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MATLAB System Simulation: Solar-Dehumidification for low Volume Water Production in Remote Regions

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By

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Abstract

Modern techniques for water desalination focus on large scale industrial plants. These plants often require a large amount of resources in capital and location for efficient operation. As a result, the worldwide toll on human deaths from poor access to clean water was estimated above 1.7 million. To provide a low cost and resource system for water production, this study aims to test the feasibility of a portable solar dehumidification system to produce enough daily drinking water for four adults (about 4 gallons). The system concept design consists of using a heat chamber (solar still) and polluted water to heat and increases the humidity of the air. A refrigeration cycle via a dehumidifier would then capture the water output from the humid air by condensing on the evaporator coil. A simulation study was conducted on the proposed idea through MATLAB and using 'RefPropMini' as a thermal database for fluid properties. Results of the study showed that the proposed system is technologically feasible and able to produce 3.6 gallons of water a day. The daily operation time of the system is within the 12 hours of solar activity on Earth. Additionally, the use a heat chamber greatly improves the energy factor of the refrigeration cycle to 1.5 L/kWh (0.40 gal/kWh) – a value less by 25% to average commercial dehumidifiers. The energy factor was matched to commercial products in a similar dimensional range for comparison. Although this study underlines good initial results for the system feasibility, there are still many more improvements that could be made to the model to better represent real world conditions and experimental designs that would validate the system.

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Variables Nomenclature

A = area	$[ft^2, in^2]$
C = heat capacity	$\left[\frac{Btu}{s \cdot °F}\right]$
COP = Coefficient of performance	
Cp = specific heat at constant pressure	$\left[\frac{Btu}{lbm \cdot R}\right]$
d = diameter	$[in]$
f = frequency	$\left[Hz, \frac{1}{s}\right]$
hc = convective heat transfer rate	$\left[\frac{Btu}{hr \cdot ft^2 \cdot R}\right]$
h = enthalpy	$\left[\frac{Btu}{lbm}\right]$
H = height	$[ft, in]$
I = solar radiation heat flux	$\left[\frac{Btu}{hr \cdot ft^2}, \frac{W}{m^2}\right]$
k = thermal conductivity	$\left[\frac{Btu}{hr \cdot ft \cdot R}\right]$
L = length	$[ft, in]$
m = mass	$[lbm]$
$\dot{m}$ = mass rate	$\left[\frac{lbm}{s}\right]$
N = number of items	
NTU = Number of transfer units	
Nu = Nusselt number	
Pr = Prandlt number	
P = Pressure	$\left[\frac{lbf}{in^2}\right]$
q = heat energy flux	$\left[\frac{Btu}{ft^2}\right]$
$\dot{Q}_\iota$ = heat energy rate	$\left[\frac{Btu}{hr}\right]$
Re = Reynolds number of fluid flow	
R= thermal resistance	$\left[\frac{hr \cdot R}{Btu}\right]$
s = entropy	$\left[\frac{Btu}{lbm}\right]$

t = time | $[s, hr]$
T = Temperature | [°F, R]
U = Overall heat transfer coefficient | $\left[\frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}\right]$
$\vec{v}$ = velocity | $\left[\frac{ft}{s}\right]$
V = Volume | $[ft^3]$
$\dot{V}$ = Volumetric flow rate | $\left[\frac{ft^3}{s}\right]$
w = humidity ratio | $\left[\frac{lbm\ of\ water}{lbm\ of\ air}\right]$
W = width | $[ft, in]$
$\dot{W}$ = work energy rate of the compressor | $\left[\frac{Btu}{h}, W\right]$
x = quality ratio of fluid liquid to vapor | [%]

Symbols Nomenclature

α = coefficient
η = efficiency of system | [%]
μ = dynamic viscosity | $\left[\frac{lbm}{ft \cdot s}\right]$
ν = specific volume | $\left[\frac{ft^3}{lbm}\right]$
ρ = density | $\left[\frac{lbm}{ft^3}\right]$
ϕ = relative humidity | [%]
ω_{rad} = angular speed | $\left[\frac{rad}{s}\right]$

Subscripts Nomenclature

A_S = surface area

A_C = cross sectional area

C_{min} = minimum heat capacity of heat exchanger

C_{max} = maximum heat capacity of heat exchanger

C_r = heat capacity ratio of heat exchanger

h_1 = properties state at entry to the compressor / exit of evaporator

h_2 = properties state at entry to the condenser / exit of compressor

h_3 = properties state at entry to the expansion valve / exit of condenser

h_4 = properties state at entry to the evaporator / exit of expansion valve

I_{pk} = peak solar radiation

l_{char} = characteristic length

R_{foul} = fouling factor thermal resistance

$R_{c,ref}$ = refrigerant fluid thermal resistance

R_{wall} = heat exchanger wall thermal resistance

$R_{c,air}$ = air fluid thermal resistance

X_i = initial properties (T,P,h,s,v,x) state of fluid into a system component

X_e = exit properties (T,P,h,s,v,x) state of fluid out of a system component

α_{ql} = coefficient of energy loss from the heat chamber

η_f = single fin efficiency

η_o = overall fin efficiency

1. Introduction

Although developed nations are normally blessed with technological advancements to provide all the basic resources to its population, this is often not true for developing worlds. In most developing nations, access to clean drinkable water is a severe issue. This can often be attributed to constant conflict/wars and limited access to education which leads to constant pollution of water bodies by the local residents through waste disposal (Ashbolt, 2004). However, these same bodies are also then used for water consumption. The end result is the ever present issue of disease and bacteria thriving in the water supply. A study by the School of Civil and Environmental Engineering in South Wales, noted that about 1.7 million deaths a year can be easily attributed to poor water and sanitation quality worldwide (Ashbolt, 2004). The natural habitat for most microbial pathogens is within drinking water for developing worlds. And given the lack of resources to treat the water supply, gastro-intestinal diseases are often more severe (Ashbolt, 2004).

Current technological innovations in developed nations have allowed for the production of several devices that can be repurposed to address the lack of clean water in developing nations. One such device is the dehumidifier. Dehumidification devices are common place and widely used to remove water moisture from an environment. The basis for this research is that a hidden application of dehumidifiers would be to produce clean drinkable water for developing remote regions. The renewable energy, solar radiation, would be working with a dehumidifier system. The hypothesis was to utilize the solar radiation to interact with air in a heat chamber (solar still) to heat up the volume.

Black/contaminated water could be added to the chamber to increase the relative humidity of the air in the chamber when evaporated. The new air mixture is then drawn through the dehumidifier (refrigeration cycle) to produce water. Polluted or contaminated water could be added to the heat chamber to increase the humidity and correspondingly the water yield. This adds to the efficiency of the system and the water filtration/treatment aspect. Utilizing the underlying thermodynamic theory, a MATLAB mathematical model was generated to simulate the thermal cycle. A successful result of the study would be to generate a functional software program that would represent the hypothesized system and show a simulation water yield sufficient for a small family (four human adults) under defined assumptions.

2. Literature Review

A primary conflict that faces the world today is how to determine low energy methods of water production. This section will discuss the key background concepts of thermodynamics and the water crisis. Other established studies and experiments are presented to demonstrate what has been done and how this study will differ from published works.

2.1.Thermal Cycles

Thermal cycles encompass a large range of applications in the modern era all the way from jet and space craft engines to refrigerators and computers. Even an everyday process of water condensing on a car or window during a cold morning can be explained from an understanding of thermal concepts. To describe the various systems, thermal engineering can be divided into three subjects: thermodynamics, heat transfer, and fluid mechanics. Understanding the concepts of each of these subjects is key to the overall theory and application of a thermal system.

The second law of thermodynamics and **entropy (**that characterizes the amount of energy needed to overcome the irreversible process and construct a reversible path) led to many breakthroughs for engine technology in the 1800s (Kaminski & Jensen, 2011). One of these breakthroughs brought about the creation Vapor Compression Heat or Refrigeration (VCR) cycle. The dehumidification system is a correspondent of the VCR cycle. A review of the overall system will be presented in the next section however; the VCR can be defined as an energy consuming cycle that transfers thermal energy from a low temperature space (i.e. inside a fridge) to a high temperature space (Kaminski & Jensen, 2011) using external power. A refrigeration cycle consists of four components; compressor, condenser, expansion or throttling valve, and an evaporator. The theory behind each component is presented in Section 5.4.

2.2.Psychometrics

The development to characterize the properties of the air in the atmosphere based on certain parameters gave rise to the fundamental concepts of thermal air systems. These systems are utilized for Heating, Ventilation, Air-Conditioning, and Refrigeration (HVAC) designs. As stated earlier, the dehumidifier and humidifier systems also use the refrigeration cycle (ASHRAE, 2009). The psychometric chart is a thermodynamic engineering tool developed to correlate some thermal and physical properties of an air mixture at atmospheric pressure. The standard properties shown are humidity ratio, wet bulb temperature, enthalpy, dry bulb temperature, saturation curve, and pressure (ASHRAE, 2009). Air by its very nature is a mixture of fluids/gasses air (nitrogen, oxygen, hydrogen) and water vapor. Such a mixture is classified as wet air. When the water vapor is removed, the air is labeled as dry air. The simple interaction of water droplets condensing around a glass of cold water is defined by the interaction of wet air being converted into dry air. In modern society, devices such as air conditioners, dehumidifiers, and humidifiers are used to convert wet air into dry air or vice versa as they follow the relationships detailed by psychometrics (Kaminski & Jensen, 2016. Ch15). To explain the tool, some further terms must be defined:

Humidity is a term that describes the amount of water vapor in the atmosphere compared to the amount of dry air (Kaminski & Jensen, 2016. Ch15).

Relative humidity is formulated as the actual mass of moisture compared to the maximum amount of moisture the air can hold (or the air-water combined mass) (Kaminski & Jensen, 2016. Ch15).

Enthalpy is a thermal property of a substance that that defines the internal energy and work energy. It is utilized in the analysis of open thermal systems where a system is considered open when mass of a fluid or object is able to move in and out of the system bounds (Kaminski & Jensen, 2011).

2.3.The Water Problem

Water and diseases are two entities that have been together since the dawn of human civilization. Pathogens are disease causing organisms that are primarily transmitted through drinking water. It is also known that the origin of most pathogens is from fecal matter. Although water treatment techniques can be dated as far back as ancient Greece 6000 years ago, modern classification and treatment systems began around the early 1800s after a massive outbreak an of cholera in Germany. It was found that one town had severely lower mortality rates due to the use of sand filtration of drinking water (Ashbolt, 2004). Several studies were then commissioned by the government that led to the revision in 1845 to utilize chorine disinfection to treat the pathogens. The success of this treatment method for pipe water caused the system to become the norm and was implemented in developing nations across Europe and the Americas (Ashbolt, 2004).

The World Health Organization (WHO), founded to monitor and record the effective health of the planet, published a report in 2003 that concluded that about 1.1 billion people drank unsafe water which is the main source of diarrheal diseases (Ashbolt, 2004). Additionally, approximately 1.7 million deaths worldwide could also be attributed to the drinking of unsafe water (Ashbolt, 2004). Many water borne diseases cause severe discomfort to the patient and if not treated quickly will lead to death. However, these diseases tend to ran rampart in underdeveloped regions as the inhabitants lack the education on the subject needed to combat the diseases (Ashbolt, 2004).

Based on the World Health Organization (WHO) studies of epidemiological studies and classification, a set of guidelines were developed for the treatment and recycle of water for public use. This is largely because waste water is often recycled for many practical uses including industrial purposes, crop irrigation, and recreational water (Gerba, 2003). Current guidelines for water treatment are based on the risk level that the contaminated water could be exposed to a population and create an epidemic disease outbreak. This means that the treatment guideline must take into account the local epidemiological, sociocultural, and environmental factors (Gerba, 2003).

Today there are many methodologies utilized for water treatment of desalination. The most common (68%) method for industrial plant sized water treatment facilities seems to be a form of reverse osmosis (RO) systems (Al-Karaghouli & Kazmerski, 2013). However, that seems to be quickly changing with the introduction of more hybrid treatment systems that combine renewable energy sources and water treatment techniques to reduce energy cost and respectively reduce water cost (Ghaffour et al., 2012). Additionally, the ability of evaporative water desalination techniques to kill of pathogens in the process is a well defined benefit. It is still possible to include additionally filtration units to ensure pure water outputs (Al-Karaghouli & Kazmerski, 2013). This means that the filtration ability of solar dehumidification can be relied upon for this study.

Solar energy is the key source of energy for hybrid systems. Solar energy refers to the direct conversion of electromagnetic radiation (light) into electrical energy for consumption (Iles, 2001). The building block of solar energy is dependent on photo cells made of semi-conductors that absorb the photons and release a small electrical current from excited electrons in the cell. The efficiency of the cell to absorb, convert, and send out electrical currents is largely dependent on the material selection of the base semiconductor (Iles, 2001). Current solar cell energy conversion efficiencies hover between 12% and 25% with an average of 17%. Solar cells are in the ever increasing state of improvements: However, this is an iterative process based heavily on the material science of semiconductor physics and chemistry (Iles, 2001).

The low energy requirement for a portable water desalination device means that renewable energy is extremely vital for the device. The use of solar energy would greatly enhance the thermal cycle. A 2013 study noted that the use of Vapor compression water treatment plants is normally reserved for the medium to low level water production

(Karaghouli & Kazmerski, 2013). This is the primary reason a dehumidifier system was selected for this study.

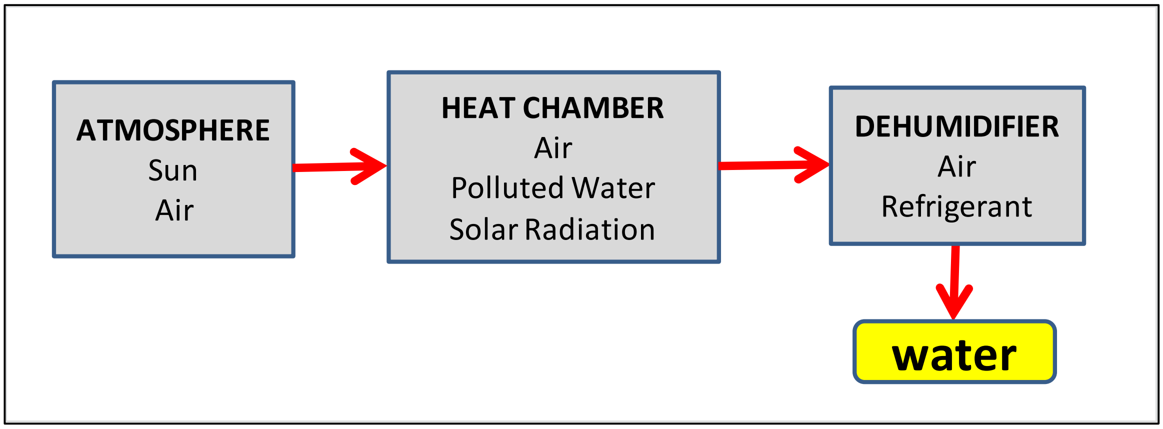
2.4.Field Studies

A collaborative study conducted by researchers in China and the USA presented the use of a solar powered humidification-dehumidification (HDH) process for water treatment (Wang et al, 2012). The system was tested under free convection flow of air and also under forced convection using solar panels. This work was more experimental in nature and did not utilize computer software to model the system: However, the researchers did utilize the fundamental theory of the thermal cycle to devise the experiment. The researchers began the study with a large focus on the holistic system level. A flowchart block diagram was utilized to plan the various components and the interaction of each system. The experiment showed that a HDH system could be used as a desalination of brackish/salted water. The use of the humidification process would then produce freshwater for consumption. The study found optimal results for a water yield of $0.873\frac{kg}{m^2 day}$ or $1.01 * 10^{-5}\frac{mm}{s}\left[960\frac{\text{gal}}{\text{hr}}\right]$ under forced convection and at an inlet evaporator temperature of 64.3℃ (146°F) (Wang et al, 2012). Lessons drawn from this study indicate that even at the industrial scale, the use of a solar humidification cycle is possible for water treatment.

Another study conducted through the Federal University of Technology in Ghana provided a comparison of a single state vapor compression refrigeration system by optimizing the refrigerant fluid (Mogaji, 2015). In the past, refrigerants used where chlorofluorocarbons (CFCs) and hydro-chlorofluorocarbons (HCFCs). Although these fluids had great thermal properties such as low freezing point and chemical stability, the recent discovery and efforts to decrease global warming found the chlorine in these fluids harmful to the environment (Mogaji, 2015). This study creates a mathematical model to study the benefits of more recently introduced refrigeration fluids such as R134a, R290, R600a and ammonia. The stated and maintained assumptions include; the ambient of the environmental and the final temperature of the quality chart. Similar to the intended study of this paper, the MATLAB software was used to model the VCR through mathematical formulas. The thermal properties of the refrigerants were taken from a COOLPACK saturation table (Mogaji, 2015). The study revealed that different fluids have different effects on the efficiency of the refrigeration cycle with ammonia being the best (highest cooling effect). It was noted that although ammonia was the most efficient refrigerant, when removed from the test, the results of R134a and R600 are the next in line (Mogaji, 2015). In other words, using R134a in this study is a valid and effective refrigerant fluid.

Other studies like that of Al-Karahouli and Kazmerski shine light on the economics of the multitude of desalination processes in use. This work also verified that most desalination systems are large and complex thermal systems with boilers, reheating stages, and super chilled heat exchangers today (Al-Karaghouli & Kazmerski, 2013). These systems have two main disadvantages, being; the large economic resources needed to construct, run, and maintain the facility and also that the need for the facility to be located near a body of water to help reduce the cost (Al-Karaghouli & Kazmerski, 2013).

Water desalination studies with small portable systems are more difficult to come by. The hypothesis for this study is based on a portable or compact system that combines a heat chamber (solar still) with a dehumidifier cycle to produce water from high humidity air. Black/Polluted water can be added to the chamber to increase the relative humidity of the air and further increase the water output. It is expected that the increase in humidity will greatly improve the water yield of the system and may also have other beneficial effects on the cycle efficiency. Figure 1 provides an illustration of the concept/hypothesis of the study.

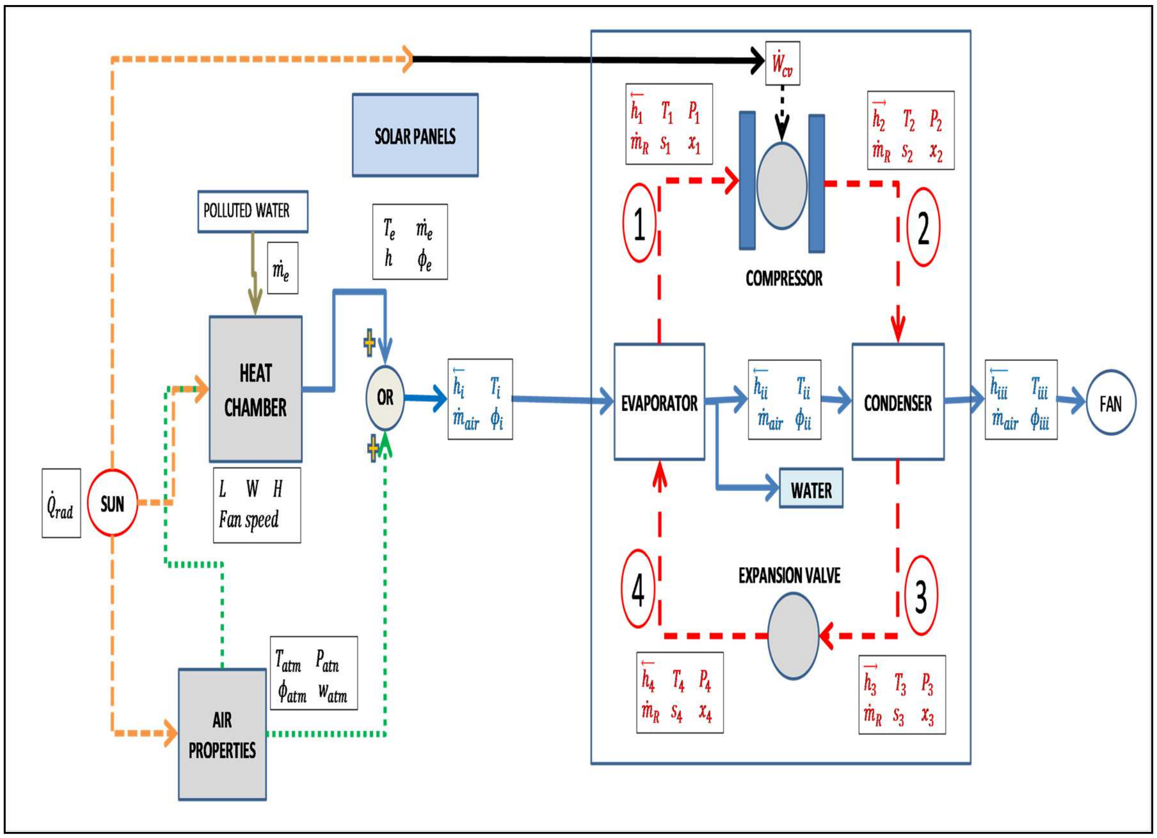


F.1 Hypothesized model: A simple depiction of the model this study will assess with a transfer of energy from the sun by radiation into the air. The heat chamber increases the air temperature and humidity thus allowing water to be extracted in the dehumidifier.

3. Methodology

Due to the conceptual nature of the hypothesis, a simulation study was planned using the MATLAB software program. The use of MATLAB as a programming tool allows for the creation of powerful research tool. For this study, the program will be used to simulate a solar dehumidification system to produce and filter low water quantities for regions with limited access to clean water. Each block or component was generated in MATLAB and then integrated to the adjacent components. Understanding of the various inputs and outputs of each component was critical to the system integration step. Programming of the MATLAB code began with the solar irradiance model and the air properties model. These models provide the backbone to the transfer system and feed into the heat-chamber and dehumidifier models. All thermal properties for the refrigerant and air mixture are pulled from the National Institute of Standards and Technology (NIST) reference database program titled 'RefpropMini' (REFPROPMINI, 2016).

The system shown in Figure 2 below can be described as the flow of air through the hypothesized model. The air interacts with solar radiation from the sun and increases in temperature. The use of polluted water filtered into the model increases the humidity and the total wet air is cooled in the dehumidifier by the evaporator. The dehumidifier runs the refrigeration cycle that aims to extract the water from the air.



F.2 System Model Transfer Diagram: Detailed breakdown of hypothesized model. The transfer diagram shows how the inputs and outputs of each component interact allowing for the critical parameters to be isolated and assessed.

4. System Assumptions

This section aims to list out the general assumptions in the model. These assumptions are justified for a conceptual simulation program but may differ in an experimental design or by nature during a physical test. The assumptions of the underlying model are defined and listed as follows:

- Quasi-Steady state system: Stable and time independent atmospheric variables of solar irradiance, atmospheric temperature, atmospheric relative humidity, and atmospheric pressure. This assumption will allow for the model to be developed by focusing on key parameters that affect the water yield with little concern of external influences. Some of the variables can be fit to a curve to simulate time variance.
- Concept shapes: the heat chamber and water tank of polluted water are conceptualized to have a rectangular and cylindrical shape. This assumption helps guide the modeling for simple physical geometries. Additionally, the heat exchangers of the evaporator/condenser are square in shape with a shell and tube physical model.
- Very low flow rate of polluted water into the heat chamber: this will allow for the assumption that all water into the chamber is evaporated to increase the relative humidity of the air.
- VCR cycle: to make this study as accurate to real world systems, the dehumidifier model will utilize the second law of thermodynamics and employ entropy for the compressor. Additionally, the inlet conditions for the evaporator

heat exchanger will be user defined to allow for an initial condition to the iterative solution.

Other defined assumptions will be discussed within the individual models along with the assumed values used in this study. This will provide more contextual support for the model assumptions. Appendix E contains all values and units used for assumed variables in the study.

5. Component Design

The various models that make the system are presented. Each model is defined and described with the corresponding inputs and outputs. The key equations/formulas that underline the model or a sample analysis of the model are also presented.

5.1.Solar Model

T.1 Solar Model I/O design

This table shows the inputs and outputs for the solar model

Input(s)	Output(s)
Solar Radiation	Sine wave radiation curve
Atmospheric Temperature	Since wave temperature curve
Atmospheric Pressure	Sine wave relative humidity curve

This model takes in user defined parameters of solar radiation, atmospheric pressure and temperature, with outputs of time varying radiation, temperature, and relative humidity. The radiation signal can be a constant value or a more realistic Gaussian/normal distribution to represent the sun energy rise and fall from sunrise to sunset. The constant source was used as a simplified version to test and develop the code while the Gaussian curve will be utilized for data collection. It is possible to get solar radiation data for a specific geo-location based on the latitude, longitude, angle of plane, and other factors (TYM3). However that is not the primary focus of this study. Thus the irradiance and atmospheric parameters are fixed in this model but could easily be adjusted to represent more dynamic simulation models. The set parameters can be used to generalize a clear sky on a sunny day.

A sine plot can be used to create a normally distributed curve that is centered about the peak radiation. If the time is stated in hours, the frequency of the irradiance should be such that the energy flow begins and ends on a 12 hour cycle (sunrise to sunset). The following formula was used to fit the solar irradiance to a time cycle:

$$I_r = I_{pk} \cdot \sin \omega t \tag{1}$$

Where I_{pk} is the peak solar radiation value in the day that is set at 1000 $\frac{W}{m^2}$ for this study. Assessing the time domain gives:

$$I_r = \ I_{pk} * \sin(2\pi f \cdot t) \tag{2}$$

$$I_r = I_{pk} * \sin\left([2\pi t] \cdot \frac{1}{T}\right) \tag{3}$$

$$I_r = I_{pk} * \sin\left(\frac{1}{24} \cdot (2\pi \cdot t)\right) \tag{4}$$

The (1/24) factor represented the frequency of a 24 hour time period for an average Earth day. The solar radiance begins at 0 on $t = 0$ and reaches maximum at $t = 6$ or midday. The temperature is also modeled to a fixed value that changes with time to mimic dynamic weather. However, unlike solar radiation, the temperature of the earth during the day began at a base value set by the user and slowly increased to a maximum in the afternoon before cooling back down. The pressure was fixed to standard atmospheric pressure of $14.7psi$ $(101kPa)$. The relative humidity of the atmosphere was also assumed at 40% and changes along with the change in temperature. The air model, discussed in the next section, was used to determine the humidity ratio and initial atmospheric enthalpy from the fixed parameters.

5.2.Air (Psychometric) Model

T.2 Air Model I/O design

This table shows the inputs and outputs for the air model

Input(s)	Output(s)
Temperature	Humidity Ratio
Relative humidity	Dry Bulb temperature
	Enthalpy
	Psychometric Plot and Air cycle Trace

This model served two primary functions. The first function was to compute the air parameters of dry bulb temperature, humidity, relative humidity, enthalpy, dew point temperature, specific volume, and wet bulb temperature. The model relies on the principles and formulas for psychometric analysis found in the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) handbook (ASHRAE, 2009). ASHRAE utilizes the pressure and saturation pressure for a given temperature to determine the humidity ratio. The saturation pressure over **ice** for the temperature range below $0°C/32°F$ is given by:

$$ln\,(p_{ws}) = C_1/T_i + C_2 + C_3T_i + C_4T_i^2 + C_5T_i^3 + C_6T_i^4 + C_7\,ln\,(T_i) \quad [5]$$

The saturation pressure over **liquid water** for the temperature range above $0°C/32°F$ is given by:

$$\ln(p_{ws}) = C_8/T_i + C_9 + C_{10}T_i + C_{11}T_i^2 + C_{12}T_i^3 + C_{13}\ln(T_i) \qquad [6]$$

T.3 ASHRAE water saturation coefficients

The coefficients for the estimation of the water saturation pressure are given below. This coefficients are for English unit calculations and will have different values in SI unit base.

Parameter	Value (English)
C1	-1.02E+04
C2	-4.89E+00
C3	-5.38E-03
C4	1.92E-07
C5	3.56E-10
C6	-9.04E-14
C7	4.16E+00
C8	-1.04E+04
C9	-1.13E+01
C10	-2.70E-02
C11	1.29E+05
C12	-2.48E-09
C13	6.55E+00

It should be noted that the coefficient values differ for liquid and solid water and also for the unit system being used (SI vs. English). Table 3 shows the coefficients for the English unit system. Additionally, the temperature must be in absolute terms (K or R).The vapor saturation pressure can then be found using the quality or relative humidity. The humidity ratio was then determined with the mass of water vapor to dry air of the mole fractions (ASHRAE, 2009).

$$p_s = X_{ws} \cdot p \qquad [7]$$

$$\phi = \frac{p_w}{p_{ws}} \quad [8]$$

$$w = \frac{m_w}{m_{da}} = 0.621945\left(\frac{X_w}{X_{da}}\right) \quad [9]$$

Through variable manipulation and the ideal gas equations for air and water vapor, the humidity ratio can be also be defined by :

$$w = 0.621945 \cdot \left(\frac{p_{ws}}{p - p_{ws}}\right) \quad [10]$$

The enthalpy of the air or water was found using the combination of liquid and vapor properties. It can also be rewritten to incorporate the specific heat, latent heat, and wet bulb temperature (T_{wb}) of the mixture.

$$h = h_{da} + w \cdot h_g \quad [11]$$

$$h = 0.240\left[\frac{Btu}{lbm \cdot °\mathrm{F}}\right] \cdot T_{wb} + w\left(1061\left[\frac{Btu}{lbm}\right] + 0.444\left[\frac{Btu}{lbm°\mathrm{F}}\right] \cdot T_i\right) \quad [12]$$

The dew point temperature (T_{dp}) was approximated by the following empirical formula with coefficient alpha (α). Again, it must be noted that the coefficients are subject to the unit system used.

$$T_{dp} = 90.12 + 26.142 \cdot \alpha + 0.8927 \cdot \alpha^2 \quad [13]$$

$$\alpha = \ln(p_w) \quad [14]$$

SITUATION 2.

