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Modeling and Testing of a Liquid Desiccant Dehumidification System Regenerated with Solar Thermal Energy

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**Modeling and Testing of a Liquid Desiccant
Dehumidification System Regenerated with Solar Thermal
Energy**

by

Michael Andrew Hatt

B.S. Electrical Engineering, University of Colorado, 2004

A thesis submitted to the
Faculty of the Graduate School of the
University of Colorado in partial fulfillment
of the requirements for the degree of
Masters of Science
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Modeling and Testing of a Liquid Desiccant Dehumidification System Regenerated with Solar
Thermal Energy
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The final copy of this thesis has been examined by the signatories, and we find that both the content and the form meet acceptable presentation standards of scholarly work in the above mentioned discipline.

Hatt, Michael Andrew (M.S., Civil Engineering)

Modeling and Testing of a Liquid Desiccant Dehumidification System Regenerated with Solar Thermal Energy

Thesis directed by Prof. Michael Brandemuehl, PhD, P.E.

A computer simulation model for a liquid desiccant and PV/T system was examined in this thesis. The model was calibrated using data from an experimental system installed in Orlando, Florida, developed by the the National Renewable Energy Laboratory, in conjunction with several other companies. The experimental system uses a warm-air cross-flow absorber/regenerator and a 11.2kW PV/T array that provides hot air to regenerate the desiccant.

The data gathered from the experimental system was used to validate the accuracy of the PV/T array temperatures, along with the effectiveness of the desiccant and regenerator in the simulation model. An expanded simulation model uses a previously validated model for a cross-flow absorber and cooling tower, along with the newly modeled regenerator and PV/T system. The combined system was modeled for various absorber, regenerator, tank, and PV/T sizes to determine appropriate component sizing to meet a given ventilation load. The simulations were used for seven different locations, including: Orlando, Houston, Atlanta, Washington DC, Phoenix, Guam, and Honolulu. Performance characteristics were examined, along with the potential energy savings of using a PV/T-desiccant system for dehumidification instead of standard air-conditioning.

Results of the simulations show that the largest amount of absorption achieved is at high flow rates at lower PV/T outlet temperature set-points. However, the highest PV percentage (ratio of PV panels to thermal collector modules) provides the most energy savings, because less fan and pump energy are required and more electrical energy is being generated. Temperature set-points above 40°C were found to be acceptable for regeneration, which also allows for a higher air flow rate than would be permitted at higher regeneration temperatures. Meeting the full latent load at the ASHRAE minimum outdoor air ventilation rate was shown to not be possible for most multi-story

office buildings, because the PV/T array sizes required are typically too large for the building roof . However, operating a PV/T-desiccant system to meet a 50% load allow for much smaller array sizes, even at higher PV percentages. When meeting the half load, climates in the very hot-humid zone 1A and hot-humid zone 2A achieved savings of 22-49% when compared to using standard air-conditioning to meet the latent load. More energy is obviously saved when the PV/T-desiccant provides more dehumidification than the standard air-conditioner.

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Chapter 1

Introduction

1.1 Latent Cooling in Humid Climates

Indoor comfort is based on the temperature and humidity of a space. In warm climates, temperature is controlled by simply cooling air to a particular set-point temperature. The energy required for the desired change in temperature can be referred to as the “sensible” load. Humidity is typically indirectly controlled by cooling air below the dew-point temperature to condense the water vapor out of the air. To effectively cool humid air to the dew-point in vapor-compression air-conditioners, the air usually must be cooled to far below the desired indoor temperature and then reheated. The additional cooling and heating is inefficient and a waste of energy. Also, the amount of energy required to condense water vapor out of air with no change in temperature is the “latent heat of condensation” or “latent load.” Since additional energy is required for dehumidification without changing the air temperature, cooling humid air requires much more energy than cooling dry air. Vapor-compression systems can be sized for various sensible and latent loads, but there are limits to the latent load for a given sensible load. Sometimes, the latent load cannot be met and humidity in the space may rise. An option for dehumidifying humid air or to reduce the latent load on a vapor-compression system is to use a desiccant system to directly absorb moisture from the incoming air.

1.2 Desiccant Dehumidification

Desiccants are substances that have the ability to attract and hold water vapor. Desiccants are able to absorb water due to the vapor pressure difference between the air and the desiccant. The water vapor will move from a higher pressure surface to lower pressure surface where there is less water vapor. The water molecules try to equalize the vapor pressure differences. The process of condensing the water molecules from air into the desiccant also releases heat due to the latent heat of condensation. Reversing the process requires heat to be added to the desiccant.

Desiccant systems can either use solid or liquid desiccants to absorb moisture from the air. Solid desiccants, such as silica gel, are usually set up in packed beds or a rotary wheel where air flows through the solid, packed desiccant material. Liquid desiccants, such as calcium chloride or lithium chloride, typically flow in a thin layer over a surface with the humid air blown across the flowing desiccant. An “absorber” is the device that where the desiccant and air stream interact and absorb moisture, as shown in Figure 1.1.

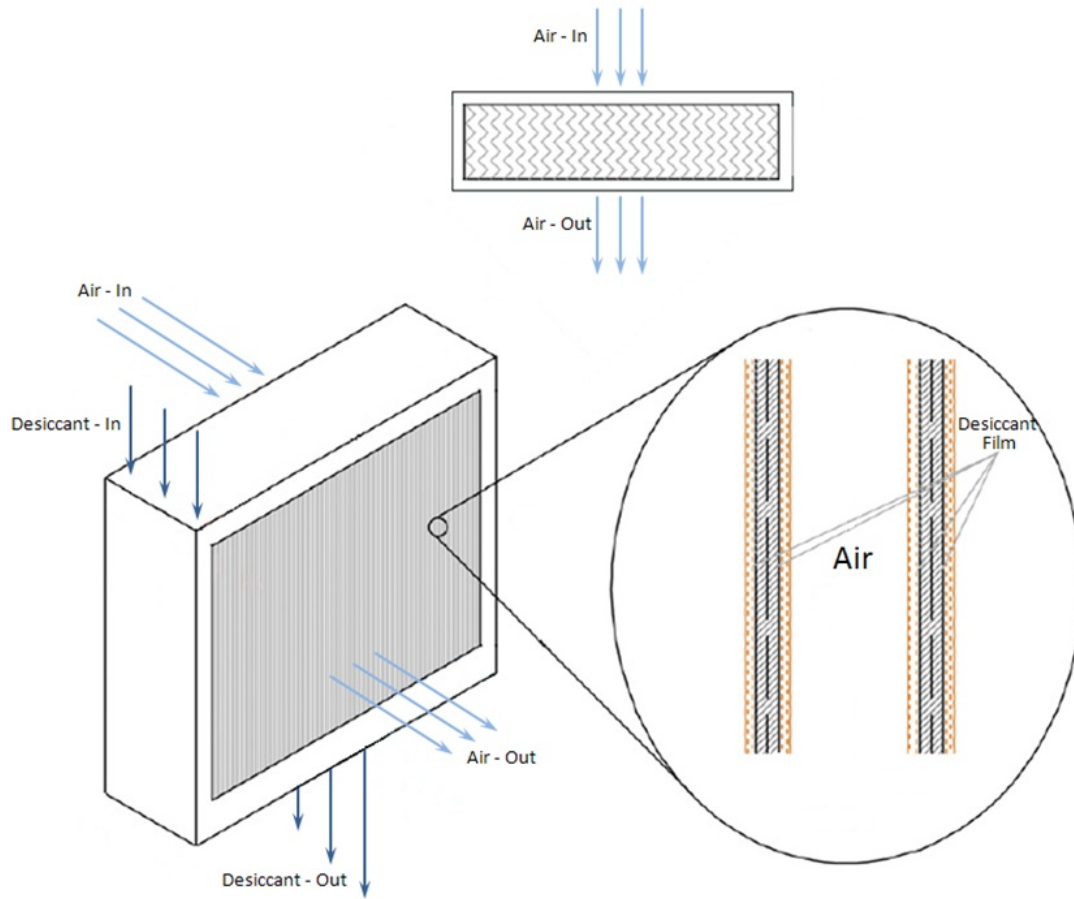


Figure 1.1: Regenerator/Absorber Flow Diagram

In both types of desiccants, the desiccant can only absorb a finite amount of moisture, and the moisture must then be removed so that it can be used again for moisture removal. To remove the moisture from the desiccant, heat must be added. The heat causes the desiccant vapor pressure to increase above that of the air, so that the moisture moves from the desiccant back to a stream of air. The process of removing moisture from the desiccant substance is called “regeneration.” The air used for this process is usually exhaust air leaving the space or air that does not need conditioning. The source of heat for the regeneration process can come from many sources, including hot water, hot air, or a gas-fired boiler. Liquid desiccants can also be regenerated from the output of a solar thermal collector, which may be part of a hybrid PV and solar thermal system.

1.3 PV/T Collectors

Photovoltaic/Thermal Collectors, or PV/T collectors, combine the electrical generation from Photovoltaic (PV) panels with the thermal energy output of a typical solar thermal collector. PV cells are semiconductor devices that convert high-energy photons from solar radiation into electricity. However, not all of the incident solar radiation is converted into electricity, and much of the energy is absorbed by the PV cells and creates heat. The heat created by the absorption of the solar radiation is typically rejected to the ambient air and is not used by the PV collector. In fact, as the temperature of the PV cells increases, the electrical generation efficiency decreases. However, PV/T collectors use some type of heat transfer fluid flowing under the PV cells to remove the waste heat from the back of the PV panels and convert it into useful thermal energy. Figure 1.2 shows a sample configuration using hot air as the fluid.

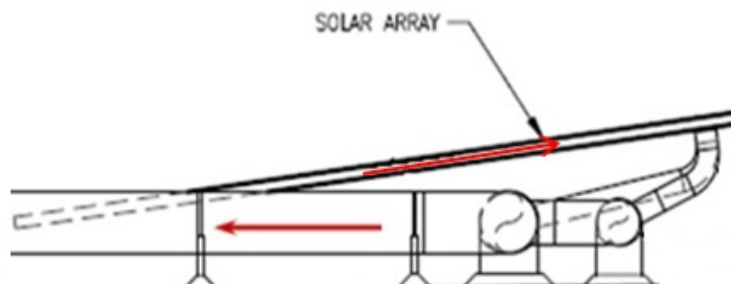


Figure 1.2: PV/T Air Collector

The heat transfer fluid can either be air or an aqueous solution, such as water. There are also many methods for transferring the heat from the back of the collector to the heat transfer fluid. Sheet-and-tube, channel, and free flow approaches can be used to move the fluid in aqueous systems. Air-based systems have large ducts or a plenum directly beneath the collector surface that basically traps the heated air. Fans, for forced-flow, or natural convection, allow the air to flow through the plenum from inlet to the outlet. Air-based systems are much simpler, and can use the roof or large sheet-metal pans to create the plenum of air below the panels.

The thermal energy transferred to the heat transfer fluid has a variety of uses. Water-based

systems can be used for pool heating or to produce domestic hot water. Air-based systems can be used for space-heating in colder months, where the hot air is simply ducted into a space. During summer months, the hot air produced by the PV/T system can potentially be used to regenerate a weak desiccant, which can then be used for dehumidification and cooling.

1.4 Combined PV/T and Desiccant System

PV/T systems provide many advantages over using separate PV and thermal collectors. By combining the two types of collectors, the energy production per surface area increases. Since the same roof area can be used for two different purposes at the same time, there is more energy production for the same roof area. Also, since most PV cell efficiency decreases as temperatures rise, adding heat removal to the back of the PV system can increase PV cell efficiency and produce more electrical output. While the thermal output of a PV/T may not be as efficient as a stand-alone thermal collector system, the PV/T system still creates useful thermal energy that is otherwise wasted in a stand-alone PV system. Any of these systems also provide roof shading that can create a possible reduction in cooling loads. The heat transfer fluid in the PV/T system also relocates and uses the thermal energy that contributes to heating of the roof.

Using the thermal output of the PV/T system for desiccant regeneration also has many advantages. In summer months, particularly in hot and humid climates, using the thermal output of a PV/T or thermal collector system for space heating or pool heating would be unnecessary. Even though the thermal output may be used for domestic hot water heating, cooling loads are the main concern in the summer. Cooling loads are much more significant than domestic hot water loads, so using the thermal output of the PV/T system to aid in reducing cooling loads would be very beneficial. Liquid desiccant systems have a significant potential for reducing the cooling loads in humid climates. Since the thermal output of the PV/T is suitable for regeneration of a liquid desiccant, combining a PV/T system with liquid desiccant regeneration should be effective in reducing overall energy use.

1.5 Objective

The overall objective of this thesis is to model and analyze the performance of an experimental cross-flow warm-air regenerator, and to use the model in understanding the performance differences in varying climates and potential energy savings of using a PV/T-desiccant system for dehumidification of outdoor air. Also, this thesis is used to aid in choosing component sizes of the PV/T array, desiccant tank, and regenerator fan, so that energy savings and performance can be maximized. To achieve the overall objective, there are several tasks that need to be completed. First, a computer model must be developed to simulate the experimental regenerator and PV/T system installed in Orlando. The data from the experimental system is used to calibrate and validate the model, and is the key to understanding the effectiveness and performance at various temperatures and outdoor air conditions. An expanded model is then created to understand the interactions between an absorber, regenerator, tank, and PV/T array, and to analyze how these components affect performance. Using the expanded model, component sizing charts can be created, and energy use of each system type can be found. Energy savings must also be analyzed by comparing the electrical energy use of a PV/T-desiccant system to a standard air-conditioning system. An estimate of latent load and energy use will be found for various climates, and compared to the energy use of the modeled PV/T-desiccant system. The climates chosen for sizing and energy savings analysis are Orlando, Houston, Honolulu, Phoenix, Atlanta, Washington DC, and Guam.

Chapter 2

Literature Review

Research on the use of desiccant absorbers for dehumidification has increased due to the increasing energy demand in hot and humid climates. To further reduce energy demand, the thermal energy provided by the sun in solar thermal systems can be used to regenerate a desiccant in a regenerator. Many types of these combined systems have been studied, and will be discussed in this review. Since this thesis involves modeling and testing of an experimental liquid desiccant system combined with a PV/T collector, liquid desiccant absorbers, regenerators, and PV/T collector components will each need to be modeled. The review also discusses previously studied liquid desiccant and air-based PV/T system modeling.

2.1 Solar Regeneration of Desiccants

Many desiccant systems have been tested using solar thermal energy as the regeneration source. Most of these systems use solid desiccants in a rotary wheel. However, more recent tests include liquid desiccant systems. The regeneration source for all of the systems discussed below use solar thermal energy from either a solar thermal collector or PV/T collector.

2.1.1 Rotary-Wheel Desiccant Systems

A 1978 study by Nelson et al. [29] analyzed the use of a rotary desiccant wheel regenerated by thermal energy from glazed solar thermal collectors. The system was simulated in TRNSYS [16] after creating theoretical computer models for each system component. Many of the components

were assumed to be ideal to allow the simulation to provide the theoretical limits of such a system. Simulations using Miami, Florida weather data showed an average COP of 0.75 during the summer season, which is similar to that of absorption systems. The study also concluded that solar thermal collectors are suitable for desiccant regeneration, and provides a reasonable amount of regeneration using a moderately sized collector system. Since the system in this study was an ideal simulation only, real systems must also be studied.

Eicker et al. [15] analyzed the performance of three separate rotary-wheel desiccant systems installed in various climates and buildings. The desiccant cooling systems were installed in 3500 m² public library in Spain, a 72 m³ laboratory in China, and an 800 m² plastics factory in Germany. All three systems used the hot air from solar thermal collectors as the regeneration source. While the three systems were different in climate and equipment sizing, it was found that the latent cooling energy demands were able to be met without using much additional energy [15]. The results show that desiccant regeneration using solar thermal energy significantly reduces cooling demand. The paper also focused on various control strategies to optimize the desiccant regeneration, and found that reduction in auxiliary energy required adjustable regeneration temperatures.

Bourdoukan et al. [8] performed experiments on a solid desiccant wheel system regenerated with thermal energy from a vacuum-tube solar collector. The system was not incorporated into a particular space, but measurements were taken for the inlet and outlet conditions. The desired inlet conditions were controlled with a pre-treatment air-handling unit. The study found that the most influential parameter on the system performance was outside humidity ratio. Performance of the system degraded at high humidity levels, and the paper concluded that the system should not be considered with an absolute humidity above 15 g/kg. However, the system showed greater potential for energy reduction in buildings in moderately humid climates [8].

Solar thermal energy from a ventilated PV facade, similar to a PV/T system, can also be used as a regeneration source for a desiccant. A system of this type was installed on a 3200 m² library in Spain and investigated by Mei et al. [27]. The overall system contained approximately 525 m² of PV panels with an additional 105 m² air-collector for improved performance. The components

of the system were modeled, and simulations were run for the summer and winter months. The study showed that the 93% of the latent cooling demand was met for the summer, with an overall COP of 0.518. It was clear from the study that the thermal energy provided by the PV facade was significant, and that the system has a “good potential [27] for reducing cooling demand.

Beccali et al. [7] studied various systems that combined solid desiccant cooling, heat pumps, standard air conditioning systems, solar collectors, and PV/T collectors. Combinations of these systems were simulated in TRNSYS using existing models. The study provided an economic analysis by comparing each of the systems along with standard A/C. The paper concluded that a desiccant cooling system with an integrated heat pump provided the most primary energy savings. Also, systems with a larger fill factor, percent PV area, provided more savings than systems with only 50% PV area. The desiccant systems also provided a reasonable economic return.

2.1.2 Liquid Desiccant Systems

Many types and configurations of liquid desiccant systems have been modeled and tested. Alizadeh et al. [2] compared a model and prototype of a forced, parallel-flow collector/regenerator, where liquid desiccant flows directly over the collector absorber plate in the same direction as the air flow. After testing the prototype system under various flow rates, solar insolation levels, and ambient conditions, the study found that the desorption rate of water vapor from the desiccant generally increased with increased air flow rate and decreased with solution flow rate. However, the outdoor conditions greatly affect the performance of the system, and there is an optimal desiccant solution to air mass flow rate that can be found to increase performance. While this system is a forced parallel flow system, other types of systems may also benefit from optimizing this flow rate ratio described in the Alizadeh [2] paper.

Another type of collector/regenerator was modeled by Peng et al. [31], but the Peng system was a counter-flow system, where the air stream and desiccant solution flow in opposite directions. The simulations of the counter-flow collector/regenerator showed that the water vapor desorption rate increased with increasing solution mass flow rate, which is different than the parallel-flow

system. However, increased air flow rate also increased the evaporation rate, which is the same for both types of systems. An optimal solution to air mass flow was not analyzed in the Peng paper, but the data indicates that there is an optimal air flow rate. Both the Peng and Alizadeh [2] systems seem to show that each system has optimal flow rates for given conditions.

Katejanekam et al. [21] performed an experimental analysis of a counter-flow liquid desiccant system installed in Thailand, where data from 19 different days and over 9 months was used. The system was also modeled and compared to this experimental data. The experiment examined the overall differences between inlet air to the outlet air conditions of the regenerator, but the trends were not discussed. The study concluded that the counter-flow conditioner/regenerator could provide up to 11% reduction in relative humidity with no almost no reduction in temperature. Since the tests were run in various seasons, the study also determined that the system should not run in all seasons due to moisture transfer between the air and desiccant going in the wrong direction under many conditions. The study shows the need for control over regeneration and dehumidification times, and as seen in other studies, the need for finding optimal strategies for these controls.

Based on a system studied by Jones et al. [20], Andrusiak et al. [3] analyzed a prototype liquid desiccant dedicated outdoor air system (DOAS) system installed in Toronto, Canada. Empirical TRNSYS [16] models were developed for the scavenging air regenerator and conditioner by Jones [20]. While the actual system used a gas-fired boiler as a regeneration source, the purpose was to simulate solar thermal regeneration. The Andrusiak study showed that the prototype system could provide between 0.9 and 9.2 tons of latent cooling, but a 200 m² solar array could only provide up to half of the required thermal energy required during the summer. Both studies concluded that desiccant storage is needed to provide additional dehumidification when solar energy cannot be used, since there is generally additional energy during the day to regenerate and store desiccant.

While none of these systems are exactly like the system modeled and examined in this paper, the previous models and experimental systems highlight the potential of using solar thermal energy to regenerate a desiccant. The previous studies also emphasize characteristics of these systems that may be important when analyzing future systems.

2.2 Liquid Desiccant Modeling

Liquid desiccant systems come in a variety of designs for which many different types of models have been developed. Many of these designs have also been developed into prototypes or commercial systems that have experimental data. All of the systems described provide dehumidification of outdoor air and have a desiccant regenerator that can be regenerated by any heat source. The other two possible components of each system are the desiccant storage tank and sensible heat air-handling unit [26]. Since the various system designs must take into consideration many factors, models and experiments have been created for determining the effects of desiccant type, packing type, packing material, flow patterns, and the use of cooling/heating water [26]. The effects and interactions of these factors have been described using analytical and numerical models, but some are also compared with actual experimental data. The various existing models for parallel-plate cross-flow regenerators/absorbers will be the focus of this literature review, since this paper examines a prototype adiabatic parallel-plate cross-flow regenerator, similar to that shown in Figure 1.1.

The prototype examined in this project is in a parallel-plate configuration, where a number of vertical plates are aligned parallel to one another with a small space in between each plate. The desiccant flows into the top of the regenerator/absorber where it drips or flows down along the vertical plates, which creates a thin-film of desiccant along the surface of each plate. Air passes between the parallel plates and comes into contact with the desiccant for dehumidification or regeneration purposes. Since the path of the air and path of the desiccant cross each other, this is also considered a cross-flow regenerator/absorber. The interactions between the air and desiccant in this particular configuration must be analyzed.

Liu et al. [23] developed theoretical analytic solutions for counter, parallel, and cross-flow regenerators and absorbers, in which the plates or packing type is not discussed. The analytical solutions provide outlet air and desiccant temperature and humidity ratios, along with moisture and enthalpy effectiveness equations. The effectiveness equations are a function of NTU, calculated from the regenerator/absorber size parameters and air flow rate, and the air to desiccant thermal

capacity ratio. All of the analytical equations can be calculated from known parameters of the system. The solutions from the Liu models were found to correlate well with results from previous literature.

A theoretical mathematical model was created by Dai and Zhang [13] that considered the heat and mass transfer of an adiabatic cross-flow dehumidifier with honeycomb packing. The model treats the paper packing material as walls with a constant spacing, which is similar to modeling parallel plate dehumidifiers. Also, as in most other models, the film thickness is assumed to be constant. The research involved finding analytical solutions for the model, and creating a program that numerically solved the differential equation based mathematical model. Both the mathematical model and analytical equations determined the Nusselt and Sherwood numbers for the liquid and the gas sides of the dehumidifier, which were then compared and analyzed. The study concluded that the analytical model produced very different results from the mathematical due to the “intimate” relationship of the air and desiccant heat and mass transfer coefficients that are part of the more complex mathematical program.

Mesquita et al. [28] developed several models to describe the heat and mass transfer of air and liquid desiccant in this parallel-plate configuration for an absorber. The first model uses differential equations to describe the heat and mass transfer correlations and assumes a constant thickness of desiccant film along the plate. The finite-difference model divides the air into 40 nodes and the desiccant film into 7 nodes, which allows programs like EXCEL to solve the differential equations from the previous model. Another model was also created to examine desiccant films of varying thickness, which improves upon and slightly modifies the constant thickness model. By modifying the equations and reducing the number of nodes to one for the air and desiccant streams, a simplified model was created for use in programs such as TRNSYS [16]. The variable film model was compared to data in previous literature and results match within 6% for lower desiccant flow rates, with increasing discrepancy as desiccant flow rate increases [28]. The finite-difference model underestimated dehumidification in the absorber, and was found to not be as accurate as the variable film thickness model.

Ali et al. [1] analyzed the heat and mass transfer between the air and a falling desiccant film. The analysis specifically addresses a regenerator/absorber in a parallel-plate cross-flow configuration. The mass, momentum, energy, and diffusion equations were determined and then solved numerically using central finite difference approximations. While the analysis was not compared to experimental data or other literature, it was used for parametric analysis to optimize regenerator/absorber design. The study concluded that better dehumidification is possible with lower air Reynolds number, increased channel height and length, and decreased channel width.

Cooling water, flowing in tubes or between plates, can also be used to remove heat during dehumidification, while heating water is used for regeneration. The performance of this type of internally cooled liquid desiccant dehumidifier was analyzed by Khan et al. [22]. The Khan [22] study analyzes a dehumidifier consisting of parallel plates with water tubes running through the plates, which is similar to a fin-tube heat exchanger. The air and desiccant streams are in cross-flow configuration, while the air and water streams are in counter-flow. The mathematical differential model was solved numerically, along with a simplified model, and several parameters were analyzed. The model showed that the NTU values, water inlet temperature, water to air flow rates, regenerator heat, and desiccant concentration effected the performance of the absorber, but all of the parameters had more of an effect on the sensible load than the latent load [22]. However, the heat carried over from the regenerator to the absorber adversely affected the absorber performance.

Another internally-cooled parallel-plate cross-flow desiccant regenerator and absorber were studied by Park et al [30] and compared to the experimental results of a prototype system. Heat and mass transfer differential equations were created to describe this particular experimental system, and solved numerically using finite difference equations. Input and output measurements from the experimental system were taken and recorded by the LabView data acquisition software. The input measurements were used as the inlet conditions for the mathematical model, while the output measurements were compared to the simulation results. The study showed good agreement between the simulation model and the experimental system, and found that lower air flow rates give better humidity control and cooling. In other words, the air flow rate should be carefully controlled for

optimal regeneration rate.

Regenerators and absorbers, regardless of packing type or material, can also be characterized according to an NTU-effectiveness type model, similar to models used for heat exchangers. The NTU and effectiveness values can be determined using specific equations, but the value can also be determined using experimental data. Lui et al. [24] created a two-dimensional mathematical model of a cross-flow regenerator/absorber that uses an NTU value and Lewis number as input parameters. The NTU value is determined using experimental data, but the Lewis number is assumed to be one. The model is discretized into an MxN mesh of nodes, or control volumes, and numerically solved as a matrix of equations. Enthalpy and moisture effectiveness values were calculated from the experimental data and the mathematical model. The average enthalpy effectiveness difference is 7.9% for the absorber, and 5.8% for the regenerator. For moisture effectiveness, the values are 8.5% for the absorber and 6.9% for the regenerator [24]. The model is reasonably accurate and can be applied to regenerators/absorbers with different types of packing types and configurations because the NTU parameter is based on experimental data.

The previous model by Lui [24] assumed the Lewis number to be one, but Yin and Zhang [35] found the Lewis number to be an important factor in desiccant heat and mass transfer calculations for a cross-flow absorber/regenerator. Yin and Zhang [35] developed a methodology for separately determining the h_d value for heat transfer, and the Le number for the coupling of heat and mass transfer. The method is based on an NTU-Le model, but substitutes the two new parameters for a more accurate model. The study showed that the air and desiccant inlet conditions had a large effect on the h_d -Le input parameters, but were simple to determine with the newly developed methodology. The model provided reasonable correlation with the experimental data, and the air humidity, air temperature, and desiccant temperature changes were only 10%, 6%, and 12% different, respectively.

Andrusiak et al. [3] used a more simplified approach for modeling an internally-cooled parallel-plate cross-flow absorber. Instead of determining NTU or Le values, the effectiveness values were determined using experimental data for moisture removal, enthalpy change, and heat

transfer between desiccant and heating/cooling water. The combination of the three effectiveness equations allows the model to determine all necessary outlet conditions using the psychrometric chart. The simplified model was used in TRNSYS to simulate a system using solar energy as a regeneration source, and the results studied in various climates. Even though the simulation showed good correlation between the model and experimental data, the simplified effectiveness model was not as accurate and an empirical model was used.

To simulate an adiabatic parallel-plate cross-flow absorber/regenerator in TRNSYS requires a more simplified model than the mathematical differential models. Also, the parameters of the prototype system designed for this research must be determined experimentally. Since a simplified experimental approach must be taken, a modified Andrusiak [3] model can be used to analyze the prototype system.

2.3 PV/T Modeling

A review of PV/T collector literature was performed by Charalambous et al. [10] in 2006, which described the types of PV/T collectors, analytical and numerical modeling, simulation and experiments, and an qualitative evaluation of various parameters affecting thermal-electrical output. In 2009, Chow [11] performed a similar review of PV/T collector technology, but also described the developments in the last decade, along with commercial applications, and the market potential for PV/T technology. Both studies agreed that PV/T has excellent potential but further research must be conducted to increase product quality.

In 1995, Sopian et al. [33] determined closed-form solutions from a differential model of single-pass and double-pass air-based PV/T collectors. Even though the study was not compared to any experimental results, the model showed that higher PV-cell efficiency could be obtained from the cooling effect that the air channel provides. Double-pass air channels were also found to further increase efficiency with only a negligible increase in production cost. Sopian [33] concluded that PV/T collectors are ideal for many applications.

An extensive study by Hegazy [18] in 1999 investigated four types of air-based PV/T col-

lectors: air flowing over the absorber, air flowing under the absorber, single-pass with air on both sides, and double-pass. The research provided the energy and heat transfer equations for all four models and solved them numerically to find energy production, required fan power, and PV and thermal efficiencies. Tiwari and Sodha [34] derived analytic expressions from a thermal resistance network model for four different models of PV/T collectors: glazed and unglazed collectors with and without a tedlar backing. The expressions determined temperatures for the solar cells, back surface, and the outlet air, and also the useful energy produced in the system. The values were compared to experimental data from a prototype system. While both studies provided models for various types of PV/T systems, only the Tiwari and Sodha [34] models were compared to experimental data. The study showed a fair agreement between the simulation and the experiment for outlet temperatures, but drastic wind changes caused a much greater error for solar cell and back surface temperatures. Incorporating heat capacitance effects into the model could potentially reduce these errors.

Aste et al. [6], under an R&D program at Politecnico di Milano, carried out the design, development, and monitoring of an air-based PV/T system. The design and development included the creation of a steady-state model that underwent six revisions before reaching the final product. Due to measuring limitations of solar irradiation data, only seven optimal days were used for the experimental data. However, when using the limited amount of data, the experiment closely matched that of the simulation. Also, some corrections were applied to the model to account for the maximum power point tracker installed in the PV/T system. While the model provided a reasonable representation of the given data, more experimental data is necessary for a more complete and accurate model that could be used to describe other similar systems.

Sarhaddi et al. [32] studied previously existing steady-state PV/T models and attempted to improve the accuracy by creating more detailed thermal and electrical models. Several corrections and additions were made to the heat loss coefficients in previous models to include conductive heat transfer through the tedlar, radiative heat transfer from the solar cell to ambient, convective heat transfer on the back of the PV/T collector, convective heat transfer inside the air duct, and the

overall heat transfer from the air collector to the surrounding air. The heat transfer inside the air duct and the overall heat transfer from the air collector to the surrounding are generally considered constant values, but were changed to calculated or variable values for the model in the Sarhaddi study. The electrical model was improved by replacing typical linear functions with a non-linear model that more accurately reflects the actual behavior of the PV cells. The model was compared to results from previous literature and showed improvement over the previous model. However, the Sarhaddi model still does not include the thermal capacitance effects of the system, and these effects could potentially show more improvement.

In 2006, Ito et al. [19] created an unsteady-state model of a solar air collector to take into account the heat capacity effects of the system. Steady-state models ignore the heat capacities of the cover, absorber plate, and bottom plate, but the Ito et al. model integrates these into the energy balance equations for greater accuracy. The steady-state and unsteady-state models were compared to experimental data, and the results showed that the unsteady-state model more closely matched that of the experimental data. The study found that unsteady-state analysis is important in understanding the transient effects and for predicting outlet temperature and other parameters due to changes in solar radiation intensity, variable flow rates, and changes in wind velocity.

Based on the findings of Ito et al., Casey [9] created an unsteady-state model of a PV/T collector that was used for parametric simulations of various systems and climates using TRNSYS. Casey considered the various TRNSYS components for modeling an experimental PV/T system, but found that none were suitable for use in the simulations. Casey created a new TRNSYS component model using assumptions similar to that in the Ito et al. [19] model, but added PV considerations into the heat, mass, and energy balances. Experimental data showed that the thermal capacitance effects were significant, and an unsteady-state model was necessary for the accurately modeling the PV/T system.

Since this thesis examines a PV/T system nearly identical to the Casey system, the model in this thesis uses the Casey model with modified assumptions and updated parameters to account for different PV cells and roofing structure.

Chapter 3

System Descriptions

The National Renewable Energy Laboratory (NREL), working with the Florida Solar Energy Center (FSEC), EchoFirst, and AIL Research, has installed a liquid desiccant regenerator/absorber in conjunction with an existing PV/T system at the Orange County Convention Center in Orlando, Florida. The adiabatic cross-flow liquid desiccant regenerator uses warm air from the output of the PV/T system as the regeneration source for the desiccant. This thesis examines the ability of the experimental combined system to provide dehumidification and cooling load reductions for hot and humid climates.

Figure 3.1 shows an overall system diagram of the experimental Orlando system. Data collected from the sensors/control system in Orlando was used to validate and calibrate simulation model components, which is discussed in Chapter 5, “Model Calibration.” The same calibrated component models were later used for further modeling, to expand on the experimental system and analyze a fully functional system that could be used in buildings. This section describes the various components of the experimental system in greater detail.

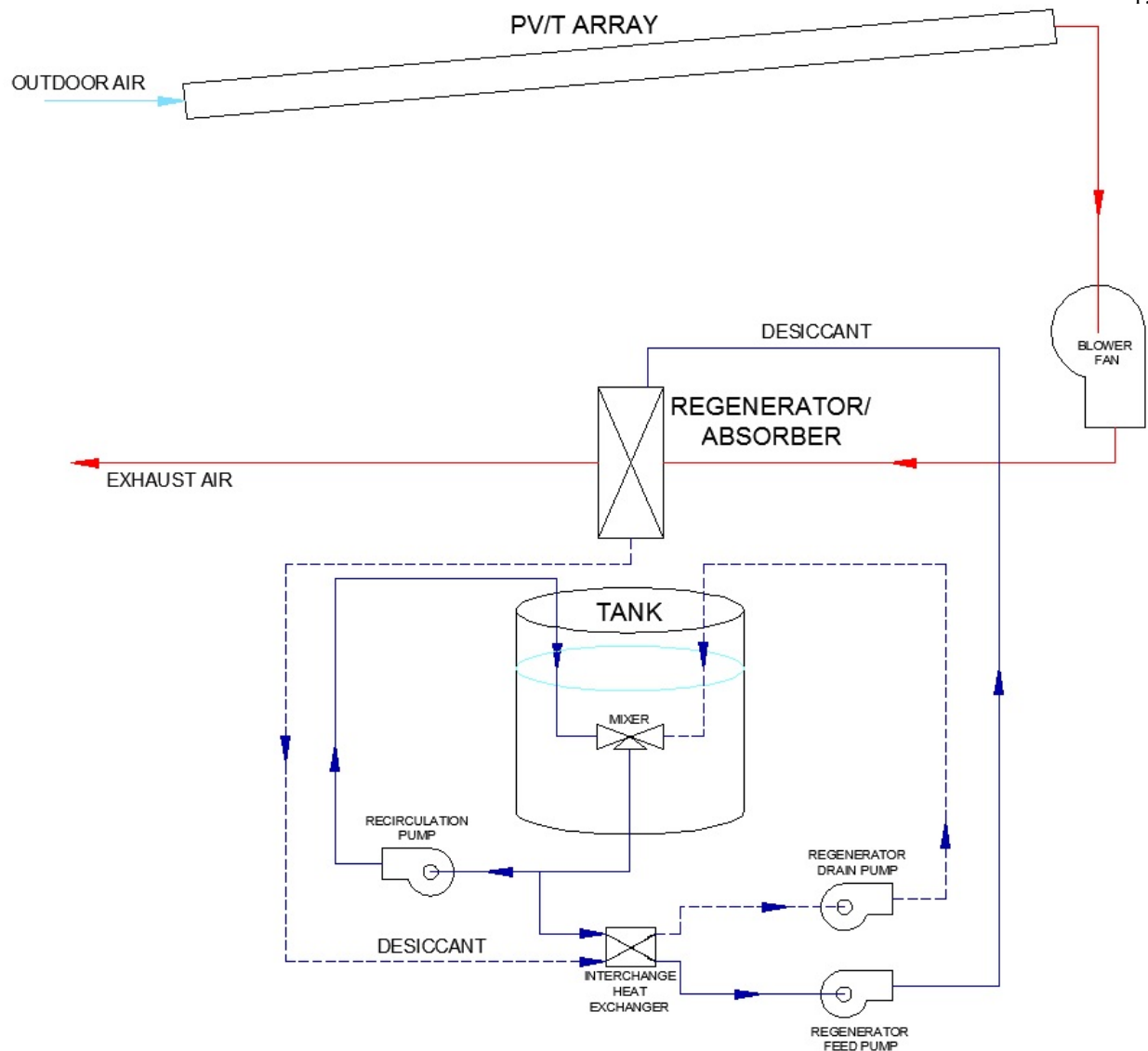


Figure 3.1: Experimental System Diagram

3.1 Photovoltaic/Thermal Air Collector System

The PV/T array, mounting system, and control system were designed and developed by EchoFirst Inc. and installed on the roof of the Orange County Convention Center. The array consists of 64 PV modules arranged in 4 rows and 16 columns, with 1 thermal module above each column of PV modules for a total of 16 thermal modules. The mounting system is a modified version of the EchoFirst TiltRack Cleanline System, and is used to hold the PV/thermal modules

and create an air plenum for heating the air. Figure 3.2 shows the initial PV/T system before installation of the duct system and insulation. Figure 3.3 and Figure 3.4 show the fully installed PV/T system that is connected to the liquid desiccant system.



