THE ENERGY EFFICIENCY OF DEHUMIDIFIED TWO-STAGE AIR COMPRESSION

A Thesis

by

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ABSTRACT

Compressed air systems (CAS) are an integral component of most manufacturing facilities yet converting electricity to useful pneumatic energy may be 10-15% efficient while the remaining 85-90% of energy is lost to the surroundings as thermal energy. Although best practices to reduce operating cost focus on eliminating inappropriate uses and system losses, none suggest improving the thermodynamic process of air compression, which is the focus of this study. For example, a two-stage air compressor (TSC) with intercooling can reduce the energy consumption by up to 18% when compared to a single-stage model. The humidity of inlet air however, can limit allowable intercooling and increase the demand of a TSC by as much as 7%. This investigation parametrically studies the dehumidification of air prior to its compression by a TSC to evaluate the net change of energy demand using a TSC equipped with a dehumidifier when compared to the standalone TSC. The equations used to model the TSC and dehumidifiers are derived by using thermodynamic principles and a fractional factorial experiment.

The dehumidifiers will be evaluated with makeup air from outdoors, which is normally the inlet air for the CAS. The makeup air temperature ranges from 70 to 110 °F and its relative humidity ranges from 30 to 90% and it is either cooled to a temperature of 45 °F by using a vapor compression refrigeration cycle (air conditioner) or dehumidified by a rotary solid desiccant dehumidifier (desiccant wheel). After dehumidification, the air is compressed by a TSC to a discharge pressure ranging from 75 to 150 psig. The results show that an air conditioner installed at the inlet of a TSC can reduce the energy demand of the system by as much as 11%. In contrast, a desiccant wheel fitted onto a TSC can cause the TSC to consume as

much as 5% more specific work. Additionally, the total demand of the desiccant wheel dehumidified TSC increases as much as 80% in the form of thermal energy necessary to regenerate the desiccant wheel.

DEDICATION

I dedicate this work to my research advisor Dr. Michael Pate for exploring technology development, and to my direct family Enrique, Eugenia, and Priscilla Pelli for supporting my engineering career and aspirations.

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All other work conducted for this thesis was completed by the student independently.

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NOMENCLATURE

$psia$	Pounds per Square Inch, Absolute Scale
$psig$	Pounds per Square Inch, Gauge Scale
P	Total Pressure $(psia)$ or $(psig)$
P_{da}	Partial Pressure of Dry Air $(psia)$
P_{wv}	Partial Pressure of Water Vapor $(psia)$
P_g	Saturated Vapor Pressure $(psia)$
P_f	Saturated Liquid Pressure $(psia)$
$°R$	Rankine, Absolute Scale
$°F$	Fahrenheit, Relative Scale
T	Temperature $(°F)$ or $(°R)$
T_{sat}	Saturation Temperature $(°F)$ or $(°R)$
T_{dew}	Dew Point Temperature $(°F)$ or $(°R)$
lbm	Pound mass
m_{da}	Mass of Dry Air (lbm)
m_{wv}	Mass of Water Vapor (lbm)
mf	Mass Fraction $(dimensionless)$
$lbmol$	Pound mole
N	Number of moles $(lbmol)$
M_{da}	Molar Mass of Dry Air $\left(\frac{lbm}{lbmol}\right)$
M_{wv}	Molar Mass of Water Vapor $\left(\frac{lbm}{lbmol}\right)$

M_{aa}	Molar Mass of Atmospheric Air $\left(\frac{lbm}{lbmol}\right)$
Btu	British Thermal Unit
$\dot{W}$	Time Rate of Work $\left(\frac{Btu}{s}\right)$
$\dot{Q}$	Time Rate of Heat $\left(\frac{Btu}{s}\right)$
$\dot{m}$	Time Rate of Mass Flow $\left(\frac{lbm}{s}\right)$
$\bar{R}$	Universal Gas Constant $\left(\frac{Btu}{lbmol-°R}\right)$
R_{da}	Gas Constant of Dry Air $\left(\frac{Btu}{lbm-°R}\right)$
R_{aa}	Gas Constant of Atmospheric Air $\left(\frac{Btu}{lbm-°R}\right)$
w	Specific Work $\left(\frac{Btu}{lbm}\right)$
h	Specific Enthalpy $\left(\frac{Btu}{lbm}\right)$
h_f	Saturated Liquid Specific Enthalpy $\left(\frac{Btu}{lbm}\right)$
C_P	Isobaric Specific Heat $\left(\frac{Btu}{lbm-R}\right)$
C_V	Isochoric Specific Heat $\left(\frac{Btu}{lbm-R}\right)$
k	Specific Heat Ratio $(dimensionless)$
$\bar{V}$	Velocity $\left(\frac{ft}{s}\right)$
V	Volume (ft^3)
v	Specific Volume $(\frac{ft^3}{lbm})$
ϕ	Relative Humidity $(decimal)$
ω	Specific Humidity $(dimensionless)$

g	Acceleration due to Gravity $(32.17\frac{ft}{s^2})$
z	Height (ft)
Σ	Sum $(dimensionless)$

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1. INTRODUCTION

Compressed air systems (CAS) are fundamental to industrial processes. In the United States, approximately 70% of manufacturers use CAS to increase productivity and to provide power for tools where electricity might otherwise be hazardous. However, compressing air is highly inefficient. For example, a single horsepower (hp) air motor requires approximately eight hp of electric power, resulting in a CAS efficiency of approximately 10-15%, while the remaining 85-90% of the total energy input is rejected to the environment as thermal energy. Because of CAS inefficiency and its widespread usage in manufacturing facilities, CAS can account for 30% or more of a facility's operating costs, which for some, significantly exceeds the cost of water, gas, or electricity (U.S. Department of Energy, 2004).

While the initial cost of an average 100-hp air compressor may be between 30,000 and 50,000 USD, annual operating costs in electricity and maintenance are comparable (Compressed Air Challenge, 1998). For example, one study revealed a ten-year cost structure of 76% electricity, 12% maintenance, and 12% purchase price (Marshall, 2012). Further CAS costs arise from inappropriate uses, unwanted air leakage, poor system design, and inadequately operating control systems. Best practices address some of these issues to increase CAS efficiency and to reduce operating costs. Alternatively, modifying the system can improve the thermodynamic processes to increase system efficiency and reduce operating costs, which is the focus of the study reported herein.

One approach that might be taken for modifying a CAS to improve system efficiency can be found in a combustion turbine packaged with an air compressor, which is not a CAS as found in manufacturing facilities but shares some similarities. A common method for increasing the efficiency of a combustion turbine system is to cool its supply air, a technique known as Gas Turbine Inlet Cooling (ASHRAE, 2016). It should be noted that in the open literature, there is no evidence of any past studies modifying a CAS to lower the air inlet relative humidity or temperatures to reduce operating cost.

A widespread CAS practice is the use of a two-stage compressor with intercooling to reduce the temperature of the air before it enters the second stage, and reducing the energy consumed by the compressor. However, humidity limits the amount of intercooling that can take place because condensate can form in the intercooler, mixing with the compressor's lubricating oil and possibly damaging the system. To prevent condensation in the intercooler and cool the air entering the compressor's second-stage, it is proposed to dehumidify the air prior to supplying it to the compressor. This thesis investigates dehumidification at the compressor inlet by using either a vapor compression refrigeration cycle or a desiccant wheel to evaluate the net change of energy demand using a TSC equipped with a dehumidifier when compared to the standalone TSC. While the focus is recuperating additional energy consumed due to the presence of airborne moisture, the added effects on compressor inlet air temperature and hence TSC performance must also be considered for each dehumidification technique.

2. RELEVANT THERMODYNAMIC DEFINITIONS

2.1 Overview

Before developing the thermodynamic models for compressed air systems (CAS) and the dehumidification systems, basic thermodynamic definitions relevant to this study are introduced. The parametric studies of both air compression and air dehumidification involve parameters, variables, and quantities that depend on the substance, location, time, and frame of reference. It is important to understand all these terms and associated properties, and to apply reasonable assumptions to simplify the analyses.

2.2 Systems, Surroundings, and the Continuum Hypothesis

A system is either a specific set of particles or a region in space through which they travel enclosed by a boundary to distinguish it from the surroundings. The boundary of a closed system prohibits mass transfer through it whereas the boundary of an open system allows mass transfer through it. Although atoms and or molecules randomly travel and collide on a length-scale of their mean free path, the proposed study uses a length-scale sufficiently large to assume substances have uniformly distributed properties at the specified location.

2.3 Properties, Scales, and Equilibrium

Properties have magnitudes relative to standard unit systems and this study uses the Inch-Pound system, which is the industry standard for CAS in the United States. Absolute scales must be used for numerical analysis because the absolute magnitude of quantities must have the same absolute-zero reference to be comparable. Relative scales are relatable for people experiencing common household phenomena, such as freezing or boiling water. An extensive property 4 depends on the size or extent of the system, an intensive property does not, and specific properties are measured per unit mass. When properties stabilize, the system has reached equilibrium, and according to the state postulate, two independent, intensive properties define an equilibrium state, while mixtures sometimes require a third. The transition between equilibrium states can be expressed by using processes, paths, and cycles. A process is the transition between equilibrium states, and a path is defined by a series of processes. A cycle is a special type of path in which the final state returns to the initial state so that the path may repeat cyclically.

2.4 Idealizations and Assumptions

Certain physical constraints can make it difficult to reach an exact equilibrium state. For example, the process could occur too quickly for the substance to settle, or losses such as friction and leaks referred to as irreversible effects may delay the process. The ideal system behaves independently of irreversible effects and provides a baseline of maximum or ideal performance with which to compare processes that may in fact reflect the real world. For example, although processes can be initiated as necessary at any time, a system can be assumed to operate constantly over time, which is an approximation called steady flow. The steady flow approximation assumes that all properties and conditions are constant over time regardless if those parameters differ by location. To assume that the substance and its characteristics are spatially similar at the location of interest is approximate the system as uniformly distributed. Some other approximations are to assume that a particular property remains constant during a process. A process that occurs at constant pressure is isobaric, while that which occurs at constant temperature is isothermal. Constant enthalpy and entropy processes are isenthalpic and isentropic, respectively. A system with sufficient insulation to prevent heat transfer at its boundary is adiabatic. There are numerous idealizations and assumptions that will be specified during their use.

2.5 Thermodynamic Summary

To summarize, the models proposed in this study are ideal baselines with assumptions such as uniformly distributed properties, steady flow, frictionless compression, and adiabatic behavior. Idealizations and assumptions help to evaluate systems by simplifying the factors that affect performance and providing a baseline with which to compare. Equilibrium states are defined with sufficient properties to gather other properties. The Inch-Pound unit system and absolute scales are used in calculations, though results are sometimes reported by using relative scales for ease of understanding based on the familiarity and expectations of industry. Different thermodynamic paths are evaluated using relevant energy interactions between systems and surroundings with equilibrium states and absolute scales.

3. SINGLE-STAGE AIR COMPRESSION

3.1 Overview

Most compressed air systems (CAS) found in industrial applications utilize multi-stage compressors that have two or more compressor stages with intercooling between the stages. To begin however, compression theory and behavior is introduced using a single-stage compressor to develop a model for determining the specific work of a compressor. A single-stage reciprocating (i.e. positive displacement) air compressor is a piston-cylinder assembly driven by a motor and crankshaft (Figure 1). The piston reciprocates in the cylinder from top dead center (TDC) to bottom dead center (BDC) in two strokes per cycle. The clearance volume is measured with the face of the piston is at TDC, and the displacement volume is measured with the face of the piston at BDC. From TDC, the piston begins drawing air through an intake valve until reaching the BDC, then the piston compresses the air, releasing it through the exhaust valve until reaching TDC again.

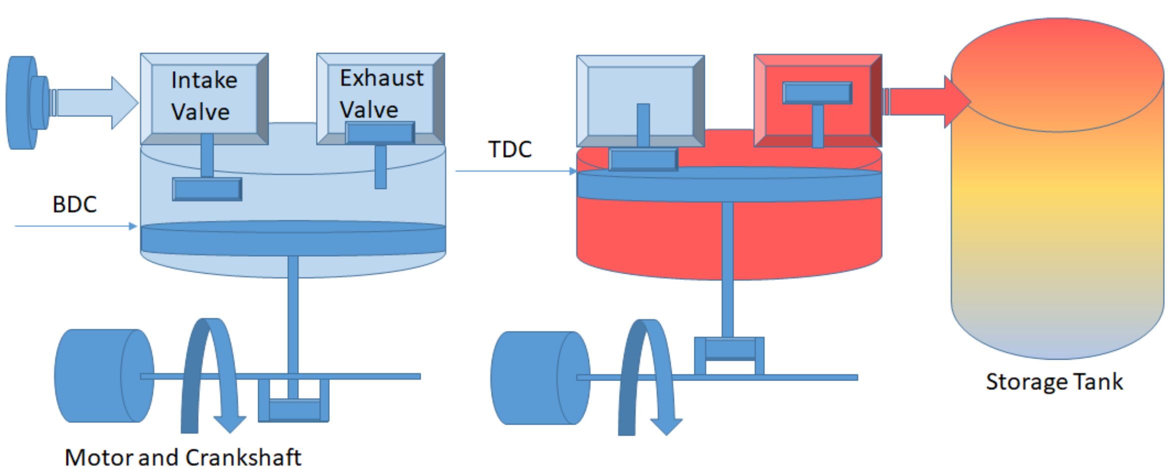

Figure 1: Single-stage air compressor piston at bottom dead center (left), and top dead center (right) discharging to a storage tank

3.2 Assumptions

The intake and exhaust valves of a reciprocating compressor are opened or closed by a differential pressure driven mechanism during the two strokes, which allows air to enter or exit. For simplifying the analysis, the differential pressure that operates the valves is neglected. The imaginary boundary separating the system and surroundings and denoting the control-volume (CV) encloses the compressor hardware and electric work crosses the boundary. The CV is assumed to denote an open system that allows air to cross the boundary, and the compressor is assumed to be insulated to evaluate the process as adiabatic. The change in kinetic and potential energy both inside the CV and at the boundary is assumed to be negligible. Although the rapid, reciprocating motion prevents the air from reaching equilibrium, the process is assumed to be quasi-equilibrium by time averaging quantities and using a steady flow approximation. Therefore, the steady flow energy balance includes electric work for the motor assembly and the change of the air's enthalpy from inlet to discharge of the compressor. The compression process is assumed to be isentropic, though an isentropic efficiency may readily be incorporated.

3.3 Single-Stage Model

In this section, the air being compressed is considered dry air. In upcoming sections, the presence of airborne water vapor will be considered. The critical pressure and temperature of dry air is 547 psia and 238.5 °R, respectively. Because the range of discharge pressure used in this study is well below the critical pressure dry air is considered an ideal gas regardless of the temperature (Çengel & Turner, 2004). The specific work of a single-stage air compressor can be
8 derived from the general energy equation and assumptions mentioned in the previous section. The subscript 1 refers to the discharge state and the subscript 0 refers to the inlet state.

$$w_{compressor} = h_f - h_0, \qquad (Eq. 3.1)$$

Based on the ideal gas assumption, the air enthalpy is assumed to be a function of temperature only. In addition, because the air temperature in this study ranges within a few hundred degrees, the specific heat is assumed to be constant. Therefore, the change of enthalpy can be approximated by the product of the isobaric specific heat C_p and the temperature change.

$$w_{compressor} = C_p \left(T_f - T_0\right), \qquad (Eq. 3.2)$$

Multiplying the numerator and denominator by the gas constant R enables some flexibility in the approximation while maintaining equality in the expression.

$$w_{compressor} = C_p \left(\frac{R}{R}\right) \left(T_f - T_0\right), \qquad (Eq\ 3.3)$$

The gas constant R is equivalent to the isochoric specific heat C_v subtracted from the isobaric specific heat.

$$w_{compressor} = \frac{C_p}{C_p - C_v} R \left(T_f - T_0\right), \qquad (Eq. 3.4)$$

Both the numerator and denominator can be multiplied by the isochoric specific heat to maintain equality.

$$w_{compressor} = \frac{\frac{C_p}{C_v}}{\frac{C_p - C_v}{C_v}} R \left(T_f - T_0\right), \qquad (Eq. 3.5)$$

The specific heat ratio k is the ratio of the isobaric specific heat to the isochoric specific heat.

$$w_{compressor} = \frac{k}{k-1} R \left(T_f - T_0\right), \qquad (Eq. 3.6)$$

For an isentropic process, the initial and final temperature and pressure are related in the following manner.

$$\frac{T_f}{T_0} = \left(\frac{P_f}{P_0}\right)^{\frac{k-1}{k}}, \qquad (Eq. 3.7)$$

Substituting the final temperature and factoring out the initial temperature, the equation for the specific work of a single-stage compressor is as follows.

$$w_{single-stage} = \frac{k}{k-1} R\, T_0 \left(\left(\frac{P_f}{P_0}\right)^{\frac{k-1}{k}} - 1\right), \qquad (Eq. 3.8)$$

3.4 A Parametric Study of Single-Stage Dry Air Compression

In the ideal case, a compressor draws in dry air, which has a specific heat ratio of k = 1.4, and gas constant of R = 0.0686 Btu/lbm-°R. The isentropic temperature-pressure relation introduced earlier requires the use of absolute thermodynamic scales for temperature and pressure, which are °R and psia, respectively. A parametric investigation was performed with dry air entering the compressor at an atmospheric pressure of 14.7 psia and temperatures ranging from 70 to 110 °F, which are typical ambient summer-time temperatures that are of interest in this study. The quantities collected include both the discharge temperature and the specific work of a single-stage air compressor resulting from a discharge pressure ranging from 75 to 150 psig. Figure 2 shows the discharge temperature in °F, and Figure 3 shows the specific work in Btu/lbm, with both parameters being shown as a function of discharge pressure and inlet temperature (i.e. ambient temperature).

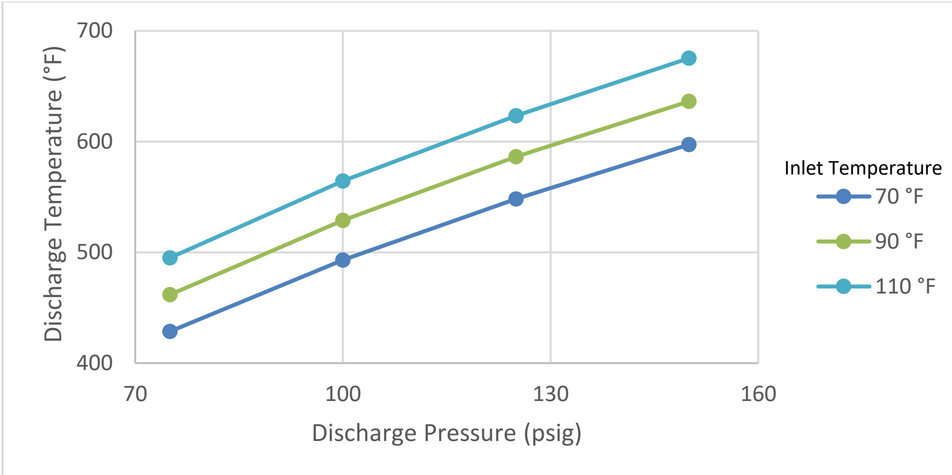

Figure 2: Discharge temperature of a single-stage air compressor supplied with dry air at atmospheric pressure and a temperature ranging from 70 to 110 °F and discharging at a pressure ranging from 75 to 150 psig

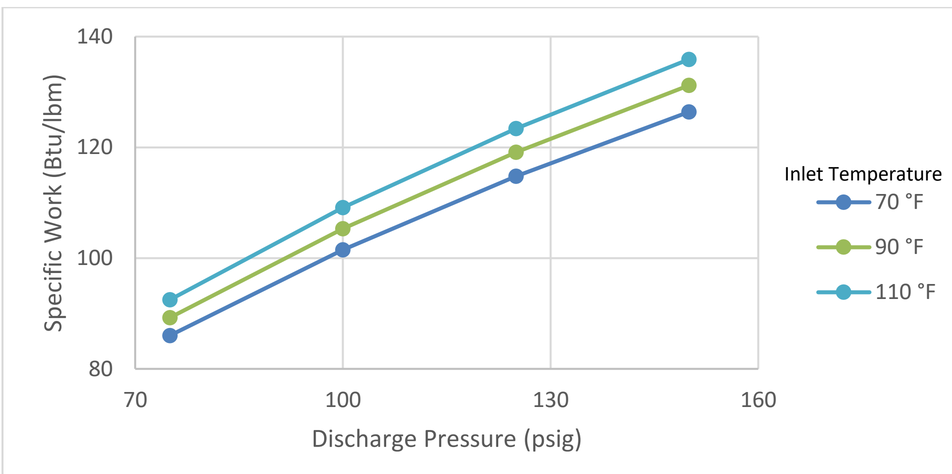

Figure 3: Specific work of a single-stage air compressor supplied with dry air at atmospheric pressure and a temperature ranging from 70 to 110 °F discharging at pressure ranging from 75 to 150 psig

Figure 2 shows that the discharge temperature of the dry air exiting a single-stage compressor is 400 ˚F when initially at 70 ˚F discharged to 75 psig, and 700 ˚F when initially at 110 ˚F and discharged to 175 psig. At a discharge pressure, when comparing dry air entering the compressor at 110 ˚F to that entering at 70 ˚F, the discharge temperature increases by approximately 80 ˚F. At the same initial temperature however, as the discharge pressure increases from 75 to 150 psig, the discharge temperature increases nearly 200 ˚F. Figure 3 indicates that at the same initial dry air temperature but comparing the minimum and maximum discharge pressure of this study, the specific work of the compressor nearly doubles. Figure 3 also indicates that at the same discharge pressure but comparing the minimum and maximum inlet temperature of this parametric study, specific work increases about 10%.

3.5 Single-Stage Dry Air Compression Processes on Pressure Versus Specific Volume Plots

The paths of dry air from initial state (0) to final state (f) can be demonstrated on pressure versus specific volume (Pv) plots. As shown in the Figure 4 Pv diagram, isotherms can be plotted from the ideal gas equation for air, and they cascade to the upper-right as temperature increases. Figure 4 also shows two dry air compression processes both from 14.7 psia to 80 psia but with one at an inlet temperature of 70 °F and the other at 110 °F. Despite the same pressure ratio, the greater inlet temperature starts further right on the Pv plot and travels to a higher discharge temperature isotherm. The specific work of an open compression system is the area enclosed between the y-axis and path line, so it can be observed in Figure 4 that greater inlet and exit temperatures of dry air require more specific work.

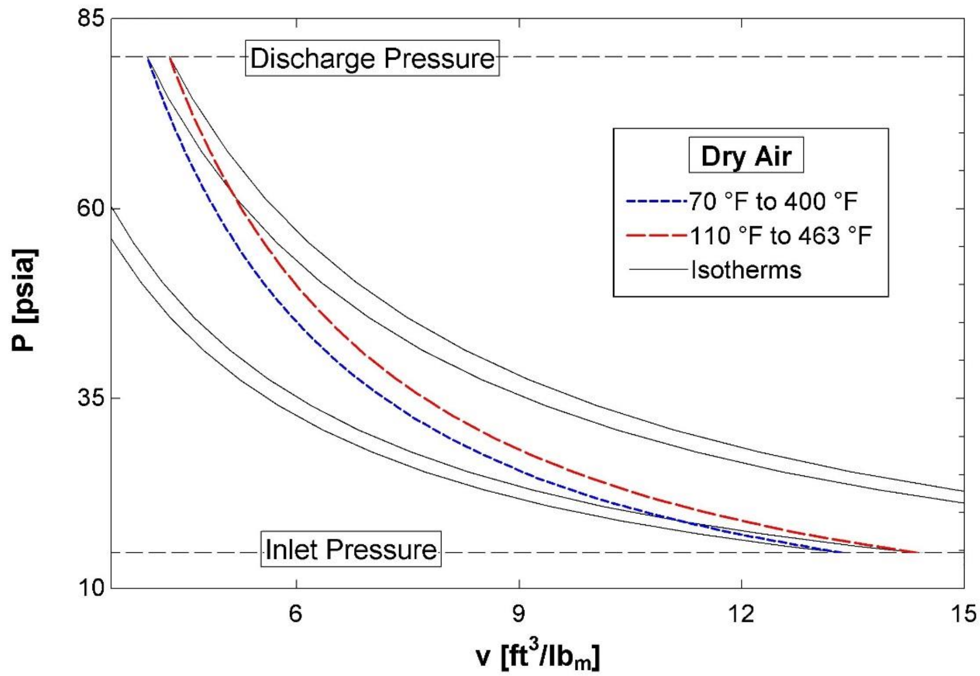

Figure 4: Single-stage compression to discharge pressure of 80 psia for air entering at a temperature of 70 °F versus 110 °F

The Figure 4 Pv plot shows that higher inlet temperature results in higher discharge temperature and requires more compression work. Alternatively, Figure 5 shows two different air compression processes both at the same inlet temperature but discharged to different pressures. Specifically, Figure 5 shows dry air entering at 70 °F being compressed from 14.7 to two different pressures, namely 50 psia and 100 psia. Starting at the same initial condition, a greater discharge pressure results in an extended process to a greater temperature isotherm. The increase of area between the process line and y-axis indicates that increasing the discharge pressure increases the specific work of compression.

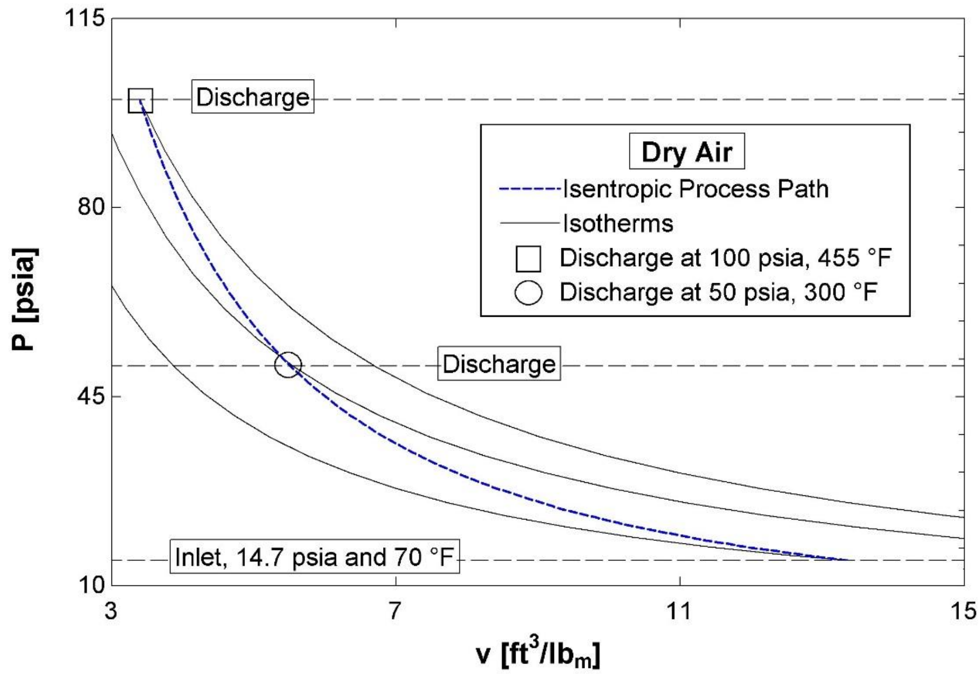

Figure 5: Single-stage compression of air entering at 70 °F discharged to a pressure of 50 psia and 100 psia

Figure 5 shows that if air enters a compressor at the same initial condition but discharges to a greater pressure, it reaches a greater discharge temperature isotherm, encloses more area between the y-axis, and thus consumes more specific work. The parametric study of dry air compression for a range of entering air temperature and discharge pressures in the form of Pv diagrams is important because it shows different process paths, discharge temperatures, and resulting specific work.

3.6 Specific Work Increase Relative to Pressure and Temperature

To complement the understanding gained from the Pv diagram and demonstrate the variation in specific work due to inlet temperature, the specific work of compression is plotted versus discharge pressure in Figure 6 for two different inlet temperatures, namely 70 and 110 ˚F.

On the other hand, Figure 7 shows the specific work as a function of inlet temperature for two discharge pressures, namely 100 and 150 psig. Comparing the two figures shows that variations in discharge pressure has a substantially greater effect on specific work than does variation in inlet temperature. For the inlet temperature range of this parametric study, the single-stage compression work can increase by up to 10%. For the discharge pressure range of the parametric study, single-stage compression work can increase by as much as 50%, which is significantly more than the increase due to temperature. It is likely that production facilities will experience air distribution line losses resulting in lower-than-expected discharge pressure and then compensate by increasing compressor discharge pressure

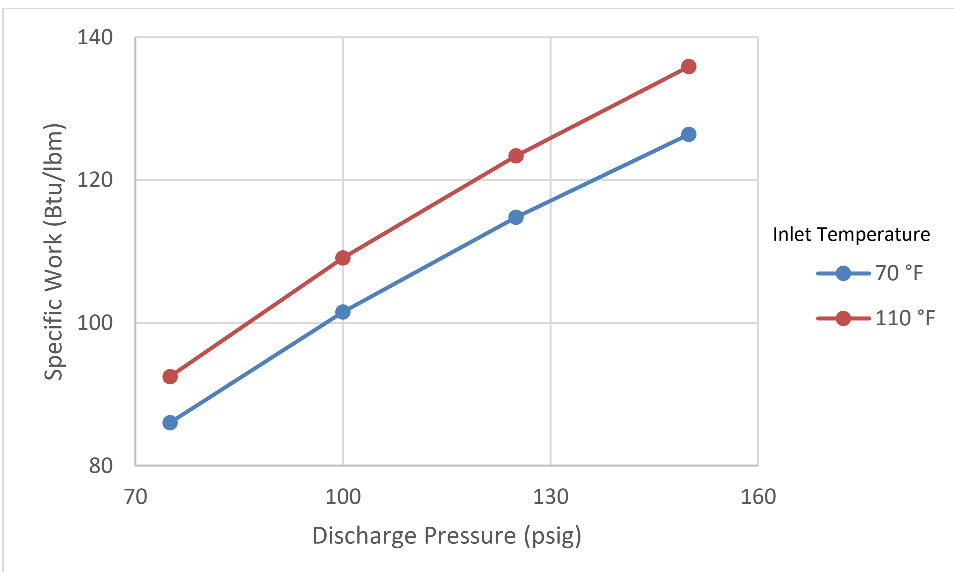

Figure 6: Variation in specific work of a single-stage air compressor for air entering at an atmospheric pressure of 14.7 psia and discharged to a pressure ranging from 75 to 150 psig when the initial air temperature is 70 °F versus 110 °F

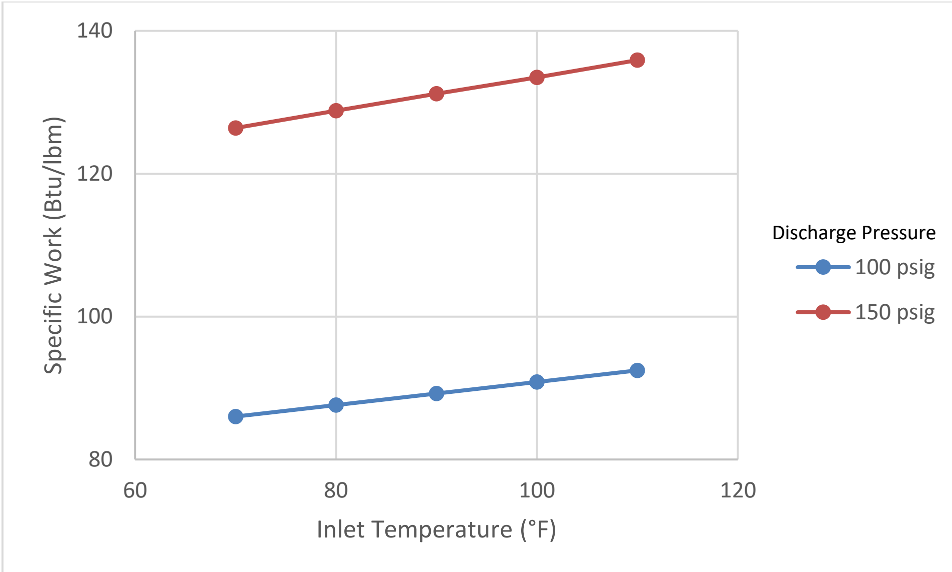

Figure 7: Variation in specific work of a single-stage air compressor for air entering at a temperature ranging from 70 to 110 °F and atmospheric pressure of 14.7 psia then discharged to 100 psig versus 150 psig

While a Pv diagram of an air compression process can show area enclosed between the y-axis and the path line, a plot of the specific work resulting from the working model clearly indicates the magnitude. Figure 6 is a plot of the specific work as a function of discharge pressure when comparing inlet air temperature, and Figure 7 shows a plot of the specific work as a function of inlet air temperature for two different discharge pressures. It can be seen that the difference in magnitude of specific work due to variation in pressure is significant, and it is clear that discharge pressure has a much more significant effect on the specific work than does inlet air temperature.

4. MULTI-STAGE AIR COMPRESSION

4.1 Overview

Because compressed air systems (CAS) are highly inefficient and operating costs run high, a primary concern is cost reduction. The equation for specific work of a single-stage air compressor shows that the specific work is proportional to the temperature difference from the inlet to discharge condition. One can then infer that a smaller temperature difference due to a lower discharge temperature reduces specific work, and in theory, if the air could be cooled during compression, the specific work would be kept to a minimum. Cooling through the compressor walls has size and heat transfer limitations that are outside the scope of this study. Alternatively, a widely used approach for cooling is to partially compress, discharge, and cool the air as many times as necessary to reach a target discharge pressure. A cooler between the outlet of one compressor and the inlet of another is referred to as an intercooler, and an assembly comprised of a series of compressors (referred to as stages in this context) with intercooling between is a multi-stage compressor.

To a reasonable extent, the time it would take to reach a target discharge pressure, the size of a multi-stage compressor, and the cost of materials and manufacturing would outweigh the benefit of many compressor stages. Consequently, the industry standard for multi-stage air compression is the two-stage compressor, which ideally cools the air temperature to its initial temperature. This section explains the assumptions of two-stage air compressor modeling, includes a parametric study of specific work resulting from ranging inlet air temperature and discharge pressure, and evaluates the increase of the specific work relative to the lowest inlet air temperature and discharge pressure of the parametric study.

4.2 Assumptions

The two-stage compressor has three components: the first stage, the intercooler, and the second stage. Modeling the two-stage air compressor relies on four states, namely the conditions outside the first-stage inlet, the first-stage discharge which is also the intercooler inlet, the intercooler discharge, and the second-stage discharge. A diagram of a two-stage compressor is shown by Figure 8, with numbers designating the component associated with each state. It is assumed that both stages of the compressor are adiabatic, and that the intercooler cools the air discharged from the first stage to the temperature at the inlet of the first stage. The quasi-equilibrium compression processes are time-averaged to approximate the flow as steady, and the discharge pressure is far below the critical pressure of dry air, so dry air is assumed to be an ideal gas. Furthermore, the changes in kinetic and potential energy as well as the pressure drop are negligible.

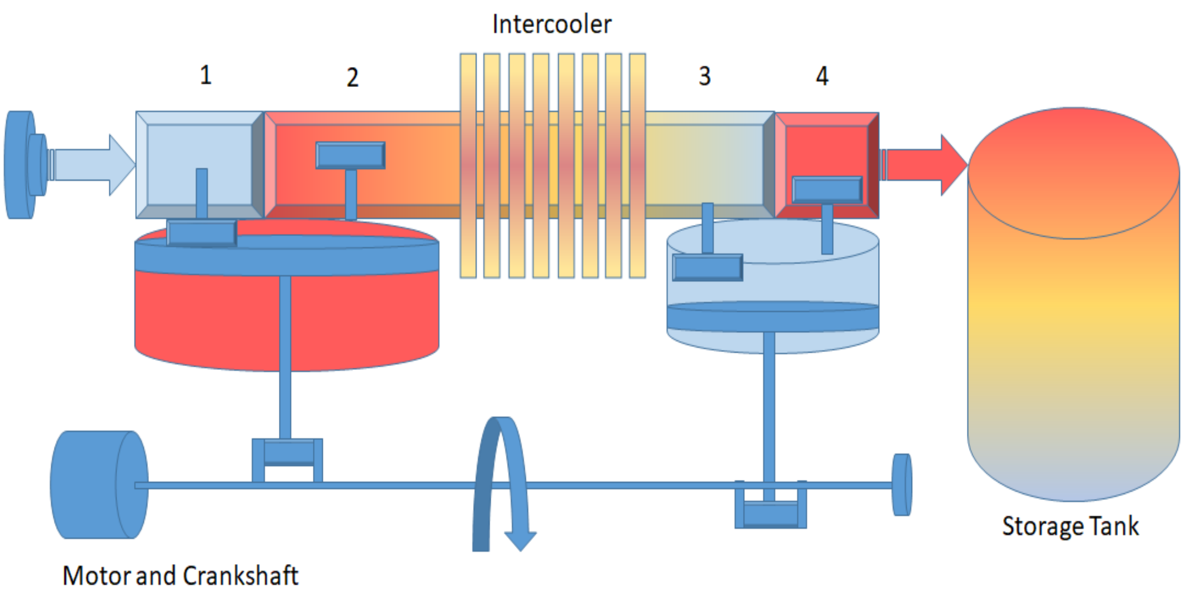

Figure 8: Reciprocating two-stage air compressor and storage tank

4.3 Two-Stage Model

The specific work of a two-stage air compressor results from the sum of the specific work of both stages. Like the model for the single-stage air compressor that was developed in the previous section, the specific work of a single stage is proportional to the isobaric specific heat multiplied by the temperature difference, and considering isentropic compression, it can be derived with the pressure ratio. A two-stage compressor has a temperature difference for each stage, where the first temperature difference is that between the inlet and discharge of the first stage, and the second is the temperature difference between the inlet and discharge of the second stage. However, the pressure of the air entering the second stage is above atmospheric pressure, and is the discharge pressure of the first stage, or the intermediate pressure. The process of intercooling occurs at constant pressure from the discharge of the first-stage to the inlet of the second-stage. Because the ideal intercooler cools the first-stage discharge air to the first-stage inlet temperature, both stages of an ideal two-stage compressor share the same specific heat ratio k and gas constant R. The key difference however, is that the first stage has an isentropic pressure ratio of an intermediate pressure to the inlet pressure, while the second stage has an isentropic pressure ratio of the desired discharge pressure to the intermediate pressure. Therefore, the specific work model for a two-stage compressor is as follows, where 0 denotes the inlet or initial state, I corresponds to the intermediate condition, and f denotes the discharge or final state.

$$w_{compressor} = \frac{kRT_0}{k-1} * \left[\left(\left(\frac{P_I}{P_0}\right)^{\frac{k-1}{k}} - 1\right) + \left(\left(\frac{P_f}{P_I}\right)^{\frac{k-1}{k}} - 1\right)\right], \qquad (Eq. 4.1)$$

The intermediate pressure that produces the minimum amount of work is the square root of the product of the inlet and discharge pressure. Proof can be shown by calculating the minima of the specific work when taking the derivative with respect to pressure and setting it equal to zero. By noting that the specific heat ratio, gas constant, and inlet temperature are independent of pressure, those terms can be eliminated from the derivative.

$$\frac{dw}{dP} = 0 = \frac{d}{dP}\left(\left(\frac{P_I}{P_0}\right)^{\frac{k-1}{k}}\right) + \frac{d}{dP}\left(\left(\frac{P_f}{P_I}\right)^{\frac{k-1}{k}}\right), \qquad (Eq. 4.2)$$

To eliminate confusion with the exponent, it is denoted by an arbitrary variable and substituted into the derivative.

$$c = \frac{k-1}{k}, \qquad (Eq. 4.3)$$

$$\frac{dw}{dP} = \frac{d}{dP}\left(\left(\frac{P_I}{P_0}\right)^c + \left(\frac{P_f}{P_I}\right)^c\right), \qquad (Eq. 4.4)$$

The inlet and discharge pressure are constant, and their derivative is equal to zero. Applying both the product rule and the power rule to the derivative of specific work with respect to pressure results in the following.

$$\frac{dw}{dP} = 0 = \left(\frac{1}{P_0}\right)^c c\, P_I^{c-1} + P_f^c\,(-c)\left(\frac{1}{P_I}\right)^{c-1}, \qquad (Eq. 4.5)$$

The negative term including the final and intermediate pressure can be subtracted from both sides of the expression to maintain equality, moving it from the right-hand side (RHS) of the equation to the left-hand side (LHS).

$$P_f^c\left(\frac{1}{P_I}\right)^{c+1} = \left(\frac{1}{P_0}\right)^c P_I^{c-1}, \qquad (Eq. 4.6)$$

By multiplying both sides of the equation with the terms in the denominators of both the LHS and RHS, the equation has the discharge and inlet pressure on the LHS, and the intermediate pressure on the RHS.

$$\left(P_f P_0\right)^c = (P_I)^{c+1+c-1}, \qquad (Eq. 4.7)$$

Using algebraic rules of exponents, the variable c can be eliminated from both sides, and the square root taken to have the intermediate pressure in terms of the square-root of the product of the discharge and inlet pressure.

$$\sqrt{P_f P_0} = P_I, \qquad (Eq. 4.8)$$

4.4 The Intermediate Pressure Resulting in Minimum Specific Work

Although the minimum specific work results from an intermediate pressure that is the square root of the discharge and inlet pressure, it is important to confirm the result with another potential value such as the average pressure. Two-stage compressor specific work is proportional to discharge temperature. Thus, the discharge temperature of the first stage of the two-stage compressor using the intermediate pressure is plotted in Figure 9.

