

Water Collection

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Abstract

This paper covers a thermodynamic simulation of a vapor compression cycle to generate drinkable water from the surrounding humid air. A vapor compression cycle is normally used for cooling air, such as a refrigerator or an air conditioner in a home. The main components of a VCC are an evaporator, condenser, compressor, and expansion valve. The VCC produces water by condensing water vapor along the outer surface of the evaporator as a result of heat transfer between the working fluid R-1234ze and the ambient air. The main goal of the simulation was to find suitable pressure, temperature, and enthalpy state points suitable to produce 500cc/hr of drinkable water and system parameters such as the energy consumed by the compressor and the cooling factor of the evaporator. This was done by producing a MATLAB code to solve a system of equations based on governing equations of the system components, as well as property correlations. Additionally, a parametric study was completed to examine how the mass flow rate, overall heat transfer coefficients and the amount of water yielded affect the coefficient of performance.

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Nomenclature

Symbol	Description	Units
U	Overall Heat Transfer Coefficient	W/m^2k
A	Area of heat Exchanger	m^2
D_i	Internal Heat Exchanger Diameter	m
D_o	External Heat Exchanger Diameter	m
L	Length of Heat Exchanger	m
T_s	Saturation Temperature of water vapor	C°
T_{1-4}	State Point Temperature	C°
Q_{in}	Evaporator Heat Rate	W
Q_{out}	Condenser Heat Rate	W
\dot{m}	Mass Flow Rate	kg/s
h_{fg}	Change in Enthalpy of Water Vapor	kJ/kg
h_{1-4}	State Point Enthalpy	kJ/kg
W_{in}	Compressor Work	W
η	Isentropic Efficiency	Unitless
R	Ideal Gas Constant	$J/mol K$
p_{1-4}	State Point Pressure	kPa
T_{iceb}	Temperature of Ice Bath	C°
T_{del}	Log Mean Temperature Difference	C°
Nu	Nusselt Number	Unitless
Pr	Prandtl Number	Unitless
Re	Reynolds Number	Unitless
h_{cond}	Convection Coefficient of Refrigerant Inside Condenser	W/m^2k
h_{evap}	Convection Coefficient of Refrigerant Inside Evaporator	W/m^2k
μ	Dynamic Viscosity	$Pa.s$

ρ	Density	kg/m^3
K	Thermal Conductivity	W/mK
k	Specific Heat Ratio	Unitless

Figure 1: Nomenclature

Executive Summary

Currently, access to clean, drinkable water is threatened all around the world. Potable water enables people to stay healthy, produce food, and improve sanitization. In turn, clean water can improve a country's economy and overall well being. As of 2017, 2.2 billion people were unable to access safely managed drinking water services. To combat this water scarcity, multiple technologies exist, including, water distillation plants, fog catchers, and cheaper methods such as charcoal filters. This project explores the use of a vapor compression cycle (VCC) to capture water along the outside surface of an evaporator. To model the VCC a system simulation was performed to understand the state points of the selected refrigerant R1234ze, as well as its system components. As a result, the coefficient of performance (COP) was 4.48, and the cooling factor of the evaporator was .3519 kW. Additionally, the goal of condensing 500cc/hr of water was achieved. The pressure, temperature, and enthalpy, at the outlet of the evaporator, was 1.77 Bars, -5 C°, and 380 kJ/kg respectively. Outside the compressor, the state point was 5 Bars, 38 C°, and 413 kJ/kg. At the outlet of the condenser, the state point was 5 Bars, 24.9 C°, and 233 kJ/kg. Finally, at the outlet of the expansion valve the state point was 1.77 Bars, -5 C°, and 233 kJ/kg. An additional portion of this project included a parametric study of the system parameters of the VCC. The design parameters that were varied were the UA value of the evaporator, the heat transfer at the evaporator, and the mass flow rate. As a result, it was shown that as the UA value increases, the COP increases, as the heat transfer at the evaporator increase so too does the COP, and finally, as the mass flow rate increases, the COP decreases.

Introduction

Currently, access to clean, drinkable water is threatened all around the world. Potable water enables people to stay healthy, produce food, and improve sanitization. In turn, clean water can improve a country's economy and overall well being (“Drinking-Water”,2019). As of 2017, 2.2 billion people were unable to access safely managed drinking water services, or water that is, “located on-premises, available when needed, and free from fecal and priority chemical contamination” (“Drinking-Water”,2019). In the future, water scarcity will be an even greater, issues such as climate change, urbanization, and demographic changes will further strain the earth’s potable water supplies(“Drinking-Water”,2019). To combat this problem, many solutions exist today, each design having benefits and drawbacks specific to geographical region limitations. One promising solution is fog nets. These nets catch water droplets in fog using a thin mesh that has holes from 1-40 millionths of a meter (“The “Fog Catcher” Project – Peru”,n.a). Six hundred square meters of netting can produce enough water for up to 500 people. One limitation of this method is that these nets must be placed at a high altitude, so there must be mountains, and the area must have a constant supply of fog. However, this solution greatly benefits regions that do not have a constant water supply in another form, such as a river. For geographical regions that are not foggy but have contaminated water supplies, filtering their water supply is often the best solution. Some cheap ways to do this include using purification salts, charcoal filters, and even using the sun’s radiation to decontaminate water.

The Omniprocessor has been in development in partnership with Bill Gates and represents a more costly solution (Gates,2015). This machine turns human feces into drinkable water and has the dual purpose of water production and waste removal. This machine will have the capability of removing the waste of 100,000 people and producing 86,000 liters of water per day. An obvious drawback of this solution is that right now it is very expensive compared to other techniques.

Using ambient air to condense water is a possible solution that does not have some of the previously mentioned drawbacks that many other solutions have. The greatest benefit to using humid air for water generation is that it can be used almost anywhere in the world, and provides reliable clean drinking water at a far reduced cost than other more complex methods. This project will focus on a VCC to produce potable water.

Water Treatment Technology

Generally, water treatment is the process of taking a water source and purifying it, to produce drinkable water. The creation of a VCC to produce drinkable water is another form of water treatment, utilizing water vapor in the air as the source that is being treated. To get a sense of the benefits and limitations of the VCC, it is useful to cover other methods of water treatment.

The most common form of water treatment to produce drinkable water is desalination. Desalination is the process of taking seawater and removing the salt within it, as well as other containments. The two most common forms of desalination are thermal distillation and reverse osmosis(RO). Thermal distillation involves boiling water to evaporate it and then condensing the vapor back into water. RO works by using high pressure to force water through a filtration system thus removing containments (Woodward,2020).

Thermal Distillation

Thermal Distillation is used to produce drinkable water through a process that replicates the hydrogen cycle. This cycle can is shown in figure 2.