Figure 3.2: PV/T Array Before Duct Installation



Figure 3.3: PV/T Array After Duct and Insulation Installation (side)



Figure 3.4: PV/T Array After Duct and Insulation Installation (rear)

3.1.1 Mounting System

The mounting system for the PV/T array is similar to the EchoFirst TiltRack Cleanline system. However, the TiltRack CleanLine mounting system is typically used on sloped roofs and integrated into the roof of the building, so that the roof surface acts as the lower part of the air plenum. Since the Orange County Convention Center has a flat concrete roof where the array is installed, the mounting system must be tilted and cannot be integrated into the roof surface. The modified TiltRack Cleanline system adds a metal pan system underneath the array to create an air plenum. Insulation was also installed under the pan system to reduce thermal losses from the plenum to the ambient air underneath the array. As shown in Figure 3.5, the mounting system consists of the compression rail system, pans, and insulation. The compression rail system is used to mount and hold the PV/Thermal modules and pans in place. While various PV/Thermal modules can be used with this rail system, the pans were designed to carefully fit this specific TiltRack CleanLine system.

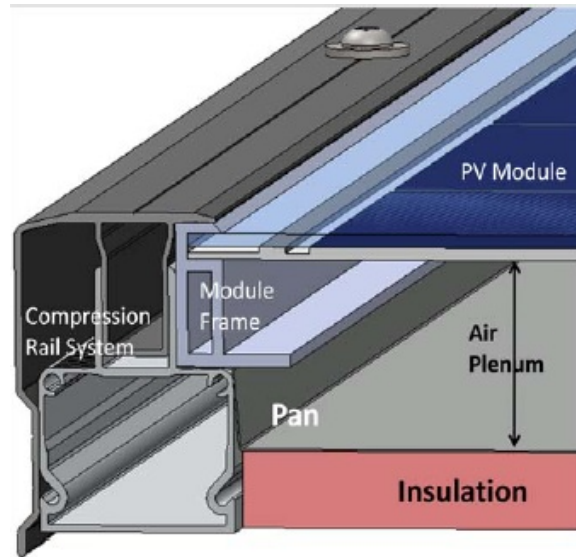


Figure 3.5: Diagram of PV/T Mounting System

The aluminum compression rail system creates an overall depth of approximately 2.9 inches. The pans are 26 gauge galvalume sheet metal, which creates an air plenum of 1.9 inches, from the rear of the PV/T modules to the top of the pan. However, metal “ribs”, as seen in Figure 3.6, were added to the sheet metal to increase air turbulence and heat transfer to the air.



Figure 3.6: CFD Analysis for Air Plenum

3.1.2 Thermal Modules

The top row of modules in the PV/T array are glazed air-heating collectors, designed by EchoFirst. As shown in Figure 3.7, the module is composed of a transparent glass cover that is approximately 0.1181 inches (3mm), an air gap of 0.7874 inches (0.02m), and a copper absorber plate measuring approximately 0.007874 inches (0.0002m). The surface of the copper absorbing plate has been sprayed with Solkote Hi/Sorb-II Selective Solar Coating to improve the emissivity and absorptivity of the copper surface for solar thermal applications. The air channel, as described

in the mounting system, is 1.9 inches deep, and backed by the galvalume sheet metal pan and foam insulation.

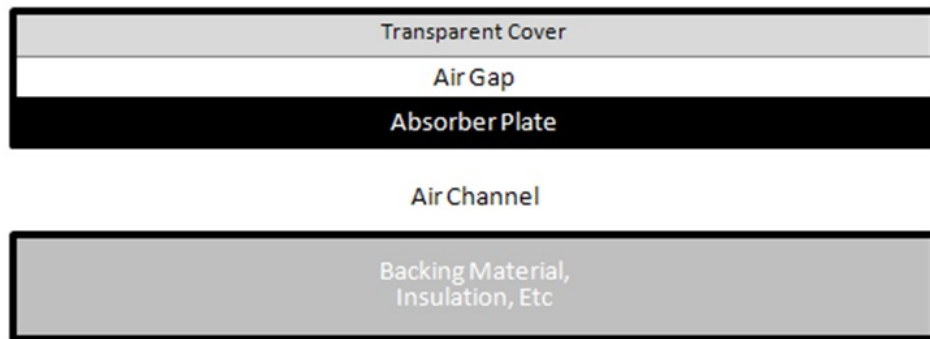


Figure 3.7: Air Heating Collector (section view)

Each module measures approximately 31.89 inches (810mm) wide by 63.39 inches (1610mm) long, which are the same size as the individual PV modules installed in the array.

3.1.3 PV Modules

There are 64 SolarWorld SunModule SW175 Mono PV panels installed in the PV array, with each panel rated at 175W for 11.2kW of maximum total power output at standard test conditions. As shown in Figure 3.8, the module is composed of a transparent, tempered glass cover that is approximately 0.1181 inches (3mm), monocrystalline solar cells embedded in EVA, and a white tedlar backing material. The combined thickness of the PV cells and tedlar backing is approximately 0.0315 inches (0.0008m).

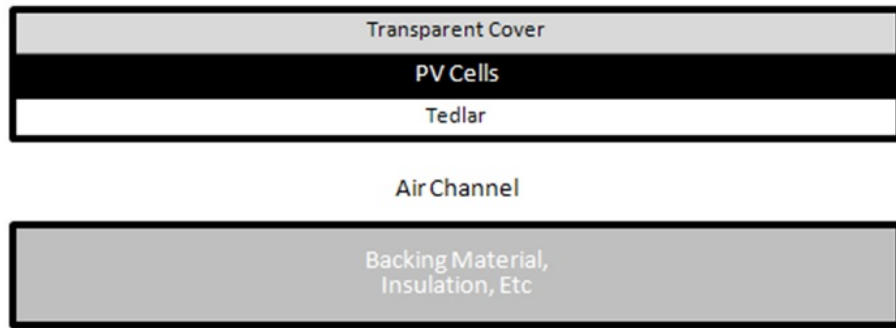


Figure 3.8: PV Module (section view)

The backing materials, air channel size, and overall panel size are the same as in the air heating collectors mentioned previously.

3.1.4 Ducts

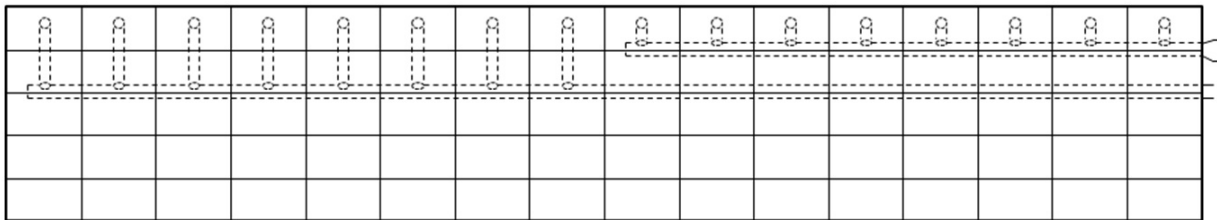


Figure 3.9: PV/T Duct Schematic (Top)

Underneath each air heating collector in the array, there is a 4 inch diameter opening through the pan system and insulation. As shown in Figure 3.9, short 4 inch ducts connect the small openings to a larger 12 inch ducts running along the length of the array. The long 12 inch ducts are then connected to the main 16 inch duct located to the right of the PV/T array, which then leads to the liquid desiccant regenerator. The configuration of the main 16 inch duct is shown in Figure 3.10.

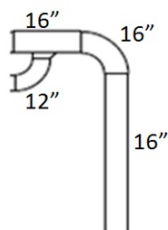


Figure 3.10: Large Duct Configuration

3.2 Liquid Desiccant System

The experimental calcium chloride liquid desiccant system consists of a regenerator/absorber, desiccant storage tank, feed pump, drain pump, re-circulator pump, fluid mixer, interchanger, and all associated pipes and tubing. Figure 3.11 and Figure 3.12 show the arrangement of equipment in the desiccant system.

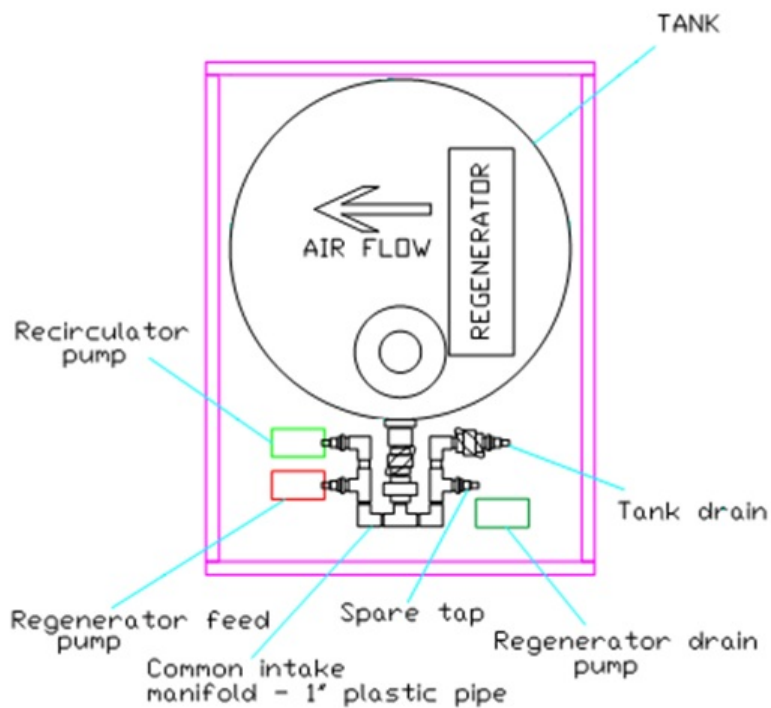


Figure 3.11: Desiccant System (Top)

Figure 3.12 also shows the flow directions for the air and desiccant in the system. As shown in the figure, the regenerator feed pump delivers desiccant to the top of the regenerator by pumping the liquid through the interchanger. The desiccant leaving the regenerator drains out of the bottom of the regenerator and into a tube that leads to the interchanger. Desiccant is then pumped out of the interchanger and into the fluid mixer using the re-circulator pump. The regenerator drain pump takes fluid from the bottom of the tank and delivers it to the other inlet of the fluid mixer. The mixed fluid is then released into the tank, which reduces stratification of the liquid in the tank.

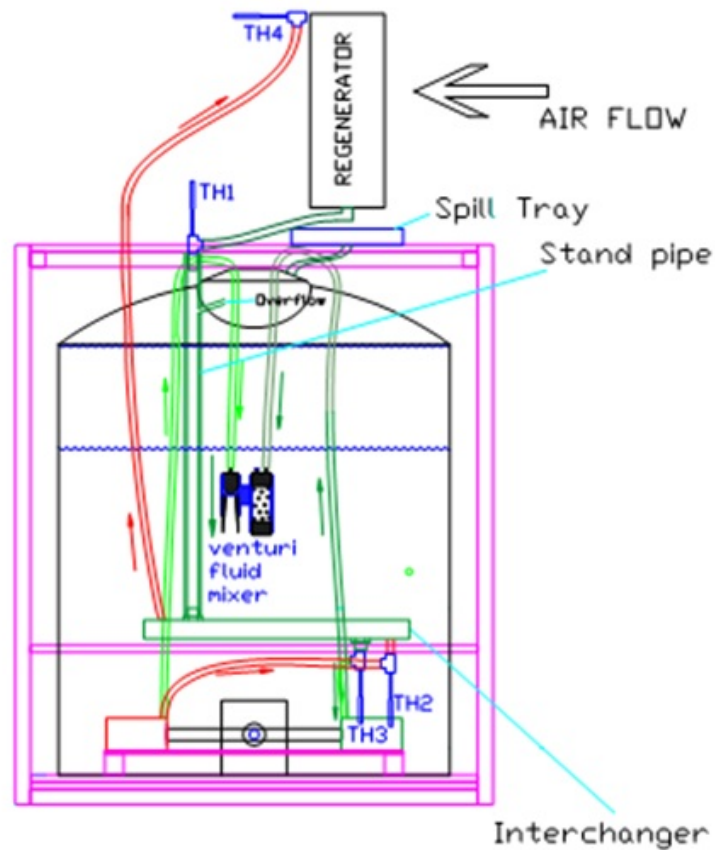


Figure 3.12: Desiccant System (Side)

Air flow into the regenerator is in a cross-flow direction through a duct system coming from the PV/T system.

3.2.1 Regenerator/Absorber

For the ease of ducting, the regenerator installed at the Orange County Convention Center also acts as the absorber, when run in reverse, at night. The device has a stainless steel, enamel-coated housing that measures approximately 2 feet high by 2 feet wide, with a depth of 6.25 inches. There are 75 folded, polyphenylsulfone (PPSU) sheets installed inside the regenerator housing, for an overall width of 22.113 inches for air to pass through. Figure 3.13 shows the constructed regenerator/absorber, before installation.



Figure 3.13: Regenerator/Absorber

Figure 3.14 shows the overall setup of the regenerator in greater detail. As shown in the top view in the figure, the PPSU plates are folded and fitted together inside the regenerator housing. The folds increase the surface area of the plates, which increases regeneration and absorption capacity. The diagram also shows the cross-flow configuration of the air and desiccant, where air enters between the plates and desiccant flows down the plates to create a thin film of desiccant.

Figure 3.13 shows that there are 3 small tube openings on the side of the regenerator housing where desiccant actually enters the regenerator. The calcium chloride desiccant is then delivered across the plates through distribution channels inside the regenerator housing, which allows the desiccant to drip down evenly along the 75 plates.

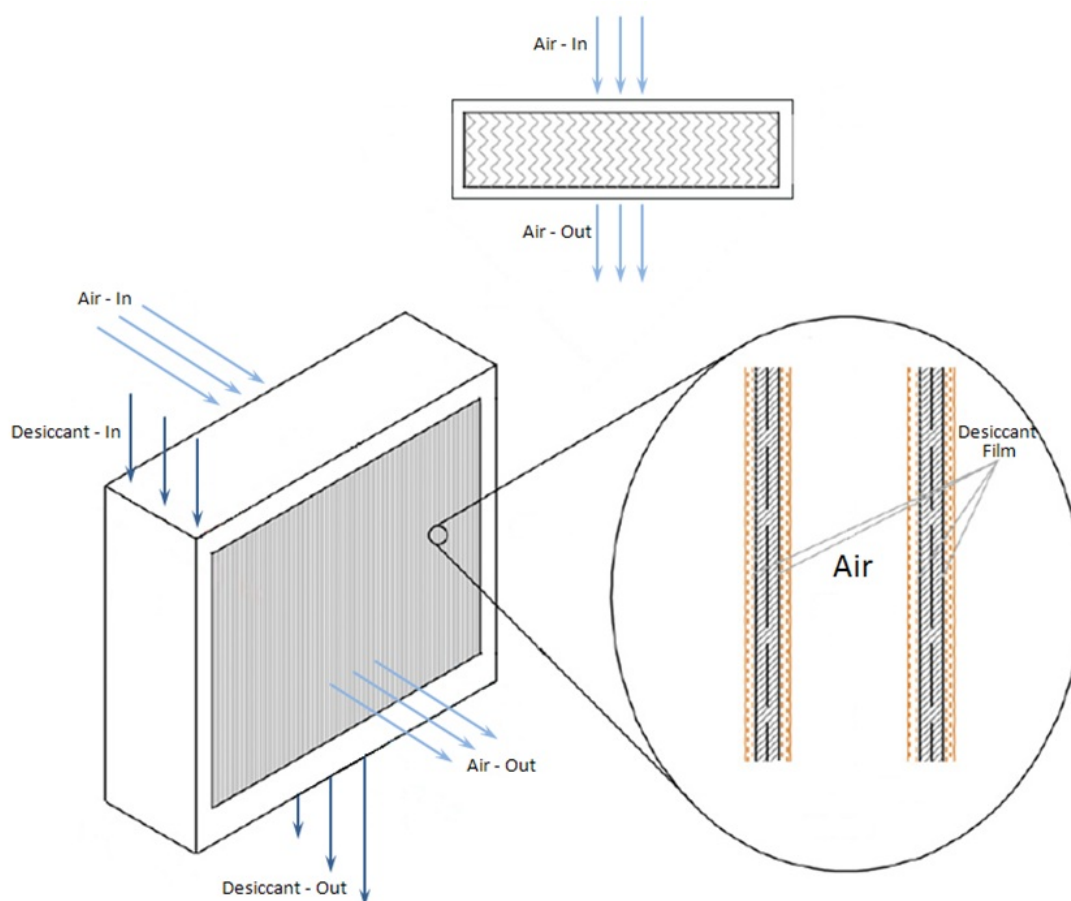


Figure 3.14: Regenerator/Absorber Flow Diagram

3.2.2 Desiccant Storage Tank

The tank used for storing the liquid desiccant is an Ace RotoMold VT0300-42. It is an un-insulated, medium-to-high density, polyethylene tank that is approximately 55 inches in maximum height with a diameter of 42 inches, and is rated to store up to 300 gallons of liquid.



Figure 3.15: Desiccant Storage Tank

3.3 Combined System

The installed, experimental system combines the PV/T and desiccant systems into one system to allow the thermal output from the PV/T to be used as the regeneration source for the liquid desiccant. To combine the systems, additional ductwork, fans, dampers, filters, and a control system were installed. The combined system also requires an automated control system to determine proper operation of the fans, pumps, and dampers. In addition, sensors are necessary to record all relevant data for the experiment, which are connected to and recorded by the control system.

3.3.1 Overall System Setup

The additional equipment allows delivery of hot air from the output of the PV/T system to the regenerator, provides a bypass path around the regenerator (for certain conditions), and takes air from the output of the regenerator to an exhaust vent. Figure 3.16 shows the path of warm

air from the PV/T array through the regenerator to the exhaust vent. The figure also shows the setup of the installed ductwork and equipment used in the complete system. Figure 3.17 shows typical air flow during operation, but also shows the installation of the bypass path around the regenerator along with the additional dampers. Both figures show the overall installation locations of all equipment on the Orange County Convention Center.

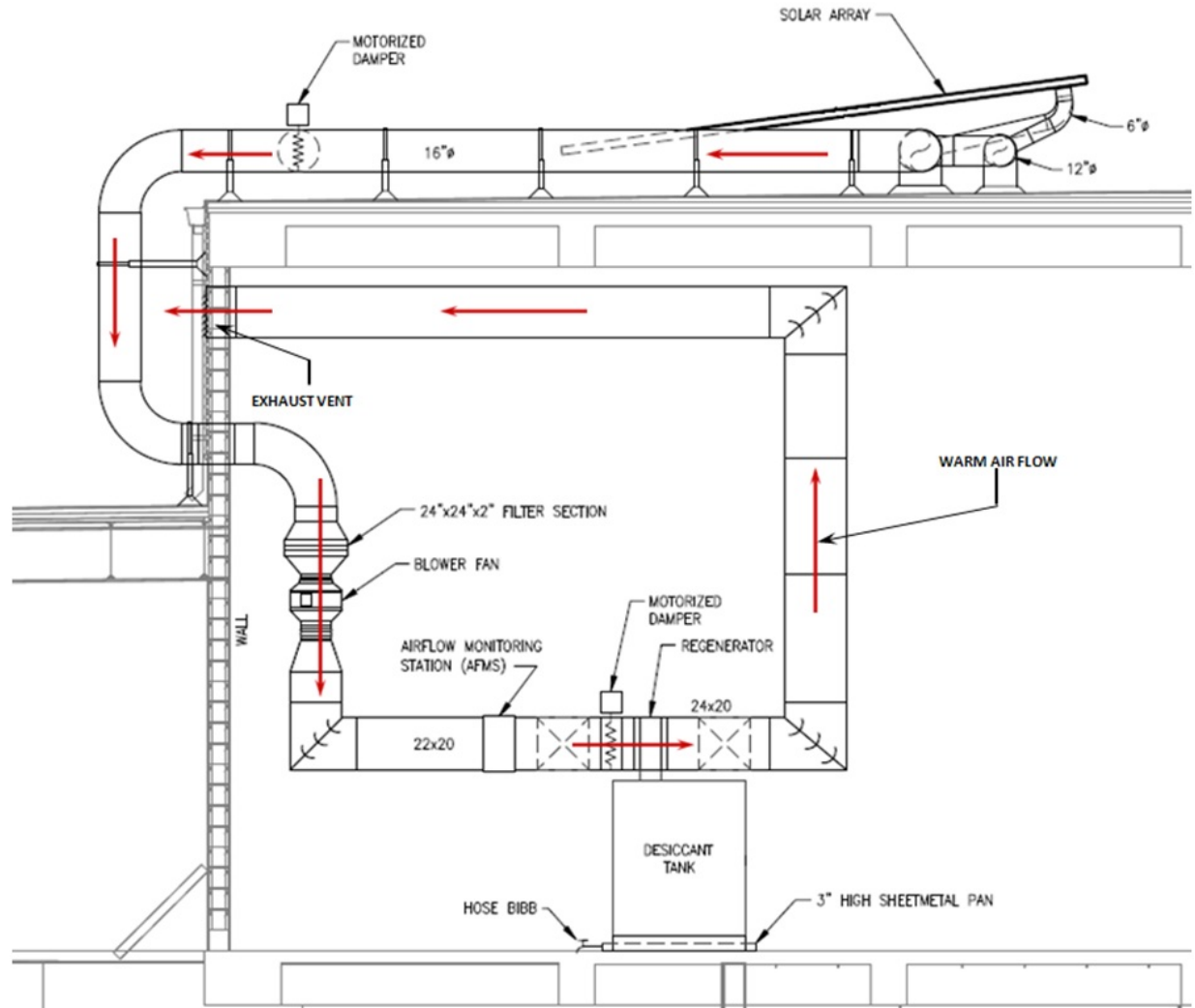


Figure 3.16: Combined PV/T-Desiccant System (Side)

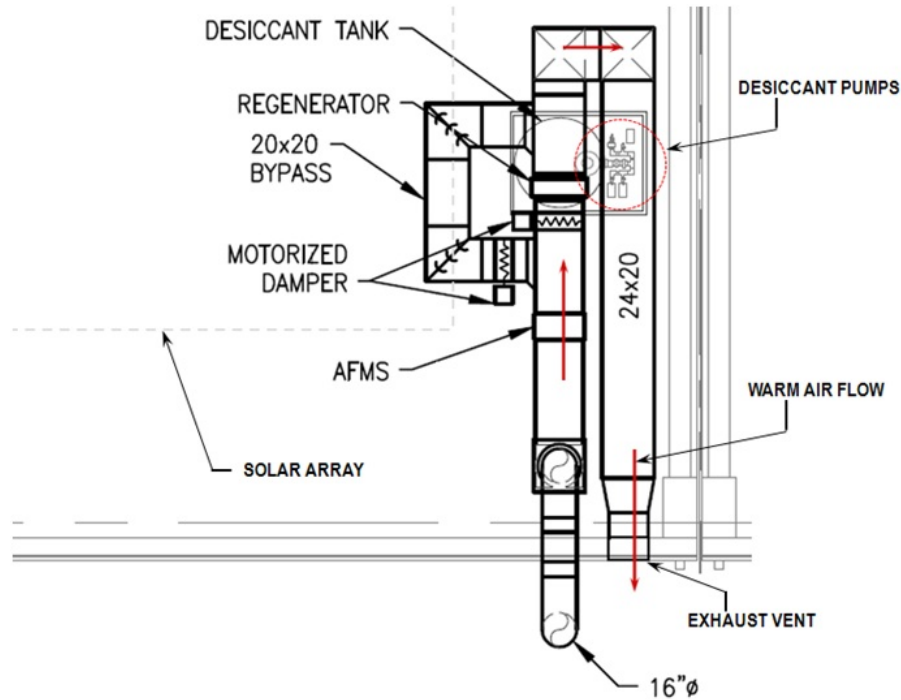


Figure 3.17: Combined PV/T-Desiccant System (Top)

3.3.2 Sensors and Control System

The PVT-1000 control system, designed by EchoFirst, was installed to control the system devices and monitor a variety of signals. The controller has the ability to read 12 RTD inputs and 8 analog inputs, and can control several triac outputs for 120V devices, 24 VAC devices, and digital outputs. A separate Campbell Scientific data logger also monitors the various sensor inputs, and allows for remote web-based viewing of the recorded data. Table 3.1 lists all of the signals simultaneously monitored by the PVT-1000 control system and Campbell Scientific data logger.

All of the temperature measurements use RTD sensors, while the remaining measurements use analog voltage inputs. Most of the sensors and measuring devices are placed in specific locations on the PV/T array or desiccant system. However, the air flow rate is measured using the Ruskin IAQ50X air flow monitoring station, which is installed in-line with the ducting system and the output signal connected to the controller. Also, the pyranometer, wind speed, and ambient temperature sensors are located on a separate weather station adjacent to the PV/T array.

Table 3.1: Monitored Signals

Signal	Units
Voltage - Minimum DAS Battery	V
Voltage - Weather Station Batteries	V
Temperature - Reference	°C
Temperature - Roof/Ambient	°C
Temperature - Air Entering Regenerator	°C
Temperature - Air Leaving Regenerator	°C
Temperature - Desiccant Entering Regenerator	°C
Temperature - Desiccant Leaving Regenerator	°C
Temperature - Desiccant Leaving Heat Exchanger	°C
Temperature - Desiccant Tank	°C
Temperature - PV 1	°C
Temperature - PV 2	°C
Temperature - PV 3	°C
Temperature - PV 4	°C
Air Pressure - Regenerator	V
Tank Level	V
Relative Humidity - Roof/Ambient	%
Relative Humidity - Air Entering Regenerator	%
Relative Humidity - Air Leaving Regenerator	%
Solar Insolation	W/m^2
Wind Speed	m/s

All of the measurements are calculated directly from the sensor readings, except for the desiccant concentration measurement. However, the concentration measurement is the voltage reading from a pressure transducer that indicates the level/height of desiccant in the tank, which

corresponds to a particular concentration of desiccant. Tank level and concentration were manually calculated, and shown by markings on the actual tank. The calculated concentration values are the values used later in the simulation models.

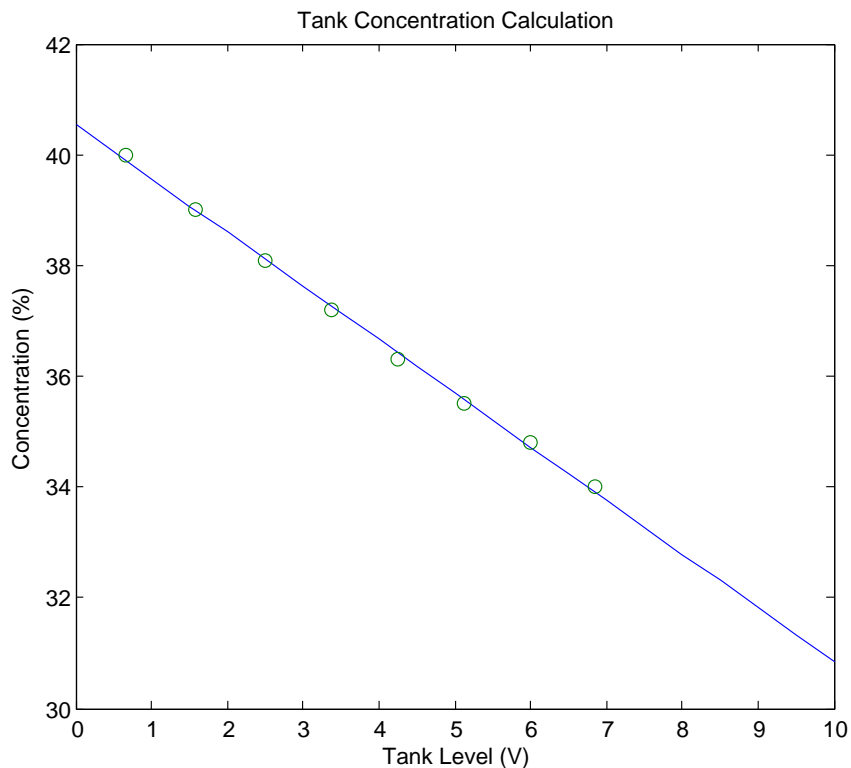


Figure 3.18: Tank Concentration Value

Figure 3.18 shows the linear correlation between transducer voltage and tank concentration, and is also described by the following equation:

$$\xi = -0.9688 * V + 40.537 \quad (3.1)$$

The PVT-1000 system not only records many data channels, it also controls six devices in this system. Three pumps, which are the drain, feed, and re-circulator pumps from the desiccant system, are connected to one power strip and controlled by a single triac output, while the blower fan is controlled by a second triac output. The other two devices connected to the controller are

the exhaust and bypass dampers that are controlled by the 24 VAC outputs.

To generate data for this experimental system setup, a simplified controller algorithm was used to keep the system running constantly day and night. The controller does not distinguish between regeneration and absorption. However, several dampers are used to maintain specified temperature limits within the regenerator and concentration limits of the desiccant tank, which are shown in Table 3.2. The maximum regeneration temperature is limited to avoid damage to the regenerator, but the minimum temperature and tank concentration limits are in place to only allow operation during optimal conditions.

Table 3.2: Control Sequence Description

Controlled Device	Controller Programming
Blower Fan	Run continuously (24/7)
	If PVT outlet temperature exceeds 200°F, open fresh air damper
	If regenerator inlet temperature is less than 130°F, blower fan runs at 50%
	If regenerator inlet temperature is above 180°F, blower runs at 10%
	If regenerator inlet temperature is between 130°F - 180°F, blower modulates to maintain maximum temperature
Regenerator/Bypass Dampers	If regenerator inlet temperature is between 130°F - 180°F and tank level is less than 0.40%, open regeneration damper and close bypass damper
	If regenerator inlet temperature is below 130°F and tank level is more than 0.34%, open regeneration damper and close bypass damper
	If regenerator inlet temperature is above 180°F, close regeneration damper and open bypass damper
Pumps	Run continuously (24/7)
	If pumps turn off for any reason, Pumps run for 2 minutes before opening regeneration damper

Chapter 4

System Modeling

In order to evaluate the prototype PVT-Desiccant system, mathematical models were developed for computer simulation and were validated with the experimental data. The computer simulation tool used in this thesis was TRNSYS, an energy simulation software package developed by the University of Wisconsin. Each component of the PVT-Desiccant system was modeled individually in the TRNSYS tool, and then combined into a theoretical, fully-operational system simulation. Many of the component models were provided by the TRNSYS software package, which includes the fans, pumps, and dampers. The PVT array modeling uses models developed by Casey [9] at the University of Colorado, while the absorber uses component models created by Jones [20] at Queen's University. New or modified models were created for the regenerator, system controllers, and desiccant storage system. The regenerator model is based on the work by Jones [20], but created to model the warm-air, cross-flow regenerator designed for the prototype system in Orlando. Absorber models, with the associated cooling tower and pumps, were previously validated with experimental data by Jones [20] and Andrusiak [3], but the PVT array and fans, regenerator combined with dampers, and the desiccant tank system with associated pumps and valves were modeled to replicate the prototype system described earlier. Combining the various models allows for simulation of a fully operational system that can be used in commercial and possibly some residential applications. Figure 4.1 shows the system diagram for the fully functional system, which is the basis for the TRNSYS model described in this chapter. The same TRNSYS model is also used for all simulations in the baseline simulation analysis and the results/discussion, which are found in

Chapters 6 and 7. The system described in the previous chapter is used for model calibration and validation only. Details of the model calibration are shown in the “Model Calibration” chapter, Chapter 5.

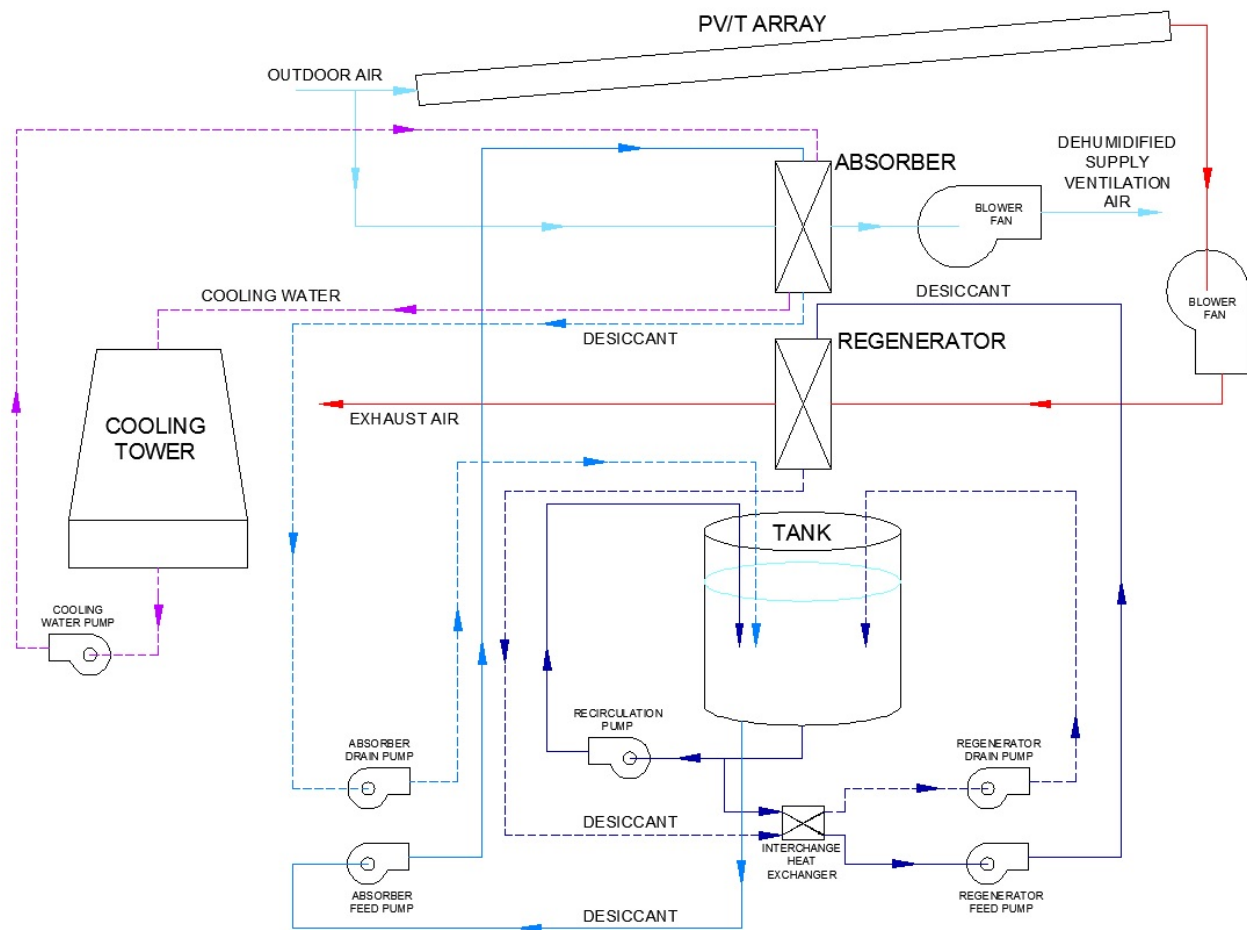


Figure 4.1: Fully Modeled System Diagram (without controls)

4.1 Photovoltaic/Thermal Air Collector System Modeling

Models for a PV/T air collector system were developed by Casey [9] for use in TRNSYS to simulate a prototype PVT system also created by EchoFirst. However, the prototype system used in this thesis requires a different mounting system that is not flush with the roof or integrated into it, which includes different construction and materials for the plenum under the PV/T panels.