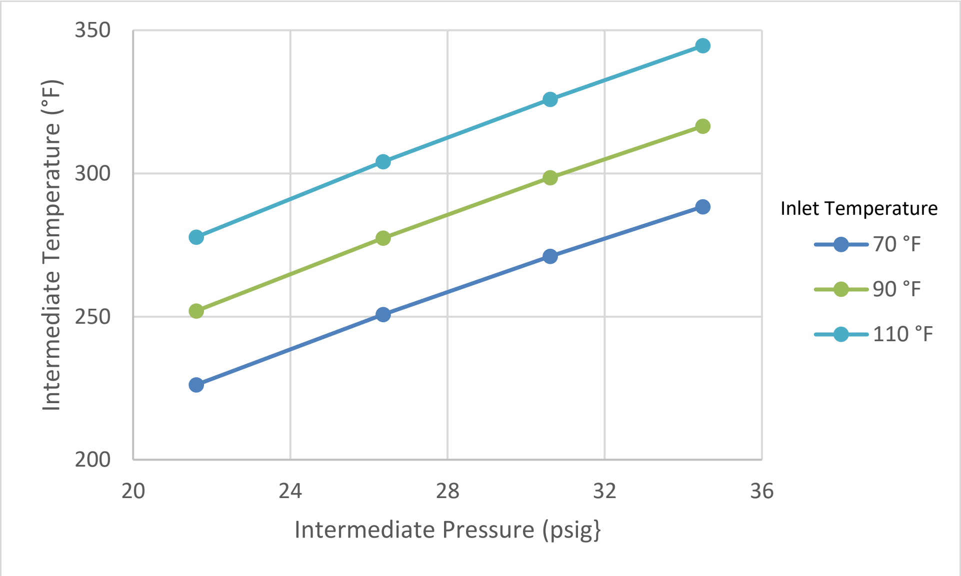

Figure 9: Discharge temperature of the first stage of a two-stage compressor discharging to the optimum intermediate pressure

The discharge temperature of the second stage of the two-stage compressor using the optimum intermediate pressure has the same values as Figure 9. Using the optimum intermediate pressure in the two-stage compressor model, the outlet temperature of both stages is equal. The reason is that the discharge temperature of an isentropic compressor depends on the pressure ratio, and the optimum intermediate pressure results in the same pressure ratio for the first and second stage of the compressor. To compare, an intermediate pressure equal to the average of the inlet and discharge pressure results in different discharge temperatures for each stage, which is shown in Figure 10 and Figure 11.

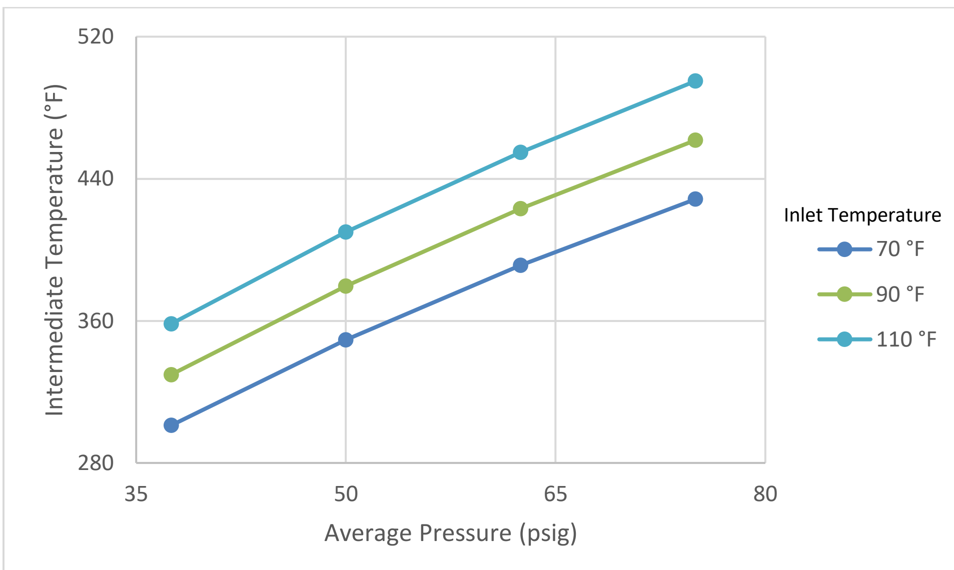

Figure 10: Discharge temperature of the first stage of a two-stage air compressor discharging to the average of the inlet and desired discharge pressure

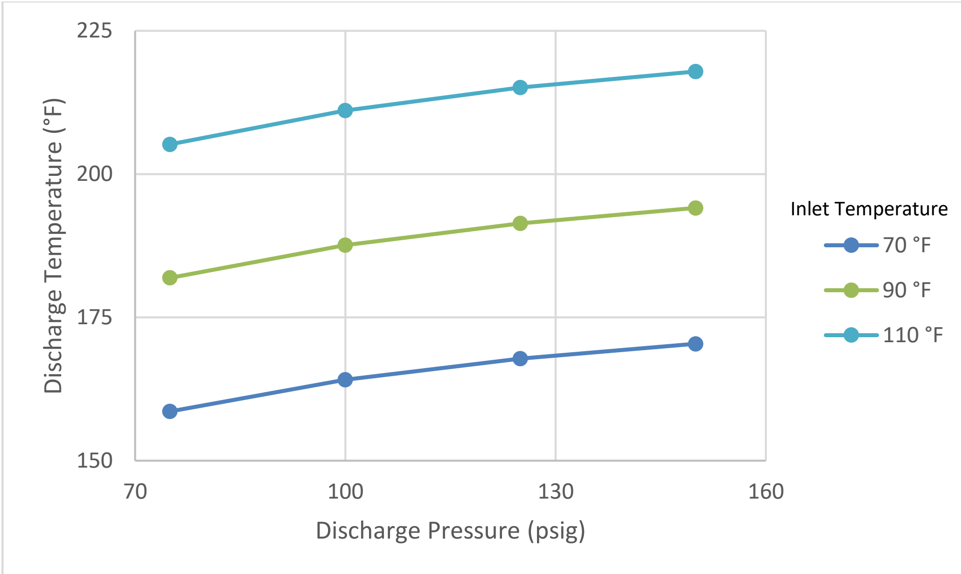

Figure 11: Discharge temperature of the second stage of a two-stage air compressor drawing air at a pressure equal to the average of the desired discharge pressure and inlet pressure, and discharging to the desired discharge pressure

Figure 10 and Figure 11 show that by using the average of the desired discharge pressure and inlet pressure, the discharge temperature of the first stage is greater than that resulting from the optimum intermediate pressure, and the discharge temperature of the second stage is less. Again, this is due to the relationship between temperature and pressure for an isentropic compression process. The arithmetic mean is independent of the pressure ratio, so although the average is the midpoint of the desired discharge pressure and inlet pressure, the resulting first stage pressure ratio is greater than the second. The specific work of compression is directly proportional to the absolute temperature rise, so it is important to observe the change in total specific work of a two-stage compressor by incrementing the intermediate pressure from the inlet pressure to the discharge pressure. For example, to compress air from 14.7 psia to 80 psia, the optimum intermediate pressure is 34 psia, while the average is 47 psia, and Figure 12 shows the specific work of each stage of a two-stage compressor as the intermediate pressure is incremented from 14.7 psia to 80 psia.

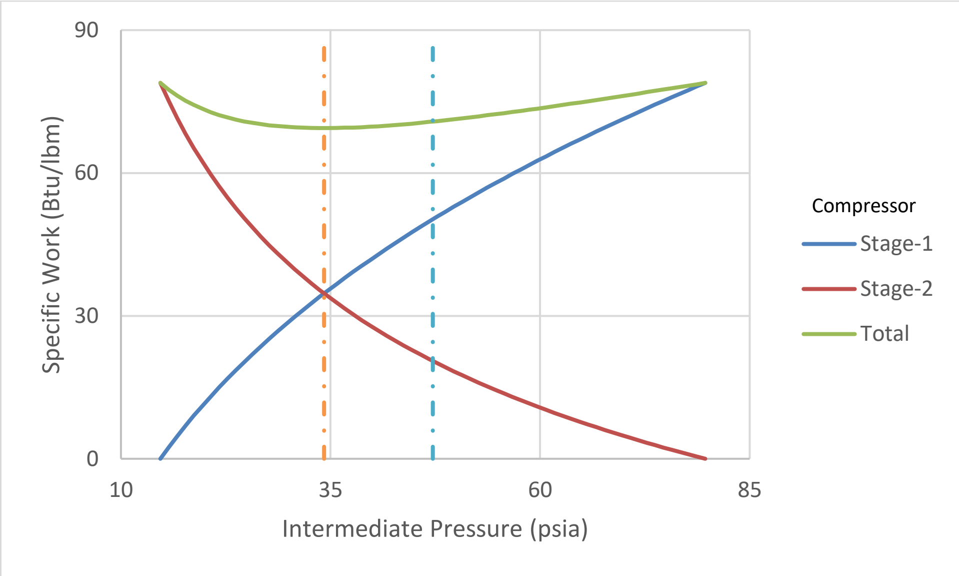

Figure 12: Incremented intermediate pressure applied to both stages of a two-stage air compressor to find the minimum total work

Figure 12 shows the specific work of each compressor stage in a two-stage assembly when the intermediate pressure is incremented from the inlet pressure to desired discharge pressure. If the intermediate pressure is the equal to the inlet pressure, it is as though the first stage has not performed any compression and the second stage performs all the compression. Conversely, if the intermediate pressure is equal to the discharge pressure, it is as though the first stage performs all the compression and the second stage does not do any compression. The contour corresponding to the total of both stages has a minima at the intermediate pressure that is the square-root of the inlet and discharge pressures. The average pressure is the midpoint of the inlet and desired discharge pressures and results in more total work than the minimum. The specific work of the two-stage air compressor can also be shown on a Pv diagram as the area from y-axis to the path line. The paths in Figure 13 and Figure 14 show that the temperature increases in the first stage, cools to the initial temperature, and increases through the second stage.

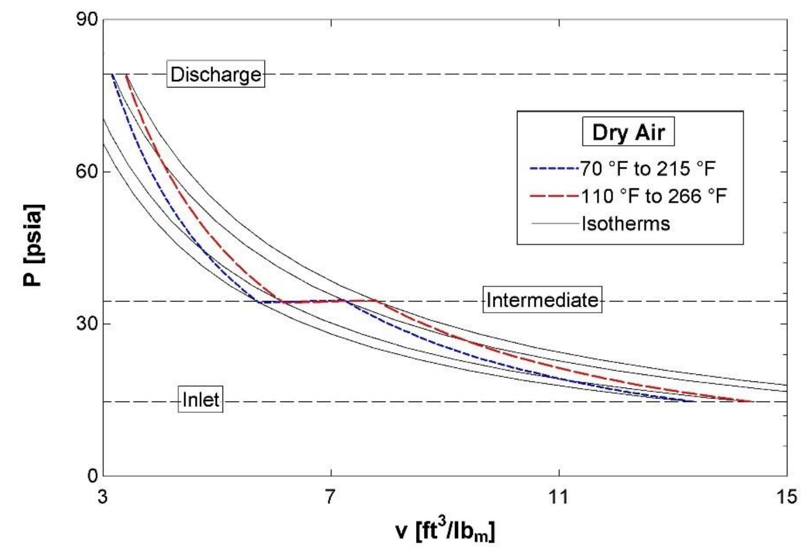

Figure 13: Pv diagram of two-stage compression of air entering at 14.7 psia and 70 °F versus 110 °F to discharge pressure of 80 psia by the optimum intermediate pressure

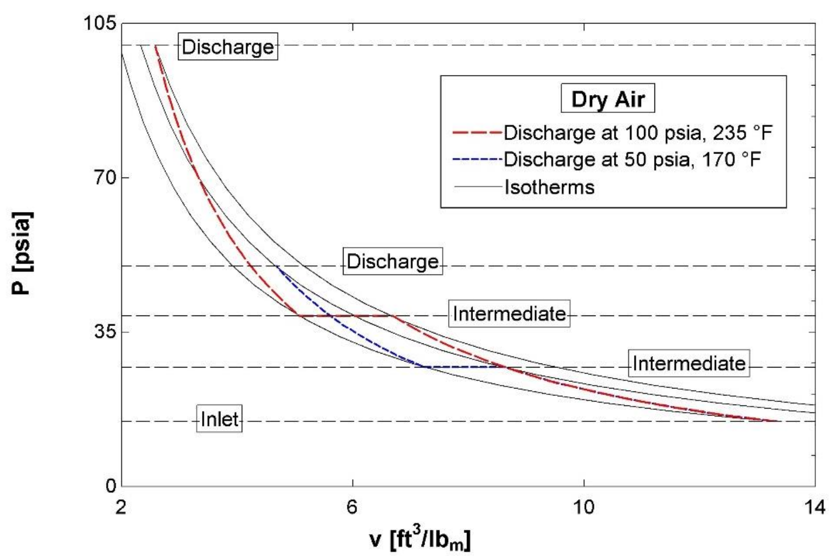

Figure 14: Pv diagram of two-stage compression of air entering at 14.7 psia and 70 °F to discharge pressures of 50 versus 100 psia by way of the optimum intermediate pressure

The Pv diagram in Figure 13 shows two different compression paths of air both initially at atmospheric pressure but with one beginning at 70 ˚F and the other at 110 ˚F. While the path line of the compression of air initially at 110 ˚F encloses more area between the y-axis than the path line of compression of air initially at 70 ˚F, the overall amount does not appear significant. On the other hand, the Pv diagram shown in Figure 14 shows a significant increase in area enclosed by the path line of two-stage compression of air to 100 psia when compared to 50 psia with both at the same initial condition. Therefore, discharge pressure has a significantly larger effect on the specific work of a two-stage compressor than does inlet air temperature.

4.5 Comparing the Optimum and Average Intermediate Pressure

Although a two-stage compressor using the square root of the desired discharge pressure and inlet pressure has equal discharge temperatures for both stages, and the discharge temperature of each stage of a two-stage compressor set to the average pressure varies, it is not clear from the Pv diagrams as to how much more specific work is required when using the average pressure. A parametric study of specific work of two-stage compression for air temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig is conducted for both the optimum intermediate pressure and the average pressure. The total specific work resulting from the optimum intermediate pressure and average pressure are plotted in Figure 15 and Figure 16, respectively. Figure 17 shows the percent reduction in specific work when referencing the path utilizing an average pressure for the intermediate pressure.

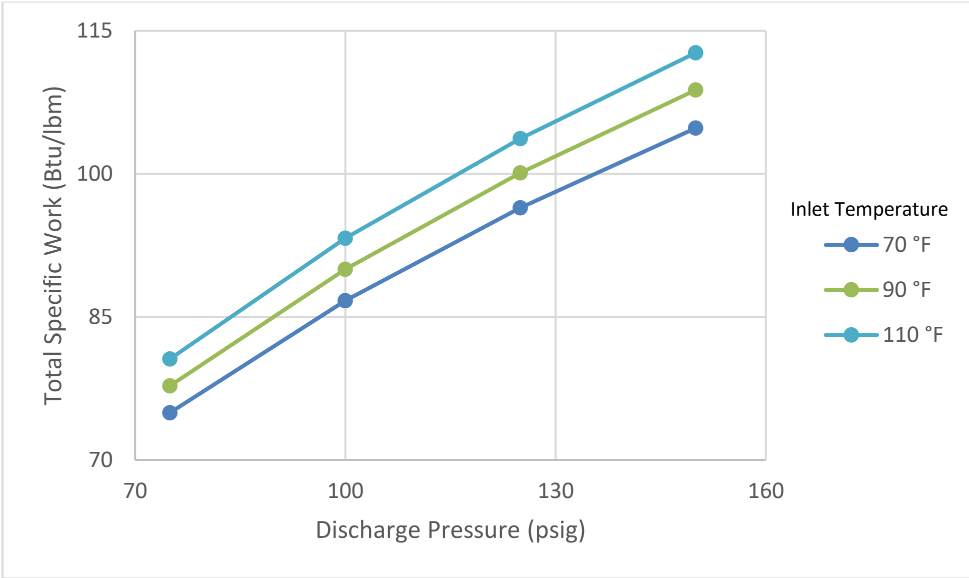

Figure 15: Specific work of a two-stage compressor using the optimum intermediate pressure

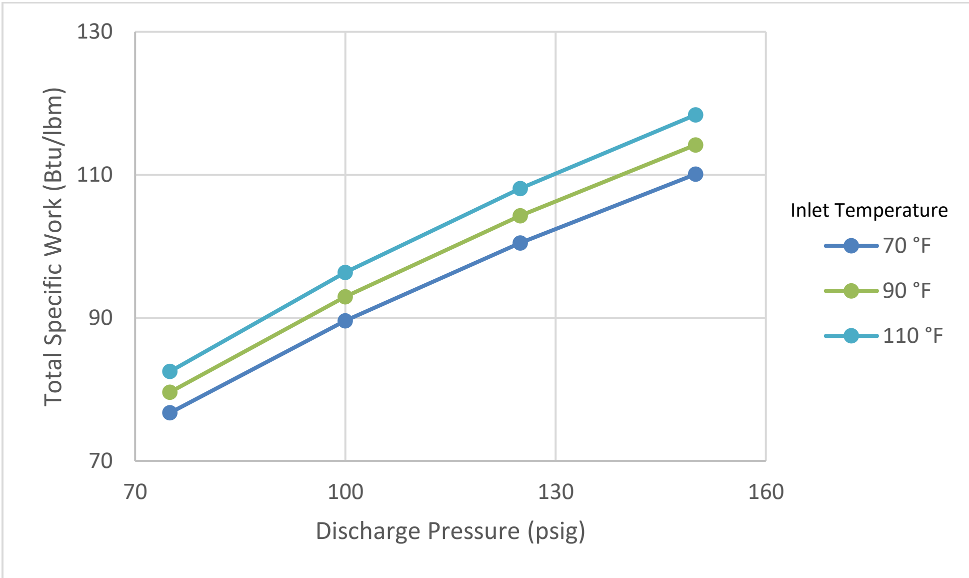

Figure 16: Specific work of a two-stage compressor using the average intermediate pressure

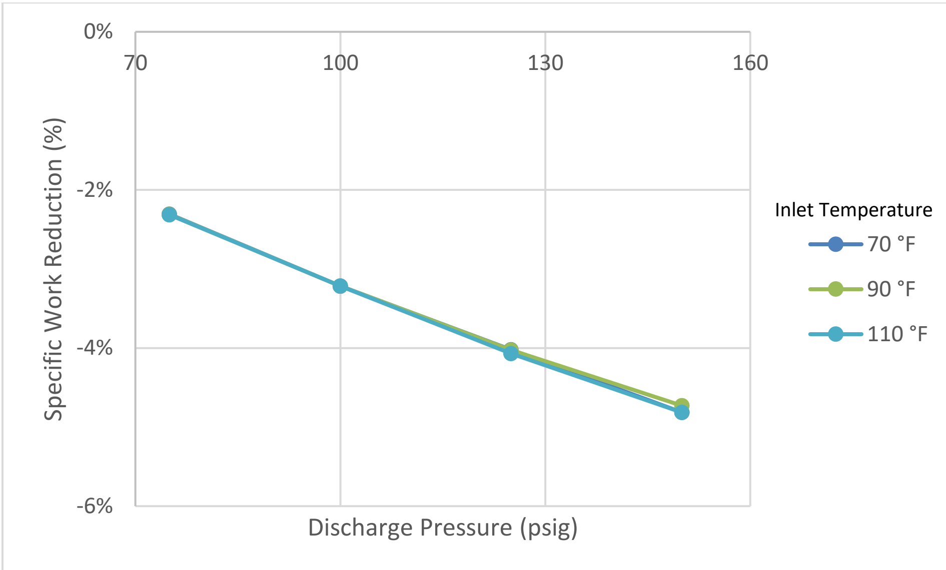

Figure 17: Percent specific work decrease using the optimum intermediate pressure instead of the average intermediate pressure

Figure 15 indicates that a two-stage air compressor utilizing the optimum intermediate pressure requires specific work ranging from 70 to 115 Btu/lbm. The specific work of a twostage air compressor utilizing the average of the desired discharge pressure and inlet pressure for its intermediate pressure is shown by Figure 16, and ranges from 70 to 125 Btu/lbm. By subtracting the optimum intermediate pressure results from the average intermediate pressure results and dividing with respect to the average intermediate pressure results, Figure 17 shows that the optimum intermediate pressure reduces the specific work consumption by 2-6% depending on the discharge pressure.

4.6 Specific Work Increase Relative to Pressure, Temperature, and Single-Stage Compression

A two-stage compressor with intercooling requires less specific work than a single-stage compressor when utilizing the optimum intermediate pressure. Within the parametric study ranges, the single-stage compressor requires up to 10% additional specific work due to temperature, and up to 45% additional specific work due to pressure. Intercooling allows the two-stage compressor to consume less specific work, and the difference due to temperature and pressure are indicated by Figure 18 and Figure 19, respectively. Compared to the single-stage compressor, the two-stage compressor also requires up to 10% extra specific work from an inlet temperature of 70 to 110 °F, but only 40% extra specific work from a discharge pressure of 100 to 150 psig. Furthermore, the two-stage compressor consumes less specific work and the amount relative to the single-stage compressor is indicated by Figure 20. Regardless of the inlet temperature, two-stage dry air compression consumes 10-20% less than the single-stage compressor.

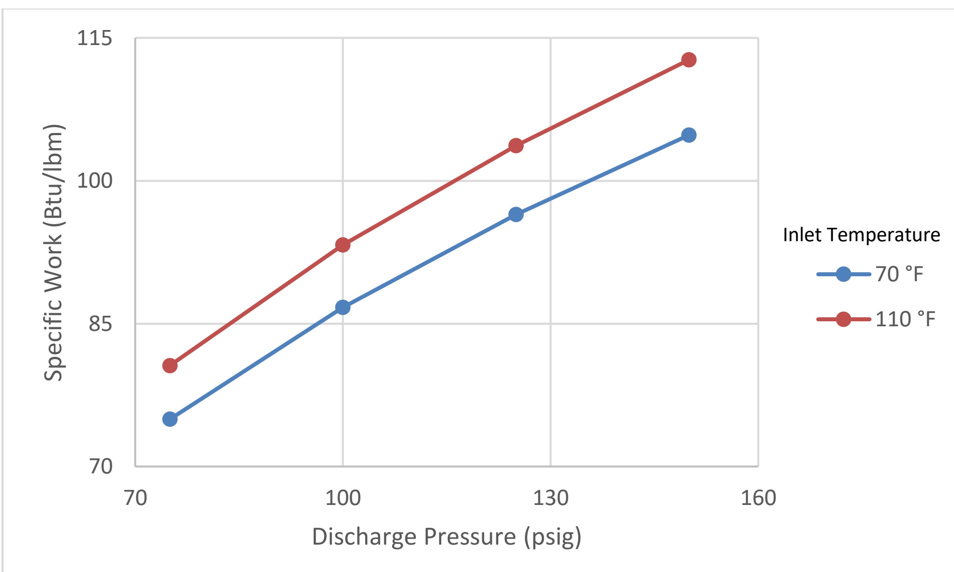

Figure 18: Specific work of a two-stage air compressor with air entering at an atmospheric pressure of 14.7 psia and a temperature of 70 °F versus 110 °F compressing to a range of discharge pressure from 75 to 150 psig

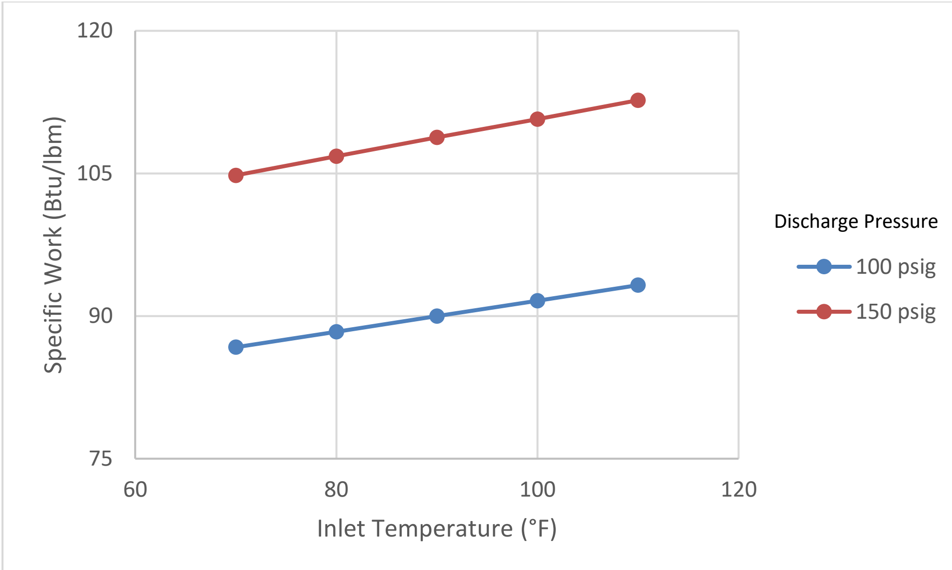

Figure 19: Specific work of a two-stage air compressor with air entering at an atmospheric pressure of 14.7 psia and a temperature ranging from 70 to 110 ˚F then discharged to 100 versus 150 psig

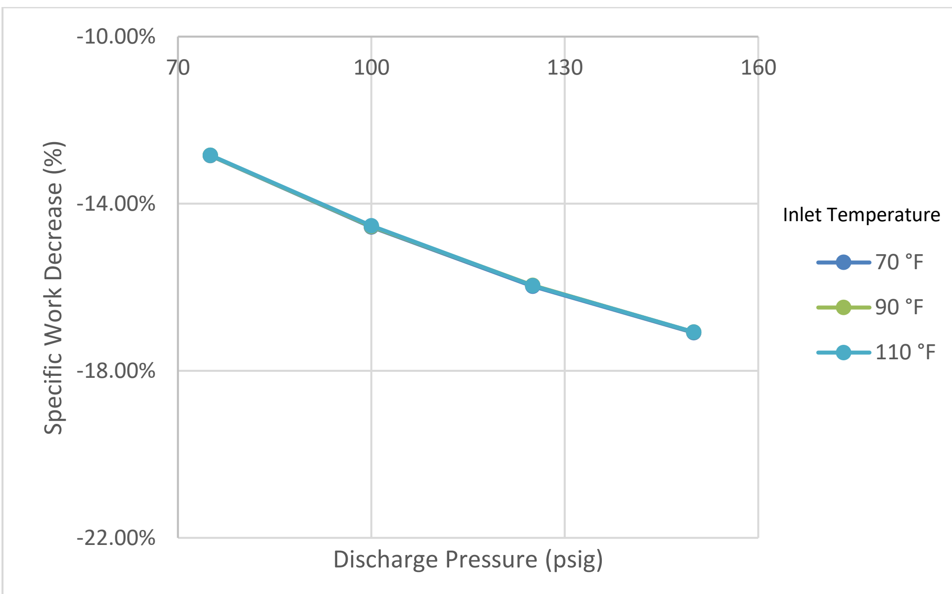

Figure 20: Specific work reduction when comparing the specific work of a two-stage air compressor to a single-stage air compressor with air entering at 14.7 psia and a temperature ranging from 70 to 110 ˚F discharged to a range of 75 to 150 psig

Figure 18 shows that the specific work of a two-stage air compressor at a discharge pressure of 75 psig increases from 75 to 80 Btu/lbm when the temperature of the air entering the compressor rises from 70 to 110 ˚F. An inlet air temperature increase of 40 ˚F increases the specific work of a two-stage compressor by less than 10%. While the specific work of a singlestage air compressor increases by 45% when comparing air at the same inlet condition discharging to 100 versus 150 psig, Figure 19 shows that the specific work of a two-stage compressor increases by 40%. The benefit of intercooling is demonstrated by Figure 20, which shows that a two-stage compressor consumes 10-20% less specific work than the single-stage model for the same range of inlet air temperature and discharge pressures.

5. ATMOSPHERIC AIR COMPRESSION

5.1 Overview

Atmospheric air (air) is a mixture of dry air and water vapor, and its composition varies by prevailing weather and other factors. The composition of air has a substantial effect on compressed air systems (CAS) because condensate can form in the two-stage compressor's intercooler, where it can mix with the piston lubricating oil entrained in the air, or in air distribution lines, possibly inhibiting performance and causing damage. The following subsection reviews the properties of dry air, water vapor, and air to quantify moisture and understand its effect on CAS. The remainder of the chapter shares the results of air compression to compare to the ideal baselines described in earlier sections and to identify opportunities for improvement.

5.2 Properties of Water Vapor, Dry Air, and Atmospheric Air

Dry air is composed of 78% nitrogen, 21% oxygen, 0.93% argon, and 0.04% carbon dioxide (Sharp, 2017). Considering each component's molar mass and their mole percentages, the molar mass of dry air is approximately 28.97 lbm/lbmol. On the other hand, water vapor has a molar mass of 18.02 lbm/lbmol. Because air is a mixture, the mass fraction of each substance is necessary to determine its molar mass, which depends on the amount of airborne moisture. Thus, it is necessary to introduce a way to measure the mass of water vapor in air.

Specific humidity is the mass-ratio of water vapor to dry air derived under the ideal gas approximation and Dalton's law of partial pressures. The critical pressure of dry air and water vapor are 547 psia and 3204 psia, respectively (Çengel & Turner, 2004). The parametric study evaluates pressures well below the critical pressures of the substances of interest. Therefore, regardless of temperature, both dry air and water vapor are approximated as ideal gases subject to the ideal gas equation of state.

$$Pv = RT, \qquad PV = mRT = \frac{m\bar{R}T}{M} = N\bar{R}T, \qquad (Eq. 5.1)$$

Using the ideal gas equation of state to solve for the mass of each component in terms of its pressure, volume, temperature, and gas constant, the derivation of specific humidity is as follows.

$$\omega = \frac{m_{wv}}{m_{da}} = \frac{\frac{P_{wv}V_{wv}}{R_{wv}\,T_{wv}}}{\frac{P_{da}V_{da}}{R_{da}T_{da}}} \quad (Eq. 5.2)$$

As a mixture in air, water vapor and dry air are in thermal equilibrium and occupy the same space, so the temperature and volume of each component are equal in magnitude and may be eliminated from the expression.

$$\omega = \frac{\frac{P_{wv}}{R_{wv}}}{\frac{P_{da}}{R_{da}}} \quad (Eq. 5.3)$$

A substance's gas constant is equal to the ratio of the universal gas constant to the substance's respective molar mass. Expressing each gas constant in terms of the universal gas constant appears in both the numerator and denominator and can be eliminated.

$$\omega = \frac{\frac{P_{wv}}{\bar{R}}M_{wv}}{\frac{P_{da}}{\bar{R}}M_{da}} = \frac{P_{wv}\,M_{wv}}{P_{da}\,M_{da}} = \frac{18.02}{28.97}\frac{P_{wv}}{P_{da}} = 0.622\frac{P_{wv}}{P_{da}}, \qquad (Eq. 5.4)$$

Under the ideal gas assumption, Dalton's law of partial pressures states explains that the total pressure of is equal to the sum of partial pressures. Thus, the total pressure of air is equal to the sum of the partial pressure of dry air and the partial pressure of water vapor.

$$P_{total} = \Sigma P_{partial}, \qquad (Eq. 5.5)$$

$$P_{total} = P_{da} + P_{wv}, \qquad (Eq. 5.6)$$

Because the total pressure is equal the sum of the partial pressures, the partial pressure of dry air is equal to the partial vapor pressure subtracted from the total pressure.

$$P_{da} = P_{total} - P_{wv}, \qquad (Eq.\ 5.7)$$

$$\omega = 0.622 \frac{P_{wv}}{P_{total} - P_{wv}}, \qquad (Eq. 5.8)$$

If the relative humidity is known, the partial vapor pressure can be calculated. Relative humidity is the ratio of partial vapor pressure to the saturated vapor pressure of water at a given dry bulb temperature, which is the sensible temperature most are familiar with.

$$\phi = \frac{P_{wv}}{P_g} = \frac{P_{wv}}{P_{sat}(T_{dry\ bulb})}, \qquad (Eq. 5.9)$$

Thus, the partial vapor pressure is the product of the relative humidity and the saturated vapor pressure of water at the given dry bulb temperature

$$P_{wv} = \phi\, P_g = \phi\, P_{sat}(T_{dry\ bulb}), \qquad (Eq. 5.10)$$

Finally, the specific humidity of atmospheric air is as follows.

$$\omega = 0.622 \frac{\phi\, P_g}{P_{total} - \phi\, P_g}, \qquad (Eq. 5.11)$$

Specific humidity quantifies airborne moisture and plays an important role in calculating the specific work of moist air compression. The model for specific work of moist air compression requires the gas constant of atmospheric air, which requires the molar mass of atmospheric air. Because the amount of moisture in the air varies, it is necessary to calculate the molar mass of atmospheric air in terms of specific humidity. To begin, the molar mass of a substance, its mass, and its current number of moles are related in the following way.

$$M = \frac{m}{N}\ , \qquad N = \frac{m}{M}\ , \qquad (Eq. 5.12)$$

Therefore, the molar mass of a mixture is equal to the mass of the mixture divided by the number of moles of the mixture. The number of moles of a mixture is the sum of the number of moles of each component, which for each component, is the mass of the component divided by the molar mass of the component.

$$M_{mix} = \frac{m_{mix}}{N_{mix}} = \frac{m_{mix}}{\sum_{i=1}^{n} N_i} = \frac{m_{mix}}{\sum_{i=1}^{n} \frac{m_i}{M_i}}, \qquad (Eq. 5.13)$$

Dividing the right-hand side (RHS) of the expression by the mass of the mixture in both the numerator and denominator maintains equality in the expression.

$$M_{mix} = \frac{1}{\sum_{i=1}^{n} \frac{\mathrm{m_i}}{\mathrm{m_{mix}}}\ M_i}, \qquad (Eq. 5.14)$$

Because the only two components are dry air and water vapor, and the mass of the mixture is the sum of the mass of each component, the molar mass of moist air is as follows.

$$M_{air} = \frac{1}{\frac{m_{da}}{(m_{da} + m_{wv})\ M_{da}} + \frac{m_{wv}}{(m_{da} + m_{wv})\ M_{wv}}}, \qquad (Eq. 5.15)$$

Because water vapor content varies, the expression is best represented with the specific humidity by multiplying the numerator and denominator of the RHS by the mass of the mixture, then dividing all terms by the mass of the dry air.

$$M_{air} = \frac{m_{da} + m_{wv}}{\frac{m_{da}}{M_{da}} + \frac{m_{wv}}{M_{wv}}} = \frac{1 + \frac{m_{wv}}{m_{da}}}{\frac{1}{M_{da}} + \frac{m_{wv}}{m_{da}\ M_{wv}}} = \frac{1 + \omega}{\frac{1}{M_{da}} + \frac{\omega}{M_{wv}}}, \qquad (Eq. 5.16)$$

Lastly, the gas constant of atmospheric air is the ratio of the universal gas constant to the air’s molar mass.

$$R_{air} = \frac{\bar{R}}{M_{air}} = \frac{\bar{R}\left(\frac{1}{M_{da}} + \frac{\omega}{M_{wv}}\right)}{1 + \omega}, \qquad (Eq. 5.17)$$

With the gas constant of the air and assuming the specific heats of air are constant over the prescribed temperature range the specific energy of air compression can be calculated.

5.3 Compressed Air Systems and Moisture

Moisture in the intercooler of a multi-stage air compressor and downstream equipment is detrimental when it mixes with airborne particulates such as lubricating oil or dirt. In the intercooler, condensate can mix with the lubricating oil entrained in the air from the piston and damage the compressor. Further downstream, the condensate mixes with particulates, coagulates, and forms scale on the inner surface of equipment, which increases resistance to airflow, reduces discharge pressure, and ultimately leads to corrosion. Corrosion further compromises system efficiency by forming openings that leak compressed air and further reduce discharge pressure. A short-term solution is to increase discharge pressure to compensate for losses, but in that case, the compressor requires more specific work. Furthermore, long-term corrosion can result in total equipment failure.

Condensate forms if the compressed air cools to the saturation temperature of the partial vapor pressure (dew point), which depends on the total air pressure and the relative humidity. Recall that relative humidity is the ratio of the partial vapor pressure to the saturated vapor pressure of water at the given dry bulb temperature. At 100% relative humidity, the partial vapor pressure in air is equal to the saturation pressure of water vapor. When saturated water vapor is cooled, the water changes phase to liquid (condenses) at constant temperature. The temperature at which the partial vapor pressure begins to condense is the dew point temperature.

$$T_{dew} = T_{sat}(P_v) = T_{sat}(\phi\, P_g), \qquad (Eq. 5.18)$$

A dew point deviating from atmospheric pressure is called a pressure dew point. The intermediate and discharge pressures have elevated dew points because compressed fluids are more concentrated than usual and require more heat to overcome intermolecular forces. The dew point temperature at the range of discharge pressures in this study is greater than the room temperature, so the walls of air distribution lines, which are at room temperature, transfer heat from the relatively hot compressed air to the surrounding air and condense airborne water vapor on the inner surface of the wall.

In the two-stage air compressor, sufficient relative humidity results in an intermediate pressure dew point greater than the inlet temperature. In that case, intercooling below the intermediate dew point forms condensate and can damage the compressor as mentioned earlier, so it is wise to buffer the intercooling above the intermediate dew point. However, if the safetybuffer results in a temperature greater than the inlet temperature, then the second stage requires more specific work for compression than would the ideal case. Therefore, it is necessary to determine the intermediate dew point and quantify its impact on the specific work of two-stage air compression.

5.4 Pressure Dew Point: Saturation Temperature of Compressed Water Vapor

The traditional method of determining the dew point of compressed air, or the pressure dew point, involves the absolute pressure ratio and grains of moisture chart. A grains of moisture chart lists the grains of moisture corresponding to a combination of dry bulb temperature and relative humidity. The product of the absolute pressure ratio and grains results in the quantity of grains present in the compressed air. This is because the absolute pressure ratio determines the volume of air compressed into a single unit of volume, and the moisture per unit volume is also compressed. After determining the number of compressed grains, the temperature on the chart corresponding 100% relative humidity is the pressure dew point (Johns, 1996).

Because there are 7000 grains of moisture per pound-mass of water vapor, specific humidity can be used to determine the pressure dew point. The atmospheric specific humidity requires the partial vapor pressure at atmospheric pressure, which depends on the relative humidity and saturation pressure at the given dry bulb temperature.

$$\omega_{atm} = 0.622 \frac{P_{wv_{atm}}}{P_{atm} - P_{wv_{atm}}}, \qquad (Eq. 5.19)$$

When the air is compressed from atmospheric pressure, it retains the same mass-ratio of moisture to dry air but exists at a greater total pressure. Thus, the partial vapor pressure at any total pressure can be expressed in terms of the atmospheric specific humidity and total pressure.

$$P_{wv} = \frac{P_{total}}{\frac{0.622}{\omega_{atm}} + 1}, \qquad (Eq. 5.20)$$

The expression can be simplified by substituting the equation of the atmospheric specific humidity is into the first term of the denominator; simplification of the first term follows.

$$\frac{0.622}{\omega_{atm}} = \frac{0.622}{0.622 \frac{P_{wv_{atm}}}{P_{atm} - P_{wv_{atm}}}} = \frac{P_{atm} - P_{wv_{atm}}}{P_{wv_{atm}}}, \qquad (Eq. 5.21)$$

Implementing the simplified first term into the expression for compressed partial vapor pressure and distributing the terms algebraically results in the following.

$$P_{wv} = \frac{P_{total}}{\frac{P_{atm} - P_{wv_{atm}}}{P_{wv_{atm}}} + 1} = \frac{P_{total}}{\frac{P_{atm} - P_{wv_{atm}} + P_{wv_{atm}}}{P_{wv_{atm}}}} = \frac{P_{total}}{\frac{P_{atm}}{P_{wv_{atm}}}}, \qquad (Eq. 5.22)$$

By multiplying the numerator and denominator by the atmospheric partial vapor pressure, the compressed partial vapor pressure at any total pressure is the ratio of the total pressure to the atmospheric pressure multiplied with the atmospheric vapor pressure.

$$P_{wv} = \frac{P_{total}}{P_{atm}} P_{wv_{atm}} = \frac{P_{total}}{P_{atm}} \phi P_g\left(T_{dry\ bulb}\right), \qquad (Eq. 5.23)$$

The pressure dew point is the saturation temperature of the compressed partial vapor pressure.

$$T_{dew_{pressure}} = T_{sat}\left(\frac{P_{total}}{P_{atm}} * P_{wv_{atm}}\right), \qquad (Eq. 5.24)$$

5.5 A Parametric Study of Pressure Dew Point

Atmospheric pressure is equal to zero on the gauge scale and 14.7 psia on the absolute scale. The following explanations are made using the absolute scale because the intermediate pressure is the square root of the inlet and discharge pressure in the absolute scale. At a discharge pressure of 114.7 psia (100 psig), the intermediate pressure is 41.06 psia. The product of the partial vapor pressure at atmospheric pressure with the pressure ratios corresponding to the intermediate and discharge pressures provide the intermediate and discharge partial vapor pressures that can be used for the pressure dew points. The increase in dew point temperature due to the increase in total pressure with initial dry bulb temperatures ranging from 70 to 110 °F and relative humidity at 30, 60 and 90%, are shown by Figure 21, Figure 22, and Figure 23, respectively. The two-stage air compressor requires additional specific work if the for intermediate pressure dew point is greater than the corresponding initial ambient temperature.