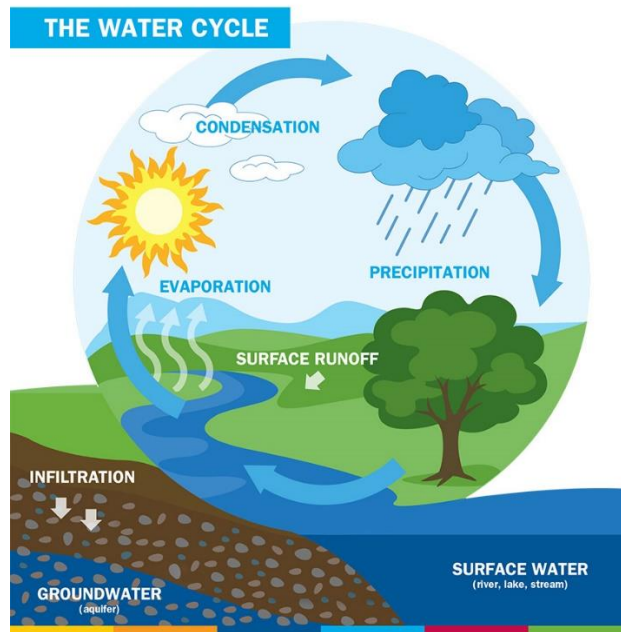


Figure 2: Hydrological Cycle (“What is the Hydrological Cycle?”, n.a)

A hydrological cycle, is the natural process of evaporating water into the atmosphere, condensing it, and returning the vapor as a liquid through precipitation. This process is leveraged by thermal distillation by evaporating water, causing it to be separated from its components, and thus creating distilled water (Kucera,2005). One method of thermal distillation involves a vapor compression cycle, like the one being utilized in this project. The only difference is that water is used as the working fluid. It goes through the same stages as the refrigerant would go through. It is first boiled using a heat exchanger, then its pressure is raised with a compressor, and finally, it is condensed with another heat exchanger. Utilizing a VCC can produce 1 gallon of water with .12 kWh of energy (Kucera,2005).

An alternative form of thermal distillation uses a process called the multiple effect. The multiple effect takes advantage of using multiple boilers to sequentially boil the water in the proceeding boilers. This method is beneficial because it saves energy by recycling the boiled

water to produce more boiling water. Compared to a VCC this system can be more efficient for smaller-scale applications (Kucera,2005).

Reverse Osmosis

Reverse Osmosis produces distilled water, but through a much simpler process than that of thermal distillation. In this process, water is forced through a semi-permeable membrane, moving water from an area high in concentration of containments to an area of low concentration (Woodward,2020). This process is shown in figure 3.

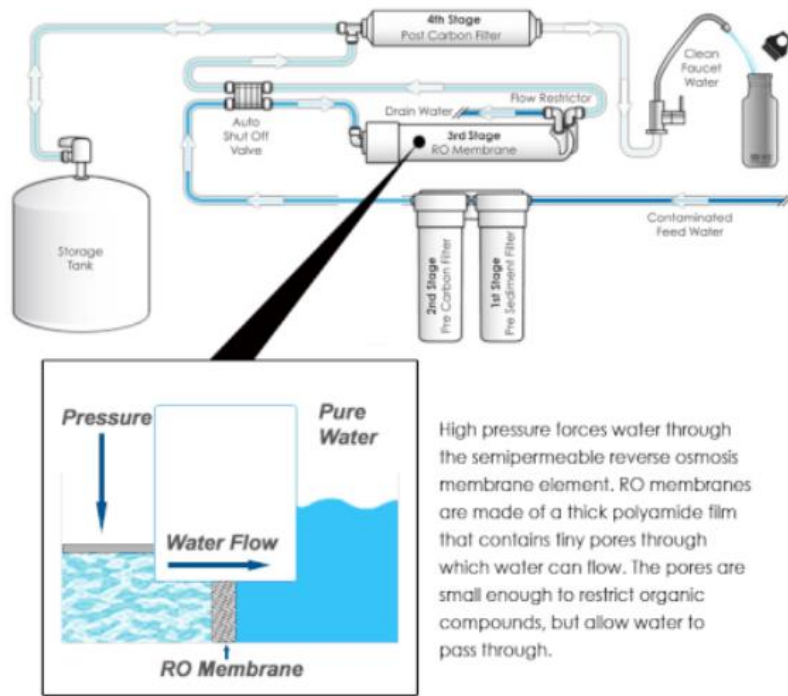


Figure 3: Reverse Osmosis (Woodward,2020)

VCC

The system that will be used in this project utilizes water vapor in the air to produce potable water is a vapor compression cycle. The VCC is around 200 years old and is normally utilized to force heat out of a hot reservoir to a cold one, requiring an input of energy, as heat normally moves from an area that is hot to a cold area. This system can be utilized in air conditioning units as well as refrigerators. A basic VCC system consists of 4 parts, a condenser, evaporator, compressor, and expansion valve, and utilizes a refrigerant as a working fluid to promote heat transfer from the ambient air to the VCC. The system begins in the evaporator, by exchanging heat between the air, and the refrigerant, this causes the air temperature to decrease, and the refrigerant to gain energy as well. This is the phase of the VCC that gives you “refrigeration”, however, the job of the system is not done, as you have a refrigerant, that can no longer cool ambient air as effectively, as it has a greater enthalpy. One solution to this problem could be to use a cooler refrigerant that could return the working fluid to its original state; however, a cooler refrigerant would be more expensive. A more economical solution to return the refrigerant to its original state is to pass the refrigerant through a compressor, a condenser, and finally an expansion valve (Brokowski,1997).

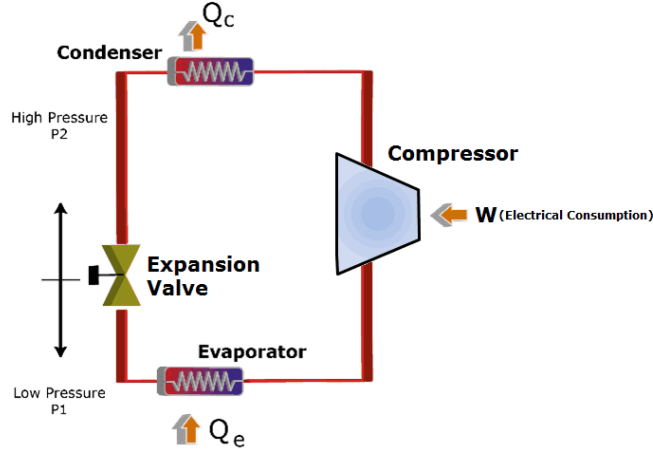


Figure 4: VCC (The Vapor Compression Refrigeration Cycle, n.a)

First, the compressor does work into the refrigerant, causing the enthalpy, and temperature and pressure to go up. By increasing the temperature and pressure of the refrigerant, ambient air can be used to cool the refrigerant, passing energy from a point of the higher temperature refrigerant, to the cooler ambient air. From here, a condenser is used to decrease the enthalpy of the refrigerant, and an expansion valve is used to decrease the temperature and pressure without changing the enthalpy. At this point, the refrigerant is at its original state and is ready to repeat the process within the cycle.

Besides refrigeration, the VCC can be used to produce potable water on the surface of the evaporator. This is caused by the air losing energy to the refrigerant, and water vapor condensing along the evaporator due to this loss of energy. Due to thermodynamics, the amount of energy gained by the refrigerant will be the same amount of energy lost by the ambient air. As a result of this, the more energy that the evaporator can transfer, the more water will condense on the evaporator (Brokowski,1997).