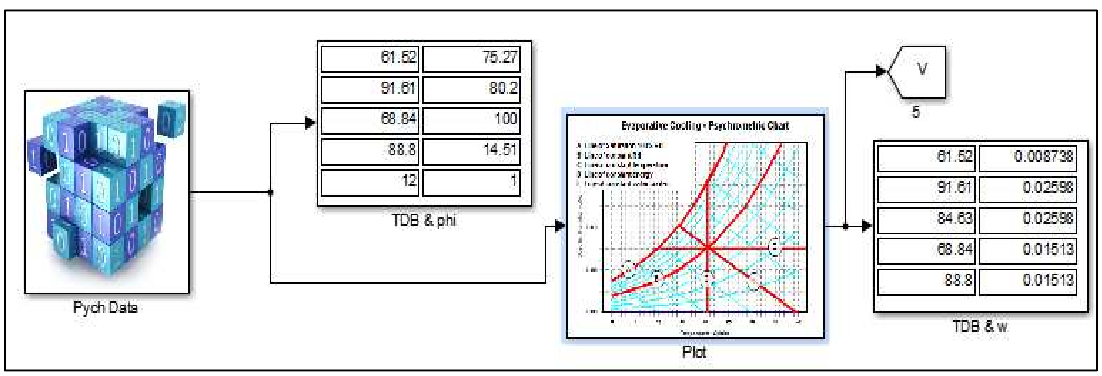
Given: Dry-bulb temperature t, Dew-point temperature t_d, Pressure p

To Obtain	Use	Comments
$p_w = p_{ws}(t_d)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t_d
W	Equation (22)	
$p_{ws}(t)$	Table 3 or Equation (5) or (6)	Sat. press. for temp. t_d
W_s	Equation (23)	Using $p_{ws}(t)$
μ	Equation (12)	Using W_s
ϕ	Equation (25)	Using $p_{ws}(t)$
v	Equation (28)	
h	Equation (32)	
t^*	Equation (23) and (35) or (37) with Table 3 or with Equation (5) or (6)	Requires trial-and-error or numerical solution method

F.3 ASHRAE Analysis Guide

The ASHRAE handbook also provided several situations based on given parameters such as dry bulb temperature and enthalpy and guides users to the equations needed to estimate the remaining air properties. These cases and situations have been modeled into the air model function to quickly find the right formulas and determine all the necessary parameters. Figure 3 provides one example to a situation from the text (ASHRAE, 2009) where dry bulb and dew point temperatures are given. The handbook list out the equations needed to determine the other psychometric parameters (ASHRAE, 2009).

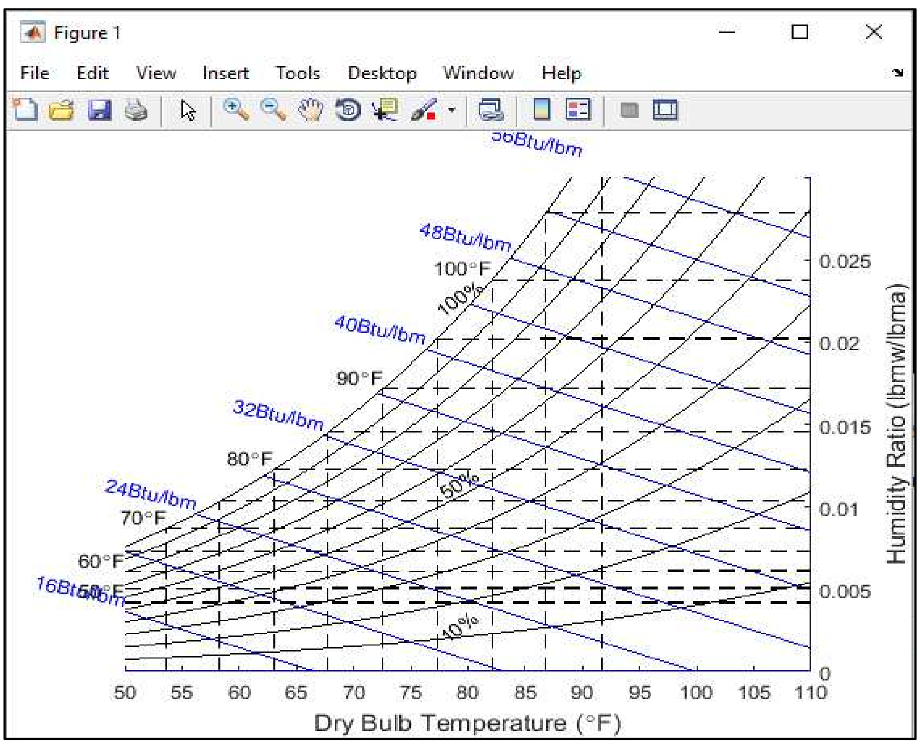
The second function of this air model was to generate a psychometric plot/trace of the air properties based on temperature and relative humidity inputs. The initial value of temperature and relative humidity represent the atmospheric conditions. Following inputs are taken from various points in the transfer system; the exit of the heat chamber, the exit of the evaporator, and the exit of the condenser. The model in Figure 4 takes in the dry bulb temperature and relative humidity and combines these values into a matrix. The MATLAB function then plots the psychometric chart to calculate the humidity ratio of the given data points and trace the air cycle path.



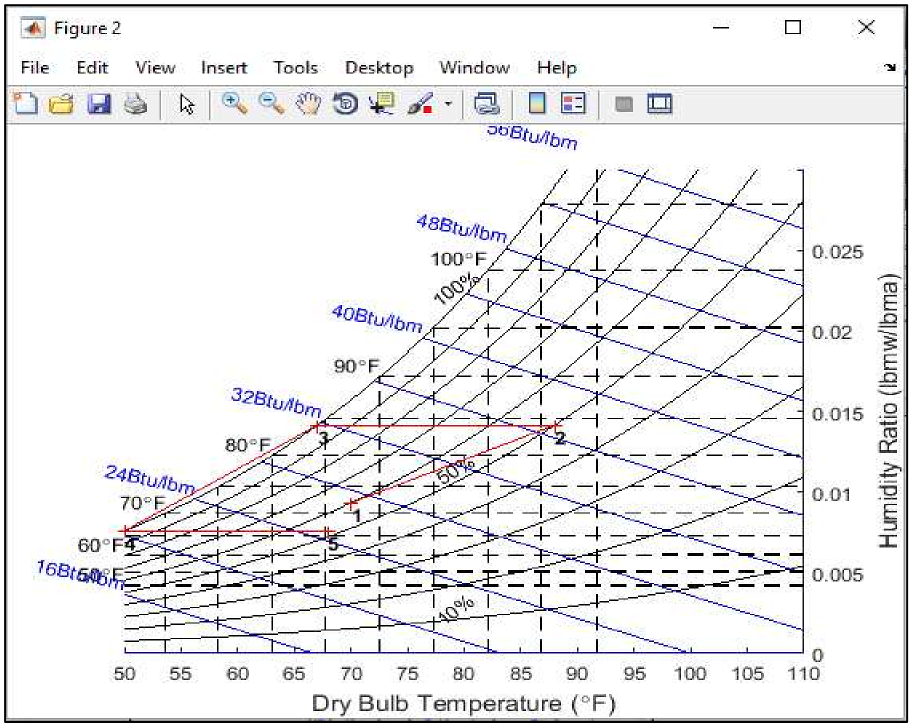
F.4 Air Model Design : shows the Simulink design of the air model pulling a matrix of temperature and relative humidity. After going through the Plot model, an output matrix of dry bulb temperature and humidity ratio is used to create a trace.

The data necessary for the trace came from various points in the system discussed earlier. It should be noted that the heat chamber and evaporator conditions constitute two data points. For the heat chamber, the first point reflects the constant increase in temperature from the initial atmospheric conditions whereas the next point shows the increase in relative humidity with the addition of evaporated polluted water. The evaporator points are similar in that the air is cooled at constant until saturation at 100% relative humidity but cooled further to a determined value by the refrigeration cycle.

Examples of the resulting plots are shown below in Figure 5 and 6. The first plot shows the psychometric chart by itself. The second plot traces the outline of the transfer path and allows for the computation of water production. The plot auto generated the axis and figure based on the given temperature and relative humidity data. The axis of the plot was set to automatically scale. This was done in order to capture the data matrix and fit the plot to the window.



F.5 Psychometric Plot : the output from the second function of the air model. This figure depicts an example psychometric plot from the air model without the air cycle trace. The plot can be used as a reference and has English base units.



F.6 Psychometric Trace : shows the output of a psychometric plot with the air cycle plot. The figure provides a perspective on the operation of the model. The change in humidity ratio between points 3 and 4 also gives a quick indication of the water yield in the captured time.

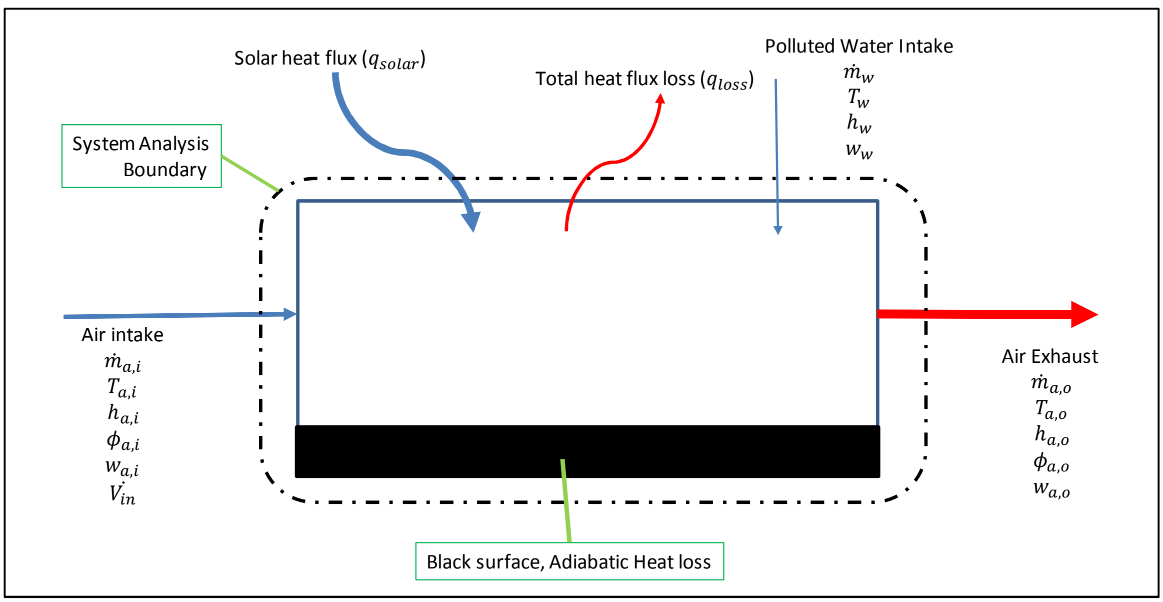
5.3.Heat Chamber Model

T.4 Heat Chamber I/O design

This table shows the general inputs and outputs for the heat chamber model

Input(s)	Output(s)
Atmospheric Temperature	Exit Air Temperature
Atmospheric Relative humidity	Exit Air Mass rate
Solar Radiation	Exit Air Enthalpy
Blackwater mass rate	Exit Air Relative Humidity
Heat Chamber dimensions	

This model represents the solar still or heat chamber that played a major role in increasing water production. The model utilizes the outputs of the sun model (irradiance and atmospheric temperature) along with a virtual rectangular prism. The rectangular box is heated by the sun to increase the temperature of the air encased. If the dimensions of the chamber have no or values then the output temperature and relative humidity should the same as the atmospheric values. The user inputs for the dimensions of the chamber are the length, width, and height units along with the fan volume rate. The dimensions used in this study were [6’ · 2’ · 0.5’] at a 200 cfm rating $(\dot{V}_{in})$. The model computes the surface area, cross-sectional area, and volume. A thermal black box analysis of the model is shown in Figure 7.



F.7 Heat Chamber Black : shows the inputs and outputs for the heat chamber. An energy balance analysis was conducted to keep track of the energy transfers and mass flows.

Using the conservation of energy equation for an open system, the thermal variables can be found.

$$[E_{solar} + E_{air,in} + E_{water} = E_{air,out} + E_{loss}] * \left(\frac{1}{t}\right) \quad [15]$$

$$(q_{solar} \cdot A_c) + \dot{m}_{a,i} \cdot h_{a,i} + \dot{m}_w \cdot h_w = \dot{m}_{a,o} \cdot h_{a,o} + (q_{loss} \cdot A_c) \quad [16]$$

The energy loss is assumed to be some smaller percent (20%) of the energy into the system:

$$q_{loss} = \alpha_{ql}(E_{solar} + E_{air,in} + E_{water}) \quad [17]$$

Other helpful equations used were the conservation of mass and relevant equations for mass rate:

$$\dot{m}_{a,i} + \dot{m}_w = \dot{m}_{a,o} \quad [18]$$

$$\dot{m} = \dot{V} * \rho \quad [19]$$

Enthalpy and specific heat are defined as a function of temperature and humidity:

$$h = f(T, P, w) = c_p T$$

Next, all the known terms could be labeled.

- The solar flux is given by the atmospheric assumptions
- The volumetric flow rate is defined and fixed
- The dimensions of the heat chamber are user defined and fixed
- The initial air enthalpy, specific volume, and relative humidity were defined by the atmospheric assumptions

The mass rate of water into the chamber can be determined with the addition of polluted water into the heat chamber through an input parameter. It is also possible to change the mass rate of water in to the chamber directly as user defined variable for a validation test. Once in the chamber, the water would be evaporated under the assumption that all the water that enters is evaporated. This is based on the very slow mass flow rate of water into the chamber. The latent heat of evaporation for water was found from the thermal ‘RefpropMini’ database. The humidity ratio of the air exiting the chamber followed a similar principle to the conservation of mass such that:

$$w_{a,o} = w_{a,i} + \frac{\dot{m}_w}{\dot{m}_{da}} \quad [20]$$

Solve Equation [6] for dry air and substitute

$$\dot{m}_{a,i} = \dot{m}_{da} + \dot{m}_{da} \cdot w_{a,i} \quad [21]$$

$$\dot{m}_{da} = \frac{\dot{m}_{a,i}}{1+w_{a,i}} \quad [22]$$

The minimum value for the temperature at the exit of the heat chamber must be high enough to vaporize liquid water so:

$$\Delta\dot{E}_{water} \rightarrow \ \dot{m} \cdot h_{fg} = \dot{m} \cdot c_p \cdot (T_{out} - T_w) \quad [23]$$

The heat flux lost from the system can be found through a thorough investigation of the heat transfer in the system. This requires specification of materials for the heat chamber such as plastic or glass for the casing and dark metal or ceramic for the base. Again, selection of construction materials for the chamber would alter this value to make the chamber more efficient at gaining heat energy. Since this is a simulation study, the heat energy lost is approximated at some percentage (10 – 20%) of the total energy into the system. With this, the temperature at the exit can be solved with the energy balance equation.

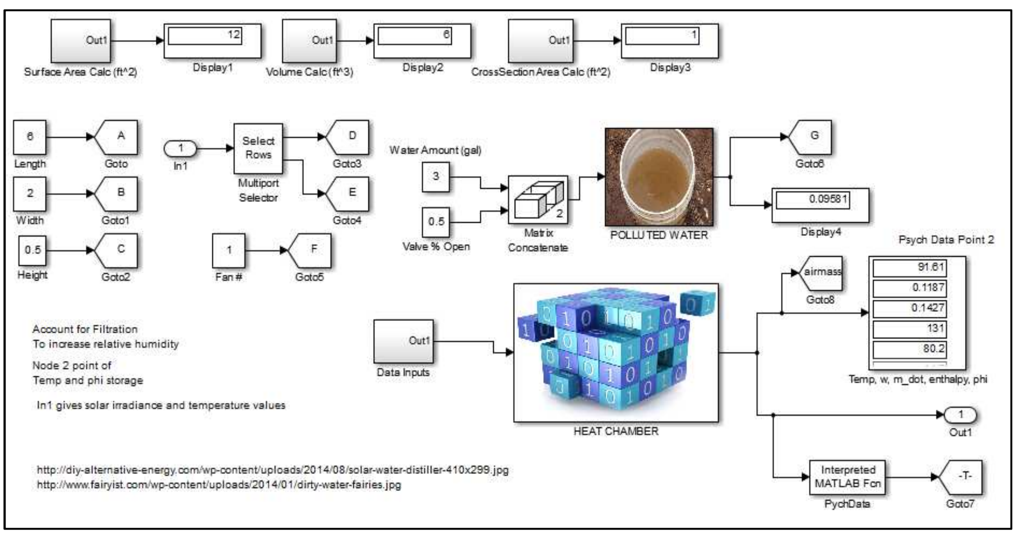
$$T = f(h, P, w, c_p) \quad [24]$$

Enthalpy is a function of temperature, pressure and humidity and is equivalent to the specific heat multiplied by the temperature. By reverse analysis, then temperature can be seen as a function of enthalpy, pressure, humidity, and specific heat. Solving the energy balance equation [16] for the exit temperature gives the following:

$$T_{a,o} = \frac{[(q_{solar} \cdot A_C) + \dot{m}_{a,i} \cdot c_{p,i}(T_{a,i}) + \dot{m}_w \cdot c_{p,w}(T_{w,i}) + \dot{m}_{a,o} \cdot c_{p,i}(T_{a,i}) - (q_{loss} \cdot A_C)]}{\dot{m}_w * c_{p,w} + \dot{m}_{a,o} * c_{p,i}} \quad [25]$$

Note that the units provide a temperature value in absolute terms. Also the time units of the heat energy flux should match that of the mass rate. The remaining properties of the exit air such as the relative humidity, wet bulb temperature, and enthalpy can be found using the temperature and humidity ratio and the air model.

Figure 8 below shows the user interface for the heat chamber model. The user inputs are grouped to the left of the interface and can be easily adjusted from simulation to simulation. The chamber dimensions and the polluted water tank opening are the primary controls for the model. This will enhance the ability for any researcher to validate the model.



F.8 Heat Chamber Design : shows the Simulink model design that depicts the user variables that can be adjusted and the output parameters.

The results of the heat chamber model are fed to the dehumidifier (refrigeration) model and also back to the air model.

5.4.Dehumidifier Model

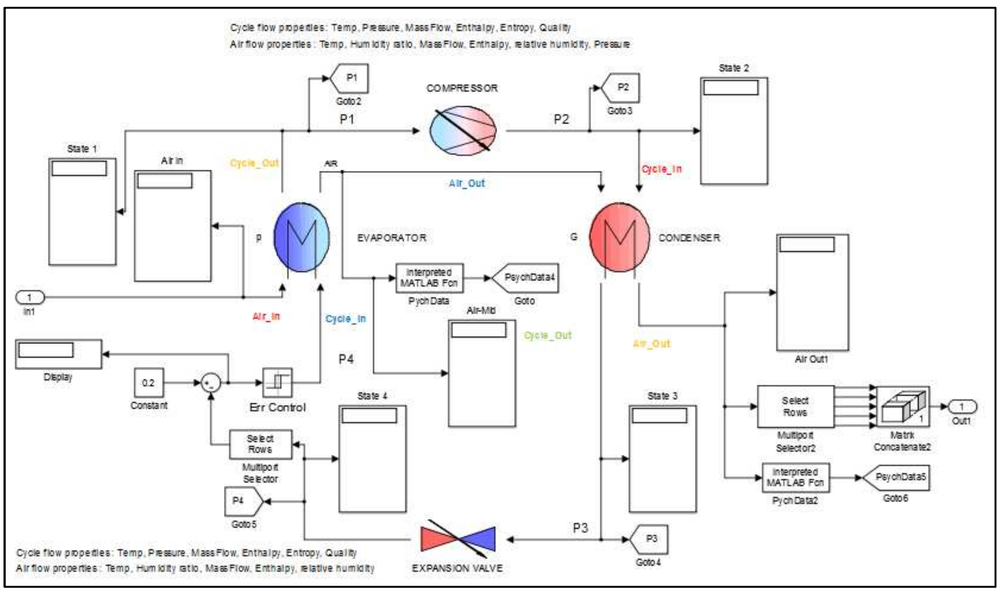
T.5 Dehumidifier I/O map

This table evaluates the general inputs and outputs for the dehumidifier system and all the components.

Component	Inputs	Outputs
Evaporator	*Refrigerant:* • Temp. • Enthalpy • mass rate • quality *Air:* • Temp. • Humidity • mass rate	*Refrigerant:* • Temp. • Enthalpy • mass rate *Air:* • Temp. • mass rate ***Water***
Compressor	*Refrigerant:* • Enthalpy • Temp. • mass rate	*Refrigerant:* • Enthalpy • Temp. • mass rate
Condenser	*Refrigerant:* • Temp. • Enthalpy • mass rate *Air:* • Temp. • mass rate	*Refrigerant:* • Temp. • Enthalpy • mass rate *Air:* • Temp. • mass rate
Expansion Valve	*Refrigerant:* • Enthalpy	*Refrigerant:* • Enthalpy • Quality

This model represents the most complex component to the transfer diagram. It incorporates the compressor, evaporator, expansion valve, and condenser component models. The system interfaces with the heat chamber and receives a data packet containing the air temperature, relative humidity, mass flow rate, and air enthalpy. The cycle runs using refrigerant R134a fluid. The 'RefpropMini' database is used to get the thermal property values based on the cycle point temperature, pressure, enthalpy, or entropy. Standard thermodynamic practice dictates that the analysis of any thermal cycle is conducted component by component and can be represented by a physical diagram and property diagram. For refrigeration cycles, a typical diagram found is the temperature vs. entropy chart.

From Figure 9, it is clear that the air properties are cycled through the evaporator and condenser. This represents the application of the dehumidifier pulling out the water from the air by cooling the air and condensing water vapor on the evaporator then recycling the cooled air to reduce the refrigerant temperature through the condenser. The other cycle flow is the refrigerant cycle through the four components. Refrigerant data packets consist of the temperature, enthalpy, mass flow rate, entropy, pressure, and specific volume properties.



F.9 Dehumidifier Design : shows Simulink design of the refrigeration cycle with inspiration from the MATLAB example model.

5.4.1. Heat Exchanger

A heat exchanger is a black box system through which heat energy is exchanged between two or more fluids. In this study, a basic two stream heat exchanger is considered. There are many types and systems for heat exchanger systems. A typical commercial refrigeration system utilizes a shell and tube design where one fluid (air or water) passes through a shell containing tubes of a refrigerant fluid. The name of a heat exchanger depends on the goal of the exchanger. An evaporator is a heat exchanger where the fluid in the tube is heated or gains heat energy from the external fluid. A condenser on the other hand allows the fluid in the tube to be cooled by giving off heat energy to the cooler external fluid. This verifies the thermal concept that heat or temperature always flows from hot to cold.

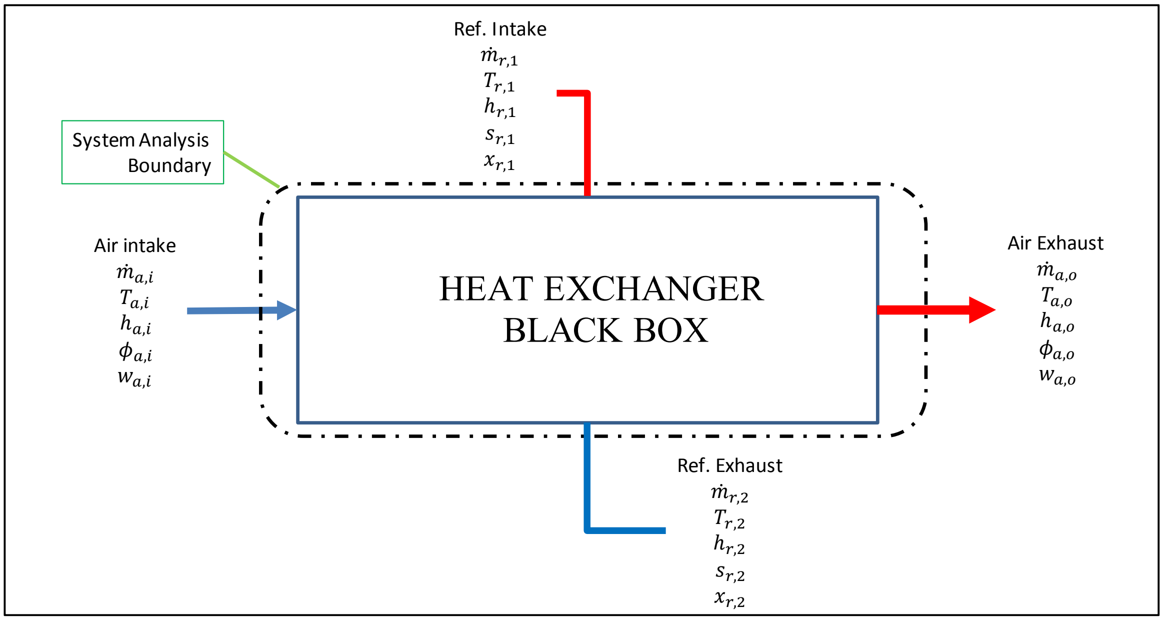
Evaporator

This component takes in the air properties from the heat chamber as well as refrigerant properties from the expansion valve. The two data packets are combined together and fed into a MATLAB function titled 'HX (heat exchanger) Function'. This function utilizes heat exchanger thermal properties to determine the output of the refrigerant and air properties. The difference between the humidity ratio of the input air stream and output air stream will be used to compute the amount of water produced in the system. The output data packet is then spilt to separate the air properties and refrigerant properties. The refrigerant data continuous the dehumidification cycle along to the compressor whereas the air packet it transferred to the condenser heat exchanger. Additionally, refrigerant properties to the evaporator inlet are given initial conditions to start the simulation iteration where: $T = 20°\text{F}, x = 20\%, \dot{m} = 0.1\,\frac{lbm}{s}$

Condenser

Similar to the evaporator model in that the same dynamics and formulas apply. This model also calls and computes the refrigeration properties using the HX function. Given that the same function is utilized for the evaporator and condenser, the code determines the current coil in use based on the state properties into and out of the component. For example, if the quality of the refrigerant into the heat exchanger is between 0 and 1.0 then the system determines that the condenser is in use. Whereas, if the quality was greater than or equal to 1.0 into the heat exchanger then the evaporator coil would be in use. This represents the cycle flow on a T-s diagram where the condenser takes the refrigerant from superheated vapor to a liquid or sub cooled fluid.

Common analysis of the heat exchanger uses either the Log Mean Temperature Difference (LMTD) method or the Effectiveness NTU method (ε). In this study, both methods are utilized to determine the effectiveness of the heat transfer and determine the exit temperatures. Both methods require the computation of the overall heat transfer coefficient for the system to determine the exit parameters (objectively the temperature). For a dehumidifier system, the first component the air interacts with is the evaporator coil. The initial air properties are given from the heat chamber model and the initial refrigerant properties are assumed by the user. Figure 10 provides an illustration of a heat exchanger analysis.



F.10 Heat Exchanger Black Box : shows the system diagram for the heat exchanger characterized by the mass flows across the boundary.

To make this model real, the physical dimensions of the exchanger are defined. A shell and tube heat exchanger is used with one shell and multiple tube passes with fins. It is assumed that the exchanger for the evaporator and condenser are of the same dimensions. The fixed parameters for the coils are the tube outer and inner diameters, the length of the shell, the number of fins on the tube, the number of tubes in the shell, and the thickness of the fin. The values used are given in Appendix E. Using the exchanger dimensions and the initial fluid properties, the heat transfer coefficients can be determined. The process to be followed here will be as shown below:

$$Re \rightarrow Pr \rightarrow Nu \rightarrow h_c \rightarrow \dot{Q} \rightarrow T$$

The Reynolds Number (Re) is based on the fluid properties and must be solved for the conduit of use. Since the fluids are given in terms of mass rate, the following derivation can be used. Note that the characteristic length is different of each fluid.

$$Re = \frac{\rho \vec{v} L_{char}}{\mu} = \frac{4 * \dot{m}_i}{\pi * \mu * L_{char}} \quad [26]$$

Internal flow is defined as laminar or turbulent based on the interaction of the fluid at a given velocity. The Prandlt (Pr) number is a thermal property of the fluid that can be found through 'RefpropMini'. The Reynolds number, Prandlt number, and conduit shape of flow (pipe versus square shell) are used to determine the Nusselt (Nu) equation. There are several characterized empirical formulas for this. The formulas selected in this study are given below however they can be adjusted to include more fluid flow characteristics that would increase data accuracy (Kaminski, 2011).

T.6 Nusselt Number Empirical Formulas

This table provides the equations used in this study to determine the Nusselt number of the fluid based on the Reynolds number.