Also, different PV panels are used to increase power production from the PV portion of the array. Because of the similarities between the two systems, the Casey models are the ideal choice for simulation of the prototype system examined in this thesis.

To model the PV/T system, Casey created two separate TRNSYS components (TYPES). TYPE 982 models an un-glazed solar collector intended to produce electrical power from PV panels while simultaneously providing heat to an air stream in the plenum below the PV array. TYPE 961 models a glazed solar collector that provides heat to an air stream in the plenum below the absorber plate. For modeling purposes, the two component types must be used together to create the PV/T system, by using the outlet conditions of the Type 982 model as inputs to the Type 961 thermal collector model. Input conditions for the PV portion of the model simply use ambient air conditions and irradiance data, which is provided a Type 15 TRNSYS model using a TMY2 weather file (TMY2). The same irradiance data is also provided to the thermal collector model. Type 999, created by Casey [9], also uses wind data from weather file to provide convection coefficients to the PV/T models. Since a blower fan provides the air flow into the system, a simple variable speed fan model, Type 662 by TRNSYS, is included in the modeling of the complete PV/T system. Figure 4.2 shows the components, as modeled in TRNSYS.

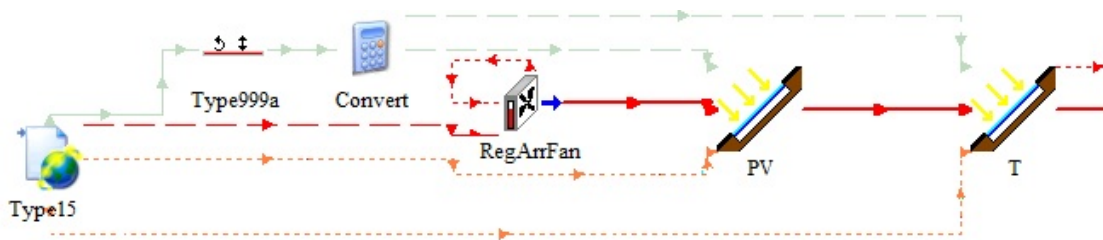


Figure 4.2: TRNSYS Model - PVT System

In addition to the model inputs simulating various conditions, model parameters describe the characteristics of the particular system being modeled. While the system described in the Casey thesis was similar, many modifications were necessary to account for the differences in plenum construction, mounting system, and PV panel selection. The remainder of Section 4.1 describes

the many parameters and calculations necessary to properly model the PV/T system. Because the PV and thermal modules basically consist of a cover, substrate/plate, air channel, and backing, the sections showing these calculations also fall into these same categories, along with PV or thermal modeling specific parameter calculations. Documentation regarding the energy balances and heat transfer equations used in the Type 982 and 961 models can be found in the Casey[9] thesis.

Drawings in Appendix A.1 detail the array and mounting system sizing and materials. The materials used affects various thermal resistance and capacitance parameters, and many of the calculations below take into account those materials. Both the PV and thermal collector modules use the same mounting and pan system, so the calculations for the backing parameters will be the same for both models and are calculated in section 4.1.1.

4.1.1 Mounting System

The mounting system frame is composed of aluminum, however, the pans below the mounting system are made of 26 gauge galvalume sheet metal. Material property information is readily available for galvanized steel, but not for galvalume coated sheet metal. The make-up of the galvalume coating was found to be 55% Aluminum, 43.4% Zinc, and 1.6% Silicon. Using the individual material properties and given metal thickness, the properties of the sheet metal were found in Table 4.1.

Table 4.1: Galvalume Sheet Metal Material Properties and Calculations

Material	%	Specific Heat	Density	Thermal Conductivity	Thickness	Resistance	Reference
	(%)	(kJ/kg * K)	(kg/m ³)	(kJ/hr*m*K)	(m)	(hr * m ² * K/kJ)	
Aluminum	55	0.896	2740	221	0.0000275	0.000000034565	[4]
Zinc	43.4	0.39	7130	110	0.0000217	0.000000054798	[4]
Silicon	1.6	0.71	2330	150	0.0000008	0.0000000014815	[4]
Steel	–	0.5	7830	45.3	0.00045	0.0000027594	[4]
Aluminum	55	0.896	2740	221	0.0000275	0.000000034565	[4]
Zinc	43.4	0.39	7130	110	0.0000217	0.000000054798	[4]
Silicon	1.6	0.71	2330	150	0.0000008	0.0000000014815	[4]
Galvalume - Total		0.53153	7249.76	–	0.00055	0.000002941	

As shown in Appendix A.1, the back of the pan system that encloses the air channel is composed of a layer of 26 gauge galvalume sheet metal, however the Dow Thermax polyisocyanurate insulation is not used in Orlando. A layer of Tiger Foam spray insulation was also added to the underside of PV/T array, which varies in thickness but has an assumed average value of 2 inches (0.0508m). The overall material properties for the entire backing of the array are calculated in Table 4.2. The values for several of the backing parameters are used directly for the PV and thermal collector models, and also indirectly in calculations for the substrate.

Table 4.2: Array Backing Material Properties and Calculations

Material	Specific Heat	Density	Thickness	Resistance	Reference
	(kJ/kg * K)	(kg/m ³)	(m)	(hr * m ² * K/kJ)	
Galvalume Sheet Metal	0.53153	7249.76	0.00055	0.000002941	Table 4.1
Exterior Insulation	1.47	28	0.069	0.53333	[4] and ?? and A.1
Backing - Total	1.4627	94.1223	0.0704	0.80316	

4.1.2 Thermal Modules

The top layers of the thermal collector modules, above the air channel, consist of a transparent cover, small air gap, and absorbing plate, as shown in Figure 3.7. The glass cover for the

thermal collectors is made of low-iron, tempered glass, with high transmittance and low absorptance. Average or typical emissivity (ε), absorptance(α), and transmittance(τ) values from various glass manufacturers are assumed for the cover parameters. The copper absorbing plate is sprayed with Solkote coating, so average values for emissivity and absorptance are also assumed for the absorbing plate surface. Also, the back of the absorbing plate acts as the upper air channel, so the same surface properties are used. The model also requires emissivity values for the top and bottom of the backing. The top of the backing, or lower air channel surface is galvalume sheet metal, and the emissivity is assumed to be the same as galvanized steel. Tiger foam coated in UV inhibiting paint covers the back of the pan system, and a typical value for white paint is used in the model. All emissivity, absorptance, and transmittance values necessary for modeling are summarized in Table 4.3.

Table 4.3: Thermal Module Surface and Optical Properties

Thermal Modules			
Surface	ε	α	τ
Cover	0.86	0.04	0.916
Absorbing Plate	0.345	0.895	—
Channel - Upper	0.345	0.895	—
Channel - Lower	0.25	—	—
Backing	0.916	—	—

Multiplying the absorptance value of the absorbing plate by the transmittance of the glass cover gives the normal transmittance-absorptance product, or $(\tau\alpha)_n$, which is 0.828. Since $(\tau\alpha)$ depends on the angle of incidence of radiation on the collector, an incidence angle modifier expression is also used in the model. The incidence angle modifier coefficient, as part of the expression, is also referred to as the “Incidence Angle Modifier” parameter in the Type 982 model. From Duffie and Beckman [14], the incidence angle modifier coefficient was found to be 0.136 for a single-cover collector.

Table 4.4: Type 961 Substrate Calculations

Type 961						
Material	Density (kg/m^3)	Specific Heat ($kJ/kg * K$)	Thickness (m)	Thermal Conductivity ($W/m * K$)	Heat Capacity ($kJ/m^2 * K$)	Thermal Resistance ($hr * m^2 * K/kJ$)
Copper Absorbing Plate	8910	0.39	0.0002	393	0.695	0.0000001414
Aluminum Frame	2740	0.896	0.00075	–	1.841	–
Substrate - Total	4038.95	0.7895	0.00095	–	1.42327	0.0000001414

Type 961 requires that the thermal capacity of the aluminum frame is calculated as part of the substrate material. For this calculation, an effective thickness for the entire frame is calculated by dividing the cross-sectional area of the aluminum frame by the array width [9]. As shown in Table 4.4, the overall properties of the substrate are calculated based on the individual material thicknesses and properties.

All of the necessary parameter values for Type 961 are summarized in Table 4.5.

Table 4.5: Type 961 Parameter List

Type 961			
Parameter	Value	Units	Reference
Collector Length	0.81	<i>m</i>	A.1
Collector Width	25.76	<i>m</i>	A.1
Cover Emissivity	0.84	–	Table 4.3
Cover Thickness	0.003	<i>m</i>	[9]
Cover Density	2470	<i>kg/m³</i>	Table 4.4
Cover Specific Heat	0.75	<i>kJ/kg * K</i>	Table 4.4
Cover Absorptance	0.0402	–	Table 4.3
Cover- Plate Spacing	0.02	<i>m</i>	[9]
Plate Emissivity	0.345	–	Table 4.3
Substrate Resistance	0.0000001414	<i>hr * m² * K/kJ</i>	Table 4.4
Plate Thickness	0.0002	<i>m</i>	[9]
Plate Density	4038.95	<i>kg/m³</i>	Table 4.4
Plate Specific Heat	0.78947	<i>kJ/kg * K</i>	Table 4.4
Upper Air Channel Emissivity	0.345	–	Table 4.3
Lower Air Channel Emissivity	0.25	–	[4]
Back Resistance	0.81185	<i>hr * m² * K/kJ</i>	Table 4.2
Back Thickness	0.0704	<i>m</i>	Table 4.2
Back Density	94.4321	<i>kg/m³</i>	Table 4.2
Back Specific Heat	1.46267	<i>kJ/kg * K</i>	Table 4.2
Back Emissivity	0.9	–	Table 4.2
Channel Thickness	0.04826	<i>m</i>	A.1
Normal Transmittance Absorptance	0.82802	–	Table 4.3
Incidence Angle Modifier Multiplier	0.136	–	[14]

Parameters not calculated earlier in this section were determined from system drawings or from the Casey[9] documentation.

4.1.3 PV Modules

The PV modules, above the air channel, consist of a transparent cover, PV cells and EVA encapsulant, and a white tedlar backing, as shown in Figure 3.8. As in the thermal modules, the glass cover is made of low-iron, tempered glass. Zondag et al.[36] describes typical emissivity values for PV glass and PV cells/EVA. Typical values from various PV module manufacturers are assumed for absorptance and transmittance. White tedlar backing forms the upper part of the air channel, so surface properties for the tedlar are used. The PV model also requires emissivity values for the top and bottom of the backing, which are the same values as in the thermal collector model. The $\tau\alpha$ product is calculated in same way as in the thermal collector model parameters, and found to be 0.85115.

Table 4.6: PV Module Surface and Optical Properties

PV Modules			
Surface	ε	α	τ
Cover	0.86	0.04	0.916
PV/EVA	0.86	0.92	—
Channel - Upper	0.87	0.895	—
Channel - Lower	0.25	—	—
Backing	0.916	—	—

Table 4.7: Type 982 Substrate Calculations

Type 982						
Material	Density (kg/m^3)	Specific Heat ($kJ/kg * K$)	Thickness (m)	Thermal Conductivity ($W/m * K$)	Heat Capacity ($kJ/m^2 * K$)	Thermal Resistance ($hr * m^2 * K/kJ$)
PV Cells, EVA	2330	0.677	0.0003	0.1404	0.473	0.00059354
Tedlar	1200	1.25	0.0005	0.1188	0.750	0.0011691
Aluminum Frame	2740	0.896	0.00075	—	1.841	—
Substrate - Total	1623.75	1.035125	0.0008	—	2.359125	0.0017626

Type 982 also requires that the thermal capacity of the aluminum frame is calculated as part

of the substrate material. An effective thickness for the entire frame is calculated by dividing the cross-sectional area of the aluminum frame by the array width [9], as calculated for the thermal collector model. Table 4.7 shows the PV module substrate material properties and associated calculations.

In addition to the parameters calculated for the Type 961 thermal collector model, the Type 982 PV model requires parameters regarding electrical generation and efficiency. The PV module data sheet, contains all information necessary for determining the PV model parameters, and is summarized in Table 4.8.

Table 4.8: PV Panel Characteristics

Parameter	Name	Value	Unit
Collector Length	L_c	1.61	m
Collector Width	W_c	0.81	m
Collector Area	A_c	1.3041	m^2
Reference Temperature (STC)	T_{ref}	25	$^{\circ}C$
Reference Radiation (STC)	G_{ref}	3600	$kJ/hr * m^2$
Maximum Power Point Voltage	V_{mpp}	35.8	V
Maximum Power Point Current	I_{mpp}	4.89	A
Open Circuit Voltage	V_{oc}	44.4	V
Short Circuit Current	I_{sc}	5.3	A
I_{sc} Current Temperature Coefficient	I_{TC}	0.036	$A/^{\circ}C$
V_{oc} Voltage Temperature Coefficient	V_{TC}	-0.14652	$V/^{\circ}C$
Efficiency Reduction Rate	η_{RR}	5	$\%$
Efficiency Reduction Difference	ΔG	800	W/m^2

Equations 4.1, 4.2, and 4.3 are used to calculate the PV reference efficiency, η_{ref} , efficiency temperature modifier, Eff_T , and efficiency radiation modifier, Eff_G , respectively. Reference temperature and radiation parameters are standard values given in the data sheet, and are considered standard test conditions. PV reference efficiency refers to the efficiency at those standard con-

ditions. The temperature efficiency modifier describes the efficiency of the PV panels at varying temperatures, in relation to that reference efficiency. The radiation efficiency modifier provides the same function, but refers to varying solar radiation levels.

$$\eta_{ref} = \frac{V_{mpp}I_{mpp}}{A_c G_{ref}} \quad (4.1)$$

$$Eff_T = \frac{V_{mpp}I_{TC} + I_{mpp}V_{TC}}{V_{mpp}I_{mpp}} \quad (4.2)$$

$$Eff_G = \frac{\eta_{RR}}{\Delta G} \quad (4.3)$$

The efficiency of the PV panels under standard operating conditions was found to be 0.13424, or approximately 13.4%. Also, the efficiency decreases by 0.37025% for every 1°C increase in temperature, so the efficiency temperature modifier is -0.0037025. Efficiency increases with increased radiation, at a rate of 0.001736% for each $kJ/hr * m^2$.

All of the necessary parameter values for Type 982 are summarized in Table 4.9. Parameters not calculated earlier in this section were determined from system drawings or from the Casey[9] documentation.

Table 4.9: Type 982 Parameter List

Type 982			
Parameter	Value	Units	Reference
Collector Length	3.24	m	A.1
Collector Width	25.76	m	A.1
Cover Emissivity	0.86	–	Table 4.6
Cover Conductance	3.6	$kJ/hr * m * K$	[4]
Cover Thickness	0.003	m	[9]
Cover Density	2470	kg/m^3	[4]
Cover Specific Heat	0.75	$kJ/kg * K$	[4]
Cover Absorptance	0.04	–	Table 4.6
Substrate Resistance	0.001763	$hr * m^2 * K/kJ$	Table 4.7
Plate Thickness	0.0008	m	Table 4.7
Plate Density	2163.87	kg/m^3	Table 4.7
Plate Specific Heat	0.9678	$kJ/kg * K$	Table 4.7
Upper Air Channel Emissivity	0.87	–	[9]
Lower Air Channel Emissivity	0.25	–	[4]
Back Resistance	0.81185	$hr * m^2 * K/kJ$	Table 4.2
Back Thickness	0.0704	m	Table 4.2
Back Density	94.4321	kg/m^3	Table 4.2
Back Specific Heat	1.46267	$kJ/kg * K$	Table 4.2
Back Emissivity	0.9	–	Table 4.2
Channel Thickness	0.04826	m	A.1
Normal Transmittance Absorptance	0.85115	–	Table 4.6
Incidence Angle Modifier Cell Type	Mono	–	[9]
PV Reference Efficiency	0.13424	–	Eqn. 4.1
PV Reference Temperature	25	$^{\circ}C$	Table 4.8
PV Reference Radiation	3600	$kJ/hr * m^2$	Table 4.8
PV Efficiency Temperature Modifier	-0.003702553	$1/^{\circ}C$	Eqn. 4.2
PV Efficiency Radiation Modifier	0.00001736	$hr * m^2/kJ$	Eqn. 4.3

4.1.4 Wind Convection Calculations

Both the PV and thermal module models require the convection coefficients for the front and back of the modules, which is calculated from the wind speed near the PV/T array. There are a number of correlations for calculating convection coefficients from wind speed. Many of the correlations predict convection coefficients for objects/plates of different sizes, buildings of specific sizes, and when wind speed is measured away from the building or surface. The PV/T array in the experimental system is not on a regular size building that is used in the correlations. Also, the wind speed measurements are taken very close to the surface of the PV/T array. Since most correlations are not appropriate for this particular system, the simple McAdams [25] wind convection correlation is used for the modeling of the experimental system. McAdams [25] uses Equation 4.4 to calculate the convection coefficient from wind velocity.

$$h = 5.7 + 3.8 * v \quad (4.4)$$

4.2 Liquid Desiccant System

Liquid desiccant systems are generally composed of a regenerator, absorber, and interchange heat exchanger. A desiccant storage tank can be added so that the absorber and regenerator can operate independently. The prototype liquid desiccant system operating in Orlando contains a combined absorber/regenerator, heat exchanger, and storage tank. Since the regenerator and absorber are combined into a single unit for the prototype system, absorption and regeneration are not independent. However, the model of the full PV/T-Desiccant system combines an independent regenerator and absorber, along with a heat exchanger and desiccant storage tank. The model provides fully independent operation of both the absorber and regenerator that uses the same tank for desiccant storage. Different models are used for the regenerator and absorber, and are described in sections 4.2.1 and 4.2.2.

4.2.1 Absorber

A validated absorber model was developed by Jones [20] for a cross-flow absorber, which is TRNSYS Type 974. A cooling tower model, Type 51a, and cooling water pump, Type 3, is also used to model the cooling tower and water provided to the absorber in order to remove the heat generated by the absorption of water vapor into the desiccant. The parameters necessary for modeling this portion of the system were provided in the validated model, provided by Andrusiak et al. [3]. A diagram of the modeled components related to the absorber is shown in Figure 4.3.

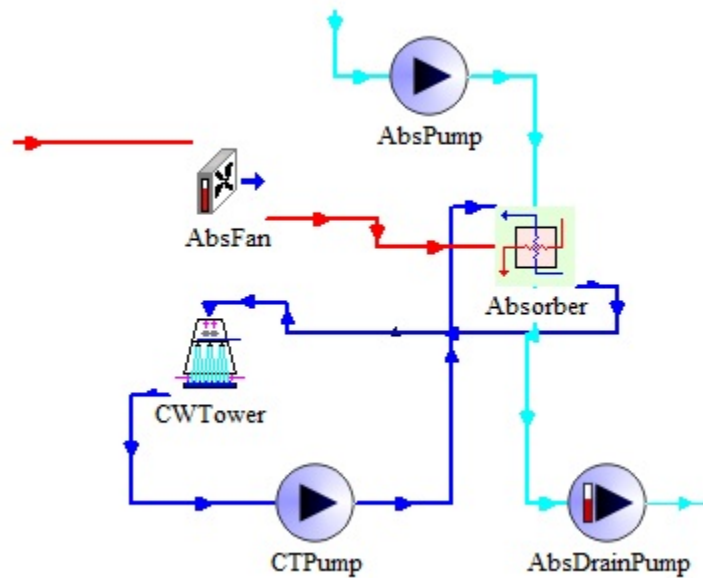


Figure 4.3: TRNSYS Model - Absorber Components

The absorber model is based on three effectiveness equations. Using the given inlet air and desiccant conditions, the effectiveness model determines all of the necessary outlet conditions, including temperatures and flow rates. Equation 4.5 is the humidity ratio effectiveness, or dehumidification effectiveness, which describes the ratio of humidity ratio reduction across the absorber to the minimum possible humidity ratio exiting the absorber.

$$\varepsilon_w = \frac{\omega_{in} - \omega_{out}}{\omega_{in} - \omega_{min}} \quad (4.5)$$

ω_{in} and ω_{out} are the inlet and outlet humidity ratios of the absorber, respectively. ω_{min} refers to the minimum possible humidity leaving the absorber, which is the humidity ratio of the air that is at the same vapor pressure as the desiccant solution at the cooling water temperature. Vapor pressure, and other calcium chloride property calculations are provided by the equations found in the Conde [12] article.

Equation 4.6 is the enthalpy effectiveness, which describes the ratio of enthalpy reduction across the absorber to the minimum possible enthalpy exiting the absorber.

$$\varepsilon_h = \frac{h_{in} - h_{out}}{h_{in} - h_{min}} \quad (4.6)$$

h_{in} and h_{out} are the inlet and outlet air enthalpies of the absorber, respectively. h_{min} refers to the minimum possible enthalpy leaving the absorber, which is the enthalpy of the air at the same vapor pressure as the desiccant solution.

Equation 4.7 refers to the desiccant-cooling water heat transfer effectiveness, which describes the amount of energy transferred from the desiccant to the cooling water in order to calculate the desiccant temperature leaving the absorber.

$$\varepsilon_w = \frac{T_{d,in} - T_{d,out}}{T_{d,in} - T_{cw,in}} \quad (4.7)$$

$T_{d,in}$ and $T_{d,out}$ are the inlet and outlet desiccant temperatures, respectively. $T_{cw,in}$ is the temperature at which the cooling water enters the regenerator.

Andrusiak et al.[3] found the humidity ratio effectiveness value to be a constant 0.63, and the enthalpy effectiveness value to be 0.45. Cooling water effectiveness was found to be a function of the entering air humidity ratio, and described by the following equation:

$$\varepsilon_{cw} = -12.5\omega_{amb} + 0.9009 \quad (4.8)$$

Entering air humidity ratio is the same as the ambient air humidity ratio, because outside air is being drawn into the absorber.

Using the effectiveness equations and basic psychrometric calculations, all of the outlet air properties can be determined. Outlet cooling water temperature, desiccant flow, and desiccant concentration are based on mass and energy balance equations. Equations 4.9, 4.10 and 4.11 are used to calculate the mass flow rates of the desiccant solution, which also allows for calculation of concentration. Equations 4.12 and 4.13 calculate the cooling water temperature leaving the regenerator.

$$\dot{m}_{abs} = \dot{m}_a(\omega_{in} - \omega_{out}) \quad (4.9)$$

$$\dot{m}_{LDout} = \dot{m}_{LDin} + \dot{m}_{abs} \quad (4.10)$$

$$\xi_{out} = \frac{\dot{m}_{sd}}{\dot{m}_{LDout}} \quad (4.11)$$

In Equation 4.9, \dot{m}_{abs} is the rate of moisture removed from the air stream leaving the regenerator. The outlet desiccant flow rate, \dot{m}_{LDout} , is then calculated as the sum of the incoming desiccant flow and the amount of moisture removed from the air stream. Leaving desiccant concentration, ξ_{out} , is found from the previously calculated outlet desiccant flow rate and the entering salt flow rate, which is also a known inlet condition.

$$Q_a = \dot{m}_a(h_{in} - h_{out}) \quad (4.12)$$

$$T_{cwout} = \frac{Q_a}{\dot{m}_{cw}C_{p_{cw}}} + T_{cw_{in}} \quad (4.13)$$

Heat transferred into the cooling water is referred to as Q_a and is used to calculate the leaving cooling water temperature T_{cwout} , where $C_{p_{cw}}$ is the heat capacity of the water. The $C_{p_{cw}}$ value used for the simulations is $4.183 \text{ kJ/kg} \cdot \text{K}$.

The cooling tower model, with associated fan and pump, are standard TRNSYS models. Performance characteristics and sizing information were obtained from the validated Andrusiak [3]

model, and power values were chosen based on typical values for fans and pumps at the given flow rates. Since the Andrusiak [3] model was validated for a specific sized system, the absorber, cooling tower, and associated fans/pumps in the model for this thesis were scaled appropriately to different sizes to maintain the same effectiveness and performance characteristics of the original system.

4.2.2 Regenerator

A cross-flow regenerator model, TRNSYS Type 937, was created specifically to model the regenerator/absorber installed in Orlando. Data generated from the experimental system in Orlando was used to validate this model. Type 937 is based on the absorber model detailed in the previous section, but it has been modified and no heating/cooling water is modeled. A diagram of the modeled components related to the regenerator is shown in Figure 4.4.

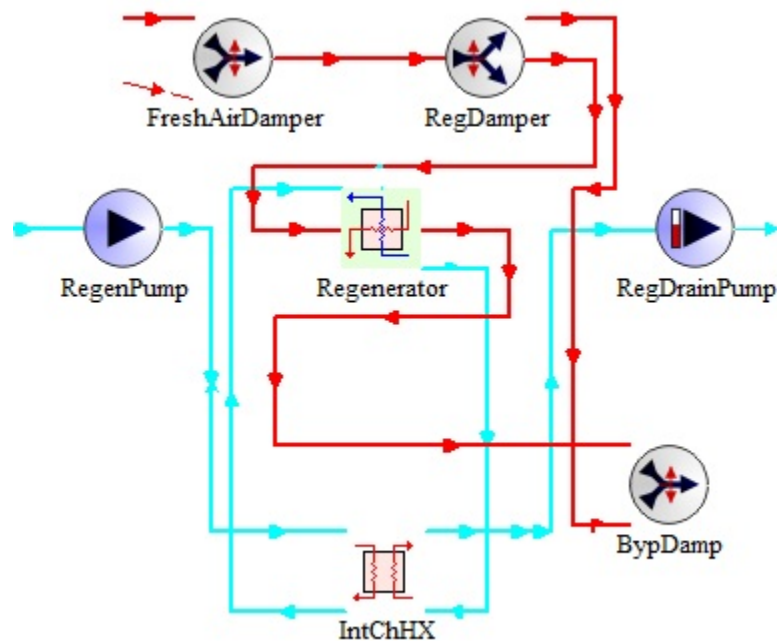


Figure 4.4: TRNSYS Model - Regenerator Components

The regenerator model is based on two effectiveness equations. As in the absorber model, the regenerator model uses the given inlet air and desiccant conditions, and determines all of the necessary outlet conditions, including temperatures and flow rates. Equation 4.14 is the humidity

ratio effectiveness, which describes the ratio of humidity ratio increase across the regenerator to the maximum possible humidity ratio exiting the absorber.

$$\varepsilon_w = \frac{\omega_{out} - \omega_{in}}{\omega_{max} - \omega_{in}} \quad (4.14)$$

ω_{in} and ω_{out} are the inlet and outlet humidity ratios of the regenerator, respectively. ω_{max} refers to the maximum possible humidity leaving the regenerator, which is the humidity ratio of the air that is at the same vapor pressure as the desiccant solution at the temperature of the entering air.

Equation 4.15 is the enthalpy effectiveness, which describes the ratio of enthalpy change across the regenerator to the maximum possible enthalpy change exiting the regenerator.

$$\varepsilon_h = \frac{h_{in} - h_{out}}{h_{in} - h_{eq}} \quad (4.15)$$

h_{in} and h_{out} are the inlet and outlet air enthalpies of the regenerator, respectively. h_{eq} refers to the maximum possible enthalpy change leaving the regenerator, which is the enthalpy of the air at the same vapor pressure as the desiccant solution. The enthalpy of the air leaving the regenerator can be more or less than the entering air, depending on ambient air conditions.

Using the effectiveness equations and basic psychrometric calculations, all of the outlet air properties can be determined. Outlet desiccant temperature, flow, and concentration are based on mass and energy balance equations. Equations 4.16, 4.17 and 4.11 are used to calculate the mass flow rates of the desiccant solution, which also allows for calculation of concentration. The concentration equation used is the same as that of the absorber. Equations 4.18 and 4.19 determine the desiccant temperature leaving the regenerator.

$$\dot{m}_{des} = \dot{m}_a(\omega_{out} - \omega_{in}) \quad (4.16)$$

$$\dot{m}_{LDout} = \dot{m}_{LDin} - \dot{m}_{des} \quad (4.17)$$

In Equation 4.16, \dot{m}_{des} is the rate of moisture added to the air stream leaving the regenerator. The outlet desiccant flow rate, \dot{m}_{LDout} , is then calculated as the difference between the incoming desiccant flow and the amount of moisture passed to the air stream. Leaving desiccant concentration, ξ_{out} , is found from the previously calculated outlet desiccant flow rate and the entering salt flow rate, which is also a known inlet condition.

$$Q_a = \dot{m}_a(h_{out} - h_{in}) \quad (4.18)$$

$$T_{dout} = \frac{\dot{m}_{LDin} C_{pd} T_{din} - Q_a}{\dot{m}_{LDout} C_{pd}} \quad (4.19)$$

Heat transferred between the desiccant and the air is referred to as Q_a and is used to calculate the outlet desiccant temperature T_{dout} , where C_{pd} is the heat capacity of the desiccant. The C_{pd} value used for the simulations $2.5833 \text{ kJ/kg} \cdot \text{K}$, which is an average value across typical concentrations and temperatures.

Figure 4.5 shows typical entering air and desiccant conditions of a regenerator on the psychrometric chart. Entering air and desiccant work to approach an equilibrium point that is somewhere between the two temperatures and enthalpy states, which is indicated by the red box. The final state of the air will be much cooler with a lower enthalpy, but humidity ratio will increase. The humidity ratio effectiveness is based on the maximum humidity that can be added, which is located in the top corner of the red box near 0.023 kg/kg and 36.5°C . Enthalpy effectiveness indicates the amount that the enthalpy decreases in relation to the lowest possible enthalpy state, indicated by the lower line of the red box. For the example in the figure, the leaving air state for regeneration will be near 0.02 kg/kg and 37°C ., with an increase in humidity ratio and decrease in enthalpy.

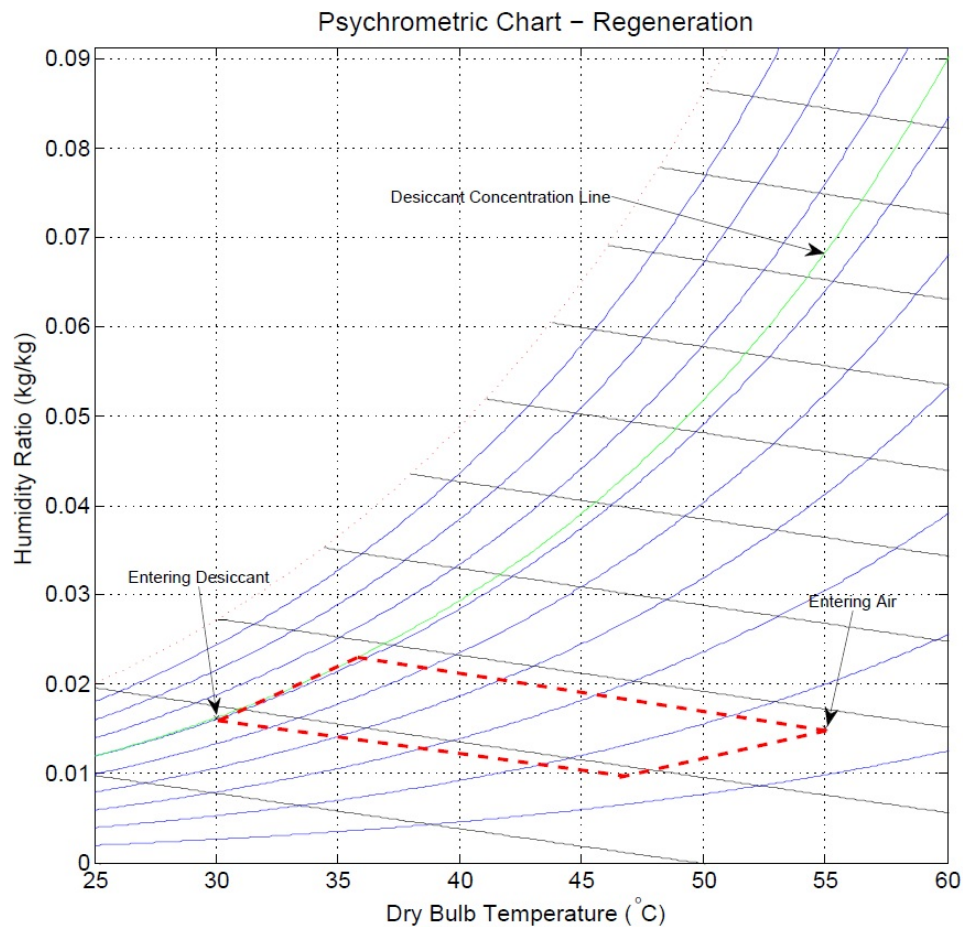


Figure 4.5: Regeneration on the Psychrometric Chart

Performance characteristics, such as effectiveness values, were determined from experimental data from the Orlando system and are detailed in the next chapter. Since the regenerator was validated for a specifically sized system, the regenerator and associated fans/pumps in the model for this thesis were also scaled to maintain the same effectiveness and performance characteristics of the original system. Fan and pump power are based on typical values for the appropriately sized system.

Included with the regenerator system in Orlando, is a heat exchanger that is intended to pre-heat the desiccant stream entering the regenerator using the warmer desiccant leaving the regenerator. Desiccant leaving the tank to go to the inlet of the regenerator passes through the heat exchanger. The other stream entering the heat exchanger is desiccant leaving the regenerator

and going to the tank. To model the heat exchanger, TRNSYS Type 5b is used. The model is a simple counter-flow heat exchanger that calculates an effectiveness value that is based on an overall heat transfer coefficient. The default overall heat transfer coefficient of $10 \text{ kJ/hr} * K$ is used, because the temperature difference between the two desiccant streams is generally negligible and since limited data was available for fully calculating a heat transfer coefficient.

4.2.3 Desiccant Storage

Liquid desiccant storage allows the regenerator and absorber to operate independently of the other. By storing the desiccant in a tank, the absorber can run at any time of day, including during the night when the regenerator will not be running. For storage of the desiccant, a large, un-insulated plastic tank can be used. The system in Orlando uses an Ace Roto-Mold 300-gallon vertical tank made from high-density polyethylene. Due to the varying volume of water levels in the tank, a variable-volume tank, TYPE 39, is used for thermal modeling of the desiccant tank. The model has one inlet and one outlet that can operate at different flow rates. A diagram of the modeled components related to the desiccant storage is shown in Figure 4.6.