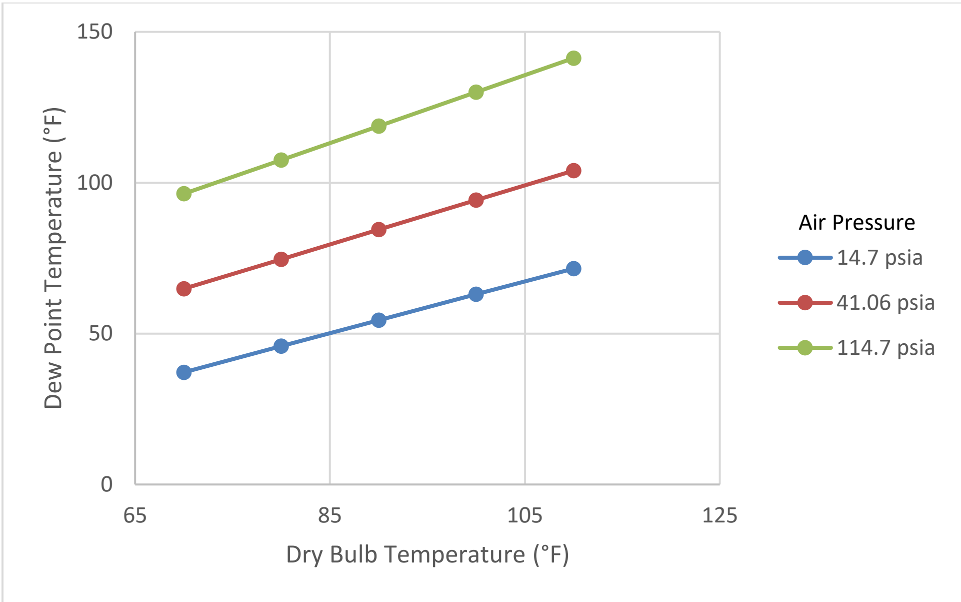

Figure 21: The atmospheric, intermediate, and discharge dew point of air at 30% relative humidity with dry bulb temperature ranging from 70 to 110 °F

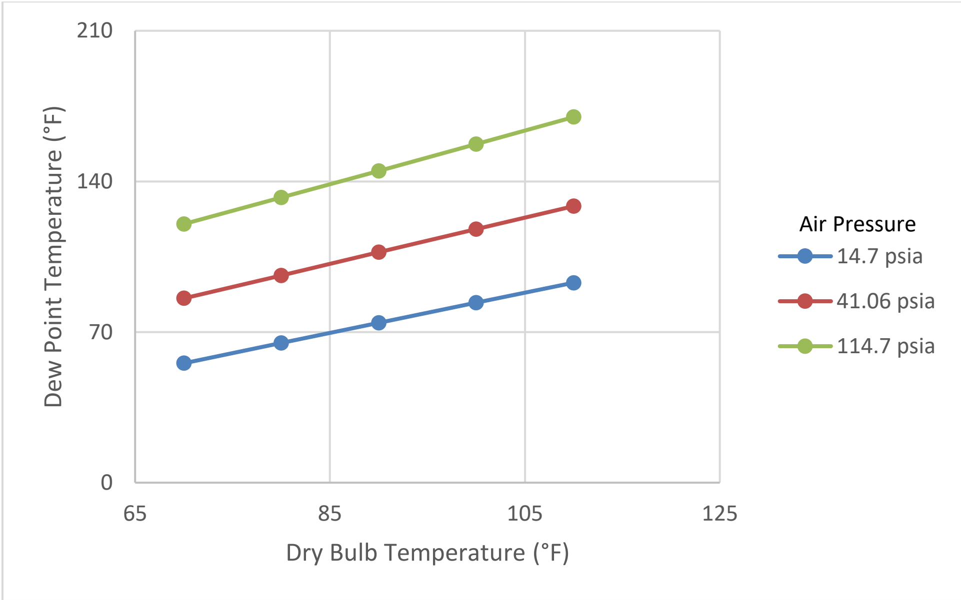

Figure 22: The atmospheric, intermediate, and discharge dew point of air at 60% relative humidity with dry bulb temperature ranging from 70 to 110 °F

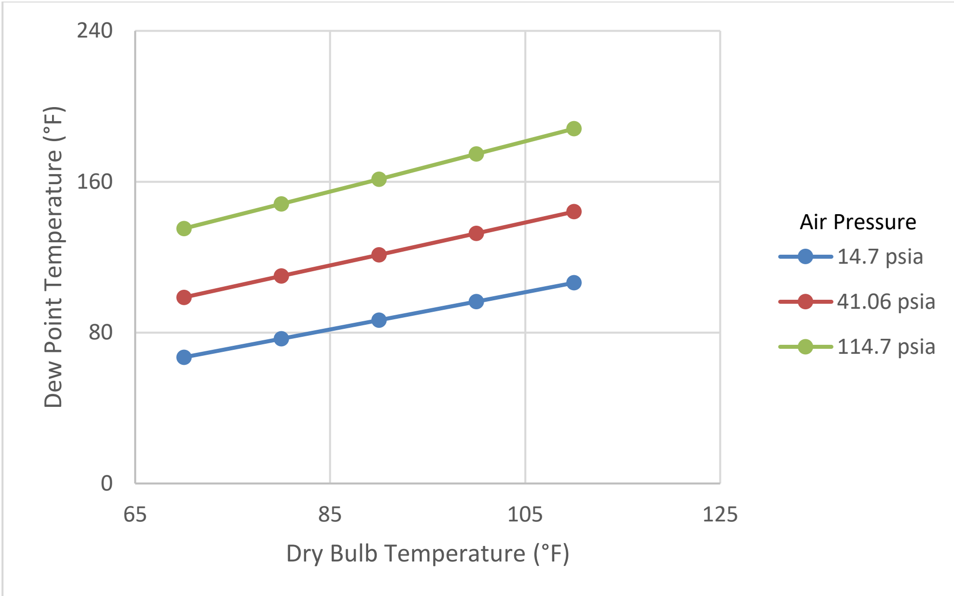

Figure 23: The atmospheric, intermediate, and discharge dew point of air at 90% relative humidity with dry bulb temperature ranging from 70 to 110 °F

At 30% relative humidity, the intermediate pressure dew points are less than their corresponding initial dry bulb temperature, so the intercooler can restore the initial air temperature without condensing water vapor. At precisely 38.1% relative humidity, the intermediate pressure dew point of 100 psig begins to exceed the initial dry bulb temperature. Compressor discharge dry bulb and dew point temperatures are extremely high compared to the temperature of the surrounding air. Therefore, the moment that air exits the compressor, it transfers heat with the air distribution lines until reaching thermal equilibrium with the surrounding air to a temperature below the discharge dew point, resulting in condensate. For that reason, many equip the system with aftercoolers and dryers, which are outside the scope of this study. In the section that follows, the processes during the compression path which result in condensation will be expressed in the form of pressure versus specific volume (Pv) plots of water vapor. Although the dry air Pv plots do not indicate any phase change, the water vapor Pv plots do when the water vapor reaches the saturation temperature of the partial vapor pressure.

5.6 Pressure Versus Specific-Volume of Water Vapor in Moist Air Compression

The potential transition of water vapor from a gas to a liquid during air compression can be shown by the water vapor Pv plots. Recall that the partial vapor pressure of compressed air is equal to the product of the atmospheric partial vapor pressure and the pressure ratio with respect to atmospheric total pressure. It is instructive to first introduce what happens to water vapor during the single-stage compression and discharge of air. From 14.7 psia to a discharge pressure of 80 psia, the path of water vapor during single-stage compression of air entering at 70 °F and 60% relative humidity, and 110 °F and 90% relative humidity, are shown by Figure 24 and Figure 25, respectively.

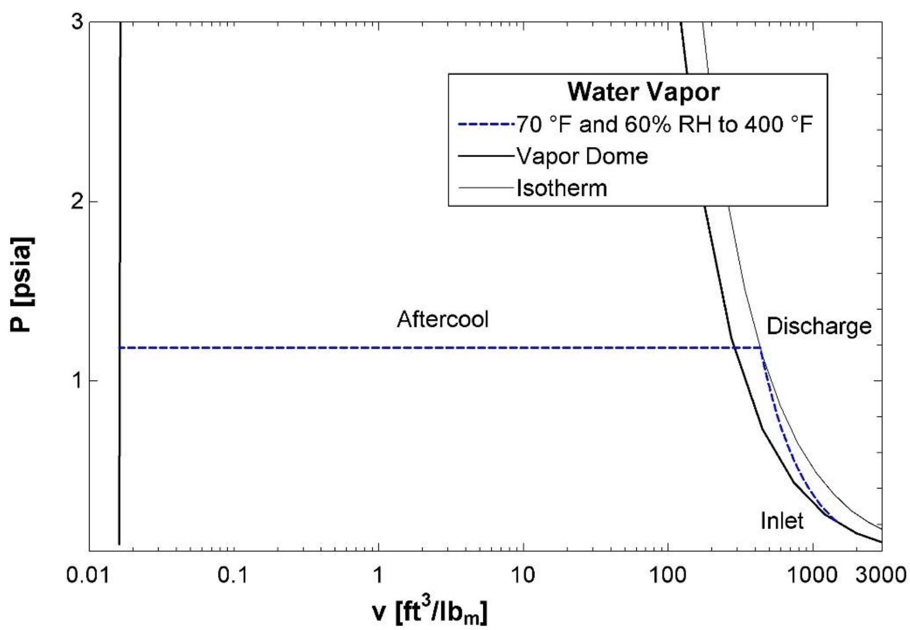

Figure 24: Path of water vapor during single-stage compression of air entering at 70 °F and 60% relative humidity

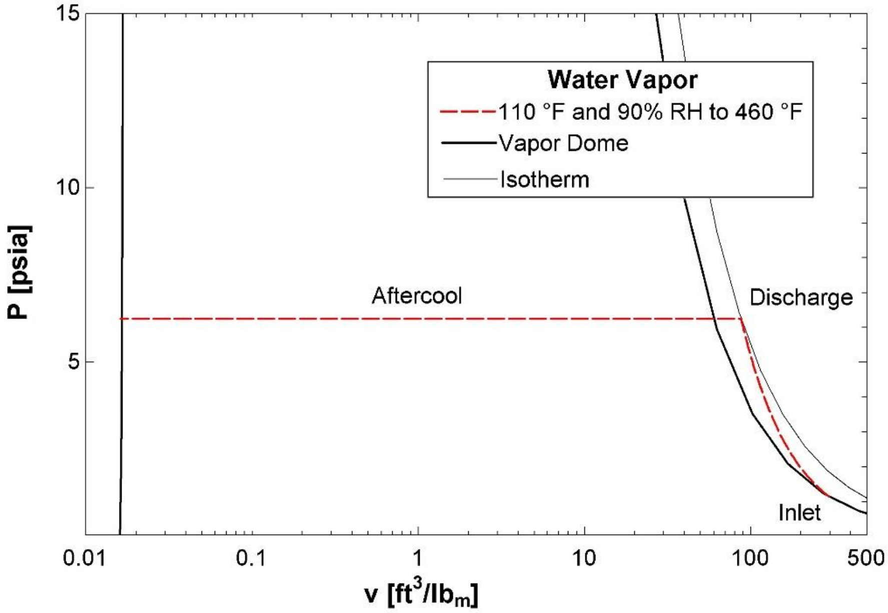

Figure 25: Path of water vapor during single-stage compression of air entering at 110 °F and 90% relative humidity from 14.7 psia discharged to 80 psia

Figure 24 shows that the partial vapor pressure of air at a temperature of 70 °F, relative humidity of 60%, and atmospheric pressure is equal to 0.218 psia, but at a total discharge pressure of 80 psia it changes to 1.187 psia, then rejects heat at constant pressure until reaching thermal equilibrium with the surroundings. Figure 25 shows that the partial vapor pressure at 110 °F, 90% relative humidity, and atmospheric pressure is equal to 1.149 psia, and changes to 6.255 psia at a total discharge pressure of 80 psia, then rejects heat at constant pressure until reaching thermal equilibrium. For a single-stage air compressor, condensation occurs after the discharge because the compressed air dew point is always greater than the ambient temperature. The hot, compressed air rejects heat to the surrounding ambient-temperature air through the walls of the air distribution system. To prevent air distribution line condensate, many install an aftercooler at the compressor discharge to collect condensate of hot, compressed air. If the heat of the compressed air exceeds the capacity of the aftercooler, or the compressed air should have very little moisture at its end use, downstream dryers can be installed to continue moisture removal. Aftercoolers and dryers are outside the scope of this study.

The focus of this study is to investigate alternative dehumidification technologies capable of eliminating the need to buffer the intercooling of a two-stage air compressor. Recall that if the intermediate pressure dew point is a greater temperature than the inlet air temperature, the intercooler would condense water vapor and a cooling buffer would be necessary to prevent damage. To demonstrate the phase-change that occurs when intercooling to the initial air temperature, the path of water vapor during two-stage air compression from atmospheric pressure of 14.7 psia to a desired discharge pressure 80 psia follows.

Figure 26 shows the thermodynamic path of water vapor during the ideal two-stage compression of air entering at 70 °F, 60 % relative humidity, and atmospheric pressure, compressed to an optimum intermediate total pressure of 34.29 psia in the first stage, intercooled at constant pressure to the inlet air temperature, compressed again to the desired total discharge pressure of 80 psia in the second stage, then discharged to the aftercooler where it rejects heat. Figure 27 shows the thermodynamic path of water vapor during the ideal two-stage compression of air entering at 110 °F, 90 % relative humidity, and atmospheric pressure, again compressed to the optimum intermediate pressure of 34.29 psia in the first stage, intercooled to the inlet air temperature at constant pressure, compressed to the desired total discharge pressure of 80 psia, and discharged to the aftercooler where it rejects heat at constant pressure to the surrounding air through the aftercooler walls.

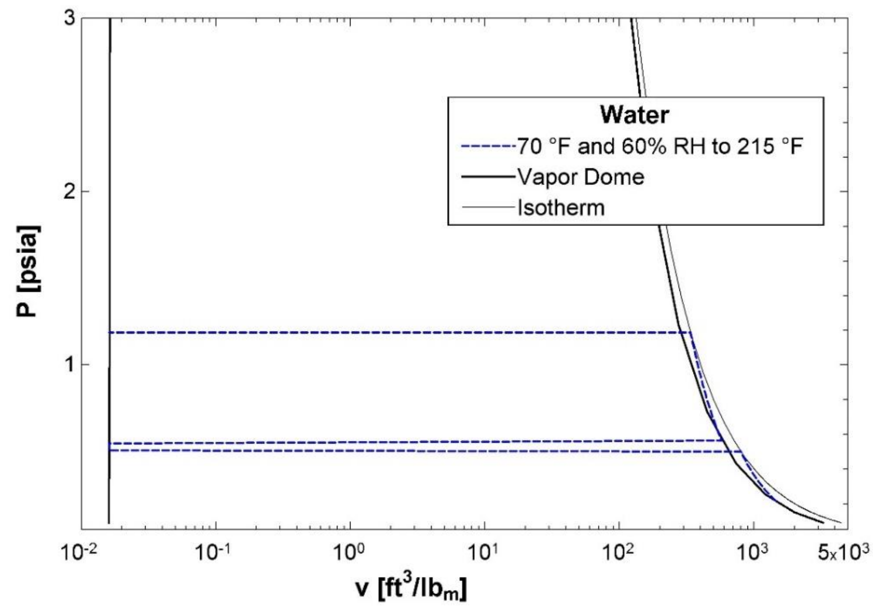

Figure 26: Path of water vapor undergoing ideal two-stage compression with air entering at 70 °F, 60% relative humidity, and atmospheric pressure of 14.7 psia compressed to a discharge pressure of 80 psia

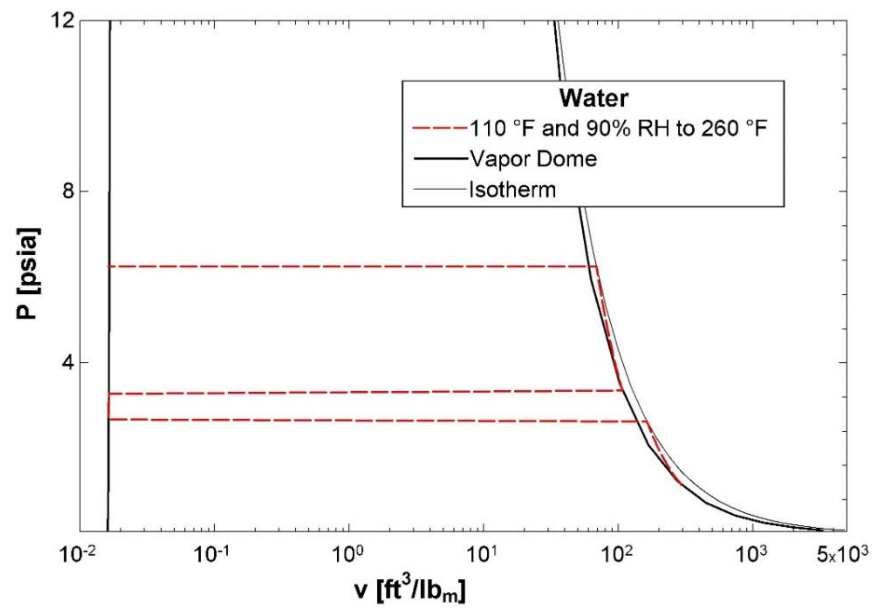

Figure 27: Path of water vapor undergoing ideal two-stage compression with air entering at 110 °F, 90% relative humidity, and atmospheric pressure of 14.7 psia compressed to a discharge pressure of 80 psia

Figure 26 shows that the partial vapor pressure of air at an atmospheric pressure of 14.7 psia, a temperature of 70 °F, and 60% relative humidity equal to 0.218 psia, increases to 0.509 psia and approximately 215 °F in the first stage of the compressor corresponding to the optimum intermediate pressure of 34.29 psia, intercools at constant pressure to the first-stage inlet air temperature of 70 °F, then increases to 1.187 psia and again 215 °F in the second stage, corresponding to a desired discharge pressure of 80 psia. Although the saturation temperature of the partial vapor pressure at an atmospheric total pressure is 55 °F, the dew point at the intermediate total pressure is 80 °F, so intercooling to the initial air temperature of 70 °F results in condensation within the intercooler. Similarly, the discharge pressure dew point is 107.5 °F and aftercooling to the first-stage inlet temperature results in condensate as well.

Figure 27 shows the path of water vapor within air initially at a temperature of 110 °F, 90% relative humidity, and atmospheric pressure undergoing the same compression, first from a partial vapor pressure of 1.149 psia to an intermediate total pressure of 34.29 psia in which the partial vapor pressure is 2.681 psia, intercooled at constant pressure to the initial air temperature of 110 °F, compressed to a total discharge pressure of 80 psia in which the partial vapor pressure is 6.225 psia, then aftercooled at constant pressure until reaching thermal equilibrium. While the initial dew point is 106.4 °F, the intermediate pressure dew point is 137.1 °F, so intercooling to the first-stage inlet air temperature results in condensate. The discharge pressure dew point is 171.9 °F and aftercooling to the first-stage inlet temperature results in condensate as well.

In both cases, the intermediate pressure dew point and the discharge pressure dew point exceed the first-stage inlet air temperature. To prevent damage of the compressor, it is necessary to buffer intercooling to a threshold above the intermediate pressure dew point. Consequently, the second-stage of the two-stage compressor receives air at temperature greater than the first-stage inlet temperature, resulting in additional specific work for the two-stage compressor. Although downstream aftercoolers and dryers may prevent condensate from accumulating within the air distribution lines or reaching the end-use, neither recuperates the energy lost to the consequence of buffered intercooling. The following subsection investigates the change in specific work of air compression due to the presence of water vapor in both the single-stage and two-stage air compressors relative to the ideal case of dry air compression without moisture.

5.7 Increase of Single-Stage Air Compressor Specific Work due to Humidity

The specific work of actual single-stage air compression is subject to the properties of humidity, and consequently uses the gas constant and specific heat ratio of air rather than dry air. The following parametric study calculates both the gas constant and specific heat of air ranging temperature from 70 to 110 °F and relative humidity from 30 to 90%, and determines the specific work resulting from a discharge pressure ranging from 75 to 150 psig. Using the values from the parametric study of dry air compression, the percent difference in specific work relative to the ideal case is calculated by subtracting the ideal value from the actual value then dividing the difference by the ideal value.

$$\Delta\%_w = \frac{w_{actual} - w_{ideal}}{w_{ideal}}, \qquad (Eq. 5.25)$$

Figure 28, Figure 29, and Figure 30 show the change in specific work of single-stage compression of air with 30, 60, and 90% relative humidity, respectively.

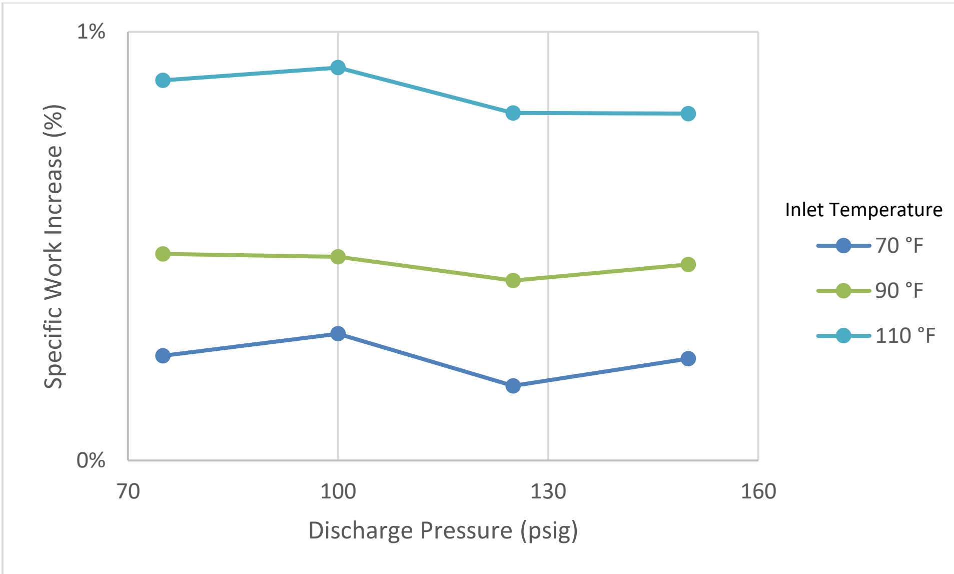

Figure 28: Specific work increase of single-stage compression of air at 30% relative humidity compared to dry air for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

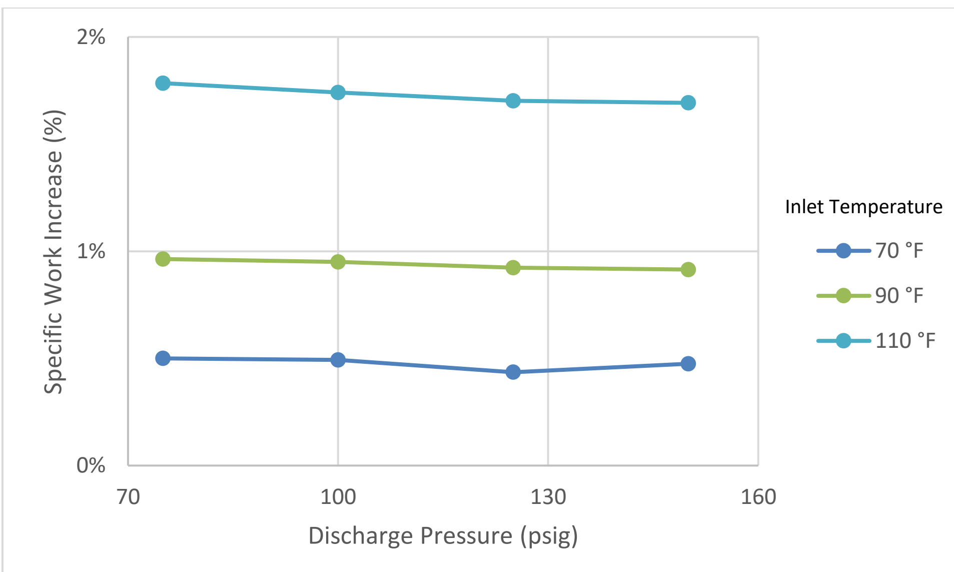

Figure 29: Specific work increase of single-stage compression of air at 60% relative humidity compared to dry air for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

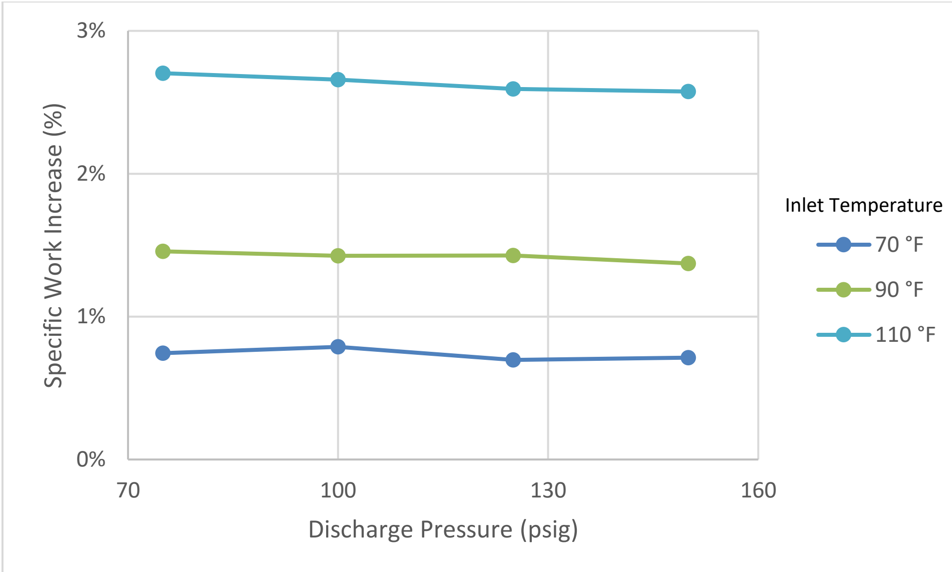

Figure 30: Specific work increase of single-stage compression of air at 90% relative humidity compared to dry air for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

The presence of moisture in the single-stage case can amount to approximately 3% additional specific work, which can have a significant cost in the long term. To further evaluate the increase of specific work due to humidity and compare to the dry air case, the specific work difference relative to the lowest inlet temperature and discharge pressure of the parametric study is calculated. In the dry air case, a single-stage air compressor costs an additional 2% specific work per 10 °F increase of inlet temperature, and 15-20% per 25 psig. Figure 31 and Figure 32 show the specific work increase relative to an inlet temperature of 70 °F at 60 and 90% relative humidity, respectively. Figure 33 and Figure 34 show the specific work increase relative to a discharge pressure of 75 psig at 60 and 90% relative humidity, respectively. Temperature based specific work increases are greater than the dry air case as humidity increases because the water vapor requires more energy to compress. Pressure based specific work increases do not change because they are relative to the lowest discharge pressure at the same level of humidity.

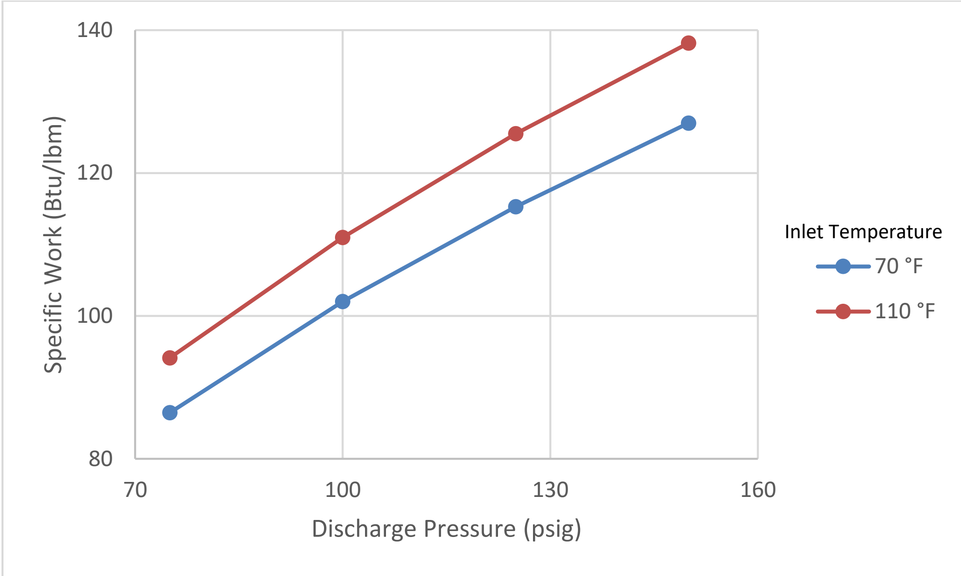

Figure 31: Increase of single-stage air compression specific work due to inlet temperature increase from 70 °F at 60% relative humidity

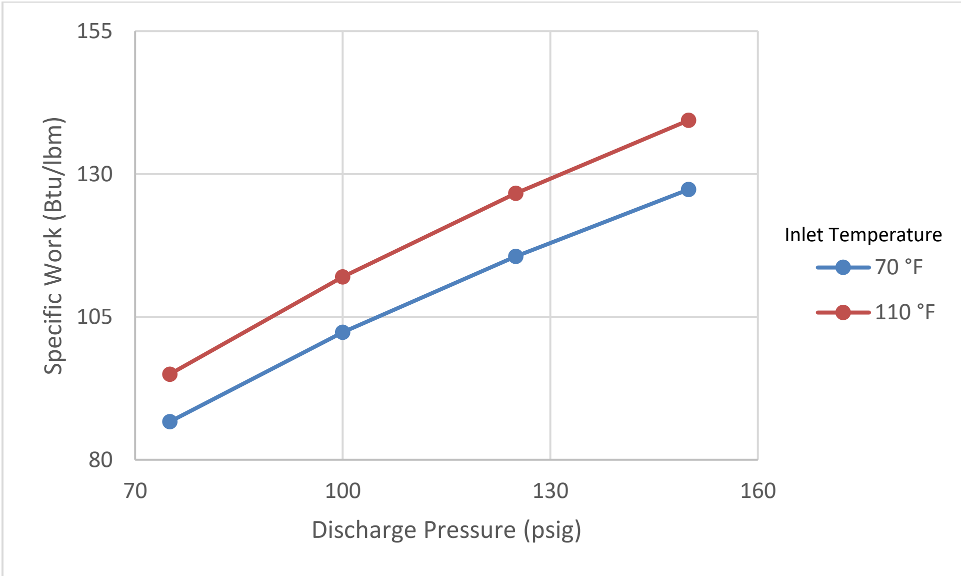

Figure 32: Increase of single-stage air compression specific work due to inlet temperature increase from 70 °F at 90% relative humidity

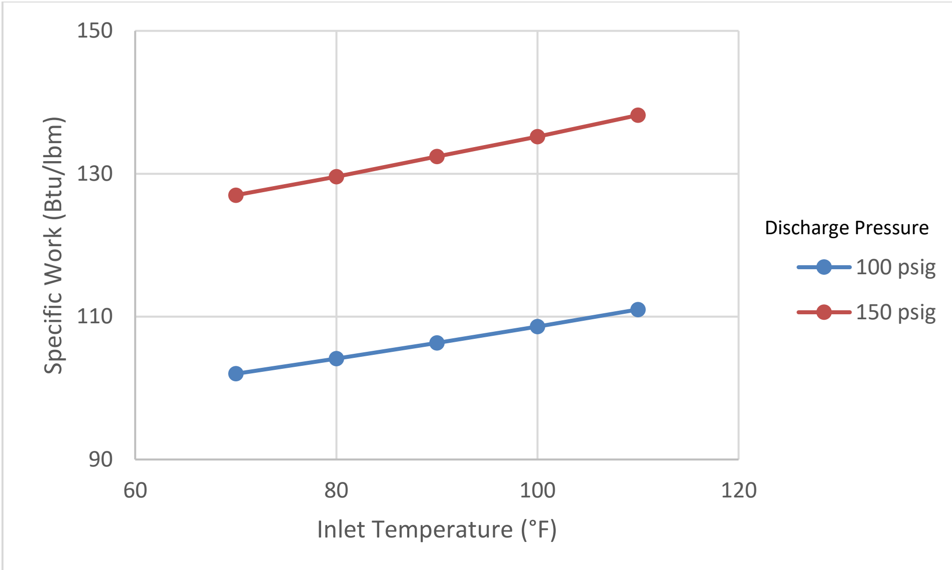

Figure 33: Increase of single-stage air compression specific work due to discharge pressure increase from 75 psig at 60% relative humidity

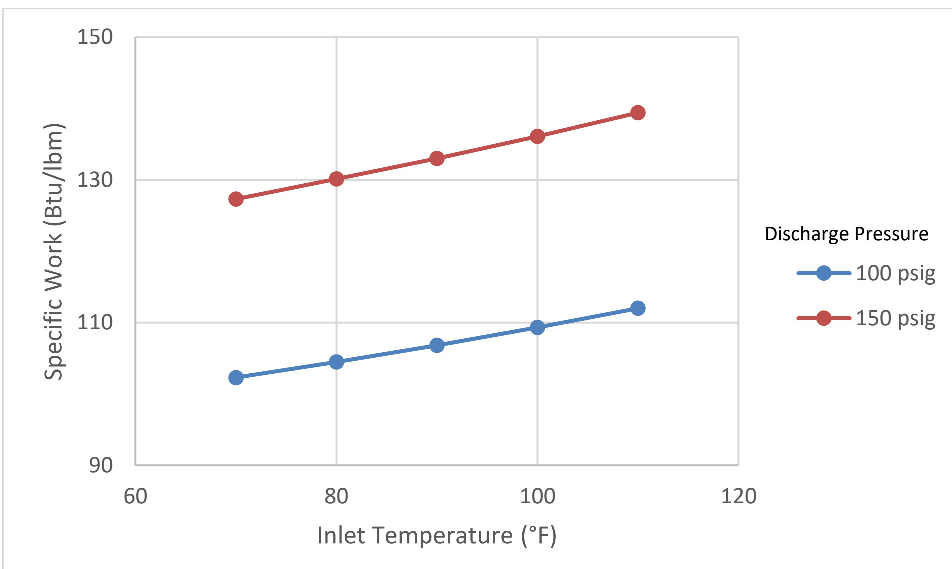

Figure 34: Increase of single-stage air compression specific work due to discharge pressure increase from 75 psig at 90% relative humidity

As can be seen from Figure 31 and Figure 32, the increase of specific work for a single-stage air compressor between 60 and 90% relative humidity is negligible. Figure 33 and Figure

34 also show that there is no additional percent increase of specific work due to pressure for moisture because the datum is compared at the same amount of humidity.

5.8 Increase of Two-Stage Air Compressor Specific Work due to Humidity

Like single-stage air compression, the model for two-stage air compression uses the air gas constant and specific heat ratio. However, to protect the integrity of the compressor, intercooling must be buffered above the intermediate pressure dew point. To prevent any possibility of condensation, the intercooling buffer utilized in this study is 5 °F. A parametric study of two-stage compression atmospheric air with inlet temperature ranging from 70 to 110 °F, relative humidity from 30 to 90%, and discharge pressure from 75 to 150 psig follows. The change in specific work of two-stage moist air compression relative to the two-stage dry air case for 30, 60, and 90% relative humidity, are shown by Figure 35, Figure 36, and Figure 37, respectively.

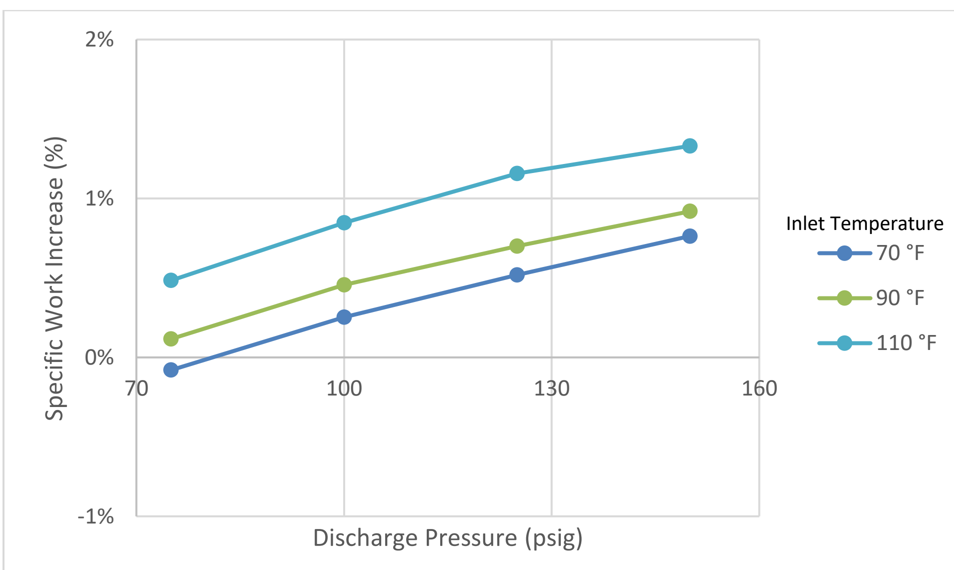

Figure 35: Increase of specific work for two-stage compression due to presence of moisture in air at 30% relative humidity for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

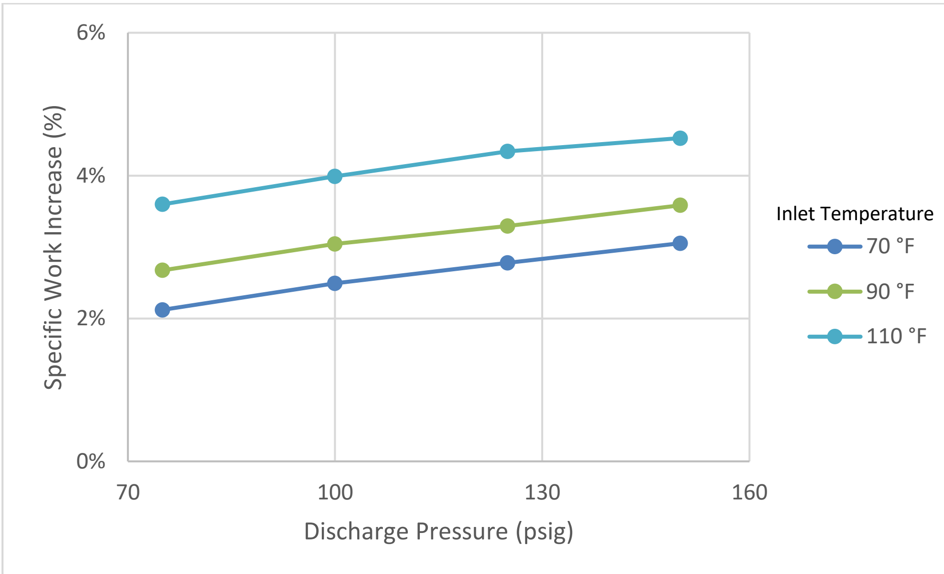

Figure 36: Increase of specific work for two-stage compression due to presence of moisture in air at 60% relative humidity for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

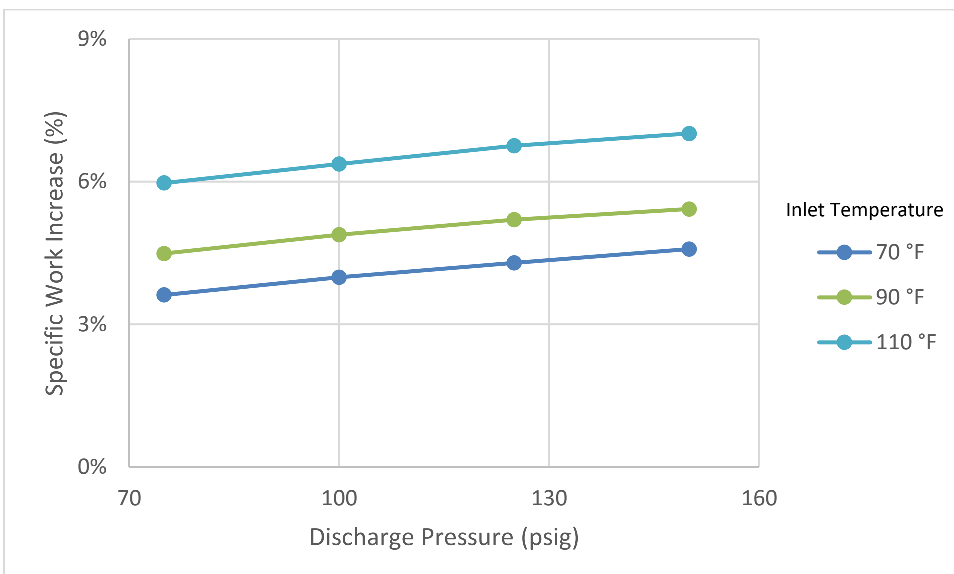

Figure 37: Increase of specific work for two-stage compression due to presence of moisture in air at 90% relative humidity for an inlet temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

Figure 35 indicates that the presence of moisture in air at 30% relative humidity results in less than 2% additional specific work in the upper limit of ambient air temperature of this study, usually approximately an additional 1% for all other ambient temperatures and discharge pressures. According to Figure 36 , air at 60% relative humidity requires additional specific work ranging from 2-5%, and Figure 37 shows an additional 3-7% specific work at 90% relative humidity. To understand the impact of inlet air temperature and discharge pressure on specific work of a two-stage compressor compressing moist air, Figure 38 and Figure 39 show the specific work difference relative to 70 °F at 60 and 90% relative humidity, respectively. Regarding temperature, there is a positive correlation between increasing specific work difference and moisture because the water requires more energy to compress.