Flow Type	Round conduit	Square conduit
Laminar	$Nu = 0.193\, Re^{0.618} P_r^{\frac{1}{3}}$ [26]	$Nu = 0.177\, Re^{0.699} P_r^{\frac{1}{3}}$ [27]
Turbulent	$Nu = 0.027 Re^{0.805} P_r^{\frac{1}{3}}$ [28]	$Nu = 0.102\, Re^{0.675} P_r^{\frac{1}{3}}$ [29]

With this information, the heat transfer coefficient of the fluid can be found be the relationship of the Nusselt Number:

$$Nu = \frac{h_{conv} * L_{char}}{k} \quad \rightarrow \quad h_{conv} = k * \frac{Nu}{L_{char}} \quad [30]$$

If fins are included, the fin efficiency is:

$$n_f = \frac{\left(\sinh(mL) + \left(\frac{h_c}{mk}\right)\cosh(mL)\right)}{\left(\cosh(mL) + \left(\frac{h_c}{mk}\right)\sinh(mL)\right)} * \left(\frac{1}{mL}\right) \quad [31]$$

$$m = \sqrt{\frac{h_c * p}{k * A}} \quad [32]$$

From this the overall fin efficiency is given by:

$$n_o = 1 - \frac{N_{fins} * A_{fins}}{A_s + N_{fins} * A_{fins}} (1 - n_f) \quad [33]$$

The overall heat transfer coefficient (U) is given by the transfer of thermal resistances between the fluids:

$$U = \left[R_{conv_{ref}} + R_{foul,i} + R_{wall} + R_{foul,o} + R_{conv_{air}} \right]^{-1} \quad [34]$$

Each resistance value is dependent on the material and fluid. The effectiveness method was then used to find the external temperatures using the heat capacity ratios and the number of transfer units.

$$NTU = \frac{UA}{C_{min}} \quad [35]$$

$$C^* = C_r = \frac{C_{min}}{C_{max}} = \frac{\left(\dot{m}_i * C_p|_{\cdot min}\right)}{\left(\dot{m}_j * C_p|_{\cdot max}\right)} \quad [36]$$

For a shell and tube design, the effectiveness of heat transfer is based on the formula:

$$\varepsilon = 2 * [(1 + C_r) + (1 + C_r^2)^{0.5} * \frac{\left(1 + \exp\left(-NTU\sqrt{1 + C_r^2}\right)\right)}{\left(1 - \exp\left(-NTU\sqrt{1 + C_r^2}\right)\right)} \quad [37]$$

The resulting temperatures are then given by:

$$T_{hot,o} = T_{hot,i} - \frac{\left(\varepsilon * C_{min} * \left(T_{hot,i} - T_{cold,i}\right)\right)}{\dot{m}_{hot} * C_{p,hot}} \quad [38]$$

$$T_{cold,o} = T_{cold,i} + \frac{\left(\varepsilon * C_{min} * \left(T_{hot,i} - T_{cold,i}\right)\right)}{\dot{m}_{cold} * C_{p,cold}} \quad [39]$$

There are other ways and other formulas that could be substituted into the model to change the characteristic for desired testing. However, it was felt that using these two techniques would provide the best approximation for a real world heat exchanger system.

5.4.2. Compressor

This component takes in the refrigerant fluid and increases the fluid pressure to create flow. The fluid properties are received from the evaporator and may be in a saturated vapor or slight superheated phase. The compressor then moves the fluid to a higher superheated vapor state. The compressor will also receive voltage and current data from the solar panels (assumed value). The compressed superheated refrigerant is then passed onto the condenser heat exchanger to continue the cycle.

A compressor normally adds pressure with the use of external work, there is little to no heat addition and the velocity of the fluid does not change nor does the potential energy. The same open system equation of thermodynamics can also be applied here. Again, it should be noted that the power supply of the compressor can be tied to the output of a solar cell array. This simplifies the equation to:

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_{in} \dot{m}_i \left(h_i + \frac{\dot{V}_i^2}{2} + gz_i\right) - \sum_{out} \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right)$$

$$0 = -\dot{W}_{cv} + \dot{m}(h_1 - h_2) \quad [41]$$

$$\dot{W}_{act} = \eta \cdot \dot{W}_{cv} \quad [42]$$

5.4.3. Expansion Valve

This model represents the simplest of all the components and acts as an isenthalpic valve that allows the refrigerant fluid to pass through with equal enthalpy on both ends. However, since the volume on the two ends of the valve differs, the temperature and pressure change across the interface. The model calls the “ExValve” function to compute the thermal properties of the component. ‘RefPropMini’ is used to determine the refrigerant properties across the device.

This device operates in steady state with very little change to kinetic and potential energy, no work added, and no heat lost. The simplest visualization would be the sudden change from a small pipe to a larger pipe. All the pressure held in the small pipe is dispersed in the large pipe however the fluid flow rate decreases.

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_{in} \dot{m}_{\iota}\left(h_{\iota} + \frac{\dot{V}_{\iota}^2}{2} + gz_{\iota}\right) - \sum_{out} \dot{m}_e\left(h_e + \frac{V_e^2}{2} + gz_e\right)$$

$$h_1 = h_2 \qquad [43]$$

5.5.Cycle Yield Model

This function aims to compute the yield characteristics and metrics of the system. These key parameters will allow for a user to assess the benefits and downfalls of the system. The primary parameters to be determined are; water yield, thermal cycle efficiency, Carnot (maximum power) cycle efficiency, and the Coefficient of Performance (COP). The water yield provides the estimated amount of water produced by the system over a unit time. The selected units of measure are gallons per hour and gallons per day. The use of gallons per day gives an estimate of daily water production based on a 12 hour daylight schedule for the solar energy.

The COP compares the heat gained in the system from the evaporator to the energy/work needed to gain that energy from the compressor. The COP gives a better insight in to the refrigeration effect of the system. COP is the more common and preferred metric for VCR systems. The model for cycle yield pulls in the properties from the air such as the mass flow rate, temperature, and humidity ratio along the four key locations in the system. The 'P' tags represent the points of the refrigeration cycle. This state properties are used to compute the cycle metrics explained earlier. The following first principle equations were used:

$$Water\ volume\ rate\ [\dot{V}] = \frac{mass\ rate\ [\dot{m}]}{density\ [\rho]} \quad [43]$$

$$Cycle\ Efficiency\ [\eta_{cycle}] = 1 - \frac{Q_L}{Q_H} = 1 - \frac{h_4 - h_1}{h_3 - h_2} \quad [44]$$

$$Carnot\ Efficiency\ [\eta_{Carnot}] = 1 - \frac{T_L}{T_H} \quad [45]$$

$$Coefficient\ of\ Performance\ [COP] = \frac{Q_L}{\dot{W}_{net}} = \frac{h_1 - h_4}{h_2 - h_1} \quad [46]$$

The subscripts for the enthalpy refer the location of refrigeration cycle where: 1 is after the compressor, 2 is the state after the condenser, 3 is the state after the expansion valve, and 4 is the state after the evaporator. Another metric that is calculated is the amount of water produced per energy unit input (energy factor). This metric would allow for a quick baseline comparison of this system to on the shelf-dehumidification systems.

$$Water\ to\ Power\ rating\ = \frac{Water\ volume\ rate\ [\dot{V}]}{Work_{compressor}} \quad [47]$$

The total water produced throughout a day can be found by integrating the water volume rate produced over time:

$$W_T = \int_{t_i}^{t_f} \dot{V} * dt \quad [48]$$

$$W_T = \sum_{t=i}^{n} \frac{V_i + V_{i+1}}{2} (t_{i+1} - t_i) \quad [49]$$

6. System Simulation/Results

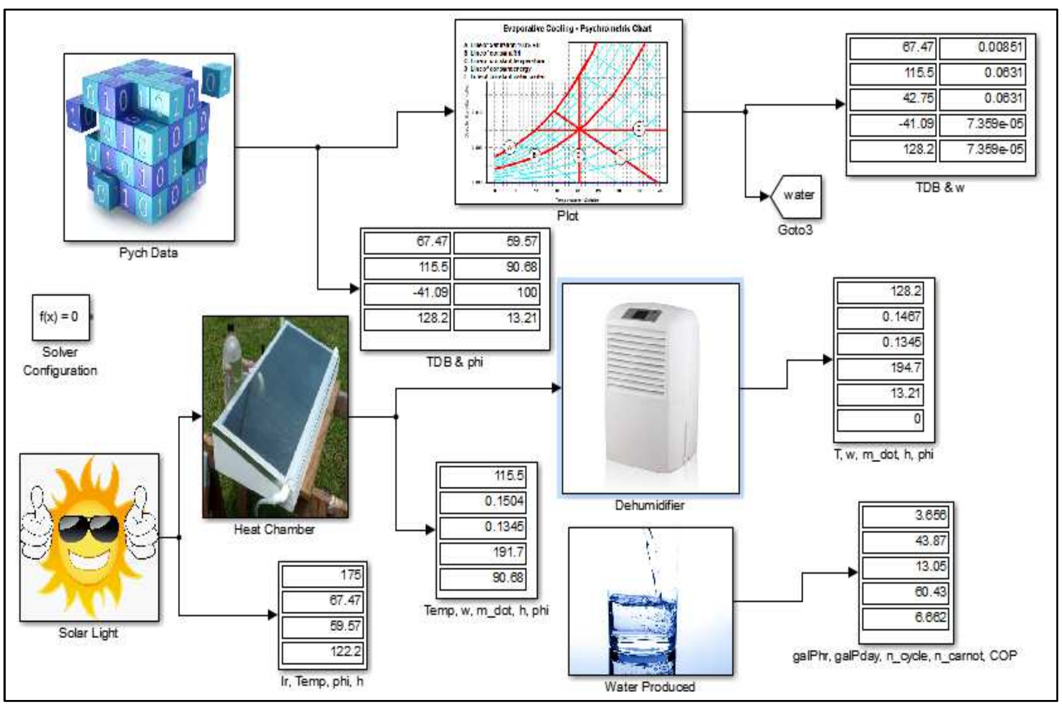
The complete system model involves the components and integration of every model discussed earlier. The goal here was to piece together a model that would function like the transfer model shown in Section 3. The display GUI (graphical user interface) in MATLAB was used to show the user key data values throughout the system such as the temperature, enthalpy, and humidity values. The values in this image primarily display the information of the air cycle. The output from the water cycle function gives the cycle yield parameters such as the amount of water produced and the system efficiencies.

6.1. Integration

As found in the Literature Review, success of any project begins with a well-defined plan of action. A combination of flow chart and input-output diagram were used to design and develop the integration plan. This ensured that the model was designed to succeed from the start. Although new parameters were added along the development, proper system planning was greater significant to the savings of development time. Parameters could be easily identified and observed for quick theory checks using display nodes. See the appendix B for the system mapping charts and I/O design. In addition, a data acquisition model was constructed to log the data at various nodes in the system to a MATLAB array file called 'Results'.

Using a 'Data2XLS' function, the results of the array are written to a Microsoft Excel file at the final time ($t = 12$). This allowed for the data to be collected and represented more easily. The data could be displayed in charts, plots, and graphs that are not easily made in MATLAB. Also, the use of excel as a statistical software would be used to quantify and compare the data. Figure 11 provides a visual of the complete

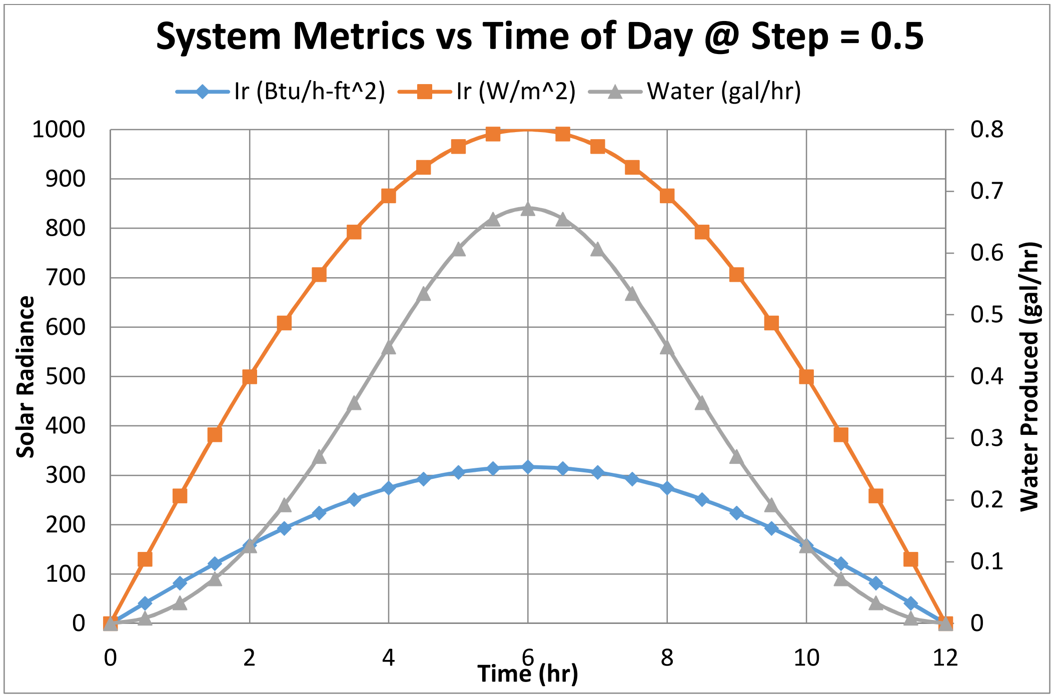
Simulink model. Close attention was paid to the look of the model to match the transfer system created in the planning phase. Along with the input-output mapping, the creation of the model was found to be more simplistic by knowing what parameters are necessary for the computation in each component model. The use of display blocks, shown in Figure 11, provide a quick check to assess the functionality of the system by following the air cycle. From the concept design, the temperature of the air should be seen to increase from atmosphere through the heat chamber, decrease through the evaporator and heat back up through the condenser. The values displayed during simulation matched this concept.



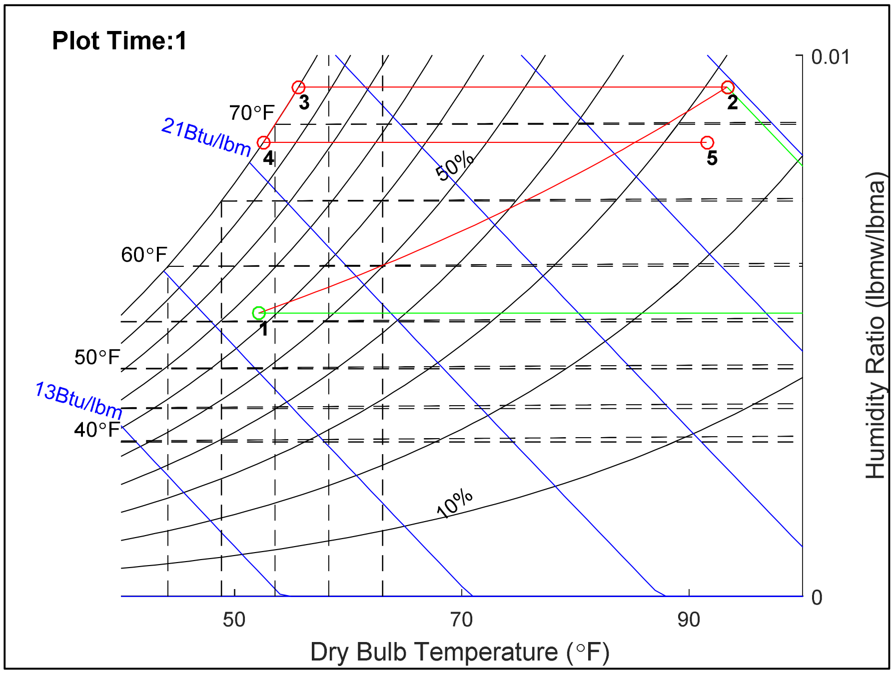
F.11 System Model Design: represents the complete system model in Simulink. Note the design layout similarity to the transfer system model designed in Figure 2. This highlights the importance of system and integration planning in a simulation study. The Input-Output pinouts, shown for each model and detailed in Appendix B, made the modeling development very simplistic and modular.

6.2.Results

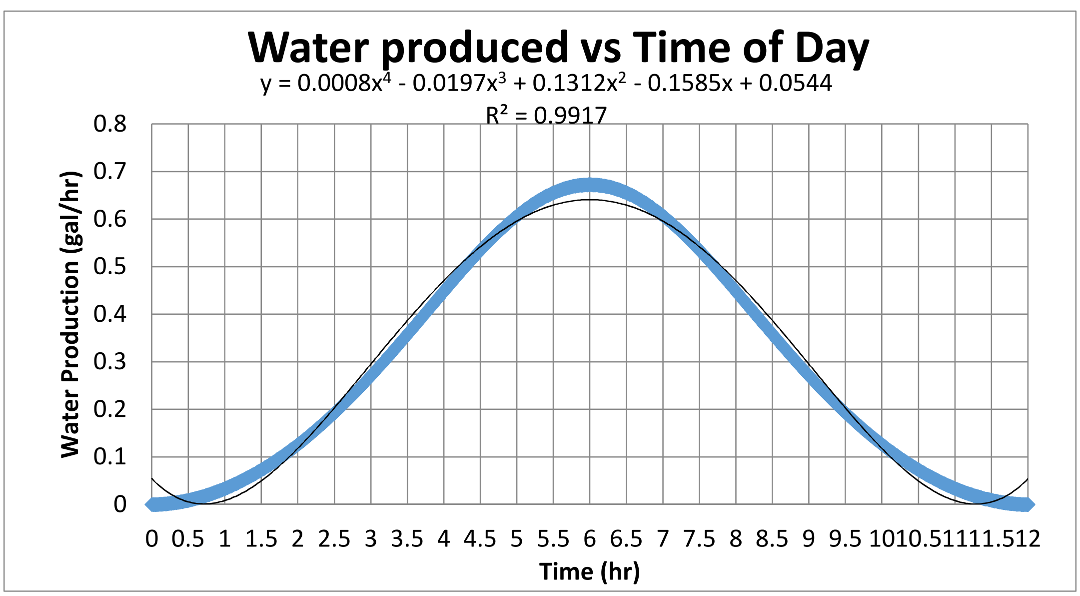
This section presents several key figures and tables during the study that display the system dynamics as well as the findings.



F.12 System Operation Curve @ step=0.5 : this figure shows the relation between the solar radiation and the amount of water produced. The higher the radiation energy, the more water is generated at a constant rate for polluted water addition. The peak radiation value of 1000 $\frac{W}{m^2}$ is centered about mid-day to generalize a typical earth day. The distribution of water produced can be described as more Gaussian and normalized through the day due to the bell shaped curve.



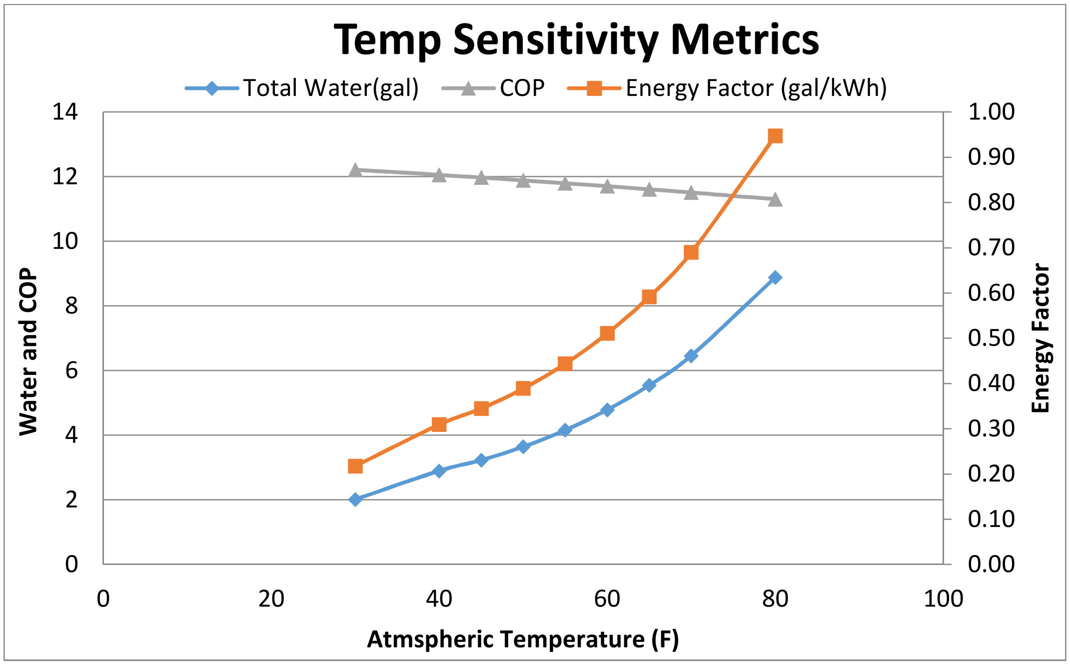
F.13 Complete Air Psychometric Chart : this figure highlights the completed psychometric plot. The red trace follows the air cycle throughout the system whereas the green trace highlights the addition and evaporation of black water in the heat chamber.



F.14 Water Produced per Day: the total water amount is computed from the integration of the water produced curve. The computation of the amount can be done in the following two ways:

$$W_T = \int_{t_i}^{t_f} \dot{V} * dt \qquad\qquad W_T = \sum_{t=i}^{n} \frac{V_i + V_{i+1}}{2} (t_{i+1} - t_i)$$

Both these formulas were used with the rectangular rule of integration being preferred due to the manner in which the data was saved to the data file.

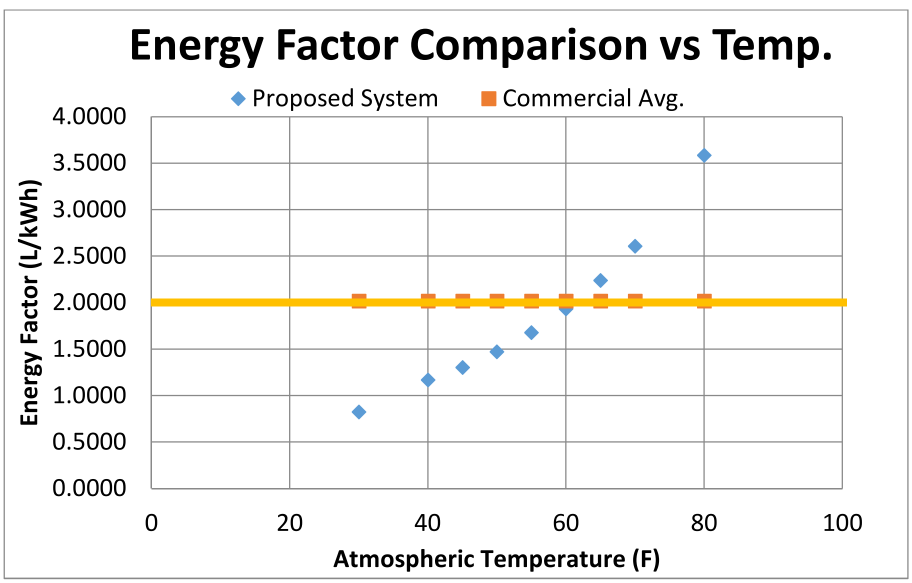


F.15 Refrigeration Cycle Performance : a comparison of the total water produced, refrigeration cycle efficiency, and the energy factor of the system. Both the total water produced and the energy factor can be seen to increase along with increases in temperature. The higher the atmospheric temperature in a particular region, the more water can be produced and the more water efficient per unit energy the system becomes as the high temperature allows for more water to be extracted. The coefficient of performance, that represents the heat transfer efficiency, decreases slightly due to a smaller thermal change from the atmosphere to the heat chamber exit. Both the mass flow of black water into chamber and the peak solar radiation are constant.

T.7 Water and Energy Factor per Time Step

The changes in water and energy factor as shown with relation to the time step selected in the simulation. The smaller the time step, the more realistic the results become. The last two columns show a relative comparison of the other time steps to t=1. From this it is clears that decreasing the time step affects the energy factor computation more than it does the total water produced.

Time Step	Average Water (gal/hr)	Total Water (gal)	Work input (kW)	Energy Factor (gal/kWh)	Energy Factor (L/kWh)	Relative % error of Total water	Relative % error of EF
1	0.2804	3.64540	0.75	0.374	1.415	0	0
0.5	0.2917	3.64565		0.389	1.472	0.0070	3.85
0.0167	0.3032	3.64573		0.404	1.530	0.0091	7.52



F.16 Energy Factor vs Base Temperature : a comparison of the proposed system and commercial products energy factor is presented. The average for commercial products of similar dimension, heat exchangers $(1' * 1' area)$, is 2 $\frac{L}{kWh}$. The proposed system is able to surpass this when the atmospheric temperature is 60°F and above.

6.3.Discussion

Since this research is a simulation study, the results of the system are iterative and can be timed for as long as needed. The time scale could be easily adjusted to run for hours or for a few seconds, depending on the input parameters. As stated earlier, the atmospheric conditions are fixed but are modified to seem dynamic to the system. It is possible to increase the realism of the model by increasing the complexity of each component model. The solar model can be made more realistic by calculating the solar irradiance to the geo-location of the earth and accounting for clouds, ground reflectance, earth emissivity, angle, and many other parameters. The more complex the model the more reliable the data becomes. The goal of this study was to provide the backbone and assess the system feasibility.

The key models that are the most complex in this study were the heat chamber and the heat exchanger. A significant amount of time was spent on these two models to make them as real as possible with the fundamental thermodynamic principles. Although there are still some assumptions made to simplify the models such as the heat energy lost from the heat chamber being assumed to a fraction of the total energy into the chamber. Most of the assumptions made were primarily due to the fact that the parameters in question required some defined material properties. Since this was not an experimental study or validation study, defining material properties was not found to be critical to the system backbone analysis. The production of a prototype and material selection is presented in the future studies section of this document. The Heat exchanger model employs the use of heat transfer equation for two fluids in a shell and tube design. It is possible to analyze other heat exchanger designs as well by changing the heat exchanger type in the function code. Both the Log Mean temperature and effectiveness method are used to compute the exit temperature streams of both fluids. The main assumption to be fixed here is the initial temperature of the refrigerant.

The heat exchanger increases the temperature and relative humidity of the air through forced convection, solar heat energy, and adding drops of polluted water into the chamber. An adequate model here means the amount of water vapor added to the air stream can be accurately predicted. The heat exchanger is the second key to the puzzle since the high humidity air is cooled through the evaporator heat exchanger. This means that an accurate model will predict the amount of water that condenses and could be purified for water drinking. The ability to clearly predict the water production is the hypothesis for this study and determines the technological feasibility of the system. This is realized in Figure 12 where the increase in solar radiation is seen to generate more water in the system.

The chart in Figure 13 provides a glimpse of the air properties through the cycle points. This chart is only intended to give the user the ability to perceive if the model is functioning as it should be. The red line traces the air through the cycle points stated in Section 5. The process of heating and adding humidity to the air in the heat chamber is represented by the green line. It is noted that the heat within the chamber is well into the 100°F range which is hot enough to cause evaporation of water droplets.

Additionally, all the data that is computed in this study is saved to an excel file that could be easily shared and used by other programs. The use of tags within the code allowed for the models to be more cleanly interfaced together. All tags in the model are global so that information can be transferred between nodes quickly. In some cases, tags were not used. Instead, arrows were shown between models to represent the flow of data and fluid from component to component. At the end of the time sequence (user defined and adjustable) the data in the 'Results.mat' file are stored to an excel file.

The other important result from the simulation is the psychometric chart. As stated in the model description, this model is important because of the psychometric figure that would allow any researcher to quickly identify the air properties and determine if the cycle is correct (based on the trace). If one were to overlay the charts, they would be able to identify how changes in one variable affect the air and thus the corresponding water production. Considerations for improvements to the system model are covered in the future research section (Section 10): However, it is clear that a design of experiments could be conducted on the simulation to optimize the output for water, hot exhaust, or chamber temperature.

As stated earlier, the time scale for the simulation could be easily adjusted to represent a 12 hour day through the time step selected. The computation of the timing sequence is shown in the verification section. Adjusting the time step allowed the simulation to represent the collection of data points at every hour, half-hour, minute, or even second. The smaller the time step, the smaller the integration error in the simulation became thus making results better at the cost of simulation run time.

The system analysis from Figure 14 shows the total amount of water that can be produced by the system based on the stated assumptions from section 4. By integrating the water produced per hour over a total time of operation, the **average total amount of 3.6 gallons** was found. The two methods of integration possible are to use equation of the polynomial fit curve for the integral or to use the trapezoidal rule of integration. The trapezoidal rule was used for the analysis. The result was close to the target value of 4.0 gallons and indicates great possibility for the system. Additionally, the COP of the system was between 11 and 12 units, indicating great performance of heat energy transfer in the system. It is noted in Figure 15 that the COP is seen to steadily decrease as the temperature rose. Since the work input to the compressor is constant, the decrease in COP comes from the numerator factor (the heat into the system). As the heat chamber increases the air temperature, the refrigeration cycle is not able to cool the hotter air to the same effect from the same power output.

A quick comparison of the simulated system to commercial dehumidifier products can be established on the energy factor basis. The energy factor, which is the amount of water removed by the system to the energy consumed, is computed for every commercial dehumidifier. According to Energy Star, the average energy factor among qualified efficient dehumidifiers is about $2.02\frac{L}{kWh}$ $(0.6\frac{gal}{kWh})$ with the best systems getting up to $4\frac{L}{kWh}$ $(1.06\frac{gal}{kWh})$ (Dehumidifier Basics, 2016). In comparison on Table 7, the simulated system was found to have an energy factor of $1.5\frac{L}{kWh}$ $(0.40\frac{gal}{kWh})$.

This result of the proposed system is 25% less than standard commercial units of a similar size dimension as the heat exchangers ($1' \cdot 1'$cross section area) modeled. The primary cause of this low energy factor may be due to the amount of energy assumed. As shown in Figure 16, the energy factor of the system can be made to outpace that of commercial items based on the compressive load required. An experimental study on this hypothesis could better evaluate the energy factor as the balance of energy needed to run the compressor or minimum energy needed to run the compressor.