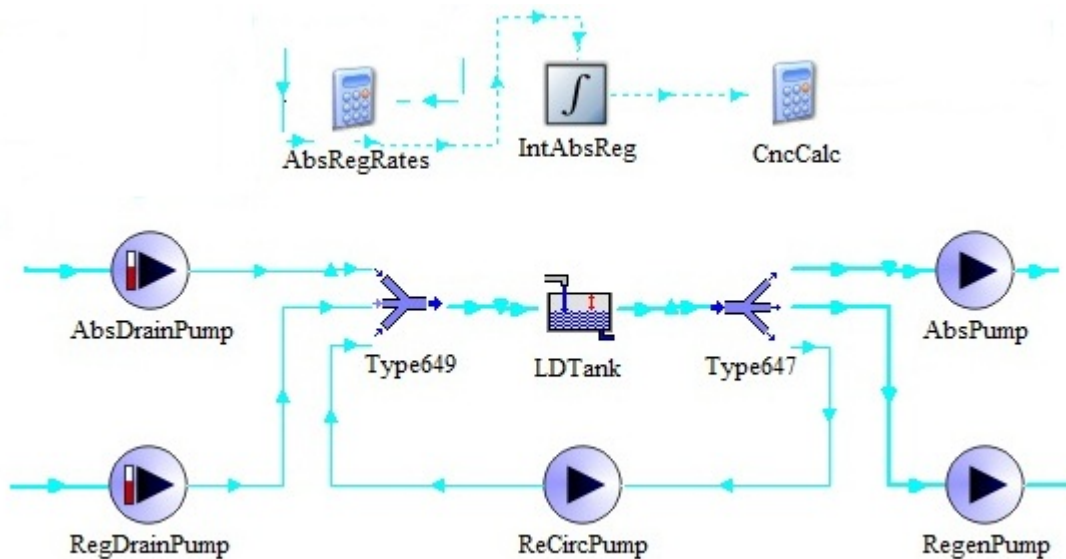


Figure 4.6: TRNSYS Model - Tank Components

Most of the model parameters simply specify the capacity and dimensions of the tank. For modeling of the tank in Orlando, the dimensions were specified as 42 inches in diameter and 55 inches in height, with a minimum capacity of 0 gallons and a maximum of 300 gallons. Parameters for cross-sectional area and circumference are calculated from the given dimensions. For the parametric studies performed in this thesis, the tank capacity and dimensions are varied for other regenerator and absorber sizes. Parameters for the specific heat and density of the the liquid in the tank are also necessary, so the average/typical values of desiccant are used, as mentioned previously. The other parameters used for the tank model are the dry and wet loss coefficients, which allow the calculation of heat loss or gain from the tank. All variations of the model assume that a high-density polyethylene tank with a wall thickness near 1/2" is used. The loss coefficient value of $55.534 \text{ kJ/hr} * m^2 * K$ is calculated from the tank wall thickness and material properties for polyethylene in the ASHRAE Handbook of Fundamentals[4] .

The variable-volume tank model does not include calculations for changes in concentration, so it only provides modeling of the overall liquid volume and desiccant temperature changes. Concentration calculations are done externally of the tank model using simple equations. The instantaneous overall tank concentration (ξ) is found by dividing the mass of salt by the current mass of liquid, using Equation 4.20.

$$\xi = \frac{m_s}{m_{LD}} \quad (4.20)$$

The mass of the salt in the tank, m_s , is a constant value and the same as the initial mass of salt in the tank. The mass of the salt never changes, only the amount of water removed or added during regeneration and absorption. The current overall mass of liquid desiccant in the tank, m_{LD} is based on the amount of water absorber and removed since filling the tank with the initial amount of liquid desiccant, using Equation 4.21.

$$m_{LD} = m_{init} - m_{reg} + m_{abs} \quad (4.21)$$

The initial mass of liquid desiccant in the tank, m_{init} is the sum of the mass of water and salt initially put into the tank. With varying tank sizes for some of the simulations, the initial mass of water and salt will also vary. However, for all simulations, the initial concentration is set to 0.36, with a minimum concentration of 0.30 when filled to the maximum tank capacity. m_{reg} and m_{abs} are the total amount of water regenerated and absorbed since filling the tank, respectively. The quantities are totaled by integrating the absorption and desorption rates over time.

Valves are added to the tank modeling to account for the multiple fluid streams entering the tank through the single inlet of the model, and leaving through the single outlet. The mixing valve model, Type 649, combines three fluid streams into a single stream, for simulation purposes. Type 647 is the diverting valve model that splits the single outlet of the variable volume tank into the three separate streams going to the desiccant system pumps.

Several pumps carry desiccant to and from the storage tank, regenerator, and absorber. The regenerator pump carries desiccant from the tank to the regenerator, while the absorber pump delivers desiccant from the tank to the absorber. The recirculator pump circulates desiccant from the outlet to the inlet of the tank, for constant mixing. These three pumps are modeled by single-speed pump models, Type 3b, since the pumps carry desiccant at a constant rate at all times. Pump sizes are scaled appropriately to the regenerator or absorber size, which is varied during the simulations. However, the pumps installed in the Orlando system are 1 GPM pumps operating at approximately 5 Watts. For other size pumps, the power used is assumed to remain at 5 W/GPM. The recirculator pumps is always sized to be 1 GPM at 5W. Two drain pumps are also included in the model for simulation purposes only. Normally, the regenerator and absorber would be directly above the tank to allow desiccant to drain, or drip into the tank. Two variable-speed pumps, Type 110, are modeled to pump at the exact speeds that the desiccant leaves the absorber and regenerator. In the model, no power is used and no heat is added to the fluid streams. The two drain pumps are just used to account for mass balancing in the TRNSYS simulations.

4.3 Fans and Additional Components

Variable-speed fans provide the outdoor air that is supplied to the absorber, and to the PVT system leading to the regenerator. Both fans are modeled using the TRNSYS supplied variable-speed fan/blower model, Type 662. The Orlando system uses a FanTech FKD 12XL blower that uses a maximum of 500 Watts and operates at a maximum of 2016 CFM. The Andrusiak [3] model uses a 2550 CFM blower. From FanTech and other manufacturer data, blowers generally operate at approximately 0.25 - 0.333 W/CFM. The value of 0.333 W/CFM is assumed to be typical for most blowers of similar size and used for the simulation power parameters. The default power coefficient parameters are also assumed in all simulations. Maximum fan sizes for both fans are chosen for the various parametric simulations, discussed in later sections of this thesis.

Other additional components modeled in the full system are the fresh air and bypass dampers associated with the regenerator, and are modeled using Type 11c and Type 11e, respectively. The fresh air damper model simply routes outdoor air into the air stream between the PVT system and the regenerator, when necessary. The bypass damper model routes air around the regenerator, and occurs under certain conditions. Other additional components include psychrometric functions and other necessary calculations.

4.4 Combined System

Figure 4.7 shows the combined systems and components of the TRNSYS model, mentioned above, with the exception of the controller components.

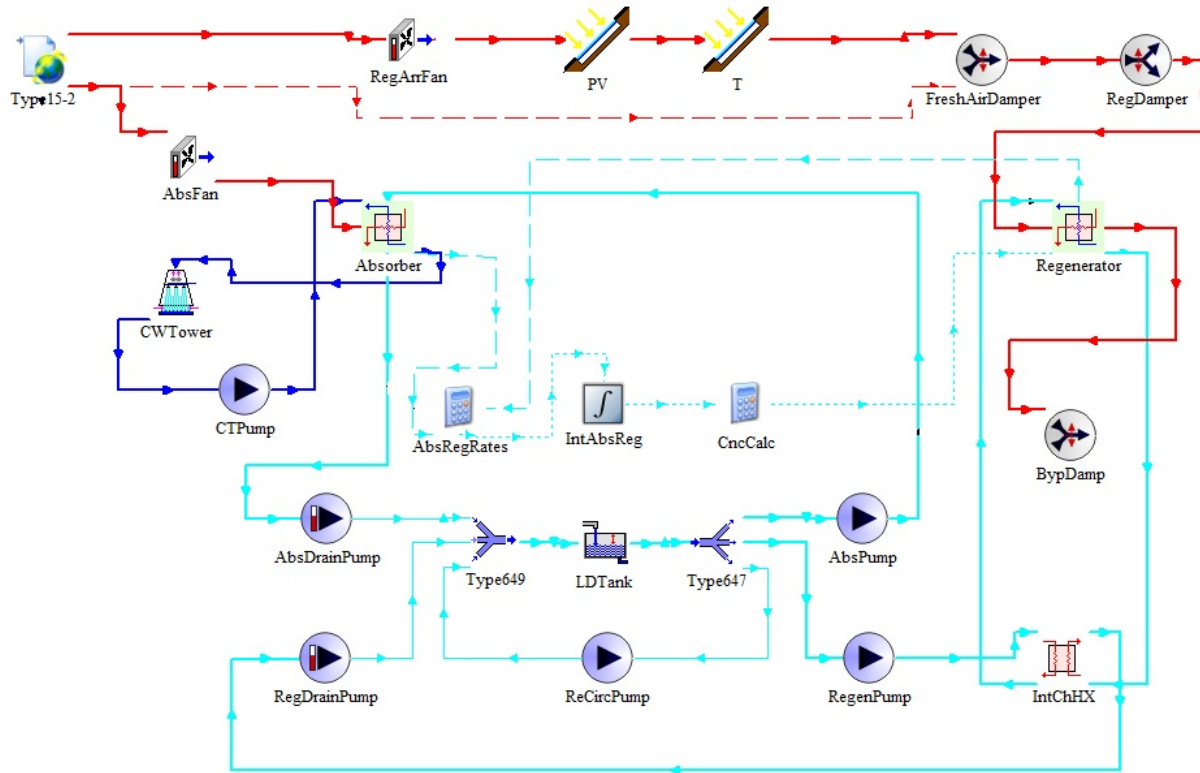


Figure 4.7: TRNSYS Model - Full System

The modeled system includes all of the components mentioned in this chapter, along with two controller components, which were written specifically for control of the system used in this thesis. One controller, designated as Type 992 or “absorber controller”, provides the functionality for operating the pumps, fan, and cooling tower that are associated with the absorber. The “regenerator controller”, Type 991, operates the fan, pumps, dampers related to the regenerator and PVT system. No control is required for operating the tank, since this functionality is provided by the absorber and regenerator pumps.

Tables 4.10 and 4.11 describe the operation of the two controllers. The physical system described in the Andrusiak [3] paper operated at a constant air, desiccant, and cooling water flow rate. The modeled system in the same paper operated to maintain a specified humidity ratio leaving the absorber, with the cooling tower on a specific schedule. For the system in this thesis, the absorber does not maintain a specific leaving air humidity ratio, but operates at a constant

air flow rate when the outdoor air is above specified humidity ratio. Also, regeneration and PVT system control differ from the Orlando system. The PVT blower is modulated to maintain a specific set-point temperature at the PVT outlet, instead of running at a constant rate. In addition, the pumps and fans only operate under certain conditions, instead of continuously. Details of the controller are mentioned in Table 4.11, and provide more realistic or typical operation than that of the Orlando system.

Table 4.10: Control Sequence Description

Controlled Device	Controller Programming
All Devices	If outdoor humidity ratio is \leq humidity ratio set-point, all pumps and fans are disabled and turned off
	If outdoor humidity ratio is $>$ humidity ratio set-point, all pumps and fans are enabled but not turned on
Fans	If fan is enabled and tank concentration is $>$ 0.305, blower turns on at 100%
	If tank concentration is \leq 0.30, blower turns off
Pumps	If pumps are enabled and tank concentration is $>$ 0.305, pumps turns on
	If tank concentration is \leq 0.30, pumps turns off

The absorber controller is designed to run the absorber pump, absorber fan, cooling tower fan, and cooling tower pump. The absorber system only operates when the outdoor humidity ratio is above a specific set-point. When below that set-point, all fans and pumps are shut off.

For most simulations, the set-point is 0.008 kg/kg . When the system is enabled and allowed to operate, absorption continues until the minimum concentration of 0.30 is reached. The system will not absorb again until the tank concentration reaches 0.305, providing hysteresis for the controller. When the absorber system is operating, all fans and pumps operate at constant flow rates designated in each simulation.

Table 4.11: Control Sequence Description

Controlled Device	Controller Programming
Blower Fan	System operates during daylight hours only
	If PVT outlet temp is $\geq 3^{\circ}\text{C}$ above the set-point, blower turns on at 5% max
	If PVT outlet temp is $> 45^{\circ}\text{C}$ and $< 85^{\circ}\text{C}$, blower modulates to maintain set-point
	If blower fan is operating at 100% and outlet temp exceeds $< 85^{\circ}\text{C}$, turn off blower
	If blower fan is operating at 5% and outlet temp drops below $> 45^{\circ}\text{C}$, turn off blower
Regeneration/Bypass Dampers	Open regeneration damper when tank concentration is < 0.415
	Close regeneration damper when tank concentration is ≥ 0.42
Fresh Air Damper	Open fresh air damper when blower is at 100% and outlet temp is $> 85^{\circ}\text{C}$
	Close damper all other times
Pumps	Turn on pumps when fan is $>5\%$ and $\leq 100\%$, and when tank concentration is < 0.415
	Turn pumps off when fan is $\leq 5\%$, or when tank concentration is ≥ 0.42

The regenerator controller is used to control the regenerator pump, regenerator blower/fan, regeneration damper, bypass damper, and fresh air damper. All of these components are different components than those discussed for the absorber controller. The PV/T and regeneration systems operate when daylight is available and electricity is produced by the PV cells. Once the systems

are enabled and the PV/T outlet temperature set-point is met, the blower fan modulates the speed to maintain the set-point. The goal of the controller is to regenerate desiccant (use regenerator pump) while the PV/T array is able to meet, or stay near, the temperature set-point. However, regeneration only continues until the maximum tank concentration of 0.42 is reached, and does not begin again until the tank concentration reaches 0.415 (for controller hysteresis). To stop regeneration during those times, the pump is turned off, but the fan remains on to maintain the PV/T outlet temperature. When the PV/T outlet temperature set-point cannot be maintained within specified bounds, regeneration stops and the fan and pump are turned off. Dampers are simply used to better maintain the desired set-point and not allow air that is too cold or too hot to enter the regenerator.

While the controllers operate independently of one another, both actively work to keep the tank concentration within the bounds of 0.30 and 0.42. Since tank concentration corresponds to the physical level of liquid inside the tank, the controllers are designed to prevent the tank from becoming too dry (high concentration) or from overflowing (low concentration). Also, absorption and regeneration become less effective when used outside the limits. Tank limits are also imposed to avoid many simulation errors that may occur if the tank overflows or dries out.

Chapter 5

Model Calibration

Parameters for the modeling of each component were found from various sources, and many were mentioned in the previous chapter. Absorber modeling, as mentioned above, was previously validated and used by Jones [20] and Andrusiak [3], but values were scaled for use in this model. Simple component models, such as the fans and pumps, use parameters found in manufacturer data and design documents for similar components in the existing Orlando system. Also, manufacturer data and design documents provide the necessary information to calculate or estimate most of the parameters for modeling the PV/T system, regenerator, and liquid desiccant system. These are also mentioned in Chapter 4. However, the PV/T and regenerator model both contain two parameters each that must be calculated using data from a working system, in order to properly model the systems. The methods for determining the parameters are discussed below, along with the calculated values for each.

5.1 PV/T Model Calibration

Two air-channel “turbulence factor” values are needed to model the PV/T system properly. One value is needed for the PV model, Type 982, and one is used for the Thermal model, Type 961. Since the air-channel is not perfectly flat or smooth, model adjustments are necessary to account for additional air mixing caused by fins and other obstructions found in the air channel. Both “turbulence” factors are used to account the additional air mixing and turbulence. Each “turbulence factor” value is a multiplication factor for the fluid convection heat transfer coefficient, h , of the air

channel, and described in Equation 5.1.

$$h = \frac{Nu_L k_f}{D_H} * f \quad (5.1)$$

The Nusselt number, Nu_L , hydraulic diameter, D_H , and air/fluid conductivity, k_f , are calculated internally to the model from other parameters and inputs. The details of the model calculations are found in the Casey [9] thesis. The “turbulence factor”, f , defines the air mixing/turbulence parameter, and is a value greater than 1.0.

Both turbulence factors are determined using experimental data from the PV/T system installed in Orlando. PV/T inlet and outlet temperatures from the experimental data were compared to the modeled values. The PV and Thermal turbulence factor values were adjusted until the best fit was found between the modeled and experimental PV/T outlet temperatures. Since temperature data is not available inside the air channel and only for inlet and outlet temperatures, both turbulence factors cannot be calculated independently. Instead, the combination of values corresponding to the best fit is found.

To determine the best fit between the model and data, two calculations were used: root mean square error (RMSE) and overall percent error. The root mean square error, or average difference between the experimental data point and the modeled value, is calculated using Equation 5.2.

$$RMSE = \sqrt{\frac{\sum_{i=1}^n (\hat{y}_i - y_i)^2}{n}} \quad (5.2)$$

The value \hat{y}_i refers to the modeled outlet temperature value for a particular time or data point, while y_i is the experimental temperature value for that same point in time. The number n is the total number of data points used compared in the calculation.

Another comparison is made between the modeled and actual data is the average overall percent error, over a given period of time. For each set of data points (specific time), the percent error is found between the experimental data and the modeled, or theoretical value. Equation 5.3 is used to calculate the percent error. The percent error is found between each set of data points,

and then averaged to find the overall percent error for the simulated time period.

$$\%Error = \frac{|Experimental - Theoretical|}{|Theoretical|} * 100 \quad (5.3)$$

Temperature data from the Orlando PV/T system, for the month of June, was used to calibrate the PV/T model turbulence factor values. The PV/T model, using default turbulence factor values of 1.0, is compared to the Orlando system in Figure 5.1. Using the previously calculated parameters and no adjustments to the model, the average percent error between the model and data is 5.4%, with an RMSE of 3.06. As shown in the figure, the model is not accurate at either high or low temperatures, which is also reflected in the high RMSE value and over 5% average error. Considerable room for improvement can be found by testing and adjusting the turbulence factor values of the model.

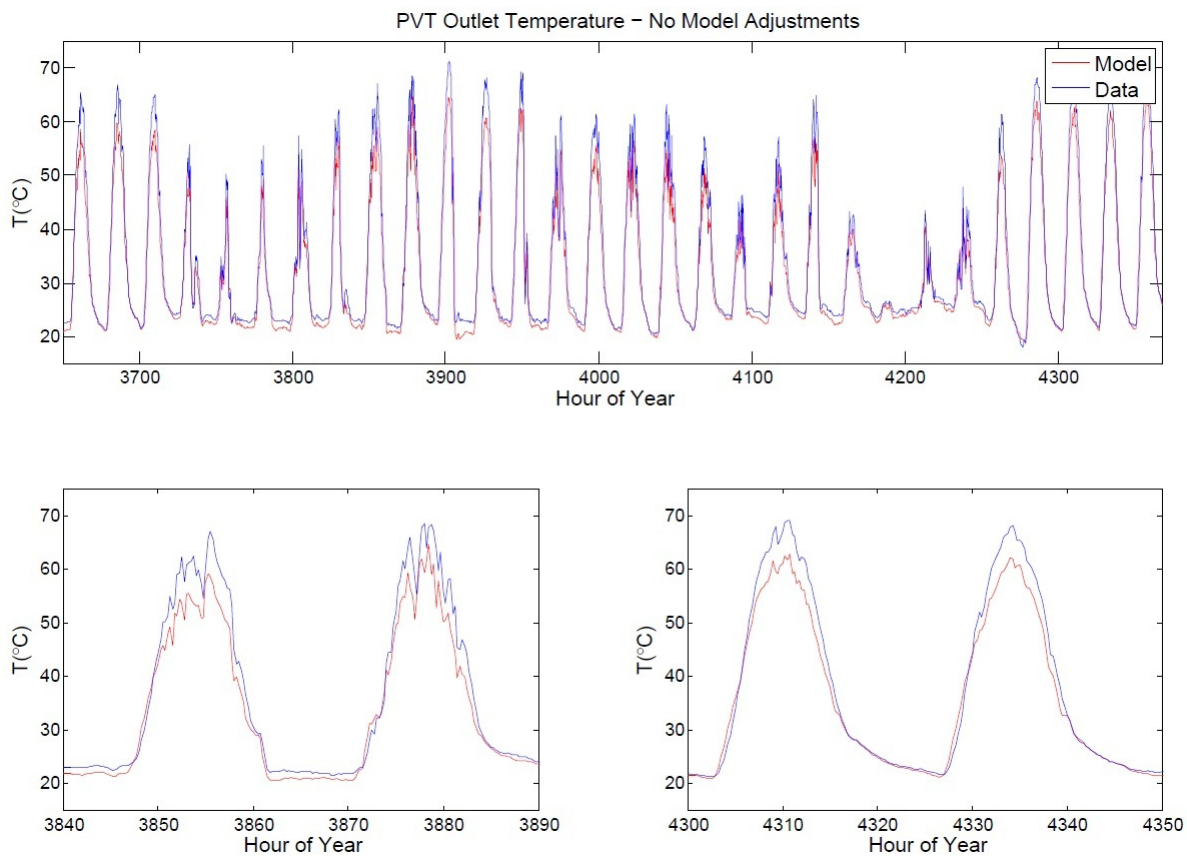


Figure 5.1: Uncalibrated PV/T Model

To find the optimal combination of turbulence factor values, the model was simulated by varying the PV and Thermal turbulence factors. PV turbulence factor values were varied from 1.0 - 2.5, in 0.1 increments, as were the thermal turbulence factor values. For all of the combinations of values, the RMSE and percent error were found. Figures 5.2 and 5.3 show the percent error and RMSE values for all combinations of turbulence factor values.

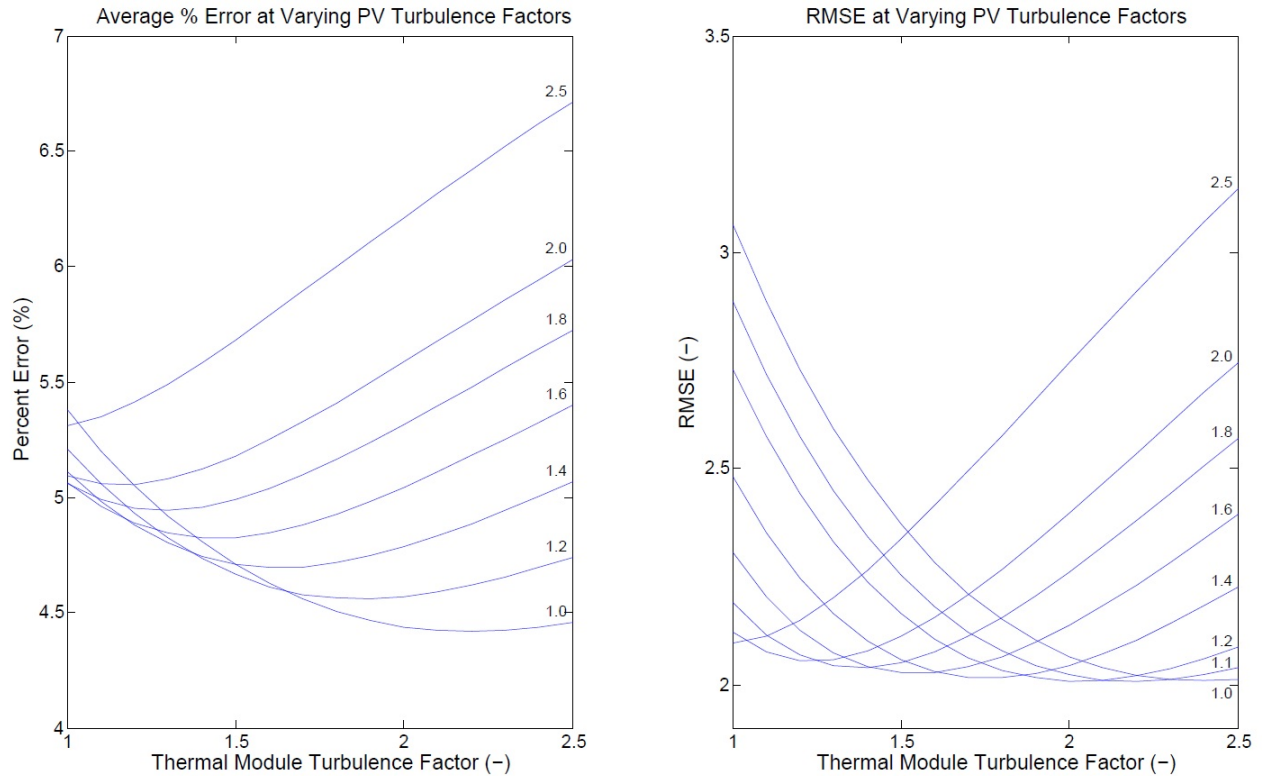


Figure 5.2: Best Fit for Thermal Turbulence Factor

Figure 5.2 shows that the lowest RMSE and percent errors occur when the thermal turbulence factor value is near 2.2, regardless of PV turbulence factor. However, the low values for percent error and RMSE occur when the PV turbulence factor is approximately 1.0. Using a PV turbulence factor of 1.0 and thermal turbulence factor of 2.2, the percent error between the PV/T data and model is 4.42% and has an RMSE of 2.02.

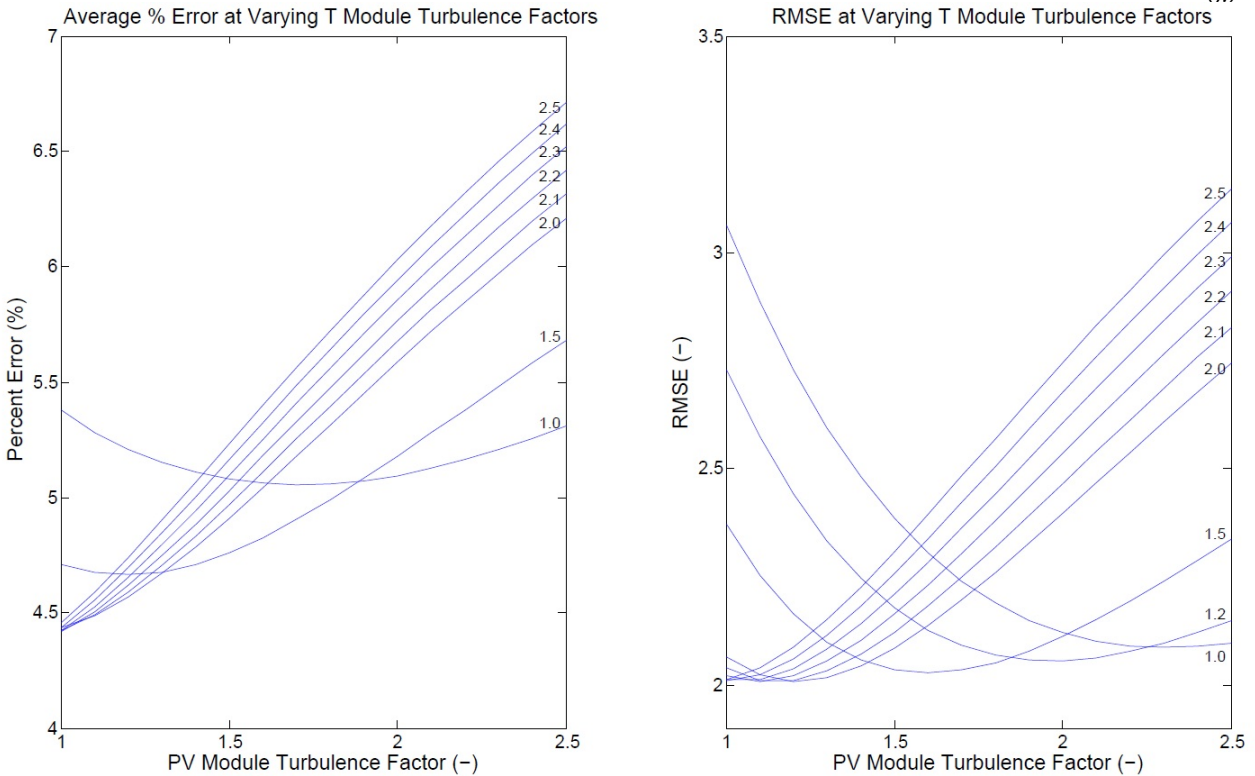


Figure 5.3: Best Fit for PV Turbulence Factor

Figure 5.3 shows the same values as in the previous figure. Again, the lowest RMSE and percent errors occur when the PV turbulence factor value is near 1.0, regardless of thermal turbulence factor. The low value for percent error occurs when the PV module turbulence factor is 1.0, at virtually all thermal turbulence factors. The low RMSE occurs when the PV turbulence factor is approximately 1.0-1.2, at higher values of the thermal turbulence factor.

For the best fit between the data and model, the combination of 1.0 and 2.2 are used for the PV and thermal turbulence factor values, respectively. The specific combination of values provides nearly the lowest percent error and RMSE. Figure 5.4 shows the outlet PV/T array temperatures for the model and data when using the optimized turbulence factors.

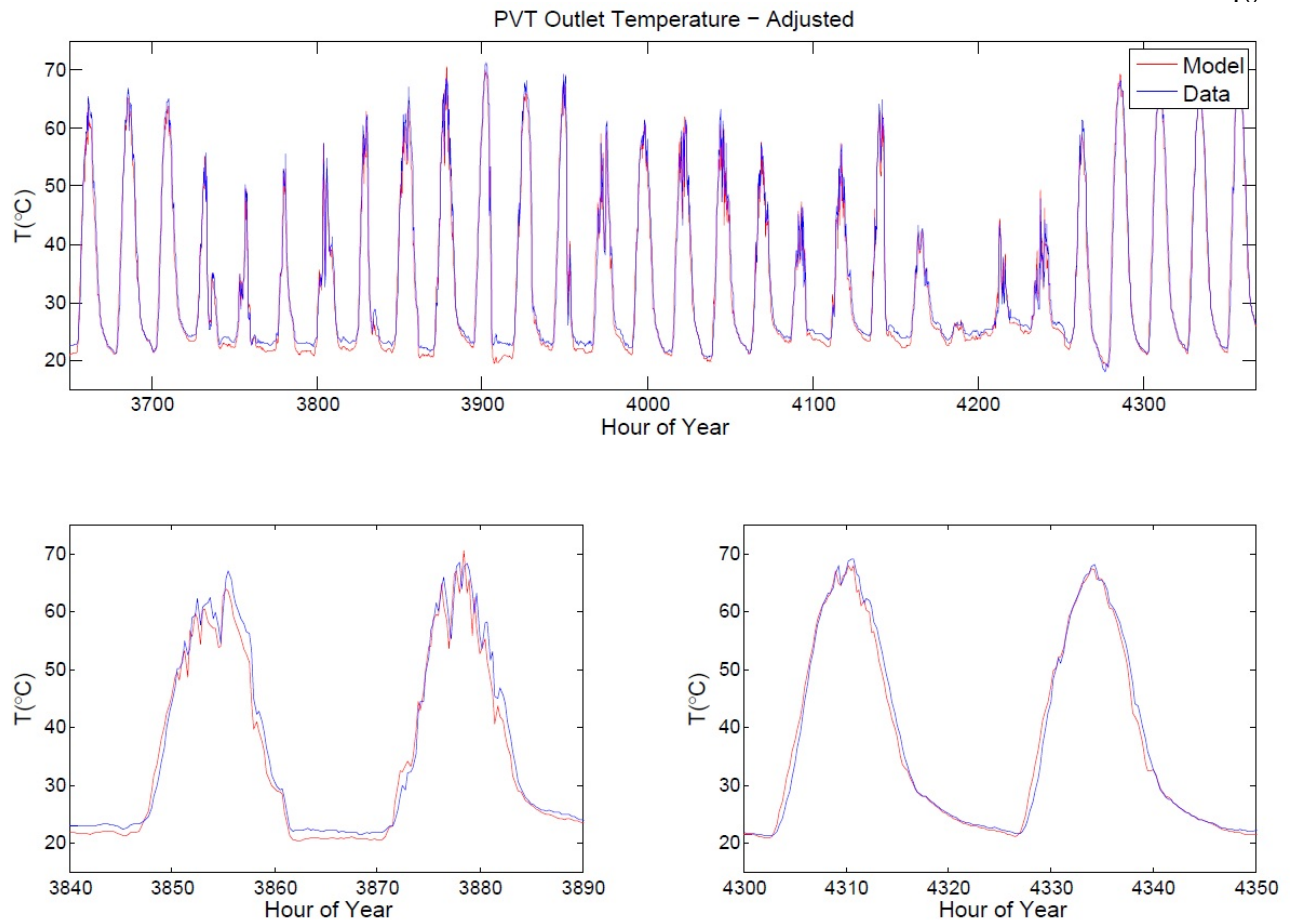


Figure 5.4: Calibrated PV/T Model

Figure 5.4 shows a much better fit than when using default turbulence factor values. At mid and peak temperatures, the model and data are nearly identical for the entire month, and low temperatures also match more closely. The model tends to predict slightly lower temperatures than the data shows at night-time hours, in certain instances. However, these hours are not considered important during simulation, since the regenerator and PV/T system are turned off and not functional during these times.

5.2 Regenerator Effectiveness Validation

In addition to the previously calculated model parameters for the regenerator model, two effectiveness parameter values are used. The enthalpy effectiveness allows the model to calculate the

change in enthalpy across the regenerator, and the humidity ratio effectiveness is used for calculating the change in humidity ratio. Since both effectiveness values are determined experimentally, data from a working system must be used. Many data sensors are necessary for these calculations, including humidity and temperature data for air entering and leaving the regenerator, and also entering desiccant temperature and concentration. Data from each sensor, in the Orlando system, was recorded at 15 minute intervals. Data from each interval was used in Equations 4.14 and 4.15 for calculation of the effectiveness values. Enthalpies and humidity ratio values were directly used in the equations, while the Conde [12] correlations were used to find the maximum humidity ratio and equilibrium enthalpy value.

Effectiveness values were calculated using all data from the period June 1st to July 9th, since the system was completely functioning and all sensors were recording accurately at those times. These effectiveness values must be accurately predicted in the model. For an accurate prediction of the values, a correlation or function, was found using the given data and experimental values. Since the experimental effectiveness values are a direct calculation using the given data, the actual values are a function of the following variables: air temperature, humidity ratio, air enthalpy, desiccant temperature, and desiccant concentration. Finding a correlation between effectiveness and the five mentioned variables is complex, and a simplified function was found for modeling purposes. Figure 5.5 shows an example of the humidity ratio effectiveness for several of the individual input variables. Using those variables shown, the data is inconsistent and effectiveness varies widely across the x-axis. However, the effectiveness over varying inlet air temperature shows a trend and is discussed later in this chapter.

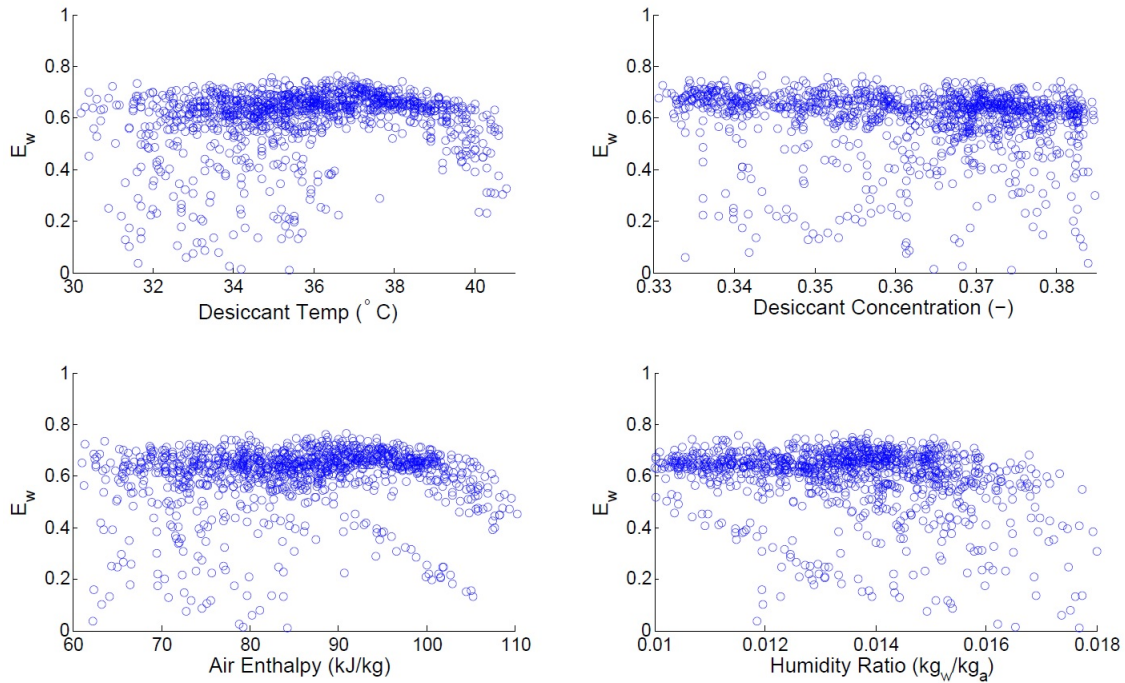


Figure 5.5: Humidity Ratio Effectiveness for Several Input Variables

Because the experimental Orlando system runs continuously, not all of the data is suitable for analysis. During the period from June 1st to July 9th, the system both absorbs during the night and regenerates during the day, depending on air and desiccant conditions. Obviously, for the regenerator model, data is needed only for when the experimental system is regenerating. Also, since there are inherent inaccuracies in measuring data, especially with multiple sensors, not all data points can be used. The accuracy of the sensors are not exact, and also erroneous readings may occur. The recorded data shows many such instances, where the reading is not similar to other recorded data under the same conditions. Many of these data points may not be considered in the effectiveness value calculations.