Potential areas of failure of VCC

Though a VCC system is successful in producing potable water, multiple design considerations must be accounted for the system to function properly. One such problem is leakage of the refrigerant from the compressor, as well as any refrigerant line. The leakage of the refrigerant can cause damage to other parts of the system and can be expensive to replace. Another factor to consider is keeping the refrigerant lines clean of sedimentation, scale formation, or slime, which can all lead to performance issues.

Last year's MQP group

This project is a continuation of an MQP group in 2019, consisting of Christopher Bolsinger and Spencer Ralphs. Their group was tasked with creating a VCC system that was capable of producing 500cc/hr of potable water at an internal ambient temperature of 70 degrees Fahrenheit and 50% relative humidity. As a result of their MQP, this group was able to design a system that was able to generate 25cc of water per hour, far below the stated goal, however acceptable as 500cc/hr was an arbitrary number for system simulation. As stated in their report some flaws in their design included low Q values for both the condenser and the evaporator and low surface area in contact with the ambient air. Their report suggested using a flat plate or a coiled heat exchanger to improve this factor.

The purpose of this project is to increase the amount of water condensed on the evaporator. For simulation purposes, 500cc/hr will be a design constraint, however, a successful project will be achieved if the new system produces more than 25cc of water per hour. To measure if the new system is more successful, the COP of performance will be compared, in this project the hope is to increase the COP.

Ways to improve heat exchangers

To improve the VCC system, the COP value, or $COP = Q_{in} / W_{in}$ must go up. This equation takes the ratio of Q into the evaporator and takes the amount of work put into the compressor and finds a ratio between them. The higher the number the better, as the design should be cooling as much as possible, while inputting as low of work as possible (COP,n.a). Last year's group was able to achieve a COP of 3.01, for this project to be successful, the VCC must achieve a higher COP.

The primary way to improve the COP value is to improve the heat exchange between the ambient air and the evaporator. Possible ways of doing this include, replacing the refrigerant with a cooler one, adding forced convection to the heat exchangers with fans, improving the conductivity of the heat exchangers or enhancing the surface area of the heat exchanger. This will cause the overall U, or overall convection coefficient to increase, thus increasing the overall heat transfer.

Refrigerant

To properly choose a refrigerant, some design considerations must be taken into account for the VCC system to properly function. These are as follows:

1. The saturation temperature of the working fluid at the operating pressure of the condenser must be greater than the temperature of the hot reservoir in order to reject heat.
2. The saturation pressure of the refrigerant should not be too high because a condenser that is capable of operating at high pressures is much more expensive and hazardous.
3. The saturation temperature of the working fluid at the operating pressure of the evaporator must be less than the temperature of the cold reservoir to absorb heat.
4. The saturation pressure of the refrigerant should not be less than atmospheric pressure because air leaks into the system can severely decrease the COP (“Criteria For Refrigerant Selection,2005).

Based on these constraints, the working fluid of R1234ze, which was chosen by the 2019 group meets these requirements.

System Simulation

To design a vapor compression cycle that produces 500cc/hr of water, a system simulation was performed to find the state points of the working fluid around the system, as well as the operating parameters such as the energy the compressor consumes, the cooling factor of the evaporator, and the amount of energy the condenser exchanges. A system simulation is a process of creating a computer model that takes it to account, the interconnection between system components, as well as governing laws such as energy conservation. To simulate the vapor compression system, a set of equations was developed with X unknowns and X variables, to solve for the unknowns. This process was completed in Matlab using a non-linear equation solver to solve for our system's unknown parameters. It took into account governing equations based on the system components, and property correlations of the working fluid.

Setting up the Simulation

To produce the system, first, the state points must be set up relative to the VCC. These state points are numbered 1-4 and represent the different pressure, enthalpy, and temperature at each state point. The numbered state points are shown in figure 4.

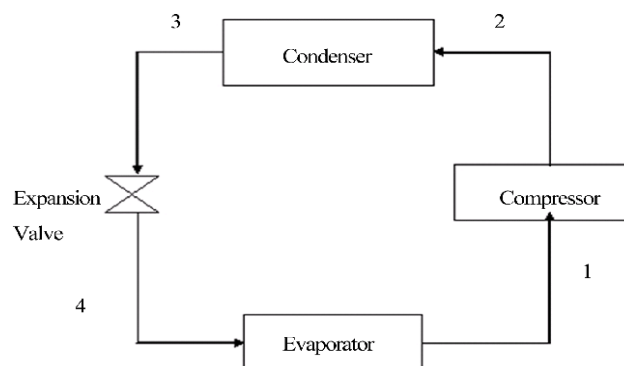


Figure 5: VCC State Point Numbers

These numbers are chosen based on the start point, and the direction of the VCC.

The first parameter that must be accounted for is the cooling factor or the Q_{in} of the evaporator. The Q_{in} is the amount of energy the refrigerant will gain, and thus the amount of energy the water vapor in the air will lose, according to the conservation of energy. To establish this value, we must first know the saturation of the water vapor within the air. The T_{sat} of the evaporator is calculated using the following formula.

$$T_{sat} = \frac{237.7 \left(\frac{17.27T}{237.7+T} + \ln(RH) \right)}{17.27 - \left(\frac{17.27T}{237.7+T} + \ln(RH) \right)} \quad (\text{Relative Humidity, 2018})$$

This formula uses the value of relative humidity and temperature of the ambient air to calculate at what temperature the water vapor in the air will condense. The VCC will be operating within an indoor lab space where it can be assumed these values are standard values within a room. Based on these factors, relative humidity of 50% was chosen, and a room temperature of 20 C°(Housh,2017). As a result, the T_{sat} of the water vapor in the air is 9.25 C°. This saturation temperature represents the temperature of the outside surface of the evaporator. A final value of 5 C° was chosen to give room for error, and make sure the water vapor will condense into water. To calculate the Q_{in} , the following formula is utilized,

$$Q_{in} = \dot{m}_w \cdot h_{fg}$$

Where the \dot{m}_w is calculated by converting the volumetric flow rate of 500 cc/hr to a mass flow rate using the known density of water of 997 kg/m³. As a result, the mass flow rate is .000139 kg/s. Additionally, h_{fg} equals 2489 kJ/kg, which is the change in enthalpy it takes to condense water vapor into water. Based on this equation Q_{in} equals, .345 kW. To fully simulate our system, the Q_{in} was not assumed, however, we will be using this value as an initial guess to produce a simulation that condenses roughly 500 cc/hr of drinkable water.

After the initial perimeters were calculated, the specific areas of where the state points could be must be factored in. This is an important step because the state point equations and governing equations rely on knowing the relative position of each state point.

State Point 1

State point one is assumed to be along the saturated vapor line of the pressure vs. enthalpy diagram. For the refrigerant R1234ze, this diagram is pictured below.