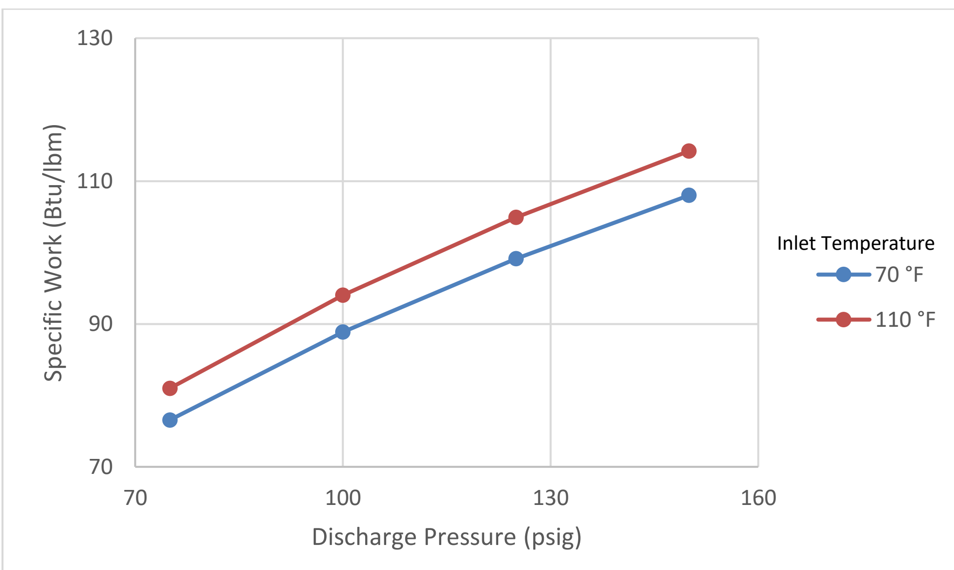

Figure 38: Comparing specific work of moist two-stage air compression for air entering at a temperature of 70 and 110 °F and an atmospheric pressure of 14.7 psia, relative humidity of 60%, and discharge pressure ranging from 75 to 150 psig

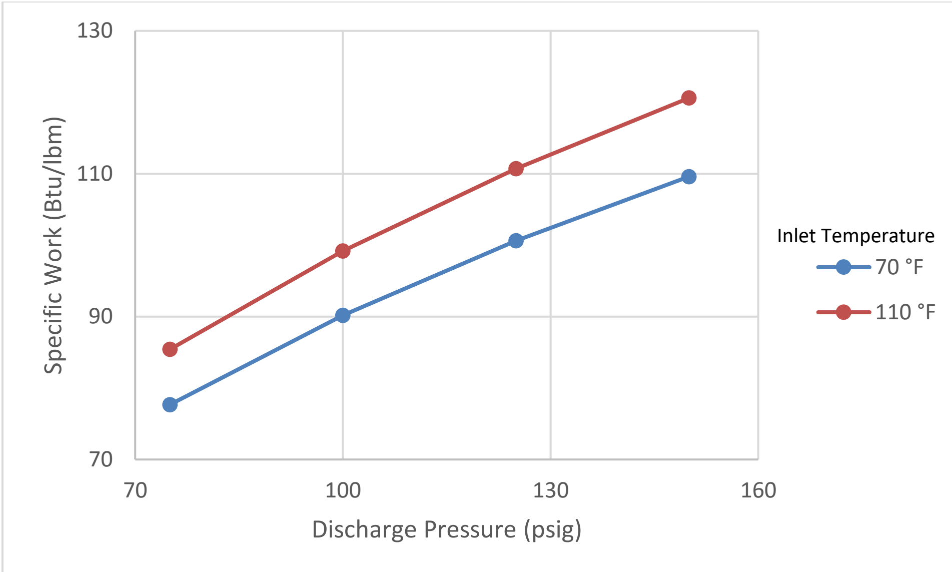

Figure 39: Comparing specific work of moist two-stage air compression for air entering at a temperature of 70 and 110 °F and an atmospheric pressure of 14.7 psia, relative humidity of 90%, and discharge pressure ranging from 75 to 150 psig

Compared to Figure 38, Figure 39 shows an increasing disparity between specific work due to inlet air temperature. This is because the more moisture in the air, the greater the saturation temperature of the intermediate partial vapor pressure, which results in a greater intercooler limitation and more specific work. Furthermore, the moisture adds energy to the air, which increases with temperature, so more moisture will in fact require more specific work per degree Fahrenheit. On the other hand, discharge pressure still has a greater effect on the specific work altogether. Figure 40 and Figure 41 show the specific work of compressing moist air with two stages to 100 and 150 psig at both 60 and 90% relative humidity.

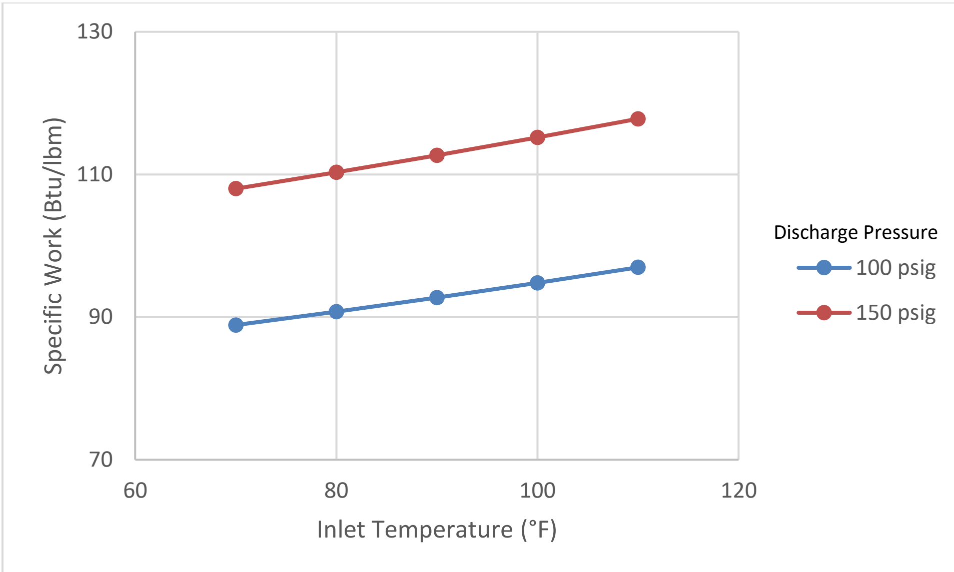

Figure 40: Comparing specific work of moist two-stage air compression for discharge pressures of 100 and 150 psig with air entering at an atmospheric pressure of 14.7 psia, relative humidity of 60%, and temperature ranging from 70 to 110 °F

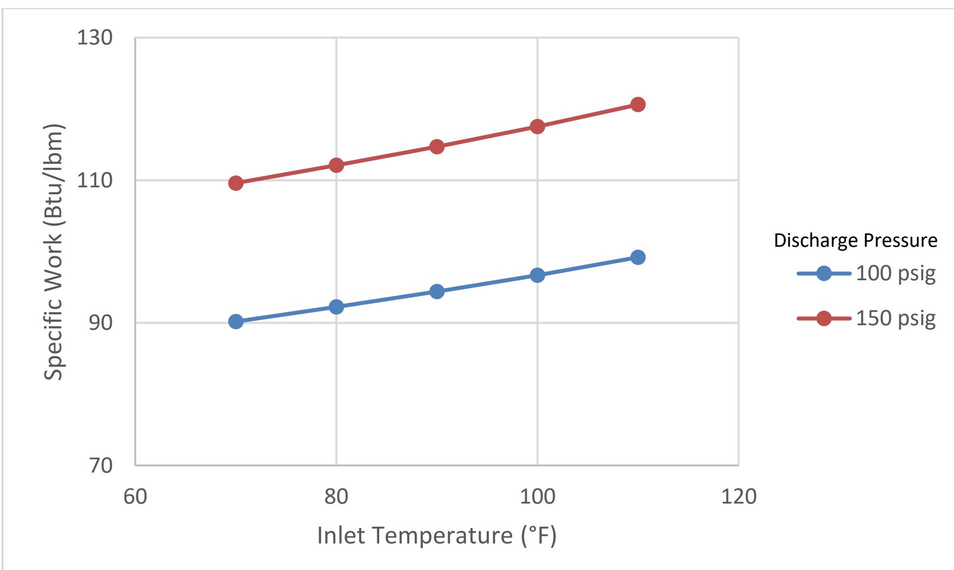

Figure 41: Comparing specific work of moist two-stage air compression for discharge pressures of 100 and 150 psig with air entering at an atmospheric pressure of 14.7 psia, relative humidity of 90%, and temperature ranging from 70 to 110 °F

While the specific work of compression due to the presence of moisture increases from Figure 40 from Figure 41, the most important takeaway from the plots is that the disparity in specific work does not because the datum in each evaluation comes from the same amount of humidity. Moisture in the air inhibits intercooling and results in more specific work per degree Fahrenheit but does not incur additional specific work per psig.

5.9 Specific Work of Two-Stage Air Compression Relative to Single-Stage Model

It is important to re-evaluate the specific work difference of a two-stage compressor relative to the single-stage model using moist air properties and limitations. In the dry-air case, the two-stage compressor reduces specific work by 10-20% depending on the discharge pressure. Atmospheric air compression requires more specific work with increasing moisture, but the difference between both compressors is ambiguous because the extent of increasing specific work is different between the models. Therefore, the specific work difference of a two-stage compressor relative to a single-stage compressor at 30, 60, and 90% relative humidity is shown by Figure 42, Figure 43, and Figure 44, respectively.

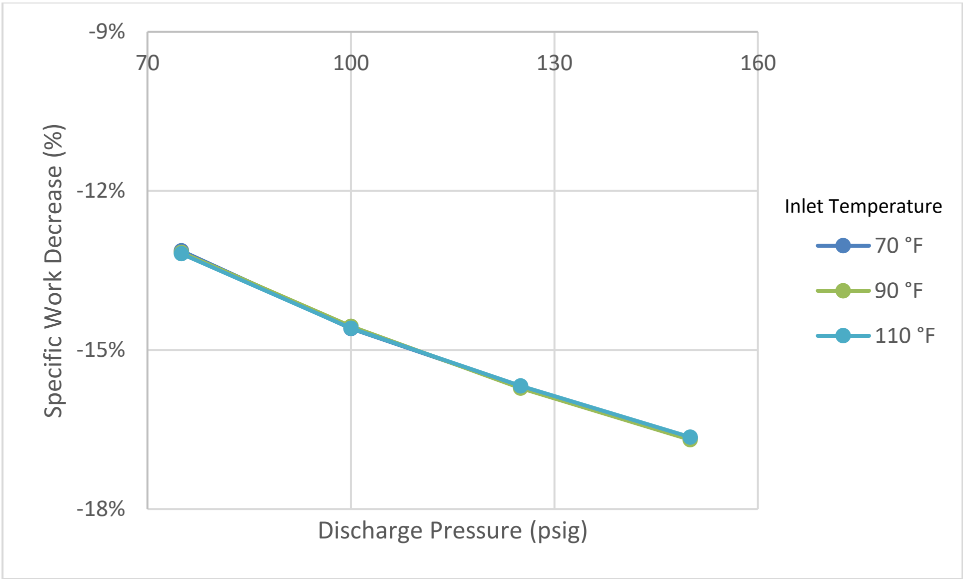

Figure 42: Change in specific work of air compression between a two-stage to a single-stage model at 30% relative humidity using atmospheric air at a temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

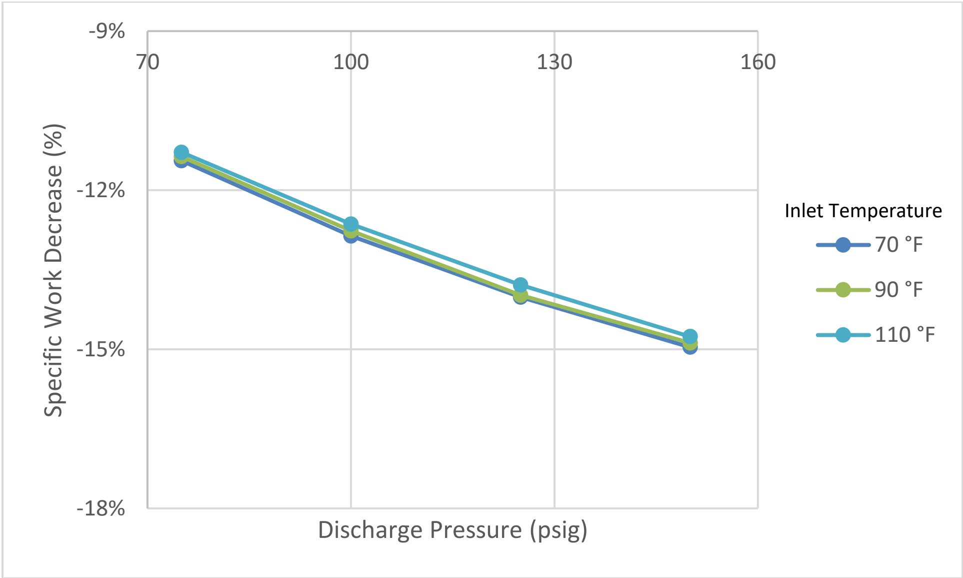

Figure 43: Change in specific work of air compression between a two-stage to a single-stage model at 60% relative humidity using atmospheric air at a temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

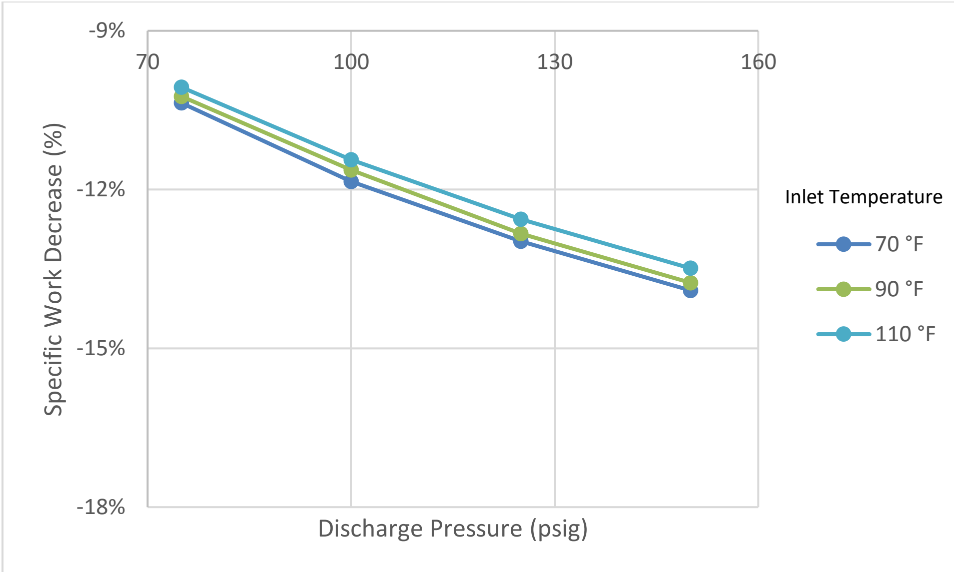

Figure 44: Change in specific work of air compression between a two-stage to a single-stage model at 90% relative humidity using atmospheric air at a temperature ranging from 70 to 110 °F and discharge pressure ranging from 75 to 150 psig

The trend from Figure 42 to Figure 43 then to Figure 44 negatively correlates with moisture. At 30% relative humidity, moisture effects two-stage compressor specific work by 1-3% depending on discharge pressure. Thus, according to Figure 42, the specific work reduction of moist air compression between a two-stage compressor and single-stage compressor is 12-18%. However, as moisture in the air increases, the intermediate pressure dew point rises above the inlet air temperature and intercooler buffering results in more specific work for the two-stage air compressor. Therefore, the elevated inlet air temperature of the second stage of the two-stage compressor results in a net specific work reduction of 10-15% at 60% relative humidity as shown by Figure 43, and 9-13% at 90% relative humidity as shown by Figure 44.

The two-stage air compressor still has a specific work advantage to the single-stage model at all levels of moisture. However, relative to the ideal two-stage compressor, actual performance suffers 3-7%. This motivates the investigation of dehumidification techniques that can recuperate the 3-7% specific work. Alternative dehumidification techniques are a vapor compression refrigeration cycle and a desiccant wheel, which are described in the following sections.

6. DEHUMIDIFYING ATMOSPHERIC AIR

6.1 Overview

The moisture in atmospheric air (air) increases its mass and energy. The parametric studies of earlier sections indicate that a single-stage air compressor drawing air at a temperature of 110 °F and 90% relative humidity and discharging at 150 psig requires less than 3% additional specific work that the ideal case. On the other hand, the two-stage air compressor (TSC) drawing air at the same condition requires an additional 7% specific work than the ideal case due to elevated intercooling temperatures designed to prevent condensation. For air at a temperature of 70 °F and 60% relative humidity discharged at 75 psig, the TSC consumes 2% additional specific work, and the greater the inlet temperature, relative humidity, or discharge pressure, the greater the specific work consumption. Therefore, the focus of this study is to investigate alternative dehumidification technologies, conduct parametric studies, and net change of energy demand using a TSC equipped with a dehumidifier when compared to the standalone TSC.

Removing moisture enables lower dew points and reduces the risk of performance decreases and system damage. A lower dew point enables further intercooling, and the lower chance of performance decrease will prevent costs incurred by maintenance, failure, and downtime. If the increase of specific work due to elevated intercooling can be reduced, and the cost of maintenance or equipment replacement avoided, dehumidification technology may be economically justifiable. One method of removing moisture from air is cooling the air to the dew point to condense and drain the moisture, which is the consequence of a conventional air conditioning. Another method of removing moisture is by attracting it to a hygroscopic material, such as a desiccant. The desiccant wheel continuously adsorbs moisture from a passing airstream and rejects it to a heated airstream. The next subsection elaborates on air conditioning, refrigerants, and the vapor compression refrigeration cycle. After these topics are covered, desiccants and the desiccant wheel are introduced. Both the vapor compression refrigeration cycle and desiccant wheel are modeled and analyzed to determine the capacity of moisture removal, side-effects, and net system efficiency when combined with the two-stage compressor.

6.2 Air Conditioning and Refrigerants

Naturally occurring heat transfer occurs by a temperature difference from one body at a certain temperature to another at a lower temperature until the two bodies reach thermal equilibrium, that is, the same temperature. A heat-exchanger is a device designed to optimize heat transfer between two materials by transferring heat first through its walls by conduction, then through a working fluid circulated outside the walls by convection. The source of heat can be a solid such as a hot electrical component, or a fluid such as hot air or hot water. For example, the TSC intercooler is a heat exchanger that transfers heat from compressed air to a working fluid outside the walls of the intercooler, whether that be the surrounding air, or a fluid within a jacket enclosing the compressor. Similarly, an air conditioner is a heat exchanger that removes heat from air by transferring it to a refrigerant within an evaporator and removes heat from the refrigerant by transferring it elsewhere and condensing the refrigerant. Refrigerants are chosen as the working fluid in air cooling due to their relatively low enthalpy of vaporization.

At a temperature of 70 °F, refrigerant 410A requires approximately 80 Btu/lbm to vaporize, whereas water needs more than 1000 Btu/lbm (ASHRAE, 2017). During phase change, the refrigerant evaporates at constant temperature but lowers the air temperature by absorbing its heat. Enough heat removal can cool the air to its dew point and condense its moisture. As the water vapor condenses and the level of airborne moisture decreases, the temperature of the air continues to decrease until it reaches the regulated saturation temperature of the refrigerant, which is controlled by a throttling valve. By charging an air conditioner with an amount of refrigerant proportional to the amount of heat that needs to be removed from the air and using appropriately designed heat exchangers for both refrigerant evaporation and condensation, the refrigerant can quickly transfer heat from one location to another, say to cool a room and reject the heat outside.

A designer selects a refrigerant that operates safely under the conditions of the system and follows environmental policies of the acting government. Environmental policies were enacted because charge losses through leaks result not only in lost system capacity, but also in ozone depletion through chemical reactions of the refrigerant in the stratosphere. Most refrigerants that are classified as chlorofluorocarbon or hydro-chlorofluorocarbon cause ozone depletion, and some more than others. The use of refrigerants with a high environmental impact is becoming illegal. An industry standard in the United States is refrigerant 410A, and that is the refrigerant selected for the vapor compression system evaluated in following sections.

6.3 Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle (VCRC) circulates a refrigerant through an evaporator, compressor, condenser, and throttling valve in a thermodynamic cycle. The throttling valve regulates the pressure of the refrigerant with a spring-loaded pin at a small inlet driven by a differential pressure and controls the saturation temperature to discharge the refrigerant as a low-quality liquid-vapor mixture into the evaporator. The refrigerant absorbs heat in the evaporator at constant pressure until it completely vaporizes. The saturated vapor enters a compressor specifically designed for the type of refrigerant and discharges as a superheated vapor into a condenser, which where the refrigerant rejects heat until it completely condenses. The throttling valve expands the high-pressure, saturated liquid to discharge at a lower pressure and temperature allowing some to vaporize, restoring the initial liquid-vapor state. A system diagram is shown by Figure 45.

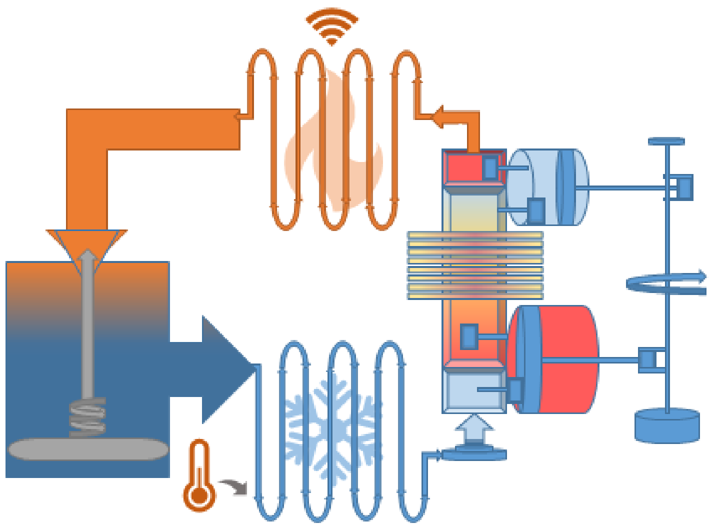

Figure 45: Vapor compression refrigeration cycle with evaporator (bottom), two-stage refrigerant compressor (right), condenser (top), and throttling valve (left).

6.4 Desiccant Dehumidification

Although cooling the air to its saturation temperature initiates condensation, the amount of moisture removal is proportional to the refrigerant cycling through the system, and significant moisture can either require more refrigerant or time to achieve the target cooling temperature. Furthermore, extended exposure can build scale on evaporator coils reducing their cooling effectiveness. When there is significant humidity and the goal is to remove moisture, dehumidification with a hygroscopic material may be more appropriate. A desiccant absorbs or adsorbs water vapor due to a vapor pressure difference. Most desiccant absorbents are liquids that attract water and chemically react with it. Most desiccant adsorbents are solids that attract water and draw it into pores with capillary action (ASHRAE, 2017).

Solid desiccant adsorbents (SDA) have extremely high surface area and inner volume because of their porosity. Dry SDA pores have very low partial vapor pressure, naturally drawing nearby water vapor at a relatively greater partial vapor pressure. In addition, the SDA surface is highly polar and condenses water. The condensate fills vacant pores through capillary action and diffusion, maintaining low partial vapor pressure at the surface and attracting moisture until the desiccant reaches its capacity. Capacity depends on total surface area, vacant volume, and capillary size, which are outside the scope of this study (ASHRAE, 2016).

The most common SDA are silica gels and molecular sieves. Manufacturers produce both substances for specific applications with relative ease and low-cost. The partial vapor pressure of water that condenses within both silica gels and molecular sieves positively correlates with temperature. Therefore, a hot airstream with low partial vapor pressure can be used to heat the SDA and increase the partial vapor pressure of the condensate within, which will migrate into the airstream by a partial vapor pressure difference, enabling the SDA to once again adsorb water. An SDA can be recharged discretely or continuously, and a common method of continuously regenerating the desiccant is the use of a rotary solid desiccant dehumidifier (desiccant wheel).

6.5 Rotary Solid Desiccant Dehumidifier

A desiccant wheel is a belt-driven ring with SDA filler that spins at low speeds such as 20-30 rotations per hour. A moist airstream passes through one portion of the wheel’s face where the wheel attracts and retains its moisture if the SDA has capacity. A hot airstream passes through another portion of the wheel’s face to heat the condensate within the SDA filler, vaporize some, and draw out the vapor by a partial vapor pressure difference. The rate of moisture adsorption and removal depends on many factors including the type of SDA filler, the wheel size, the rotating speed, the moist air conditions, the reactivation temperature, the return air conditions, the airstream flow rates, and the airstream surface area. Figure 46 is a diagram of the most basic desiccant wheel configuration with color coding to suggest relative temperatures.

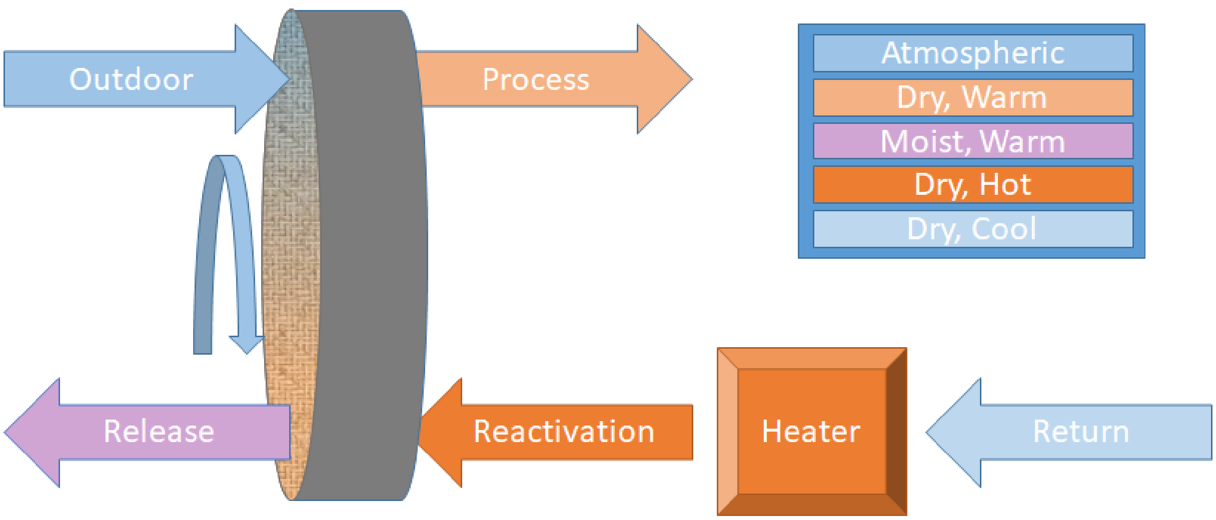

Figure 46: Basic desiccant wheel configuration removing moisture from an outdoor airstream with a reactivation airstream composed of heated return air

When the pores of the SDA have low partial vapor pressure relative to an incoming airstream, the wheel attracts moisture from the airstream. During adsorption and condensation however, the water vapor rejects heat, and Figure 46 indicates that the temperature of the outdoor air increases after it passes through the wheel due to the heat released by adsorption. To restore the wheel's moisture removing capacity, an airstream with lower partial vapor pressure must heat and vaporize the moisture condensed into the pores of the SDA. The moisture scavenging airstream, usually referred to as a reactivation airstream, is sourced from indoor return air that is heated to reduce its partial vapor pressure. The hot return air, or reactivation air, heats the wheel and its condensate, vaporizing some, drawing the vapor away because of a vapor pressure difference, and restoring the wheel's capacity to adsorb water vapor. The reactivation air acquires moisture and exits the wheel with at a lower temperature due to the heat absorbed by vaporized condensate. The primary energy consumption of a desiccant wheel is the heater, which can use many sources of energy such as direct- or indirect-fired natural gas, resistive electric coils, and so on.

The following section reviews the methodology for investigating the efficiency of dehumidified two-stage air compression. The methodology section reviews the general assumptions applied in earlier background sections that also apply in other situations, as well as the location of states utilized for thermodynamic evaluation. In addition to the methods of model development and the software used to do so, the coupling of systems is explained. The expectation is that the methodology will provide accountability, as well as a reference for reproducing the efforts pursued in this investigation.

7. METHODOLOGY

7.1 Overview

The following subsections describe the methods used to develop the thermodynamic models investigated in this study to provide a reference should others wish to continue the efforts of the reported results. General assumptions were applied to multiple thermodynamic systems, yet different software or approaches were sometimes used to develop models due to the absence of information or to simplify the analysis. The vapor compression refrigeration cycle readily modeled thermodynamically because it is subject to refrigerant selection and the temperature difference between the refrigerant and ambient conditions. On the other hand, the desiccant wheel was semi-empirically modeled using manufacturing data because performance varies according to the manufacturer specification.

7.2 General Assumptions

The Inch-Pound unit system is used in calculations in terms of the British thermal unit (Btu) for energy, pound mass (lbm) for mass, pound per square inch absolute (psia) for pressure, and Rankine (°R) for temperature. The range of ambient conditions investigated are atmospheric air (air) temperature from 70 to 110 °F and relative humidity (RH) from 30 to 60%. All calculations are performed using absolute thermodynamic scales so that magnitudes are comparable relative to absolute zero.

$$°R = °F + 459.67\,°R, \qquad (Eq. 7.1)$$

$$psia = psig + 14.7\,psia, \qquad (Eq. 7.2)$$

$$\phi = \frac{\%\,RH}{100}, \qquad (Eq. 7.3)$$

Although processes such as compression subject the fluids to rapid motions, it is assumed the processes have uniformly distributed properties and occur between defined equilibrium states. All equipment is also assumed to be operating steadily, with time-averaged quantities when questionable. Steady flow indicates that all quantities remain constant in time. Using a dot above the variable to denote a time rate of change, the steady flow assumption implies that the mass and energy of a system are constant over time. For an open system enclosed by a boundary, the mass at any time is equal to the difference between the rate of mass flowing into the system and exiting the system, and the energy at any time is equal to the difference between the rate of energy entering the system and the rate of energy exiting the system.

$$\Sigma\dot{m} = 0 = \Sigma\dot{m}_{in} - \Sigma\dot{m}_{out}, \qquad (Eq. 7.4)$$

$$\Sigma\dot{E} = 0 = \Sigma\dot{E}_{in} - \Sigma\dot{E}_{out}, \qquad (Eq. 7.5)$$

The energy of a system at any time is also equal to the sum of the time rate of change of heat and work, as well as the mass flow rate multiplied by the enthalpy, kinetic energy due to translation, and potential energy due to elevation.

$$\Sigma\dot{E} = \Sigma\dot{Q} + \Sigma\dot{W} + \Sigma\dot{m}\left(h + \frac{1}{2}\bar{V}^2 + gz\right), \qquad (Eq. 7.6)$$

Because the steady flow approximation means that the mass and energy of a system are constant at any time, the flow rate of mass into the system is equal to the flow rate of mass out of the system, and the flow rate of energy into the system is equal to the flow rate of energy exiting the system.

$$\Sigma\dot{m}_{in} = \Sigma\dot{m}_{out}, \qquad (Eq. 7.7)$$

$$\Sigma\dot{E}_{in} = \Sigma\dot{E}_{out}, \qquad (Eq. 7.8)$$

By denoting the sum of enthalpy, kinetic energy, and potential energy due to elevation with the arbitrary variable c, using the steady flow approximation and substituting the time rate of change for energy and mass, the steady flow energy balance of a system is as follows.

$$\dot{Q}_{in} - \dot{Q}_{out} + \dot{W}_{in} - \dot{W}_{out} = \Sigma \dot{m}_{out} c_{out} - \Sigma \dot{m}_{in} c_{in}, \qquad (Eq. 7.9)$$

Because all changes in potential and kinetic energy and pressure drops within equipment are assumed to be negligible, the following apply to all models.

$$\Delta KE = \frac{1}{2}\dot{m}(\bar{V}_{out}^2 - \bar{V}_{in}^2) = 0, \qquad (Eq. 7.10)$$

$$\Delta PE = \dot{m}g(z_{out} - z_{in}) = 0, \quad (Eq. 7.11)$$

$$\Delta P_{equipment} = 0, \qquad (Eq. 7.12)$$

7.3 Engineering Equation Solver

Engineering Equation Solver (EES) is software used to call property data and conduct parametric studies of all models. EES is a program that identifies variables and solves their numerical relationships independently of placement within an equality. With built-in property data as well as unit-checking capabilities, EES is used to call property functions that require the name of the substance and sufficient independent properties. It is important to recognize however, that EES does not verify the physical significance of the numbers or arguments provided by the user. Engineering judgment is used to decide or investigate whether the results are valid.

7.4 Air Compressor Modeling

The air compressor is assumed to be isentropic to determine the temperature difference between two states using the pressure ratio and specific heat ratio, and to analyze a system that does not lose energy through heat transfer. An isentropic process is reversible and adiabatic, where a reversible process is free from losses such as friction and leaks, and an adiabatic process has sufficient insulation to prevent energy in the form of heat from inadvertently transferring past the system boundary. While the reciprocating piston moves faster than the air can reach equilibrium, the compressor in this study is modeled as a quasi-equilibrium process to describe a system where the properties of the air are uniform at any point and have a defined state. The intercooler is assumed to have sufficient cooling capacity to reach the initial temperature. The compressor is assumed to be able to reach any pressure of interest without complication. In addition, all general assumptions apply.

The two-stage compressor (TSC) has four states: the inlet and outlet of the first stage, and those of the second stage. The first stage of a standalone compressor draws air at inlet temperature, pressure, and humidity, and discharges it at the intermediate pressure with a corresponding temperature from the isentropic relation. The intercooler reduces the air temperature at constant pressure to five degrees greater than the pressure dew point. The second stage draws air from the intercooler at the intercooler buffer temperature and increases its pressure to the discharge pressure and increases its temperature according to the isentropic relation. The resulting quantity is specific work in Btu per lbm of air.

7.5 Vapor Compression Refrigeration Cycle Modeling

Refrigerant 410A (R410A) is chosen for the vapor compression refrigeration cycle (VCRC) because it is the current standard in the United States due to lower ozone depletion than the prior standard, R-22. R410A is a mixture that requires high pressure equipment. The type of refrigerant compressor is not the focus of this study, and its specific work is assumed to be the change in enthalpy of the inlet and discharge states. In concept, the VCRC system is configured so that the TSC supply air passes its evaporator coils before entering the inlet of the TSC (Figure

47). The water vapor in the air condenses on the evaporator coils and drains while the air temperature decreases. It is assumed that there is sufficient refrigerant charged in the system to handle the heat load. The mechanical design of the air conditioner is beyond the scope of this study.

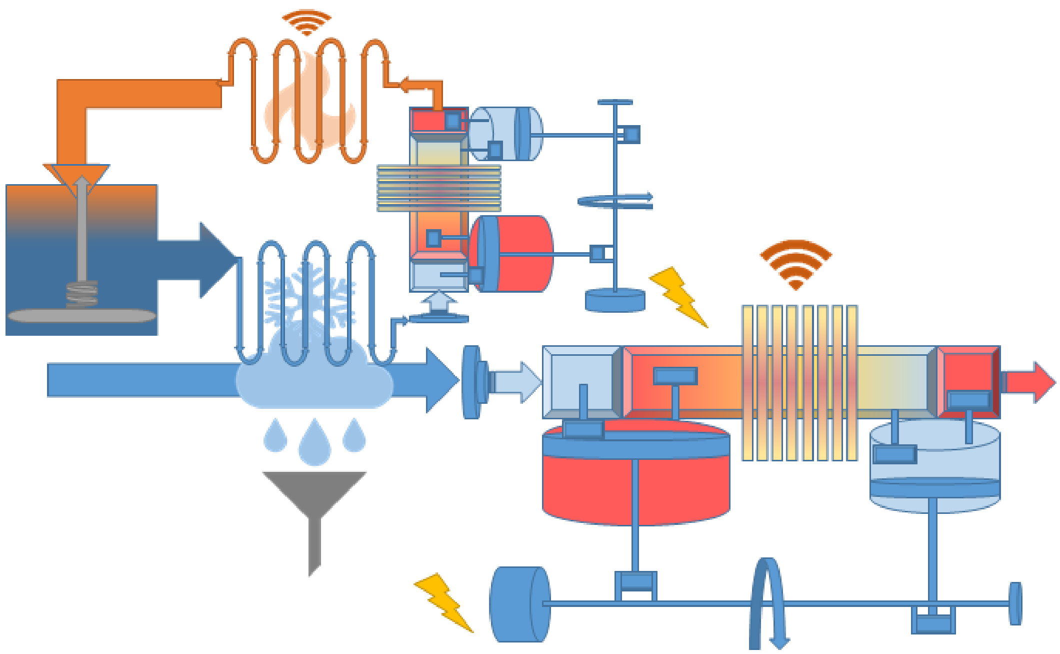

Figure 47: Cooling the supply air of a two-stage air compressor using the evaporator of a vapor compression refrigeration cycle

The VCRC’s four components are evaluated in the steady state so that the mass and energy are constant over time. The first state is defined at the refrigerant compressor inlet, the second at the condenser inlet, the third at the throttling valve inlet, and the fourth at the inlet of the evaporator. The evaporator and condenser are heat exchanging devices that do not require work. In the evaporator, the heat absorbed by the refrigerant is the enthalpy difference of the first and fourth state.

$$\dot{Q}_{in} = \dot{m}(h_1 - h_4), \qquad q_{evaporator} = h_1 - h_4 \left(\frac{Btu}{lbm_{R410A}}\right), \qquad \text{(Eq. 7.13)}$$

In the condenser, the heat rejected by the refrigerant is the enthalpy difference between the second and third state.

$$-\dot{Q}_{out} = \dot{m}(h_3 - h_2), \qquad q_{condenser} = h_2 - h_3 \left(\frac{Btu}{lbm_{R410A}}\right), \qquad (Eq. 7.14)$$

The specific work of a VCRC comes from the motor driving the compressor and is equal to the enthalpy difference between the first and second state.

$$\dot{W}_{in} = \dot{m}(h_2 - h_1), \qquad w_{compressor} = h_2 - h_1 \left(\frac{Btu}{lbm_{R410A}}\right), \qquad (Eq. 7.15)$$

The throttling process occurs quickly without work or heat transfer and is an isenthalpic process.

$$0 = h_4 - h_3, \qquad h_3 = h_4, \qquad (Eq\ 7.16)$$

Assumptions define the equilibrium states for the ideal VCRC. To define the first ideal state, the evaporator is assumed to cool the ambient air to 45 °F using refrigerant at ten degrees below the ambient air temperature. The pressure of the refrigerant is assumed to be the saturated vapor pressure at 35 °F, and the evaporator changes the phase of the refrigerant from a liquid-vapor mixture to a saturated vapor.

$$T_{evaporator} = 35\ °F, \qquad (Eq. 7.17)$$

$$P_{evaporator} = P_g(R410A, 35\ °F), \qquad (Eq. 7.18)$$

The condenser between the second and third state is assumed to reject heat at constant temperature ten degrees above the ambient air temperature at the saturated liquid pressure of the refrigerant, which is the pressure of the second and third state.

$$T_{condenser} = (T_{ambient} + 10°F), \qquad (Eq. 7.19)$$

$$P_{condenser} = P_f(R410A, T_{condenser}), \qquad (Eq. 7.20)$$

The refrigerant compressor is assumed to be isentropic so that the entropy of the first state and saturated liquid pressure of the condenser define the ideal second state. The throttling valve is isenthalpic so that the enthalpy and temperature define the fourth ideal state. Achieving precise states can be difficult because the phase of a substance is very sensitive at the exact saturation temperature. Therefore, states are buffered before subjecting the refrigerant to the equipment. For example, wet compression can damage the compressor, so the refrigerant is superheated by 15 °F prior. High pressure vapor entering the throttling valve reduces effectiveness, so the condensed refrigerant is subcooled by 5 °F after it exits the condenser. Furthermore, actual refrigerant compressors deviate from the isentropic case, so the compressor efficiency is assumed to be 80%, which is the ratio of actual compressor performance to its isentropic counterpart. The refrigerator efficiency provides the actual enthalpy of the second state.

$$T_{compressor} = T_{evaporator} + 15°F, \qquad (Eq. 7.21)$$

$$T_{valve} = T_{condenser} - 5°F, \qquad (Eq. 7.22)$$

$$\eta_{isentropic} = \frac{w_{actual}}{w_{isentropic}}, \qquad (Eq. 7.23)$$

The specific work of the TSC is in terms of lbm air. To determine the net energy change of a system comprised of the TSC and VCRC relative to the standalone TSC, the specific work of the refrigerant compressor must be converted from lbm R410A to lbm air. The heat the absorbed by the refrigerant in the evaporator is proportional to the heat rejected by the air.

$$q_{R410} = q_{air}, \qquad (Eq. 7.24)$$

In the steady flow approximation, the air enters the air conditioner, rejects heat to the evaporator coils where its moisture condenses and drains, then exits the system. The condensate is assumed to drain after reaching the final temperature. The mass of dry air entering is equal to the mass of dry air, while some of the moisture entering the system becomes condensate and the remainder is water vapor.

$$\dot{m}_{da_{in}} = \dot{m}_{da_{out}} = \dot{m}_{da}, \qquad (Eq. 7.25)$$

$$\dot{m}_{wv_{in}} = \dot{m}_{wv_{out}} + \dot{m}_{condensate}, \qquad (Eq. 7.26)$$

Recall that the specific humidity is the mass ratio of water vapor to dry air; the mass of water vapor is equal to the specific humidity multiplied by the mass of dry air.

$$\dot{m}_{wv} = \dot{m}_{da}\, \omega, \qquad (Eq. 7.27)$$

$$\dot{m}_{condensate} = \dot{m}_{wv_{in}} - \dot{m}_{wv_{out}} = \dot{m}_{da}(\omega_{in} - \omega_{out}), \qquad (Eq. 7.28)$$

Considering the work of the evaporator's circulating pump to be negligible, neither the system nor the surroundings perform work while the refrigerant passes the evaporator. Therefore, the energy balance to determine the heat rejected by the air is as follows.

$$-\dot{Q}_{out} = \dot{m}_{da} h_{mix_{out}} + \dot{m}_{condensate} h_{w_f} - \dot{m}_{da} h_{mix_{in}}, \qquad (Eq. 7.29)$$

$$q_{out} = h_{mix_{in}} - h_{mix_{out}} - h_{w_f}(\omega_{in} - \omega_{out}) = q_{air}\left(\frac{Btu}{lbm_{air}}\right), \qquad (Eq. 7.30)$$

To convert the specific work in lbm R410A to lbm air, the specific work of the refrigerant compressor should be divided by the heat absorbed by the refrigerant then multiplied by the heat rejected by the air.

$$\frac{w_{VCRC}}{q_{evaporator}} * q_{air} = w_{air}\left(\frac{Btu}{lbm_{air}}\right), \qquad (Eq. 7.31)$$

As a side note, the heat extracted by the refrigerant divided by the specific work of the refrigerant compressor is the coefficient of performance (COP) of the refrigeration cycle. Therefore, the specific work of the refrigerant compressor in lbm air is the heat rejected by the air divided by the coefficient of performance.

$$COP = \frac{q_{R410A}}{w_{R410A}}, \qquad (Eq. 7.32)$$

$$w_{air} = \frac{q_{air}}{COP}, \qquad (Eq. 7.33)$$

The VCRC compressor's specific work is converted in terms of lbm air to calculate the net specific energy when combining the VCRC with the TSC. The two components consuming specific work are the TSC and the refrigerant compressor. The refrigerant compressor's work varies according to the parametric study. The VCRC evaporator cools the air below the initial dew point so that moisture drains, and the air temperature is decreased. The TSC draws cooler air with a lower intermediate dew point than without the VCRC. Thus, the total specific work of the conditioned two-stage air compressor (CTSC) is the sum of the specific work of VCRC compression in lbm air and the cooled inlet two-stage air compressor (CIC).

$$w_{CTSC} = w_{air} + w_{CIC}, \qquad (Eq. 7.34)$$

The CTSC specific work is compared both to the ideal two-stage compressor (ITSC) specific work, and the actual two-stage compressor (ATSC) specific work. The percent differences are all compared relative to ITSC specific work because it is the baseline with which gains or losses were initially referenced. If the difference exceeds the additional specific work resulting from intercooler buffering, the CTSC package would be economically viable. Additional costs of artificial demand and downtime would also provide justification, but those are outside the scope of the study.