Another factor to be considered is the cost of the system. A brief assessment of the how this proposed system would relate to other commercial products and other desalination systems. Since this concept envisions a small portable device with a possible total area of $12ft^2 (1.115m^2)$, the initial investment and material cost for a single unit would be far less compared to industrial plants of standard water desalination systems. Published research by Ghaffour and co., reveals the range of water cost per unit produced for industrial plants is between $0.7 - 1.0\frac{\$}{m^3}\left[2.65 - 3.79 * 10^{-4}\frac{\$}{gal}\right]$ with typical production capacity values above $50{,}000\frac{m^3}{day}\left[1.32 * 10^7\frac{gal}{day}\right]$ (Ghaffour et al., 2012). Industrial sized thermal systems on the other hand have higher unit production cost of $2\frac{\$}{m^3}\left[0.008\frac{\$}{gal}\right]$. (Ghaffour et al., 2012).

In contrast, the proposed system does seem to be more costly per unit with a result of $15.3\frac{\$}{m^3}\left[0.0579\frac{\$}{gal}\right]$, energy if a $0.15\frac{\$}{kWh}$ estimate is used. These values seem logical as a small device does not produce large amounts of water to significantly reduce the unit cost. On the other hand decreasing the size of the system dramatically will lend to reduce the other cost such as investment, materials, labor, maintenance, and space/location. Based on the current analysis, the system viability based on cost is dependent on the components used. A more power efficient compressor would greatly improve the production unit cost. Additional changes that could be made to improve the model, efficiency, and cost parameters are discussed in section 8 and 10.

7. Verification/Sensitivity Analysis

7.1.Computational precision Test

During the construction/programming of the model, the dehumidifier system was tested to ensure near realistic design. The test involved running through two refrigeration examples from a thermal textbook and matching the results of the simulation to the hand calculation. One example involved the analysis of an ideal vapor compression refrigeration cycle and the other assessed a real thermal cycle with irreversibility's (entropy). The results of the verification are shown below. The values between the hand calculation and the simulation show an average of 1% difference in Tables 8 and 9. It is believed that this difference stems from the thermal data base for which the properties of the fluids are referenced.

A simple test on the enthalpy of air at a given temperature and pressure found that the values received from 'RefpropMini' and a two thermal reference sheets differed by 100 units. This was unexpected as the thermal properties of the same fluid should be near equal regardless of database. It is therefore noted that care should be taken by any researcher working between 'RefpropMini' and other thermal reference databases.

T.8 Refrigeration Verification Example 1

A setup of example 8.1, from the *Introduction of Thermal Fluids and Engineering* textbook, by Kaminski & Jensen. This example focusing on the calculations for an Ideal VCR cycle and the results of the comparison between the simulation and hand calculations are shown:

Parameter	Simulation	Expected	Percent Difference (%)
Work Rate (Hp)	3.251	3.27	0.58
Heat Rate Out$\left(\frac{Btu}{min}\right)$	743.4	740	0.46
COP cooling	4.352	4.33	0.51

T.9 Refrigeration Verification Example 2

A setup of example 8.3, from the *Introduction of Thermal Fluids and Engineering* textbook, by Kaminski & Jensen. This example focusing on an ideal VCR heat pump system and the results of the comparison between the simulation and hand calculations are shown:

Parameter	Simulation	Expected	Percent Difference (%)
Work Rate (kW)	6.52	6.61	0.76
COP heating	5.261	5.29	0.55

7.2.Simulation Convergence Test

In addition to the verification of the refrigeration cycle model, a time sensitivity test was conducted. The purpose of which would show how changes in the time step of the simulation would benefit the results of the study. It was expected that the smaller the time step used, the smaller the simulation integration error. Additionally, decreasing the time step was also expected to increase the overall time needed to complete the simulation. The table below illustrates how the time step could exponentially affect the total simulation run time. The base for this section is that the 12 time periods designed into the model corresponds to 12 Earth hours. The significance of the 12 hour time is the day light for which the system would be operational and be maximized with the solar energy.

From the table, it is clear that changes in the time step affect the model as predicted. The graphs from the time analysis can be found in appendix C, however two images are presented below for a quick comparison. Although the smaller time step shows a smoother solar irradiance curve (more accurate), it took twice as long to complete compared to the figure on the left. Minimizing the integration error allows for better predictability of the system. The solar radiation is tied to the temperature value although the values are fixed. Better control and observation in the radiation curve allows for the prediction of water produced and the corresponding changes in the air conditions.

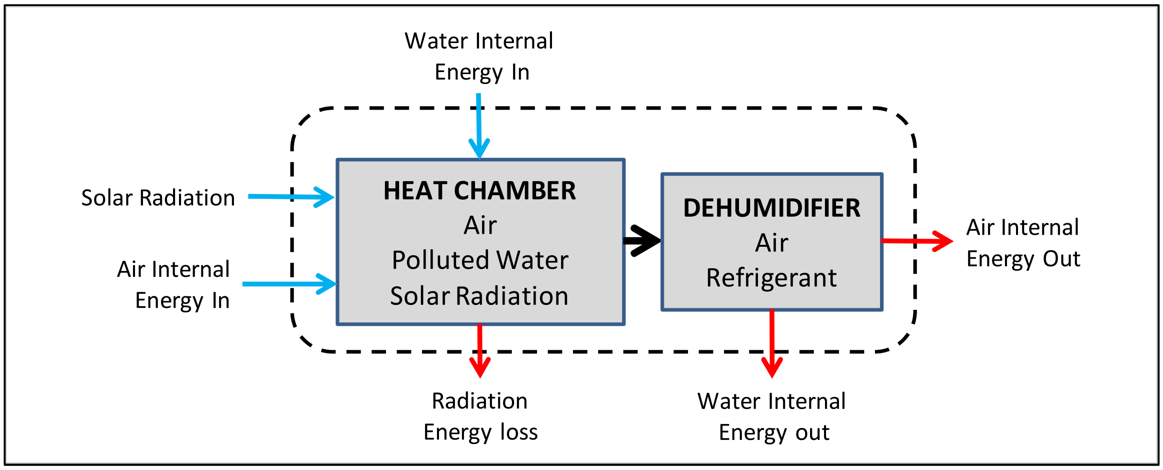
T.10 Simulation Time Analysis

This table presents the simulation run times based on the change in time step. As discussed in the Section 6, changes to the time step improve the convergence of data points in the simulation. As seen in Table 7, higher convergence was necessary for certain parameters but not all such as the energy factor.

Simulation final time (t_f) = 12 $\rightarrow 12\ hrs$		
Time step(Δt)		Total Run Time (t_T)
1	$1\ hr$	$12\ s$
$\frac{1}{2} = 0.5$	$0.5\ hr$	$24\ s$
$\frac{1}{60} = 0.0167$	$1\ min$	$720\ s\ \approx 12\ mins$
$\frac{1}{3600} = 2.78 * 10^{-4}$	$1\ s$	$43200\ s\ \approx 720\ mins\ \approx 12\ hrs$

7.3.System Energy Balance Check

A system energy balance was conducted to ensure that model was theoretically logical. The system boundary was placed around the heat chamber and dehumidifier as shown in Figure 17. The energy inputs to the system noted are the solar radiation, internal energy of the air and polluted water. On the other hand, the energy outputs are the exhaust air from the dehumidifier, drinking water, and the radiation loss from the heat chamber.



F.17 Total System Energy Balance: an energy balance diagram to aid in the identification of the energy parameters that pass through the system boundary.

The energy balance equation for the system is defined as follows:

$$[E_{solar} + E_{air,in} + E_{water,in} = E_{air,out} + E_{water,out} + E_{loss}] \quad [50]$$

$$(q_{solar} \cdot A_c) + \dot{m}_{a,i} \cdot h_{a,i} + \dot{m}_{w,i} \cdot h_{w,i} = \dot{m}_{a,o} \cdot h_{a,o} + \dot{m}_{w,o} \cdot h_{w,o} + (q_{loss} \cdot A_c) \quad [51]$$

Using values throughout the simulation and output data file, all these parameters were found and computed:

$Energy\ In\ -$	$Energy\ Out$	$\sim Difference/Percent\ Error$
27.92	28.31	0.39/1.4%

The average results for system found a 1.4% difference between the two energy sides. This difference is most likely accounted for with the initial assumption of the evaporator conditions and the simplified conversion factors used in the modelling of come components. Rounding of numbers has an effect on any calculation and most scientific methods recommend carrying all numbers and rounding at the end of the computation.

8. Validation Consideration

This section aims to outline a methodology that will allow for the created model to be checked and validated in future studies. The constructed model relies on the continuous flow of air and refrigerant fluid. To validate this, a prototype must be designed and constructed that will allow for the testing and collection of data at various points in the system. The definition validation used in this work means to compare how well the computer model matches the conditions and parameters in the real world application. The work or experimental instructions that may be used to validate this model are represented. Ideally, following these steps will provide any researcher with a built prototype, the information necessary to compare with the simulated model.

Needless to say that the validation of the research presented cannot be achieved without a physical model of the system for which must be built. Care should be taken in building a prototype to account for methods and ways of testing the system. Much like the care taken to ensure model component integration in the design phase of this research, future studies to validate the simulation must be aware of which parameters are easy to test and which are more difficult. Success to model validation lies in the ability to find the critical parameter of a system that can be directly tested. For example the temperature and pressure of the air and refrigerant may be easy to test directly however the enthalpy and entropy are parameters that need to be measured indirectly.

8.1. Critical Parameters

This study simply provides the backbone for feasibility simulation of a new hypothesis for low volume with solar energy and a refrigeration cycle. No tests were conducted to validate the system although all results from the simulation indicated that the system is feasible. This document presents the findings of the study such that a researcher could easily take the code and construct a physical prototype to test and validate the characteristics of the system or improve on the complexity of the model to account for new situations or parameters. Therefore, this section will detail the believed critical parameters of the system that would be valuable to testing an experiment.

As stated early, the heat exchanger and the heat chamber models were the most important to the accuracy of the simulation. However, in terms of testing the, the heat chamber becomes the more important critical parameter. This is because there are several characteristics of the heat chamber that can be adjusted in an experiment. Some key parameters are:

- The dimensions of the chamber (length, width, and height). These parameters can be grouped together to test as a volume constraint or tested separately.
- The amount of water flow allowed into the chamber.
- The angle of incline of the chamber.
- The air volumetric flow into the chamber. Using a fan for forced convection or testing natural convection flow.
- And also the materials used in the design of the heat chamber to increase chamber temperature or light.

The second critical parameter is still testable although it may not be as straight forward. By test and changing the heat exchangers used, one can better assess the effect on the water and cycle dynamics. This is probably more difficult to test and may require the construction/purchase of multiple prototypes. A tube and shell heat exchanger was used in the conducted simulation study. However in practice, one could use a two tube counter flow system or a shell and tube with a different number of passes. The possibilities are many meaning that a well-defined research goal is needed. Another critical parameter that may be worth validating is the compressor power of the refrigeration cycle.

One would expect that increasing the compressor power would allow for colder refrigerant flow and higher mass rate that could alter the water cycle yield. It should be noted that the hypothesis for this system is to function solely based on solar energy thus the more power drawn by the compressor the more solar panels and energy needed. It may be better for a prototype to select a portable sized solar array and then determine the best compressor available for the given solar energy (size the compressor to the solar panels).

8.2.Sample Test Procedure

A sample work instruction details the possible steps that may be taken to validate this computer model. See Appendix D for the complete testing guide.

The software model visualizes a small portable unit for low volume water production and filtration. The system relies solely on solar energy thus making it environmentally friendly with an open loop air cycle and a closed loop refrigerant cycle. The computer model is characterized by state properties of the air and refrigerant at nodes within the respective cycles. Physical measurement of the properties is required to match them with the simulated values.

The instructions aim to provide an experimenter with an idea of what variables can be measure and which must be inferred. For example, if may be easy to measure the air temperature, relative humidity, and seep given the right instruments: However, it would be difficult for anyone to measure the heat gain of the air directly. During the design of an experiment for this system, it is very important to consider what data is useful and how it will aid. Taking data for the sake of it will make the characterization of the system more difficult and not easily comparable to the data found here.

9. Conclusion

The results of the study based on well-defined and stated assumptions: such as steady state system, slow flow rate of contaminated water into the chamber, heat chamber and heat exchanger physical characteristics. These assumptions help guide to the conclusion that the system is technologically feasible for low water production.

The primary components of the model were the heat chamber and the heat exchanger. These components were responsible for heating the air and the extraction of liquid water. The heat chamber was fundamental to increasing the atmospheric air temperature using solar radiation and also increasing the humidity of the air by evaporating water vapor from contaminated/polluted water. An open system thermal energy balance formula was used to model the heat chamber. The heat exchanger was the second critical component for this model in that the net yield of water was computed through this component. Both the log mean temperature difference and effectiveness NTU methods common in thermal heat exchanger analysis were utilized to determine the exit temperatures of the air and refrigerant fluids. The refrigerant used in this study was R134a and the thermal database for generating fluid parameters was 'RefpropMini'.

The energy factor of the system was found at $1.5\frac{L}{kWh}$, which is within 25% of commercial product values. Additionally, the system was able to produce 3.6 gallons of water within a 12 hour sunlight day. Granted the assumed atmospheric conditions were fixed and were not an ideal representation of a typical day with clouds, solar refraction and/or rain. However, it is expected that a real world condition system would still provide results similar to those found in this study. From a cost perspective, this model would not be able to compete with the low unit production cost found with industrial reverse osmosis plants. However, since the size and goal of this system is much smaller, a unit production cost of water would be higher. The next step of this study would be to improve the variables of the solar model and add solar panels to better predict the water output. Further future research ideas are described in the next section.

10. Future Research

Future researchers working on the subject of refrigeration cycles or dehumidifiers have a great tool available from this study. The code provided here can be utilized for a variety of studies and applications other than dehumidification systems. Additionally, for studies that encompass solar dehumidification systems, this simulation package would provide a great baseline for an early estimate. As stated in the introduction, most research into solar dehumidification for water desalination tends to focus on large scale industrial systems. However, this model was built for small and portable water production systems. Therefore the model would be most effective when utilized to study smaller solar stills or solar dehumidifiers. That being the case, it is still possible for a researcher to modify the code and sizing of the system to test industrial scale designs.

Although the MATLAB model created was complex enough to run the necessary computations and show results, the software model could still be improved in a variety of methods. To begin with, the solar irradiance model could be updated to clearly reflect the real world changes in temperature. The current model uses a set of fixed assumptions to simulate dynamic sun and temperature changes. However, the model could be changed to use the Typical Meteorological Year (TMY) data. This dataset is a compilation of real environmental conditions from 1961 to 2005. The latest release, TMY3, contains data for over 1000 location across the world and provides the most accurate log of solar radiation values. In addition to using the TMY data, the solar model could also be improved upon to include real time atmospheric conditions that determine the temperature and relative humidity based on the solar radiation, geo-location (longitude and latitude), surface angle, cloud density, ground reflectance, and several other factors.

Another area of improvement would be the compressor and throttling valve components of the refrigeration cycle. In this model, these two components were represented with the first law principles of an open thermal system: However, there are other factors that could be added to make the model more accurate including the dimensions of the throttling valve, the efficiency of the compressor, and the operational conditions of the compressor. For the system to be truly considered to be energy efficiency and self-producing, a solar panel model is needed to interface the compressor with the solar model. The solar model can be used to size the compressor and further refine the feasibility of the system. For example, one must consider how much power the solar panels can produce based on the solar radiation and when the power threshold is enough for the compressor to be operational. This power threshold further reduces the operational time of the compressor during the day and thus affects the maximum amount of water that can be produced.

The presented ideas for future improvements have been software based, however, one major key for a scientific research study is the ability for others to validate and verify the conclusions of the study. The perceived critical parameters of the system were discussed in section 8.1. (Validation Considerations: Critical Parameters). There are many test and experiments that could be conducted to check the results of this study. Ideally, the results of this study and any experimental test should correlate to the general theory of the system. It is possible for the entire system to be validated at once with a prototype that contains all the functional components or for a researcher to validate each component on a singular bases. For example, a study could be conducted on which heat exchanger type would allow for the maximum extraction of water from the humid air through the evaporator. Regardless of the validation system studied, this future study would also be able to optimize the system for real world working conditions. Despite the advancement and cost savings given by computer simulations, there are natural phenomenon's that cannot always be accounted for in a simulation. Therefore, constructing a physical model to test and experiment with would provide the most accurate data for real operating conditions.

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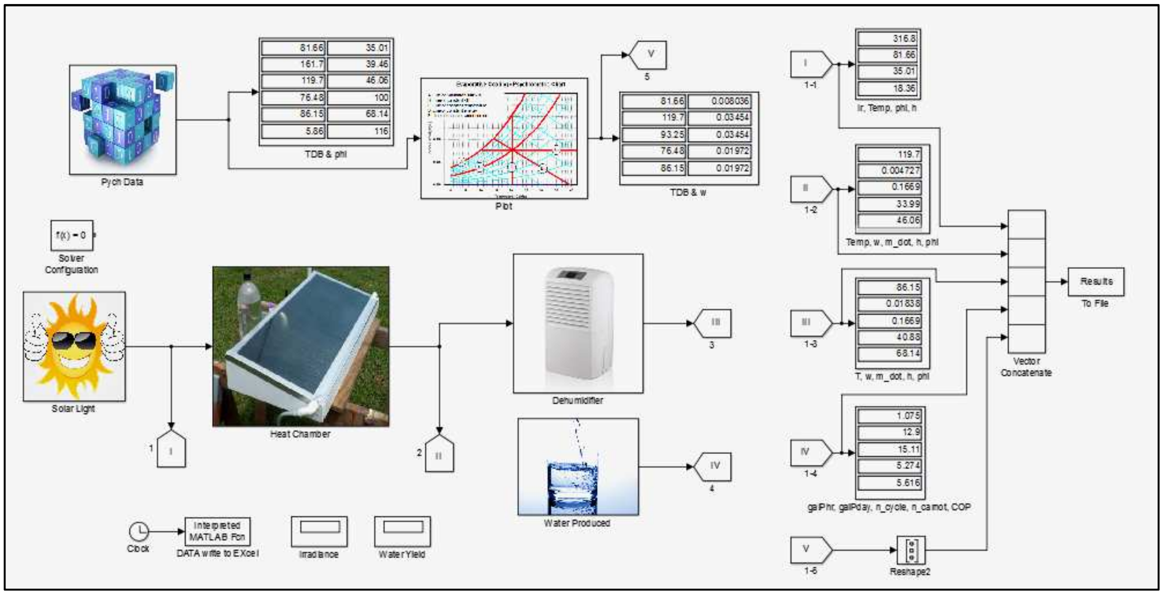
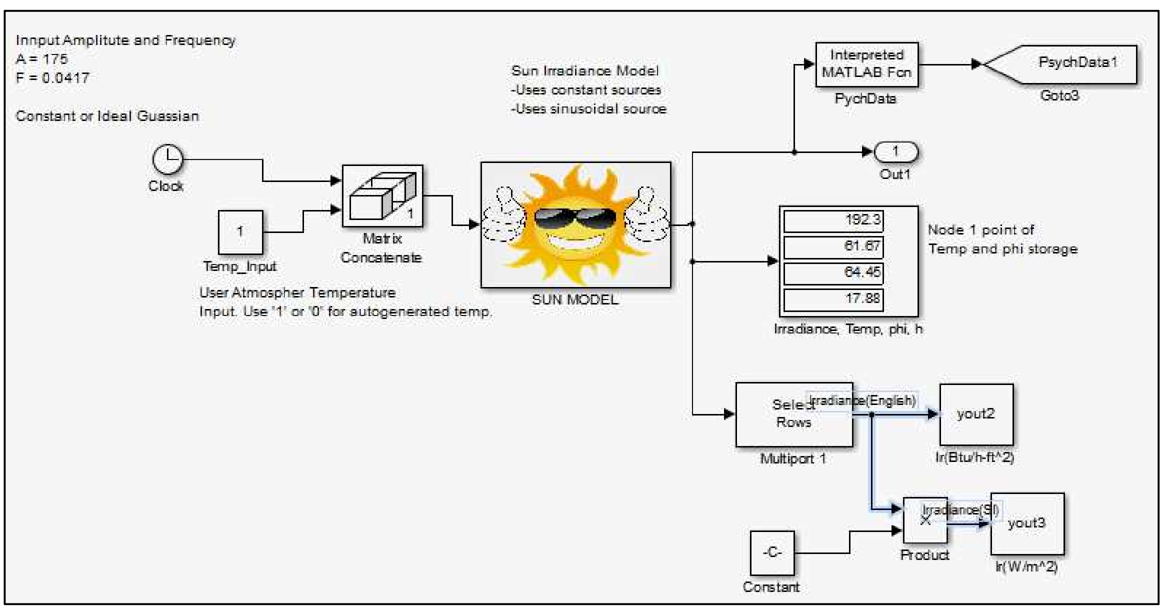
Appendix

A. System Code



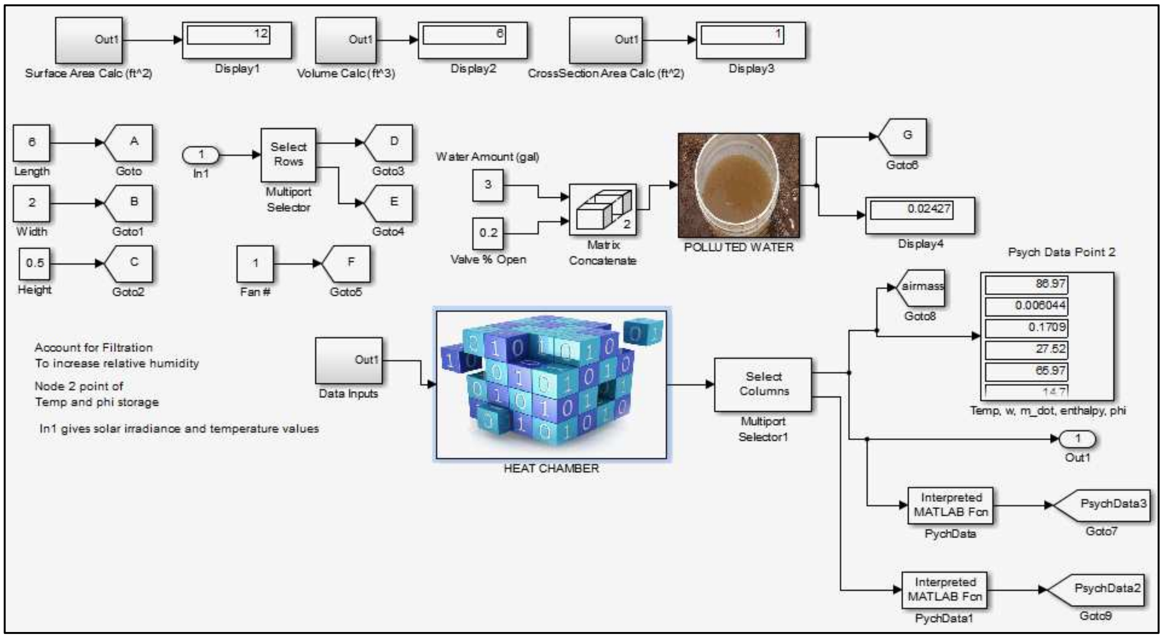
Solar Radiation Model Code




```
function y = SunLight(u)
%Find the estimated earth temperature
Time = u(1);
Temp = u(2);

Pressure = 14.686; %psi
```

```
% Fit to curve for dynamic interaction
Irradiance = 1000 *sin((1/24)*(Time*2)*pi);

    %Conver to english units
Sun_power = Irradiance * W2Btuph(1)/m22ft2(1);

if Temp <= 1
    %Assume the atmoshper temp to follow the solar radiation
    Temp_atm = (Sun_power/10)*sin((1/24)*(Time*2)*pi) + 50;
else
    %If user input data, curve to 12 hour cycle with peak at noon
    Temp_atm = (Temp/2)*sin((1/24)*(Time*2)*pi) + (Temp/2);
end

%Get the entalpy of the air at temp and pressure
[H] = refpropm('H','T',F2K(Temp_atm),'P',psi2kPa(Pressure),'AIR.PPF');
offset = 10;
enthalpy = (H * J2Btu(1)/kg2lbm(1))/ offset;

%Use entalpy to find relative humidity
[Temp_atm, w, RH, enthalpy,Tdp ,v ,Twb] = Psychrometricsnew
('tdb',Temp_atm, 'h', enthalpy,'p',Pressure);
%Offset the relative humidity from 40 percent
phi = RH ;%

%Normalize phi to 100
if phi > 100
    phi = 100;
elseif phi < 20
    phi = RH + 20;
end

%Pass out data for next system block
data = zeros(4,1);
data(1) = Sun_power ; data(2) = Temp_atm; data(3) = phi; data(4) =
enthalpy;

y = data;
```

Heat Chamber Code



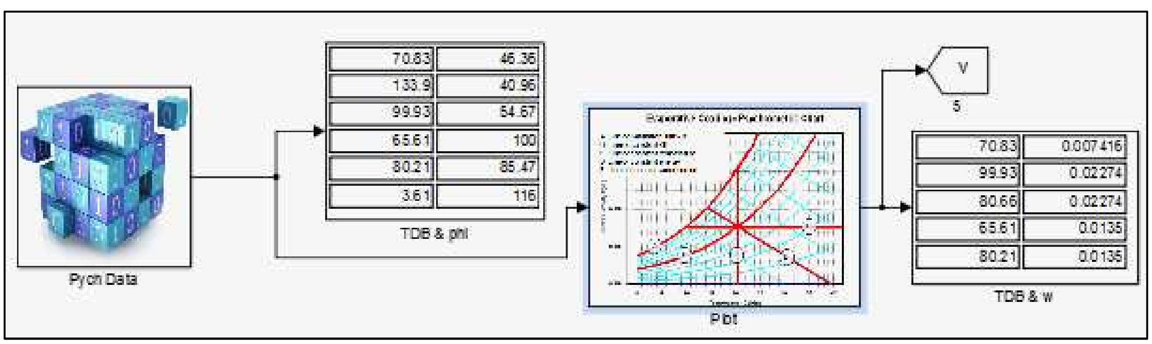

```
function y = HeatChamber(u)
%Function used to compute some parameters of the heat chamber such as
%   black body surface temperature, air mass rate, final temperature
%Input parameters: still lenght, still width, still height,
%   Solar irradiance, atmospheric temperature, air volume intake
%Output parameters: Air exit temperature, time, mass rate flow
%Assume atmospheric pressure in system 1atm ~ 14.7psi
%**********************************************************************
**
    %Data In
lenght = u(1); %ft
width = u(2); %ft
height = u(3); %ft
Sun_irradiance = u(4); %Watts/m^2
Temp_i = u(5); %F
Air_Intake = u(6); %cfm
mass_dot_water = u(7); %lbm/s

    %Constant
Patm = 14.696; %psi
Fluid = 'AIR.PPF';

%*******************************************************
[H C] = refpropm('HC','T',F2K(Temp_i),'P',psi2kPa(Patm),Fluid);
offset = 10;
%There seems to be a factor of 10 difference between Refprop and
%   Psychrometrics

h_i = (H * J2Btu(1)/kg2lbm(1));%offset; %;
C_p_air = C * J2Btu(1)/(kg2lbm(1)*K2R(1));
%Use entalpy to find relative humidity
```

```
[~, w, RH_i, ~,Tdp_i ,v_i ,Twb_i] = Psychrometricsnew ('tdb',Temp_i,
'h', h_i,'p',Patm);

%Offset the relative humidity from 40 percent
phi_i = RH_i;

%Normalize phi to 100
if phi_i > 100
    phi_i = 100;
elseif phi_i < 20
    phi_i = RH_i + 20;
end
[~, w_i, ~, ~,~ ,~ ,~] = Psychrometricsnew ('tdb',Temp_i,
'phi',phi_i,'p',Patm);

%Find the amount of air volume intake
if Air_Intake == 1
    Cfm_e = 200; %cfm of 1 fan at exit of input
    Cfm_i = Cfm_e/lenght;
    Cfm_avg = (Cfm_e+Cfm_i)/2;
else
    Cfm_e = 200;
    Cfm_i = 200;
    Cfm_avg = (Cfm_e/lenght+Cfm_i/lenght)/2;
end

Cfs = Cfm_avg/min2s(1); %cubic feet per second
mass_air_i = Cfs/v_i;

%Find the Chamber characteristics
Volume = lenght * width * height; %ft^3
S_A = lenght * width;               %ft^2
CS_A = width * height;            %ft^2

mass_dryair = mass_air_i/(1 + w_i);%lbm dry air

%*****************************************************
%Find the Solar energy within the chamber per unit area
q_solar = Sun_irradiance/hr2s(1);% * W2Btuph(1)/m22ft2(1);
    %Energy per unit area Btu/hft^2

%*****************************************************
%Water properties
T_water = 65; %F
[H_w H_fg C] =
refpropm('HYC','T',F2K(T_water),'P',psi2kPa(Patm),'water'); %kg/m^3
h_fg = H_fg * J2Btu(1)/kg2lbm(1);
h_w = H_w * J2Btu(1)/kg2lbm(1);
C_p_water = C * J2Btu(1)/(kg2lbm(1)*K2R(1));

%*****************************************************
% Assume heat loss is some fraction of qsolar in
%Get the energy loss from system
E_dot_solar = (q_solar*(S_A));%(Cfs/S_A)*lenght);
E_dot_airI = (mass_air_i*h_i);
```

```
% E_dot_water = mass_dot_water*h_fg;
E_dot_water = mass_dot_water* C_p_water * (F2R(T_water));
    %Find q_loss of the system Btu/ft^2
% Assume the energy lost to be some fraction of the energy into the
system
fraction = 0.2;
q_loss = (1/S_A)*(E_dot_solar+E_dot_airI+E_dot_water)*fraction;
E_dot_loss = q_loss*(1/S_A);

%*****************************************************
%Exit air properties without water
mass_air_o1 = mass_air_i;

T_o1 = (mass_air_i*C_p_air*(Temp_i-0)+
mass_air_o1*C_p_air*F2R(Temp_i)...
     + E_dot_solar - E_dot_loss)/(mass_air_o1*C_p_air );
Temp_o1 = abs(R2F(T_o1));% F

[~, ~, RH_o1, h_o1,~ ,~ ,~] = Psychrometricsnew ('tdb',Temp_o1,'w',
w_i,'p',Patm);
phi_o1 = RH_o1;
if phi_o1 > 100
    phi_o1 = 100;
% elseif phi_o1 < 40
%     phi_o1 = RH_o1 + 40;
end

%Exit air properties with water
mass_water_evap = mass_dot_water *(C_p_water*(Temp_o1-Temp_i))/h_fg ;
%lbm/s
w_plus =  mass_water_evap/mass_dryair;

%combine initial and water humidity
w_f = w_i + w_plus;
mass_air_out = mass_air_i + mass_dot_water; %lbm/s

%*****************************************************
%Determine the temperature from the energy balance
T = (mass_air_i*C_p_air*(Temp_i-0) +
mass_air_out*C_p_air*F2R(Temp_i)...
    + mass_dot_water*C_p_water*F2R(T_water) + E_dot_solar -
E_dot_loss)...
    /(mass_air_out*C_p_air + mass_dot_water*C_p_water );
Temp_f = abs(R2F(T));% F

%Get the enthalpy and relative humidity from water
 [~, ~, RH_f, h_f,Tdp_f ,v_f ,Twb_f] = Psychrometricsnew
('tdb',Temp_f,'w', w_f,'p',Patm);
    %Find the energy of air out
E_dot_airF = mass_air_out*h_f;

phi_f = RH_f;
if phi_f > 100
    phi_f = 100;
end
```

```
%Not using a heat chamber
%Tie output to input conditions
if Volume == 0
    Temp_f = Temp_i;
    w_f = w_i;
    mass_air_out = mass_air_i;
    h_f = h_i;
    phi_f = phi_i;
    Temp_o1 = Temp_i;
    phi_o1 = phi_i;
end

%*****************************************************
%Transfer outputs to array to the next model
data1 = [ Temp_f       %F
          w_f           %lbmw/lbma
          mass_air_out  %lbm/s
          h_f           %Btu/lbm
          phi_f         % percent/lbm_air
          Patm ];

%Only purpose of data2 is to pass intermediate step variables
data2 = [ Temp_o1   %F
          0
          0
          h_o1
          w_plus    %percent/lbm_air
          0      ];

data = [data1 data2]; %Empty cell to make array match
y = data;
```