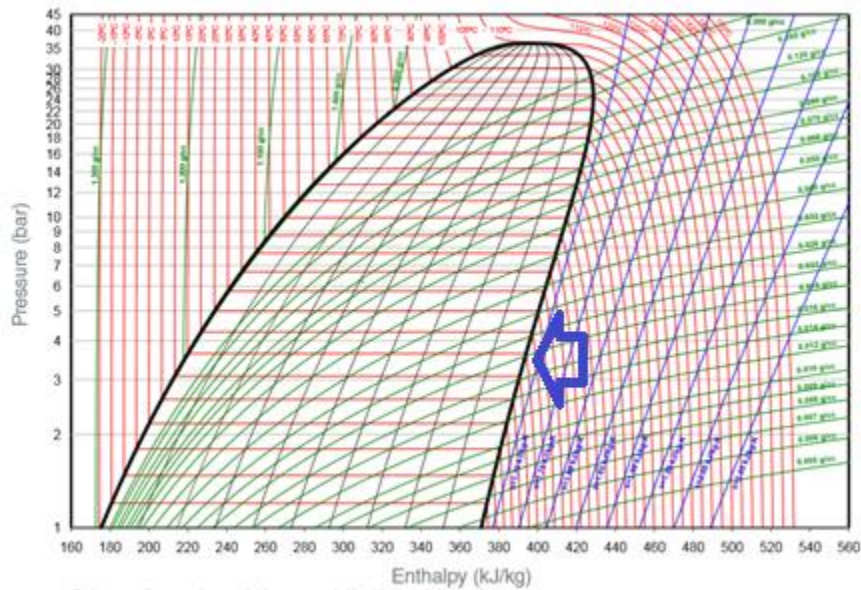


Figure 6: Pressure Vs. Enthalpy of R1234ze

In this figure, the saturated vapor line is the bold black line on the right side of the dome in the center of the graph, pointed out by a blue arrow. At this point, the refrigerant is a vapor, however, it remains the same temperature it would be within the dome. It is the same temperature, because any point within the dome, along a horizontal line, all the energy being gained is changing the fluids state, and not its temperature or pressure (Brokowski,1997).

State Point 2

State point 2 is assumed to be within the superheated region outside of the saturation dome. This is because the compressor will cause the refrigerant to increase in temperature and pressure. In an ideal case, the compressor will follow an isentropic process, forming a linear line from state point 1-2, where the entropy of the refrigerant remains the same. These lines are represented by the diagonal blue lines to the right of the saturation dome (“Isentropic Efficiency”, n.a). For the case of this simulation, the isentropic efficiency is assumed to be 80%, which will require more work to produce the same outlet pressure and at a greater temperature. This will result in a line that has a slope that is less than that of the isentropic efficiency line. The superheated region is shown in figure 7 (“Isentropic Efficiency”, n.a).

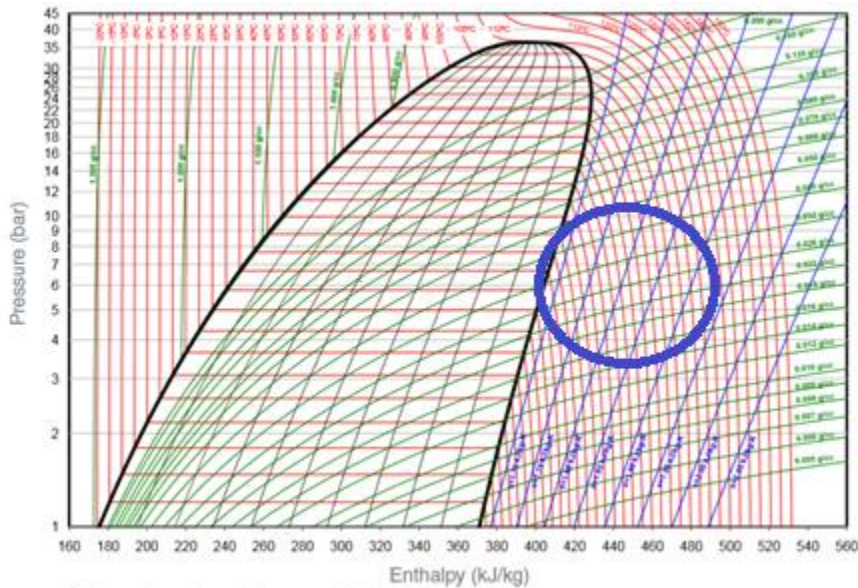


Figure 7: Super Heated Region

State Point 3

Like state point 1, state point three is assumed to be along a saturation line, except this time it is along the saturated liquid line. The saturated liquid line is the bold black line along the left side of the saturated line, shown in figure 6. For both state points 1 and 3, being on the

saturated liquid and vapor lines is desirable because isothermal heat transfer is more efficient than that of a non-isothermal process (Brokowski,1997).

State Point 4

State point 4 is assumed to be within the saturation dome. This is a result of the refrigerant going through an expansion valve and decreasing in temperature, and pressure from state point 3, while enthalpy remains the same. This area is shown in figure 8.

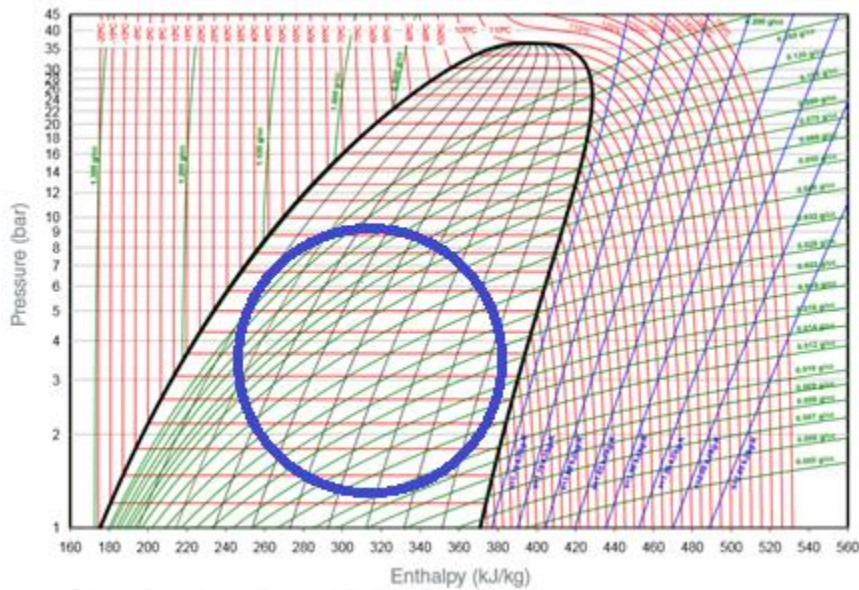


Figure 8: Saturation Dome

Governing Equations

The governing equations for the system simulation are based on the system components within the VCC. The components include an evaporator, a compressor, a condenser, and an expansion valve. For the evaporator and the condenser, there should be 2 equations represented for each. The compressor and expansion valve should each have 1 equation represented.

Evaporator

The two equations that represent the evaporator are as follows,

$$(1). Q_{in} = U_1 A_1 (T_s - T_1)$$

$$(2). Q_{in} = \dot{m}(h_1 - h_4).$$

The first equation is representative of the heat transfer between the internal temperature of the working fluid within the evaporator, T_1 , to the outside surface temperature of the evaporator or T_s . U_1 is representative of the overall heat transfer coefficient through the thermal circuit that is defined by the convection coefficient of the refrigerant, and the conduction coefficient through the evaporator. A_1 is the overall area of the evaporator the heat being transfer through. In this case, it is assumed the internal temperature at the inlet and outlet of the evaporator is the same because both temperatures are within the saturation dome, so the temperature remains constant. Because of this, we can substitute T_1 with T_4 because $T_1 = T_4$.