7.6 Desiccant Wheel Modeling

The desiccant wheel is a belt-driven ring filled with a fluted, solid desiccant adsorbent (SDA) coated substrate that rotates its face so that a moist airstream continually charges the passing flutes with moisture, provided the pores have low partial vapor pressure. After passing the moist airstream, the flutes meet a reactivation airstream that will heat the condensate within the SDA pores, vaporizing some, and increase its partial vapor pressure, and if the partial vapor pressure of the SDA's moisture exceeds that of the airstream, the moisture migrates to the reactivation airstream, recharging the desiccant wheel's capacity to remove moisture.

In the steady state the mass of dry air passes through the wheel at a constant rate, the mass of dry air entering is equal to that exiting.

$$\dot{m}_{da_{in}} = \dot{m}_{da_{out}} = \dot{m}_{da}, \qquad (Eq. 7.35)$$

The SDA coating dehumidifies the atmospheric air by adsorbing water vapor to its surface then drawing the condensate into its pores with capillary action.

$$\dot{m}_{wv_{in}} = \dot{m}_{wv_{out}} + \dot{m}_{f_{out}}, \qquad \dot{m}_{f_{out}} = \dot{m}_{da}(\omega_{in} - \omega_{out}), \qquad (Eq. 7.36)$$

Adsorption of water vapor into condensate releases the heat within the water vapor to the dry air, increasing its temperature. However, the heat of sorption is proportional to the quantity of moisture condensed, which depends on the properties of the SDA.

$$q_{sorption} = \omega_{in} h_{g_{in}} - \omega_{out} h_{g_{out}}, \qquad (Eq. 7.37)$$

The wheel rotates slowly at speeds such as 20-30 rotations per hour. Thus, the specific work of the motor driving the wheel is considered negligible. Return air passes a heater at steady state so that the mass of dry air entering is equal to the mass of dry air exiting.

$$\dot{m}_{da_{in}} = \dot{m}_{da_{out}} = \dot{m}_{da}, \quad (Eq. 7.38)$$

A heater increases the temperature of return air so that it can regenerate the desiccant. However, because moisture remains constant during the heating, it is eliminated from the heat balance.

$$\dot{m}_{wv_{in}} = \dot{m}_{wv_{out}} = \dot{m}_{wv}, \qquad \dot{m}_{da}(\omega_{in}) = \dot{m}_{da}(\omega_{out}), \qquad (Eq. 7.39)$$

$$q_{heating} = h_{da_{out}} - h_{da_{in}}, \qquad (Eq. 7.40)$$

The heated return air, called reactivation air, passes another portion of wheel's face and heats the SDA's condensate, which absorbs heat, vaporizes, and joins the passing airstream. Again, the mass of dry air entering is equal to the mass of dry air exiting.

$$\dot{m}_{da_{in}} = \dot{m}_{da_{out}} = \dot{m}_{da}, \qquad (Eq. 7.41)$$

The vaporized condensate joins the reactivation airstream, which has moisture to begin with.

$$\dot{m}_{wv_{in}} + \dot{m}_{f_{in}} = \dot{m}_{wv_{out}}, \qquad \dot{m}_{f_{in}} = \dot{m}_{da}(\omega_{out} - \omega_{in}), \qquad (Eq. 7.42)$$

Because the condensate absorbs some of the reactivation air's heat during vaporization without any work done by or onto the system, the theoretical steady energy balance is as follows.

$$-\dot{Q}_{out} = \dot{m}_{da}\left(h_{da_{out}} - h_{da_{in}} - h_{g_{in}}(\omega_{out} - \omega_{in})\right), \qquad (Eq.\ 7.43)$$

$$q_{evaporation} = h_{da_{in}} - h_{da_{out}} + h_{g_{in}}(\omega_{out} - \omega_{in}), \qquad (Eq. 7.44)$$

To evaluate the system of equations, the location of equilibrium states must be defined and with sufficient intrinsic properties are required. As a mixture, humidity, pressure, and temperature are selected to define any states. The first state corresponds to the outdoor air that enters the desiccant wheel, and second state corresponds to the process air that exits the desiccant wheel. The third state defines the state of the return air that enters the heater, the fourth state defines the state of the reactivation air that exits the heater and enters the regeneration portion of the wheel, and the fifth state defines the state of the release air that exits the wheel from the reactivation airstream to the discharge area. The desiccant wheel is positioned in the dehumidified compressor package so that the process air enters the TSC. Waste heat from the intercooler, aftercooler, and possibly the motors, could potentially supplement the desiccant heater to reduce the amount of energy consumed by the heater from external sources. The heat transfer of those devices however, is outside the scope of this study. Therefore, the primary specific energy consumers are the reactivation heater, and air compressor motor.

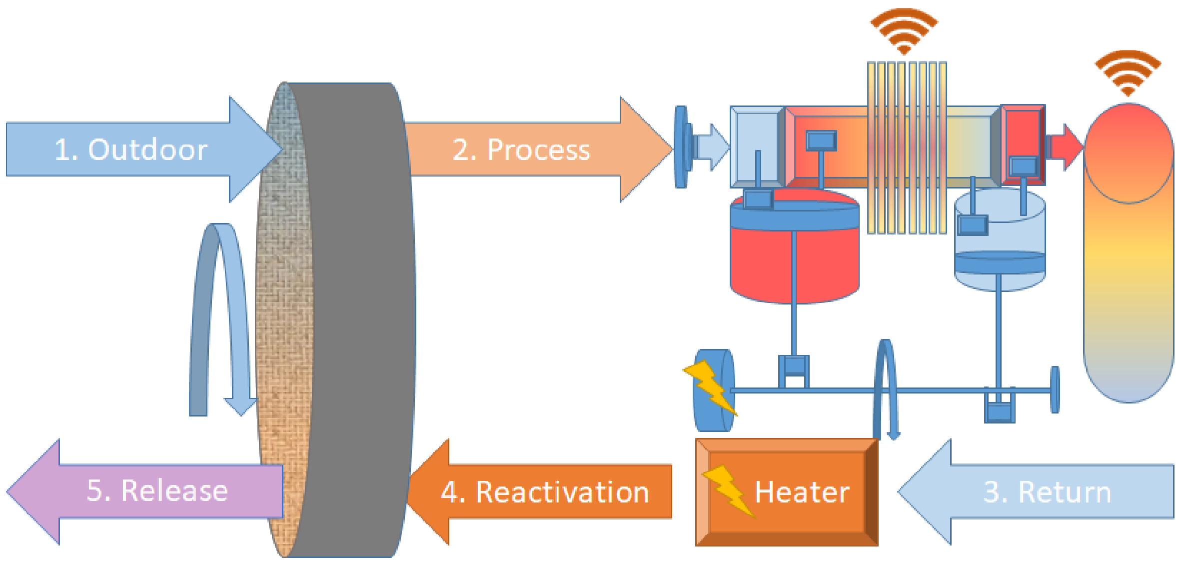

Figure 48: Desiccant Wheel Air Compressor Package

The parametric study evaluates ambient outdoor air from a temperature ranging from 70 to 110 °F at a relative humidity ranging from 30 to 90% at an atmospheric pressure of 14.7 psia. The return air is assumed to come from an indoor room at a temperature of 75 °F and relative humidity of 50%. Although the states can be defined, the change in moisture depends on the properties of the SDA. Desiccants have properties that vary by material, production process, and manufacturer. The heat of sorption includes the heat of condensation and chemical heat that depends on desiccant composition and target specific humidity (ASHRAE, 2016). The heat of sorption is unknown, so it is not possible to model the wheel without assuming the amount of moisture removal, which defeats the purpose of the model.

Behavior of common desiccants including silica gels and molecular sieves differ according to chemistry and manufacturer specification. While a widespread practice is to assume 110% the latent heat of vaporization (Fischer, 2011), manufacturers such as NovelAire Technologies (NovelAire) provide more information. NovelAire develops different configurations of air quality equipment for original equipment manufacturers. Their primary products are energy conservation wheels, desiccant wheels, and dehumidifiers. Their simulation software helps end-users decide an appropriate material, size, and configuration of silica gel, synthetic zeolite, and low-temperature regeneration desiccant wheels.

While NovelAire's software does not specify the heat of adsorption, it allows users to identify results of a single test point among several factors such as rotational speed, airflow rate, outdoor and reactivation air conditions, and heating temperature. Rather than collect a data point for each increment of the parametric study, results from NovelAire's software are collected in the extremes of the parametric study ranges so the data can be fit with a least-squares regression. The regression determines significance of factors and corresponding coefficients to approximate a response variable, such as moisture removal and resulting process air temperature. The benefit of modeling the software is to re-use the coefficients in EES to rapidly conduct large-scale parametric studies, instead of single point evaluations. The response variables that are relevant to the TSC include the resulting specific humidity and resulting process air temperature. In addition, the specific heat to increase the temperature of the return air to regenerative levels will be collected and compared to the air's enthalpy change to validate the regression's ability to model the phenomena.

7.7 NovelAire Desiccant Wheel Software Regression

7.7.1 Overview

NovelAire refers to reactivation air as regeneration air and that terminology will be continued from this point forward to prevent confusion. The NovelAire Desiccant Wheel Simulation Program (simulation) settings allow the user to select units and variables for moisture, regeneration characterization, wheel type, and size. Relative humidity is chosen to define the moisture of the outdoor air, the ratio of regeneration to process air is chosen for the regeneration capacity, and humidity ratio in pounds water per pounds air is chosen to define the moisture of the process air.

Relative humidity is selected for input and specific humidity for output because the parametric study for the TSC uses relative humidity to calculate specific humidity, which is used to determine pressure dew points. Regeneration ratio is selected because most texts (ASHRAE, 2016) mention the ratio of one to three regardless of the flowrate. The smallest size is selected to reduce the footprint of the compressor package.

The WSG and LT3 wheel types are explored, which are silica gel and synthetic zeolite, respectively. Another desiccant type listed in the simulation software is NAC, yet NovelAire does not describe NAC in their brochure, so it is not tested. Each desiccant is tested separately because WSG dehumidifies air for relative humidity of 60% or greater, while LT3 is for deep drying of air with relative humidity less than 50% (NovelAire Technologies, 2004). Furthermore, 81 silica gel's performance varies, while molecular sieves tend to be very consistent (ASHRAE, 2017).

7.7.2 Brainstorm for Factorial Experiment

Following the settings, nine inputs in the simulation software for each type of desiccant, including outdoor air flowrate (scfm), outdoor air dry-bulb temperature, outdoor moisture, ratio of regeneration airflow to outdoor airflow, regeneration coverage, return air dry-bulb temperature, return air moisture, regeneration air heating temperature, and rotations per hour. A two-level factorial experiment with k factors requires 2^k runs to estimate the main and interaction effects. Because conducting 2^9 trials is impractical, a brainstorm to reduce factors to be tested follows. Two factors are eliminated by assuming the return air comes from the conditioned environment indoors at a temperature of 75 °F and 50% relative humidity. In addition, the scfm depends on the application, and rotational speed would be directly related, so these are kept at default values, eliminating another two factors.

NovelAire suggests that lesser dehumidification can use the 50/50 regeneration/process coverage, and while deep drying should use 25/75. However, there is no statement regarding its relation to the airflow ratio. Some suggest a ratio regeneration to process air of at least 1/1.3 and others, 1/3 (ASHRAE, 2016). Therefore, both the airflow coverage and regeneration ratio are tested at the previously mentioned extrema. The desiccant wheel is used to condition outdoor air and follows the ranges of the other parametric studies, a temperature ranging from 70 to 110 °F and relative humidity ranging from 30 to 90%. Furthermore, the simulation only works with heating temperature between 150 °F to 347°F, so the range of regeneration temperature chosen is 175 to 325 °F. The brainstorm reduces the number of factors to five, resulting in 32 trials per wheel type, and 64 trials total.

7.7.3 Background of Factorial Experiments

The significance of each factor in a two-level experiment is approximated with normalized, linearly independent factors, and the significance provides criteria for selecting variables of a model developed from a linear regression (Wu & Hamada, 2009). To normalize the factors, they are assigned a coded variable, which may be an arbitrary letter, as well as a value of positive one (+1) or negative one (-1) for the high and low level, respectively. For example, the factor that is outdoor air temperature can be assigned an arbitrary coded variable such as the letter A, and the high level of outdoor air temperature that is 110 °F codes with +1, and the low level of outdoor air temperature that is 70 °F codes with -1.

Actual variables are assigned coded variables that are typically letters of the alphabet so that they are easy to refer to when defining them in a numerical analysis. In a linear regression, not only the factors, but also their interaction can affect the observed response. Thus, with the actual variables changed to coded variables, pairs of factors in the experiment can easily be organized. The pairs of factors must have an equal number of pairs of levels to prevent the model from skewing towards any factor. Thus, the pairs of levels between two factors (1, 1), (1, -1), (-1, 1), and (-1, -1) must have the same number of trials in the experiment, for every possible pair of factors. To summarize, each factor takes the value +1 or -1 in a trial, with an equal number of high and low levels and equal number of level-pairs in the experiment.

To further reduce the number of trials in the experiment, a fractional factorial approach is used, which reduces trial count by aliasing a main effect with multiple-factor interaction-effects. Aliasing means that the effects of some high-level interactions cannot be estimated because it is coupled with a lower level interaction or main effect. In most cases, the number of significant effects is small, main effects are more significant than interaction effects, and an interaction effect should only be significant if one of its parent factors is significant. Thus, aliasing can still approximate the effect of different factors if the main effects are strongly-clear or at least clear. Fractional factorial experiments and aliasing is explained in the next subsection.

7.7.4 Background of Fractional Factorial Experiments

Outdoor air temperature, outdoor air relative humidity, heating temperature, regeneration ratio, and regeneration area are labeled with coded variables A, B, C, D, and E, respectively. The product of the coded values of all factors is the value of the so-called design generator, which is generally assigned the letter I. As a result, the design generator also has a high and low code of +1 and -1. For example, if in a trial, the coded variables A, C, and D take the value of +1, and B and E take the value of -1, the product of all letters, the design generator, is equal to +1. The level of the design generator depends on the product of the levels of all the factors.

A fractional factorial experiment ignores one of the levels of the design generator to halve the number of experiment trials at the consequence of aliasing effects, which neglects higher-order interaction effects of multiple factors. By multiplying the design generator by the code of one of the factors, the factor’s contribution to the product is squared and aliased. For example, multiplying the design generator with the coded value for factor A will alias A with BCDE. Returning to the previous example, if A is equal to positive one, multiplying the design generator by positive one results in positive one, which is equal to the product of B, C, D, and E. When an effect is aliased, its contribution to the product of the levels of all the effects becomes squared, so that regardless of whether the value is positive or negative one, its product is positive one and the remaining factors multiply with unity.

An effect is strongly clear if it aliases with a four-level interaction or more, but only clear if with three-level interactions. The main effects of a 2^{5-1} experiment are strongly clear because the design generator is equal to the product of A, B, C, D, and E so that A is equal to the product of four factors, B, C, D, and E. Similarly, B is equal to the product of A, C, D, and E and the other main effects alias with four-level interactions. The two-factor interaction effects are only clear because the product of A and B is equal to the product of C, D, and E, and the other two-factor interactions alias with three-level effects. The following section covers the design of the fractional factorial experiment to determine significance of factors and their coefficients in the regression model.

7.7.5 Fractional Factorial Experiment for Significance

The factors outdoor air temperature, outdoor air relative humidity, heating temperature, airflow ratio, and airflow coverage, are assigned letters A, B, C, D, and E, respectively, and given a high and low level in accordance with this parametric study (Table 1).

Table 1: Coded variables for high and low level of factors of a fractional factorial experiment

	Temperature	Humidity	Heating	Ratio	Coverage
Code	A	B	C	D	E
+1	110 °F	90%	325 °F	1/3	50%
-1	70 °F	30%	175 °F	1/1.3	25%

A 2^{5-1} fractional factorial experiment is created by aliasing E with ABCD; the choice of factor E is arbitrary and any other is equally viable. Every other factor, namely A, B, C, and D, has an equal number of trials with the high and low level, as well as an equal number pairs of high and low levels with the other factors. Therefore, a 2^{5-1} (16) run experiment, should have eight trials with a high level of each factor, and the remaining eight with the factor’s low level. A cascading, alternating pattern for the level of the coded variables is organized in a table to calculate the level of the aliased variable E, which is the product of the level of factors A, B, C, and D for each trial. Finally, after organizing the design matrix, its coded values are translated to the actual values to create a planning matrix for the actual settings of factors during each trial of the experiment. An example of the cascading alternating pattern is shown by Table 2.

Table 2: Example of a 2^{4-1} design matrix with a cascading, alternating pattern

A	B	C	D=ABC
1	1	1	1
1	1	-1	-1
1	-1	1	-1
1	-1	-1	1
-1	1	1	-1
-1	1	-1	1
-1	-1	1	1
-1	-1	-1	-1

7.7.6 Randomization and Least-Squares Regression

To prevent learning bias during experimentation, the rows of the planning matrix are numbered and randomized, the experiment is conducted in the random order, and the results reorganized using the run number. From the desiccant wheel simulation and the actual values of the five factors, the outlet specific humidity, outlet temperature, and Btu/h are collected. Heater Btu/h is converted to Btu/lbm with the actual volumetric flow rate and air volume at the state defined by the trial settings to calculate the specific energy and determine the net effect with a TSC. A least-squares regression is used to find the coefficients of the factors and their interactions, and their significance investigated to simplify the model.

With an orthogonal design matrix, a column of ones is included to calculate the zero-offset or intercept of the function, and only first and second order terms, which are the main effects and two-factor interactions, are considered. The least squares regression solves the following equation, where are the regression coefficients are organized by element of column matrix *C*, the levels of each factor organized by trial in the columns of the design matrix *R*, and the collected responses such as outlet humidity or temperature are organized by trial into the columns matrix *y*.

$$C_i = (R^T * R)^{-1} * R^T * y_i, \qquad (Eq. 7.45)$$

The design matrix of coded variables, as well as the observed responses for outlet specific humidity, outlet temperature, and heater Btu/lbm in the form of column matrix organized by trial are compiled in MATLAB, then the coefficients corresponding to the factors of the experiment are collected for each response variable. Trials are not replicated because the computer simulation produces the same result when using identical settings and inputs. Without replication, analysis of variance does not provide the significance of each effect. Therefore, significance is instead determined by Lenth's method with the individual error rate (IER) test-statistic, and the p-value of the IER test-statistic.

7.7.7 Lenth's Method for the Screening Experiment

Lenth's method compares the absolute value of the pseudo standard error (PSE) test-statistic with the individual error rate (IER) test-statistic. There are fifteen effects to consider, namely the five main effects A, B, C, D, and E, as well as the ten two factor interactions AB, AC, etcetera. Factor effects (E) are equal to twice their regression coefficients, and to calculate the PSE test-statistic, the PSE of the response is calculated, and each of the response's effects is divided by the PSE (Wu & Hamada, 2009).

$$E_i = 2 * C_i, \qquad (Eq. 7.46)$$

The IER test-statistic corresponding to a 5% significance level and 15 effects is 2.16, so the absolute value of the PSE test-statistics that exceed 2.16 are declared significant.

$$IER_{\alpha=0.05,\ n=15} = 2.16, \qquad (Eq. 7.47)$$

In addition, the absolute value of the PSE test statistic can be evaluated with a MATLAB student t-distribution function to find the p-value at a 5% significance level. To calculate the PSE, the initial standard error (s_0) must be calculated, and it is 1.5 times the median of the absolute values of the effects.

$$s_0 = 1.5 * median(|E_i|), \qquad (Eq. 7.48)$$

The criteria (c) for PSE is the absolute value of the effects less than 2.5 times the s_0.

$$c = |E_i| < 2.5 * s_0, \qquad (Eq. 7.49)$$

The median of those that meet the criteria must be calculated, and the PSE is 1.5 times the result.

$$PSE = 1.5 * median(c), \qquad (Eq. 7.50)$$

The PSE test-statistic of each factor in the design matrix is the effect of each factor divided by the PSE.

$$t_{i_{PSE}} = \frac{E_i}{PSE}, \qquad (Eq. 7.51)$$

The factors that have a significant impact on the regression model are those whose absolute PSE test-statistic is greater than the IER test-statistic. Each response has its own set of regression coefficients, effects, s_0, and PSE.

$$|t_{i_{PSE}}| > IER_{\alpha=0.05,\ n=15}, \qquad (Eq. 7.52)$$

The experiment is conducted using a fractional factorial design, and the significance of each effect is evaluated with Lenth's method of calculating the PSE test-statistic and the corresponding p-value. The significant effects are A, B, C, and in some cases, factor D, which are outdoor air temperature, outdoor air moisture, heating temperature, and the airflow ratio, respectively. The summary of significant effects in the screening experiment is organized in Table 3.

Table 3: Screening Experiment Significant Factor Evaluations

	WSG T_O	WSG ω_O	WSG q_h	LT3 T_O	LT3 ω_O	LT3 q_h
t_{PSE}	A, B, C, D	A, B, C, AB	C	A, C, D	A, B, C, D, AB	C, D, CD
p_{PSE}	A, B, C, D	A, B, C, AB	C	A, C, D	A, B, C, D, AB	C, D, CD

7.7.8 Empirical Models Composed of the Significant Effects

The regression coefficients result from the least-squares regression using the coded values in the design matrix. Only regression coefficients corresponding to significant factors are included the model. While the regressions apply to the parametric study range, the wheel types perform differently. The WSG wheel is best suited for environments with relative humidity greater than 60%, and the LT3 wheel best for those where relative humidity is less than 50%. The models are only valid within ambient temperatures from 70 to 110 °F and relative humidity from 30 to 90%, as well as heating temperature ranging from 175 to 325 °F. Furthermore, parametric studies need to be performed using code values of positive or negative one, resulting in a response in the actual expected magnitude. For example, using coded values of positive or negative one for the coded variables in the model will result in an actual value of specific humidity, outlet temperature, or heating energy. Recall that the process air is the outdoor air dehumidified by the desiccant wheel.

The empirical model for the specific humidity of the process air exiting the WSG wheel is as follows.

$$\omega_{out_{WSG}} = 10^{-3} * (15.15 + 11.1 * A + 8.74 * B - 1.34 * C + 6.19 * CD), \qquad (Eq. 7.53)$$

The empirical model for heating the return air to the state of reactivation air prior to entering the WSG wheel is as follows.

$$q_{WSG} = 41.25 + 17.68 * C, \qquad (Eq. 7.54)$$

The empirical model for the temperature of the process air exiting the WSG wheel is as follows.

$$T_{out_{WSG}} = 137 + 22.51 * A + 8.825 * B + 16.3 * C - 5.15 * D, \qquad (Eq. 7.55)$$

The empirical model for the specific humidity of the process air exiting the LT3 wheel is as follows.

$$\omega_{out_{LT3}} = 10^{-3} * (15.77 + 11.4 * \mathrm{A} + 9.69 * \mathrm{B} - 2.27 * \mathrm{C} + \mathrm{D} + 6.96 * \mathrm{AB}), \qquad (Eq. 7.56)$$

The empirical model for heating the return air to reactivation air entering the LT3 wheel is as follows.

$$q_{LT3} = 41.21 + 17.66 * C - 0.1738 * D - 0.00745 * CD, \qquad (Eq. 7.57)$$

Lastly, the empirical model for the temperature of the process air exiting the LT3 wheel is as follows.

$$T_{out_{LT3}} = 140.3 + 21.79 * A + 21.43 * C - 7.681 * D, \qquad (Eq. 7.58)$$

8. RESULTS AND COMMENTARY

8.1 Overview

This section reviews the results of the vapor compression refrigeration cycle (VCRC) and the desiccant wheel according to the methodologies described earlier. Each dehumidification technology (dehumidifier) removes moisture from the atmospheric air (air) and changes the state of the air. To measure the specific energy consumption of each system, parametric studies are conducted for each dehumidifier, as well as for the compression of the corresponding dehumidified air using a two-stage air compressor (TSC). The total specific energy consumption of the package combining the dehumidifier and the dehumidified TSC is compared to the specific energy consumed by the standalone TSC using air at the same initial condition.

8.2 Vapor Compression Refrigeration Cycle

To avoid freezing the airborne water vapor, the vapor compression refrigeration cycle (VCRC) cools the air passing over the coils of the evaporator to a temperature of 45 °F using refrigerant 410A (R410A). For inlet air temperatures ranging from 70 to 110 °F and relative humidity ranging from 30 to 90%, the parametric study evaluates the heat rejected by the air, the remaining moisture in the air after condensation, the heat absorbed by the R410A, the specific work of the R410A compressor, and the heat rejected by the R410A. The remaining moisture in the air after passing the evaporator coils is as follows. Except for the case of inlet air temperature of 70 °F and 30% relative humidity, in which case the specific humidity is 0.00465, the cooling temperature of 45 °F is below the dew point of all test points in the parametric study, resulting in a specific humidity of 0.00631.

The VCRC circulates R410A from the evaporator at a temperature of 10 °F below the cooling temperature of 45 °F resulting in a temperature of 35 °F, to 10 °F above the ambient air temperature in the condenser, which would be a temperature ranging from 80 to 120 °F. The saturated vapor pressure of R410A at 35 °F is 122 psia, while the saturated liquid pressure corresponding to a temperature ranging from 80 to 120 °F ranges from 251 to 435 psia. The ideal temperature-entropy (Ts) diagram for refrigerant operating in an outdoor air temperature of 70 °F and 110 °F is plotted in Figure 49.

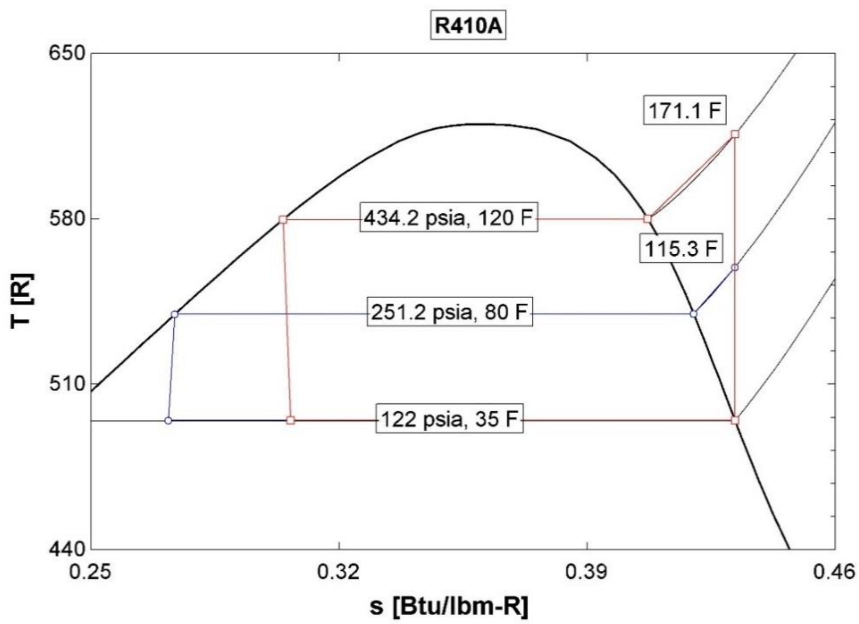

Figure 49: Ideal R410A vapor compression refrigeration cycle Ts diagram comparing air entering at 70 °F with 110 °F

Figure 49 is the overlay of two different R410A cycles, namely for inlet air temperature of 70 °F as well as 110 °F. To cool the air to 45 °F, the refrigerant is at 35 °F and its saturated vapor pressure is 122 psia. In the ideal case, the R410A enters the isentropic refrigerant compressor at the state designated by the point on the saturated vapor line. The refrigerant compressor compresses the R410A until it reaches the isobar designated by the saturated liquid pressure corresponding to 10 °F above the outdoor air temperature. As shown by Figure 49, the discharge pressure of R410A at 120 °F is nearly twice as high as that at 80 °F. Because the refrigerant compressor is isentropic, the entropy from its inlet to its discharge is constant and appears as a vertical line on the Ts plot. The heat absorbed by the refrigerant is the area below the evaporation process line to the x-axis, which is the horizontal line inside the vapor dome at 35 °F. The heat rejected by the refrigerant after being compressed is the area below the condenser process line to the x-axis, which is the horizontal line at 80 or 120 °F. The work of the compressor is the heat absorbed by the refrigerant subtracted from the heat rejected by the refrigerant, which is the area under the evaporation process subtracted from the area of the area under the condensation process.

To compare the ideal VCRC with an actual VCRC, the refrigerant compressor is evaluated with 80% isentropic efficiency. The actual process differs from the ideal process because it is difficult to precisely control the phase of the substance at the saturated state. Thus, the saturated vapor is superheated by 15 °F to completely evaporate the refrigerant and to prevent wet compression, and the saturated liquid is subcooled by 5 °F to completely condense the refrigerant and to ensure effective throttling. The actual VCRC is plotted in Figure 50. The area beneath the low-temperature vaporization process contour is the heat absorbed by the refrigerant. The area beneath the high-temperature condensation process contour is the heat rejected by the refrigerant.

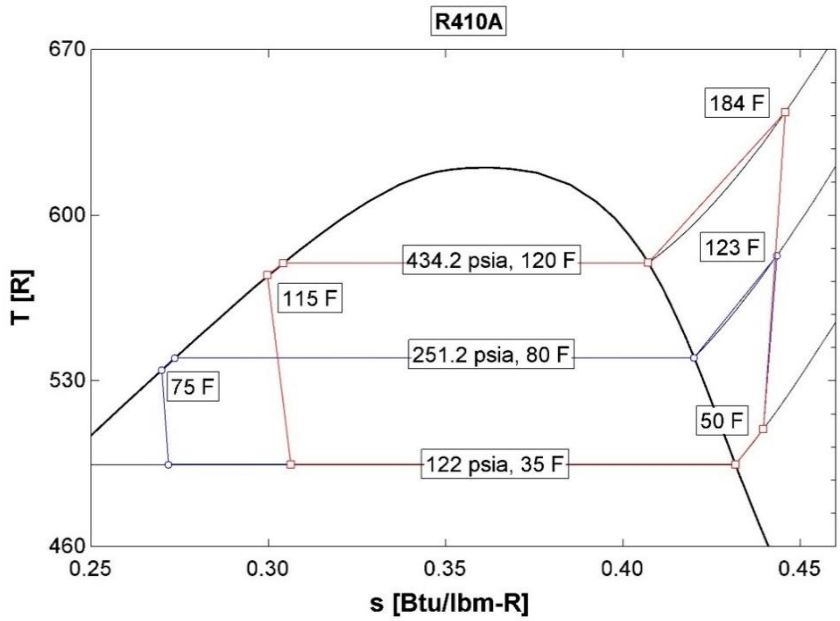

Figure 50: Actual R410A vapor compression refrigeration cycle Ts diagram conditioning air at 70 °F versus 110 °F

As depicted by Figure 50, the main difference between the actual VCRC and the ideal VCRC is that the compressor is no longer isentropic, so the compression process is a slanted line rather than vertical. In addition, the compressor requires an additional 25% specific work due to the 80% isentropic efficiency. Furthermore, the need to superheat the vapor and sub-cool the condensate means that the refrigerant absorbs more heat before entering the refrigerant compressor and rejects more heat before entering the throttling valve. In both the ideal case and actual case, the difference between the heat rejection and absorption is equivalent to the specific work of the refrigerant compressor. In the actual case however, the evaporation process line extends beyond the saturated vapor line until the R410A reaches the superheated temperature, and the condensation process line extends beyond the saturated liquid line until reaching the subcooled temperature. The coefficient of performance (COP) of the refrigeration cycle is the ratio of heat removed from the air by refrigerant to the specific work consumed by the refrigerant compressor. The values of the parametric study are tabulated in Table 4, and the amount of heat absorption (qL) and rejection (qH) are both plotted as a function of inlet air temperature in Figure 51.

Table 4: Specific energy and COP of an 80% efficient R410A VCRC

Temperature (°F)	qH (Btu/lbm)	qL (Btu/lbm)	w (Btu/lbm)	COP
70	94.09	83.03	11.06	7.508
80	92.32	78.99	13.33	5.924
90	90.35	74.81	15.55	4.812
100	88.18	70.47	17.70	3.981
110	85.76	65.95	19.81	3.329

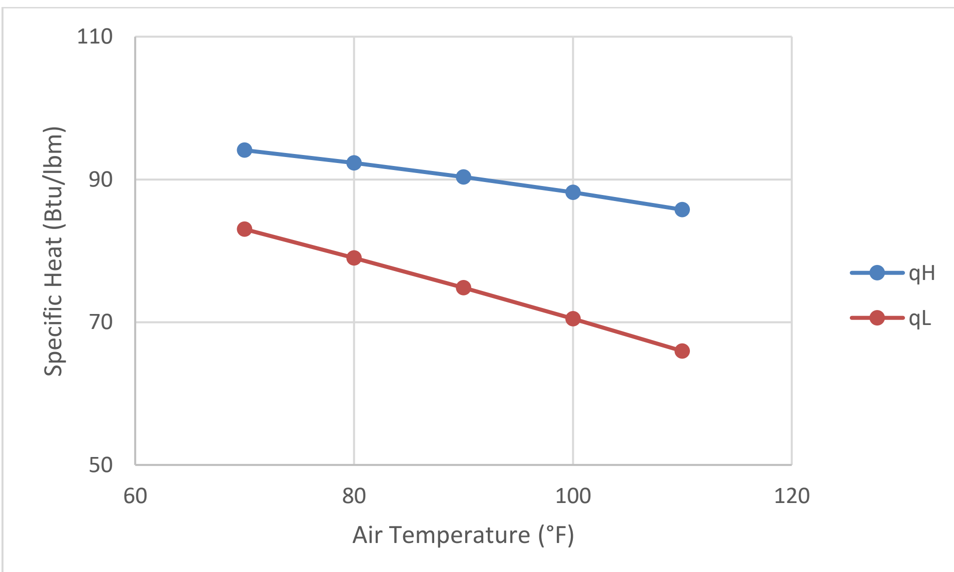

Figure 51: Actual R410A vapor compression refrigeration cycle specific heat rejected by refrigerant from condenser (blue) and specific heat absorbed from the air in the evaporator (red)

The specific work of refrigerant compression is expressed per unit mass of R410A. In order to evaluate the net change in energy consumption of the VCRC packaged with the TSC, he specific work must be per pound mass of air. However, because the mass of the R410A charged into the system is designed to absorb a certain amount of heat, the specific work can be expressed per unit mass of dry air by multiplying with the ratio of heat rejected by the air to heat absorbed by the R410A. The heat rejected by the air is equal to the sum of both the saturated liquid enthalpy of water multiplied by the change in specific humidity and enthalpy of the moist air exiting the evaporator coil chamber subtracted from the enthalpy of the moist air entering the evaporator coil chamber; values are plotted as a function of inlet air temperature in Figure 52. Recall that the dew point at an inlet air temperature of 70 °F and 30% relative humidity is less than 45 °F, so in that case, the data is corrected to only account for the change in enthalpy of the air without condensation.

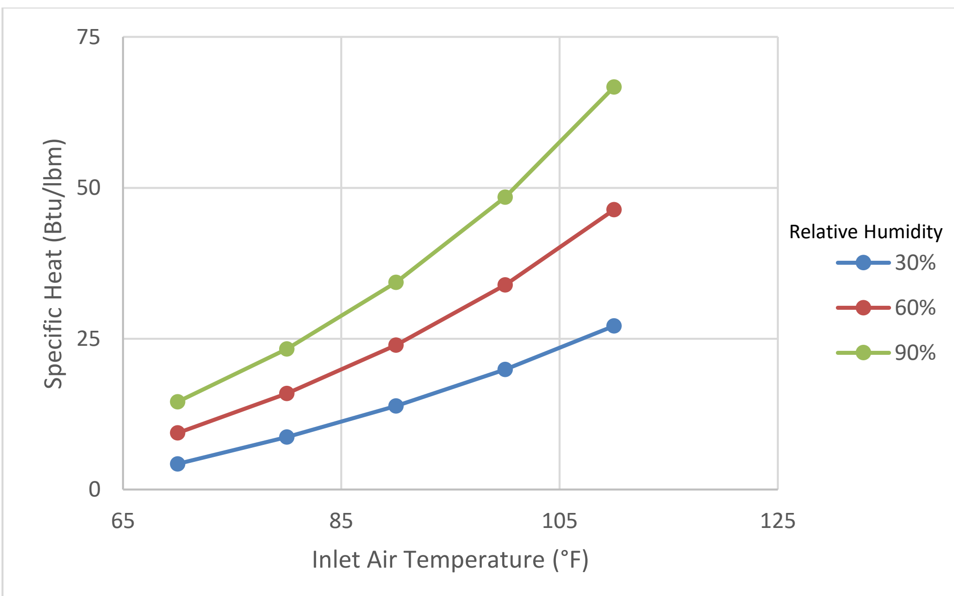

Figure 52: The specific heat rejected by the air passing the evaporator coils of the R410A VCRC with air entering at a temperature ranging from 70 to 110 °F and relative humidity ranging from 30 to 90%

Figure 52 shows the heat rejected by the moist air from an inlet air temperature of 70 to 110 °F and relative humidity ranging from 30 to 90%. As the inlet air temperature increases, the air must reject more heat to reach the cooling temperature of 45 °F. As the moisture in the air increases, the air must reject more heat because the water vapor releases heat during condensation. At an inlet air temperature of 70 °F and 30% relative humidity, the air transfers approximately 4 Btu/lbm of heat to the R410A for its temperature to reach 45 °F. On the other hand, at an inlet air temperature of 110 °F and relative humidity of 90%, the air transfers approximately 65 Btu/lbm to the R410A to reach the same cooling temperature. Figure 53 shows the specific work of the refrigerant compressor per unit mass of dry air that is calculated by using the ratio of heat rejected by the air to the heat absorbed by the R410A.