Psychrometric Plot Code




```
function y = PsychData( input )
%Function called to collect air data on temperature
%   and relative humidity to be fed into the psych plot

%Find the number of rows in the input data array
nrows = size(input,1);
```

```
%Use if statement to find temperature and relative humidity values
if nrows <= 4
    Temp = input(2);
    phi = input(3);
else
    Temp = input(1);
    phi = input (5);
end

%Cap the relative humidity
if phi > 100
    phi = 100;
end

data_out = [Temp, phi];
y = data_out;

%*******************************************************************

%*******************************************************************

%*******************************************************************

%% Curve Trace
%Original Code by Muhammed Ali
%   Generated SI Pyschrometric Plot
%Modified by Daniel Appiah-Mensah
%   Generate Inch-Pound Pyschrometric Plot
%********************************************************************
**
function y = PsychPlot(data)

%Allocate arrays to save time

TDB = zeros(5,1);
RH = zeros(5,1);
w = zeros(5,1);
h = zeros(5,1);
Twb = zeros(5,1);
processdata = data;
Time = data(6);

Pamb = 14.7; %psi
addpath unit_converters/;
%Get Array of Dry Bulb Temp and Relative Humidity
%Seperate values for Dry Bulb and RH
TDB = [ processdata(1,1)
        processdata(3,1)
        0
        processdata(4,1)
        processdata(5,1) ];

RH = [ processdata(1,2)
        processdata(3,2)
```

```
        0
        processdata(4,2)
        processdata(5,2) ];

T_HC = data(2,1);
w_HC = data(2,2);

%Heat Chamber max temp
 [~, w_HC, RH_HC,H_HC,~ , ~,~] = Psychrometricsnew ('tdb',T_HC, 'w',
w_HC,'p',Pamb);

%Use a switch case to determine the various air cycle points
for i=1:5
switch i
    case 1
        [TDB(1), w(1), RH(1), h(1),~ ,~ ,Twb(1)] = Psychrometricsnew
('tdb',TDB(1), 'phi', RH(1),'p',Pamb);
%         if w(1) ~= w_HC
        if max(w(1),w_HC)== w_HC
            w(1) = w_HC;
            [TDB(1), w(1), RH(1), h(1),~ ,~ ,Twb(1)] =
Psychrometricsnew ('tdb', TDB(1), 'w', w(1),'p',Pamb);
        end
    case 2
        if TDB(2)==TDB(1)
            w(2)=w(1);
            [TDB(2), w(2), RH(2), h(2),~ ,~ ,Twb(2)] =
Psychrometricsnew ('tdb',TDB(2), 'w', w(2),'p',Pamb);
        else
            [TDB(2), w(2), RH(2), h(2),~ ,~ ,Twb(2)] =
Psychrometricsnew ('tdb',TDB(2), 'h', H_HC,'p',Pamb);
        end
    case 3
        w(3)=w(2);
        RH(3) = 100;
        [TDB(3), ~, ~, h(3),~ , ~,Twb(3)] = Psychrometricsnew ('w',
w(3), 'phi', RH(3),'p',Pamb);
        %Test or limit case 3

        if TDB(4) > TDB(3)
         TDB(3) = TDB(4);
         [~,w(3), RH(3),~,~,~,~] =  Psychrometricsnew ('tdb', TDB(3),
'w',w(3),'p',Pamb);
         RH(4) = RH(3);
        end
    case 4
        [TDB(4), w(4), RH(4), h(4),~ , ~,Twb(4)] = Psychrometricsnew
('tdb',TDB(4), 'phi', RH(4),'p',Pamb);
    case 5
        w(5)=w(4);
        [TDB(5), w(5), RH(5), h(5),~ , ~,Twb(5)] = Psychrometricsnew
('tdb',TDB(5), 'w', w(5),'p',Pamb);
end
end

%*************************************************
```

```
%Set limits for pysch plot
T_min = roundsd(min(TDB)-10,1,'floor');
T_max = roundsd(max(TDB)+10,2,'ceil');
    %Get extra space for heat chamber trace
if Time == 3 || Time == 6 || Time == 9
    T_max = roundsd(max(processdata(:,1))+5,2,'ceil');
end

w_max = roundsd(max(w)+0.005,1,'ceil');

%*************************************************
    %Get trace for system outline
Trace = [TDB,w];

%***************************************************
% %Create a pysch plot when the modolo of time is 0
if mod(Time,3) == 0 && sum(TDB) > 20
        %plot axis and figure
    axhandle=psychplotting(T_min,T_max,0,w_max); %TDB(F) and w
(lbmw/lbma)
    hold on;

    for i=1:4
        switch i
            case 1 %Increasing temp and RH in the heat chamber
                %Increases exponentially so find the exponential
coefficients
                % y = a*b^x
                b = (w(2)/w(1))^(1/(TDB(2)-TDB(1)));
                a = (w(2)/(b^(TDB(2))));
                %Get y values for temp
                if TDB(1) ~= TDB(2)
                    v = TDB(1):1:TDB(2);
                    for j=1:length(v)
                        y = a*(b.^(v));
                     end
                plot(axhandle,v,y,'-r');
                %Add line to close gap between V and TDB(2)
                x = [ v(length(v)) ; TDB(2) ];
                z = [ y(length(y)) ; w(2) ];
                plot(axhandle,x,z,'-r');
                plot(axhandle,TDB(1),w(1),'-ro');
                else %Set y to humidity ratio
                    plot(axhandle,TDB(1),w(1),'-ro');
                    plot(axhandle,TDB(2),w(2),'-ro');
                end

                    %Get extra space for heat chamber trace
%                 if Time == 3 || Time == 6 || Time == 9
                    x_a = [ TDB(1) ; T_HC ];
                    y_a = [ w(1) ; w_HC ];
                    plot(axhandle, x_a, y_a, '-go');
                    x_b = [ T_HC ; TDB(2) ];
                    y_b = [ w_HC ; w(2) ];
                    plot(axhandle,x_b, y_b, '-go');
%                 end
```

```
            case 2 %Cooling from point 2 to point 3 on pysch plot
                %Point 3 must stop at 100 RH
                %Trace a horizontal line
                %Evaporator coils
                v = [ TDB(2)
                      TDB(3) ];
                y = [ w(2)
                      w(3) ];
               plot(axhandle,v,y,'-ro')

            case 3  %decreasing temp and RH in the heat chamber
                 %decreases exponentially so find the exponential
coefficients
                %Decreases on 100% relative humidity
                %Evaporator coils
                v = TDB(4):1:TDB(3);
                    %Set phi and output to same matrix dimensions as v
                phi = ones(1,length(v))*100;
                y = zeros(1,length(v));
                for j=1:length(v)
                    [~, y(:,j), ~, ~,~ , ~,~] = Psychrometricsnew
('tdb',v(:,j), 'phi', phi(:,j),'p',Pamb);
                end
                plot(axhandle,v,y,'-r');
                %Connect the gap in the data
                x = [ v(length(v))
                      TDB(3) ];
                z = [ y(length(y))
                      w(3) ];
                plot(axhandle,x,z,'-r');

                case 4 %Heating from point 4 to point 5 on pysch plot
                %Condenser coils
                %Trace a horizontal line
                v = [ TDB(4)
                      TDB(5) ];
                y = [ w(4)
                      w(5) ];
               plot(axhandle,v,y,'-ro')
        end
    end

    for i=1:length(Trace) %plot process data numbers
    htext =
text(Trace(i,1),Trace(i,2),num2str(i),'color','k','horizontalalignment
','left','verticalalignment','top','fontweight','bold');
    end

    title(strcat('Plot Time:
',num2str(Time)),'fontsize',12,'fontname','arial','position',[T_min
w_max]);
end

y=Trace;
```

```
%*********************************************************************

%*********************************************************************

%*********************************************************************

% VARIABLES
% Tdb (dry bulb temperature) and Tdp(dew point temperature) in F
% w (humidity ratio) in lbm/lbm of water to dry air
% phi (relative humidity) in %
% h (enthalpy) in Btu/lbm of dry air
% v (specific volume) in ft3/lbm of dry air
% Twb (wet bulb temperature) in F
% P (atmospheric pressure) in psi

% The following cases are present:
% Tdb, w; Tdb, phi; Tdb, h; w, phi; w, h; phi, h; Tdb, Twb; w, Twb;
phi, Twb;

% Following ASHRAE 2013 Fundamentals SI Psychrometrics chapter
equations are used:
% Eq6:Pws=f(Tdb); Eq22: w=f(Tdb, phi, p); Eq24: phi=f(Tdb, w, p);
Eq28:v=f(Tdb, w and p); Eq32:h=f(Tdb, w and p);
% Eq35:Twb=f(Tdb,w); Eq39:Tdp=f(Tdb, p);
function [Tdb, w, phi, h, Tdp, v, Twb] = Psychrometricsnew (varargin)

if length(varargin)<4
    display('Need four inputs:''prop1'',value1,''prop2'',value2''');
    Tdb=[];w=[];phi=[];h=[];Tdp=[];v=[];Twb=[];
    return
elseif length(varargin)>4 && length(varargin)<6
    display('Need six
inputs:''prop1'',value1,''prop2'',value2'',,''Pamb'',value in psi''');
    Tdb=[];w=[];phi=[];h=[];Tdp=[];v=[];Twb=[];
    return
elseif length(varargin)==4
Tdb_in=[];w_in=[];phi_in=[];h_in=[];Twb_in=[];

prop(1) = {lower(char(varargin(1)))};
prop(2) = {lower(char(varargin(3)))};
propVal(1) = cell2mat(varargin(2));
propVal(2) = cell2mat(varargin(4));
P = 14.7; %psi, atmospheric pressure value
elseif length(varargin)==6
Tdb_in=[];w_in=[];phi_in=[];h_in=[];Twb_in=[];
prop(1) = {lower(char(varargin(1)))};
prop(2) = {lower(char(varargin(3)))};
propVal(1) = cell2mat(varargin(2));
propVal(2) = cell2mat(varargin(4));
P = cell2mat(varargin(6));
end

for i=1:2
switch prop{i}
    case 'tdb'
        Tdb_in=propVal(i);
```

```
    case 'w'
        w_in=propVal(i);
    case 'phi'
        phi_in=propVal(i);
    case 'h'
        h_in=propVal(i);
    case 'twb'
        Twb_in=propVal(i);
end
end
if (~isempty(Twb_in) && ~isempty(h_in))
    display('function not available');
    Tdb=[];w=[];phi=[];h=[];Tdp=[];v=[];Twb=[];
    return
end
c_air = 0.240; %Btu/lbmF, value from ASHRAE 2013 Fundamentals eq. 32
hlg = 1061; %,Btu/lbm value from ASHRAE 2013 Fundamentals eq. 32
cw  = 0.444; %Btu/lbmF, value from ASHRAE 2013 Fundamentals eq. 32

%++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
++++
%CASE of Dry Bulb Temp and Humidity Ratio
if (~isempty(Tdb_in) && ~isempty(w_in))
    Tdb=Tdb_in;w=w_in;

    % phi calculation from Tdb and w
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    Pws=Saturation_pressure(Tdb);
    phi=Pw/Pws*100;
%     phi=Pw/Pws;

    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb); %ASHRAE 2013 fundamentals eq. 32

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P; %ASHRAE 2013 fundamentals
eq. 28
end
%++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
+++++
%CASE of Dry Bulb Temp and Relative Humidity
if (~isempty(Tdb_in) && ~isempty(phi_in))
    Tdb=Tdb_in;phi=phi_in;

    % w calculation from Tdb and phi
    Pws=Saturation_pressure(Tdb);
    Pw=(phi/100)*Pws;
%     Pw=phi/10*Pws;
    w=0.621945*Pw/(P-Pw);

    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
```

```
end
%+++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
+++++
%CASE of Dry Bulb Temp and Enthalpy
if (~isempty(Tdb_in) && ~isempty(h_in))
    Tdb=Tdb_in;h=h_in;

    % w calculation from Tdb and h
    if Tdb > 0
    w=(h - c_air*(Tdb))/(hlg+cw*(Tdb));
    else
    w=(h + c_air*(Tdb))/(hlg- cw*(Tdb));
    end
%     w=abs(w);
    % phi calculation from Tdb and w
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    Pws=Saturation_pressure(Tdb);
    phi=Pw/Pws*100;
%     phi=Pw/Pws*10;
%     phi=Pw/Pws;
    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
end
%+++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
++++
%CASE of Humidity Ratio and Enthalpy
if (~isempty(w_in) && ~isempty(h_in))
    w=w_in;h=h_in;

    % Tdb calculation from w and h
    Tdb=(h - w*hlg)/(c_air+w*cw);

    % phi calculation from Tdb and w
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    Pws=Saturation_pressure(Tdb);
    phi=Pw/Pws*100;
%     phi=Pw/Pws;

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
end
%+++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
++++
%CASE of Humidity Ratio and Relative Humidity
if (~isempty(w_in) && ~isempty(phi_in))
    w=w_in;phi=phi_in;

    % Tdb calculation from phi and w
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    Pws=Pw/(phi/100);
%     Pws=Pw/phi*10;
    options=optimset('LargeScale','off','Display','off');
    [y,val,exitflag]=fsolve(@Iteration_function_1, 20,options);Tdb
=y(1);
    if exitflag<1
        disp('Iteration error')
```

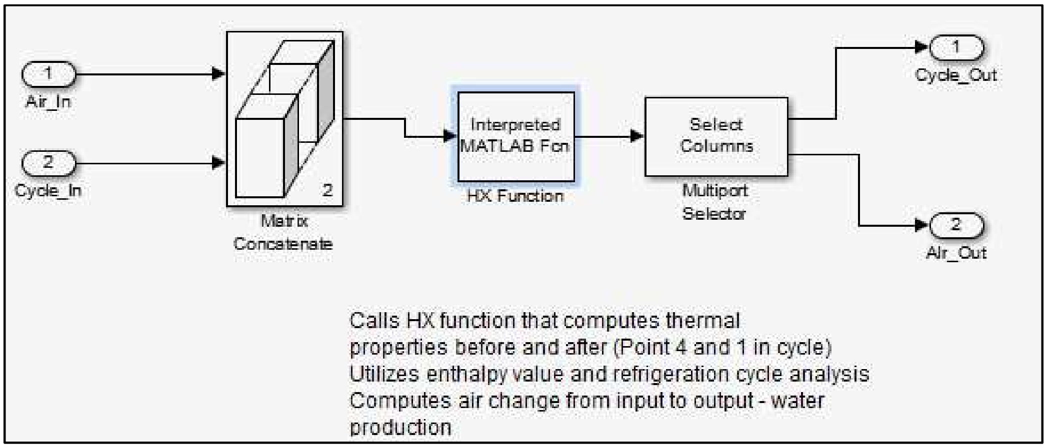
```
    end

    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
end
%+++++++++++++++++++++++++++++++++++++++
%CASE of Relative Humidity and Enthalpy
if (~isempty(phi_in) && ~isempty(h_in))
    phi=phi_in;h=h_in;

    % Tdb calculation from phi and h
    options=optimset('LargeScale','off','Display','off');
    [y,val,exitflag]=fsolve(@Iteration_function_2, 20,options);Tdb
=y(1);
    if exitflag<1
        disp('Iteration error')
    end

    % w calculation from Tdb and phi
    Pws=Saturation_pressure(Tdb);
    Pw=(phi/100)*Pws;
%     Pw=phi*Pws;
    w=0.621945*Pw/(P-Pw);

    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
end
%+++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
+++++
%CASE of Dry Bulb and Wet Bulb Temperature
if (~isempty(Tdb_in) && ~isempty(Twb_in))
    Tdb=Tdb_in;Twb=Twb_in;

    % w calculation from Tdb and Twb
    Pws=Saturation_pressure(Tdb);
    Pwsasterik=Saturation_pressure(Twb);
    ws=0.621945*Pwsasterik/(P-Pwsasterik);
    w= ((hlg-2.326e3*Twb)*ws-c_air*(Tdb-Twb))/(hlg+cw*Tdb-
4.186e3*Twb);

    % phi calculation from Tdb and w
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    phi=Pw/Pws*100;
%     phi=Pw/Pws;
    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
```

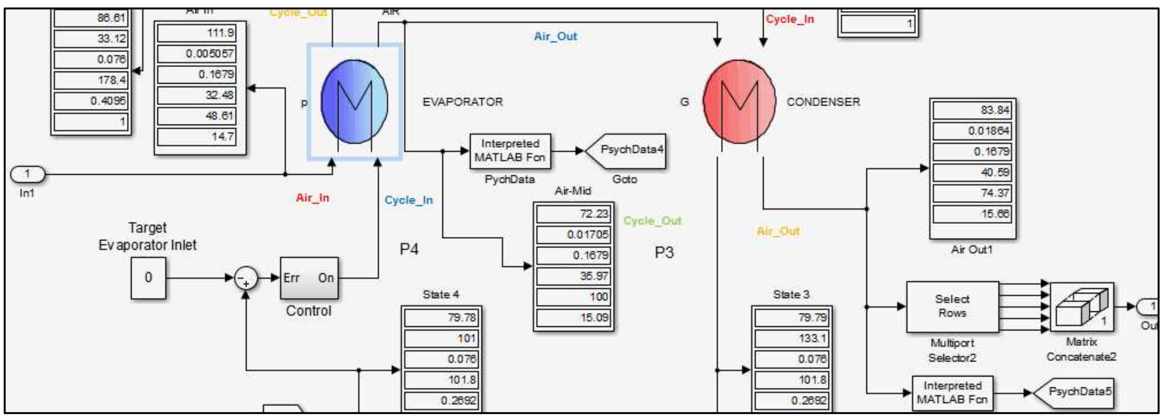
```
end
%++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
+++++
%CASE of Humidity Ratio and Wet Bulb Temperature
if (~isempty(w_in) && ~isempty(Twb_in))
    w=w_in;Twb=Twb_in;

    % Tdb calculation from Twb and w
    Pwsasterik=Saturation_pressure(Twb);
    ws=0.621945*Pwsasterik/(P-Pwsasterik);
    options=optimset('LargeScale','off','Display','off');
    [y,val,exitflag]=fsolve(@Iteration_function_4, Twb,options);Tdb
=y(1);
    if exitflag<1
        disp('Iteration error')
    end

    % phi calculation from Tdb and w
    Pws=Saturation_pressure(Tdb);
    Pw=w*P/(0.621945+w); %partial pressure of water wapor
    phi=Pw/Pws*100;
%     phi=Pw/Pws;
    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
end
%++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
+++
%CASE of Relative Humidity and Wet Bulb Temperature
if (~isempty(phi_in) && ~isempty(Twb_in))
    phi=phi_in;Twb=Twb_in;

    % Tdb calculation from phi and Twb
    Pwsasterik=Saturation_pressure(Twb);
    ws=0.621945*Pwsasterik/(P-Pwsasterik);
    options=optimset('LargeScale','off','Display','off');
    [y,val,exitflag]=fsolve(@Iteration_function_5, Twb,options);Tdb
=y(1);
    if exitflag<1
        disp('Iteration error')
    end

    % w calculation from Tdb and phi
    Pws=Saturation_pressure(Tdb);
    Pw=phi/100*Pws;
%     Pw=phi*Pws;
    w=0.621945*Pw/(P-Pw);

    % h calculation from Tdb and w
    h=c_air*Tdb+w*(hlg+cw*Tdb);

    % v calculation from Tdb and w
    v=0.370486*(Tdb+460)*(1+1.607858*w)/P;
```

```
end

% dew point calculation from w Eq (39 and 40)
Pw=(P*w)/(0.621945+w); % water vapor partial pressure in psi
alpha=log(Pw);
if Tdb < 32
    Tdp = 90.12 + 26.142*alpha + 0.8927*(alpha^2);
else % valid for Tdp between 32F and 200F
    C1= 100.45; C2 = 33.193; C3 = 2.319; C4 = 0.17074; C5 = 1.2063;
    Tdp = C1 + C2*alpha + C3*(alpha^2)+ C4*(alpha^3)+ C5*(Pw^0.1984);
end
%+++++++++++++++++++++++++++++++++++++++
if nargout>6 && isempty(Twb_in)
% Note: this Twb calc. equations are good for patm=101325 Pa only.
if abs(Tdb - Tdp) < .001, Twb=Tdb;return;end
options=optimset('LargeScale','off','Display','off');
[y,val,exitflag]=fsolve(@Iteration_function_3, Tdb,options);Twb=y(1);
if Twb > Tdb,Twb=Tdb;end
if Twb < Tdp,Twb=Tdp;end
end

function [Pws] = Saturation_pressure(Tdb) %saturated water vapor
pressure ASHRAE 2013 fundamentals eq. 6
    if Tdb > 32
        Tf=F2R(Tdb);%F to R
        C1=-1.0440397E4; C2=-1.129465E1; C3=-2.7022355E-2;
C4=1.289036E-5;
        C5=-2.4780681E-9; C6=6.545967;
        %ASHRAE inch-pound Eq.(6)
        Pws = exp(C1/Tf + C2 + C3*Tf + C4*(Tf^2) + C5*(Tf^3) +
C6*log(Tf)); %psia
    else
        Tf=F2R(Tdb);%F to R
        C7=-1.0214165E4; C8=-4.8932428E0; C9=-5.3765794E-3;
C10=1.9202377E-7;
        C11=3.5575832E-10; C12=-9.0344688E-14; C13= 4.1635019E0;
        %ASHRAE inch-pound Eq.(5)
        Pws = exp(C7/Tf + C8 + C9*Tf + C10*(Tf^2) + C11*(Tf^3) +
C12*(Tf^4)+ C13*log(Tf)); %psia
    end
  end

    function result = Iteration_function_1(y) %calc Tdb from phi and w
        Tdb_as=y(1);
        Pws=Saturation_pressure(Tdb_as);
%         phi_as=Pw/Pws*100; %ASHRAE 2013 fundamentals eq. 24
        phi_as=Pw/Pws*100;
        % equation to satisfy
        result=phi_as-phi;
    end
    function result = Iteration_function_2(y) %calc Tdb from phi and h
        Tdb_as=y(1);
        % w calculation from Tdb and phi
        Pws=Saturation_pressure(Tdb_as);
%         Pw=phi/100*Pws;
        Pw=(phi/100)*Pws;
```

```
        w_as=0.621945*Pw/(P-Pw); %ASHRAE 2013 fundamentals eq. 22
        % h calculation from Tdb and w
        h_as=c_air*Tdb_as+w_as*(hlg+cw*Tdb_as);
        % equation to satisfy
        result=h_as-h;
    end
    function result = Iteration_function_3(y) %calc Twb from Tdb and w
using ASHRAE 2013 fundamentals eq. 35
        Twb_as=y(1);
        Pws_as=Saturation_pressure(Twb_as);
        ws=0.621945*Pws_as/(P-Pws_as);
        w_as= ((hlg-2.326e3*Twb_as)*ws-c_air*(Tdb-
Twb_as))/(hlg+cw*Tdb-4.186e3*Twb_as);
        result=(w-w_as);
    end
    function result = Iteration_function_4(y) %calc Tdb from Twb and w
Tdp using ASHRAE 2013 fundamentals eq. 35
        Tdb_as=y(1);
        w_as= ((hlg-2.326e3*Twb)*ws-c_air*(Tdb_as-
Twb))/(hlg+cw*Tdb_as-4.186e3*Twb);
        result=(w-w_as);
    end
    function result = Iteration_function_5(y) %calc Tdb from Twb and
phi Tdp using ASHRAE 2013 fundamentals eq. 35
        Tdb_as=y(1);
        w_as= ((hlg-2.326e3*Twb)*ws-c_air*(Tdb_as-
Twb))/(hlg+cw*Tdb_as-4.186e3*Twb);
        Pw_as=w_as*P/(0.621945+w_as); %partial pressure of water wapor
        Pws_as=Saturation_pressure(Tdb_as);
        phi_as=Pw_as/Pws_as*100;
%         phi_as=Pw_as*10/Pws_as;
        result=phi-phi_as;
    end

    function result = Iteration_function_6(y) %calc Pw from Tdb and
Tdp using ASHRAE 2013 fundamentals eq. 39
        Pw_as=y(1);
        Tdp_as=
6.54+14.526*log(Pw_as)+0.7389*(log(Pw_as))^2+0.09486*(log(Pw_as))^3+0.
4569*(Pw_as^0.1984); % valid for Tdp between 0 C and 93 C
        result=Tdp-Tdp_as;
    end
end
```

