

Alternatively, it was identified both of the BHNP type dryers are only used for high flow operations. According to the user manuals, the lowest BHNP design flow was around 2,000 acfm. The BHNP dryers have the highest installation cost of the dryers observed, this may explain why so few are seen in industry.

4.7 Conclusion

Annual energy consumption was estimated, and a decision tool was developed for five types of twin tower desiccant dryers and 13 compressed air systems. The dryer systems were normalized in order to compare dryer operations within a similar facility. It was concluded that the BHNP – with pump dryer was the most efficient when it was able to be applied. Though the BHNP dryers are the most expensive at installation, they had the least annual operating cost calculated. The
BHNP characteristic that is different from the other dryers is the lack of compressed air purging. This key characteristic of not purging compressed air is why the cost was so low in many different facilities. Compressed air is considered an additional utility in many facilities and should be treated as such, as is reflected in the findings. The decision tool developed is aimed to assist plant personnel make informed decisions when evaluating their compressed air system and to propel the academic community to do more research and to drive the energy efficiency standards and expectations further.
5 Future Work

The energy analysis conducted in this research used real time collected data as well as information available through the department of energy. This information provides an unbiased knowledge source for the academic community to further advance the industry and a unbiases source for industry leaders to consider when evaluating their compressed air systems. However, the following data and information would improve the developed energy profiles.

5.1 Effects of Inlet Temperature and Pressure

Air temperature and pressure have an effect on the moisture content in the atmosphere. The dew point of compressed air will increase concurrently with pressure. As seen in figure 5.1, the dew point of compressed air increases depending on the pressure of the air. As an example, if air was compressed at an original atmospheric pressure dewpoint of -10°F to 400 psig the new normalized dew point would be closer to 20F. This shows that as air is compressed the moisture content will saturate at a higher temperature. This is relevant to compressed air dryers in the fact that the higher the inlet dewpoint the harder the dryer would have to work in order to reach the desired dewpoint setpoint.

![Figure 5.1: Relation between Dewpoint and Pressure](image-url)
Similarly, by observing a fundamental psychrometric chart in figure 5.2, the effect of temperature on humidity can be established. The chart shows that as the dry bulb temperature increases the possible moisture content in the air also increases. As an example, at $80^\circ F$ (dry bulb) at 20% humidity the moisture content in the air would be near 30 grains per pound of dry air; where if the temperature is increased to $100^\circ F$ the moisture weight would be near 60 grains per pounds of dry air, nearly double that of the previous reading.

![Psychrometric chart](image)

*Figure 5.2: Psychrometric chart*

For the facilities that were observed for this research, the inlet temperature going into the dryers was assumed to be extremely close. Thus, the changing moisture content of the inlet air was negated. The temperature of the compressor housing rooms were near 75°F at each facility and each facility had similar ambient climate conditions. Each of the compressed air systems observed had manufacturer ratings with similar outlet temperature, near 185°F. Finally, the outlet pressure from the compressors were all within 20 psig of each other, thus, this research omitted the minor
variation in moisture content in the compressed. However, under different circumstances with more extreme variation these factors should be accounted for in future work.

5.2 Relationships

Verification of the relationships used in the normalization process would provide a more accurate comparison of the dryers. This research assumed a linear relationship. Though a relationship has been established, the results need to be evaluated and confirmed. This would be done with a real time dew point monitor along with a system that has varying flows and real time current draw data. The data in this research consisted of 13 compressed air systems and five types of dryers. However, some of the dryer types only occurred once. In the future, more data should be gathered to help reinforce the findings in this research. Data would need to be collected and the relationships would need to be established and confirmed.
6 References


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7 Appendix I

7.1 Preliminary Data

This section shows the flow data that was collected for the preliminary facility (Manufacturer E).

Figure 7.1: Total flow from all plant compressors (preliminary Data)
Figure 7.2: Average hourly flow from each of the plant compressors in acfm (preliminary Data)

Figure 7.3: Flow profile for 270 kW plant compressor A (Preliminary Data)
Figure 7.4: Flow profile for 270 kW plant compressor B (Preliminary Data)

Figure 7.5: Flow profile for 150 kW plant compressor (Preliminary Data)
7.2 Raw Collected Data

This section shows the screen shots of the raw data in HOBOware®.