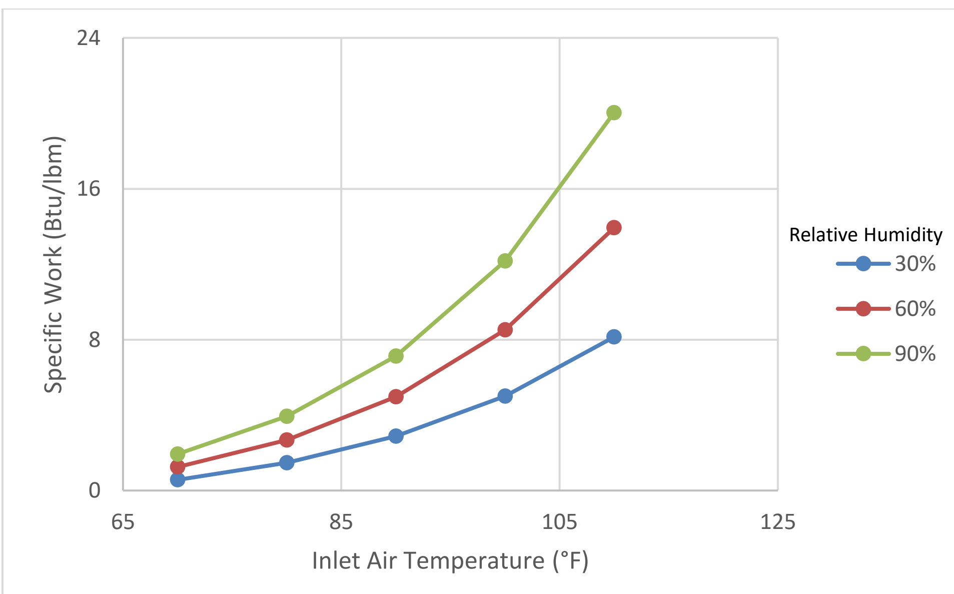

Figure 53: Specific work of R410A compression per unit mass of dry air for an inlet air temperature ranging from 70 to 110 °F and relative humidity ranging from 30 to 90%

Figure 53 shows the specific work of R410A compression per unit mass of dry air for the values of this parametric study. Multiplying the specific work of R410A compression per unit mass of R410A with the ratio of heat rejected by the air to the heat absorbed by the R410A is equivalent to dividing the heat rejected by the air by the COP of the cycle, which is why the specific work of R410A compression per unit mass air is much less than the specific heat rejected by the air. The specific work of the refrigerant compressor must be added to the specific work of the TSC using the cooled air to determine the net specific energy of the package and compare with the standalone TSC.

Except for the case of 70 °F and 30% relative humidity where the dew point temperature is approximately 37 °F, the specific humidity is 0.004647, and the partial vapor pressure is 0.109 psia, the other points of the parametric study result in air that is conditioned to a dry-bulb and dew point temperature of 45 °F with a specific humidity of 0.006308 and a partial vapor pressure of 0.1476 psia. The two-stage compression of conditioned air (CTSC) is parametrically studied with discharge pressure ranging from 75 to 150 psig and intercooling buffered by 5 °F above the intermediate dew point. The CTSC specific work is calculated and added to the VCRC specific work, both of which are per unit mass dry air. Figure 54, Figure 55, and Figure 56 show the difference from the ideal two-stage compressor specific work at 30, 60, and 90% relative humidity, respectively.

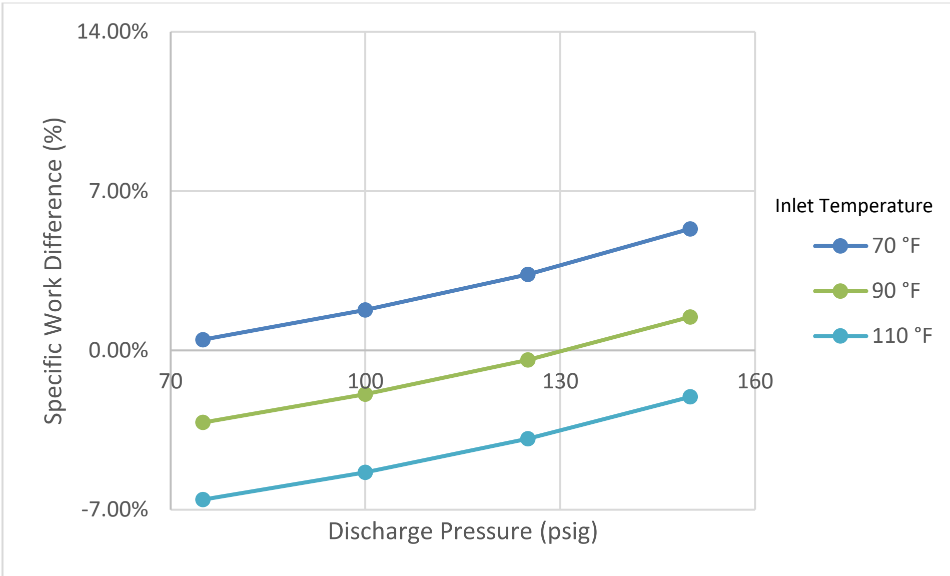

Figure 54: Specific work variation of conditioned two-stage air compression at 45 °F from ideal case with inlet air temperatures ranging from 70 to 110 °F and 30% relative humidity with discharge pressure ranging from 75 to 150 psig

Figure 54 indicates that for air initially at a temperature 70 °F and the relative humidity is 30%, the specific work of ideal two-stage air compression (ITSC) is less than the specific work of CTSC, regardless of the discharge pressure. On the other hand, if the inlet air temperature is 90 °F and the relative humidity is 30%, the CTSC package consumes up to 3% less specific work, except when the discharge pressure is greater than 130 psig. If the inlet air temperature is 110 °F and the relative humidity is 30%, then the CTSC consumes less specific work than the ideal case for all discharge pressures of the parametric study.

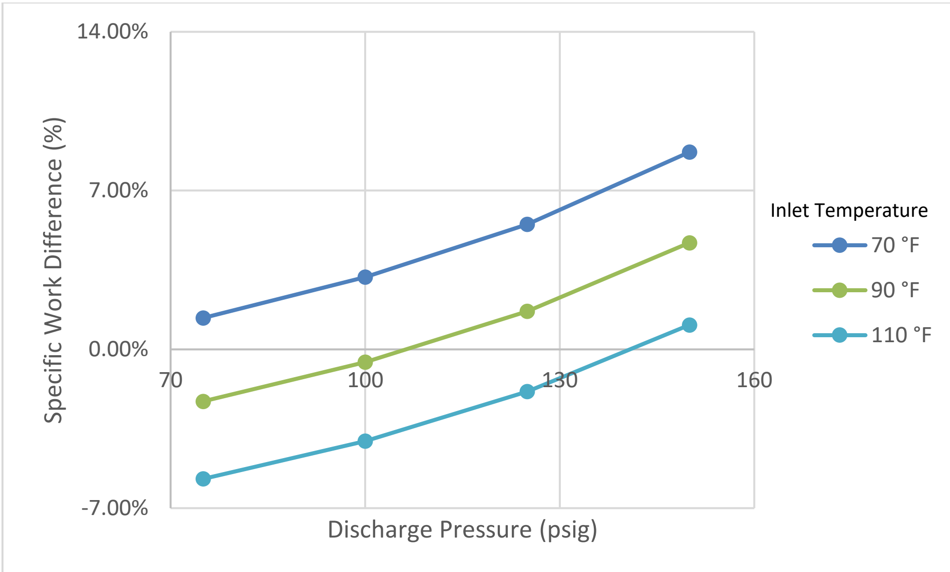

Figure 55: Specific work variation of conditioned two-stage air compression at 45 °F from ideal case with inlet air temperatures ranging from 70 to 110 °F and 60% relative humidity with discharge pressure ranging from 75 to 150 psig

According to Figure 55, for air entering at a temperature of 70 °F and relative humidity of 60%, the CTSC requires more specific work than the ITSC. For air entering at a temperature of 90 °F and relative humidity of 60%, the ideal case requires less specific work than the CTSC if the discharge pressure is greater than 100 psig. If the air at the inlet of the R410A VCRC is at a temperature of 110 °F and relative humidity of 60%, and the TSC is to discharge to a pressure of less than 140 psig, then the CTSC requires less specific work than the ITSC.

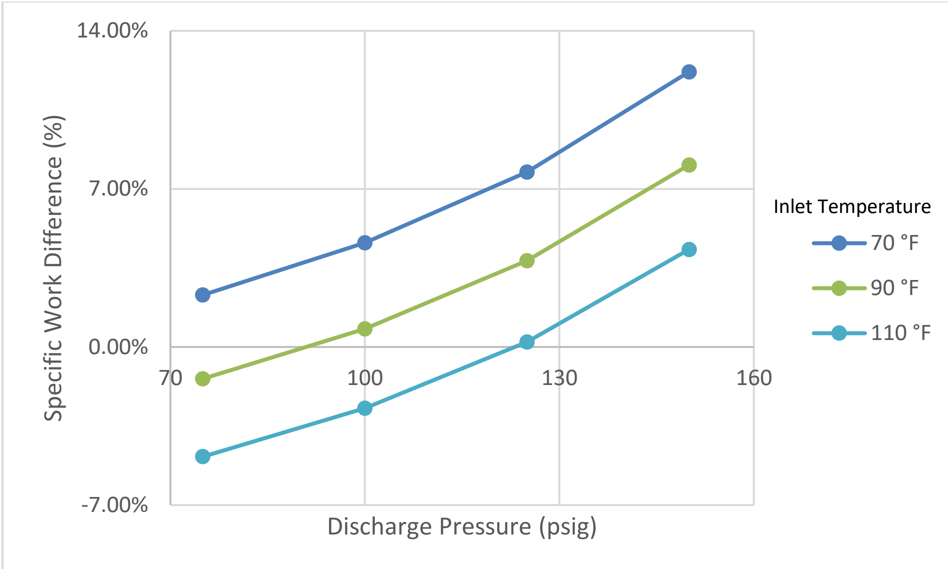

Figure 56: Specific work variation of conditioned two-stage air compression at 45 °F from ideal case with inlet air temperatures ranging from 70 to 110 °F and 90% relative humidity with discharge pressure ranging from 75 to 150 psig

he case of air entering the evaporator of the R410A VCRC at a temperature ranging from 70 to 110 °F and a relative humidity of 90% is exhibited by Figure 56. If the air inlet temperature is 70 °F, the CTSC requires 3-14% more specific work than the ideal case. For an air inlet temperature of 90 °F, the CTSC requires up to 8% more specific work than the ITSC when the discharge pressure exceeds 85 psig. For the inlet air temperature of 110 °F, the CTSC can require up to 5% less specific work than the ITSC if the discharge pressure is less than 125 psig and up to 5% more specific work if the discharge pressure is greater than 125 psig.

Recall that airborne water vapor must release heat to condense. Although significant airborne moisture can condense in a TSC intercooler and result in damage, condensing the water with a VCRC can cost excess specific work of refrigerant compression because the refrigerant must absorb significant heat from the water vapor to condense it. Acknowledging that there is tradeoff between the specific work of CTSC and two-stage compression of moist air (MTSC), the change of specific work from the MTSC to the CTSC is evaluated relative to the moist case to determine whether the specific work of the VCRC exceeds the specific work incurred due to intercooler buffering. Figure 57, Figure 58, and Figure 59 show the difference between CTSC and MTSC at 30, 60, and 90% relative humidity, respectively.

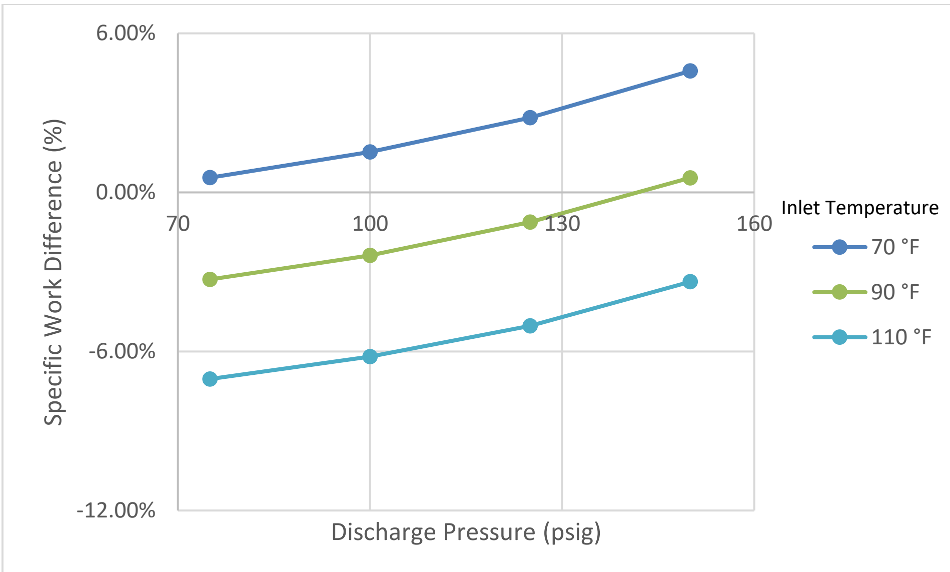

Figure 57: Specific work difference of conditioned two-stage air compression relative to the two-stage compression of moist air at an inlet temperature of 70 to 110 °F at a relative humidity of 30%

In the introduction to MTSC, it was shown that air with relative humidity of 30% has a minor impact on the ability of the intercooler to reach the inlet air temperature, resulting in the increase of specific work by 1-2% when compared to the ITSC. The difference is accounted for by Figure 57, which shows an increasing disparity in specific work difference because the MTSC requires more specific work than does the ITSC. For example, at an inlet air temperature of 110 °F and relative humidity of 30% that is discharged to a pressure of 150 psig, the CTSC requires 2% less specific work than the ITSC, but 3% less specific work than the MTSC because the

MTSC requires 1% more specific work than the ITSC. Therefore, the more specific work incurred due to the presence of airborne moisture, the more feasible becomes the CTSC.

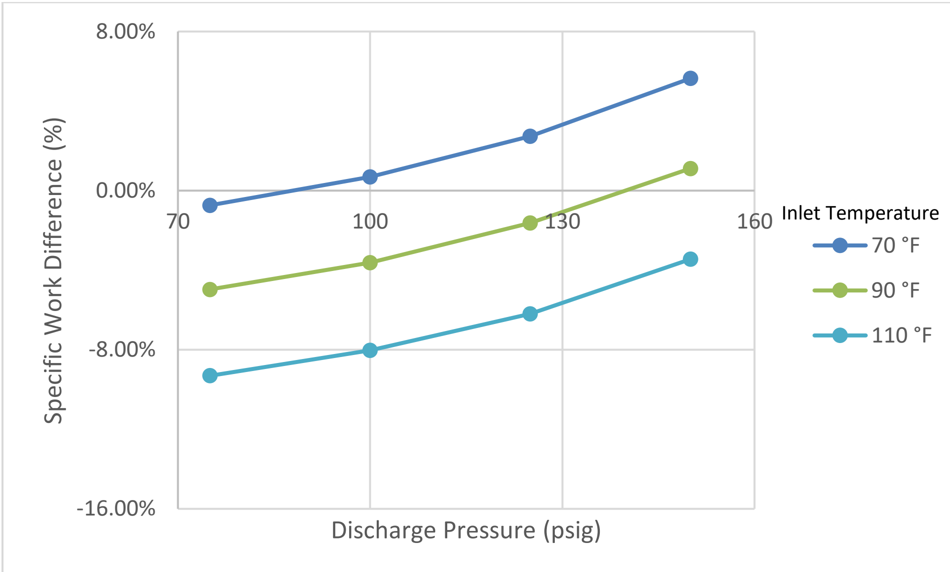

Figure 58: Specific work difference of conditioned two-stage air compression relative to the two-stage compression of moist air at an inlet temperature of 70 to 110 °F at a relative humidity of 60%

Figure 58 shows that for inlet air at temperature of 70 °F and a relative humidity of 60%, the CTSC requires 1% less specific work than the MTSC if the discharge pressure is 75 psig, but up to 6% additional specific work if the discharge pressure exceeds 85 psig. For the inlet air at a temperature of 90 °F and a relative humidity of 60%, the CTSC requires up to 5% less specific work if the discharge pressure is 140 psig or less, and 1% additional specific work if the discharge pressure is 150 psig. In addition, if the inlet air temperature is 110 °F and the relative humidity is 60%, the CTSC requires up to 9% less specific work if the discharge pressure is 75 psig.

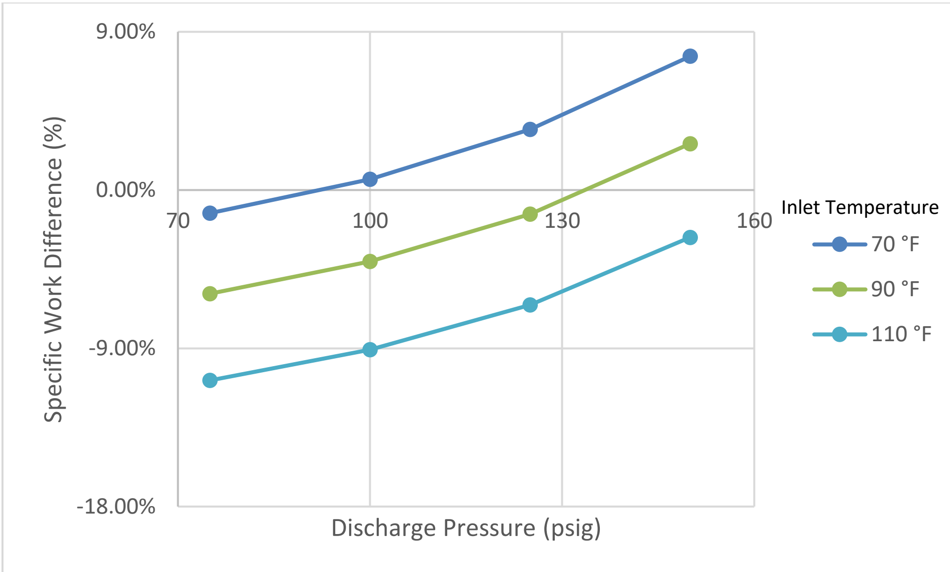

Figure 59: Specific work difference of conditioned two-stage air compression relative to the two-stage compression of moist air at an inlet temperature of 70 to 110 °F at a relative humidity of 90%

When the airborne moisture reaches 90% relative humidity, Figure 59 indicates that specific work of the CTSC can reach up to 11% less than the specific work of the MTSC if the inlet air temperature is 110 °F and the discharge pressure is 75 psig and at least 3% if the discharge pressure is 150 psig. If the inlet air temperature is 70 °F, the CTSC requires 1% less specific work than the MTSC when the discharge pressure is 75 psig, but up to 8% more as the discharge pressure increases to 150 psig. At an inlet air temperature of 90 °F and relative humidity of 90%, the CTSC requires 3% more specific work than the MTSC at a discharge pressure of 150 psig, but up to 6% less as the discharge pressure decreases from 135 psig to 75 psig.

Because discharge pressure has a much more significant impact on the specific work of two-stage air compression than do either inlet air temperature or airborne moisture, the reduction of specific work by dehumidifying the MTSC with the R410A VCRC mainly depends on the discharge pressure. Even at a low relative humidity of 30%, specific work reduction of up to 7% is possible if the air is at a temperature of 110 °F and the discharge pressure is 75 psig. At the same inlet air temperature and discharge pressure at a relative humidity of 90% the specific work of CTSC can be 11% less than the MTSC. In contrast, at a discharge pressure of 150 psig, savings are only possible if the air is at an inlet temperature of 110 °F and 90% relative humidity. At an inlet air temperature of 70 °F and relative humidity of 30%, where the dew point approaches the freezing point of water, the VCRC increases the specific work of the system by 5%.

8.3 NovelAire Desiccant Wheel Experiment

To simplify the experiment and the analysis, the settings kept constant in the simulation include a ratio of regeneration airflow to process airflow of one to three, a return air temperature of 75 °F, and return air relative humidity of 50 %. A silica gel wheel (WSG) was tested with the regeneration air temperatures ranging from 175 to 325 °F and inlet air temperature ranging from 70 to 110 °F with relative humidity ranging from 60 to 90%. The molecular sieve (LT3) desiccant wheel is appropriate for air at a relative humidity of less than 50% and is not tested because the air at 30% relative humidity has a negligible effect on the specific work of two-stage air compression, but the heat of sorption of the LT3 wheel increases the air temperature so that the specific work increases significantly. The parametric studies of WSG are conducted for air entering at a relative humidity of 60 and 90% and the results are as follows.

8.3.1 WSG Dehumidifying Air at a Relative Humidity of 60%

The WSG's ability to dehumidify incoming moist air depends on the partial vapor pressure inside of its pores. If the pores are filled with condensate, then the WSG's ability to adsorb additional moisture will depend on the temperature of the regeneration airstream, which will vaporize the condensate and gradually increase the pore partial vapor pressure. The regeneration airstream has low partial vapor pressure due to its high temperature and will attract the water vapor from the pores when the pore partial vapor pressure exceeds the regeneration air partial vapor pressure. The greater the regeneration air temperature, the more condensate will evaporate from the WSG pores, resulting in a high rate of moisture removal.

Although the heat of sorption increases the temperature of the air being dehumidified, the intercooling that can occur prior to condensation is independent of the inlet air temperature, and rather depends on the partial vapor pressure. Because the focus of this study to remove airborne moisture and reduce the intermediate pressure dew point of a TSC, it is important to first examine the intermediate pressure dew point, and then to compare with the dehumidified intermediate pressure dew point resulting from the WSG processing. Figure 60 shows the intermediate pressure dew point of air at an inlet temperature of 70 °F and relative humidity of 60%.

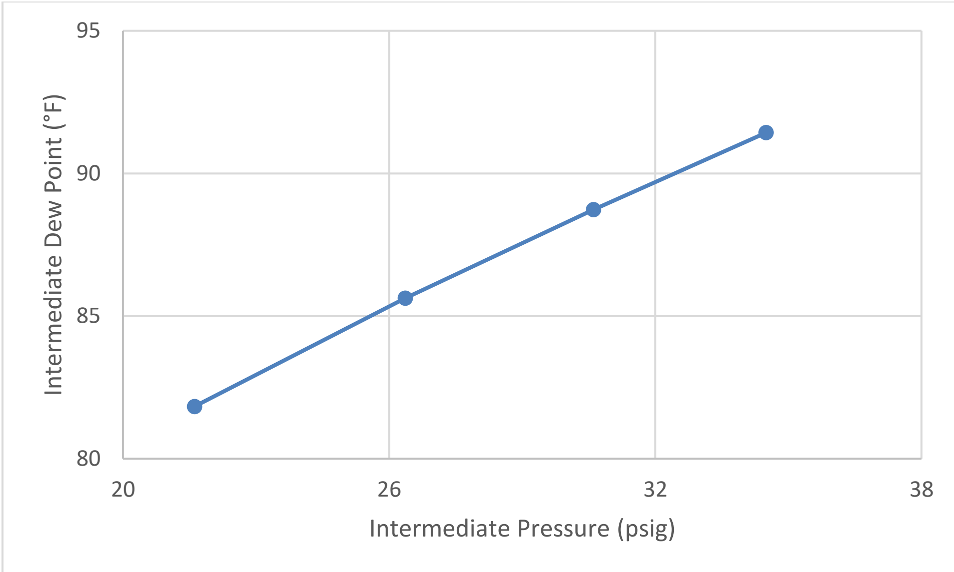

Figure 60: Intermediate pressure dew point of air entering a two-stage compressor at a temperature of 70 °F and relative humidity of 60% with the discharge pressure ranging from 75 to 150 psig

Figure 60 indicates that the intermediate pressure dew point of air entering a TSC at a temperature of 70 °F and relative humidity of 60% discharged to a range of pressure from 75 to 150 psig ranges from 80 to 95 °F, which means that intercooling to the inlet air temperature will condense water vapor and potentially damage the air compressor. Furthermore, to prevent the possibility of condensation within the intercooler, the intercooling is buffered by 5 °F above the intermediate dew point, so the inlet air temperature of the second stage of a TSC increases even more. Figure 61 is a plot of the intermediate pressure dew point for air initially at a temperature of 70 °F and relative humidity of 60% after being dehumidified by the WSG at various heating temperatures, namely 175, 250, and 325 °F.

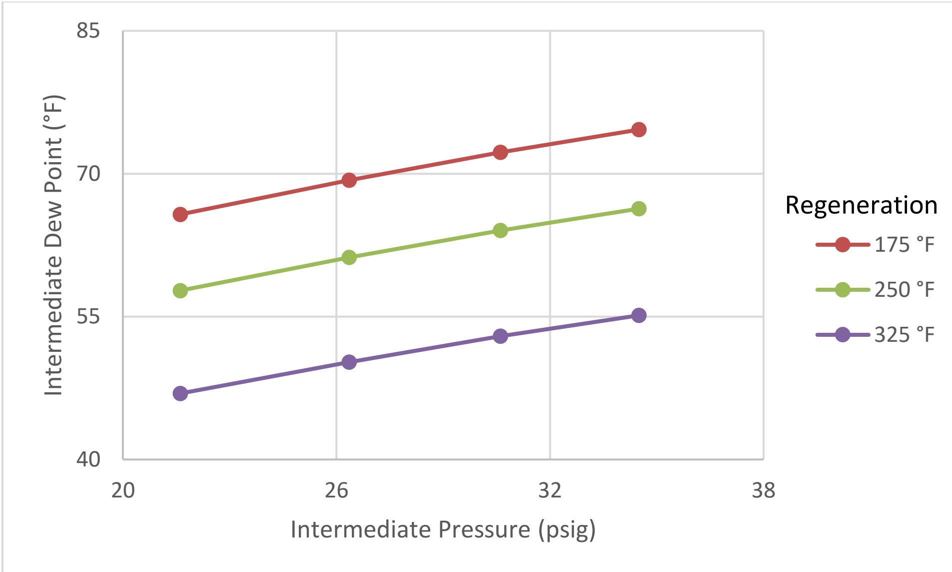

Figure 61: Intermediate pressure dew point of air initially at a temperature of 70 °F and relative humidity of 60% that is dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

The intermediate pressure dew point would ideally be at least 5 °F less than the inlet air temperature so that the intercooler can restore the inlet air temperature without condensing water vapor. Figure 61 shows that for a regeneration air temperature of 175 °F, the WSG can decrease the intermediate pressure dew point to a temperature ranging from 65 to 75 °F corresponding to a range of discharge pressure of 75 to 150 psig. Due to the intercooling buffer however, the intercooler would cool to a temperature ranging from 70 to 80 °F. When the WSG is regenerated by air at a temperature of 250 °F, the dehumidified intermediate dew point ranges from 55 to 70 °F, resulting in a second-stage inlet air temperature of 60 to 75 °F. Finally, with the WSG regenerated by air at a temperature of 325 °F, the intermediate pressure dew point ranges from 45 to 55 °F, resulting in intercooling to a range of 50 to 60 °F. Thus, for an inlet air temperature of 70 °F and relative humidity of 60%, the WSG can remove sufficient moisture when the regeneration temperature is high, or the discharge pressure is low for the medium and low regeneration temperatures. Figure 62 indicates the extent of intermediate dew point suppression relative to the air at a temperature of 70 °F and relative humidity of 60%.

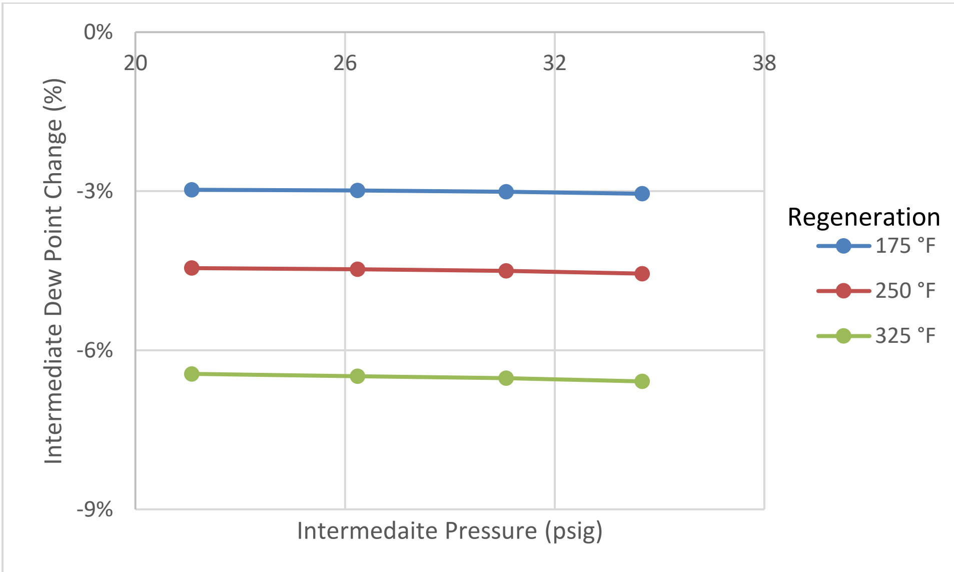

Figure 62: Intermediate pressure dew point suppression of air initially at a temperature of 70 °F and relative humidity of 60% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

According to Figure 62, at an inlet air temperature of 70 °F and relative humidity of 60%, the WSG can consistently reduce the intermediate pressure dew point by 3% when regenerated by air at a temperature of 175 °F, 4% when regenerated by air at a temperature of 250 °F, and by 6% when regenerated by air at a temperature of 325 °F. To continue the investigation, the intermediate pressure dew point of air initially at a temperature of 90 °F and relative humidity of 60% is plotted as a function of intermediate pressure corresponding to a range of discharge pressure of 75 to 150 psig in Figure 63.

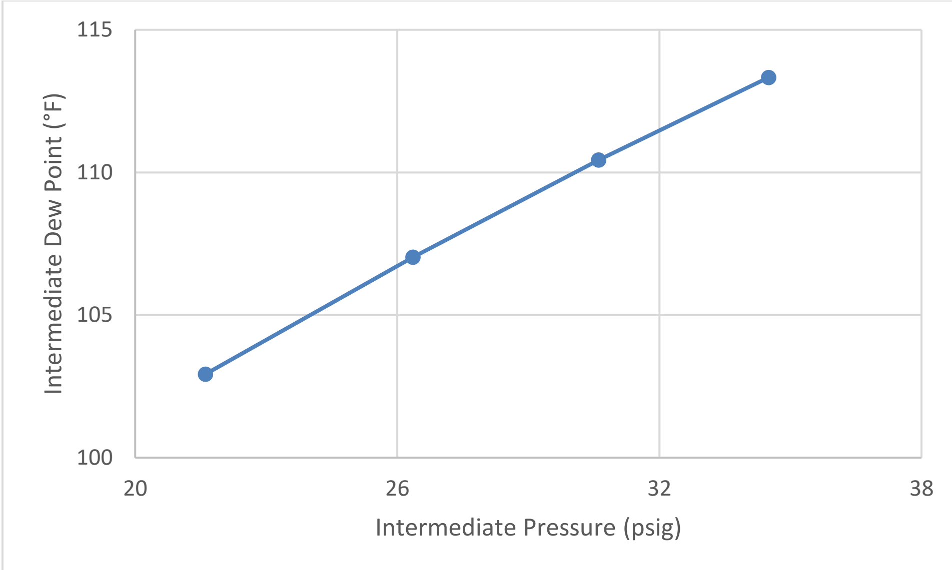

Figure 63: Intermediate pressure dew point for air entering at a temperature of 90 °F and relative humidity of 60% for a two-stage air compressor ranging discharge pressure from 75 to 150 psig

For air entering at a temperature of 90 °F and relative humidity of 60% Figure 63 shows that the intermediate pressure dew point corresponding to a range of discharge pressure of 75 to 150 psig is approximately 100 to 115 °F, increasing to a range of 105 to 120 °F when considering the 5 °F intercooling buffer. The intermediate pressure dew point must be 85 °F or less for the intercooler to restore or cool below the inlet air temperature without condensing water vapor. Figure 64 shows the intermediate pressure dew point resulting from the dehumidification of air initially at 90 °F and relative humidity of 60% using the WSG and regeneration temperatures of 175, 250, and 325 °F.

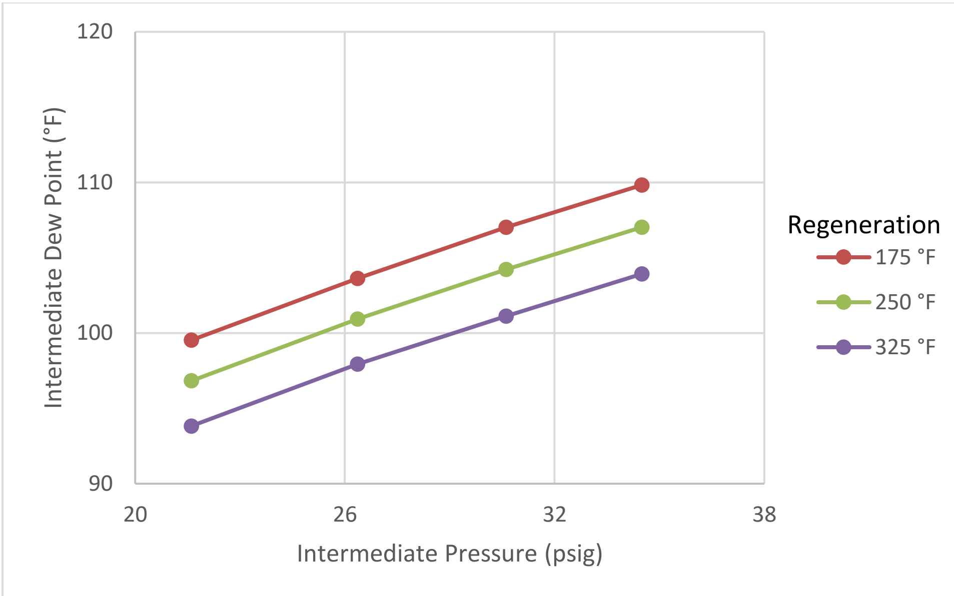

Figure 64: Intermediate pressure dew point of air initially at a temperature of 90 °F and relative humidity of 60% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

Figure 64 indicates that the intermediate pressure dew point corresponding to a discharge pressure ranging from 75 to 150 psig using air at a temperature of 90 °F and 60% relative humidity dehumidified by the WSG using a regeneration temperature of 175 °F ranges from 100 to 110 °F, resulting in intercooling yielding a temperature ranging from 105 to 115 °F. When the WSG dehumidifies this air using a regeneration temperature of 250 °F, the intermediate pressure dew point ranges from a temperature of 95 to 110 °F, resulting in intercooling to a temperature ranging from 100 to 115 °F. If the WSG uses a regeneration temperature of 325 °F for this air, the intermediate pressure dew point temperature ranges from 90 to 105 °F, resulting in intercooling to a temperature of 95 to 110 °F. In no case can the intercooler restore the inlet air temperature. The percent intermediate dew point suppression for this air using the three different regeneration temperatures is plotted in Figure 65.

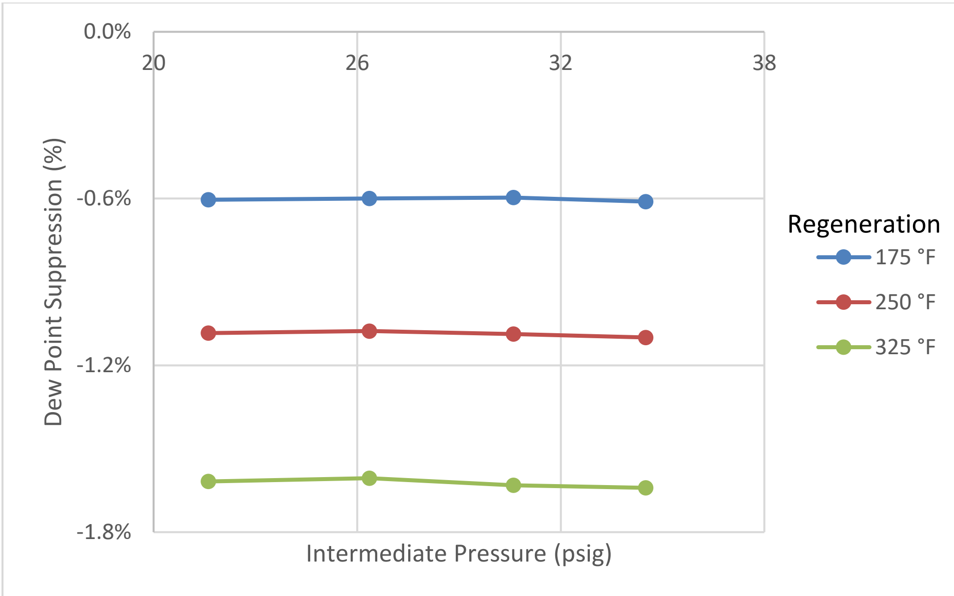

Figure 65: Intermediate dew point suppression of air at initially at a temperature of 90 °F and a relative humidity of 60% dehumidified by the WSG wheel with regeneration temperatures ranging from 175 to 325 °F

Recall that air initially at a temperature of 70 °F and a relative humidity of 60% dehumidified by the WSG wheel at a regeneration temperature ranging from 175 to 325 °F has its intermediate pressure dew point suppressed by 3-6%. Figure 65 indicates that for air initially at 90 °F and a relative humidity of 60%, the WSG wheel suppresses the intermediate pressure dew point by less than 1% using a regeneration temperature of 175 °F, approximately 1% using a regeneration temperature of 250 °F, and 1.5% for a regeneration temperature of 325 °F. The results of Figure 65 suggest that the greater the temperature of the air to be dehumidified, the less the WSG can remove moisture. To further investigate this trend, the intermediate dew point of air at initially at 110 °F and 60% relative humidity without and with the WSG wheel at various heating temperatures are plotted in Figure 66 and Figure 67, respectively.

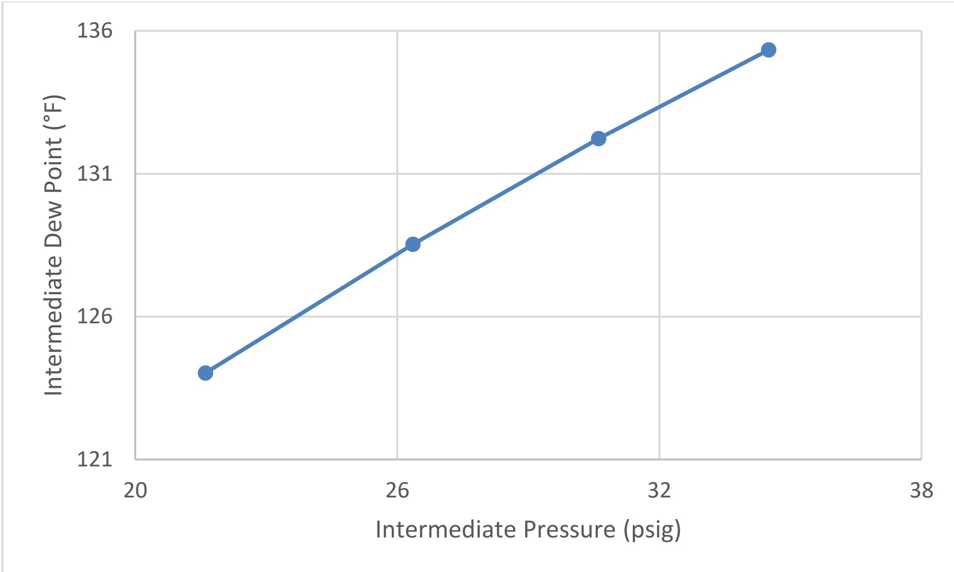

Figure 66: Intermediate pressure dew point of air at a temperature of 110 °F and relative humidity of 60% corresponding to the discharge pressure of a two-stage air compressor ranging from 75 to 150 psig

Figure 66 shows that the intermediate pressure dew point temperature corresponding to a discharge pressure of a two-stage air compressor ranging from 75 to 150 psig for an inlet air temperature of 110 °F and relative humidity of 60% ranges from 120 to 140 °F. To buffer the intercooler by an additional 5 °F, the intercooling results in a temperature ranging from approximately 125 to 140 °F. It was shown in the parametric study of intermediate pressure dew point temperature resulting from WSG dehumidification using regeneration temperatures ranging from 175 to 325 °F that as the moist air temperature increases, the WSG becomes less effective at moisture removal. Therefore, the ability of the WSG to remove moisture for air at a temperature of 110 °F and relative humidity of 60% for a TSC discharge pressure ranging from 75 to 150 psig using a WSG wheel and regeneration temperatures ranging from 175 to 325 °F is shown by Figure 67.