The accuracy of the sensors creates uncertainty in the measurements and the values calculated from these measurements. Table 5.1 shows the accuracy of the sensors used in the Orlando system measurements. The values were found in sensor data sheets for the exact sensor model numbers, when available, or from typical values for similar sensors.

Table 5.1: Sensor Accuracy

Sensor	Accuracy (+/-)	Units
Temperature	0.3	°C
Relative Humidity	3.5	%
Concentration	0.1	%

The inlet and outlet temperature measurements, for air and desiccant, provided the most accurate measurements at $\pm 0.3^\circ\text{C}$ of the actual value, since these were measured directly. Inlet and outlet desiccant concentration have an estimated accuracy of 0.1%, which is a measured value based on the level of desiccant in the tank. Relative humidity measurements are only accurate to $\pm 3.5\%$. The accuracy of these sensors lead to uncertainties in the calculated output values.

Using Engineering Equation Solver (EES), the uncertainties and percent uncertainty of the model output values were found for the Orlando data set. The uncertainty in the outlet humidity ratio ranged from $0.0005\text{--}0.0027\text{kg}_w/\text{kg}_a$, which corresponds to a percent uncertainty (ratio of uncertainty to calculated value) of 6-16%. For all values in the data set, the uncertainty in outlet enthalpy is between $1.6\text{--}6.9\text{kJ/kg}$, averaging about 3.17kJ/kg . These values correspond to a percent uncertainty that is always less than 6.5%. Some data points may be excluded for high uncertainties in the next two sections.

Data points are also excluded for other reasons. Erroneous data points are those values in the data set that are surprisingly high or low values, outside the range of the typical, or majority of values. To remove the extra data points in the next two sections, the statistical outliers and suspected outliers are removed. The full data set is divided into smaller groupings, which are temperature groupings. Both types of outliers are based on the IQR, or interquartile range. For a given set of data, the median is found. All data values larger than the median are separated and the median of these values is found, and referred to as the third quartile. All data values smaller than the median are separated and the median of these values is found, and referred to as the first quartile. The difference between the third and first quartile is the IQR value. As discussed by Frigg

et al. [17], any values greater than 1.5X the IQR above the third quartile, or less than 1.5X less than the first quartile are considered suspected statistical outliers. These outliers are erroneous and removed from the data set, and described in the sections below.

5.2.1 Humidity Ratio Effectiveness

Analysis of the data showed no strong correlation between effectiveness and four of the five variables. Fortunately, a strong correlation was found between effectiveness and entering air temperature. The raw data points are shown in Figure 5.6 and are between 30°C and 72°C. Below 40°C, the humidity ratio effectiveness is inconsistent and is highly dependent on ambient conditions. Regeneration or absorption may occur below this temperature. However, above 40°C, regeneration always occurs, and with a high effectiveness similar to effectiveness values at high temperatures. The difference in effectiveness between 40°C and 65°C is almost negligible.

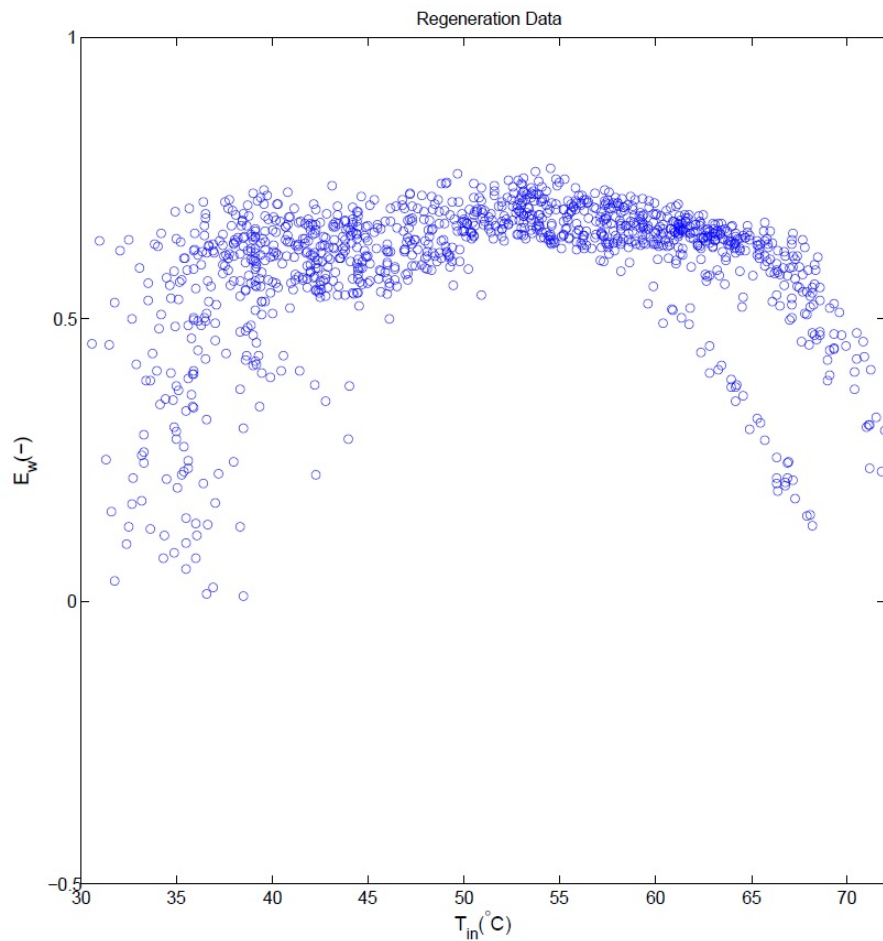


Figure 5.6: Humidity Ratio Effectiveness - Raw Data

Humidity ratio effectiveness values were next calculated for all data points where the humidity ratio increases and temperature decreases across the regenerator. These are shown in Figure 5.7. The filtering of this data ensures that only regeneration data points are used. However, data points considered to be erroneous are not excluded from the figure.

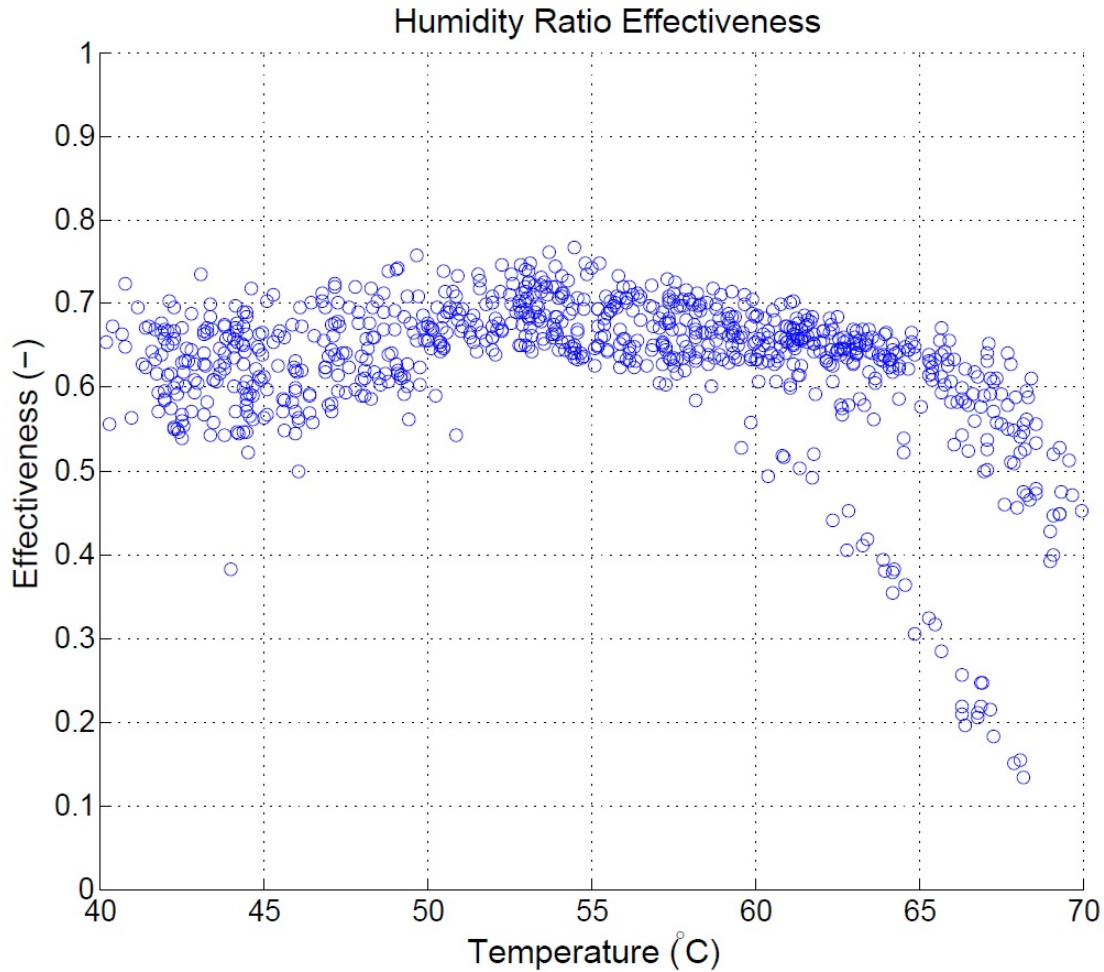


Figure 5.7: Humidity Ratio Effectiveness - Regeneration Data

Statistical outliers were then removed for each temperature and shown in Figure 5.8. Using only data from the reduced data set, the average effectiveness value is 0.642 and indicated by the red line in the figure. At temperatures below 50°C and above 65°C, the data shows a decrease in humidity ratio effectiveness, while temperatures within that range are more constant.

No data points were removed due to high uncertainties. There were also no data points removed because the difference between entering and leaving humidity ratio was within the calculated uncertainty. While initially some of these values existed, all of them occurred during absorption, low temperatures, or were statistical outliers.

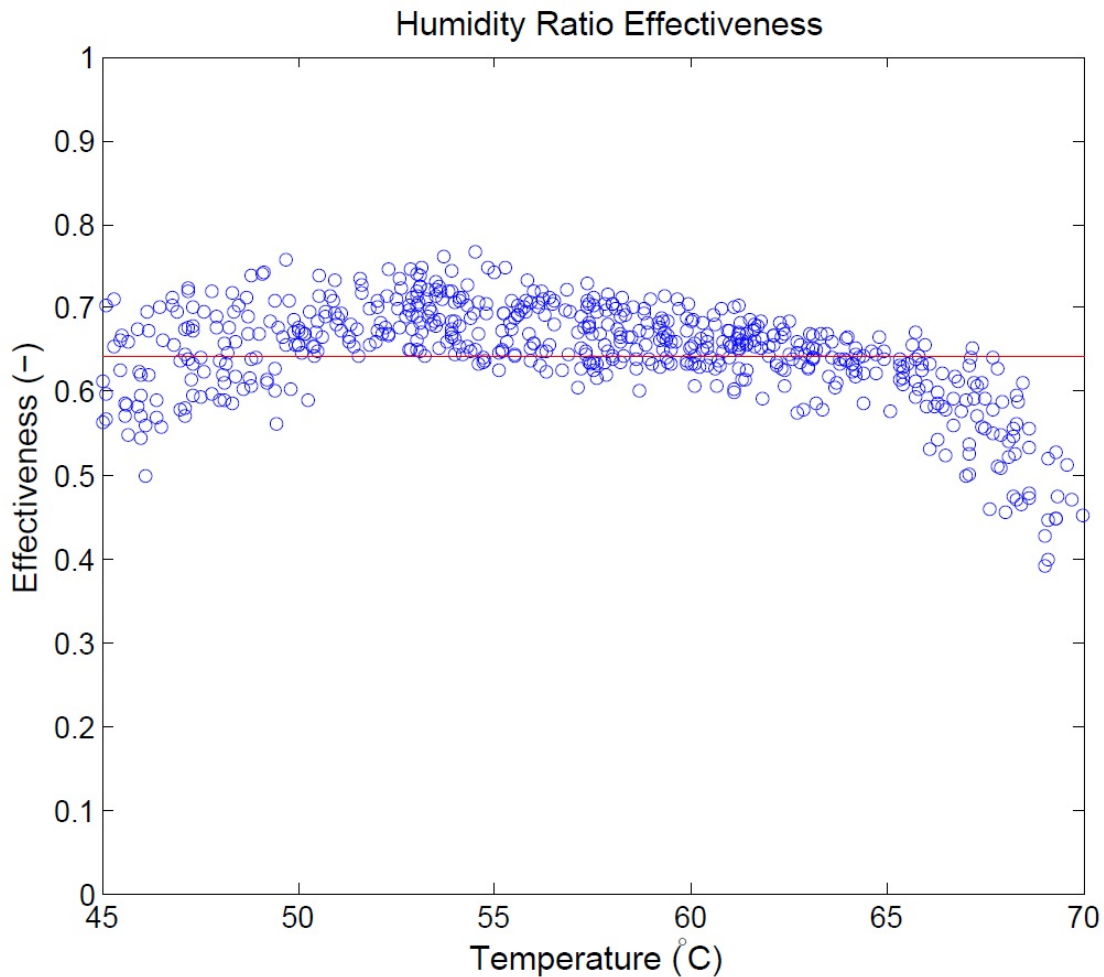


Figure 5.8: Humidity Ratio Effectiveness - Data Removed

The average value may be used as a fixed, or constant, effectiveness value. However, a simple constant effectiveness under all conditions, may result in large errors between the modeled outlet humidity ratio and the humidity ratio from the experimental data. Using the same data set, as mentioned above, and a fixed effectiveness value of 0.642, the modeled outlet humidity ratios were compared to the experimental values. The comparison shows a maximum percent error of 14.53% for all temperatures. However, over the temperature range of 50°C - 65°C, the maximum percent error is 4.72%, with an average of 2.2451%. Most of the large differences in outlet humidity ratio occur above 65°C, because the average value does not take into account the reduction in effectiveness at very high temperatures, as shown in the data. Figure 5.9 shows the percent error between the

data and model across various entering air temperatures. As shown in the figure, a majority of the modeled values are under 5% error.

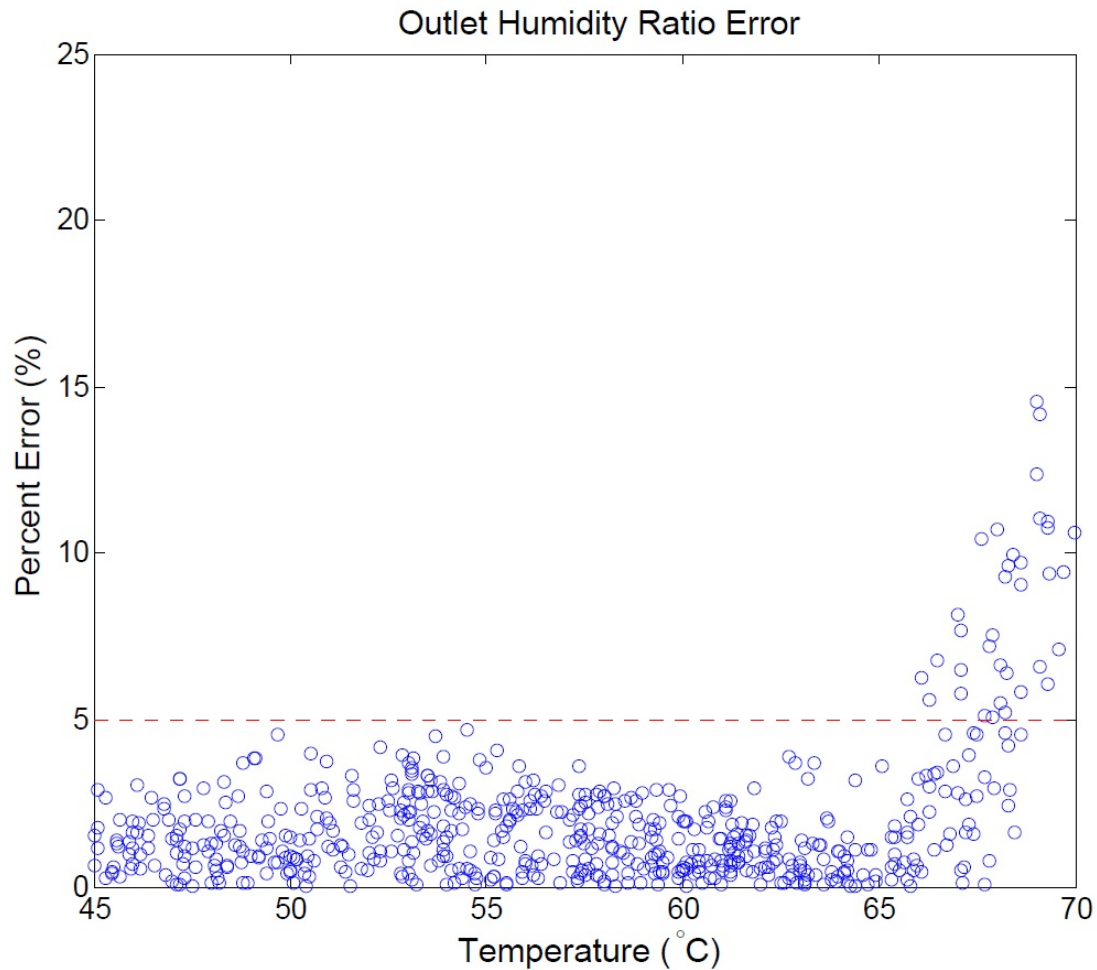


Figure 5.9: Outlet Humidity Ratio - Percent Error Between Data and Model

For the model to make a more accurate calculation of the humidity ratio effectiveness and outlet humidity ratio, a non-constant function for effectiveness was used. Many curve fitting options were tested to find the best fit for the data, including various polynomials. The best fit was found to be a rational function using a quadratic polynomial for the numerator and 4th degree polynomial for the denominator. The function is shown in Equation 5.4, and contains the coefficients from Table 5.2.

$$\epsilon_w(T_{in}) = \frac{p_1 * T_{in}^2 + p_2 * T_{in} + p_3}{T_{in}^4 + q_1 * T_{in}^3 + q_2 * T_{in}^2 + q_3 * T_{in} + q_4} \quad (5.4)$$

$$\epsilon_w(T_{in}) = \frac{392.1 * T_{in}^2 + 650.4 * T_{in} + 24.37}{T_{in}^4 - 109.8 * T_{in}^3 + 3592 * T_{in}^2 + 72.39 * T_{in} - 0.6792} \quad (5.5)$$

Table 5.2: Coefficients for Fitted Function

	p	q
1	392.1	-109.8
2	650.4	3592
3	24.37	72.39
4	-	-0.6792

The higher order polynomials were necessary in order to account for the slowly increasing effectiveness from 45°C - 55°C, and the steep decrease in effectiveness above 65°C. Also, while most functions produce a very steep drop below zero effectiveness at high temperatures, the rational function used only approaches zero. The approach to zero at high temperatures is more accurate, because increasing to higher temperatures will not cause the effectiveness to become negative and for absorption to start occurring. Figure 5.10 shows the data compared to the fitted function.

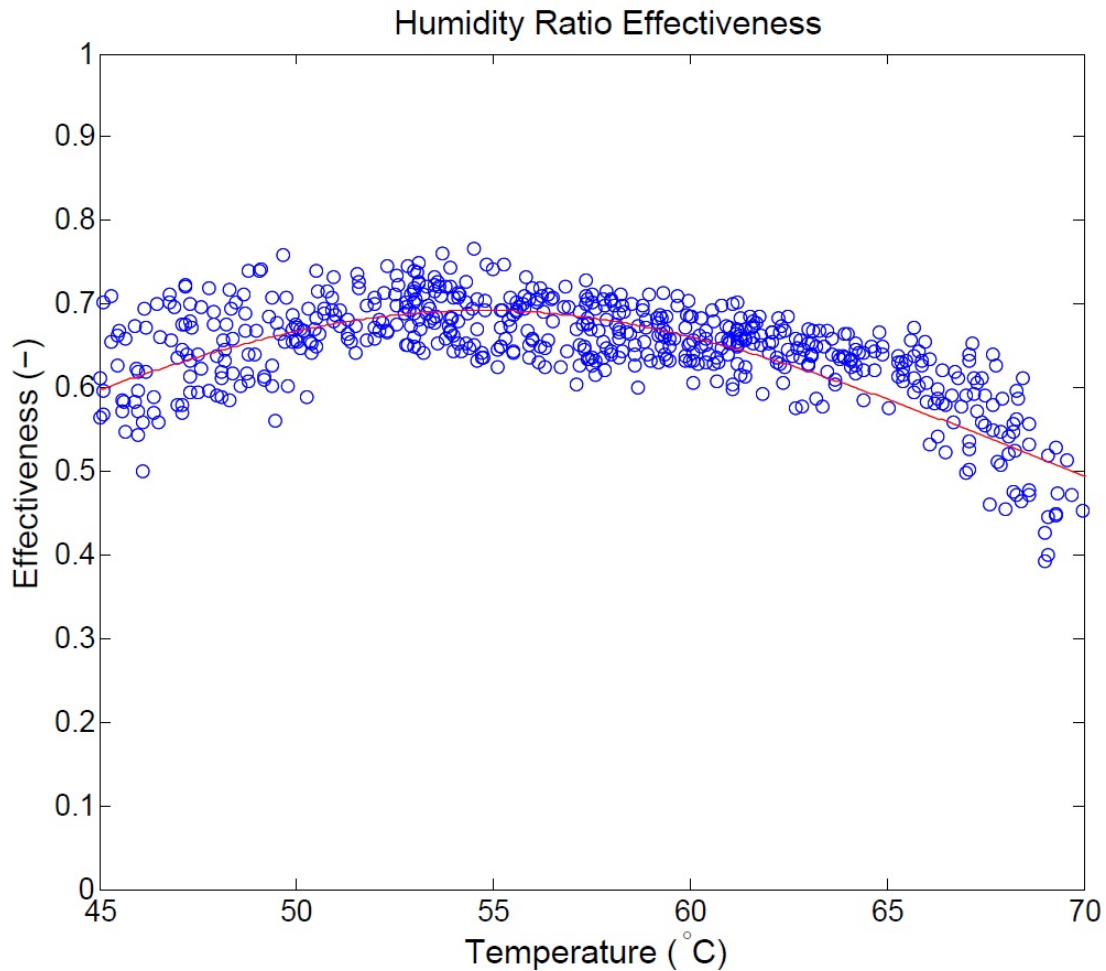


Figure 5.10: Humidity Ratio Effectiveness - Best Fit

The R-square value for the fitted curve is 0.6329, which is reasonable considering the number of data points and number of variables affecting the effectiveness data values. Also, even though the effectiveness data may not exactly match the effectiveness curve, the outlet air humidity ratio calculated from the effectiveness curve has much less of a percent error than using a constant effectiveness. The percent error between the data and model for the outlet air humidity ratio shows a maximum percent error of 7.04% for all temperatures. Over the temperature range of 45°C - 65°C, the maximum percent error is 5.02%, with an average of 1.57%. As when using the constant effectiveness, the largest percent error occurs above 65°C. By using the rational function for determining the effectiveness value, the maximum percent error is reduced by over 7% and the

average error is reduced by 0.68%. Figure 5.11 shows the percent error between the data and model across various entering air temperatures. As shown in the figure, a majority of the modeled values are under 5% error.

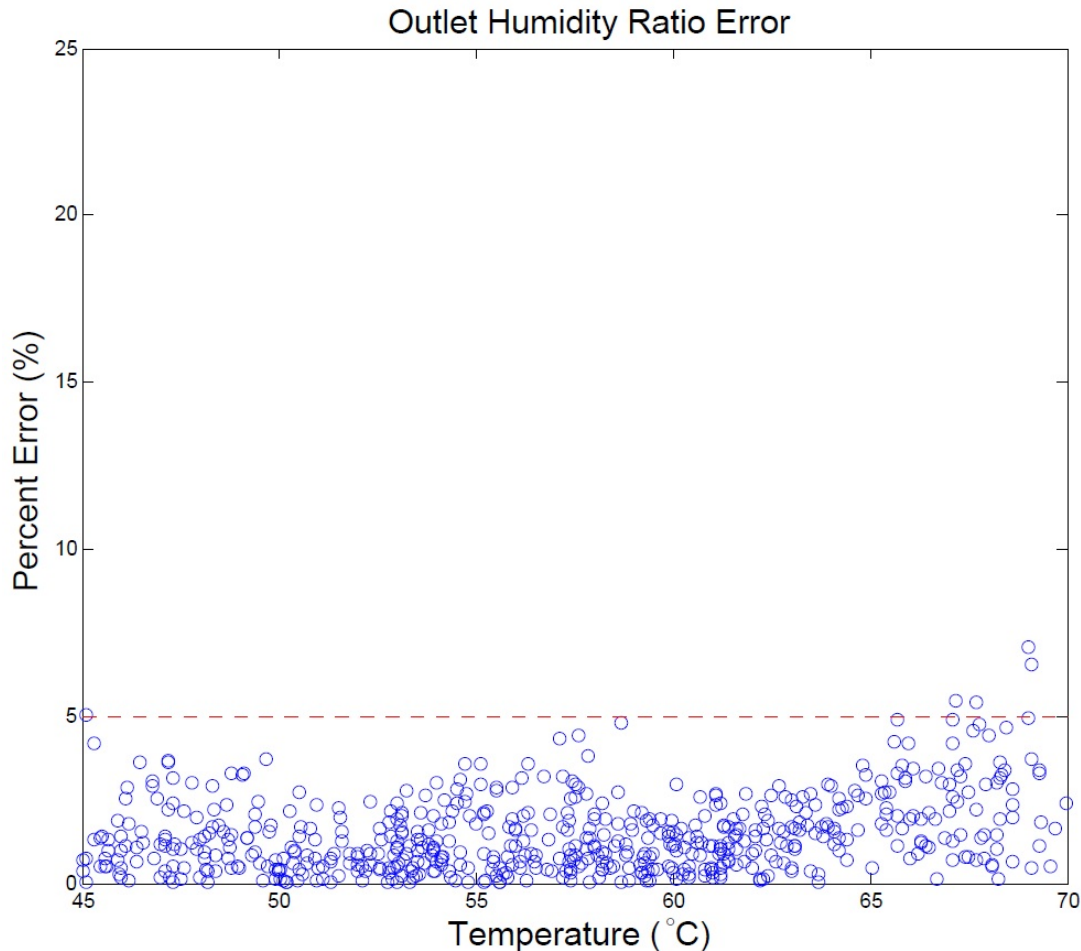


Figure 5.11: Outlet Humidity Ratio - Percent Error for Best Fit

Since the regenerator fan was run at a constant flow rate, all data used to calculate the humidity ratio effectiveness is for that given flow rate. However, in reality, effectiveness varies at different flow rates. Since the modeled system uses a variable speed fan that changes flow rates, the variation in effectiveness must be taken into account. A simple calculation is used to adjust the effectiveness value for simulation, and shown in Equation 5.6. It is based on a simple heat exchanger effectiveness equation found in McAdams [25]. The equation basically shows that when

the overall heat transfer coefficient of the heat exchanger is constant, effectiveness increases as flow rate decreases. However, it follows the exponential curve found in the equation.

$$\epsilon_{adj} = 1 - \exp\left(\frac{\dot{m}_{des}}{\dot{m}_{in}}\right) \quad (5.6)$$

The adjusted effectiveness, ϵ_{adj} , provides an effectiveness value between 0 and 1 that takes into account different flow rates. The value \dot{m}_{des} is the design flow rate, or the flow rate that was used for all data. This value is constant like the heat transfer coefficient, but \dot{m}_{in} is the adjusted or changing flow rate. To calculate the new effectiveness value, the adjusted effectiveness is normalized and multiplied by the real effectiveness value fitted from the data.

5.2.2 Enthalpy Effectiveness

Enthalpy effectiveness values were calculated for all data points where regeneration occurred, as with humidity ratio effectiveness, and is shown in Figure 5.12. Data points considered to be erroneous are not excluded from the figure.

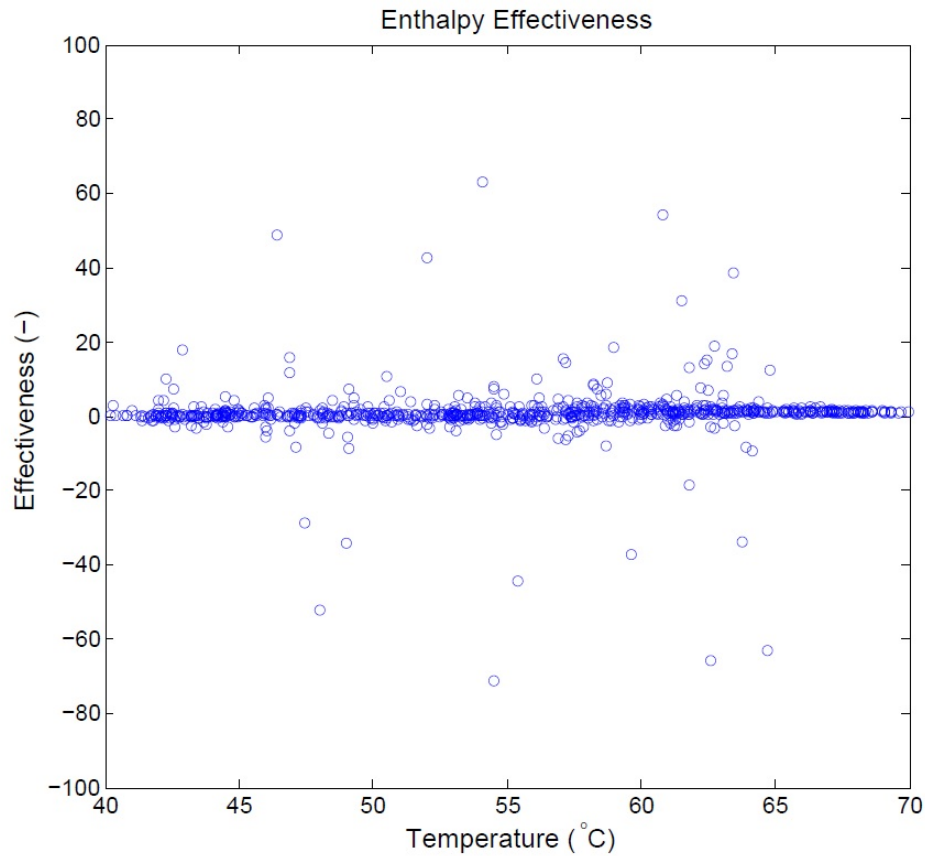


Figure 5.12: Enthalpy Effectiveness - All Data

As in humidity ratio effectiveness, the statistical outliers and erroneous data points were removed, since these likely correspond to errors in measurement. The reduced data set for enthalpy is shown in Figure 5.13. Using this data, all data points are centered around 0.0, with effectiveness values ranging from -2.7 to 6.1. The zero value is indicated by the dashed black line in the figure, the average effectiveness value is 0.793. At lower temperatures, the enthalpy effectiveness is generally negative, but positive at higher temperatures.

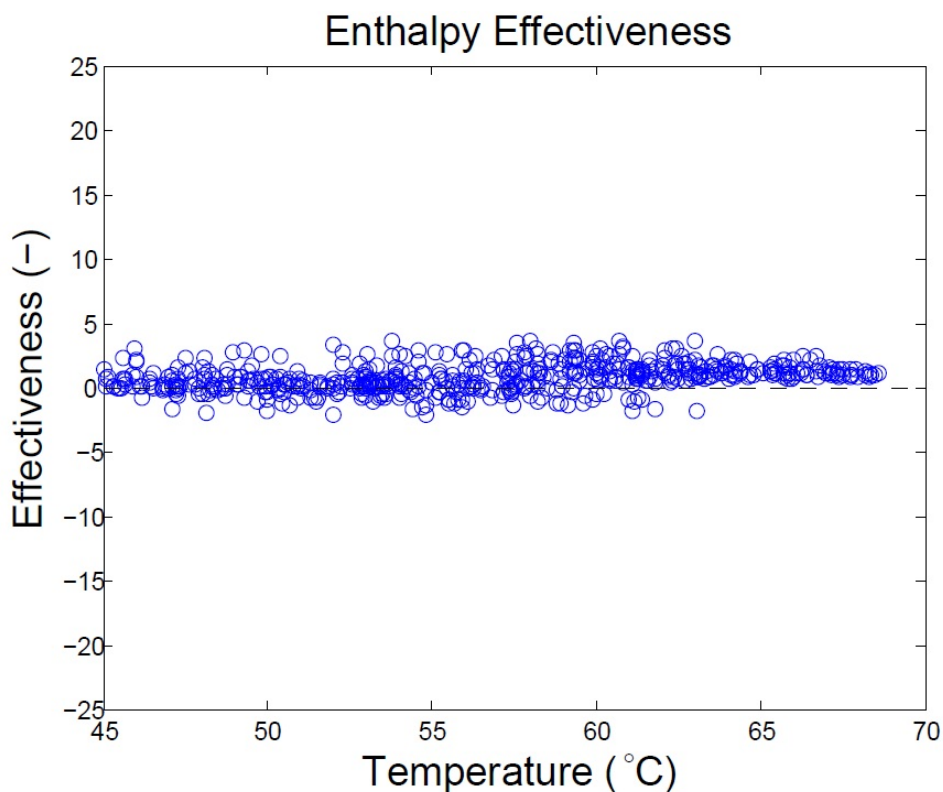


Figure 5.13: Enthalpy Effectiveness - Data Removed

At this point, it is possible to remove many other data points from the data set, due to high uncertainties in the inlet and outlet enthalpies. There are data points where the difference between entering and leaving enthalpy is smaller than the calculated uncertainty. A figure and example is given to understand these conditions.

Figure 5.14 shows an exaggerated example of regeneration. As shown in the figure, the entering air enthalpy moves closer to the maximum/equilibrium enthalpy, while the desiccant equilibrium enthalpy moves towards the original entering air enthalpy. Equation 5.7 describes how effectiveness is calculated from these conditions. In typical conditions, the red box shown in the figure is much smaller, and the actual difference in enthalpy across the regenerator is very small, with an average difference of 2.11 kJ/kg . In all cases, an enthalpy effectiveness value of ± 5 corresponds to less than a 2.5 kJ/kg difference in entering and leaving enthalpies, which is negligible.

$$\varepsilon_h = \frac{h_{in} - h_{out}}{h_{in} - h_{eq}} \quad (5.7)$$

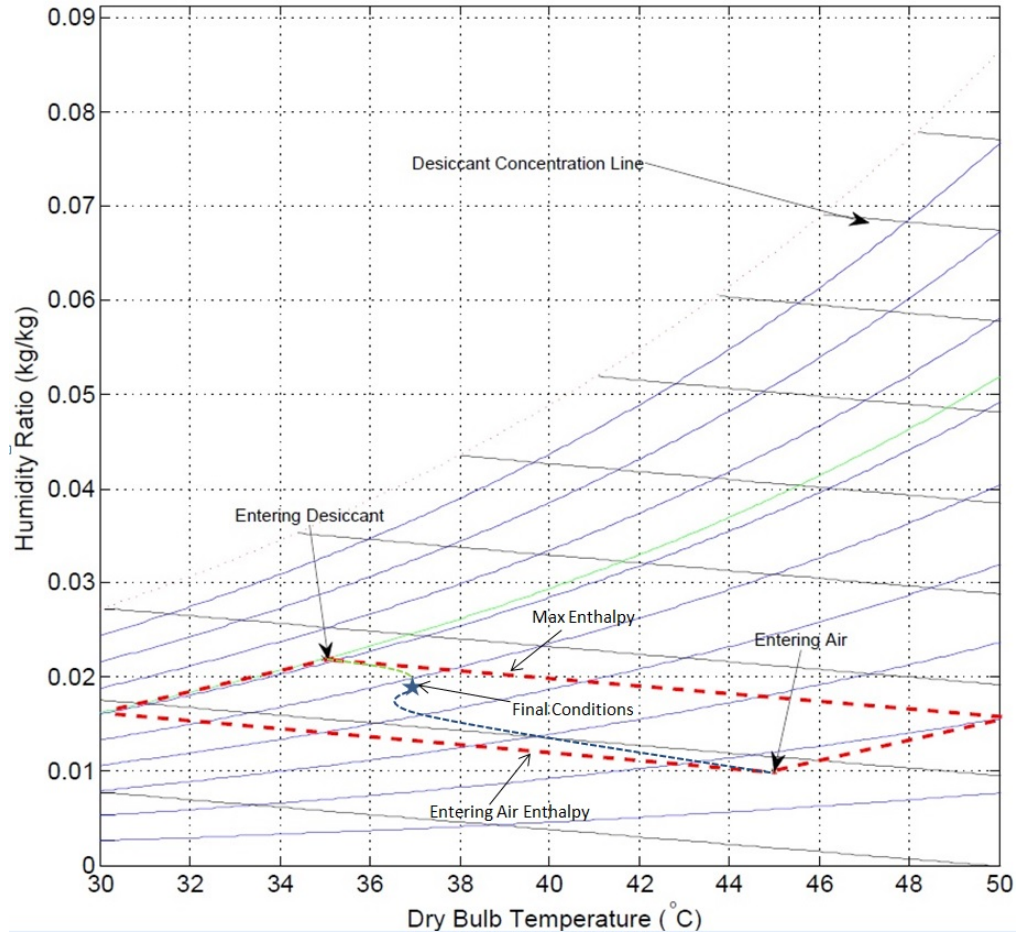


Figure 5.14: Regeneration on the Psychrometric Chart

Entering air and desiccant enthalpy states are much closer together than shown in the figure, generally within 1-2 kJ/kg . The actual uncertainty of these calculated values ranges from 1.5-15 kJ/kg for the entering air enthalpy and 1.5-9 kJ/kg for leaving air enthalpy. For example, using average values, when the entering air enthalpy is 71.92 kJ/kg , the actual value will range from 67.7-76.2 kJ/kg , due to the uncertainty in the measurements and calculations. In this same case, the leaving air enthalpy will be 70.4 kJ/kg , which is only a difference of 1.5 kJ/kg less than the entering air enthalpy. The real value of this calculation ranges from 67.3-73.6 kJ/kg . If the actual

entering air enthalpy is 70 kJ/kg and the leaving enthalpy is 71.5 , then the real difference shows an increase 1.5 kJ/kg .