The second equation represents the overall change in enthalpy of the refrigerant as a result of the gain in energy from transferring heat with the ambient air. The overall energy lost from the ambient air should equal the energy gained by the refrigerant, resulting in an increase in enthalpy within the refrigerant.

Compressor

For the compressor the governing equation is,

$$(3). W_{in} = \frac{\dot{m}R(T_1+273)\frac{k}{k-1}}{\eta} \left(\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right).$$

This equation relates the ratio of outlet and inlet pressure, the mass flow rate, and the initial temperature to the work done of the refrigerant within the compressor. This equation calculates the work if it followed isentropic efficiency, then divides by an assumed isentropic efficiency to account for the process of not being an ideal case. The assumed isentropic efficiency is 80%. Based on this, the isentropic work for the compressor will be lower than the actual work after the isentropic value is divided by .8(“Isentropic Efficiencies”,n.a).

Condenser

The equations for the condenser are very similar to that of the evaporator. However, there is one key difference, which is the fact that at state point 2 the refrigerant is superheated. This means that T_2 is not equal to T_3 and we cannot use the same heat transfer formula as used previously. To account for this difference, the log mean temperature difference (LMTD) was utilized (Herring,2012). The LMTD takes a logarithmic average of the two temperatures between state points 2 and 3. As a result, the equation for the heat transfer of the condenser is,

$$(4). Q_{out} = U_2 A_2 (T_{del})$$

Where T_{del} equals,

$$T_{del} = \frac{(T_2 - T_{iceb}) - (T_3 - T_{iceb})}{\log\left(\frac{T_2 - T_{iceb}}{T_3 - T_{iceb}}\right)}$$

An additional modification for the condenser is utilizing an ice bath to improve the heat transfer of the condenser. The assumed value of the ice bath is $5C^\circ$. Ideally, an ice bath is $0C^\circ$, however, over time it will increase its temperature as it loses energy to the ambient air, $5C^\circ$ was chosen to account for this fact. Also, because the surface temperature of the evaporator is not assumed, the thermal circuit is from the inside the condenser to the ice water outside of the condenser.

The second equation for the condenser is the same as the evaporator, however, it represents the change in energy of the refrigerant from state point 2 to state point 3, and is equal to the Q_{out} of the condenser.

$$(5). Q_{out} = \dot{m}(h_2 - h_3).$$

Expansion Valve

The final component equation for the system is for the expansion valve. An expansion valve works by decreasing the pressure and temperature of the refrigerant while doing no work

on it (Throttling Process Thermodynamics, n.a) As a result, the enthalpy of the refrigerant remains constant.

$$(h_3 = h_4).$$

Property Equations

In addition to governing equations, the system of equations relies on property equations to fill in the missing unknowns which relates the pressure, enthalpy, and temperature of the refrigerant at different state points. For many refrigerants, these properties are available, however since R1234ze is not a commonly used refrigerant, these equations were not available in a form that would be appropriate for this simulation. A way to solve this issue is to curve fit property correlations based on many different state points within an acceptable region for each state point of the simulation.

The data for the state points were gathered using Honeywell's Gentron software, which gives data points based on using two input values of temperature, pressure, or enthalpy. Curve fits were produced using MATLAB's curve fitting software. The steps to do so are as follows.

1. Create matrices of each property with the different values, 20 unique points were used in each matrix.
2. Upload matrices into curve fitting tool
3. Select the appropriate correlation, ie. linear, power, polynomial
4. Make sure the goodness of fit value is as close to 1 as possible
5. Check the appropriate of each equation by solving for different state points and see how accurately it represents the state point

Property Correlation for State Point 1

To produce a curve fit for state point 1, data points were produced along the saturated vapor line. Since along the saturated vapor line the enthalpy only depends on either temperature or pressure and not both, data points of enthalpy vs. pressure were collected. Doing so produces an equation that relates the enthalpy of the refrigerant to its pressure. The resulting equation is,

$$h_1 = gp_1^{b_1}$$

(All variables other than, p,T, and h, are coefficients produced by the curve fit)

Property Correlation for State Point 2

For state point, 2 data points were collected within the superheated region as depicted in figure 7. For state point 2, data points were collected based on h(T,p), because in this region the enthalpy is dependant on pressure and temperature. The curve fit produced was,

$$h_2 = 389.4 - .02426p_2 + .8569T_2 + .0001352T_2p_2 + .0005619 T_2^2$$

Property Correlation State Point 3

For state point 3 the same method was used as state point one, however, data points were collected along the saturation liquid line. This produced the following equation,

$$h_3 = g2p_3^{b_2} + C$$

Other Considerations

The final two sets of equations that will be used to fully constrain the system is noting that the pressure drop across the evaporator and condenser are zero. The lack of pressure drop is assumed based on an ideal case. In a real-world system, there would be pressure drop through the entirety of the VCC, however, to simplify the simulation it is assumed that there is zero pressure drop.

$$p_2 = p_3$$

$$p_1 = p_4$$

Additionally, it is assumed that the mass flow rate will be .0024kg/s. This number is based on solving the system one variable at a time where the state points were pre-determined, doing so gives an idea of what the mass flow rate will be when the system is not constrained by set state points.

Results of Simulation

Once all the equations are defined, to perform the system simulation the equations were put in MATLAB. The simulation was performed using `fsolve`, as well as initial guesses to arrive at the following solution.

State 1	Pressure = 1.77Bars Temperature = -5 C h= 380 kJ/kg
State 2	Pressure = 5.00 Bars Temperature = 38 C h = 413 kJ/kg
State 3	Pressure = 5.00 Bars Temperature = 24.9 C h = 233 kJ/kg
State 4	Pressure = 1.77 Bars Temperature = -5 C h = 233 kJ/kg