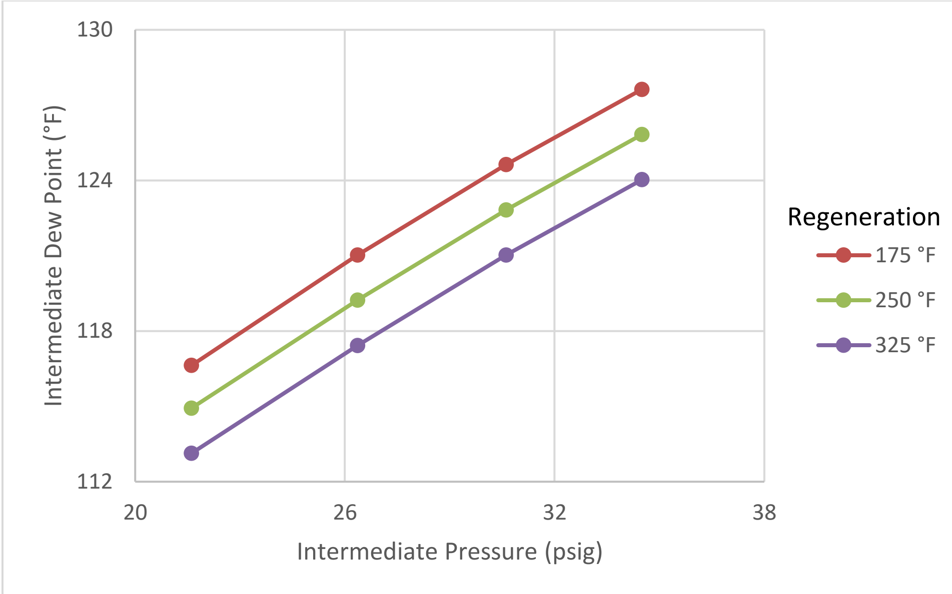

Figure 67: Intermediate pressure dew point of air initially at a temperature of 110 °F and relative humidity of 60% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

The intermediate pressure dew point of air initially at 110 °F and 60% relative humidity that is dehumidified by the WSG prior to being compressed by the TSC to a discharge pressure range of 75 to 150 psig is shown by Figure 67. The intermediate pressure dew point ranges from a temperature of 115 to 130 °F at a regeneration temperature of 175 °F, resulting in intercooling to a temperature of 120 to 135 °F. At a regeneration temperature of 250 °F, the intermediate pressure dew point temperature ranges from 110 to 125 °F, which means intercooling ranges from a temperature of 115 to 130 °F. Lastly, at a regeneration temperature of 325 °F, the intermediate pressure dew point ranges from 110 to 125 °F, and the air can be intercooled to a temperature ranging from 115 to 130 °F. For air at a temperature of 110 °F and 60% relative humidity, in no case can the WSG remove sufficient moisture so that the intercooler can restore the inlet air temperature. The percent intermediate pressure dew point suppression corresponding to this set of results is plotted in Figure 68.

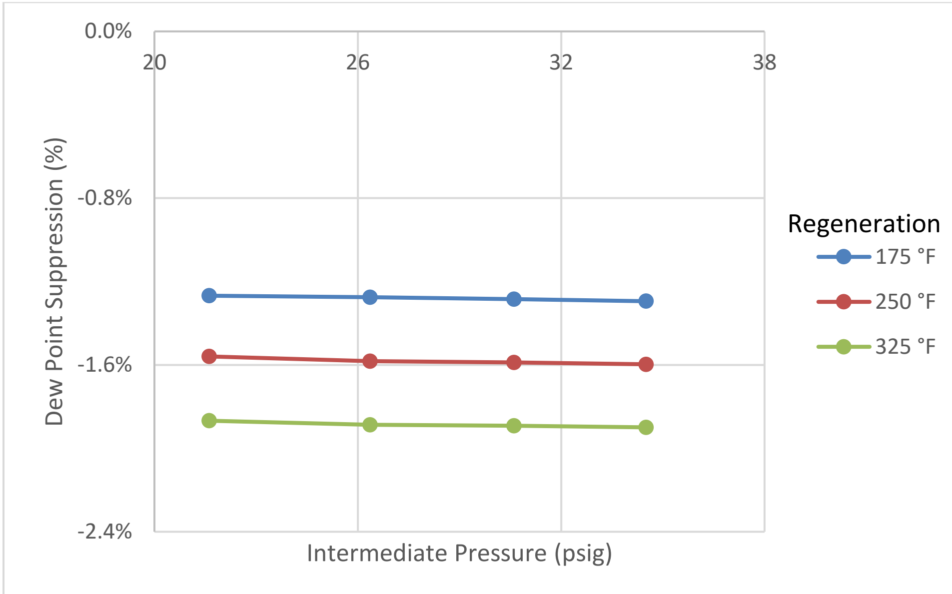

Figure 68: Intermediate dew point suppression of air at initially at a temperature of 110 °F and a relative humidity of 60% dehumidified by the WSG wheel with regeneration temperatures ranging from 175 to 325 °F

Figure 68 indicates that the percent intermediate pressure dew point suppression for air at a temperature of 110 °F and relative humidity of 60% is less than that of air at a temperature of 70 °F with the same amount of moisture, but more than that of the air at a temperature of 90 °F with the same amount of moisture. For air at 60% relative humidity, the WSG suppresses intermediate dew point of air initially at 70 °F by 3-6% and 1-2% for air initially at 110 °F. Dew point suppression deviates from linear trend between within that range, indicating that WSG wheel performance should be investigated in a narrower temperature range depending on the application.

WSG dehumidification negatively correlates with inlet air temperature because the water vapor adsorbed to the wheel is at the temperature of the air, and the regeneration air cannot sufficiently raise the temperature of the condensate to vaporize and scavenge the moisture.

Therefore, greater air inlet temperature requires higher regeneration temperature. However, because the WSG cannot operate at a temperature greater than 350 °F, it will not be able to dehumidify air at a temperature of 110 °F sufficiently to allow intercooling to the inlet air temperature.

While in some cases the WSG can dehumidify air enough to enable a TSC intercooler to restore the initial air temperature, the regeneration airstream is heated return air that must receive energy to reach the regeneration air temperatures. Figure 69 shows that the specific energy required to increase the temperature of the return air to regenerative ranges, which is independent of the temperature of the air that needs to be dehumidified and is instead proportional to the energy required to increase the return air's enthalpy at the return air temperature to the enthalpy of the regeneration air at the specified regeneration temperature.

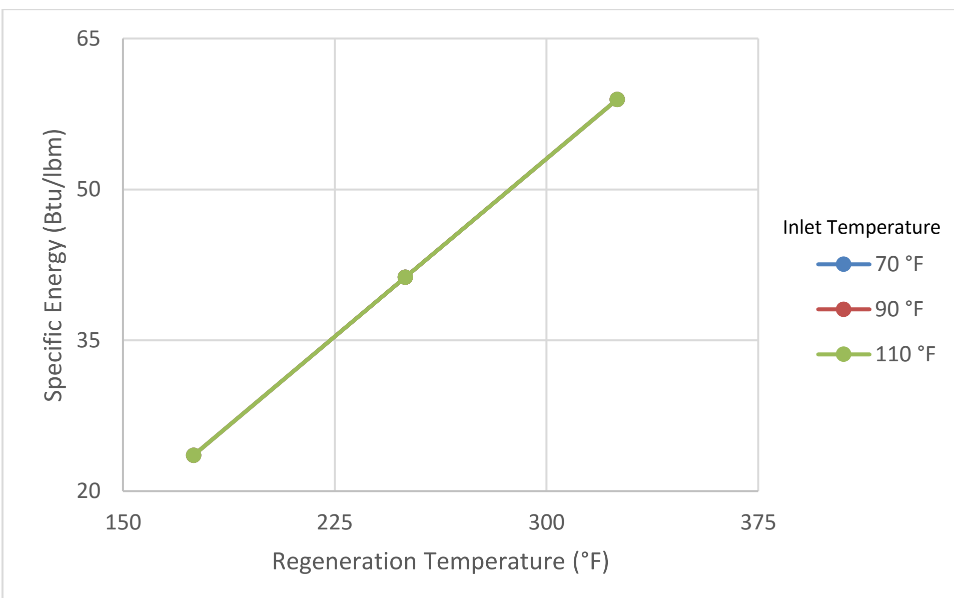

Figure 69: Specific energy required to reach regeneration temperatures ranging from 175 to 325 °F when using return air at a temperature of 75 °F and relative humidity of 50% and dehumidifying process air with a temperature ranging from 70 to 110 °F at 60% relative humidity

Figure 69 shows that the specific heat required to heat return air at a temperature of 75 °F and relative humidity of 50% to a regeneration temperature ranging from 175 to 325 °F ranges from 20 to 65 Btu/lbm and is independent of the conditions of the air that is to be dehumidified, which is again referred to as process air. Figure 70 shows the resulting temperature of the moist air after it is dehumidified for air initially at a temperature ranging from 70 to 110 °F and relative humidity of 60% dehumidified by the WSG using regeneration temperature ranging from 175 to 325 °F.

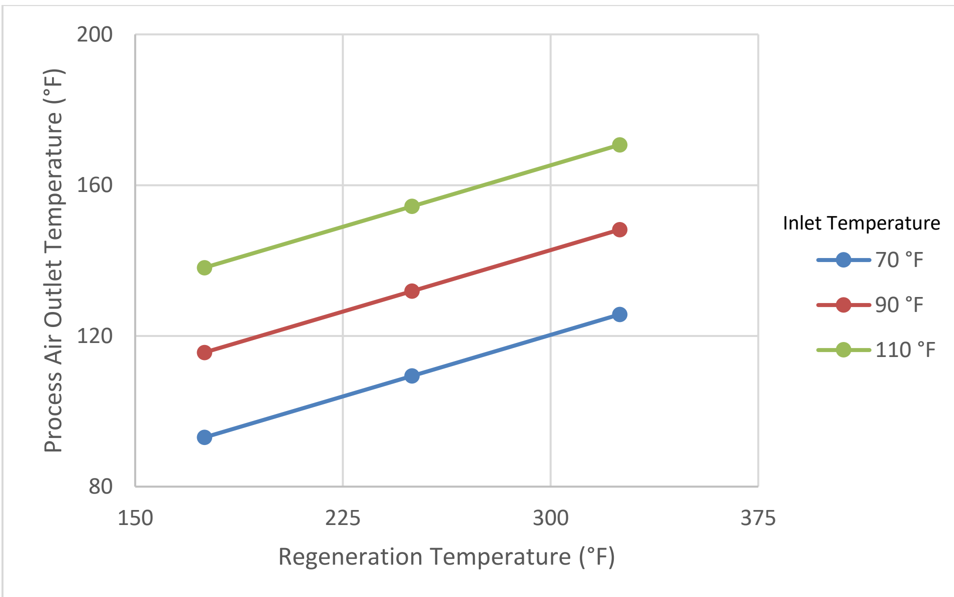

Figure 70: Process air outlet temperature of air entering the WSG wheel at temperatures ranging from 70 to 110 °F and 60% relative humidity dehumidifying with regeneration temperatures ranging from 175 to 325 °F

According to Figure 70, air initially at a temperature of 70 °F and 60% relative humidity dehumidified by the WSG using the range of regeneration temperatures has its temperature increased to a range of 90 to 130 °F because of the heat of sorption. Air initially at a temperature of 90 °F at 60% relative humidity has its temperature increased to a range of 115 to 150 °F corresponding to the regeneration temperatures used in this parametric study. Finally, air at a temperature of 110 °F and 60% relative humidity increases in temperature to a range of 135 to 170 °F due to the range of regeneration temperature of 175 to 325 °F. Because the condensation of airborne water vapor onto the surface of the WSG releases the heat of sorption, the temperature of the air increases significantly, as demonstrated by Figure 71.

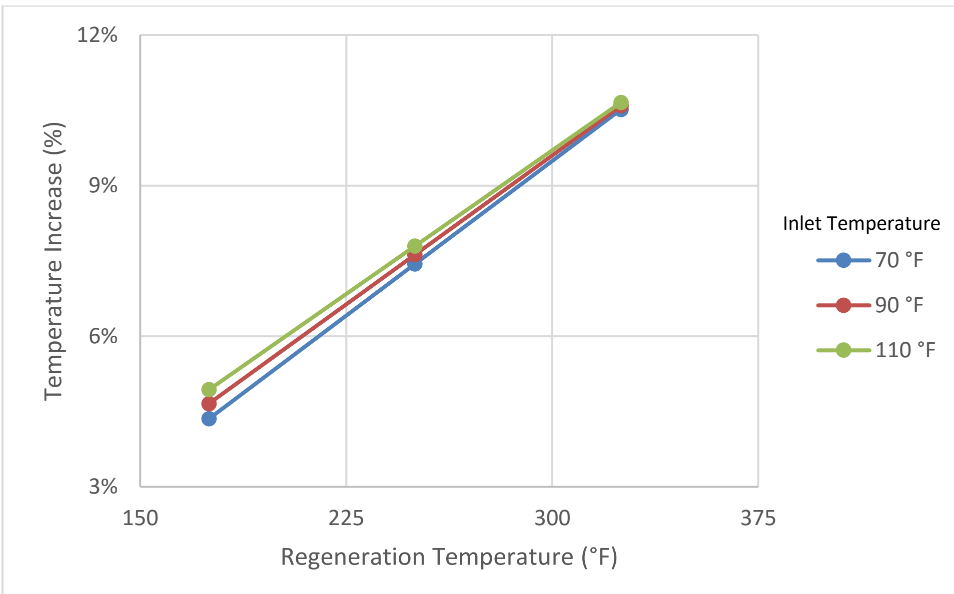

Figure 71: Change in temperature of air at 60% relative humidity and a temperature ranging from 70 to 110 °F due to the heat of sorption generated by dehumidification with the WSG regenerated by a temperature ranging from 175 to 325 °F

According to Figure 71, the percent increase in temperature of the air due to the heat of sorption is consistent for all inlet air temperatures, equal to 4-5% at a regeneration temperature of 175 °F, 8% at a regeneration temperature of 250 °F, and 11% at a regeneration temperature of 325 °F. The process air temperature increase is proportional to regeneration temperature because the higher the regeneration temperature, the more moisture will be scavenged by the regeneration airstream, restoring more of the WSG's total capacity to adsorb moisture but also releasing more heat when adsorption recurs.

Because the WSG both dehumidifies the air and increases its temperature, the specific work of a TSC must be evaluated according to the dehumidified inlet air temperature and intermediate pressure dew point buffer. Although the specific work of air compression is linearly dependent on the inlet air temperature, the dehumidified intermediate pressure dew point can be less than the dehumidified inlet air temperature, meaning that each stage of the TSC would consume a different amount of specific work. The change in specific work from the MTSC of the dehumidified two-stage air compressor (DTSC) equipped with a WSG using return air at a temperature of 75 °F and 50% relative humidity that is heated to a regeneration temperature of 175 to 325 °F to dehumidify air initially at 70 °F and 60% relative humidity is plotted and shown by Figure 72.

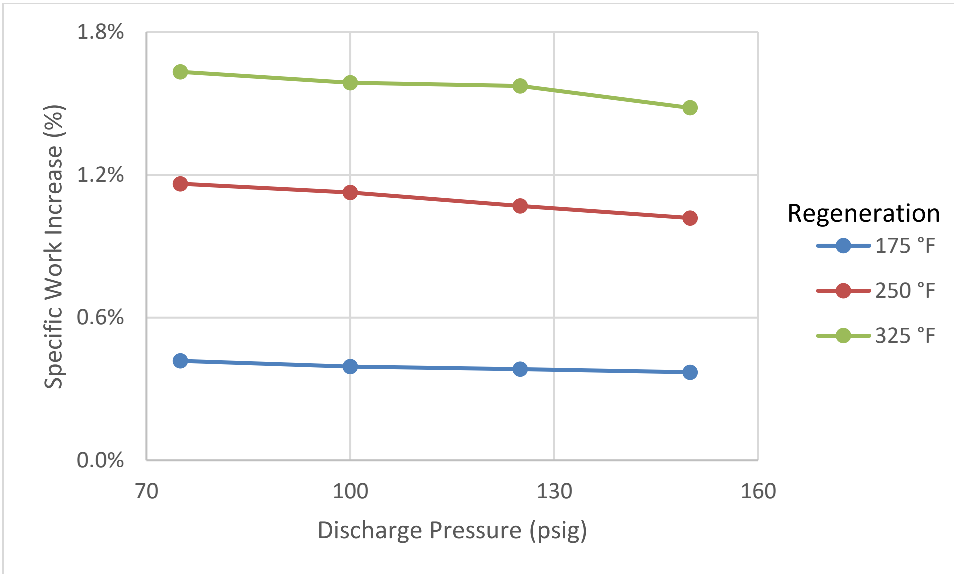

Figure 72: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 70 °F and 60% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

Figure 72 shows that for air initially at a temperature of 70 °F and relative humidity of 60%, the specific work of the TSC supplied with air dehumidified by a WSG using a regeneration temperature of 175 °F consistently increases by 0.5% regardless of the discharge pressure. If the regeneration temperature used with the WSG is 250 °F, the specific work increases by approximately 1%, and if the regeneration temperature used with the WSG is 325 °F, the specific work increases by approximately 1.5%. Figure 72 therefore indicates that although the intermediate pressure dew point is reduced through dehumidification, the increase of air temperature used at the inlet of the TSC outweighs the increased ability to intercool. The total change in specific energy resulting from the sum of the specific energy to increase the temperature of the return air to regenerative temperatures and the increase in specific work of the

TSC using air dehumidified by the WSG at the prescribed regeneration temperatures is shown by Figure 73.

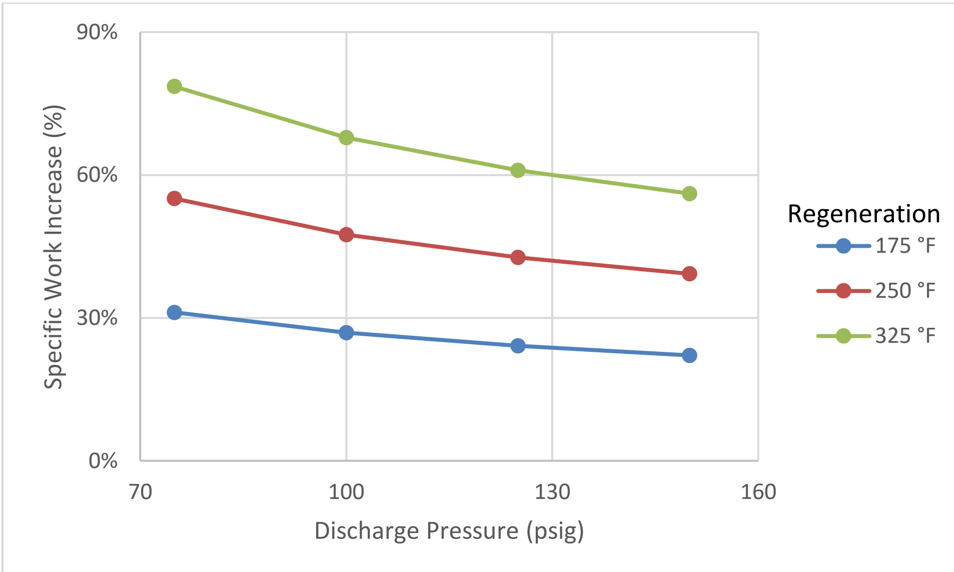

Figure 73: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 70 °F and 60% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

Figure 73 shows that when added to the work of the DTSC, the specific energy required to heat the return air to regenerative quality increases the specific energy of the two-stage air compression system by 22-30% when the regeneration air temperature is 175 °F and is negatively correlated with discharge pressure. If the regeneration air temperature is 250 °F, then the specific energy of heating the return air in the DTSC package increases the total specific energy consumption by 39-55% and is also negatively correlated with discharge pressure. Using the WSG with a regeneration air temperature of 325 °F, the total specific energy of the system increases by 56-79%, again negatively correlated with discharge pressure. The increase of specific work of the TSC of dehumidified air initially at 90 °F and 60% rh and total specific energy increase including the heating of air initially at 75 °F and 50% rh are shown by Figure 74 and Figure 75, respectively.

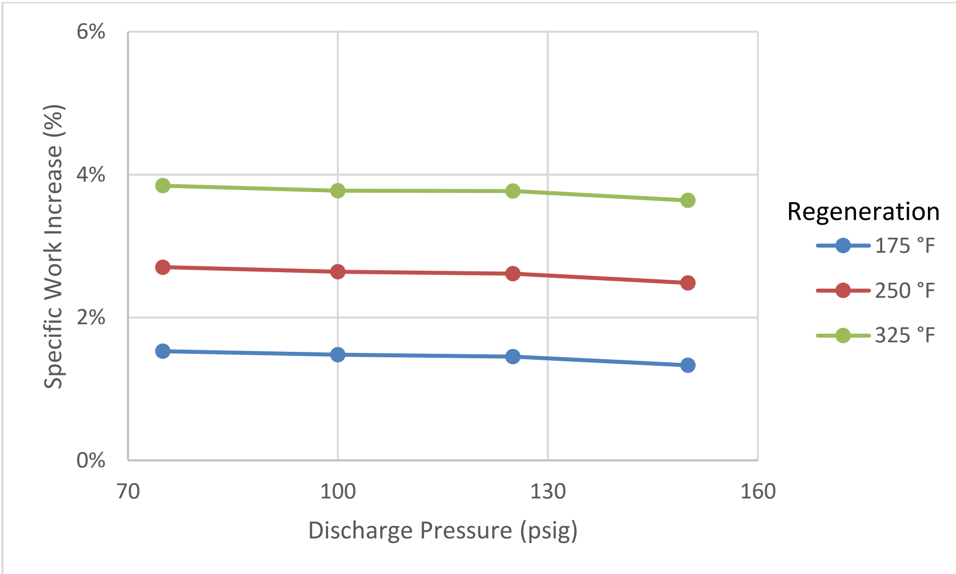

Figure 74: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 90 °F and 60% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

At 60% relative humidity, although the specific work of a TSC using air initially at a temperature of 70 °F dehumidified by the WSG at the prescribed range of regeneration temperatures increases by 0.5-1.5%, Figure 74 shows that the disparity increases as the temperature of the air to be dehumidified increases to 90 °F. At a regeneration air temperature of 175 °F, the DTSC requires 2% more specific work than the MTSC. When the regeneration temperature is 250 °F, the DTSC consumes 3% more specific work than the MTSC, and when the regeneration temperature is 325 °F, the specific work increases by 4%. The specific work increase again corresponds to the dominance of the dehumidified air temperature over the dehumidified intermediate pressure dew point in the specific work model for the two-stage air compressor. Figure 75 indicates the increase of specific energy consumed by the DTSC package when compared to the MTSC.

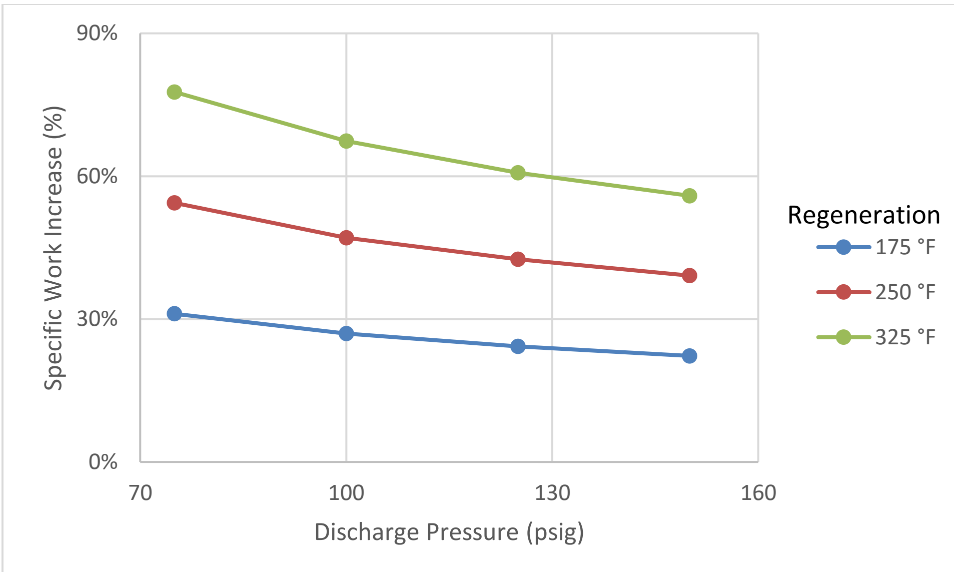

Figure 75: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 90 °F and 60% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

Like the total specific energy increase of air initially at 70 °F, the addition of the specific energy required to increase the temperature of the return air to regenerative levels significantly increases the total specific energy consumed by the system. For the WSG dehumidifying air entering at a temperature of 90 °F and relative humidity of 60% to supply a TSC with dehumidified air, the specific energy of the system increases by 22-30% when using a regeneration temperature of 175 °F, by 39-54% when regenerated by air at a temperature of 250 °F, and by 56-78% if regenerated at a temperature of 325 °F.

The increase of the system's specific energy consumption negatively correlates with discharge pressure. Because the specific work of the TSC depends linearly on the air temperature, the specific work at a higher temperature is offset by the temperature difference so that the increase of the specific work remains the same because it is relative to an offset baseline. To continue evaluating trends that exist due to the increase of inlet air temperature, the increase of specific work of the two-stage air compressor of air initially at a temperature 110 °F and 60% relative humidity dehumidified by a WSG using return air initially at temperature of 75 °F and 50% relative humidity heated to regeneration temperatures ranging from 175 to 325 °F is plotted in Figure 76.

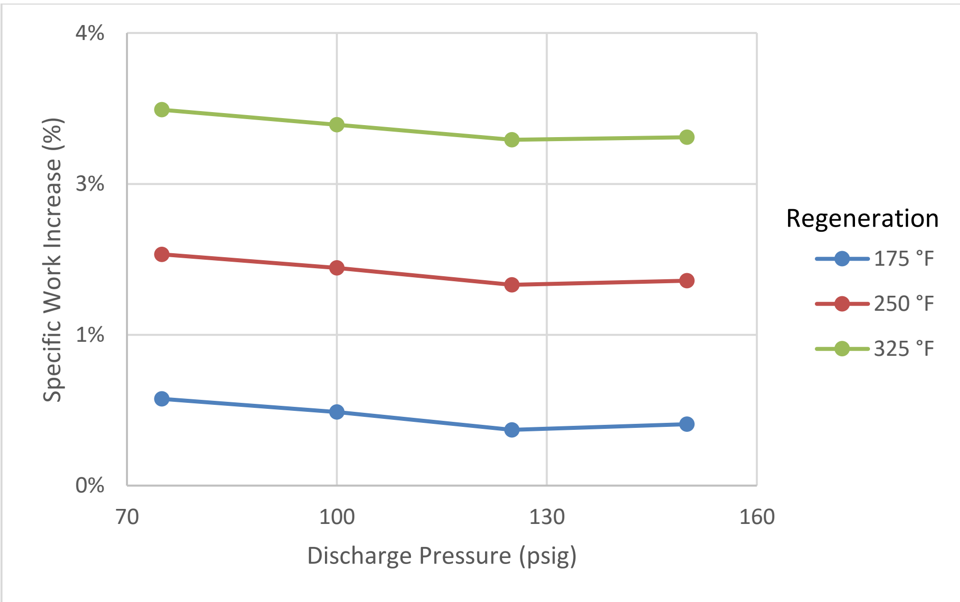

Figure 76: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 110 °F and 60% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

According to Figure 76, the increase of specific worker of the DTSC relative to the MTSC for air initially at a temperature of 110 °F and 60% relative humidity is approximately 0.5% for a regeneration temperature of 175 °F, 2% for a regeneration temperature of 250 °F, and 3.5% for a regeneration temperature of 325 °F. Recall that the WSG model had non-linear performance in its ability to reduce the intermediate pressure dew point temperature of air initially at a temperature ranging from 70 to 110 °F. Although at 60% relative humidity the increase of specific work for the DTSC using air dehumidified from a temperature of 110 °F and is greater than that of air dehumidified from a temperature of 70 °F, it is less than that of air dehumidified from a temperature of 90 °F. This is because for air at a relative humidity of 60%, the WSG's suppression of the intermediate pressure dew point for air at a temperature of 90 °F is less than that of air at a temperature of 110 °F, meaning the intercooler can cool the air off more in the latter case and reduce the specific work of the second stage of the TSC. Figure 77 shows the change in specific work of the DTSC relative to the MTSC.

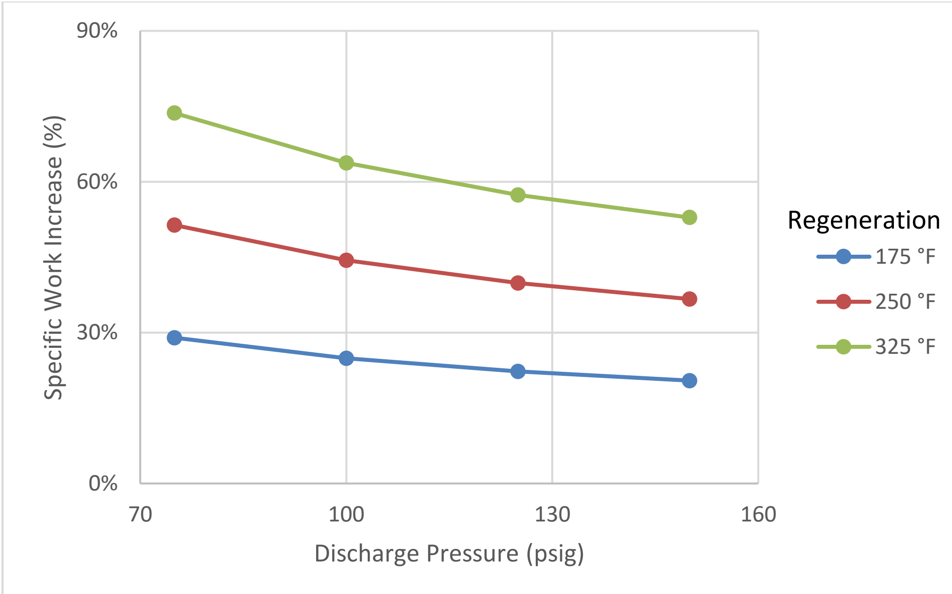

Figure 77: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 110 °F and 60% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

For air initially at a temperature of 110 °F and relative humidity of 60%, the increase of specific work of two-stage air compression compounded with the specific heat required to increase the temperature of the return air to regenerative levels is significant. Using a regeneration temperature of 175 °F, the increase of specific energy for the DTSC package ranges from 20-29%. To dehumidify the air with a regeneration airstream at a temperature of 250 °F, the total increase of specific energy for the DTSC package ranges from 37-51%, and to do so with a regeneration airstream temperature of 325 °F requires an additional 53-74% specific energy.

For air at 60% relative humidity, the sum of the increase in air temperature due to the heat of sorption and the suppression of the intermediate pressure dew point results in a net increase of temperature of 1-3% for air initially at a temperature of 70 °F, and net temperature increase of 3-7% for air initially at a temperature of 110 °F, resulting in the increase of specific work for the two-stage compressor. Furthermore, the specific energy of heating the return air from at a relative humidity of 50% from 75 °F to a regeneration temperature ranging from 175 to 325 °F results in substantial specific energy consumption for the DTSC package ranging from 20-80%. The following subsection details the investigation of dehumidifying air at a relative humidity of 90% and temperature ranging from 70 to 110 °F using a WSG and regeneration temperature ranging from 175 to 325 °F.

8.3.2 WSG Dehumidifying Air at a Relative Humidity of 90%

For air at a relative humidity of 90% and temperature ranging from 70 to 110 °F, the TSC can require 3-7% additional specific work corresponding to the inlet air temperature range. The following results determine the change in specific work of the TSC using dehumidified air instead of moist air, as well as the net energy consumption of the DTSC package. The investigation begins with the parametric study of the intermediate pressure dew point of air initially at a temperature 70 °F and 90% relative humidity with a TSC discharge pressure ranging from 75 to 150 psig as shown by Figure 78.

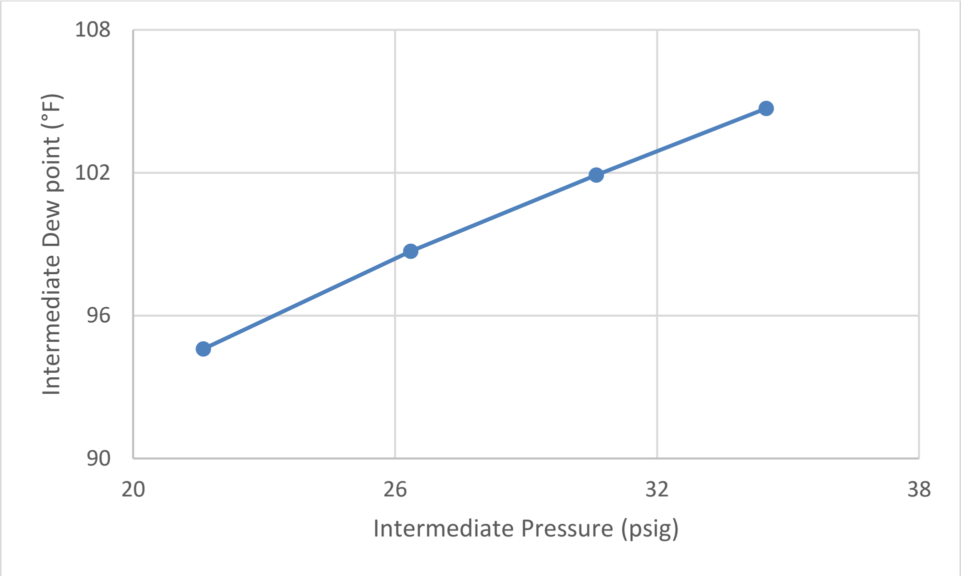

Figure 78: Intermediate pressure dew point of air entering a two-stage compressor at a temperature of 70 °F and relative humidity of 90% with the discharge pressure ranging from 75 to 150 psig

The intermediate pressure dew point of air initially at a temperature of 70 °F and relative humidity of 90% compressed by a TSC to a discharge pressure ranging from 75 to 150 psig ranges from 94-105 °F. The need to buffer intercooling by 5 °F results in a second-stage inlet temperature of 99-110 °F. It would be advantageous to dehumidify the air so that the intermediate pressure dew point would be 65 °F or less so that the intercooler can restore the initial air temperature or further cool it. The intermediate pressure dew point of air initially at 70 °F and 90% relative humidity dehumidified by the WSG at regeneration temperatures ranging from 175 to 325 °F prior to being compressed by a two-stage compressor to a discharge pressure ranging from 75 to 150 psig is shown by Figure 79.

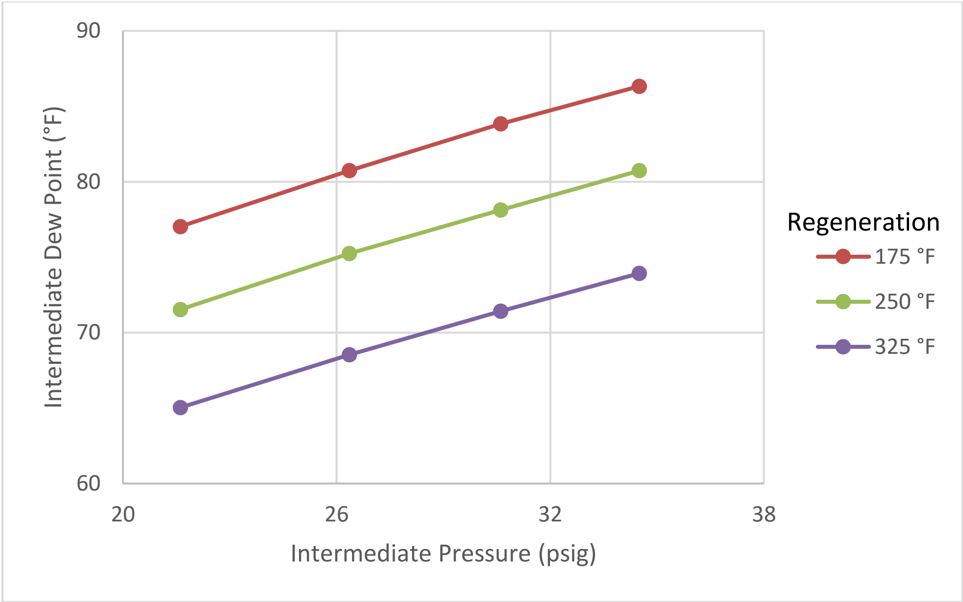

Figure 79: Intermediate pressure dew point of air initially at a temperature of 70 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

Figure 79 indicates that the intermediate pressure dew point of air initially at a temperature of 70 °F and 90% relative humidity dehumidified by a WSG at a regeneration temperature of 175 °F then compressed by a TSC to a discharge pressure of 75 to 150 psig ranges from 77-87 °F, which can be intercooled to a temperature ranging from 82-92 °F. If the regeneration temperature of the WSG is increased to 250 °F, the intermediate pressure dew point ranges from 71-81 °F, and can be intercooled to 76-86 °F. Finally, if the WSG is regenerated using a temperature of 325 °F, the intermediate pressure dew point reduces from a range of 94-105 °F to a temperature ranging from 65-74 °F, which can be intercooled to a temperature of 70-79 °F. Figure 80 compares all the ranges of dehumidified intermediate pressure dew point corresponding to the regeneration temperatures.

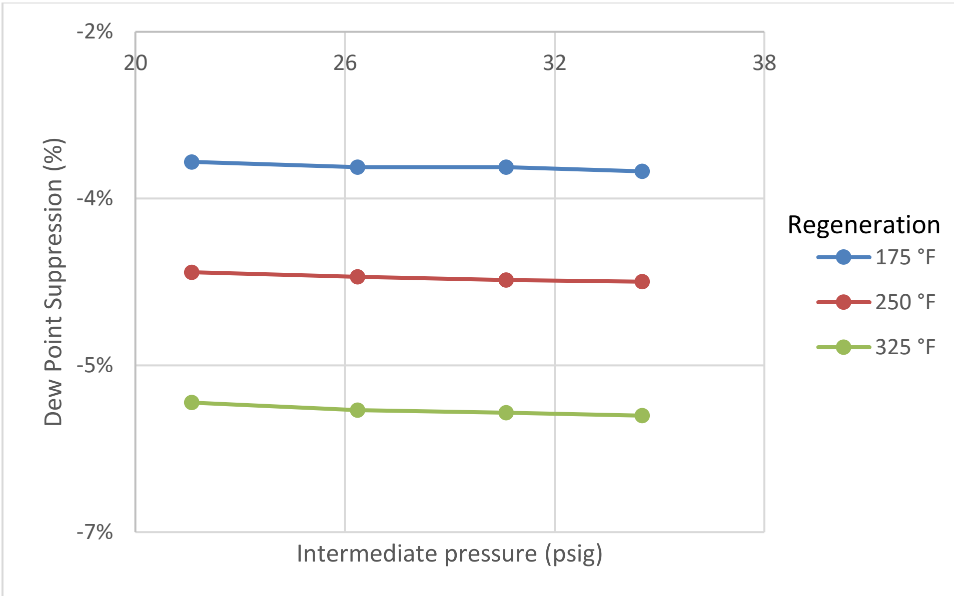

Figure 80: Intermediate pressure dew point suppression of air initially at a temperature of 70 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

For air with a relative humidity of 90% and initial temperature of 70 °F, the percent intermediate dew point suppression ranges is approximately 3.5% for a regeneration temperature of 175 °F, 4.4% regenerating the WSG with air at a temperature of 250 °F, and 5.3% using a regeneration air temperature of 325 °F to scavenge the WSG. The intermediate pressure dew point suppression must outweigh the increase of air temperature due to the heat of sorption to reduce the specific work consumed by the TSC, which will be considered after checking the other intermediate pressure dew points of the study. The intermediate pressure dew point of air initially at a temperature of 90 °F and 90% relative humidity corresponding to the two-stage air compression to a discharge pressure ranging from 75 to 150 psig is shown by Figure 81.

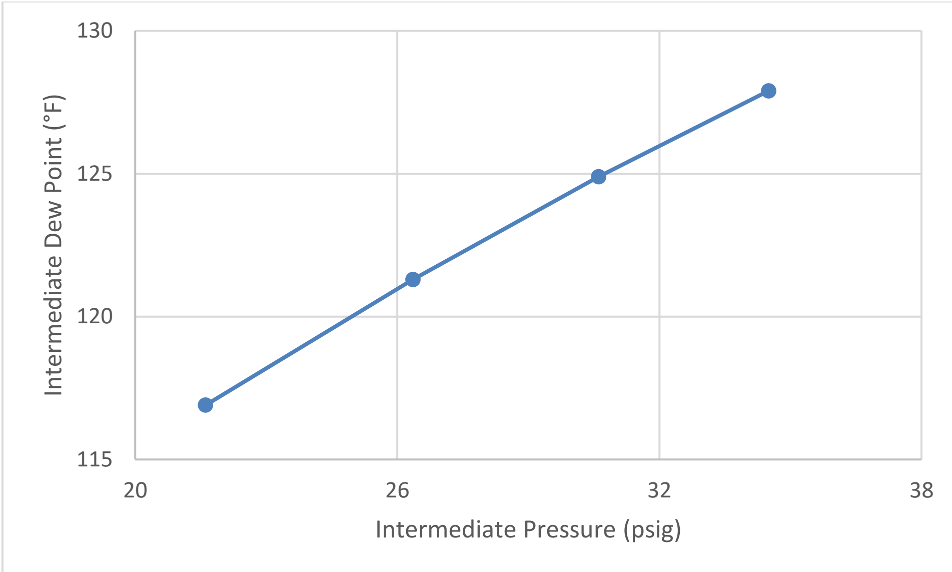

Figure 81: Intermediate pressure dew point of air entering a two-stage compressor at a temperature of 90 °F and relative humidity of 90% with the discharge pressure ranging from 75 to 150 psig

The intermediate pressure dew point corresponding to the two-stage air compression of air initially at 90 °F and 90% relative humidity to a discharge pressure of 75 to 150 psig ranges from 116 to 128 °F, which can be intercooled to a temperature ranging from 121 to 133 °F. To investigate the possibility of dehumidifying the air to restore the ability to intercool to a temperature of 90 °F, the air is dehumidified by the WSG using a regeneration temperature ranging from 175 to 325 °F and the resulting intermediate pressure dew points are plotted in Figure 82.