Heat Exchanger Unit Code (Evaporator and Condenser)






```
function y = HeatExchanger(data)
%function y = HX_Analysis(data)
%General Heat Exchanger Function that takes in
%   Refrigerant and Air properties to mix
% Computations utilize the UA method of HX analysis

%Read in the cycle and air data
cycle_data_in = data(:,2);
air_data_in = data(:,1);
%Properties A and B refer to the refrigerant
%Properties 1 and 2 refer to the fluid (air)

%% ===== defining variables =====
Refrigerant = 'R134a';
Fluid = 'AIR.PPF';
offset = 10; %offset between Refprop and Psych

%Define cycle data as Temp, Pressure, m_dot, enthalpy, entropy, quality
%************************
% State Air intitial Properties
T_1 = air_data_in(1);
w_1 = air_data_in(2);
m_dot_air = air_data_in(3);
```

```
h_1 = air_data_in(4);
RH_1 = air_data_in(5);
P_1 = air_data_in(6);

%Use given air properties and thermal database to get other properties
[S] = refpropm('PS','T',F2K(T_1),'P',P_1, Fluid);
s_1 = S * J2Btu(1)/kg2lbm(1);
C = refpropm('C','T',F2K(T_1),'P',P_1, Fluid);
c_p_air = C * J2Btu(1)/(kg2lbm(1)*K2R(1)); % KJ/kg-K

%***********************
% State Ref intitial Properties

if sum(cycle_data_in)> 25
    %Read in refridgerant properties
    T_A = cycle_data_in(1);
    P_A = cycle_data_in(2);
    m_dot_ref = cycle_data_in(3);
    h_A = cycle_data_in(4);
    s_A = cycle_data_in(5);
    x_A = cycle_data_in(6);
else

    %Assume refrigerant properties
    T_A = cycle_data_in(1);
    m_dot_ref = cycle_data_in(3);
    x_A = cycle_data_in(6);
        %Find the other properties at State A
    [P H S] = refpropm('PHS','T',F2K(T_A),'Q',x_A, Refrigerant);
    P_A = (P)*kPa2psi(1);
    h_A = (H)*(J2Btu(1))/(kg2lbm(1));
    s_A = (S)*(J2Btu(1))/(kg2lbm(1)*K2R(1));
end
C = refpropm('C','T',F2K(T_A),'Q',0, Refrigerant);
c_p_ref = (C) *(J2Btu(1))/(kg2lbm(1)*K2R(1));

%Find the overall heat transfer coefficient and area
%Assume the pipe area = pi*(d^2)/4
%Assume copper pipe for thermal conductivity and fouling
%Assume the use of fins
%Shell box area =l*l
d_tube = 0.25;%in
d_inner = 0.2; %in
l_shell = 12; %in
N_fins = 40; %number of fins in the exchanger
N_tube = 50; %number of tube passes in shell
t_fin = mm2in(1); %Fin thickness given in mm, convert to in

[U,A] = Overall_U(m_dot_air,m_dot_ref,d_tube,...

d_inner,l_shell,N_fins,N_tube,t_fin,T_A,T_1,P_A,P_1);
%------------------------------------------------
% HE_Type defines the type of heat exchanger: (see reference)
%    'Parallel Flow'
%    'Counter Flow'
```

```
%   'One Shell Pass'
%   'N Shell Pass'
%   'Cross Both Unmixed'
%   'Cross Cmax Mixed'
%   'Cross Cmin Mixed'
HX_Type = 'One Shell Pass';
T_hot_in = max(T_1,T_A);

%Assign hot and cold values for HX analysis
if T_hot_in == T_1
    m_dot_hot = m_dot_air;
    c_p_hot = c_p_air;
    T_cold_in = T_A;
    m_dot_cold = m_dot_ref;
    c_p_cold = c_p_ref;
else
    T_hot_in = T_A;
    m_dot_hot = m_dot_ref;
    c_p_hot = c_p_ref;
    T_cold_in = T_1;
    m_dot_cold = m_dot_air;
    c_p_cold = c_p_air;
end


[T_hot_out,T_cold_out]=HX_Analysis(m_dot_hot,c_p_hot,T_hot_in,...
    m_dot_cold,c_p_cold,T_cold_in,U,A,HX_Type,N_tube);

%Assign exit vales based on the input temperatures
if T_hot_in == T_1
    T_2 = T_hot_out;
    T_B = T_cold_out;
else
    T_B = T_hot_out;
    T_2 = T_cold_out;
end

Q_air = m_dot_air * c_p_air *(T_1 - T_2);
Q_ref = m_dot_ref * c_p_ref *(T_A - T_B);

LMTD = ((T_1-T_B)-(T_2-T_A))/log((T_1-T_B)/(T_2-T_A));
Q_exchange = U * A * LMTD;

%************************
%State B Ref Properties
% P_B = P_A;
    %Find exit refrigerant properties based on quality
if x_A >= 1.0 %Condenser cooling down
    x_B = 0.0;
else          %Evaporator heating up
    x_B = 1.0;
end
    %Exit exchanger properites
[H S] = refpropm('HS','T', F2K(T_B), 'Q', x_B, Refrigerant);
s_B = (S)*(J2Btu(1))/(kg2lbm(1)*K2R(1));
```

```
h_B = (H)*(J2Btu(1))/(kg2lbm(1));
P = refpropm('P','T',F2K(T_B),'Q',x_B,Refrigerant);
P_B = kPa2psi(P);

%------------------------------------------------
%Determine heat exchange
%Assuming atmospheric pressure, find density and entropy of air
%   at entry
Patm = 14.686;

%Exit air properties
%Find the exit conditions based on evaporator or condenser using exit
%quality of refrigerant
if x_B == 1.0
    h_2 = -(m_dot_ref/m_dot_air)*(h_B - h_A) + h_1;
    RH_2 = 100;
    %Find the humidity ratio and relative humidity
    [~, w_2, RH_2,h_2,Tdp ,v ,Twb] = Psychrometricsnew ('tdb',T_2,
'phi', RH_2,'p',Patm);
else
    h_2 = (m_dot_ref/m_dot_air)*(h_A - h_B)+ h_1;
    [~, w_2, RH_2,h_2,Tdp ,v ,Twb] = Psychrometricsnew ('tdb',T_2, 'h',
h_2,'p',Patm);
    %Cut the RH at 100 percent
    if RH_2 > 100
        RH_2 = RH_2/offset;
    end
end
%Determine the pressure at the exit
P_2 = P_1;

    %-***************************************
%Data packet
dataout_ref = [ T_B
                P_B
                m_dot_ref
                h_B
                s_B
                x_B];
dataout_air = [ T_2
                w_2
                m_dot_air
                h_2
                RH_2
                P_2   ];
%Collect and pass data packet to next model
dataout = [dataout_ref,dataout_air];
y = dataout;
end

%*********************************************************
function [ U, A] = Overall_U(m_dot_air,m_dot_ref,d_tube,...

d_inner,l_shell,N_fins,N_tube,t_fin,T_A,T_1,P_A,P_1)
%This model aims to determine the overall heat transfer coefficient
%   of the heat exchanger using both the LMTD and effectivness NTU
```

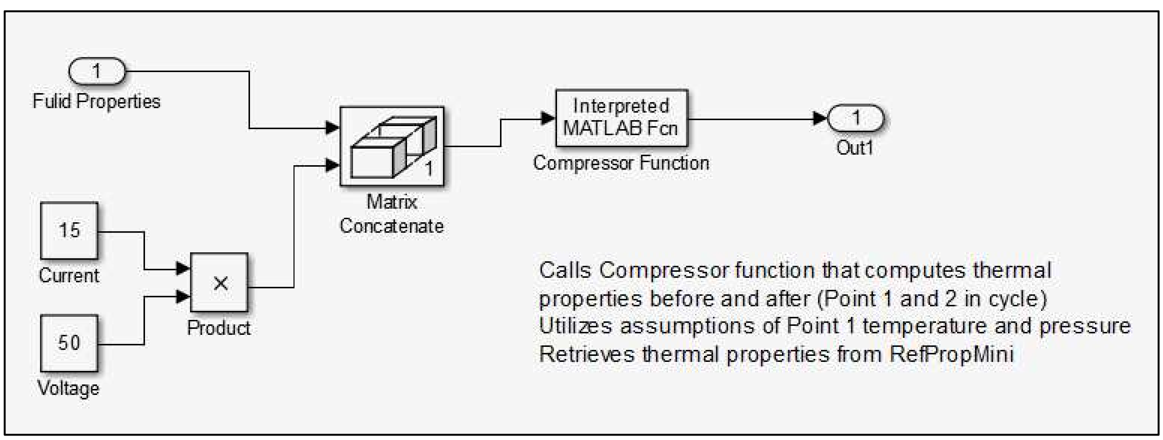
```
%   methods

%System parameters
Refrigerant = 'R134a';
Fluid = 'AIR.PPF';
l_fin = l_shell-2; %Fin lenght as long as shell
K_copper = 400 * W2Btuph(1)/(m2in(1)*K2R(1)); %Btu/(h-in-R)
l_tube = l_shell - 1;

%------------------------------------------------
%Find the area of the shell and base tube
A_shell = l_shell * l_shell; %in^2
A_base = pi*(d_inner)^2/4; %in^2
A_tube = pi*(d_tube)^2/4;
SA_base = (2*pi*(d_tube/2)*N_tube*l_tube)+(2*pi*(d_tube/2)^2);
A_fin = t_fin*l_fin; %in^2
perimeter = t_fin*(2) + l_fin*2; %fin perimeter

%Find the heat transfer coeff for ref and fluid
[C u L Pr] = refpropm('CVK^','T',F2K(T_A),'P',P_A, Refrigerant);
cp_ref = C * (J2Btu(1))/(kg2lbm(1)*K2R(1));%Btu/(lbm-R)
u_ref = u * Pa2psi(1); %psi*s
K_ref = L * W2Btuph(1)/(m2in(1)*K2R(1)); %Btu/(h-in-R)
Pr_ref = Pr; %Prandlt number unitless

Re_ref = (4* m_dot_ref/(32.2*12))/(pi* u_ref * d_inner); %Reynolds
number
%Kaminski Table 12-1
%   Nu formulas for a Round conduit
if Re_ref < 40000
    %Laminar flow check entrance effects
    Nu_ref = 0.193*Re_ref^(0.618)*Pr_ref^(1/3);
    L_char = d_inner; %in
else
    Nu_ref = 0.027*Re_ref^(0.805)*Pr_ref^(1/3);
    L_char = d_inner; %in
end
%Use found values to get air convection coefficient
convection_h_ref = (K_ref * Nu_ref)/L_char; %Btu/(h*in^2*F)

%Find the heat transfer coeff for ref and fluid
[C u L Pr] = refpropm('CVK^','T',F2K(T_1),'P',P_1, Fluid);
cp_air = C * (J2Btu(1))/(kg2lbm(1)*K2R(1));%Btu/(lbm-R)
u_air = u * Pa2psi(1); %psi*s
K_air = L * W2Btuph(1)/(m2in(1)*K2R(1)); %Btu/(h-in-R)
Pr_air = Pr; %unitless

Re_air = (4* m_dot_air/(32.2*12))/(pi* u_air * l_shell); %Reynolds
number
 %Kaminski Table 12-1
 %  Nu formulas for a Square conduit
if Re_air < 8000
    %Laminar flow check entrance effects
    Nu_air = 0.177*Re_air^(0.699)*Pr_air^(1/3);
%     L_char = l_shell; %in
```

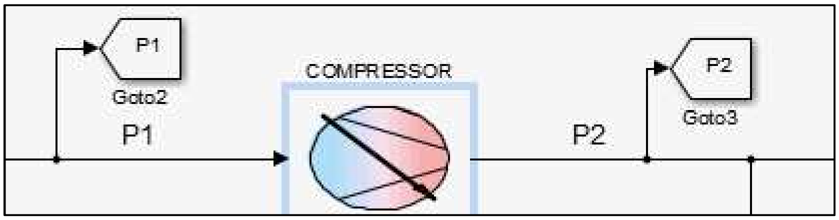
```
    L_char = A_shell/(N_fins*d_tube); %in
else
    Nu_air = 0.102*Re_air^(0.675)*Pr_air^(1/3);
    if N_fins == 0;
        N_fins = 1;
    end
    L_char = A_shell/(N_fins*d_tube); %in
end
%Use found values to get air convection coefficient
convection_h_air = (K_air * Nu_air)/L_char; %Btu/(h*in^2*R)

%Determine fin properties
%Get the fin efficiencies
if N_fins == 1
    n_f = 1;
    n_o = 1;
else
    m = sqrt(convection_h_air*perimeter/(K_copper*A_fin));
    %Fin convection from tip
    a = (sinh(m*l_fin)+(convection_h_air/(m*K_copper))*cosh(m*l_fin));
    b = (cosh(m*l_fin)+(convection_h_air/(m*K_copper))*sinh(m*l_fin));
    n_f = (a/b)*(1/(m*l_fin)); %fin efficiency
    n_o = 1 - (N_fins*A_fin)/(SA_base+N_fins*A_fin)*(1 - n_f); %Overall
fin efficiency
end
%Thermal theory of UA model
    %R_tot = R_conv,i + R_foul,i + R_wall + R_foul,o + R_conv,o
    %UA = U_i*A_i = U_o*A_o
R_fouling = 0.000175; %m^2*K/W
R_f = R_fouling * (m22in2(1)*K2R(1))/(W2Btuph(1));

R_conv_i = 1/(n_o*convection_h_ref);
R_foul_i = (R_f/n_o);
R_wall = A_base*log(d_tube/d_inner)/(2*pi*K_copper*l_tube);
R_foul_o = (R_f/n_o)*(A_base/A_tube);
R_conv_o = 1/(n_o*convection_h_air)*(A_base/A_tube);
    %Determine U based on A_i
U = (R_conv_i + R_foul_i + R_wall + R_foul_o + R_conv_o)^(-1);
A = A_base;
end

%**********************************************************
function
[T_hot_out,T_cold_out]=HX_Analysis(m_dot_hot,c_p_hot,T_hot_in,m_dot_col
d,c_p_cold,T_cold_in,U,A,HE_Type,N_tube);
%
[T_hot_out,T_cold_out]=HeatExchanger(c_p_hot,m_dot_hot,T_hot_in,c_p_col
d,m_dot_cold,T_cold_in,U,A,HE_Type);
% This function calculates the outlet temperatures of a heat exchanger
% using Epsilon-NTU method. This function uses effectiveness.m as a
% function and should have access to that function.
%
% The inputs are as follows:
% Hot Flow: c_p_hot, m_dot_hot, T_hot_in.
% Cold Flow: c_p_cold, m_dot_cold, T_cold_in.
% Heat exchanger design parameters: U,A, HE_Type.
```

```
%
%Modified by Daniel Appiah-Mensah
% Reference:
% Frank P. Incropera, Introduction to heat transfer. New York:Wiley,
1985, Section 11.4.
% Programmer: Seyyed Ali Hedayat Mofidi (seyyed4li@yahoo.com)

C_hot = m_dot_hot*c_p_hot;
C_cold = m_dot_cold*c_p_cold;
C_min = min(C_hot,C_cold); % finds the flow with lower heat capacity
and higher temperature change.
C_max = max(C_hot,C_cold); % finds the flow with higher heat capacity
and lower temperature change.
C_r=C_min/C_max;
NTU = U*A/C_min;

N_shell = 1;
epsilon = effectiveness (NTU,C_r,HE_Type,N_tube,N_shell);

Q_max = C_min*(T_hot_in-T_cold_in);
Q = epsilon * Q_max ;
T_hot_out = T_hot_in - Q/C_hot ;
T_cold_out = T_cold_in + Q/C_cold ;
end

%**********************************************************
function epsilon=effectiveness(NTU,C_r,HE_Type,N_tube,N_shell)
% This function calculates the effectiveness of a heat exchanger.
% NTU is the number of transfer units of the Heat Exchanger:
%
% Regardless of heat exchanger type, if C_r=0, either hot flow is
% condensing ( means no change in T_hot) or cold flow is evaporating (
no
% change in T_cold), therefore C_max =inf its temperature does not
change.
%
% Reference:
% Frank P. Incropera, Introduction to heat transfer. New York:Wiley,
1985, Section 11.4.
% Programmer: Seyyed Ali Hedayat Mofidi (seyyed4li@yahoo.com)
N = N_tube;
M = N_shell;

if nargin == 3
    N=1;
end
%% ===== Calculating effectiveness =====
% Special case of boiling or condensing:
if C_r == 0
    epsilon = 1-exp(-NTU);
    return;
end

switch HE_Type
    case 'Parallel Flow'
```

```
        epsilon = (1-exp(-NTU*(1+C_r)))/(1+C_r);
    case 'Counter Flow'
        if C_r==1
            epsilon = NTU/(1+NTU);
        else
            epsilon = (1-exp(-NTU*(1-C_r)))/(1-C_r*exp(-NTU*(1-C_r)));
        end
    case 'One Shell Pass'
        NTUN = NTU*N;
        epsilon = 2/(1+C_r+sqrt(1+C_r^2)*(1+exp(-
NTUN*sqrt(1+C_r^2)))/(1-exp(-NTUN*sqrt(1+C_r^2))));
    case 'N Shell Pass'
        NTUN = NTU/N;
        epsilon1 = 2/(1+C_r+sqrt(1+C_r^2)*(1+exp(-
NTUN*sqrt(1+C_r^2)))/(1-exp(-NTUN*sqrt(1+C_r^2))));
        epsilon = (((1-epsilon1*C_r)/(1-epsilon1))^M-1) / (((1-
epsilon1*C_r)/(1-epsilon1))^M-C_r);
    case 'Cross Both Unmixed'
        epsilon = 1-exp(1/C_r * NTU^0.22 * (exp(-C_r*NTU^0.78)-1));
    case 'Cross Cmax Mixed'
        epsilon = 1/C_r*(1-exp(-C_r*(1-exp(-NTU))));
    case 'Cross Cmin Mixed'
        epsilon = 1 - epx(-1/C_r*(1-exp(-C_r*NTU)));
    otherwise % the type is not in the list, therefore we assume
there's no heat exchanger.
        epsilon = 0;
end
end
```

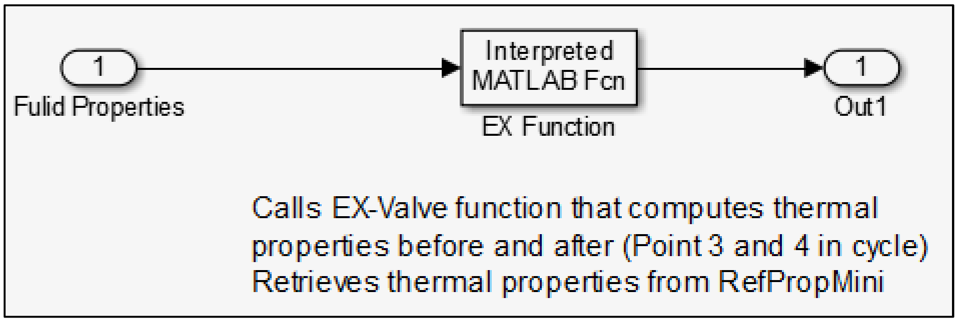
Compressor Unit Code

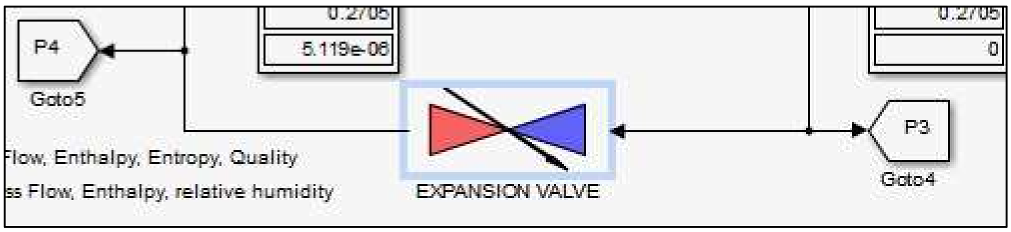





```
function y = Compressor( input )
%Compressor function used to estimate the exit thermal properties of
%the refrigerant

%State 1 of fluid
%**********************************************************
%Assumptions of Temp and Pressure
   %Define cycle data as Temp, Pressure, m_dot, enthalpy, entropy,
qualixy

Compression_ratio = 100;
%     Work = Current*Voltage; %Watts
Work = input(7);
Work_BTU = W2Btuph(Work)/hr2s(1);
Refrigerant = 'R134a';

%FROM A LOOKUP TABLE OR PROGRAM
%*******************'
%State 1 =Saturated Gas
T_1 = input(1);
P_1 = input(2);
m_dot = input(3);
h_1 = input(4);
s_1 = input(5);
x_1 = input(6);

%******************
%State 2 = Superheated Gas
s_2 = s_1;
P_2 = P_1 + Compression_ratio;
[H T] = refpropm('HT','P',psi2kPa(P_2),'Q',x_1,Refrigerant);
h_2s = (H)*(J2Btu(1))/(kg2lbm(1));
T_2 = K2F(T);
x_2 = x_1;

%-------------------------------------------------
%Determine the work done
efficiency = 0.85;
W_actual = efficiency * Work_BTU;
h_2 = W_actual/(m_dot) + h_1;
%Look up or interpolate s_actual based on h_2

%Parcel data output
%   s_2 should be s_2actual from a lookup table or interpolated
dataout = [T_2; P_2; m_dot; h_2; s_2; x_2];
y = dataout;
```

Expansion Valve Code

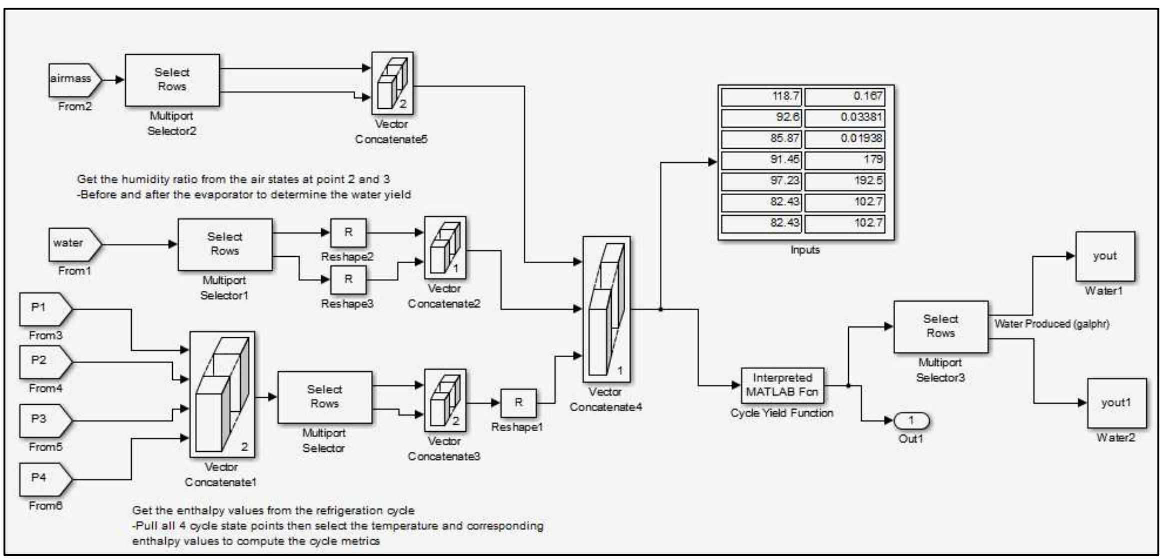


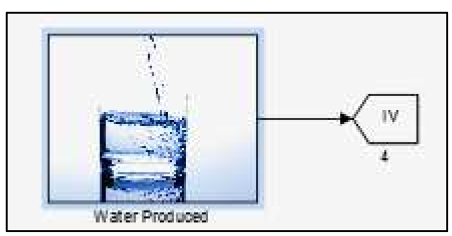



```
function y = ExValve(data)
%Function to model the expansion valve component:
%data input is the refrigerant properties from state 3
Refrigerant = 'R134a';

%STATE 3 properties
%--------------------------------------------------------
    % T_A = data(1);
T_3 = data(1);
P_3 = data(2);
m_dot = data(3);
h_3 = data(4);
s_3 = data(5);
x_3 = data(6);

%STATE 4 properties
%--------------------------------------------------------
h_4 = h_3;

H_4 = h_4* Btu2J(1)/lbm2kg(1);
[T] = refpropm('T','P',psi2kPa(150),'H',H_4, Refrigerant);
T_4 = K2F(T); %F

[H_f] = refpropm('H','T',F2K(T_4),'Q',0, Refrigerant);
[H_v] = refpropm('H','T',F2K(T_4),'Q',1, Refrigerant);
h_f = (H_f)*(J2kJ(1))*(kJ2Btu(1))/(kg2lbm(1));
h_v = (H_v)*(J2kJ(1))*(kJ2Btu(1))/(kg2lbm(1));

x_4 = (h_4 - h_f)/(h_v - h_f); %percent
%Hard stop quality at zero
if x_4 < 0
    x_4 = 0;
end
```

```
P = refpropm('P','T',T,'Q',x_4,Refrigerant);
P_4 = kPa2psi(P);

[S] = refpropm('S','P', psi2kPa(P_4), 'Q', x_4, Refrigerant);
s_4 = (S)*(J2kJ(1))*(kJ2Btu(1))/(kg2lbm(1)*K2R(1));

%---------------------------------------------------
dataout = [ T_4
            P_4
            m_dot
            h_4
            s_4
            x_4 ] ;

y = dataout;
```

Cycle Yield Code






```
function y = CycleYield(input)
%This function computes the volume of water produced per
% unit time (gal/hr) to give an estimate of the system yield.
%System metrics are
%   -water yield(volume/time)
%   -cycle COP
%   -cycle efficiency
```

```
%   -back work ratio (BWR)
%   -carnot efficiency

%Get data points from input
Temp = input(:,1);
m_dot = input(1,2);
w1 = input(2,2);
w2 = input(3,2);
h1 = input(4,2);
h2 = input(5,2);
h3 = input(6,2);
h4 = input(7,2);
Time = input(8,1);

%**************************************************
%WATER YIELD
P = 14.686; %psi
%Set the temperature of the water as the average between the
evaporator
T = (Temp(1)+Temp(3))/2; %F

%Find the mass rate from the air mass flow and humidity ratios
water_massRate = m_dot*(w1 - w2); %lbm/s*lbw/lbm = lbmw/s

%Find the density of water to convert mass rate to volume rate
[D] = refpropm('D', 'T', F2K(T), 'P', psi2kPa(P),'water');
water_density = D * kg2lbm(1)/m32ft3(1); %lbmw/ft3
    %Get the volume rate and convert to gallons
water_Volume_rate = water_massRate / water_density; %ft3/s
water_gallonsPhrx = water_Volume_rate * ft32gal(1)/s2hr(1);

if Time == 0
    normalize = water_gallonsPhrx;
     s1.a = normalize;
    save('waterOffset.mat','-struct','s1')
end

A = load('waterOffset','-mat');
 offset = A.a;

water_gallonsPhr = water_gallonsPhrx - offset;
    %Determine water per day on 1/2 day light for solar
kW = 0.75; %Recall current * voltage = 750W
water_gallonsPkWh = water_gallonsPhr/kW;

%**************************************************
Work_in = (h2 - h1);
Heat_in = (h1 - h4); %Evaporator
Heat_out =(h2 - h3); %Condenser

%CYCLE Efficiency
    %Intro to Thermal book (Kaminski)
%n is the energy in over the heat gain (evap)
n_cycle = (Work_in/Heat_out) * 100;
```

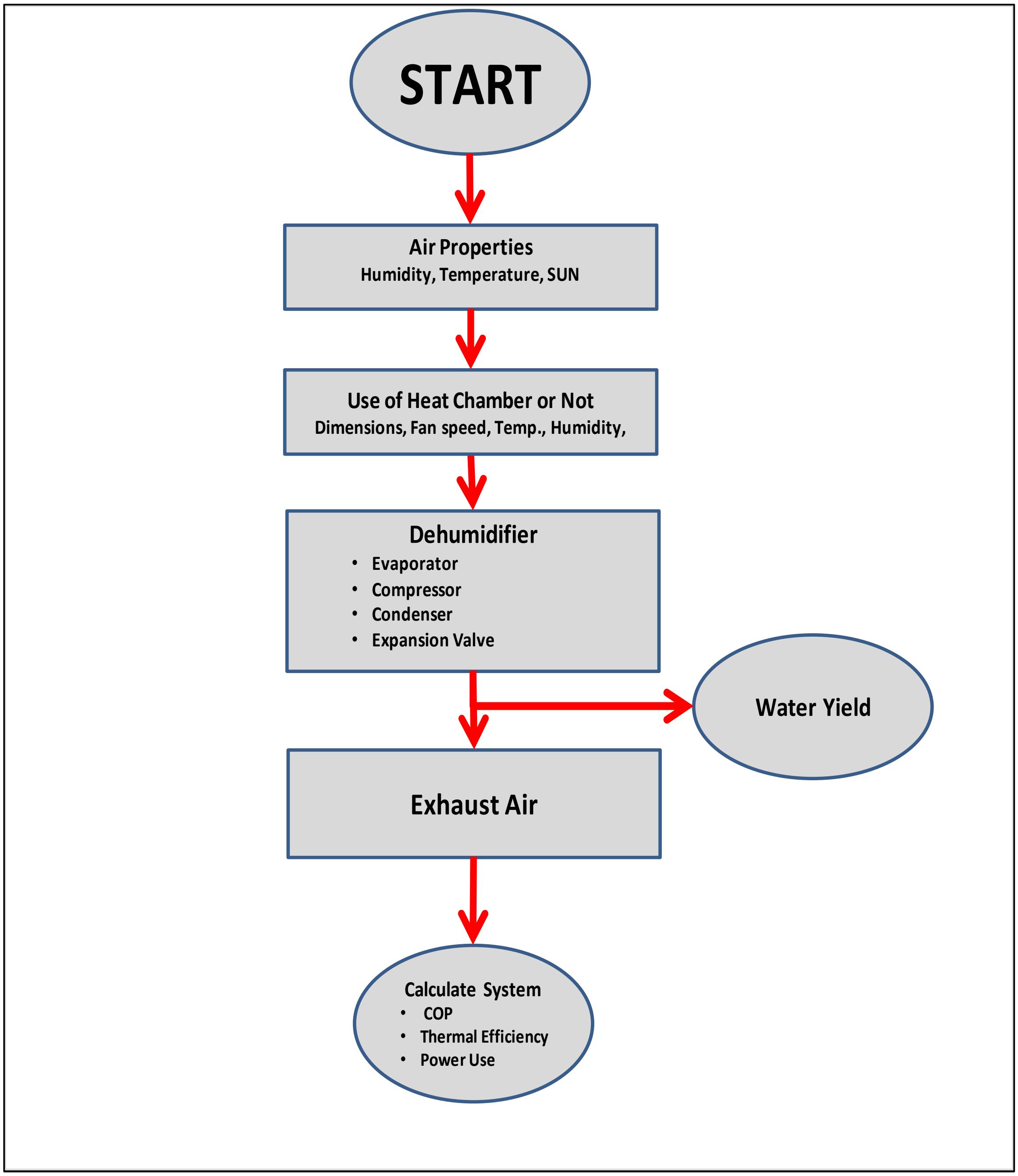
```
%*************************************************
%CYCLE COP
    %Intro to Thermal book (Kaminski) Eq (8-5)
COP_ref = Heat_in/Work_in;

%*************************************************
%CARNOT Efficiency
    %Intro to Thermal book (Kaminski) Eq (7-6)
%Reversible cycle
T_L = F2R(Temp(7));
T_H = F2R(Temp(5));
n_carnot = (1 - (T_L/T_H))*100;

%-----------------------------------------------------------
    %Data outputs
dataout = [ water_gallonsPhr
            water_gallonsPkWh
            n_cycle
            n_carnot
            COP_ref ];

y = dataout;
```