7.2.1 Hospital A

Hospital A consisted of two single stage screw compressors that were both 20hp.

Figure 7.6: Hospital A Compressor 1 Current Profile

Figure 7.7: Hospital A Compressor 2 Current Profile
7.2.2 Hospital B1

Hospital B1 consisted of two 15hp single stage screw compressors.
Figure 7.10: Hospital B1 Compressor 2 Current Profile

Figure 7.11: Performance profile for Hospital B1 (15hp)
7.2.3 Hospital B2

Hospital B2 consisted of four 10hp single state screw compressors.

*Figure 7.12: Hospital B2 Compressor 1 Current Profile*

*Figure 7.13: Hospital B2 Compressor 2 Current Profile*
Figure 7.14: Hospital B2 Compressor 3 Current Profile

Figure 7.15: Hospital B2 Compressor 4 Current Profile
7.2.4 Hospital B3

Hospital B3 consisted of four single stage 15hp compressors.
Figure 7.18: Hospital B3 Compressor 2 Current Profile

Figure 7.19: Hospital B3 Compressor 3 Current Profile
Figure 7.20: Hospital B3 Compressor 4 Current Profile

Figure 7.21: Performance profile for Hospital B3 (15hp)
7.2.5 Manufacturer A1

Manufacturer A1 had a CHP style dryer. Data was collected on the dryer’s heater and on three compressors. Compressor 3 was not running during the assessment and is not shown in the figures below. The compressed air system was composed of three rotary screw compressors, each 150hp.

Figure 7.22: Manufacturer A1 Heater Current Profile
Figure 7.23: Manufacturer A1 Compressor 1 Current Profile

Figure 7.24: Manufacturer A1 Compressor 2 Current Profile
7.2.6 Manufacturer A2

Manufacturer A2 had a CHP style dryer. Data was collected on the dryer’s heater and on three compressors. Compressor 3 was not running during the assessment and is not shown in the figures below. The compressed air system was composed of three rotary screw compressors, each 250hp.

Figure 7.25: Performance profile for Manufacturer A1 (150hp)

Figure 7.26: Manufacturer A2 Heater Current Profile
Figure 7.27: Manufacturer A2 Compressor 1 Current Profile

Figure 7.28: Manufacturer A2 Compressor 2 Current Profile
7.2.7 Manufacturer B1

Manufacture B1 had a CHP style dryer. Data was collected on the dryer’s heater and on four compressors. The compressed air system was composed of four rotary screw compressors, two single stage 100hp (with different flow capacities) and two - two stage 100hp. The data for compressor 3 had to be scaled accordingly for this analysis.
Figure 7.31: Manufacturer B1 Compressor 1 Current Profile

Figure 7.32: Manufacturer B1 Compressor 2 Current Profile
Figure 7.33: Manufacturer B1 Compressor 3 Current Profile

Figure 7.34: Manufacturer B1 Compressor 4 Current Profile
Figure 7.35: Performance profile for Manufacturer B1 Compressor 1 and 4 (100hp – two stage)

Figure 7.36: Performance profile for Manufacturer B1 Compressor 3 (100hp – single stage)
7.2.8 Manufacturer B2

Manufacture B2 had a BHNP – no pump style dryer. Data was collected on the dryer’s heater, blower and on three compressors. The compressed air system was composed of three rotary screw compressors, two where 150hp with a low flow and one was 150hp with a higher flow.
Figure 7.39: Manufacturer B2 Blower Current Profile

Figure 7.40: Manufacturer B2 Compressor 1 Current Profile

Figure 7.41: Manufacturer B2 Compressor 2 Current Profile
Figure 7.42: Manufacturer B2 Compressor 3 Current Profile

Figure 7.43: Performance profile for Manufacturer B1 Compressor 1 and 2 (150hp - low flow)
### 7.2.9 Manufacturer C1

Manufacturer C1 consisted of a BHP style dryer. Data was collected on the heater, blower, and on three 200hp compressors. However, only one compressor was running during the assessment and is shown in the figures.
Figure 7.46: Manufacturer C1 Blower Current Profile

Figure 7.47: Manufacturer C Compressor 1 Current Profile

Figure 7.48: Performance profile for Manufacturer C (200hp)
7.2.10 Manufacturer C2

Manufacturer C2 consisted of a BHP style dryer. Data was collected on the heater, blower, and on three 200hp compressors. However, only one compressor was running during the assessment. Manufacturer C2 shared a compressor with C1. The current profile for that compressor can be seen in the previous section.