Figure 9: State Points of Refrigerant

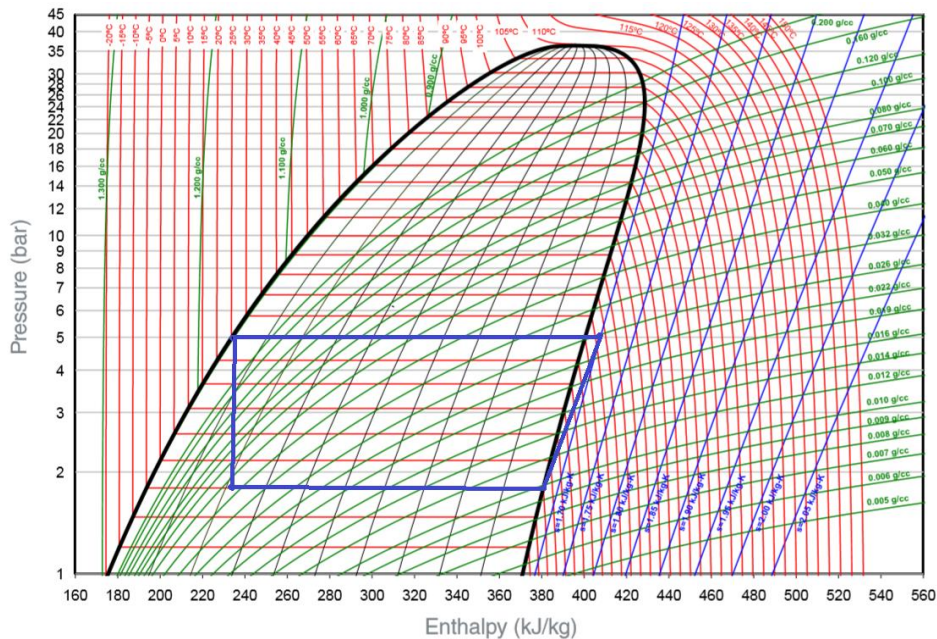


Figure 10: Pressure Vs. Enthalpy State Points

T_{sat}	9.25 C Set to 5 C
Q_{in}	.3519 kW
$\dot{m}_{\text{refrigerant}}$.0024 kg/s
UA_{evap}	$UA_{\text{evap}} = 24.1813 \text{ W/K}$
UA_{cond}	$UA_{\text{cond}} = 16.8 \text{ W/K}$
Q_{out} W_{in} $\text{COP} = Q_{\text{in}}/W_{\text{in}}$	$Q_{\text{out}} = .4322 \text{ kW}$ $W_{\text{in}} = .0785 \text{ kW}$ $\text{COP} = 4.48$

Figure 11: System Parameters

Based on these results, the simulation was successful in increasing the overall COP, which is a measure of the overall efficiency of the VCC. Additionally, the VCC can yield greater than 500cc/hr of water.

Sizing Heat Exchangers

The next step in the process of simulating the VCC is defining the size of the heat exchangers to understand how large the evaporator and the condenser must be to allow the system to function properly. To find the lengths of the heat exchangers relies on understanding the thermal circuit the heat is flowing through as well as the UA values calculated based on the simulation. A thermal circuit is a method of defining the total thermal resistance that heat has to travel through, from an area of high temperature to low temperature. It is analogous to an

electrical circuit. Were the heat acts as the current, the thermal resistance is the resistance, and the temperature difference is the voltage (“Fundamentals of Thermal Resistance”,n.a). For the evaporator, the thermal resistance involves convection through the refrigerant inside the evaporator, and conduction through the pipes of the evaporator. For both the evaporator and the condenser, 3/8th inch copper piping was utilized for the design. Utilizing an equation for the overall heat transfer coefficient and solving for the length of the evaporator produces the following equation,

$$L_1 = \frac{\frac{1}{\pi h_{evap} D_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi k_2}}{U_1 A_1}$$

For the condenser the equation is similar, however since the outside temperature of the condenser is not assumed, the thermal circuit has to also include the convective resistance of the water in the ice bath. As a result, the equation is,

$$L_2 = \frac{\frac{1}{\pi h_{cond} D_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi k_2} + \frac{1}{500\pi i D_o}}{U_2 A_2}$$

The two variables within these equations that must be solved for are h_{cond} and h_{evap} . Normally, these values can be solved for using a Nusselt number correlation and then solving for the convection coefficient, however, in this case, the working fluid is in two phases within the saturation dome, where heat transfer is taking place, which must be accounted for when solving for the convection coefficients. To account for this, the equations utilized were,

Evaporator

$$Nu = (1.3 + 39D_i) \left[Pr_l^{0.9} Re_l^{0.23} Re_v^{0.34} \left(\frac{\rho_l}{\rho_v} \right)^{.25} \left(\frac{\mu_v}{\mu_l} \right) \right]$$

(Coulson McNelly correlation)

Where Re_l and Re_v equal,

$$Re_{v/l} = 4\dot{m}/\pi D_i \mu_{v/l}$$

And finally, the convection coefficient equals,

$$h_{evap} = Nuk/D_i$$

Condenser

$$h = (k/D_i).05Re^{-8}Pr^{.33}$$

where Re equals,

$$Re = Re_v(\rho_l/\rho_v)^{.5}(\mu_v/\mu_l) + Re_l$$

Based on the equations the final results for the lengths of the condenser and evaporator are,

Evaporator	$h_{ref} = 1,099 \text{ W/m}^2\text{K}$ $L = .69 \text{ m}$
Condenser	$h_{ref} = 594 \text{ W/m}^2\text{K}$ $L = 1.903 \text{ m}$

Figure 12: Length of Heat Exchangers

Parametric Study

The final process of this project was to perform a parametric study on the different variables within the system to understand how different parameters will affect the coefficient of performance. The three parameters that were varied were the UA value of the evaporator, the mass flow rate, and the amount of water yielded by the evaporator.

UA of Evaporator

The different values for the UA of the evaporator that were used in this study are shown in the table below.

UA ₁ (W/K)	COP
6	1.25
12	2.59
24	4.48
48	5.096
96	7.2

Figure 13: UA Values

Based on these values and results the following graph was produced.

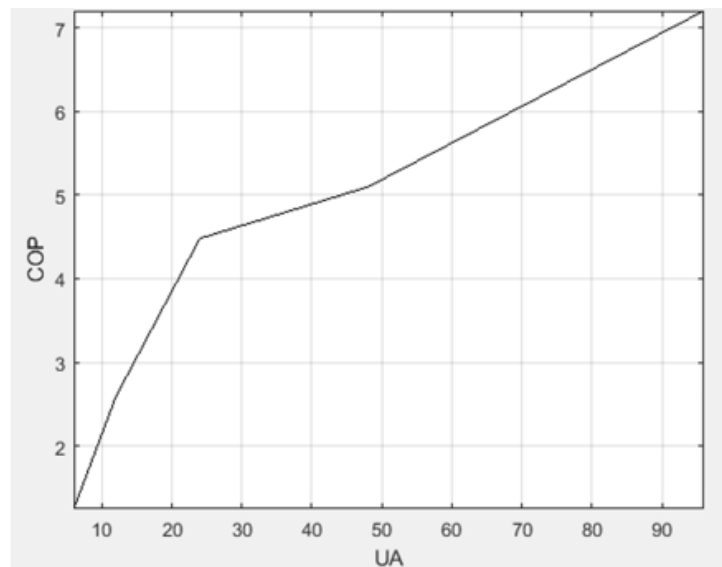


Figure 14: COP vs. UA

As a result of increasing the UA value of the evaporator, the COP will increase. The COP will increase based on the relation of UA to Q_{in} , which can be seen in equation 1. As UA increases, Q_{in} will increase because they are directly proportional to each other. Since COP is defined by the equation $COP = \frac{Q_{in}}{W_{in}}$, increasing the Q_{in} will increase COP. The UA value can be increased a number of ways. The best way to increase this value would be to use enhanced surface areas for the piping of the evaporator. Increasing the surface area leads to more space for heat transfer to take place. Another way to increase the UA value would be to increase the convection coefficient for the refrigerant, this could be done by decreasing the diameter of the pipes, or increasing the mass flow rate, which would lead to a greater Reynold's number, and thus a higher convection coefficient.