All remaining data points fall within this margin of error, where the calculated uncertainty is larger than the enthalpy difference across the regenerator. In other words, every data point would have to be removed due to the high uncertainty. There is no way to determine if the actual values show an increase or decrease in enthalpy, or to calculate an accurate enthalpy effectiveness. However, the difference in enthalpy across the regenerator is shown to be very small and negligible, so assuming an enthalpy effectiveness value of 0 is reasonable.

Due to the uncertainty, negligible effect of effectiveness on leaving air enthalpy, and since both positive and negative values need to be considered, a constant value of 0.0 enthalpy effectiveness is tested further and the data from Figure 5.13 will be used.

The value of 0.0 is used as a fixed, or constant, effectiveness value, since it lies between both the positive and negative values found from the data. An enthalpy effectiveness value of 0.0 does not result in large errors between the modeled outlet enthalpy and the enthalpy from the experimental data. Using the selected value of 0.0 , the modeled outlet enthalpies were compared to the experimental values. The comparison shows a maximum percent error of 6.79% for all temperatures, with an average of 1.51% . Figure 5.15 shows the percent error of enthalpy between the data and model across various entering air temperatures. As shown in the figure, no values are greater than 10% and almost all values are under 5% error.

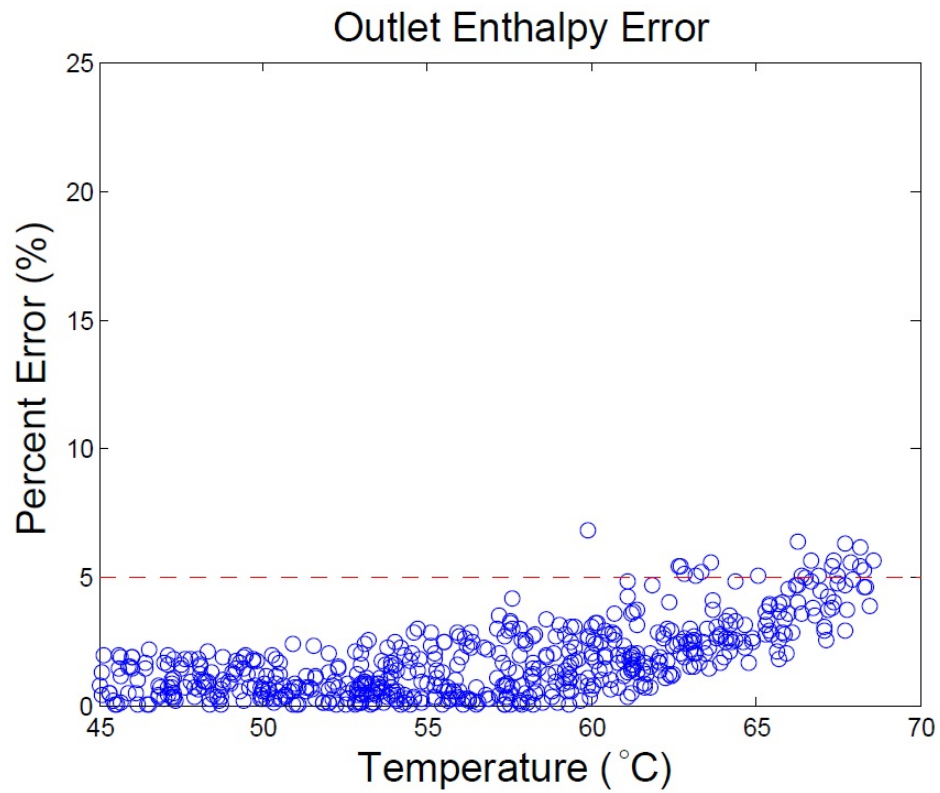


Figure 5.15: Outlet Enthalpy - Percent Error Between Data and Model

The constant effectiveness of 0.0 is used in the calculations for all simulations, since the overall error is low. Also, enthalpy effectiveness was shown to have little effect on leaving air enthalpy.

Chapter 6

Baseline Modeling Results

The components mentioned in the previous chapter were combined to provide a simulation model used for various sized systems operating in different climates. Simulations are used to determine proper sizing for each component of the system, and the effect that the sizing has on meeting a load. The maximum load that the system can meet is limited by the capacity of the regenerator and of the tank, and varies by climate. To understand the effects of component sizing, a single system is first analyzed and used as the “baseline” system model. A system similar to the experimental Orlando system is used as this “baseline” model by using the same size PV/T system as in Orlando.

6.1 Maximum Load Calculation

Since the purpose of the absorber is to dehumidify outdoor air, the amount of humidity that can be removed from the air is calculated. The absorber will dehumidify while the outdoor air is above a set-point of $0.008 \text{ kg}_v/\text{kg}_w$, which corresponds to approximately 70°F and 50% relative humidity. The difference between the outdoor air humidity ratio and the humidity ratio set-point is the mass of water vapor that can be absorbed, per mass of air. To find the load, or maximum amount of water vapor that the absorber can remove, the difference is multiplied by the absorber flow rate at each time step, as shown in Equation 6.1.

$$Load = \dot{m}_{abs}(\omega_{in} - \omega_{SP}) \quad (6.1)$$

Figure 6.1 shows the humidity ratio in Orlando, for a year. During the summer months, the figure shows that the outdoor humidity ratio is far above, and rarely drops below the set-point of 0.008. During the colder months, that humidity ratio regularly drops below the set-point.

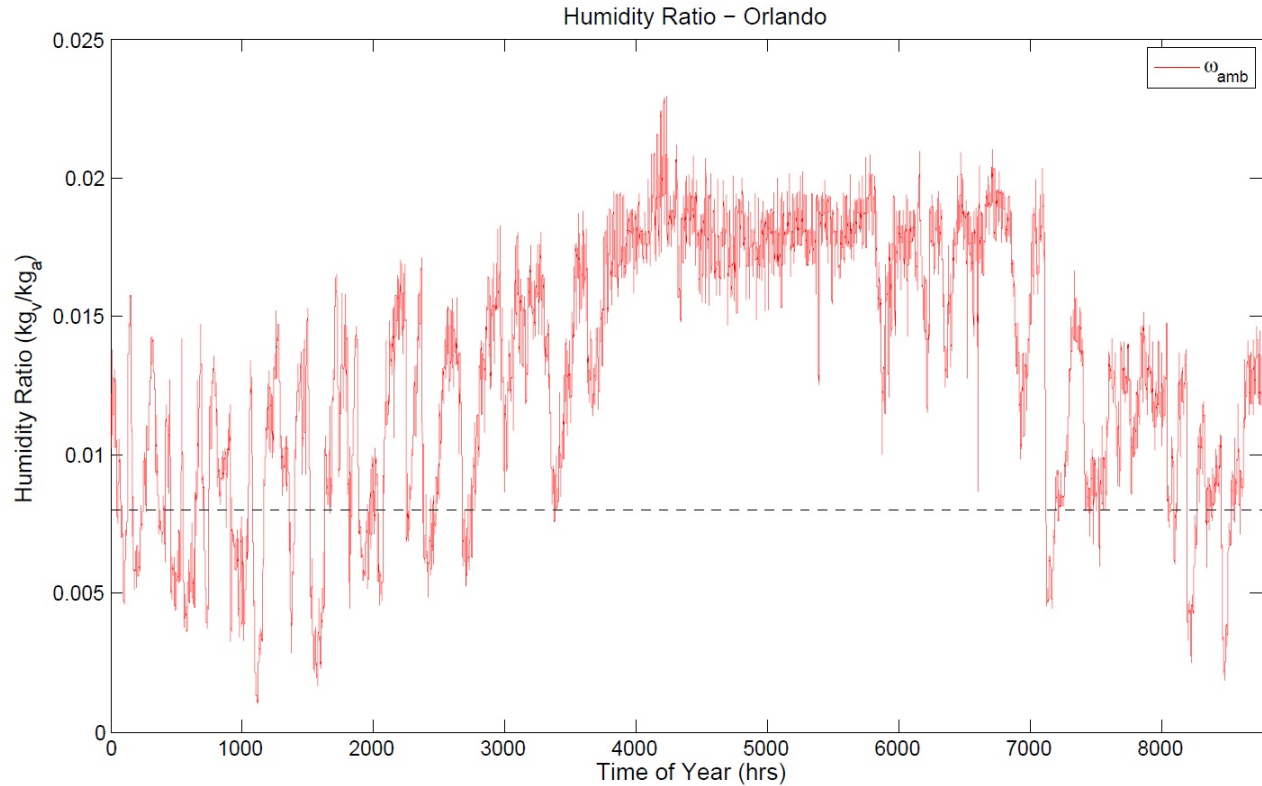


Figure 6.1: Outdoor Humidity Ratio - Orlando

Figures 6.2 and 6.3 display inlet and outlet humidity ratios at the absorber, for an example system in Orlando. When the humidity ratio is below the set-point, both the inlet and outlet humidity ratio values are the same, because the absorber is not running and the inlet/outlet air conditions do not change. Figure 6.3 also shows, in some instances, that absorption still occurs when the outdoor humidity ratio is slightly above the set-point. During these times, the air is dehumidified to a value lower than the specified set-point. The additional absorption during these times is added to the total load calculation, in order to account for the extra dehumidification that occurs.

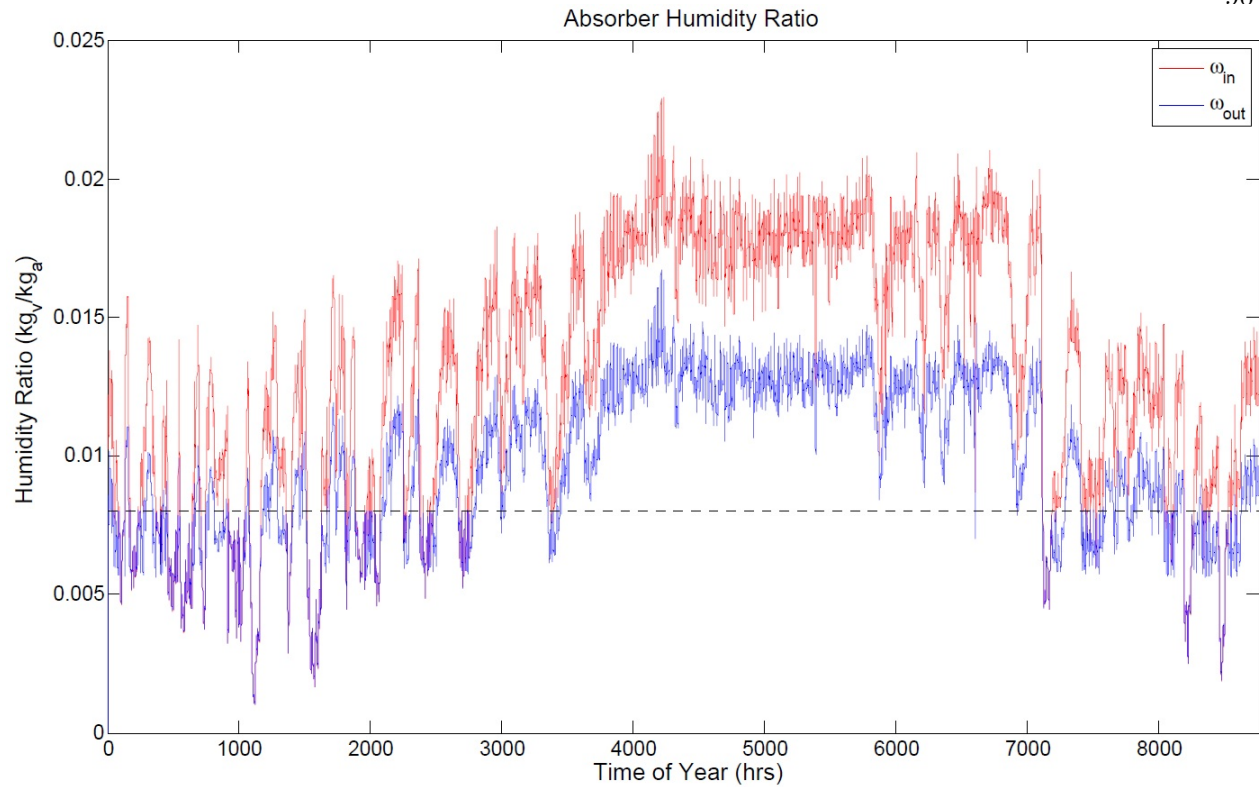


Figure 6.2: Humidity Ratios at Absorber - Yearly

The absorber cannot always absorb enough moisture to reach the user-defined set-point. Figure 6.2 shows this during the summer months. The inability to reach the set-point occurs for approximately 65% of the year, for a typical system in Orlando. Therefore, the maximum load that any system can meet will approach 65%, not 100%.

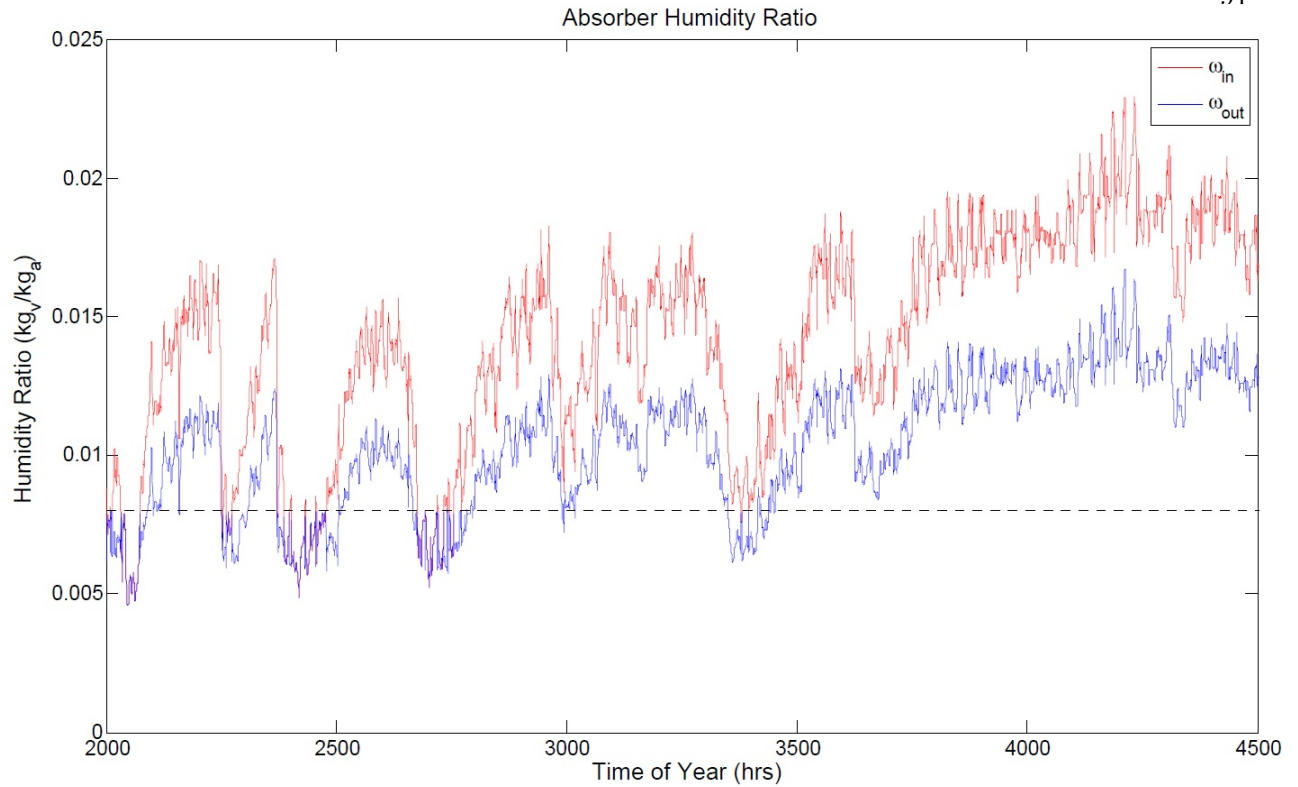


Figure 6.3: Humidity Ratios at Absorber - Partial Year

6.2 Regenerator and Fan Sizing

The blower fan speed is modulated to provide hot air, of a specific temperature, from the PV/T system to the regenerator. At the fastest fan speed, the fan must be able to maintain the set-point temperature under the hottest conditions for a given climate. Equation 6.2 describes the amount of solar energy transferred into the air channel, while Equation 6.3 refers to the amount of energy moved away from the PV/T system and out of the air channel. When the two values are equal, Equation 6.4, the flow rate necessary to maintain the desired set-point is found.

$$\dot{q}_{max} = \eta I_{max} A \quad (6.2)$$

Maximum incident solar radiation, I_{max} , is found in the weather data for a given climate, and PV/T area, A , is constant for a given simulation. Efficiency of the system, η varies with different

operating conditions, so the maximum efficiency of a typical PV/T system is used to account for maximum heat transferred to the air stream.

$$\dot{q}_{max} = \dot{m}_{min} C_p (T_{sp} - T_{in}) \quad (6.3)$$

T_{in} varies throughout the year, so the lowest temperature at the maximum incident solar radiation is used. The value is found in the yearly weather data. Set-point temperature, T_{sp} , is fixed for a given simulation, and the typical value of 1.006 kJ/kgK for the specific heat capacity of air, C_p , is used.

$$\dot{m}_{min} = \frac{\eta I_{max} A}{C_p (T_{sp} - T_{in})} \quad (6.4)$$

The minimum fan air flow rate needed to maintain the set-point temperature is referred to as \dot{m}_{min} , and calculated using the values in the Equations 6.2 and 6.3.

Fans for the PV/T system and regenerator are sized to maintain the set-point and use no more energy than necessary, so the maximum fan flow rate is equivalent to \dot{m}_{min} . The fan sizes for the Orlando system are 684 L/s (1449.2 CFM), 451.3 L/s (956.3 CFM), and 336.8 L/s (713.6 CFM), for temperature set-points of 55°C , 65°C , and 75°C , respectively.

The actual Orlando system uses a 944 L/s (2000 CFM) fan for the PV/T system and regenerator, which would allow the system to maintain all temperature set-points. The experimental system only runs at approximately 354 L/s (750 CFM) at all times, and data showed that the maximum temperature set-point of 75°C was never reached during summer conditions. However, a smaller regenerator fan can be used to reduce power consumption and still meet any of the desired set-point temperatures.

6.3 Absorber Fan Sizing

Any building or space requires a certain amount of outdoor air, which can be provided by an absorber fan that also dehumidifies the air. While small residential buildings may only need

47.2L/s (100 CFM) of outdoor air, larger buildings can require 472L/s (1000 CFM) or more. Four different absorber fan sizes are simulated with the other system components: 47.2, 118, 236, and 472 CFM. The fan sizes are used to calculate the total maximum dehumidification load, as mentioned previously. In addition, the fraction of the maximum load that the absorber can meet is also calculation, along with actual amount of water that is absorbed. For tank size to have little to no effect on the absorption load, a large tank is used in simulations for sizing the absorber fan. For the baseline system, a tank size of approximately 11500 Liters is used. Table 6.1 shows the load calculations at various absorber fan sizes and PV/T temperature set-points.

Table 6.1: Load Calculations - Orlando

Fan Size (L/s)	Temp SP (°C)	Total Absorbed (kg)	Load Fraction (kg/kg)
47.2	55	6519	0.7068
	65	6384	0.6955
	75	5479	0.6062
118	55	13115	0.5813
	65	11569	0.5168
	75	8354	0.3765
236	55	17048	0.3835
	65	13166	0.2974
	75	8724	0.1976
472	55	17869	0.2025
	65	13348	0.1517
	75	8632	0.0981

The table shows that with the PV/T system and regenerator installed in Orlando, only a 47.2L/s absorber can be used. In other words, a 47.2L/s absorber would be appropriately sized

for the PV/T array size in Orlando.

Overall maximum load increases at larger absorber sizes, but less of the load is met as absorber size increases. The load fraction actually drastically decreases with increasing absorber fan size. Also, a PV/T temperature set-point of 55°C provides the greatest amount of regeneration, and therefore, more absorption. Since more regeneration occurs at the lower set-point, in all cases, the tank has more capacity for absorption.

6.4 Tank Sizing

The tank adds another limitation to the regeneration and absorption capacity of the system. If the tank is large enough, the regenerator can always regenerate when enough warm air is available, and the absorber can always dehumidify when the outdoor humidity ratio is above $0.008 \text{ kg}_v/\text{kg}_a$. With a limited tank size, the tank could potentially run out of desiccant or overflow with too much absorbed water vapor. Very large tanks are costly and take up space. Also, the cost of salt and the actual tank can be a consideration. Properly sizing the desiccant tank allows much of the maximum load to be met, while reducing cost and space requirements. Simulations are used to determine the tank size associated with meeting a majority of the maximum load.

For simulation of the baseline Orlando system, the previously calculated PV/T-Regenerator fan sizes are used, along with a $47.2\text{L}/\text{s}$ absorber fan. Figure 6.4 shows how load fraction is affected by tank size.

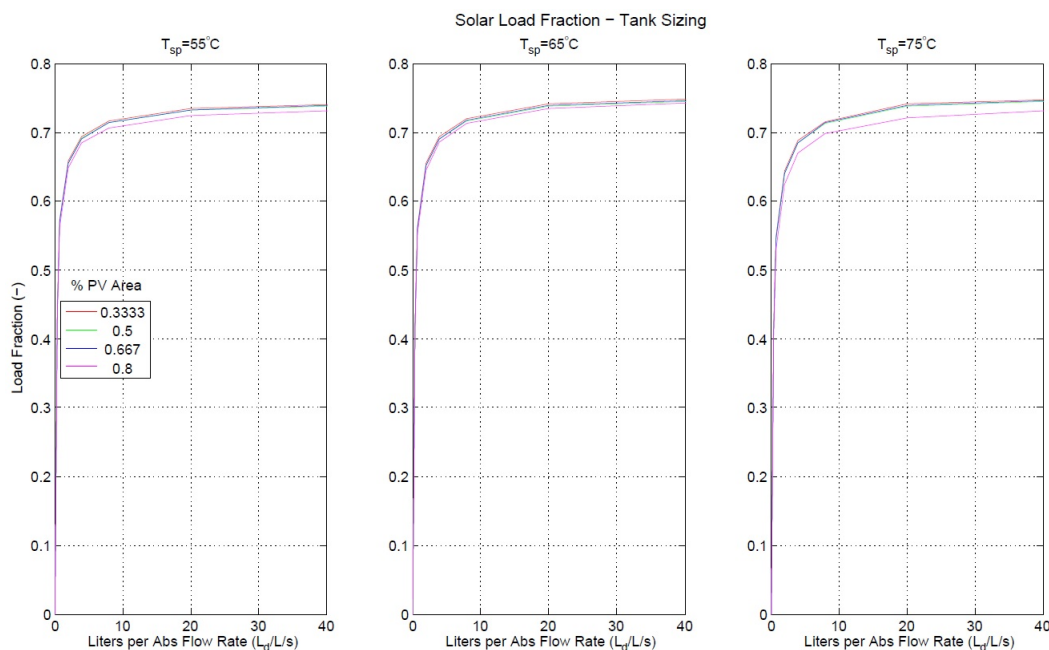


Figure 6.4: Effect of Tank Sizing

As shown in the figure, tank sizes above 20 Liters per L/s absorber flow rate (for all temperature set-points) allow the entire maximum load fraction to be reached. The proper size tank for the baseline system with a $47.2L/s$ absorber fan is approximately 944 Liters (250 gallons), or 20 Liters for each L/s of absorber flow rate. Further simulations use $L/(L/s)$ to discuss tank sizing.

6.5 Baseline System Performance

The baseline Orlando system, with approximately 104 m^2 of PV/T area (80% PV), requires a $684L/s$ regenerator fan to maintain 55°C . An appropriate size for the regenerator fan is $684L/s$, since it provides for maximum regeneration. A tank size of $20\text{ L}/(L/s)$, or 944 liters, is also necessary so that there is enough storage and no significant limits on absorption or regeneration.

Using a $47.2L/s$ absorber, there is adequate regeneration capacity provided by the regenerator, as shown in Figure 6.5. The tank concentration generally remains around a high concentration, since the average regeneration rate will generally out-pace the absorption rate. However, due to concentration and capacity limits of the desiccant tank, regeneration stops when the limits are

reached. For the $47.2L/s$ absorber, absorption does not stop due to filling the tank.

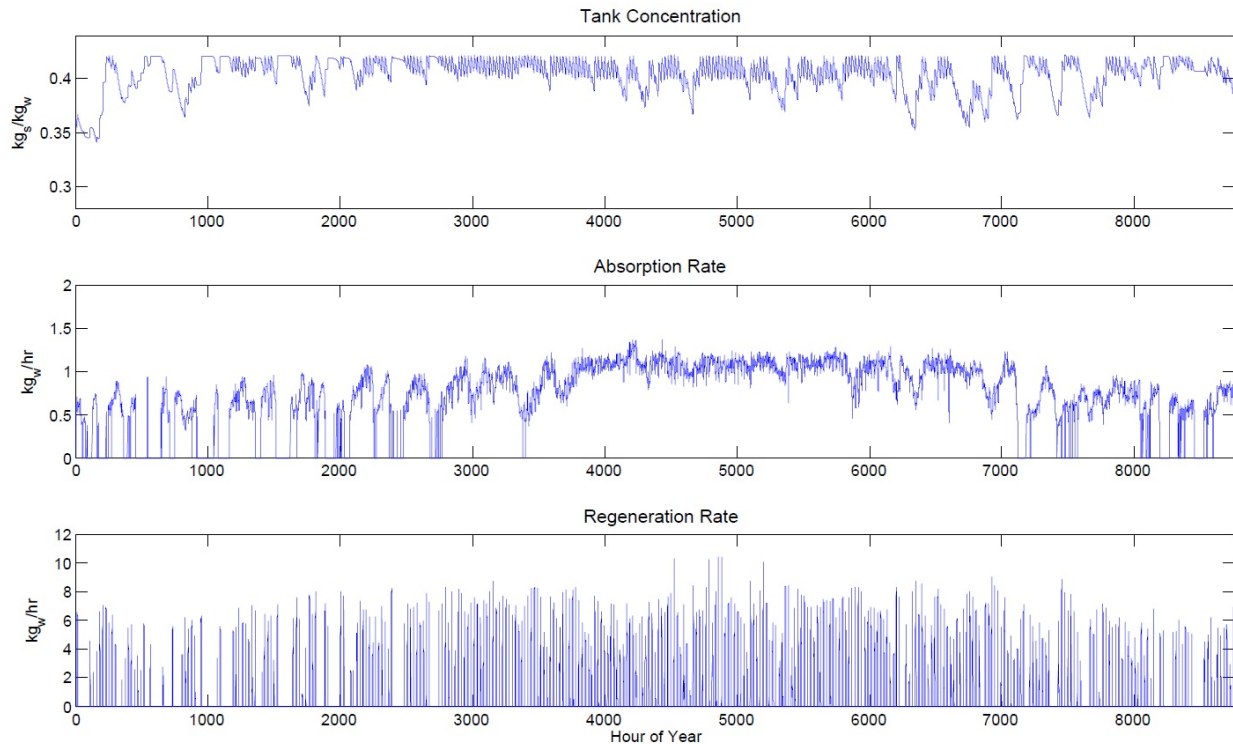


Figure 6.5: Absorption/Regeneration Performance - Baseline System

However, with a $118L/s$ absorber, too much absorption occurs and the average absorption rate is greater than regeneration rate. In other words, the absorber flow rate is too large for the PV/T array and regenerator installed in Orlando. Figure 6.6 shows that the tank concentration for a $118L/s$ absorber will reach the lower concentration limit many times over the course of a year. During these times, absorption will stop and the maximum absorption load will not be reached. Since the Orlando regenerator and PV/T system are adequately sized for a $47.2L/s$ absorber, the $47.2L/s$ absorber will be used in the baseline system model.

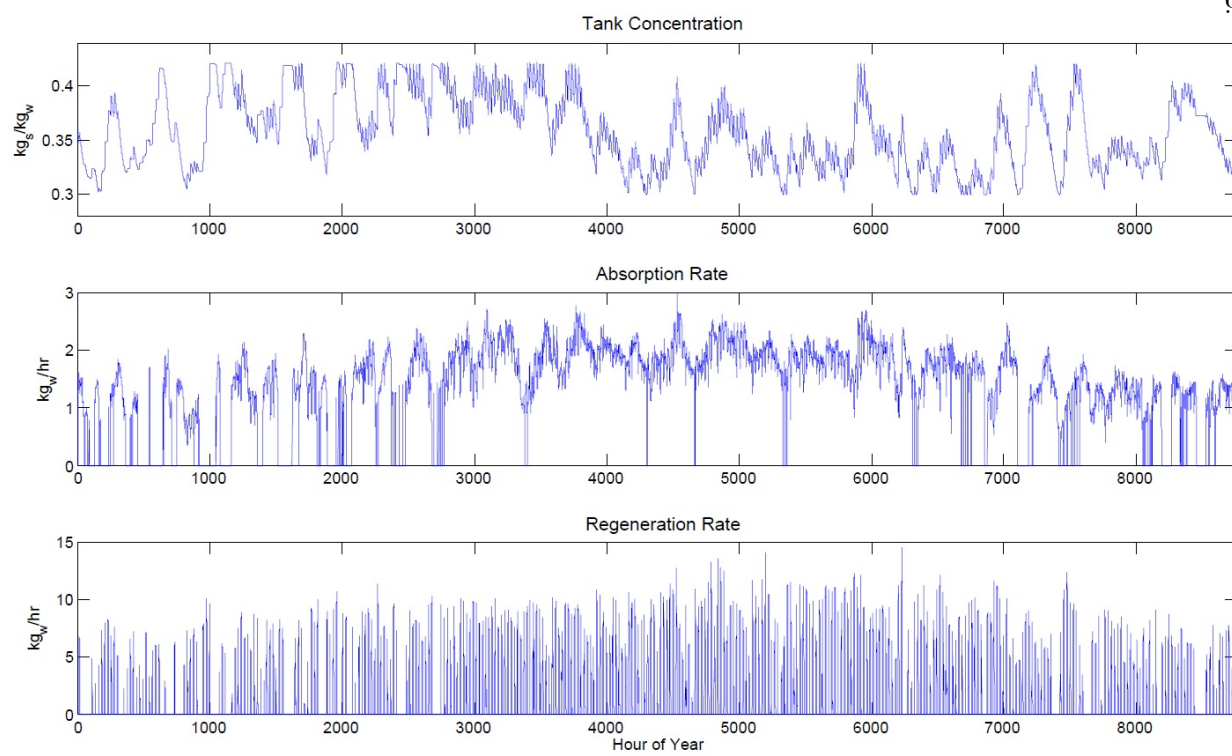


Figure 6.6: Increased Absorption - Baseline System

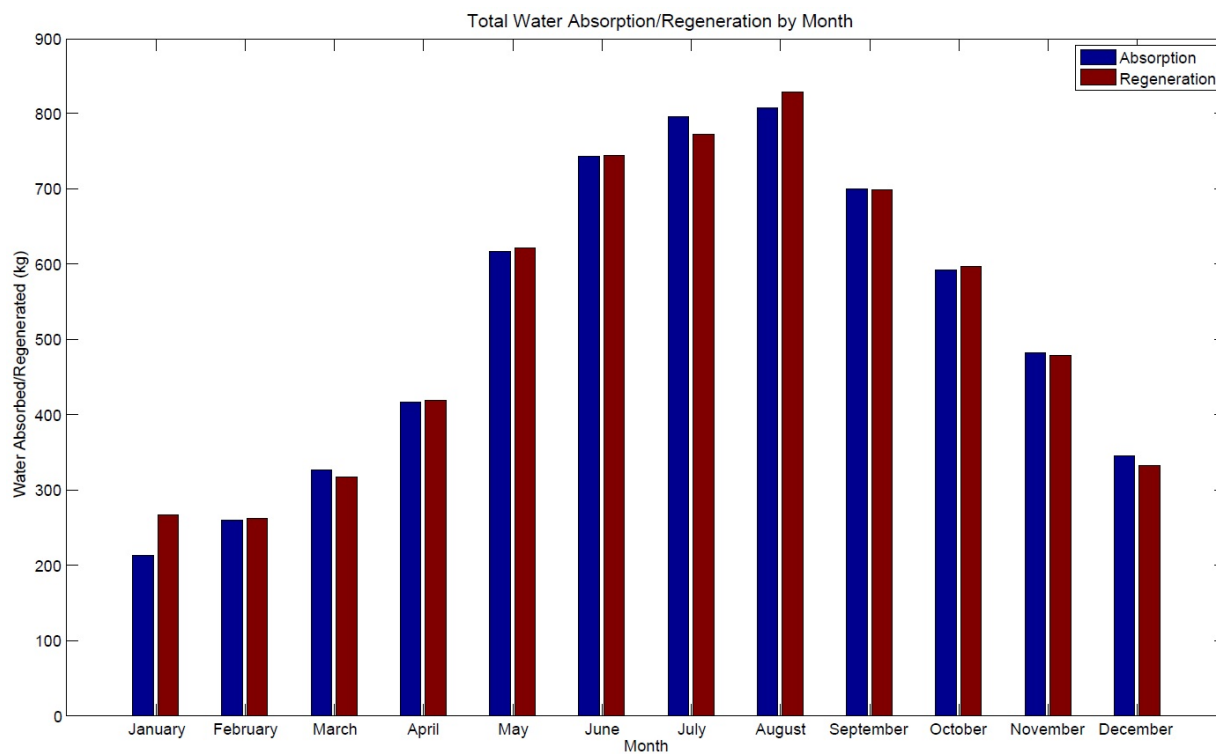


Figure 6.7: Monthly Water Absorption/Regeneration - Baseline System

Figure 6.7 compares the overall amount of water vapor absorber and regenerated for each month. Each month, both amounts are relatively equal, because of the limits imposed by the tank. If the tank concentration at the beginning of the month, or other specified time period, is higher, there is more capacity in the tank for water to be absorbed. In this case, more absorption can occur for that particular month. Also, in some months, outdoor humidity is too low for absorption to be necessary, so more regeneration will occur until the tank limits are reached.