B. System Charts

I/O Integration Design

Component	Inputs	Output
Inlet Air	*User defined*	*Air:* • Humidity • Temperature • Volume rate (mass rate)
Heat Chamber	*User defined:* • Dimensions (L X W X H) • Sun Energy • Fan speed • Polluted water mass rate *Air:* • Humidity • Temperature • Air quality • Volume rate (mass rate)	*Air:* • Humidity • Temperature • Air quality • Volume rate (mass rate)
Evaporator	*Refrigerant:* • Temp • Enthalpy • mass rate • quality *Air:* • Temp • Humidity • mass rate	*Refrigerant:* • Temp • Enthalpy • mass rate *Air:* • Temp • mass rate ***Water***
Compressor	*Refrigerant:* • Enthalpy • Temp • mass rate	*Refrigerant:* • Enthalpy • Temp • Mass rate
Condenser	*Refrigerant:* • Temp • Enthalpy • mass rate *Air:* • Temp • mass rate	*Refrigerant:* • Temp • Enthalpy • mass rate *Air:* • Temp • mass rate
Expansion Valve	*Refrigerant:* • Enthalpy	*Refrigerant:* • Enthalpy • Quality

System Flow Chart



C. Data, Plots, Figures

Sample Results Excel output

Time step = 1

Time (hr)	0	1	2	3	4	5	6	7	8	9	10	11	12
Solar Irradiance (Btu/h-ft^2)	0	82.04521	158.4992	224.1517	274.5286	306.1969	316.9983	306.1969	274.5286	224.1517	158.4992	82.04521	3.88E-14
Solar Iradiance (W/m2)	0	258.8531	500.0657	707.1997	866.1392	966.0528	1000.131	966.0528	866.1392	707.1997	500.0657	258.8531	1.22E-13
Atmoshperic Temp (F)	50	52.12349	57.92496	65.84992	73.77487	79.57635	81.69983	79.57635	73.77487	65.84992	57.92496	52.12349	50
Atmospheric Relative humidity (%)	67.96053	57.66226	35.28528	34.91463	22.42502	16.60014	14.98452	16.60014	22.42502	34.91463	35.28528	57.66226	67.96053
Atmospheric enthalpy (Btu/lbm)	17.59867	17.64969	17.78908	17.9795	18.16996	18.3094	18.36044	18.3094	18.16996	17.9795	17.78908	17.64969	17.59867
Chamber exit Temp (F)	84.47472	93.33527	106.894	122.6069	137.1857	147.4561	151.1523	147.4561	137.1857	122.6069	106.894	93.33527	84.47472
Chamber exit humidity (lbmw/lbma)	0.011845	0.013497	0.016836	0.02176	0.027572	0.032577	0.034598	0.032577	0.027572	0.02176	0.016836	0.013497	0.011845
Chamber exit mass rate (lbm/s)	0.145895	0.145411	0.14411	0.142378	0.140697	0.139497	0.139065	0.139497	0.140697	0.142378	0.14411	0.145411	0.145895
Chamber exit enthalpy (Btu/lbm)	32.11961	36.139	43.35407	52.99706	63.42678	71.8585	75.15402	71.8585	63.42678	52.99706	43.35407	36.139	32.11961
Chamber exit R.H (%)	42.70405	37.2295	31.54472	26.51405	22.88489	20.80811	20.14394	20.80811	22.88489	26.51405	31.54472	37.2295	42.70405
Chamber exit Pressure (psi)	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696	14.696
Dehumidifier exit Temp (F)	88.07241	91.52818	96.93353	103.3296	109.3743	113.6917	115.257	113.6917	109.3743	103.3296	96.93353	91.52818	88.07241
D_exit humidity (lbmw/lbma)	0.04862	0.049596	0.051431	0.053961	0.056719	0.058929	0.059783	0.058929	0.056719	0.053961	0.051431	0.049596	0.04862
D_exit mass rate (lbm/s)	0.145895	0.145411	0.14411	0.142378	0.140697	0.139497	0.139065	0.139497	0.140697	0.142378	0.14411	0.145411	0.145895
D_exit enthalpy (Btu/lbm)	74.62415	76.60412	80.04625	84.52699	89.18308	92.78397	94.15044	92.78397	89.18308	84.52699	80.04625	76.60412	74.62415
D_exit R.H (%)	16.18945	14.79527	12.9525	11.16839	97.91436	89.56715	86.80507	89.56715	97.91436	11.16839	12.9525	14.79527	16.18945
Water gallons per hr	0	0.033722	0.126292	0.270894	0.448415	0.607075	0.672604	0.607075	0.448415	0.270894	0.126292	0.033722	0
Water gallons per kWh	0	0.044963	0.168389	0.361192	0.597886	0.809433	0.896805	0.809433	0.597886	0.361192	0.168389	0.044963	0
Thermal cycle efficiency (%)	7.517936	7.569651	7.658007	7.773675	7.894796	7.988984	8.024832	7.988984	7.894796	7.773675	7.658007	7.569651	7.517936
Carnot cycle efficiency (%)	16.44317	16.85595	17.50127	18.26946	18.99755	19.51701	19.70496	19.51701	18.99755	18.26946	17.50127	16.85595	16.44317
Coeff. of Performance	12.30152	12.21065	12.05823	11.86393	11.66657	11.51724	11.46132	11.51724	11.66657	11.86393	12.05823	12.21065	12.30152
Air Cycle Temp 1 (F)	50	52.12349	57.92496	65.84992	73.77487	79.57635	81.69983	79.57635	73.77487	65.84992	57.92496	52.12349	50
Air Cycle Temp 2 (F)	84.47472	93.33527	106.894	122.6069	137.1857	147.4561	151.1523	147.4561	137.1857	122.6069	106.894	93.33527	84.47472
Air Cycle Temp 3 (F)	59.35134	63.36174	70.37037	78.49855	85.98668	91.26926	93.17751	91.26926	85.98668	78.49855	70.37037	63.36174	59.35134
Air Cycle Temp 4 (F)	48.65779	52.529	58.3213	64.89821	70.88324	75.03483	76.51614	75.03483	70.88324	64.89821	58.3213	52.529	48.65779
Air Cycle Temp 5 (F)	88.07241	91.52818	96.93353	103.3296	109.3743	113.6917	115.257	113.6917	109.3743	103.3296	96.93353	91.52818	88.07241
Air Cycle Humidity 1 (lbmw/lbma)	0.007628	0.008261	0.010235	0.013615	0.017969	0.021916	0.023547	0.021916	0.017969	0.013615	0.010235	0.008261	0.007628
Air Cycle Humidity 2 (lbmw/lbma)	0.010781	0.01246	0.015965	0.021128	0.027183	0.032368	0.034456	0.032368	0.027183	0.021128	0.015965	0.01246	0.010781
Air Cycle Humidity 3 (lbmw/lbma)	0.010781	0.01246	0.015965	0.021128	0.027183	0.032368	0.034456	0.032368	0.027183	0.021128	0.015965	0.01246	0.010781
Air Cycle Humidity 4 (lbmw/lbma)	0.007251	0.008387	0.010384	0.013162	0.016253	0.018767	0.019746	0.018767	0.016253	0.013162	0.010384	0.008387	0.007251
Air Cycle Humidity 5 (lbmw/lbma)	0.007251	0.008387	0.010384	0.013162	0.016253	0.018767	0.019746	0.018767	0.016253	0.013162	0.010384	0.008387	0.007251

Refrigeration Model Verification

Verification conducted with textbook examples from ‘Introduction to Thermal Fluids Engineering’ by Deborah Kaminski and Michael Jensen [18]

Example 8-1: Ideal Vapor compression refrigeration cycle

Parameter	Simulation	Expected	Percent Difference (%)
Work Rate (Hp)	3.251	3.27	0.58
Heat Rate Out$\left(\frac{Btu}{min}\right)$	743.4	740	0.46
COP cooling	4.352	4.33	0.51

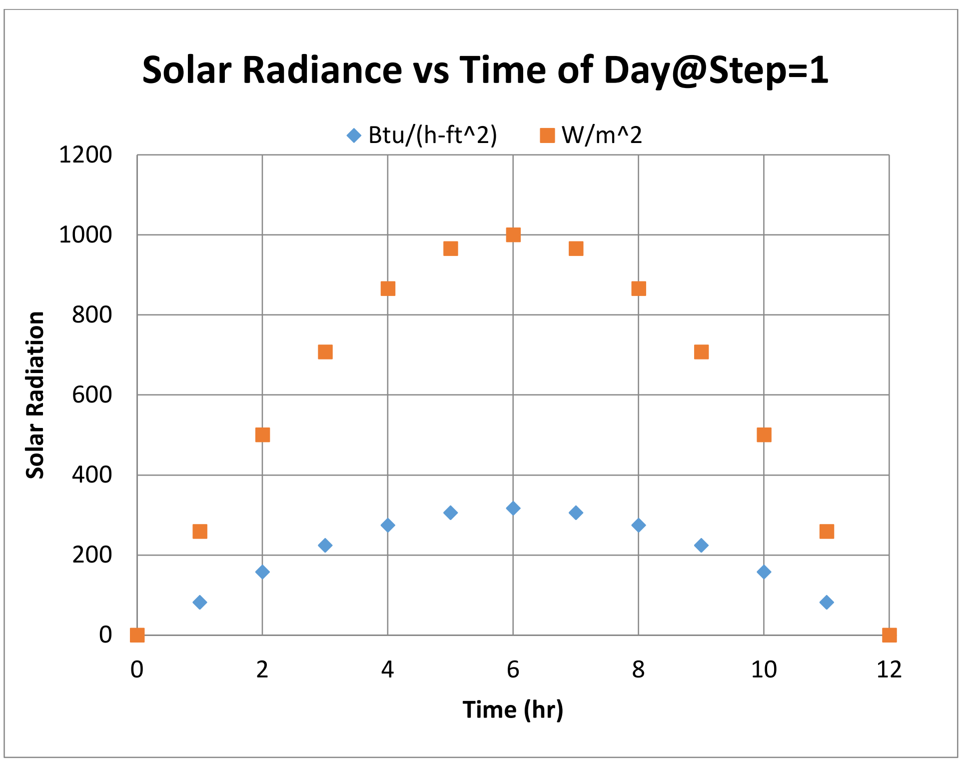
Example 8-3: Ideal Vapor compression heat pump

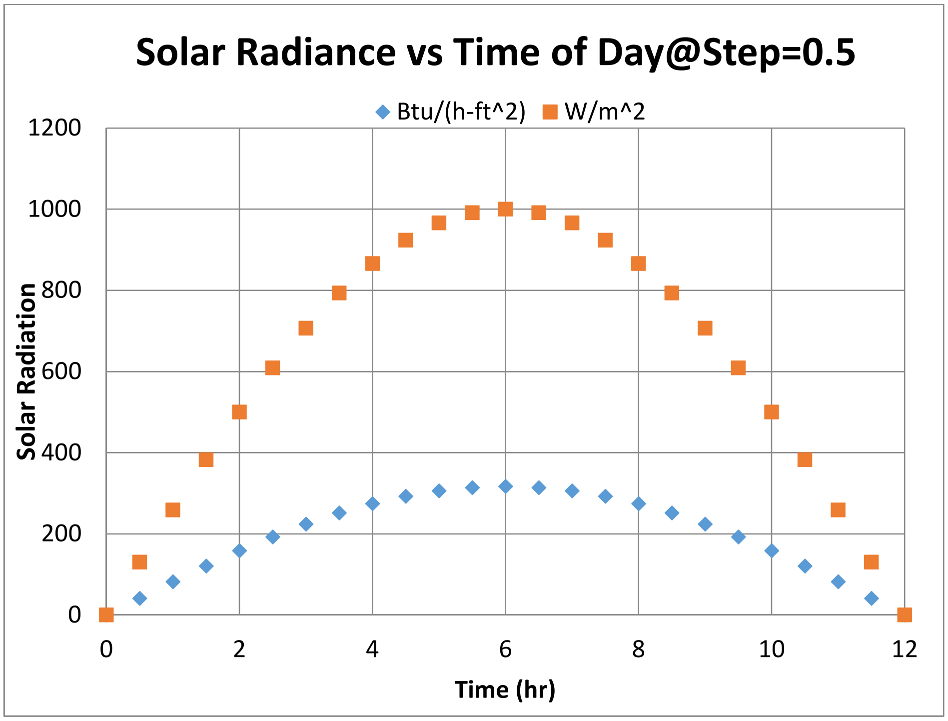
Parameter	Simulation	Expected	Percent Difference (%)
Work Rate (kW)	6.52	6.61	0.76
COP heating	5.261	5.29	0.55

Time step Sensitivity Test

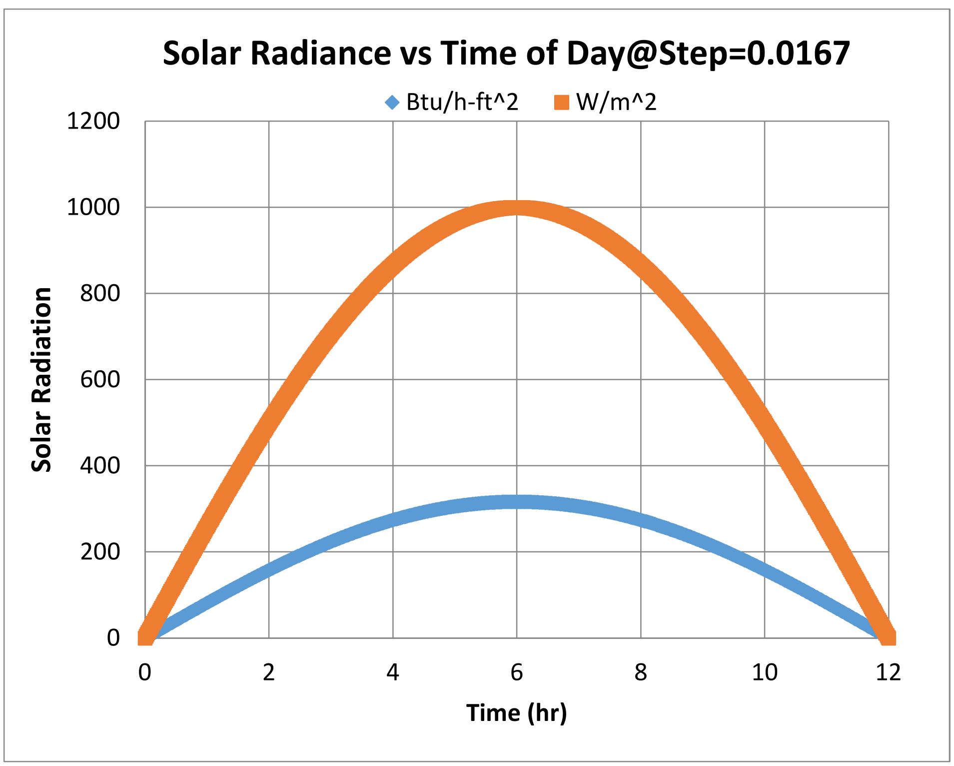
Simulation final time $(t_f) = 12 \rightarrow 12\ hrs$		
Time step(Δt)		Total Run Time (t_T)
1	$1\ hr$	$12\ s$
$\frac{1}{2} = 0.5$	$0.5\ hr$	$24\ s$
$\frac{1}{60} = 0.0167$	$1\ min$	$720\ s \approx 12\ mins$
$\frac{1}{3600} = 2.78 * 10^{-4}$	$1\ s$	$43200\ s \approx 720\ mins \approx 12\ hrs$

Effect of time step change on data accuracy shown below





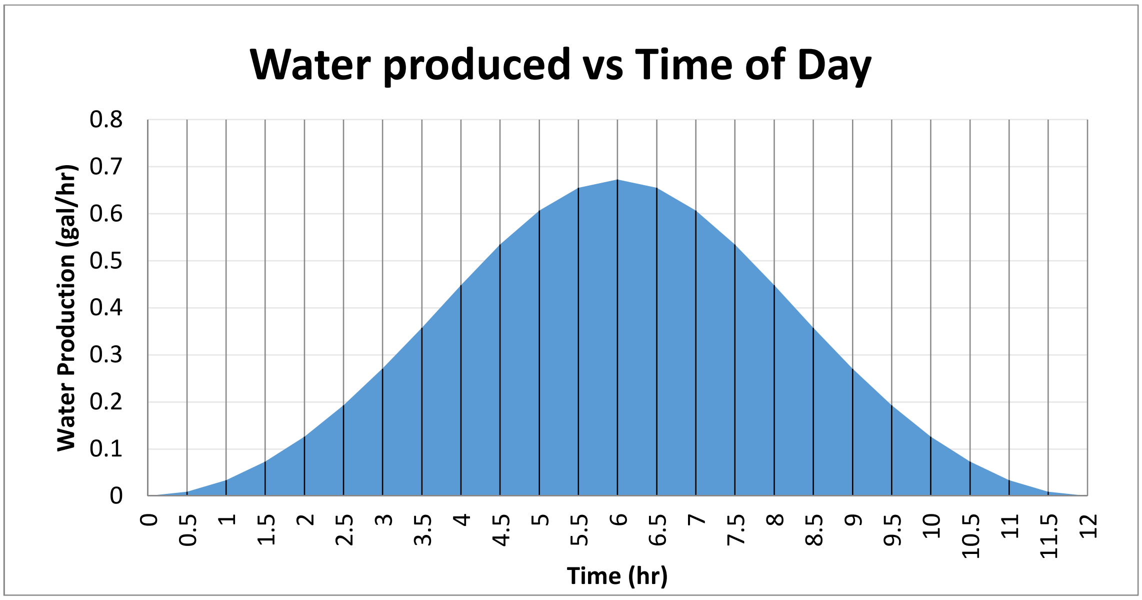
Smaller time step provides more curved plot indicating greater accuracy:



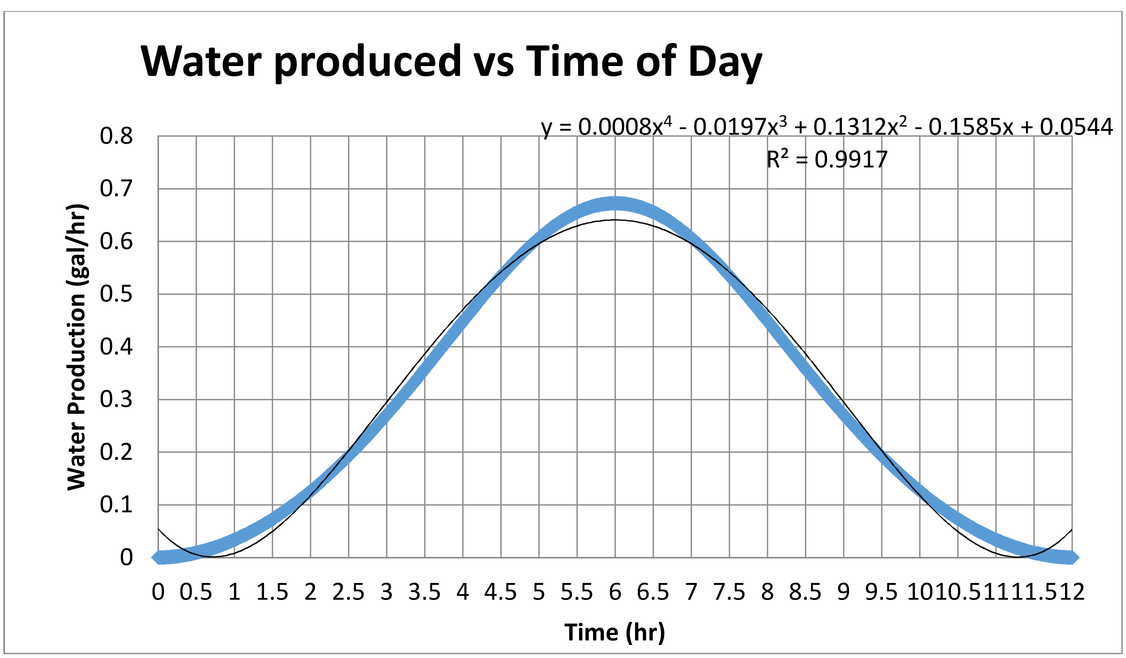
Total Water Produced

$$W_T = \int_{t_i}^{t_f} \dot{V} * dt$$

$$W_T = \sum_{t=i}^{n} \frac{V_i + V_{i+1}}{2}(t_{i+1} - t_i)$$



Or integration by trend line



Energy Factor

$$E.F = \frac{\dot{V}}{\dot{W}_{in}}$$

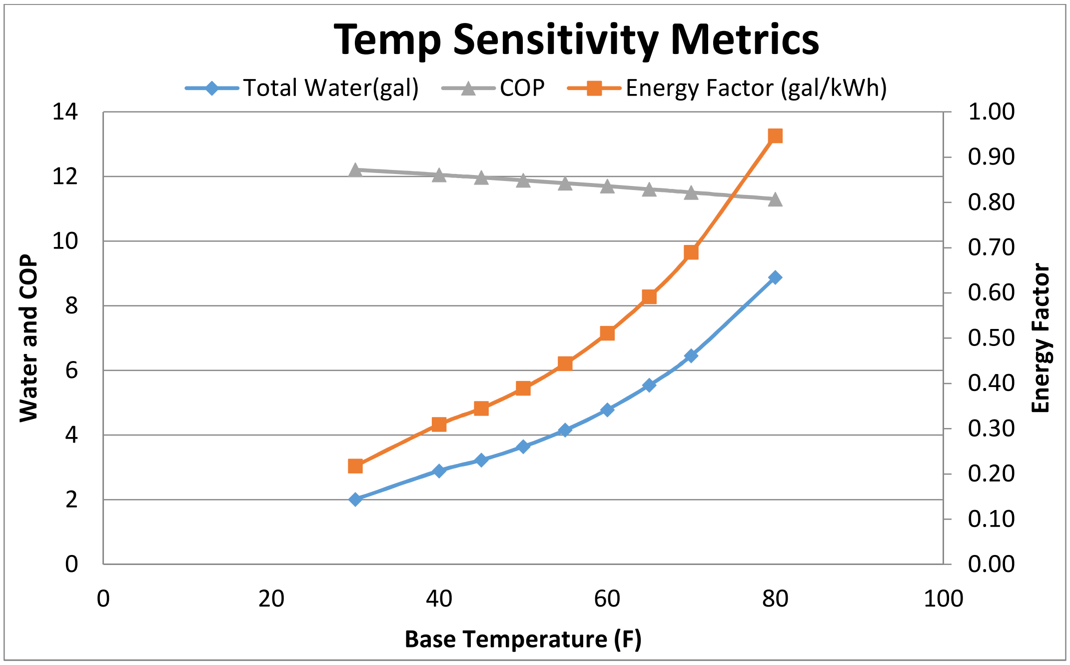
The energy factor represents the amount of water produced by the system for a corresponding unit input of energy. For this study, a constant energy input of 750W was used.

Time Step	Average Water (gal/hr)	Total Water (gal)	Work input (kW)	Energy Factor (gal/kWh)	Energy Factor (L/kWh)
1	0.2804	3.64540	0.75	0.374	1.415
0.5	0.2917	3.64565		0.389	1.472
0.0167	0.3032	3.64573		0.404	1.530

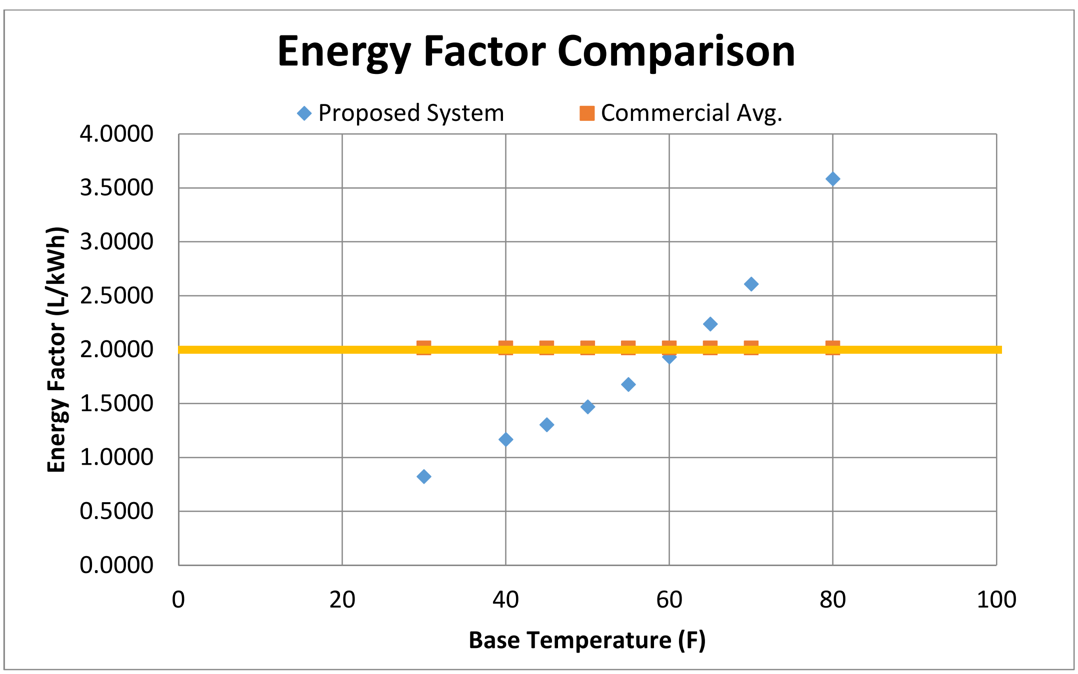
Total water produced with varying time step with initial temperature of 50°F

Base Temperature	Total Water (gal)	Energy Factor (gal/kWh)	Energy Factor (L/kWh)	Commercial Average (L/kWh)	COP
30	2.015	0.218	0.8238	2.0171	12.214
40	2.898	0.309	1.1701	2.0171	12.054
45	3.230	0.345	1.3042	2.0171	11.970
50	3.646	0.389	1.4720	2.0171	11.883
55	4.159	0.444	1.6793	2.0171	11.794
60	4.786	0.511	1.9326	2.0171	11.701
65	5.547	0.592	2.2396	2.0171	11.606
70	6.463	0.689	2.6097	2.0171	11.508
80	8.880	0.947	3.5855	2.0171	11.302

Total water produced at varying initial temperature with time step = 0.5

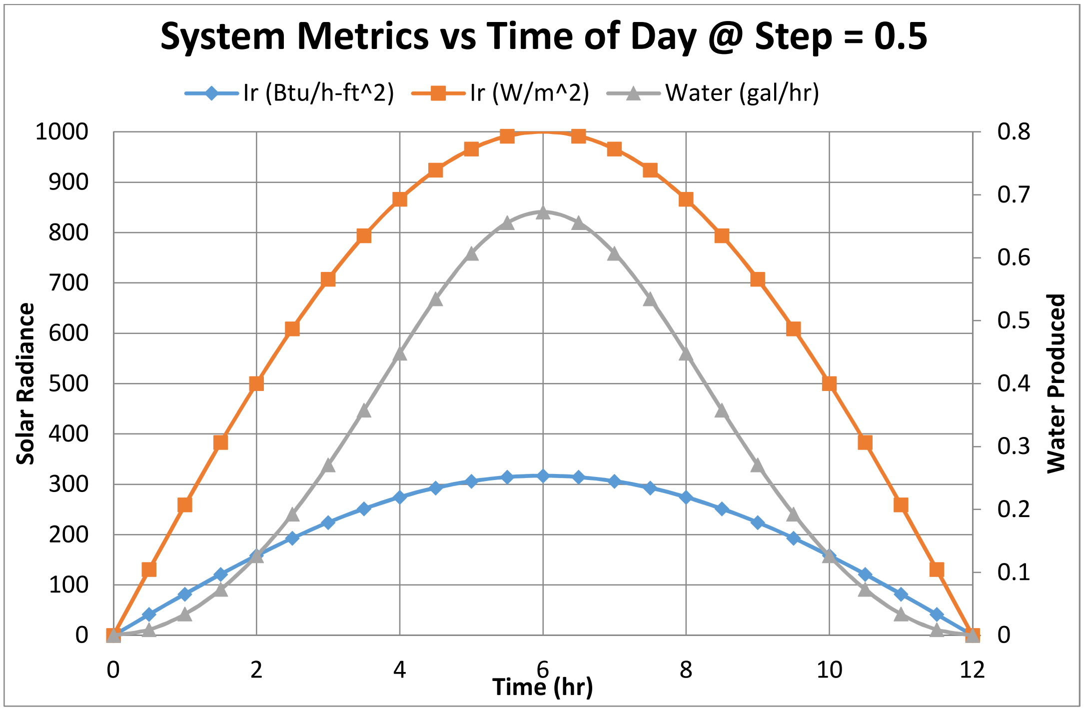


Increasing base temperature affects water and energy output positively while COP decreases slightly:

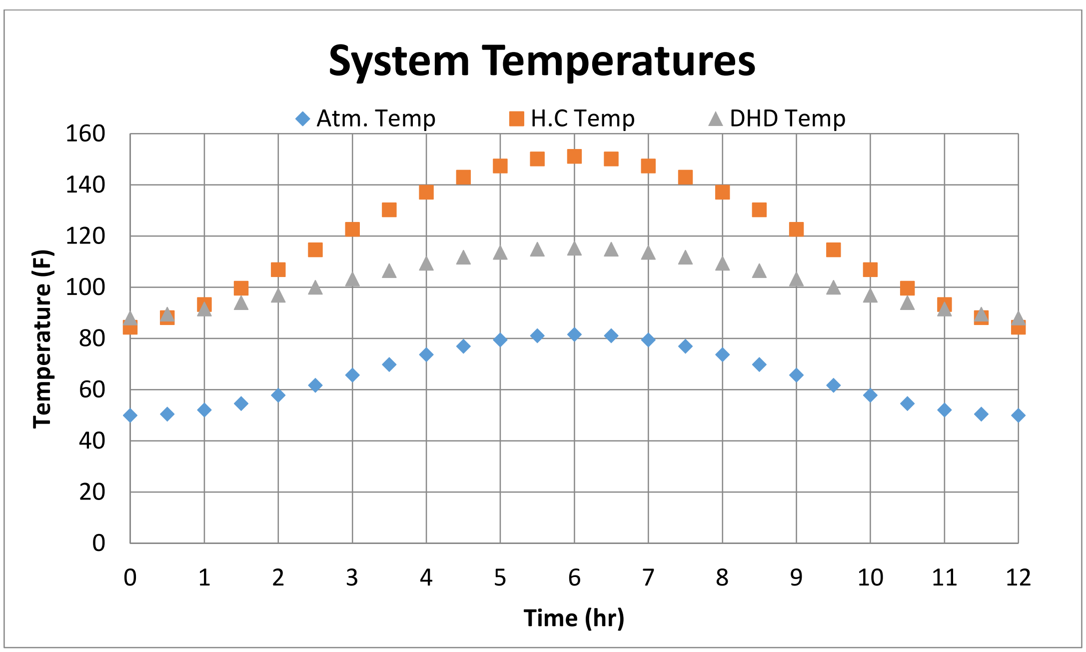


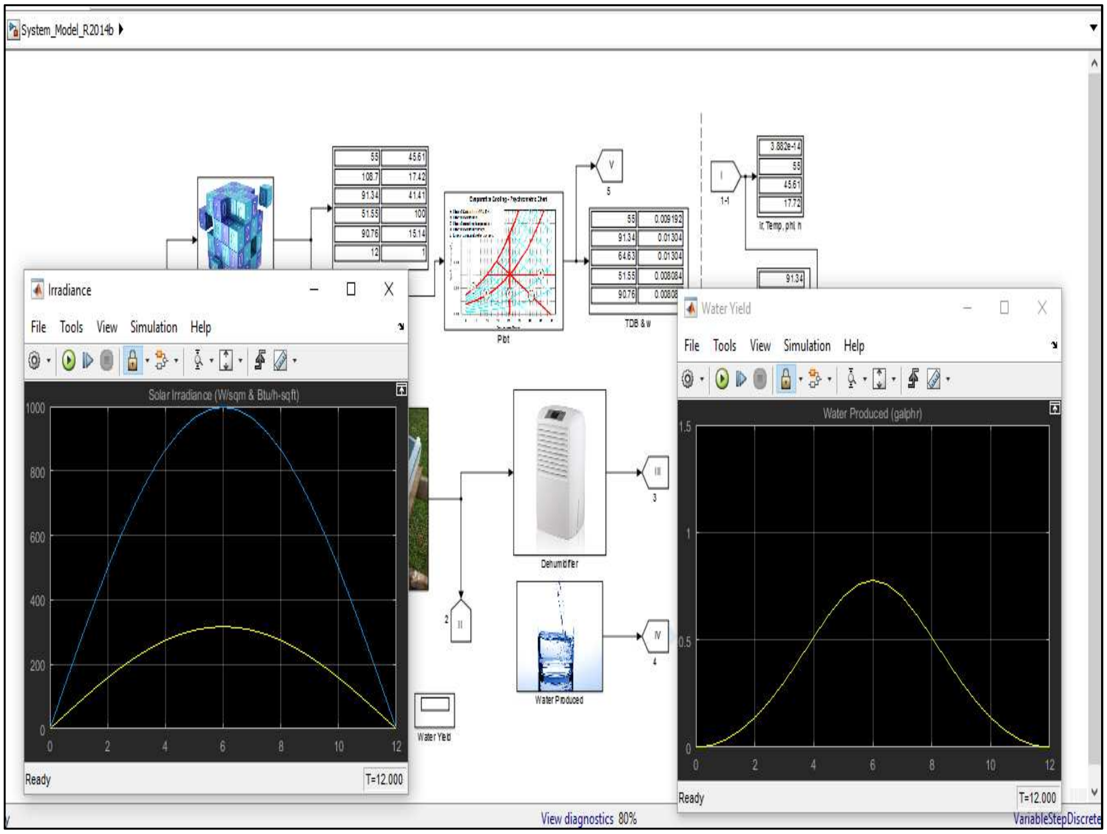
Comparing modeled system to commercially available units by energy factor, threshold is above 60°F

Simulation Figures

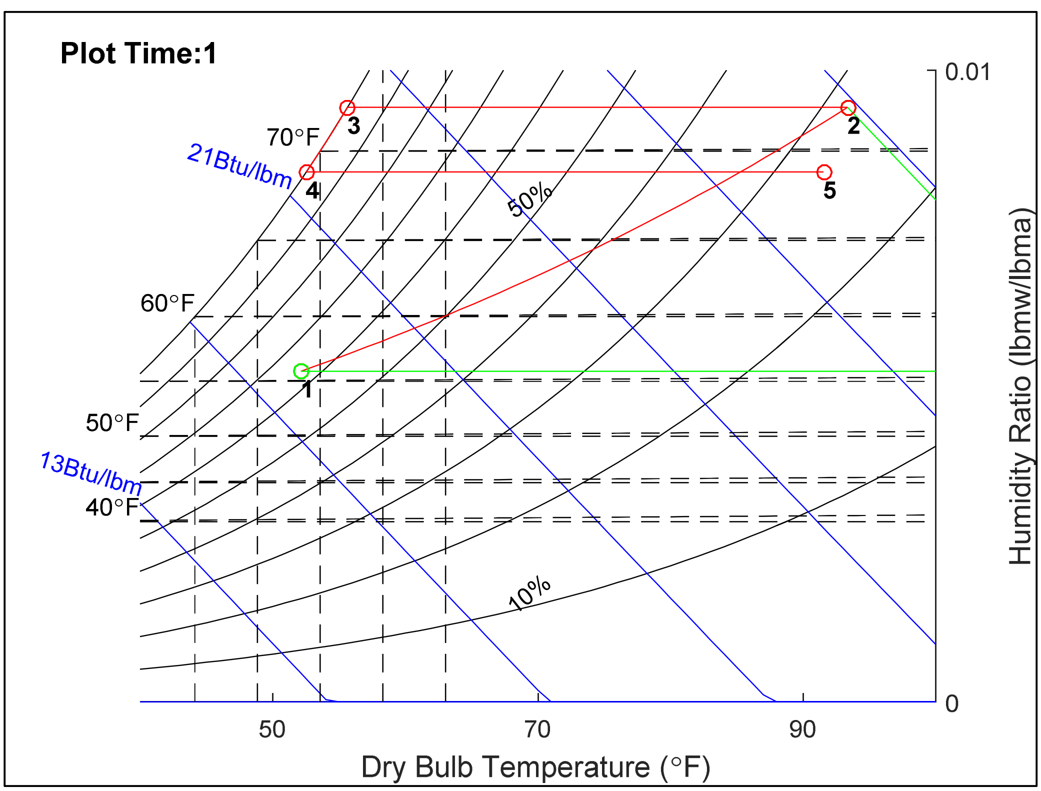


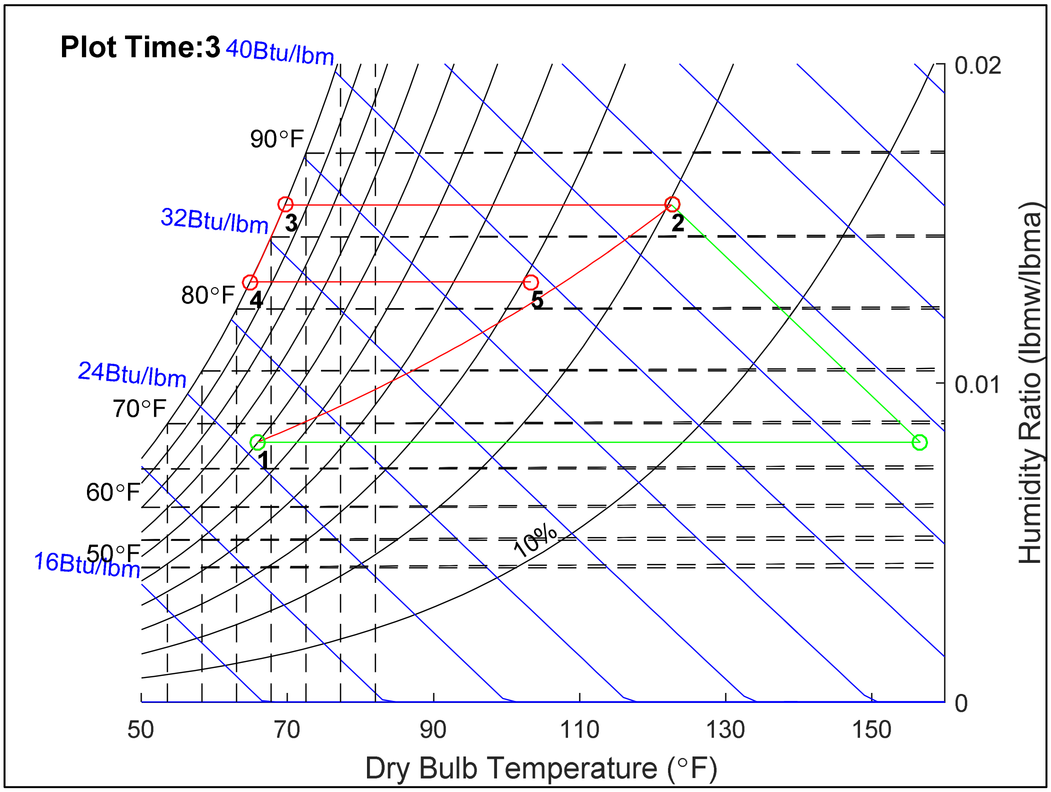
Direct relationship between solar radiance energy and water yield

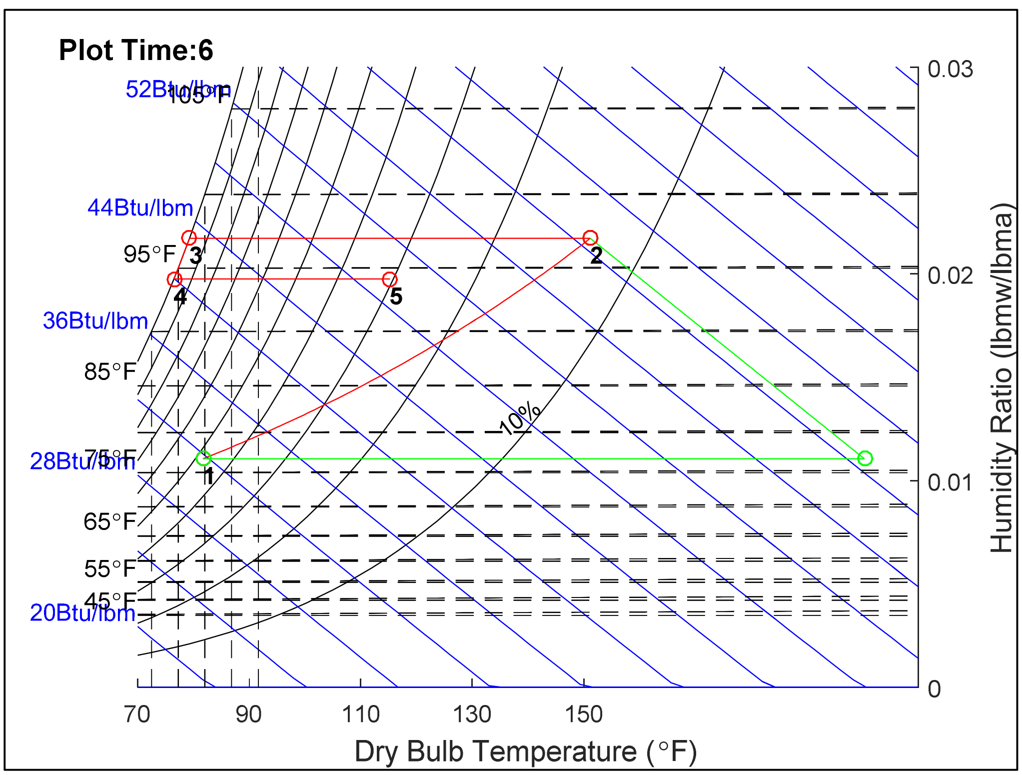


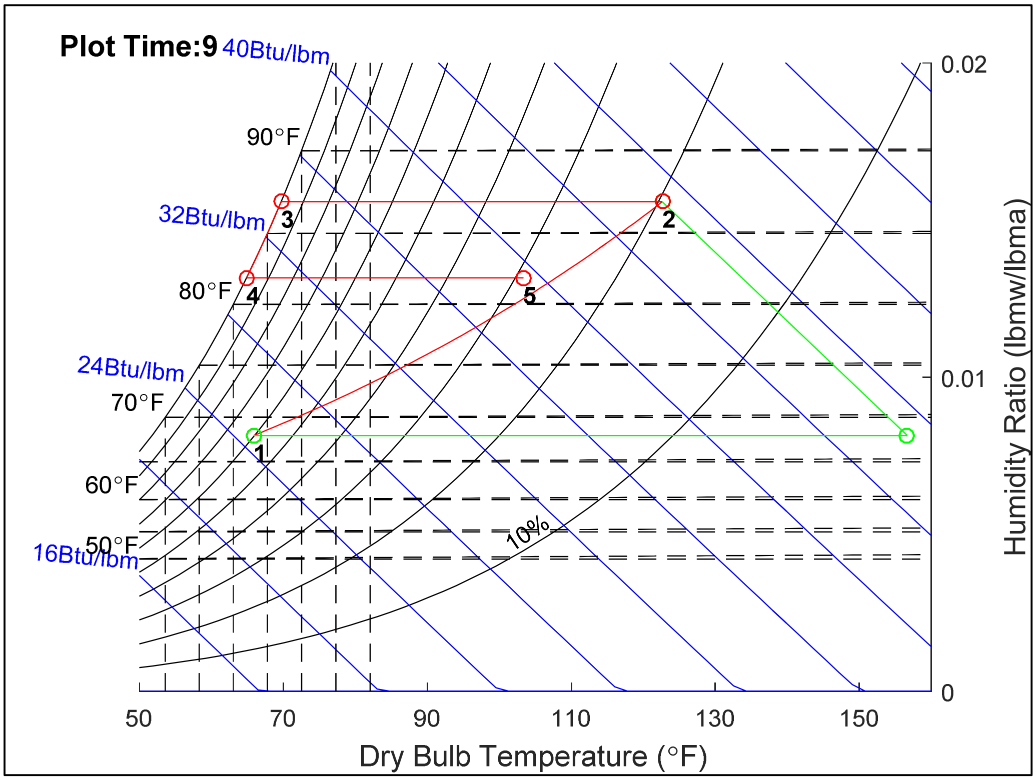


Model in run time simulation




Plot Time:3
40Btu/lbm
32Btu/lbm
24Btu/lbm
16Btu/lbm
90°F
80°F
70°F
60°F
50°F
10%
1
2
3
4
5
0.02
0.01
0
50
70
90
110
130
150
Dry Bulb Temperature (°F)
Humidity Ratio (lbmw/lbma)


Plot Time:6
52Btu/lbm
105°F
44Btu/lbm
95°F
36Btu/lbm
85°F
28Btu/lbm
75°F
65°F
55°F
45°F
20Btu/lbm
10%
1
2
3
4
5
0.03
0.02
0.01
0
70
90
110
130
150
Dry Bulb Temperature (°F)
Humidity Ratio (lbmw/lbma)



Psychometric plot of air cycle properties and cycle points

D. Sample Testing Instructions

PROBLEM STATEMENT:

Given a computer simulation from a graduate student at the Rochester Institute of Technology, the objective is to build and measure the real world operating conditions of the proposed system. The software model visualizes a small portable unit for low volume water production and filtration. The system relies solely on solar energy thus making it environmentally friendly with an open loop air cycle and a closed loop refrigerant cycle. The computer model is characterized by state properties of the air and refrigerant at nodes within the respective cycles. Physical measurement of the properties is required to match them with the simulated values.

EQUIPMENT:

The information provided here is classified as the equipment necessary to test the system and not to build it. Building materials can be user specified from wood to metal. The heat chamber can be made from plastic or rubber along with various other components. The dehumidifier can be recycled from an older model or put together by using the bare components of heat exchangers, a compressor, and a valve. The listed equipment could be utilized in testing a working prototype

- Electric mass scale
- Thermocouple
- Tachometer
- Cables/Wires
- Dial caliper/micrometer
- Volume measuring cup
- PASCO temperature/ pressure sensor
- Statistical software
- Wire cutters/Pliers
- GLX Pro data acquisition device
- Measuring Tape
- Solar array panel and DMM
- Wire current sensor
- Stopwatch
- PASCO humidity sensor
- Thermal properties software

PROCEDURE:

Part 1 – Setup / Calibration of Equipment

1. Setup the GLX pro device
    a. Turn on the device
    b. Connect the temperature thermocouple sensor and relative humidity sensors to the device
2. Calibrate the GLX device to room temperature
    a. Repeat this 25 to 40 times to get a good baseline reference for future measurements
    b. Average the data collected and find the deviation in the sample
3. Measure the dimensions of the prototype
    a. Measure and record the dimensions of the heat chamber using the measuring tape
        i. Get the height, length, and width
        ii. Record all measurements in the same base units

    b. Measure and record the dimensions of the heat exchanger coils with the measuring tape
        i. Repeat 30 times to average the width and height of the coils
    c. Measure the fin thickness on the coils with the micro-meter
        i. Repeat 30 times to average the width and height of the coils

Part 2 – Run the prototype model

1. Place the system outside on a sunny clear day.
    a. The system should be capable of functioning in rainy/dark weather as well. However, to simplify the validation procedure, ideal conditions of a clear sunny day should be tested.
    b. Measure the temperature, pressure, and relative humidity at least 3 times on every day of testing
2. Connect the dehumidifier to the power source (wall outlet or solar panel)
3. Check the evaporator coils
    a. Record any observations to the surface of the coils when the device begins running
        i. One should notice a cold frost buildup on the coils from the heat exchanger
4. Check the fan exhaust at the back of the device
    a. Record any observations to the exhaust when the device begins running
        i. One should notice hot and dry air being blown out at the exhaust

Part 3 – Collection of Raw Data on Heat chamber

1. Measure the fan speed using the tachometer
    a. Collect 40+ data points from the exit of the dehumidifier
        i. Average and normalize the data
    b. Collect 40+ data points from the entrance to the heat chamber
        i. Average and normalize the data
2. Measure the 1 gallon of polluted water with the measuring cup
    a. Weight the mass of water in the cup with the scale
    b. Pour the water into the empty tank
    c. Weight the mass of the empty cup with the scale

3. Open the valve fully and start the timer on the stopwatch
    a. Stop the timer once the tank is empty and record the time
    b. Repeat this 10 more times and normalize the distribution
    c. Measure the diameter of the valve outlet with the dial calipers
    d. Determine the number of turns or angle of turn needed to fully open the valve
        i. May need a protractor to make accurate measurements
        ii. Repeat this 5 – 20 times
4. Using the GLX device, measure the temperature within the heat chamber
    a. Mark 3 or more locations on the surface of the heat chamber along the length of the unit
        i. The more locations used the better the data evaluation becomes
    b. Use the GLX to sample the temperature range over time at each of those locations
        i. Use the same timespan for each of the locations
            1. Time greater than 2 mins
        ii. Save the data to a flash drive for each of the marked locations
    c. Use the GLX to sample the pressure range over time at each of those locations
        i. Use the same timespan for each of the locations
            1. Time greater than 2 mins
        ii. Save the data to a flash drive for each of the marked locations
    d. Use the GLX to sample the relative humidity range over time at each of those locations
        i. Use the same timespan for each of the locations
            1. Time greater than 2 mins
        ii. Save the data to a flash drive for each of the marked locations

NOTE: there are some sensors with combined temperature and pressure modes or temperature and humidity modes that may make the data sampling faster

Part 4 – Collection of Raw Data for Dehumidifier

1. Use the GLX device to measure the temperature on the evaporator coil fin

a. Note the minimum specification for the sensor in use as the evaporator coils will be frost covered
b. Repeat this three times for a set time interval
c. Repeat this step to measure and log the temperature for the condenser coils or fins

2. Using the current sensor, measure the current drawn by the compressor
    a. Repeat this measurement 5 to 30 times
    b. Record the standard voltage output for the geographic location of testing
    c. If using solar panels to produce energy, use the current measure device to measure the current produced by the solar panels to the compressor and record the voltage output
        i. Also measure the surface temperature of the solar panels with the GLX device. This will have to be done over the course of one day to generate an energy plot of the solar panel
3. Gain access to the external pipes of the dehumidifier and measure the temperature at all four points in the refrigerant cycle
    a. After the compressor
    b. After the condenser coils
    c. After the expansion valve
    d. After the evaporator coils
4. Measure the outer diameter of the coils using the dial calipers or micro meter
5. Collect and weight the condensed water from the evaporator coils
    a. Collect the water every hour for 30 samples
    b. Collect a full day of condensed water (12 hours of sunlight)

NOTE: Always remember to save each data log to a unique filename on the flash drive

REPORT COMPONENTS:

The following section details how the instruction and the data collected could possibly be compiled and analyzed to study the system. Following the same sequence in which the data was collected, the computations necessary to characterize the system are outlined below:

Part 1 – Setup / Calibration of Equipment

1. Collect and summarize the data.
2. Compare the mean value to that of a thermostat within the room
    a. Also possible to use a smart phone app to measure and validate the mean and standard deviation numbers from the GLX logger.
    b. Verify that the average temperature is within 1 standard deviation of other secondary device measurements
3. Calculate the parameters of the heat chamber from the recorded measurements

$$Volume = lenght * width * height$$

$$Surface\ Area = lenght * width$$

$$Cross\ section\ Area = width * height$$

4. Determine the cross sectional area of the heat exchangers; evaporator and condenser
5. Summarize the mean and variance of the heat exchanger fin widths and length
    a. Could be used to help determine the heat loss rate from the refrigerant to the air

Part 2 – Run the prototype model

6. Summarize the mean and standard deviation of the atmospheric parameters:
    a. Temperature, Pressure, and Relative Humidity
    b. This list must be logged for everyday of testing to assess the changes in weather conditions
7. Document all physical changes to the prototype when running
    a. Temperature increase on the black surface
    b. Frost build on the evaporator coil
    c. Hot dry air at the exit
    d. Compressor sounds when running

Part 3 – Collection of Raw Data on Heat chamber

8. Average and deviate the data on flow velocity from the fan(s)
    a. Compute the mean using excel or any statistical program
    b. Lookup/reference of the air density using the temperature and pressure measurements taken in part 2

i. The mass flow rate can be derived from the density and flow rate

$$\dot{m} = \frac{\dot{V}}{\rho}$$

ii. The mass flow can be found at the entrance to the chamber and exit of the dehumidifier. It is also possible to determine the mass flow rate into the dehumidifier if one is able to adequately measure the flow rate of air at that section.

9. From the tank water timing data, compute the volume flow rate out of tank.
   a. Use the volume measured into the tank and the time it took to empty the tank at 100% open valve. The mass flow rate of polluted water into the tank can then be found:
10. Determine the number of turns on the water tank valve needed to fully open the valve.
    a. Record and repeat this step 20 times
    b. Estimate the approximate percent open for the valve
       i. May need a protractor to determine the percent open for valves with less than 1 revolution turn
    c. This will aid in determining how the open valve affects the mass rate into the chamber

$$\dot{V}_{polluted} = \frac{V_{tank}}{t_{empty}} \qquad \dot{m}_{polluted} = \frac{\dot{V}_{polluted}}{A_{valve} * (\%open)}$$

11. Pull the temperature and humidity from the data logger into an excel file.
    a. Compute the average and standard deviation for each location
    b. Create a normal distribution scatter plot for each location temperature, relative humidity and pressure values
    c. Using the mean and deviation values for the three points, create a data table using the length along the heat chamber as the x values
    d. Compute the approximate temperature, humidity, and pressure values along the heat chamber
       i. Provide a reasoning behind the interpolation method used in the estimation

1. Linear interpolation vs Trapezoidal integration or other methods

e. Create a plot or graph based on the estimates to show the change in temperature, pressure, and humidity along the heat chamber

   i. One would expect all three parameters to show increasing values along the length of the heat chamber. The closer to the dehumidifier, the higher the parameter values should be.

12. Based on the temperature change from the inlet of the chamber to the outlet, calculate the total heat gained by the air within the chamber

    a. May need a thermal database to get the average specific heat for air

$$\dot{Q}_{gain} = \dot{m} * C_p * (T_f - T_i)$$

13. Using the total heat gain by the air, compute the temperature on the surface of the black body within the heat chamber

    a. Estimate the surface temperature:

       i. Use the convective heat transfer from the air to material surface

       ii. Use the radiation heat transfer from the sun to the surface

$$\dot{Q}_{surface} = h_{air} * A * \frac{(T_{surface} - T_{air})}{\Delta time} \qquad \dot{Q}_{surface} = \varepsilon * \sigma * A * \frac{(T_{surf}^4 - T_{air}^4)}{\Delta time}$$

    b. Verify this calculation by measuring the surface temperature 10 – 30 times

    c. Determine any efficiencies in the heat transfer from the chamber black body to the air in the chamber

Part 4 – Collection of Raw Data for Dehumidifier

14. Calculate the net-work drawn by the compressor from the current measurement and voltage supply.

    a. Note that the measured current may be AC for which the value must be converted to the equivalent DC value to represent the real work done.

       i. AC signals are composed of real power and apparent power

$$Real\ Power(DC) = \dot{W}_{net} = V * I * \cos\theta$$

    b. Create a scatter distribution plot of the power over time for the compressor.

        i. One should notice a more normal distribution if the solar panels are used and a more constant source supply from a wall outlet

   c. Note and record any times for which the compressor may be struggling to operate. This will become the baseline power the compressor needs to function

        i. Plot this baseline to the power distribution plot to determine when and how the power supply can support the compressor load

15. Compute the average temperature on the evaporator coils along with the standard deviation
    a. The standard deviation should be rather large as the temperature of the refrigerant changes drastically from the inlet to the exit of the coils.
    b. Using the measurement along the coil fins from one side to the other, create a plot of temperature variation along the evaporator.
    c. Using the thermal database software, determine the state properties at the exit and entrance of the evaporator based on the temperature and assumed characteristics
        i. Quality after the evaporator is saturated vapor or superheated vapor
        ii. Quality before the evaporator is mixed quality
16. Repeat step 15 for the condenser coils
    a. Average and normalize the temperature data
    b. Create a thermal contour plot along the width of the coil
    c. Estimate cycle properties based on temperature
        i. Quality into the condenser is superheated from compressor
        ii. Quality after the condenser is saturated liquid or sub cooled
17. Use the net-work to the compressor and the determined state properties from steps 15 and 16 to estimate the mass flow rate of the refrigerant:

$$\dot{m}_{ref} = \frac{\dot{W}_{net}}{h_1 - h_4}$$

a. If using solar panels, create the distribution of power produced from the solar energy over the course of a day through the average of the data points collected

18. Analyze the collected water samples
    a. Find the mean water produced per hour
    b. Add all the water values together to estimate the daily water production and run the system for a day to verify the result.
        i. Integrate between delta time
    c. Using the water produced per day, create a plot of the solar power and water produced over a day time frame
19. Compute the performance metrics of the system from the state properties

$$Cycle\ Efficiency\ [\eta_{cycle}] = 1 - \frac{Q_L}{Q_H} = 1 - \frac{h_4 - h_1}{h_3 - h_2}$$

$$Carnot\ Efficiency\ [\eta_{Carnot}] = 1 - \frac{T_L}{T_H}$$

$$Coefficient\ of\ Performance\ for\ Cooling\ [COP] = \frac{Q_L}{\dot{W}_{net}} = \frac{h_1 - h_4}{h_2 - h_1}$$

    a. Compare the performance metrics to the simulated values and plot the comparison
    b. Determine the energy factor based on the energy consumed by the compressor

Part 5 – Further analysis

20. Conduct short term Design of Experiment test that could determine the effect of certain parameters to the water yield. The proposed DOE is simply one example of many that could be conducted. Use a 2^3 factorial design of 3 parameters with 2 states each. The factors proposed at the flow rate of polluted water into the heat chamber, the number of fans used to draw air into the chamber, and the size of the heat chamber. The size of the heat chamber could easily be separated into multiply factors such as varying the length, width, or height to determine the relative effects.

a. The goal of this DOE is not to optimize the system but rather to determine which factors have the most effect on the output or if there are any key interactions that must be noted.

$$2^5 = 32\,runs$$

E. Simulation Parameter Values

The key values of parameters used for the simulation are provided here:

Parameter	Value	Units
Atmospheric relative humidity	40	%
Atmospheric air temperature	40-90	°F
Atmospheric pressure	14.69	psi
Blackwater mass flow	0.024	$\frac{lbm}{s}$
Evaporator inlet Ref. temperature	20	°F
Evaporator inlet Ref. quality	20	%
Evaporator inlet Ref. mass rate	0.1	$\frac{lbm}{s}$
Heat chamber dimensions	6'x2'x0.5'	ft
Heat exchanger tube diameter	0.25	in
Heat exchanger tube inner diameter	0.2	in
Heat exchanger shell length	12	in
Heat exchanger fin number	40	
Heat exchanger tube passes	50	
Heat exchanger tube material	Copper	
Compressor work load	750	W
Heat chamber energy lost	20% of $E_{T,in}$	
Heat chamber fan rating	200	$\frac{ft^3}{min}$
Solar peak radiation $(\boldsymbol{I_{pk}})$	316/1000	$\frac{Btu}{hrft^2}/\frac{W}{m^2}$