Water Yielded

The next parameter that was varied was the amount of water being produced by the evaporator. The numbers that were used are in the table below,

$Q_{in_1}(kW)$	COP
.25	.73
.28	1.2
.3	1.64
.35	4.48
.38	8

Figure 15: Q_{in} Values

Q_{in} was the variable that was changed to represent the amount of water that was produced because the more heat that is being transferred the more volume of liquid is being produced. The two variables are directly proportional to each other. As a result, the graph is,

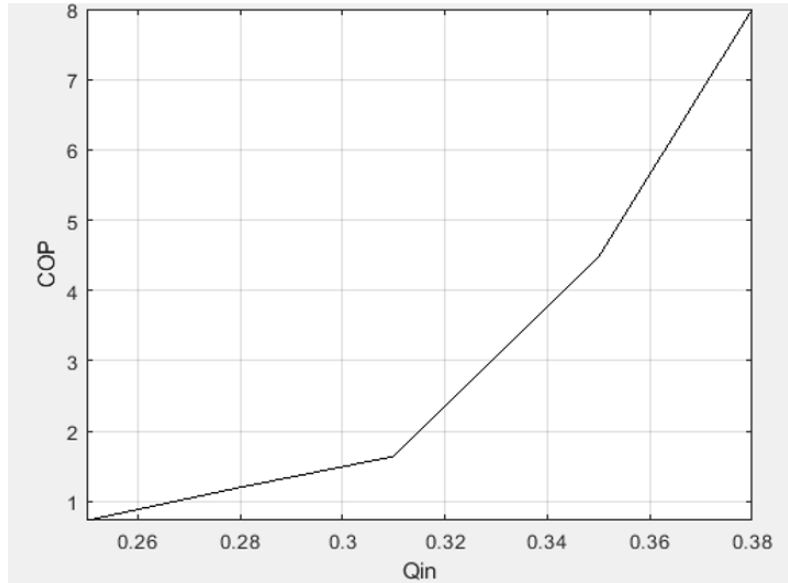


Figure 16: COP Vs. Qin

The relationship between the water produced and the COP is, as the Q_{in} increases the COP increases. This is consistent with the equation $COP = \frac{Q_{in}}{W_{in}}$, because Q_{in} as Q_{in} increases so too will the COP.

Mass Flow Rate

The final parameter that was varied was the mass flow rate of the system. The numbers that we used for the mass flow rate are as follows,

\dot{m} (kg/s)	COP
.0008	6.188
.0012	5.4
.0024	4.8
.0048	4.5
.0072	2.64
.0096	2.6

Figure 17: Mass Flow Rate Values

The graph that was produced based on these values is,

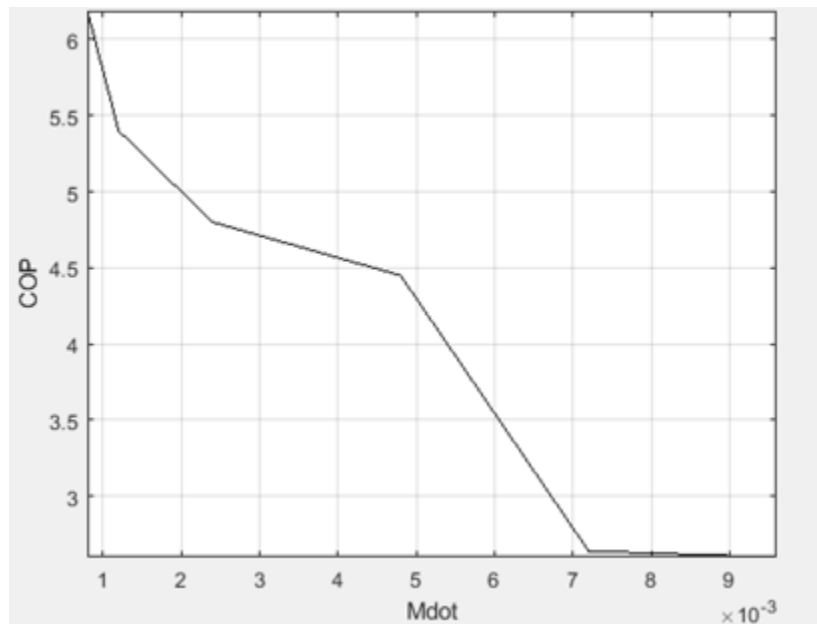


Figure 18: COP Vs. Mass Flow Rate

The relationship that exists between the mass flow rate and the COP is not as intuitive as the other variables changed in this study. The relationship between them relies on equation 1 and equation 3, which are the equation for the Q_{in} and the W_{in} . The mass flowrate is present in both these equations; the COP will change based on these values increasing or decreasing as the mass flow rate is varied. If the W_{in} is more sensitive to this change the COP will decrease. In this case, W_{in} is increasing more than the Q_{in} as a result of changing the mass flow rate.

Conclusion

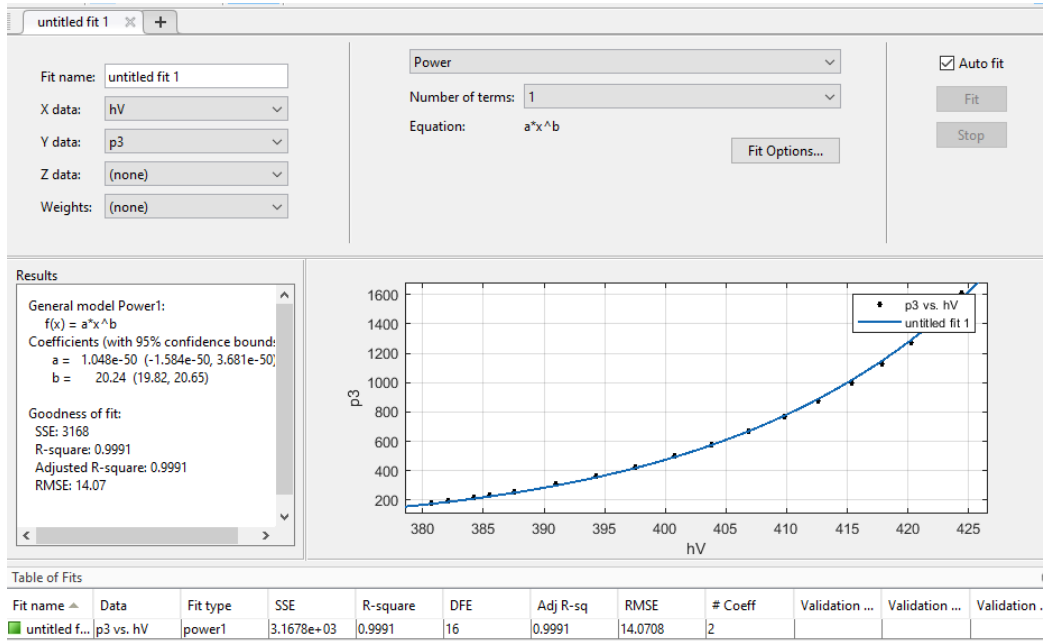
Based on the results of the simulation, the goal of increasing the COP was achieved. The COP achieved by the 2019 MQP group was 3.01, and in this study, it was increased to 4.48. Additionally, the parametric study was successful in demonstrating how different parameters could affect the coefficient of performance. To improve the coefficient of performance, enhanced tubing could be utilized as well as forced convection. Additionally, the system could be optimized using the Lagrange multipliers.