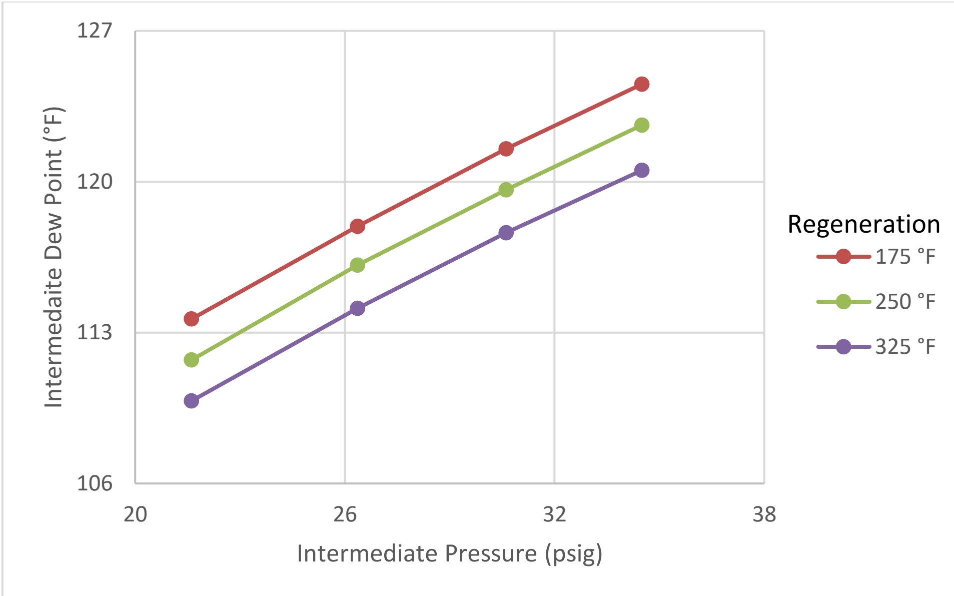

Figure 82: Intermediate pressure dew point of air initially at a temperature of 90 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

Originally, the intermediate pressure dew point corresponding to a TSC discharge pressure ranging from 75 to 150 psig for air initially at a temperature of 90 °F and relative humidity of 90% ranged from 116 to 128 °F, which could be cooled to a temperature ranging from 121 to 133 °F. Using a WSG and regeneration temperature of 175 °F, the intermediate pressure dew point reduces to a range of 113 to 125 °F, which can be intercooled to a temperature ranging from 118 to 130 °F. For a WSG dehumidifying this air with a regeneration temperature of 250 °F, the intermediate pressure dew point ranges from 111 to 123 °F, and can be intercooled to a temperature ranging from 116 to 128 °F. If the WSG dehumidifies this air with a regeneration airstream at a temperature of 325 °F, the intermediate pressure dew point ranges from 109 to 121 °F, which can be intercooled to a temperature ranging from 114 to 126 °F. The percent intermediate dew point suppression is plotted and shown by Figure 83.

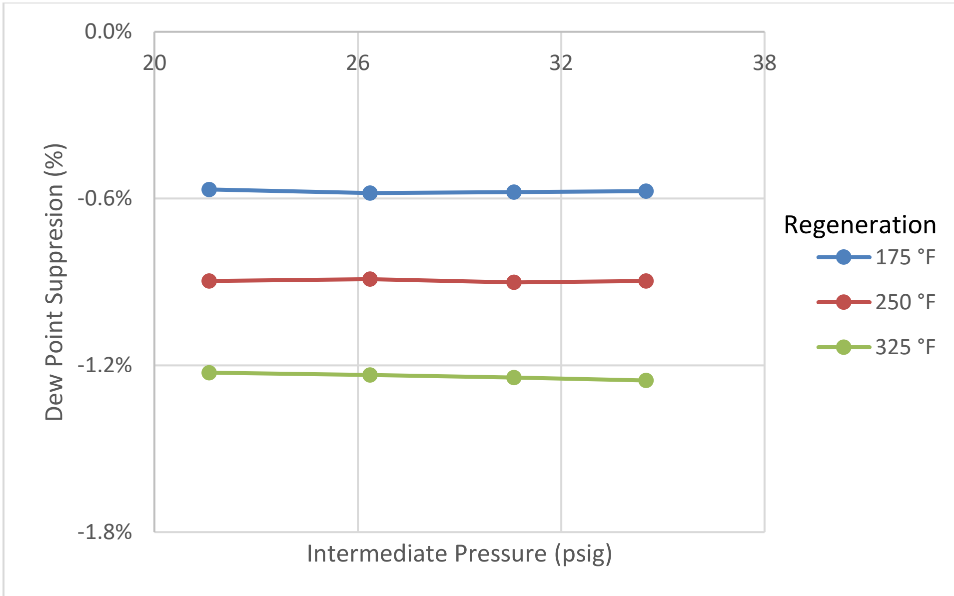

Figure 83: Intermediate pressure dew point suppression of air initially at a temperature of 90 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

The suppression of the intermediate dew point of air initially at a temperature of 90 °F and 90% relative humidity resulting from dehumidification with the WSG is approximately 0.5% when the regeneration temperature is 175 °F, 1% when the regeneration temperature is 250 °F, and 1.2% when the regeneration temperature is 250 °F. It is important to note that the trend from the experiment of the WSG dehumidifying air at 60% relative humidity is non-linear. To further investigate the ability of the WSG to dehumidify the air, the intermediate pressure dew point of air initially at 110 °F and 90% relative humidity corresponding to a TSC discharge pressure range of 75 to 150 psig is plotted and shown in Figure 84.

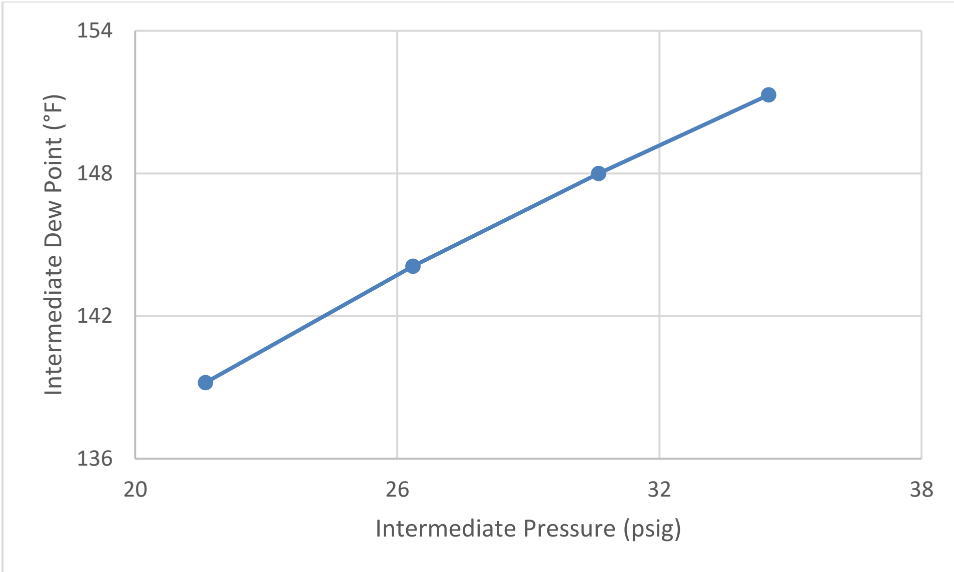

Figure 84: Intermediate pressure dew point of air entering a two-stage compressor at a temperature of 110 °F and relative humidity of 90% with the discharge pressure ranging from 75 to 150 psig

The intermediate pressure dew point of air initially at a temperature of 110 °F and relative humidity of 90% corresponding to a discharge pressure ranging from 75 to 150 psig is equal to a temperature ranging from 139 to 152 °F, which can be intercooled to a temperature ranging from 144 to 157 °F. The intermediate pressure dew point of this air dehumidified by a WSG using regeneration temperature ranging from 175 to 325 °F is plotted and shown by Figure 85.

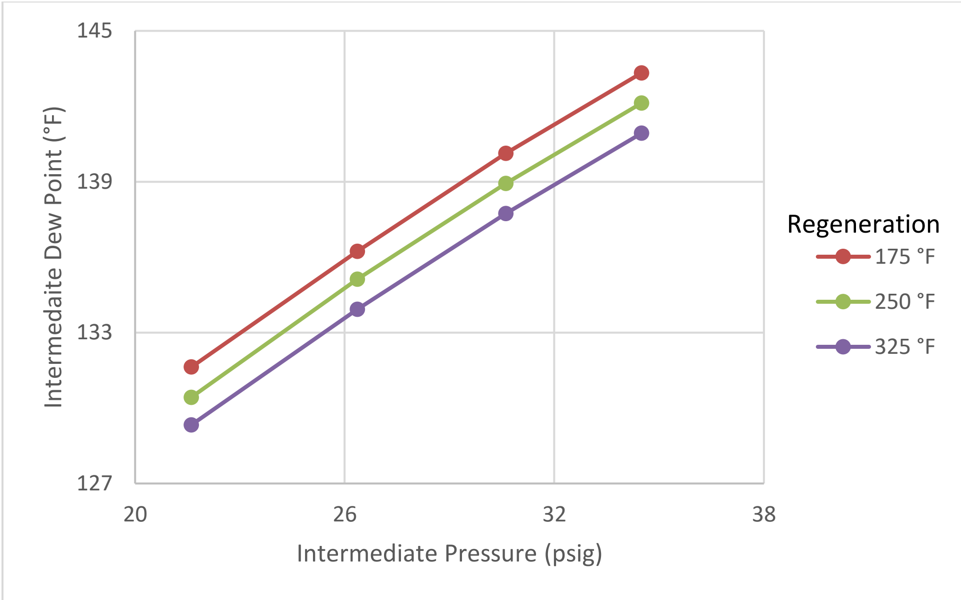

Figure 85: Intermediate pressure dew point of air initially at a temperature of 110 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

Compared to the intermediate pressure dew point of air at a temperature of 110 °F and 90% relative humidity that is compressed by a TSC to a discharge pressure ranging from 75 to 150 psig that ranges from 139 to 152 °F, the intermediate pressure dew point of the air dehumidified by the WSG with a regeneration temperature of 175 °F ranges from 131 to 144 °F, which can be intercooled to a temperature ranging from 136 to 149 °F. With a regeneration temperature of 250 °F, the WSG dehumidifies the air so that the intermediate pressure dew point ranges from 130 to 143 °F, and the intercooling reaches a cooling temperature ranging from 135 to 148 °F. If the regeneration temperature is 325 °F, the intermediate pressure dew point of this air dehumidified by the WSG ranges from 129 to 141 °F, and can be intercooled to a temperature ranging from 134 to 146 °F. The suppression of the intermediate pressure dew point is plotted and exhibited by Figure 86

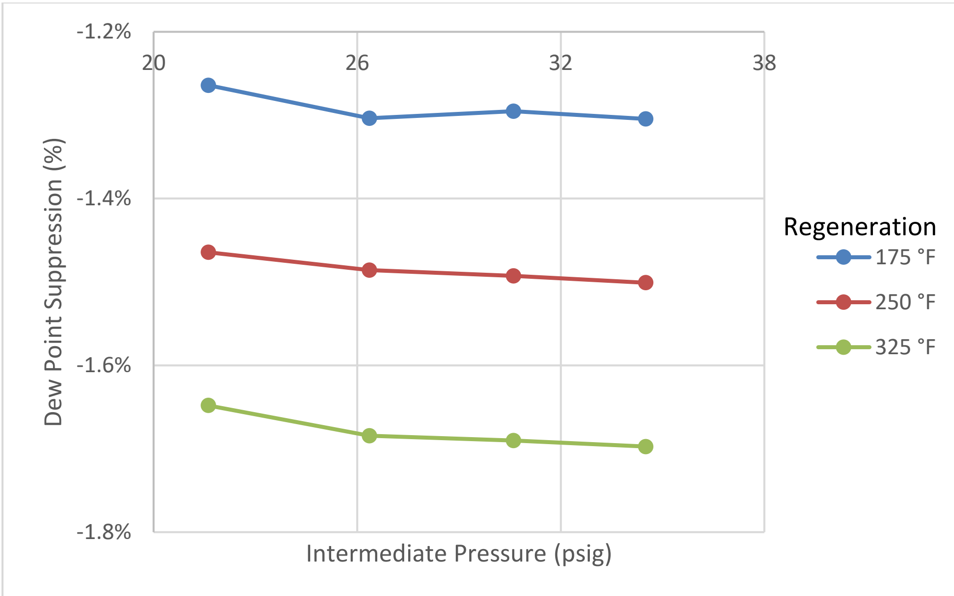

Figure 86: Intermediate pressure dew point suppression of air initially at a temperature of 110 °F and relative humidity of 90% dehumidified with the WSG wheel using regeneration temperatures ranging from 175 to 325 °F

If the air is initially at a temperature of 110 °F and a relative humidity of 90%, the intermediate pressure dew point suppression of a WSG using a regeneration temperature of 175 °F is approximately 1.3%. For the same case but instead using a regeneration temperature of 250 °F, the intermediate pressure dew point suppression is approximately 1.5%. If the regeneration temperature for the same air is instead 325 °F, the intermediate pressure dew point suppression is approximately 1.7%. The trend for air in the prescribed temperature range and relative humidity of 90% follows the expected trend of dehumidification by the WSG, which is non-linear.

For air at 90% relative humidity, the WSG wheel suppresses intermediate dew point by 3-6% for air initially at a temperature of 70 °F and 1-2% for air initially at a temperature of 110 °F. Dew point suppression is non-linear within the initial air temperature range and should be investigated at a narrower range depending on the application. As air inlet temperature increases, suppression decreases because the vapor adsorbed into condensate in the WSG is at the temperature of the air, and at as the temperature increases, it will require more heat to vaporize the condensate to migrate it with the regeneration airstream. However, a higher regeneration temperature requires more specific energy, as shown by Figure 87.

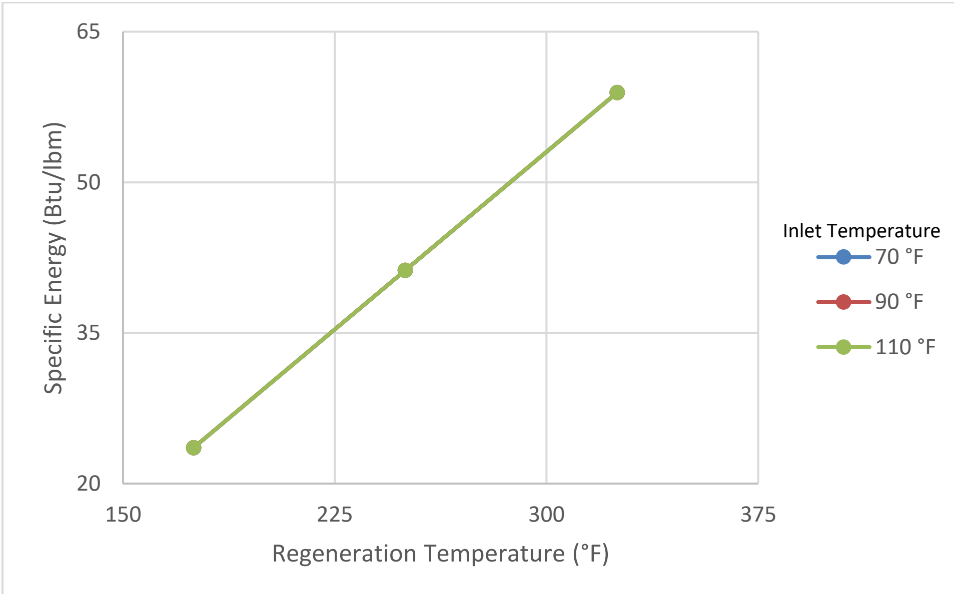

Figure 87: Specific energy required to reach regeneration temperatures ranging from 175 to 325 °F when using return air at a temperature of 75 °F and relative humidity of 50% and dehumidifying process air with a temperature ranging from 70 to 110 °F at 90% relative humidity

As anticipated, the specific energy to heat the return air from a relative humidity of 50% and temperature of 75 °F to a regeneration temperature ranging from 175 to 325 °F is independent of the air to be dehumidified, and instead depends on the change of the return air's enthalpy from 75 °F to the regeneration temperature. The temperature of the air initially at a relative humidity of 90% and temperature ranging from 70 to 110 °F exiting the WSG after dehumidification is plotted as a function of regeneration temperature in Figure 88.

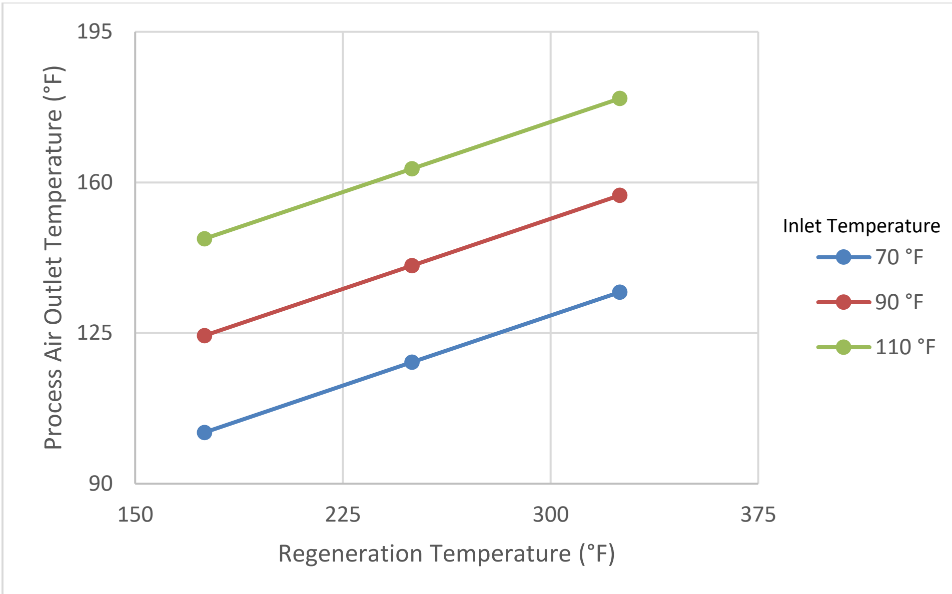

Figure 88: Process air outlet temperature of air entering the WSG wheel at temperatures ranging from 70 to 110 °F and 90% relative humidity dehumidified with a WSG wheel dried by regeneration temperatures ranging from 175 to 325 °F

The increase of temperature caused by heat of sorption during dehumidification is increasingly driven by regeneration temperature, as shown by Figure 88. For air at 90% relative humidity, at an initial temperature of 70 °F the resulting temperature corresponding to a regeneration temperature ranging from 175 to 325 °F ranges from 101 to 135 °F. Under the same conditions, but instead at an initial temperature of 90 °F, the temperature increases to a range of 124 to 157 °F. If the air were instead initially at a temperature of 110 °F, the resulting temperature range would be 146 to 180 °F. The percent increase of temperature of the air being dehumidified is plotted as a function of regeneration temperature and shown by Figure 89.

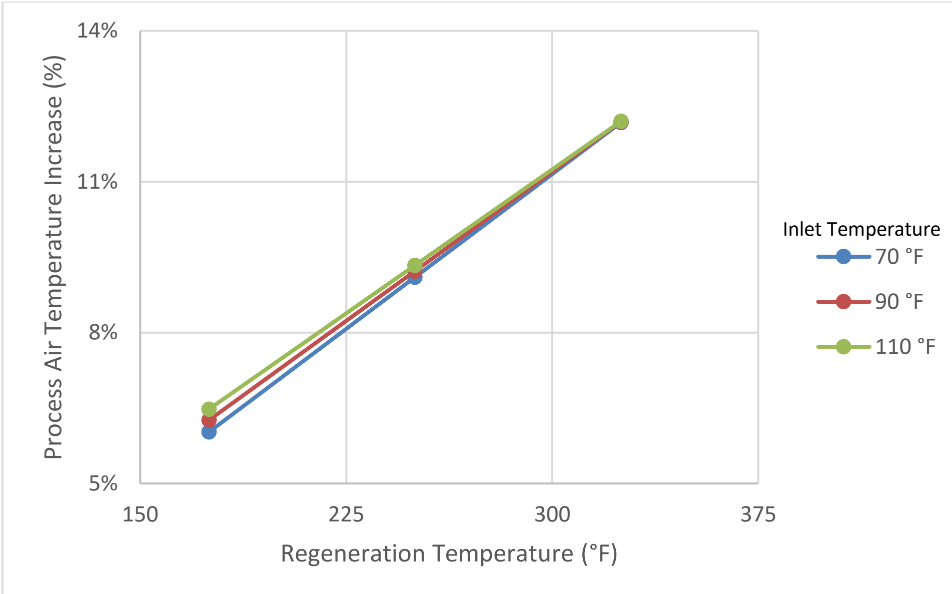

Figure 89: Change in temperature of air at 90% relative humidity and a temperature ranging from 70 to 110 °F due to the heat of sorption generated by dehumidification with the WSG regenerated by a temperature ranging from 175 to 325 °F

In contrast to the percent increase of temperature of the air being dehumidified in the case of 60% relative humidity that ranges from 5-11%, the air being dehumidified from 90% has a percent increase in temperature ranging from 6-12%. Therefore, air with more water vapor experiences a greater increase of temperature when dehumidified by the WSG. The suppression of the intermediate pressure dew point for air at 90% relative humidity is 3-6% if the air is initially at a temperature of 70 °F, and 1-2% if initially at a temperature 110 °F. However, for air with 90% relative humidity, the inlet air temperature is only recoverable when the air is initially at a temperature of 70 °F and is dehumidified by the WSG using a regeneration temperature of 325 °F.

Air at 90% relative humidity increases the specific work of a TSC by 3-7%. In all cases of air with that much moisture, the WSG significantly increases the temperature of the air, and can only enable intercooling to the initial air temperature when the air enters at a temperature of

70 °F and is dehumidified by the wheel using a regeneration temperature of 325 °F, the specific work of the DTSC will always increase in these conditions due to the massive heat influx required to supply regeneration air. The change of specific work of the DTSC dehumidified with regeneration temperatures ranging from 175 to 325 °F relative to the MTSC for air at a relative humidity of 90% and a temperature ranging from 70 to 110 °F discharged to pressures ranging from 75 to 150 psig is shown by Figure 90.

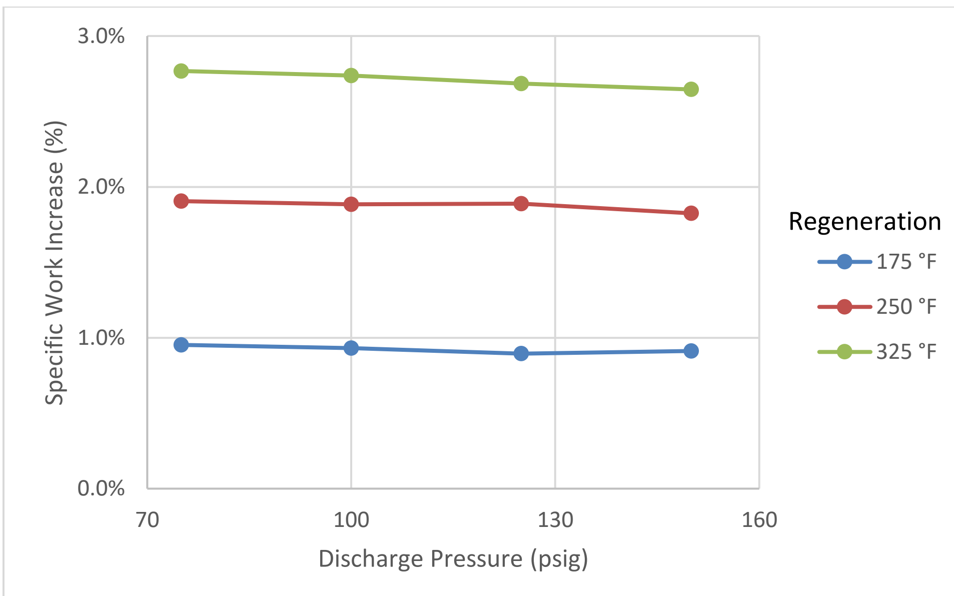

Figure 90: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 70 °F and 90% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

For air with 90% relative humidity and an initial temperature of 70 °F, the specific work of the DTSC is greater than the MTSC, as shown by Figure 90. At a regeneration temperature of 175 °F, the specific work increases by 1%. When the WSG is dehumidified by a regeneration airstream at a temperature of 250 °F, the specific work of the compressor increases by 2%. If the regeneration temperature is 325 °F, the specific work of the compressor increases by approximately 3%. The change of specific energy consumed by the system when considering the specific heat required to increase the temperature of the return air from a temperature of 75 °F at 50% relative humidity to the regeneration temperatures is shown by Figure 91.

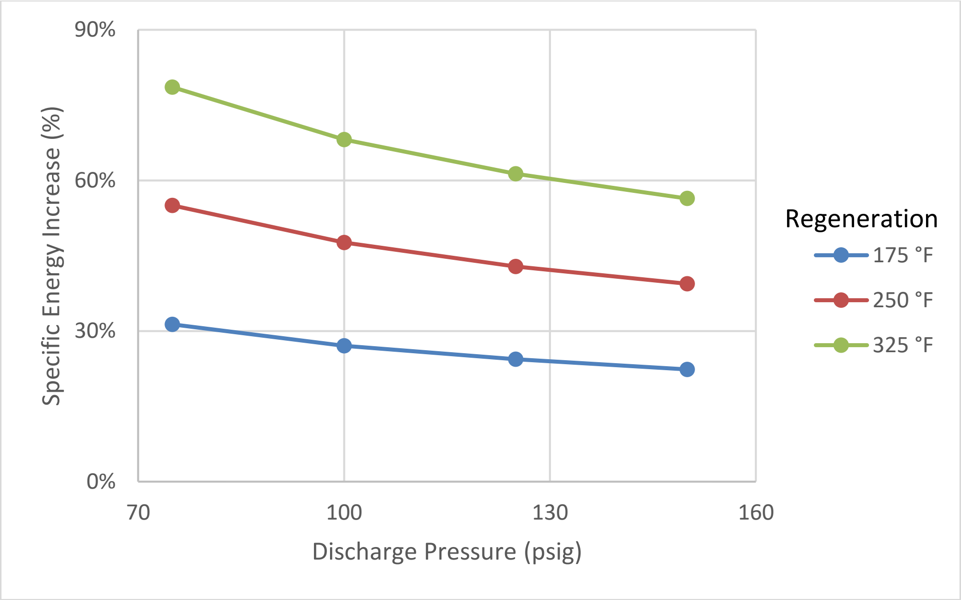

Figure 91: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 70 °F and 90% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

Like the DTSC package compared to the MTSC at 60% relative humidity, the specific energy consumption of the system increases by 22-31% if the regeneration temperature is 175 °F, 39-55% if the regeneration temperature is 250 °F, and 56-79% if the regeneration temperature is 350 °F. As the initial temperature of the moist air increases, the WSG will require hotter regeneration air to scavenge the moisture. At the same regeneration temperatures, the intermediate pressure dew point suppression falls as the moist air temperature increases, and the intercooler will not be able to restore the initial air temperature. The specific work of the twostage air compressor of dehumidified air initially at a temperature of 90 °F and 90% relative humidity is shown by Figure 92.

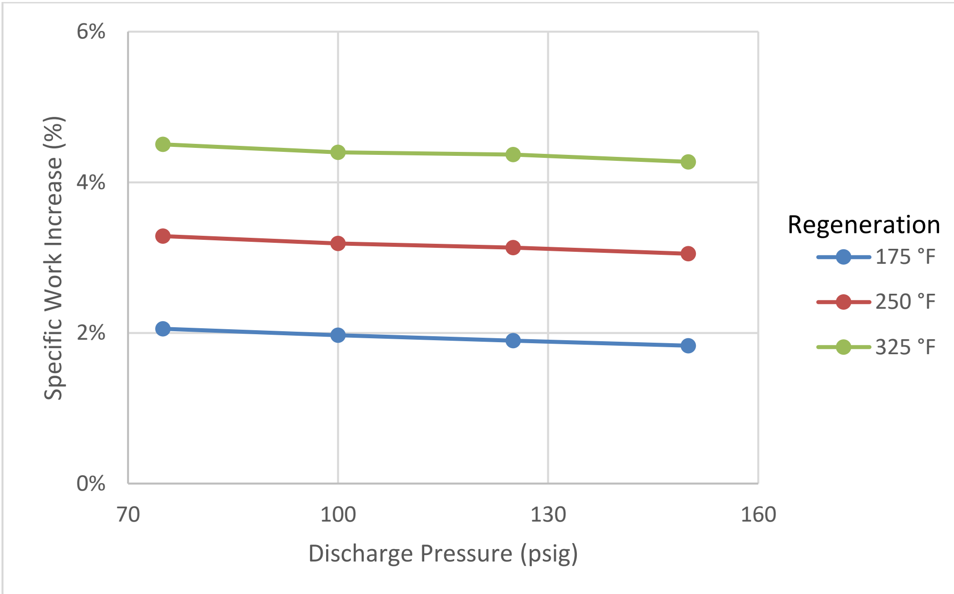

Figure 92: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 90 °F and 90% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

The increase of air temperature caused by the heat of sorption coupled with the decrease of intermediate pressure dew point suppression results in greater specific work increase for TSC of air initially at a temperature of 90 °F and 90% relative humidity rather than an initial temperature of 70 °F. Figure 92 indicates that the increase in regeneration temperature results in greater work for a TSC. At a regeneration temperature of 175 °F, the DTSC requires an additional 2% specific work than the MTSC. The DTSC using a regeneration temperature of 250

°F requires approximately 3.5% more specific work than the MTSC. Being regenerated by an airstream at a temperature of 325 °F, the DTSC requires 5% more specific work than the MTSC.

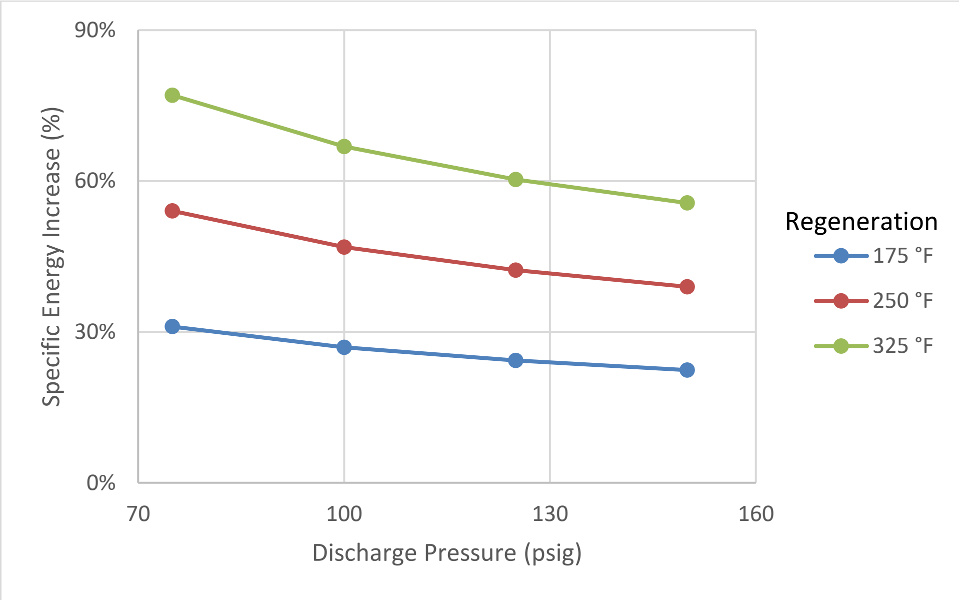

Figure 93: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 90 °F and 90% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

The increase of the specific energy of the system is significantly compounded by the heat required to increase the temperature of the return air to the regeneration temperatures. Figure 93 shows that if the regeneration temperature is 175 °F, the specific energy of the system increases by 22-31%, while it increases by 39-54% if the regeneration temperature is 250 °F, and 56-77% when the regeneration temperature is 325 °F. If the non-linear behavior continues, the air initially at a temperature of 110 °F at 90% relative humidity should have a lesser increase of specific work than that of the air initially at 90 °F. The change in specific work of the DTSC relative to the MTSC is shown by Figure 94.

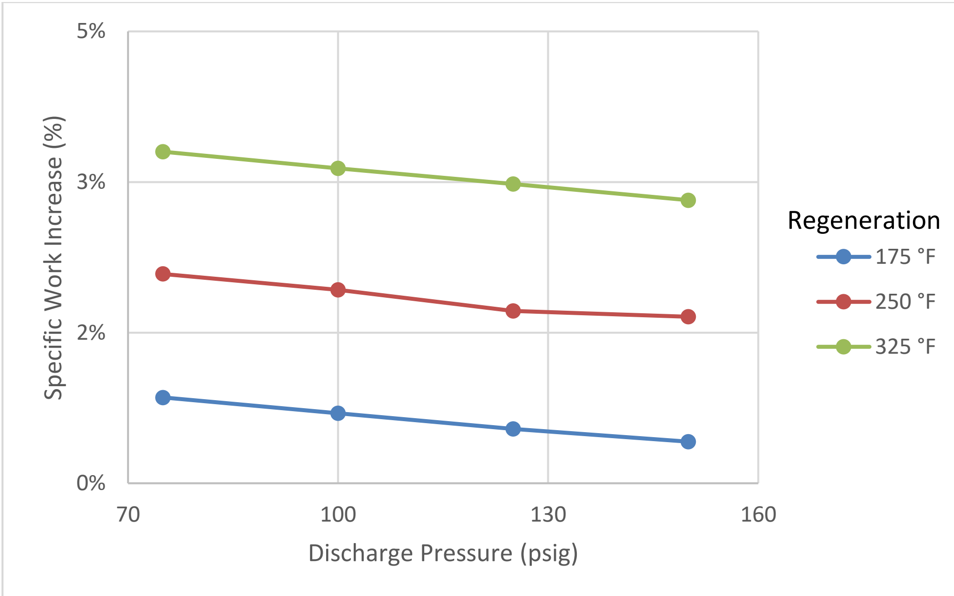

Figure 94: Change in specific work of the two-stage air compressor with a discharge pressure ranging from 75 to 150 psig from the moist air case using air initially at a temperature of 110 °F and 90% relative humidity that is dehumidified by a WSG using a regeneration airstream at a temperature ranging from 175 to 325 °F

Recall that the increase of specific work of the TSC for air initially at a temperature of 90 °F and 90% relative humidity ranged from 2-5%. Figure 94 shows that the increase of specific work of two-stage air compression of air initially at a temperature of 110 °F and 90% relative humidity ranges from 1-4% as the regeneration airstream temperature of the WSG increases from 175 to 325 °F, which is expected because of the non-linear intermediate pressure dew point suppression. The total specific energy consumption of the DTSC system using air initially at a temperature of 110 °F and 90% relative humidity should be very similar to the previous results because the order of magnitude of the specific heat to increase the return air's temperature outweighs the change in specific work of a TSC, as shown by Figure 95.

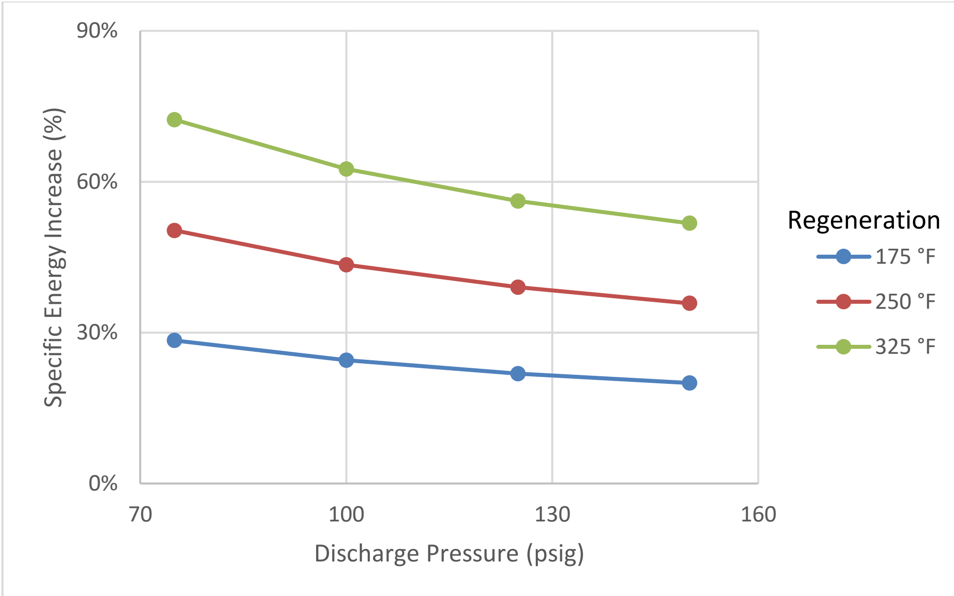

Figure 95: Change in specific energy when compounding the specific work of the dehumidified two-stage air compressor discharging the air to a pressure ranging from 75 to 150 psig using air initially at a temperature of 110 °F and 90% relative humidity with the specific energy to heat the return air from a temperature of 75 °F and 50% relative humidity to a regeneration temperature ranging from 175 °F to 325 °F

According to Figure 95, if the regeneration temperature is 175 °F, the specific energy consumption of the system increases by 20-28%, while it increases by 36-50% if the regeneration temperature is 250 °F, and by 52-72% for a regeneration temperature of 325 °F. The increasing specific work results from the process outlet temperature increase. The specific energy required to heat the return air is equivalent to the change in enthalpy from a temperature of 75 °F and relative humidity of 50% to the regeneration temperature. Because the process outlet temperature outweighs the dew point suppression, the TSC requires additional specific work. Furthermore, the specific energy of heating the return air compounds the specific energy demand.

9. CONCLUSION

To investigate dehumidification of inlet air for compressed air systems (CAS), it was necessary to introduce an understanding of the theory and physics of air compression with the single-stage air compressor. Compression theory was then extended to the two-stage air compressor (TSC), which is a more efficient system because the air is partially compressed in the first stage of the compressor to an optimum intermediate pressure, cooled in an intercooler, and compressed to the desired discharge pressure in the second stage of the compressor. The presence of humidity in the inlet air however, is of special importance for this study, because it inhibits intercooling if condensation is to be avoided, which means that opportunities for reducing specific work with the TSC are diminished by as much as 7%. In preparation for quantifying the effect of dehumidifying the TSC inlet air, it was necessary to understand the characterization of atmospheric air (air), as well as the calculation of humidity and its behavior during compression. To complete the analysis, a range of inlet air temperature and humidity was investigated for both the single-stage and two-stage air compressor with a focus on deviations from the ideal case of dry-air compression to identify the benefits associated with dehumidifying the air prior to compression.

Two methods of dehumidification were parametrically studied to determine the net changes in specific energy consumption if a CAS was modified by installing a dehumidifier to the inlet of a TSC when compared to a standalone TSC without dehumidification. The first method of dehumidification investigated was a vapor compression refrigeration cycle (i.e. air conditioner) that cooled the air to a temperature of 45 °F by circulating refrigerant 410A at an isentropic efficiency of 80%, which was modeled using principles of thermodynamics. The second method of dehumidification was a silica gel desiccant wheel that was modeled by using a fractional factorial experiment evaluated at assumed test conditions.

Air conditioning by using vapor-compression technology and inlet air cooling to 45 °F reduced the specific work consumption of the TSC to a maximum value of 11% as the inlet air temperature increased and discharge pressure decreased. In contrast, the desiccant wheel increased the specific work consumption of the TSC by up to 5% as the initial temperature of the air increased. Although moisture removal with the desiccant wheel enabled lower pressure dew points, the TSC required more specific work because the heat released during adsorption of the water vapor to the desiccant further increases the temperature of the air entering the compressor. In addition, considering the direct heat required to regenerate the desiccant wheel, the system can consume up to 80% more specific energy in the form of thermal energy than the standalone TSC without dehumidification.

Despite the results reported in this study, analysis of the trends of the dehumidified systems suggests that developing other features of the models, along with combining the desiccant wheel with the air conditioner may present beneficial results. For example, an ideal air conditioner performs as the model describes, but significant humidity may interfere with cooling the air, in which case dehumidifying the air with a desiccant wheel prior to the air conditioner may be beneficial. In addition, rather than supplying direct heat to the airstream that regenerates the desiccant, waste heat from the compressors, the compressor intercooler, and the high temperature air exiting the second stage of the compressor, as well as other sources, can supplement the heat for the regeneration airstream. Furthermore, if combined with the air conditioner, additional waste heat could be recovered from the refrigeration cycle condenser.

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