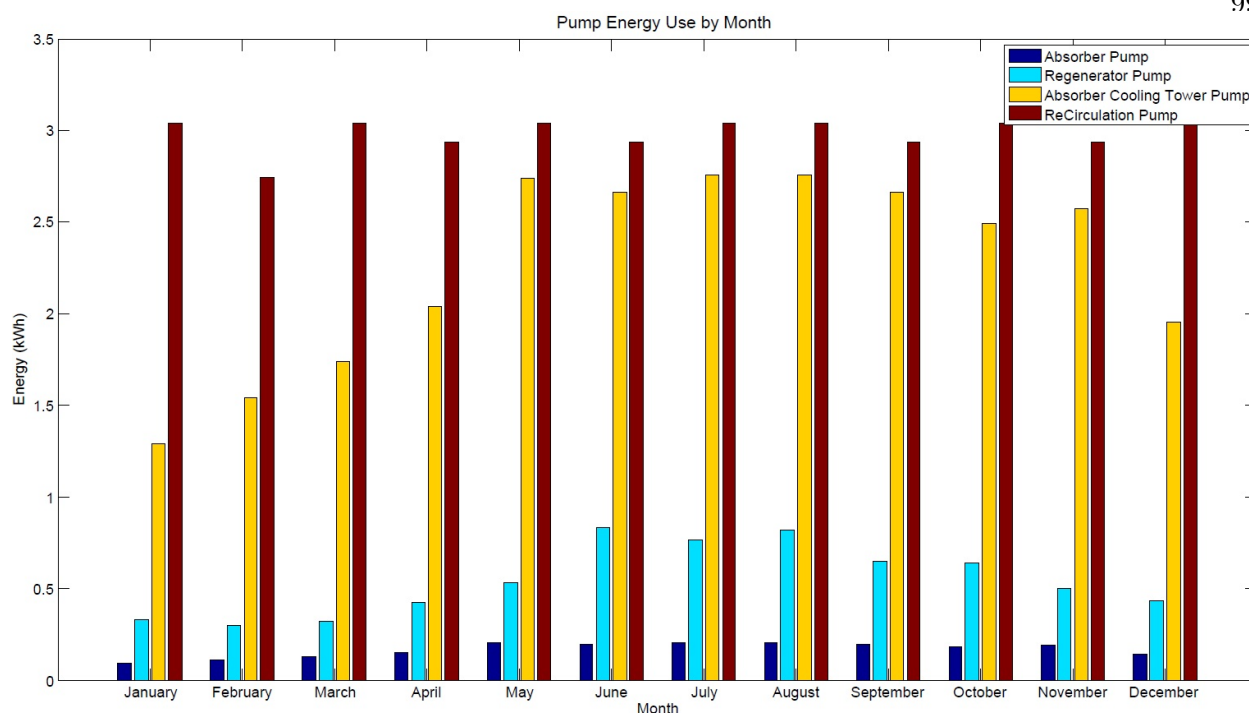


Figure 6.8: Pump Energy Usage - Baseline System

Figure 6.8 shows pump energy use for the various pumps in the system. All of the pumps in the baseline system use very little power, less than 3 kWh, compared to the fans and other system components. The recirculator pump uses the most power in the Orlando system, because it is constantly running to provide a fully mixed desiccant tank. For larger regenerator systems, the other pump sizes are also larger. The absorber pump is larger than the regenerator pump, because the regenerator is sized much larger than the absorber. The cooling tower pump is always much larger than the absorber pump, but operates only when the absorber is running. These two factors mean it uses much more energy than the or regenerator pump.

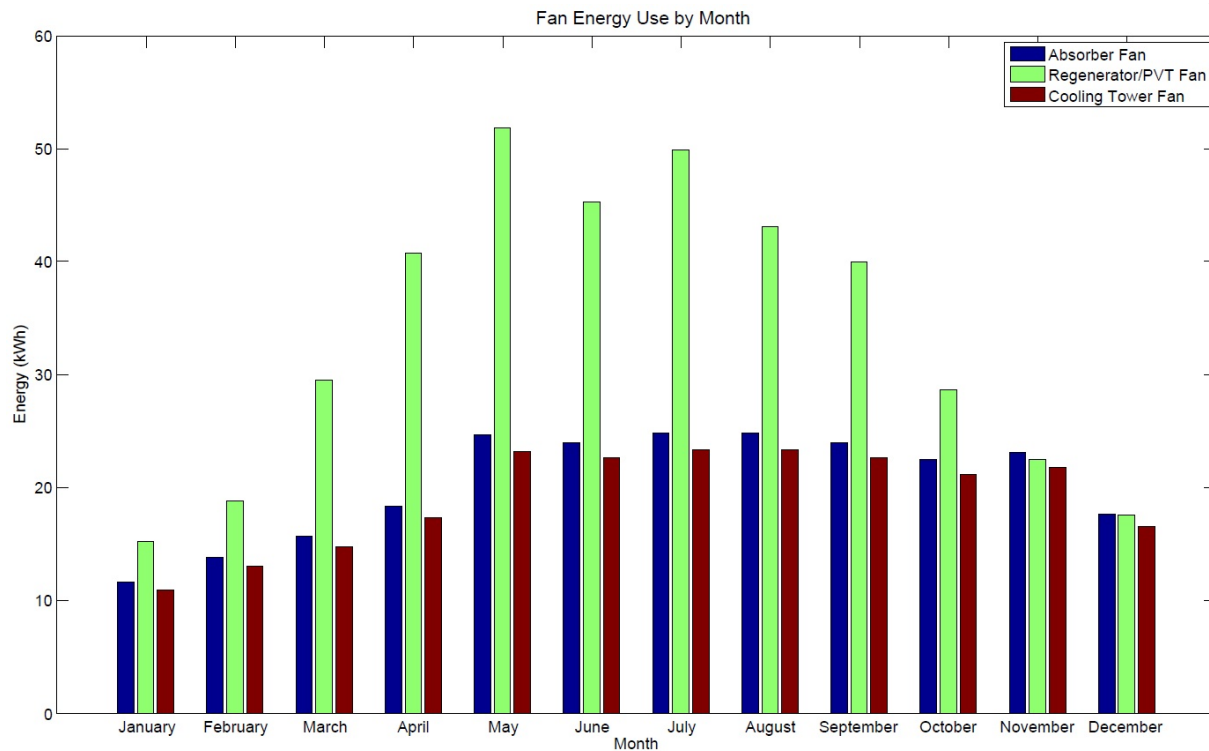


Figure 6.9: Fan Energy Usage - Baseline System

Figure 6.9 shows similar energy use compared to the pumps. Since the regenerator fan is much larger than the absorber fan, much more energy is used by the regenerator. The cooling tower and absorber fans are similarly sized for this and all other systems. Combined energy use for all system components is generally less than 100 kWh per month. The regenerator fan, while much larger than the absorber fan, is a variable speed fan that is not generally running at maximum speed. In fact, it is rarely necessary to run the fan at the maximum rate, so energy consumption stays low.

PV energy production for the 10kW system produces far more energy than is required by the pumps and fans in the desiccant system. Larger PV/T arrays may require more fan and pump energy to maintain a specific set-point, but the larger array would also produce more energy from the PV cells.

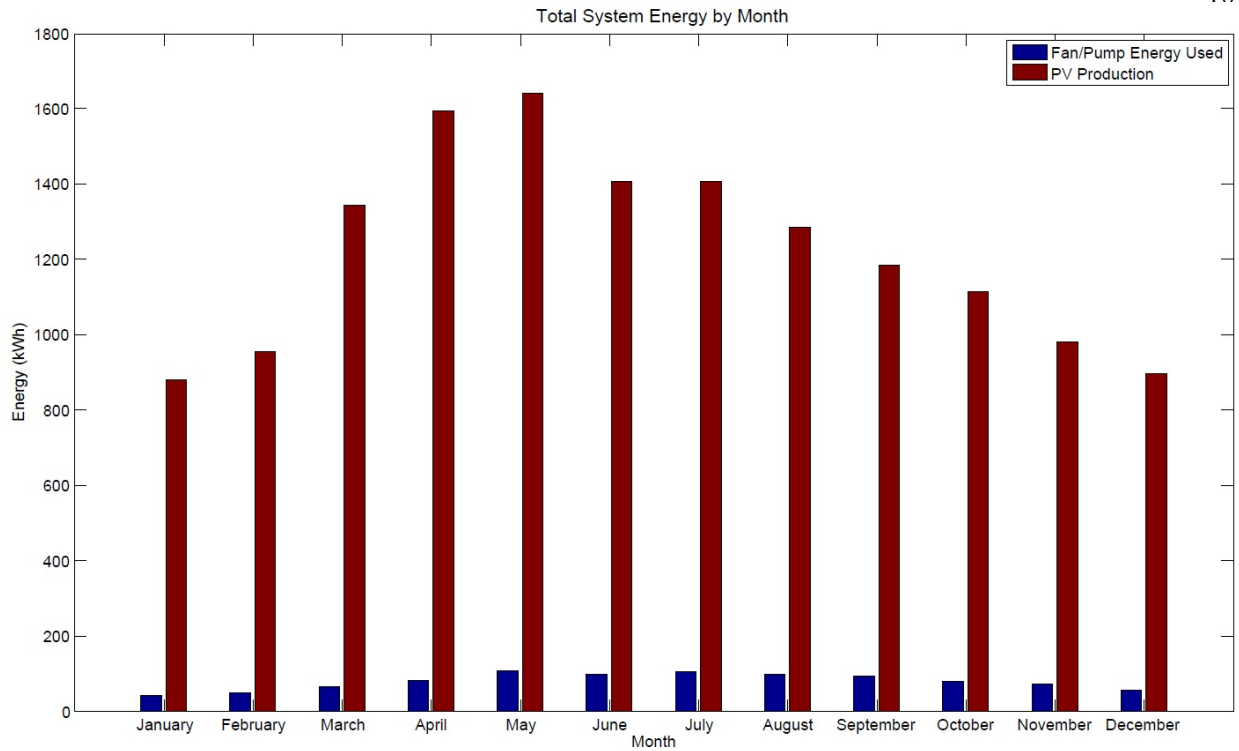


Figure 6.10: Total Energy Usage - Baseline System

The baseline system model shows that a 10kW PV/T system with 80% PV can provide adequate desiccant regeneration to meet a 47.2L/s absorber rate. The system can also produce enough energy to power all equipment related to the desiccant system, along with additional energy for the building. Additionally, only a small tank is required to provide storage for the desiccant.

Chapter 7

Modeling Results and Discussion

Each of the components of PV/T-liquid desiccant system must be sized appropriately to meet a load in a given location. Generally, loads can be based on ventilation requirements of a building. However for these simulations, four flow rates of 47.2 L/s (100 CFM), 118 L/s (250 CFM), 236 L/s (500 CFM), and 471.9 L/s (1000 CFM) are tested, which are intended to be sample outdoor air flow rates for different building sizes. The building load calculation is the amount of outdoor air that the absorber can dehumidify in a given climate, and is based on flow rate and the outdoor air conditions. Several climates are analyzed due to differing characteristics, and include the following:

- Orlando, FL
- Guam, US Territory
- Honolulu, HI
- Houston, TX
- Atlanta, GA
- Washington DC
- Phoenix, AZ

The description for each chosen climate includes the ASHRAE climate zone, ASHRAE cooling design values, and details of the humidity levels throughout the year. An index value is given to

describe the amount of dehumidification required throughout the year, which is calculated using a method similar to calculating cooling “degree days.” Using TMY3 data for each climate, the average of the high and low humidity ratio for each day of the year is calculated. The difference between the average value and the set-point humidity of $0.008 \text{ kg}_v/\text{kg}_a$ is then found. The sum of the positive values creates the index value, which will be referred to as “dehumidification index.” The calculation of cooling degree days is calculated in the same way, except using temperature instead humidity ratio. While the cooling degree days values can be used to estimate cooling load, no such method has been determined for dehumidification load. The index is simply for comparison between climates. Most of the characteristics described are generally used for estimating/calculating cooling loads of buildings in various climates, especially the given ASHRAE design values.

Different PV/T array sizes and absorber fan sizes are simulated to determine the percentage of the load that can be met in each climate. For an appropriately sized PV/T array and absorber, the majority of the maximum load can be achieved. The regenerator fan size, as discussed in the baseline system modeling, is calculated to maintain the set-point temperature of a PV/T array in a given climate, at any time of year. In addition, tank capacities also affect the percentage, or fraction, of the load that can be met by the system. Different tank sizes are modeled to find the minimum tank size necessary to not affect the absorbed load. This chapter discusses the appropriate sizing for each component, for the selected climates.

7.1 Load Calculations

The absorber load calculation, or amount of water vapor that can be absorbed from the outdoor air, is affected by the humidity ratio of the air and the humidity at which the absorber does not operate. Outdoor air above the humidity ratio set-point of $0.008 \text{ kg}_v/\text{kg}_a$ requires dehumidification and is used to calculate the absorption load. When the outdoor air is below the set-point, the absorber does not need to operate and is not considered as part of the load. However, at certain times, the outdoor air humidity is higher than the set-point but dehumidification causes the outlet humidity ratio of the absorber to drop below the set-point. To account for the addi-

tional dehumidification, the extra water vapor absorbed during these times is added to the total load calculation for each simulation. The solar load fraction calculations, later in this chapter, add the additional absorbed water vapor to the maximum load value and the actual absorbed water vapor value. For each climate, the humidity ratios over the course of a year vary significantly, and affect each load calculation differently. This section discusses how the climate affects these load values. Simulations will analyze the amount of energy needed for dehumidification using a regenerator/absorber. However, an estimated value for each climate will be found for dehumidification using standard air-conditioning processes. Using the following Equation 7.1 from the ASHRAE Handbook for common air-conditioning processes involving the transfer of heat via air transport:

$$\frac{q_l}{Q} = C_l \Delta W \quad (7.1)$$

From the ASHRAE Handbook, q_l is the latent heat transfer rate in Watts. C_l is the air latent heat factor in $W/(L*s)$, which is 3010 at or near sea level. Q is the air volumetric flow rate in L/s , which is depends on the chosen absorber flow rate for the system. ΔW describes the air humidity ratio difference, in kg_w/kg_{da} . The equation is used as a simple estimate for how much energy is necessary for dehumidification using standard air-conditioning for each climate. The value in the equation is described in Watts (or kW) for a given air flow rate (in L/s), over the course of a year. Final units are shown in $kWh/(L/s)$ per year.

Orlando is described as a hot-humid climate falling under climate zone 2A. There are approximately 4571 cooling degree days (CDD10) per year, with cooling design temperatures of 34°C dry-bulb and 24°C wet-bulb. Using these values, the design humidity ratio is 0.014733 kg_v/kg_a . The dehumidification index for Orlando is estimated at 1.9845. The design values show that there will be a high sensible and latent cooling load during the summer months, but the high dehumidification index also indicates a large latent load throughout the year.

Figure 7.1 shows the outdoor air humidity ratio for Orlando, over the course of a year.

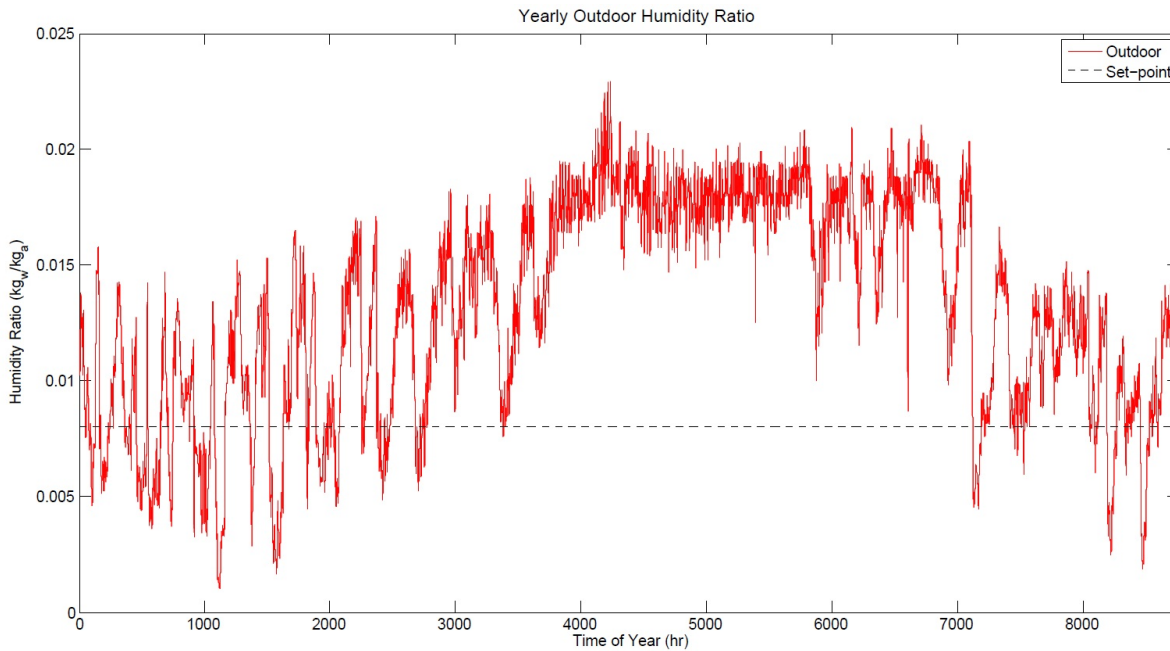


Figure 7.1: Yearly Outdoor Humidity Ratio - Orlando

The average humidity for the year is $0.0132 \text{ kg}_v/\text{kg}_a$. During the summer months, the humidity is far above the set-point, and requires much more dehumidification to reach $0.008 \text{ kg}_v/\text{kg}_a$. During these times, the set-point may not be reached. During the remaining months, the outdoor is only slightly above the set-point and the maximum dehumidification is possible, or the absorber is not running. Because the humidity ratio set-point cannot be reached during the summer months, the actual absorbed load will be less than the maximum load, for a load fraction less than 1.0. For standard air-conditioning, the estimated yearly energy use is $145.25 \text{ kWh}/(L/s)$.

Houston is also a hot-humid climate falling under climate zone 2A. There are approximately 4087 cooling degree days (CDD10) per year, with cooling design temperatures of 34°C dry-bulb and 25°C wet-bulb. Using these values, the design humidity ratio is $0.01635 \text{ kg}_v/\text{kg}_a$ with a dehumidification index estimated at 1.7188. The average humidity of Houston is $0.0123 \text{ kg}_v/\text{kg}_a$. The design values indicate a high sensible and latent cooling load during the summer months, but only slightly higher than Orlando. However, the lower dehumidification index indicates a large latent load throughout the year, but less than that of Orlando. While both of these climates are

very similar, Houston will have a larger dehumidification load during the peak season but less for the year. Figure 7.2 shows the outdoor air humidity ratio for Houston, over the course of a year.

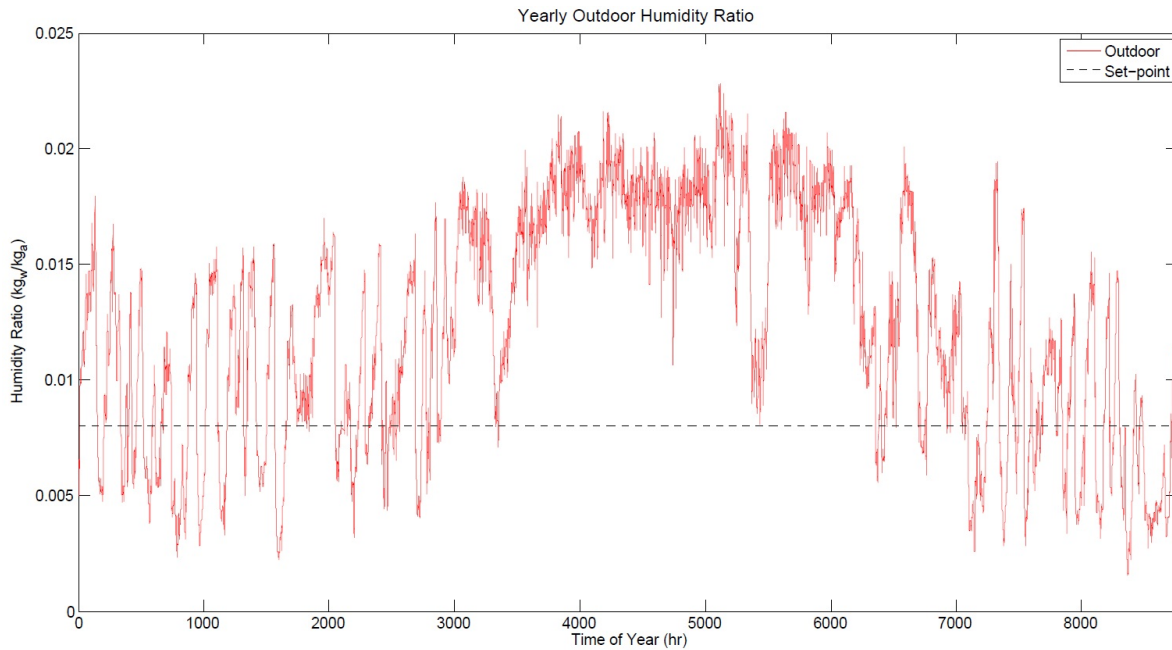


Figure 7.2: Yearly Outdoor Humidity Ratio - Houston

As in Orlando, the summer months require much more dehumidification than other months. Again, the set-point may not be reached for several months. The actual absorbed load will be less than the maximum load, for a load fraction less than 1.0. For standard air-conditioning, the estimated yearly energy use in Houston is $128.30 \text{ kWh}/(L/s)$.

Atlanta is a warm-humid climate falling under climate zone 3A. There are approximately 2799 cooling degree days (CDD10) per year, with cooling design temperatures of 33°C dry-bulb and 23°C wet-bulb. Using these values, the design humidity ratio is $0.01423 \text{ kg}_v/\text{kg}_a$ with a dehumidification index estimated at 0.9396. The average humidity of Atlanta is $0.009 \text{ kg}_v/\text{kg}_a$. The design values indicate a high sensible and latent cooling load during the summer months, similar to Orlando. However, significantly less cooling and dehumidification will be required throughout the year, based on the dehumidification index. Figure 7.3 shows the outdoor air humidity ratio for Atlanta, over the course of a year.

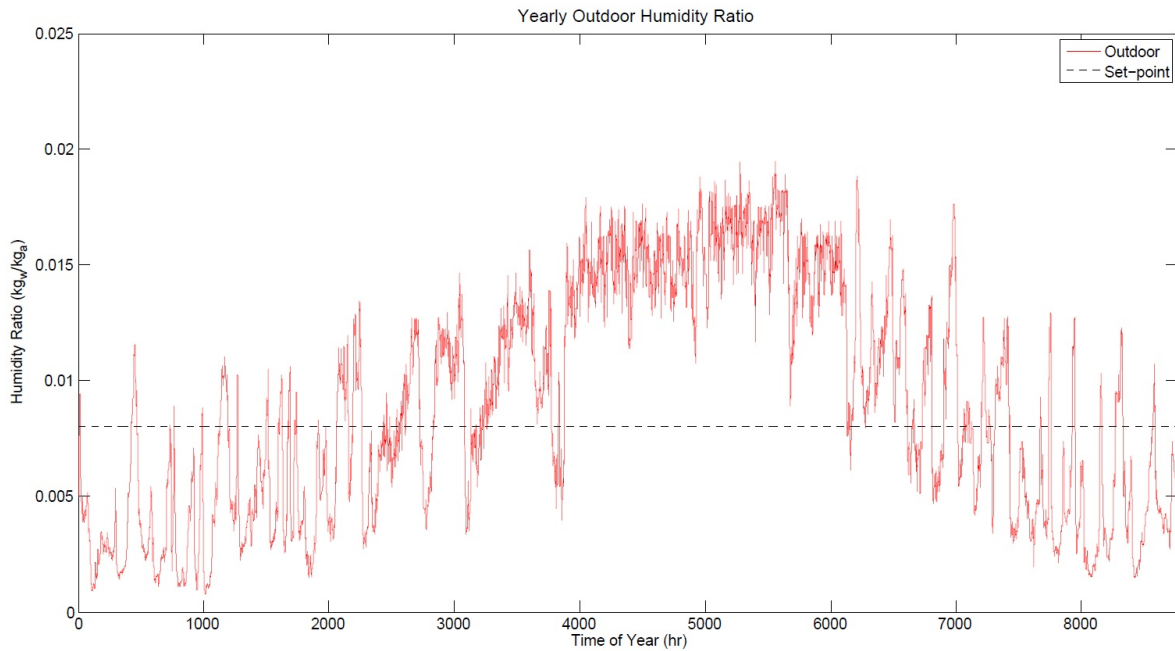


Figure 7.3: Yearly Outdoor Humidity Ratio - Atlanta

From the figure, the set-point is more likely to be maintained throughout the year. Assuming that the regenerator is sized properly, the load fraction will be closer to 1.0 than those other two mentioned climates. For standard air-conditioning, the estimated yearly energy use in Atlanta is $70.23 \text{ kWh}/(L/s)$.

Washington DC is a mixed-humid climate falling under climate zone 4A. There are approximately 2439 cooling degree days (CDD10) per year, with cooling design temperatures of 34°C dry-bulb and 24°C wet-bulb. Using these values, the design humidity ratio is $0.01472 \text{ kg}_v/\text{kg}_a$ with a dehumidification index estimated at 0.6722. The average humidity of DC is $0.0075 \text{ kg}_v/\text{kg}_a$. The design values show that at peak summer conditions, the climate can be similar to the other climates. Based on the dehumidification index, there will be much less cooling and dehumidification load compared to the other climates. Figure 7.4 shows the outdoor air humidity ratio for Washington DC, over the course of a year.

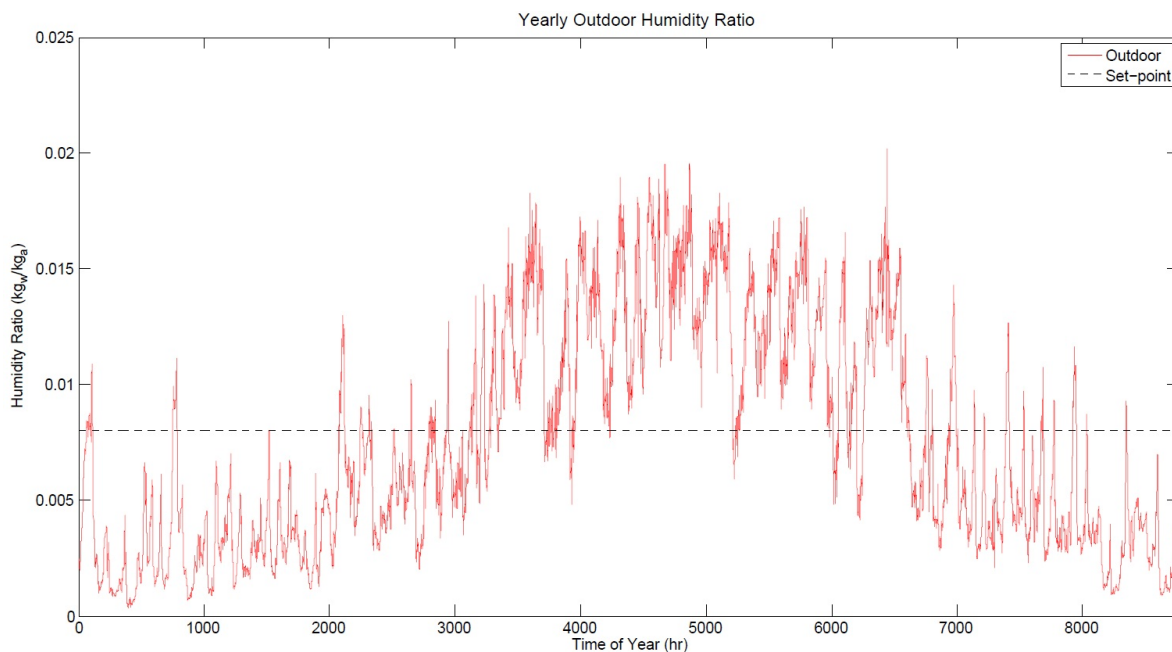


Figure 7.4: Yearly Outdoor Humidity Ratio - Washington DC

The figure verifies the previously mentioned assumptions. It also indicates that the maximum load fraction will be much closer to 1.0 than those other climates. However, it will be similar to that of Atlanta. For standard air-conditioning, the estimated yearly energy use in DC is $49.58 \text{ kWh}/(L/s)$.

Phoenix is a hot-dry climate falling under climate zone 2B. There are approximately 4681 cooling degree days (CDD10) per year, with cooling design temperatures of 42°C dry-bulb and 21°C wet-bulb. Using these values, the design humidity ratio is $0.00761 \text{ kg}_v/\text{kg}_a$ with a dehumidification index estimated at 0.2439. The average humidity of Phoenix is $0.0061 \text{ kg}_v/\text{kg}_a$. Based on the dehumidification index, design humidity, and average humidity, there will be an almost non-existent dehumidification load. The sensible load is dominant in this climate, and will only be analyzed because of a small rainy season that occurs. Since little dehumidification is required and assuming a reasonably sized regenerator, the maximum load fraction will be very close to 1.0. For standard air-conditioning, the estimated yearly energy use in Phoenix is $18.20 \text{ kWh}/(L/s)$. See Figure 7.5 for the outdoor air humidity ratio in Phoenix, over the course of a year.

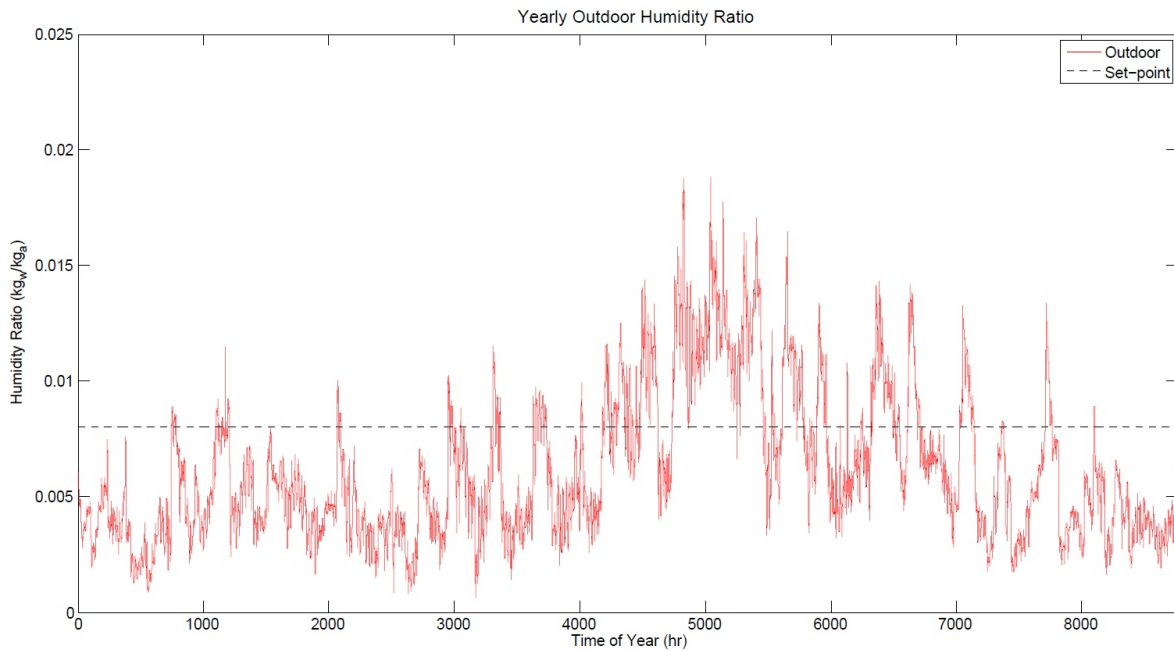


Figure 7.5: Yearly Outdoor Humidity Ratio - Phoenix

Figure 7.6 shows the outdoor air humidity ratio for Guam, over the course of a year. Guam is a very hot-humid climate falling under climate zone 1A. There are approximately 5939 cooling degree days (CDD10) per year, with cooling design temperatures of 31°C dry-bulb and 26°C wet-bulb. Using these values, the design humidity ratio is 0.0197 kg_v/kg_a with a dehumidification index estimated at 3.7642. The average humidity of Guam is 0.0183 kg_v/kg_a . Based on all design and index values, Guam will have a very heavy cooling load, both sensible and latent. The dehumidification load throughout the year, will be far greater than any of the other climates analyzed. The maximum load fraction will likely be much less than 1.0. For standard air-conditioning, the estimated yearly energy use in Guam is 271.75 $kWh/(L/s)$.

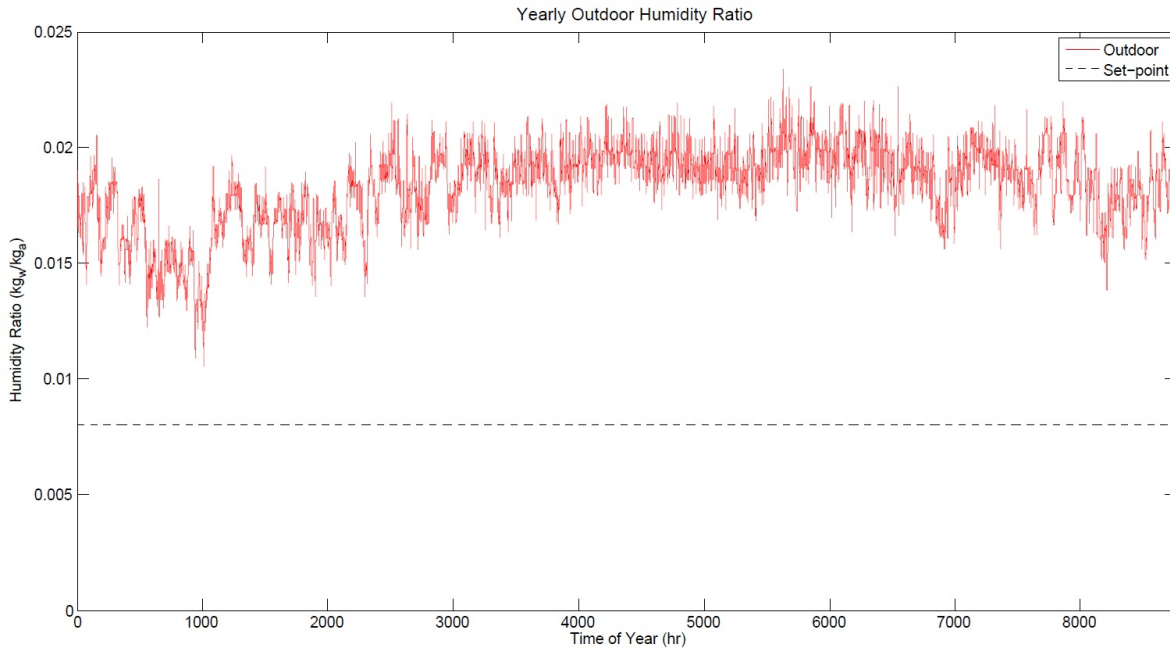


Figure 7.6: Yearly Outdoor Humidity Ratio - Guam

Figure 7.7 shows the outdoor air humidity ratio values for Honolulu. Honolulu is a very hot-humid climate falling under climate zone 1A. There are approximately 5527 cooling degree days (CDD10) per year, with cooling design temperatures of 31°C dry-bulb and 23°C wet-bulb. Using these values, the design humidity ratio is $0.01439 \text{ kg}_v/\text{kg}_a$ with a dehumidification index estimated at 1.9455. The average humidity of Honolulu is $0.0133 \text{ kg}_v/\text{kg}_a$. Based on all design and index values, Honolulu has a similar load profile of the zone 2A climates. The main difference between Honolulu and the other climates is the more constant cooling and dehumidification load throughout the entire year. There is less seasonal variation. The load fraction will be much less than 1.0, but higher than in Guam. For standard air-conditioning, the estimated yearly energy use in Honolulu is $140.04 \text{ kWh}/(L/s)$.