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Appendix A

Appendix A is the curve fit tool used to create property correlations



Appendix B

Appendix B shows the data used to produce curve fit.

```

p3 =
[179.423384, 193.628279, 216.550035, 232.957276, 259.344331, 308.3569, 364.159272, 4
27.343404, 498.521486, 578.326097, 667.410798, 766.451234, 876.146786, 997.222865, 1
130.434039, 1276.568444, 1436.454115, 1610.968179]
hL=
[193.4, 196.05, 200, 202.65, 206.63, 213.32, 220.08, 226.9, 233.8, 240.78, 247.84, 255, 2
62.27, 269.64, 277.14, 284.78, 292.58, 300.56]
hV =
[380.73, 382.11, 384.18, 385.55, 387.59, 390.96, 394.27, 397.53, 400.72, 403.83, 406.87
, 409.8, 412.63, 415.33, 417.88, 420.26, 422.45, 424.4]

h =
[380, 390, 400, 420, 440, 430, 410, 430, 390, 380, 420, 418, 420]
temp1 =
[-5, 5, 20, 35, 65, 50, 30, 70, 20, -10, 55, 80, 40]
pressure =
[2, 1.7, 3, 3, 7.5, 4.5, 4.3, 10, 3.7, 1.4, 10, 10.5, 2.7]
t2 =
[60, 45, 30, 40, 5, -5, -5, -5, 10, 13, 55, 70, 75, 40, 10, 25, 20, 55, 70, 73]
h2 =
[432.57, 423.64, 410.52, 419.27, 388.99, 381.22, 382.2, 381.71, 391.62, 394.58, 428.37,
441.2, 442.53, 411.28, 391.39, 401.81, 399.04, 431.61, 444.86, 450.49]
  
```

```
p2 =  
[650,300,250,280,200,160,120,140,280,270,600,750,1000,700,290,450,360,400,500  
,300]  
t3 = [-5,-3,0,2,5,10,15,20,25,30,35,40,45,50,55,60,65,70]
```

Appendix C

Appendix C is the matlab code used to produce the simulation

```
function F =root91(x)  
  
clc;  
options=optimset('MaxFunEvals',100000);  
  
% Define variables  
%t4 = input(' = ');% tempature in degrees celsius  
%t1 = t4; %heat transfer coefficient  
%U = input(' = ');  
tair = 20;  
tsat = 9.25;  
tamb = 5;  
%p1 = 178;  
% p2 = 780 ;  
%p4 = 178;  
t1 = -5;  
ts = 9.25;  
a2 = 17.12;  
b2 = .3358;  
c = 95.82;  
  
%t3 = 40;  
%p3 = 780;  
  
%h3 = 279;  
  
pz = -2.769e-05 ;  
px = 0.09852;  
py = -19.12;  
a1 =293;  
b1 = 0.05041;  
gx = 0.8154;  
gz = 1.955;  
gy = 23.47;  
gg = 26.97 ;  
fx = 1.163;  
fy = 34.95;  
fz = 239.1;  
A = -1115.58;  
A1 = -6.78;
```

```

A2 = 4.52;
gx = 0.8154;
gz = 1.955;
gy = 23.47;
gg = 26.97 ;
fx = 1.163;
fy = 34.95;
fz = 239.1;

R = 0.08134; % kJ/kgK
k =1.4; % 5C & atm

Mw = .000139; % Mass flow rate of the condensed water (kg/s)

hfgw = 2489;
%Qin = Mw*hfgw*1000;

F(1) =((a1*x(1)^b1)-x(2)); % Curve fit for state 1 p1& h1
%F(2)= (10.^ (A./(x(3)+273.15+ A1) +A2).)*100)-x(1);%curve fit state 1 (t1 p1
F(3)=389.4 - .02426*x(4) + .8569*x(5) + .0001352*x(5)*x(4) + .0005619*x(5)^2-
x(6); %curve fit state 2 t2,h2,p2
%F(4)=(10^ (A/(x(14)+273.15+ A1) +A2)*100) - x(4); %curve fit state 3 t1,p1
F(5)=(a2*x(4)^b2+c)-x(7); %curve fit state 3 p3 h3
F(6)= (x(8)*(tsat - x(3))/1000) - x(12); % heat transfer through evap
F(7)=(x(12)) - .0024*(x(2)-x(7)); % energy change of refrig through evap
F(8)=x(9) - ((.0024*R*(x(3)+273)*(k/(k-1)))/.8)*((x(4)/x(1))^( (k-1)/k)-1);%
Compressor

F(9)=(x(10)*(x(13))/1000)- x(11); %energy transfer through condenser
F(10)=(.0024*(x(6) - x(7))) -x(11);% energy change of refrigerant through
condenser
%F(11) =x(12) + x(9) -x(11); % energy conservation of entire system
F(11)=((x(5) - tamb) - (x(14) - tamb))/(log((x(5)-tamb)/(x(14)-tamb)))-x(13);
% log mean temp
%F(13)=x(8) - (fx*x(7)^2 + fy*x(7) + fz) ; % curve fit of t3 and h3
%F(14) = x(2)-(gx*x(3)^3 + gz*x(3)^2 + gy*x(3)+ gg); % curve fit t1 and h1
%F(15) = (x(9)/.0096 +x(2)) - x(6); % h2 out of compressor
end

% lp1 = x(1)
%h1 = x(2)
%t1 = x(3)
%p2 = (x(4)
%t2 = x(5)
%h2 = x(6)

```

```

%h3 = x(7)
%U= x(8)
%Win = x(9)
%Mr = .0096
%U2 = x(10)
%Qout = x(11)
%QIN = x(12)

%f(1) a1*p1^b1-h1 ==0 ;
%f(2) (10.^ (A./(t1+273.15+ A1) +A2).*100)-p1==0;
%f(3) 389.4 - .02426*p2 + .8569*t2 + .0001352*t2*p2 + .0005619*t2^2-h2==0;
    %st 3
%f(4) (10^ (A/(t3+273.15+ A1) +A2)*100) - p2 == 0
%f(5) (a2*p2^b2+c)-h3 ==0;
%f(6) (U*(tsat - t1)) - Qin == 0; % Heat into evaporator from air to
refrigerant (kJ/s)
%f(7) Qin/1000 - Mr*(h1-h4) == 0
%f(8) Win - ((Mr*R*(t1+273)*(k/(k-1)))/.8)*((p2/p1)^((k-1)/k)-1) ==0;
%f(9) (U2*(tdel)/1000)- Qout == 0;
%f(10) (Mr*(h2 - h3)) - Qout== 0;
%f(11) x(12) + Win -Qout == 0
%f(12) ((t2 - tamb) - (t3 - tamb))/(log((t2-tamb)/(t3-tamb)))-tdel==0;
%f(15) h2 - ((Win/Mr) + h1) == 0

```

Executable code:

```

fun = @root92;

x0 = [178 380 -5 500 40 413 235 24 .075 13.4 .42 .345 24 25 ];
x = fsolve(fun,x0,optimset('MaxFunEvals', 5000,'Maxiter',5000,'FunValCheck',
'on', 'PlotFcns', @optimplotfval))
% lp1 = x(1)
%h1 = x(2)
%t1 = x(3)
%p2 = x(4)
%t2 = x(5)
%h2 = x(6)

%h3 = x(7)
%U= x(8)
%Win = x(9)
%Mr = .0024
%U2 = x(10)
%Qout = x(11)
%QIN = x(12)
% og guesses foubd in other rootfile

```