**Figure 7.49: Manufacturer C2 Heater Current Profile**

**Figure 7.50: Manufacturer C2 Blower Current Profile**
7.2.11 Manufacturer D1

Manufacturer D1 consisted of a CHP style dryer. Data was collected on the heater and on one 75hp compressor.

Figure 7.51: Manufacturer D1 Heater Current Profile

Figure 7.52: Manufacturer D1 Compressor 1 Current Profile
7.2.12 Manufacturer D2

Manufacture D2 had a BHNP – no pump style dryer. Data was collected on the dryer’s heater, blower and on three compressors. The compressed air system was composed of four compressors, only three were running the day of the assessment. The three compressors that were running were two 150hp and one 200hp compressor.
Figure 7.55: Manufacturer D2 Blower Current Profile

Figure 7.56: Manufacturer D2 Compressor 1 Current Profile
Figure 7.57: Manufacturer D2 Compressor 2 Current Profile

Figure 7.58: Manufacturer D2 Compressor 3 Current Profile
Figure 7.59: Performance profile for Manufacturer D2 (150hp)

Figure 7.60: Performance profile for Manufacturer D2 (200hp)
7.2.13 Manufacturer E

Manufacturer E was the preliminary data used in this research. The facility consisted of a BHNP – with pump style dryer. Data was collected on the dryer’s blower and heater. The facility also consisted of three air compressors, one 150 kW and two 270 kW. Both of the 270kW compressors (1 and 2) required scaling in order to conduct the data collection.

![Manufacturer E Heater Current Profile](image1)

*Figure 7.61: Manufacturer E Heater Current Profile*

![Manufacturer E Blower Current Profile](image2)

*Figure 7.62: Manufacturer E Blower Current Profile*
Figure 7.63: Manufacturer E Water Pump Current Profile

Figure 7.64: Manufacturer E Compressor 1 Current Profile
Figure 7.65: Manufacturer E Compressor 2 Current Profile

Figure 7.66: Manufacturer E Compressor 3 Current Profile
Figure 7.67: Performance profile for Manufacturer E (150kW)
### 7.3 Normalization Characteristics

Table 7.1: Normalization Results for BHNP - with pump

<table>
<thead>
<tr>
<th>Facility</th>
<th>Dew Point (C)</th>
<th>Blower avg (kW)</th>
<th>Peak Blower kW</th>
<th>Blower time</th>
<th>Heater avg (kW)</th>
<th>Peak Heater kW</th>
<th>Heater time</th>
<th>Cooling Pump avg kW</th>
<th>Cooling Pump Peak (kW)</th>
<th>Cooling Pump Time</th>
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<td>Manufacturer A1</td>
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<td>1.1</td>
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*Used for Normalization
Table 7.2: Normalization Results for BHNP - no pump

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<th>Dew Point (C)</th>
<th>Blower avg (kW)</th>
<th>Peak Blower kW</th>
<th>Blower time</th>
<th>Heater avg (kW)</th>
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*Manufacturer B2

*Manufacturer D2

*Used for Normalization

98
Table 7.3: Normalization Results for BHP

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<th>Peak Blower kW</th>
<th>Blower time</th>
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<td>0.1</td>
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<td>9.6</td>
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</tr>
<tr>
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<td>11.2</td>
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<td>14.8</td>
<td>14.9</td>
<td>75%</td>
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</tr>
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<td>3.3</td>
<td>3.5</td>
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</tr>
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*Used for Normalization
Table 7.4: Normalization Results for CHP

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<tr>
<th>Facility</th>
<th>Dew Point (C)</th>
<th>Heater avg (kW)</th>
<th>Peak Heater kW</th>
<th>Heater time</th>
<th>Purge Rate</th>
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<tr>
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<td>12%</td>
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<tr>
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</tr>
<tr>
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<td>12%</td>
</tr>
<tr>
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<td>31.5</td>
<td>34.0</td>
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<td>12%</td>
</tr>
<tr>
<td>*Manufacturer D1</td>
<td>-40</td>
<td>10.4</td>
<td>10.5</td>
<td>57%</td>
<td>12%</td>
</tr>
</tbody>
</table>

*Used for Normalization