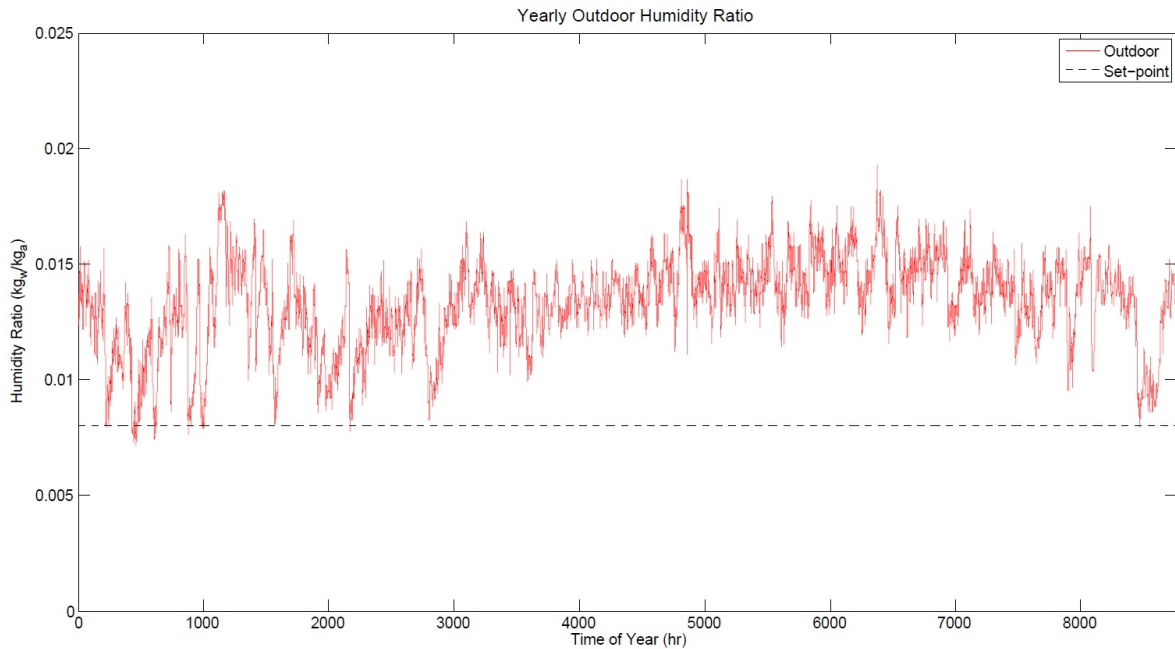


Figure 7.7: Yearly Outdoor Humidity Ratio - Honolulu

7.2 Regenerator and Fan Sizing

The calculated maximum load can only be reached if used with an appropriately sized PV/T array and regenerator, which means that over a given time period (one year) the regenerator must remove nearly the same amount of water vapor from the desiccant as is added from the absorption. Since the PV/T array and regenerator use the same air stream and blower fan, the regenerator fan is sized for a given PV/T area. The regenerator fan is simply sized to maintain a specified set-point temperature at the outlet of the PV/T array. Also, by sizing a fan based on set-point, the effects of temperature on regeneration rate can be assessed.

For each climate, solar insolation and temperature vary, which affect the amount of heat generated by the PV/T system. Flow rate increases to reduce the temperature of the air stream. The air flow rate needed to maintain the outlet temperature set-point is found for each PV/T array size. This section discusses the maximum fan flow rates for varying PV/T areas and temperature set-points.

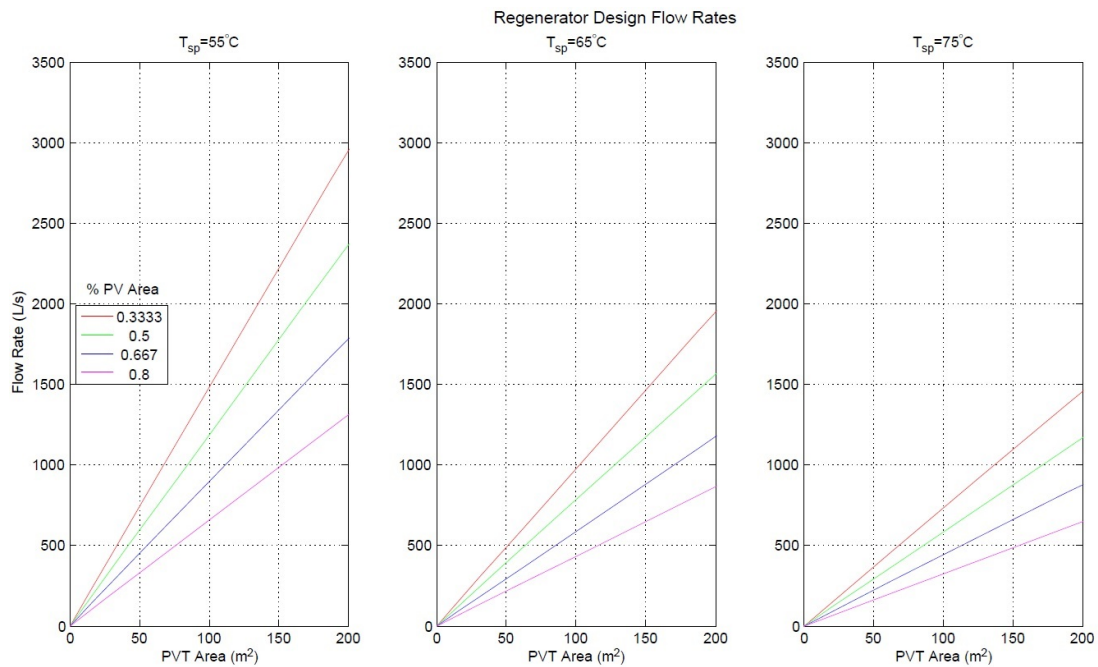


Figure 7.8: Regenerator Fan Sizes - Orlando

Figure 7.8 shows the regenerator fan sizes needed for Orlando at various PV/T sizes and % PV amounts. The baseline system is approximately 104 m^2 , 80% of which is PV. According to the figure, a 675 L/s regenerator fan is required for a temperature set-point of 55°C . For 65°C and 75°C , the approximate regenerator fan sizes are 450 and 350 L/s , respectively. Lower PV percentages require larger fans to maintain a set-point, since the thermal modules produce more heat than the PV panels. Also, at lower temperature set-points, maximum flow rates must be higher, so that flow rate can increase to reduce air temperature. At a low temperature set-point of 55°C , the fan sizes must increase significantly to be able to maintain 55°C . For some array configurations, the fan sizes may be unreasonably large by requiring much more power and a significant amount of air to pass through the small air channel.

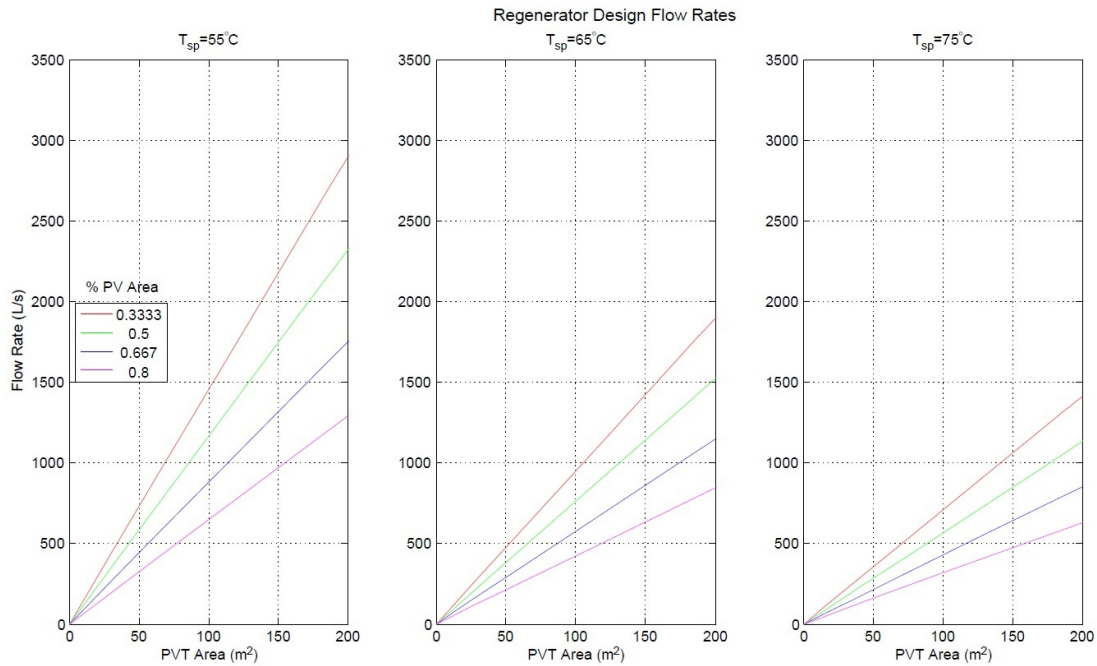


Figure 7.9: Regenerator Fan Sizes - Houston

Houston fan sizes, as shown in Figure 7.9, are virtually the same as those in Orlando. The two climates receive similar amounts of solar insolation, and temperatures are nearly the same. However, as discussed in the previous section, the Houston humidity ratios are slightly less than that of Orlando, meaning that less regeneration is required to meet the absorption load. In other words, a higher load fraction can be achieved with less regeneration, or smaller PV/T array size.

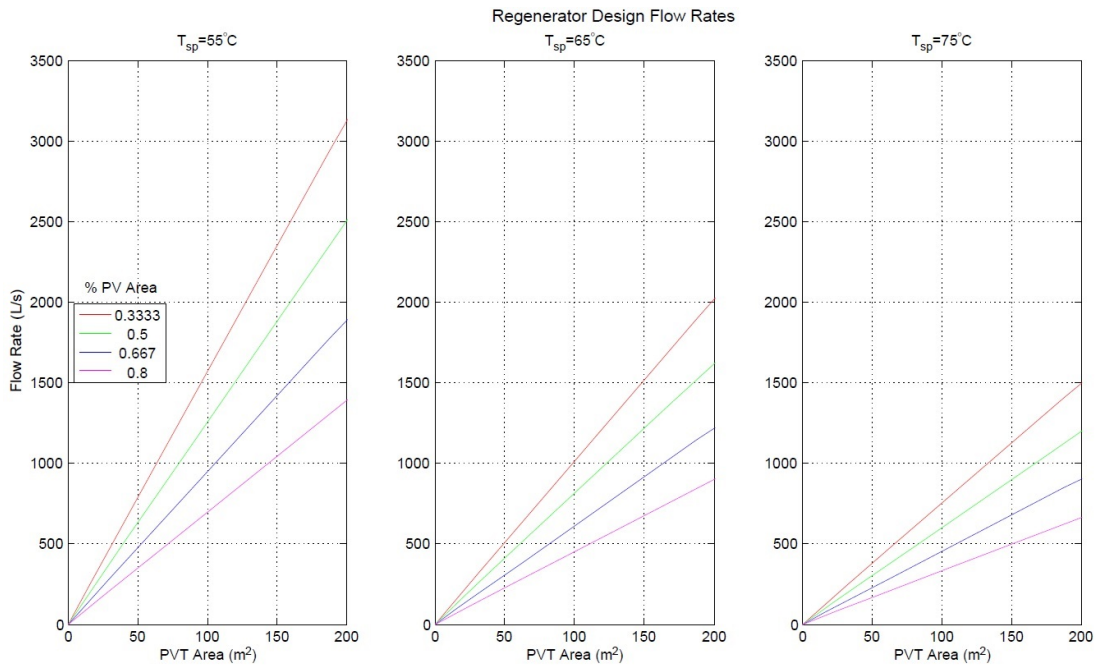


Figure 7.10: Regenerator Fan Sizes - Atlanta

Atlanta fan sizes, shown in Figure 7.10, are also similar to those in Orlando and Houston. Atlanta generally has only slightly lower air temperatures in the summer than Orlando and Houston, but less rainy days since it is further away from the coast. Atlanta receives similar amounts of solar radiation to that of Houston. According to the weather data, worst-case conditions mean that the fan sizes for Atlanta are modestly larger than the other two climates. However, the fan sizes are not significantly larger, and fan sizes for Orlando or Houston could be used. For the simulations, the calculated fan sizes from the figure are used. The Atlanta humidity ratios, as mentioned previously, are less than Orlando and Houston. Less regeneration is required to meet the absorption load, and a higher load fraction can be achieved with a smaller PV/T array size and fan flow rate.

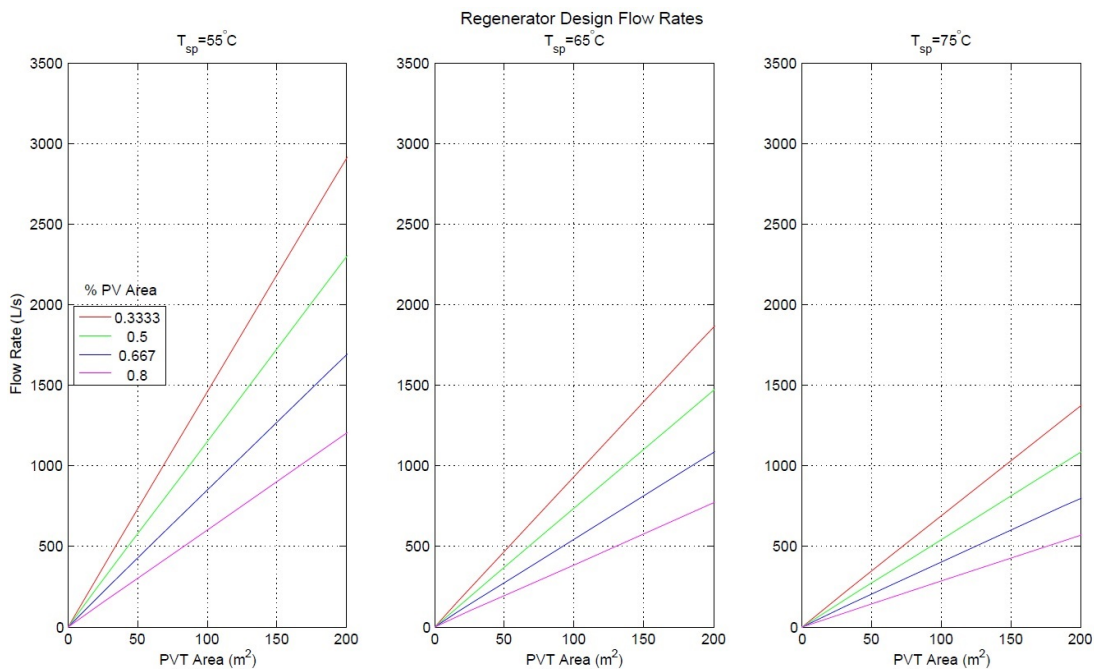


Figure 7.11: Regenerator Fan Sizes - Washingto nDC

Fan sizes for Washington DC, shown in Figure 7.11, are lower than in the previously mentioned climates. Less sunlight and lower temperatures do not produce as much heat in the PV/T array, so smaller fan flow rates can be used to maintain the same set-point temperatures. Larger PV/T areas may be necessary in Washington DC to increase the amount of regeneration. However, since little absorption is necessary, less regeneration is also necessary to reach the maximum load fraction. The trade-offs between these two will be discussed in the next section.

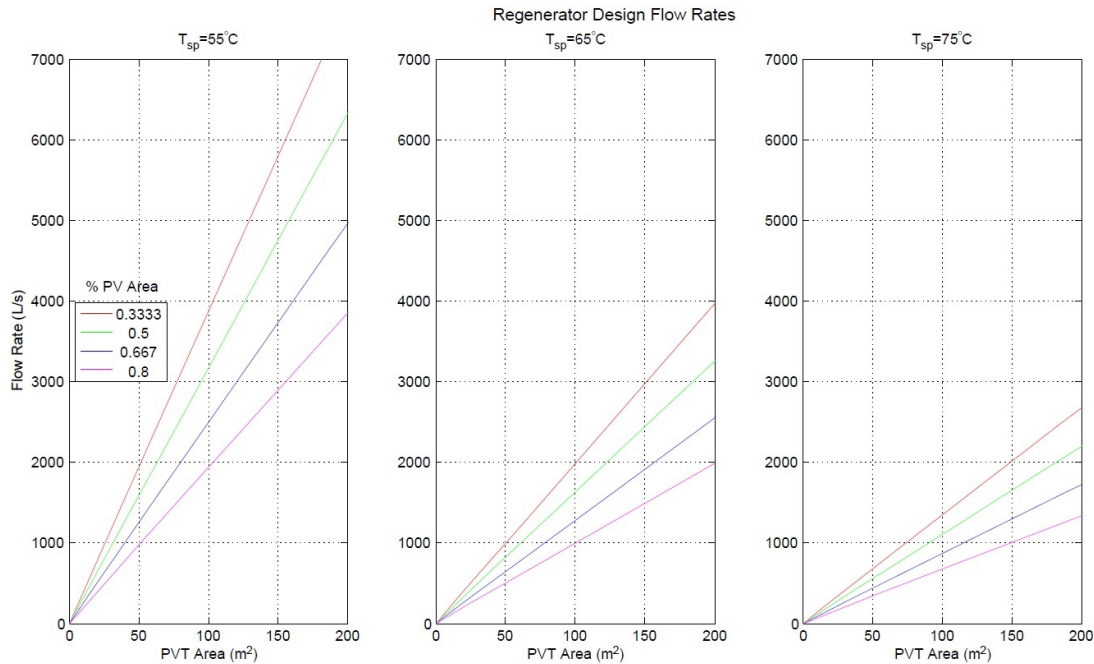


Figure 7.12: Regenerator Fan Sizes - Phoenix

Fan sizes for Phoenix, shown in Figure 7.12, are obviously much higher than all the other climates discussed. There is far more solar energy that is produced and very high temperatures, so the PV/T system must use much larger fans to reduce temperature and maintain the set-point temperatures. In some configurations, such as 33% PV, the fan sizes seem very large for the air channel. Higher PV percentage systems could be used instead to produce more PV energy and use less fan energy, and at the same time, create appropriate amounts of heat to maintain the given set-point temperatures. Using the fan sizes from the figure, the regeneration rates will be very high, and smaller PV/T array sizes can be used. Also, since only a little absorption is necessary for the year, required regeneration is also minimal and the necessary PV/T array size for Phoenix will be much smaller than in the other climates.

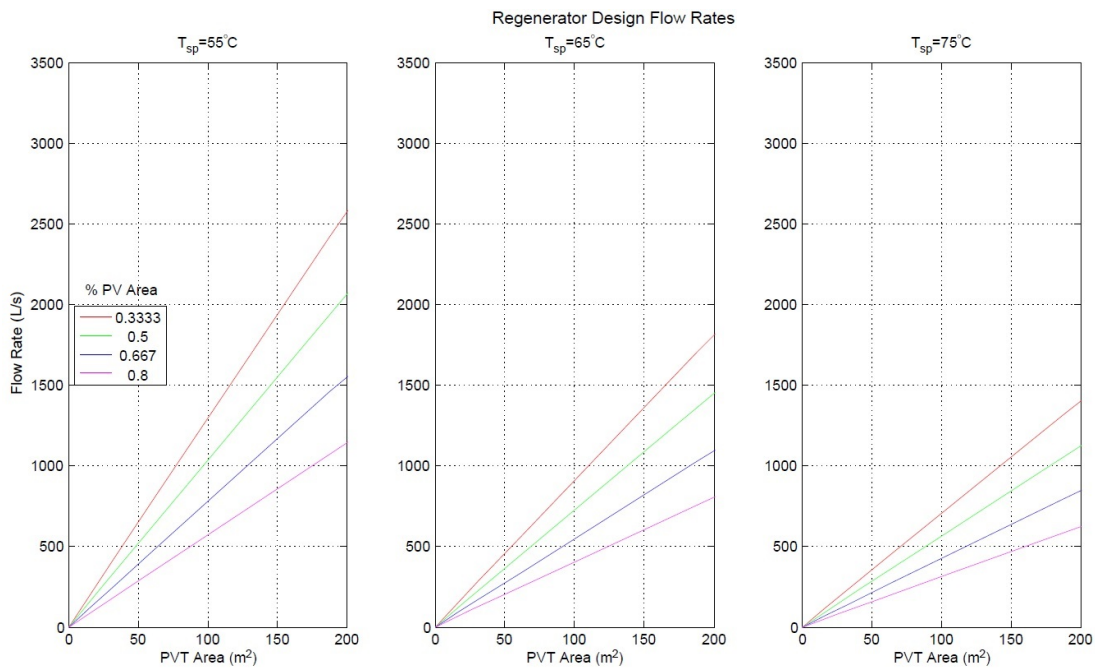


Figure 7.13: Regenerator Fan Sizes - Guam

Guam fan sizes, shown in Figure 7.13, are significantly lower than all the other climates. Average temperatures are consistent throughout the year, and far lower than the summer temperatures in the other climates. Also, since Guam is a marine climate, there is consistent rainfall reducing the amount of solar energy that can be produced. Worst-case conditions allow the PV/T array to maintain a set-point at much lower fan flow rates. Guam has the most humid climate in this analysis and requires far more absorption, so very large PV/T arrays are necessary to produce the heat and flow rate needed to regenerate the amount of water vapor that needs to be absorbed. Only large PV/T arrays can meet the maximum load fraction for the Guam climate.

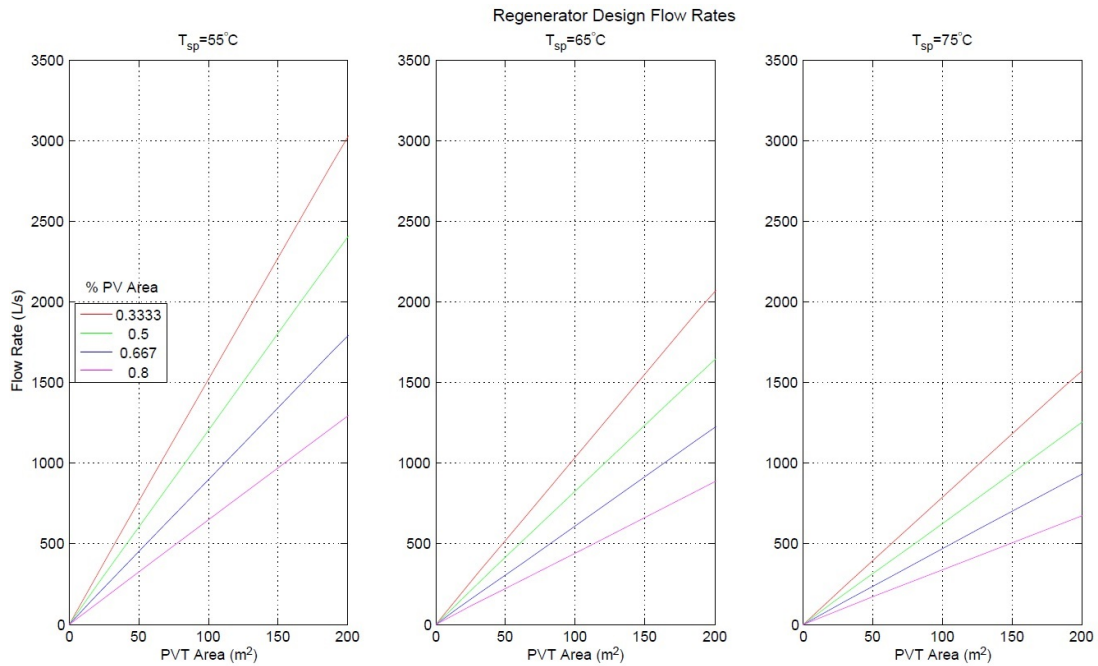


Figure 7.14: Regenerator Fan Sizes - Honolulu

Fan sizes for Honolulu, under worst-case conditions, are shown in Figure 7.14. These fan sizes are similar to those of Houston or Orlando. Honolulu has similar temperatures to that of Guam, but much less humid. Also, Honolulu has sunlight similar to that of Orlando. The constant, high level of humidity in Honolulu indicates that more absorption is necessary to meet the humidity ratio set-point. Since the PV/T flow rates are similar to Orlando but with a higher absorption requirement, larger PV/T array sizes may be necessary to meet the maximum load fraction.

7.3 Array and Absorber Sizing

Using the regenerator fan sizes calculated in the previous section, simulations are used to determine the load fraction met at different absorber flow rates and PV/T areas. For each climate, the solar load fraction is displayed as a function of “Total PVT Area per Absorber Flow Rate”, or $\text{m}^2/(\text{L}/\text{s})$. The total PV/T area is the total combined area for the PV panels and thermal collector modules. Each line displayed corresponds to different percentages of PV area. The black line refers to 0.80 or 80% PV area, which is 80% PV and 20% thermal modules. For any desired combination

of absorber flow rate and PV/T area, the solar load fraction can be found from the plot. Also shown on the plots is the maximum load fraction that can be reached.

An acceptable size for the PV/T array is one that reaches the maximum possible load fraction. The point on each curve where the load fraction reaches approximately 99% corresponds to the smallest PVT Area per L/s value, also referred to as f_{min} . Using the $m^2/(L/s)$ value, f_{min} , the maximum absorber flow rate for a specific PV/T area can be determined. Also, using the same value, the minimum PV/T area for a desired absorber flow rate (or building ventilation load) can be found. For example, if the solar load fraction curve reaches 99% - 100% at $0.75 m^2/(L/s)$, f_{min} is 0.75. If the PV/T array size, A_{pvt} , is already known, such as the $104 m^2$ in the baseline system, the maximum absorber flow rate, $\dot{V}_{abs_{max}}$, is found with Equation 7.2. The example shows that the maximum absorber flow rate is $138.67 L/s$. If a building in this example requires $100 L/s$ of dehumidified outdoor air, then the PV/T array in this system allows the highest load fraction to be met. If $250 L/s$ is required, then this is not an acceptable PV/T array size, since only a portion of maximum load fraction will be reached.

$$\dot{V}_{abs_{max}} = \frac{A_{pvt}}{f_{min}} \quad (7.2)$$

When the ventilation/absorber flow rate is known, the minimum PV/T array size, $A_{pvt_{min}}$ can be determined using Equation 7.3.

$$A_{pvt_{min}} = \dot{V}_{abs_{max}} f_{min} \quad (7.3)$$

For the same example, when f_{min} is $0.75 m^2/(L/s)$, the minimum PV/T area to meet a $100 L/s$ absorber load is $75 m^2$.

This section discusses PV/T array sizing for various absorber loads for each of the climates. Absorber rates are then compared to the minimum ventilation rates for two building sizes, 300 and $1000 m^2$. The $300m^2$ building is assumed to be two stories, with approximately $75-100m^2$ of available roof area. The $1000m^2$ building is assumed to be a three story building, with approximately

200-300 m^2 of available roof area.

ASHRAE Standard 62.1 [5], uses “Area Outdoor Air Rate”, or R_a , to describe the amount of outdoor air required for a given building size, in $L/s * m^2$. Values of R_a range from 0.3-2.4 $L/s * m^2$. 0.3 $L/s * m^2$ is the typical value for most types of spaces, including that of office spaces, and is selected as the value of comparison. Using the selected value of 0.3 $L/s * m^2$, the minimum ventilation rates are 90 and 300 L/s .

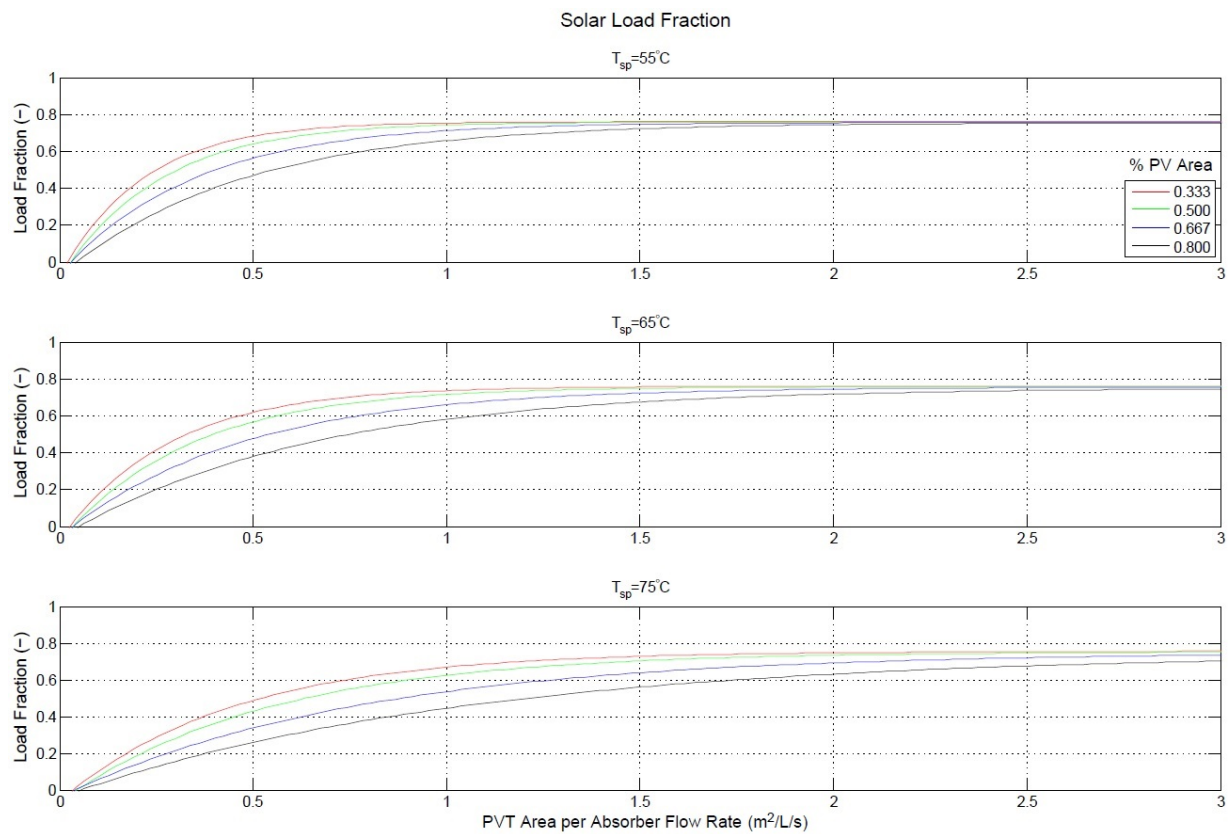


Figure 7.15: Solar Load Fraction - Orlando

Figure 7.15 shows the solar load fractions for Orlando at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Orlando climate. The four different absorber L/s rates were simulated for a variety of PV/T array sizes and PV percentages. Figure B.1 in the Appendix displays the curves fitted to the simulation data.

In the Orlando climate, the smallest array sizes that can be used are with a PV/T array

set-point of 55°C. The maximum load fraction is the same across all temperature set-points, but at higher temperature set-points, such as 75°C, the maximum load fraction can only be reached at much higher $m^2/(L/s)$ values. This indicates that the lower temperature set-points allow smaller PV/T arrays to be used to meet as much of the load as possible. While cost is not analyzed, PV panels and thermal collector modules are expensive, so smaller array sizes reduce overall cost of the system. It also decreases the space needed for the array, which may be a consideration.

At the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately 1.00 - 2.21 $m^2/(L/s)$. For a typical 80% PV area system, a 198.9 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 663 m^2 PV/T array would be needed. Obviously, those array size are much too large for the example multi-story buildings. The array may be acceptable for single level buildings with a much larger available roof area. In order to meet a smaller array footprint or use an increased absorber flow, a lower PV percentage, with more thermal collector modules, can be used instead. For a 33% PV area system, a 90 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 300 m^2 PV/T array would be needed. These PV/T array sizes would be reasonable for the example buildings, but are still near the maximum available roof space.

Table 7.1: Absorbed Water Vapor - Orlando

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	6675	0.76	1.48
117.99	16658	1.90	3.64
235.98	33194	3.79	7.21
471.95	65454	7.47	14.15

Table 7.1 shows the amount of absorbed water vapor, and the average and maximum rates at which water is absorbed in Orlando system. The values refer to PV/T array sizes where the maximum load fraction is met. If a specific amount of vapor needs to be absorbed, or an absorption rate is desired, the table is helpful in determining the absorber flow rate needed for those goals.

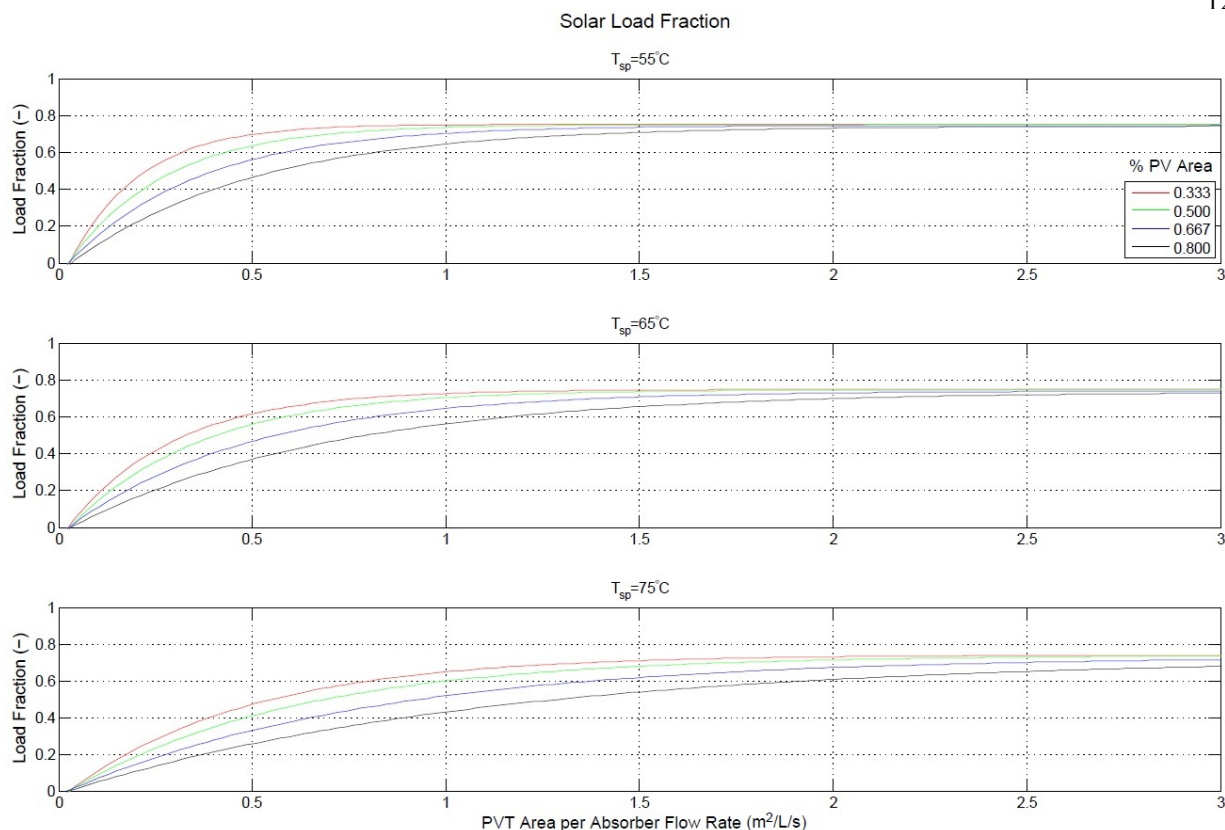


Figure 7.16: Solar Load Fraction - Houston

Figure 7.16 shows the solar load fractions for Houston at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Houston climate. Figure B.5 displays the curves fitted to the simulation data.

In the Houston climate, the smallest array sizes for any absorber rate can be used with a PV/T array set-point of 55°C, which is the same as in Orlando. In fact, the curves for all PV area percentages and temperature set-points are nearly identical to those for Orlando, so PV/T array sizing and absorber flow rates will be similar. As in Orlando, at the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately 0.86 - 2.23 $m^2/(L/s)$. For a typical 80% PV area system, a 200.7 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 669 m^2 PV/T array would be needed. These sizes are similar to those in Orlando, and also too large for the given multi-story building sizes. For a 33% PV area system, a 77.4 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2

building, an 258 m^2 PV/T array would be needed.

Table 7.2: Absorbed Water Vapor - Houston

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	5892	0.67	1.49
117.99	14692	1.68	3.67
235.98	29195	3.33	7.33
471.95	57242	6.53	14.58

Table 7.2 contains the water vapor absorption rates for systems in Houston. The values refer to PV/T array sizes where the maximum load fraction is met. As expected, Houston requires less dehumidification than in Orlando, so average and total absorption rates are slightly lower.

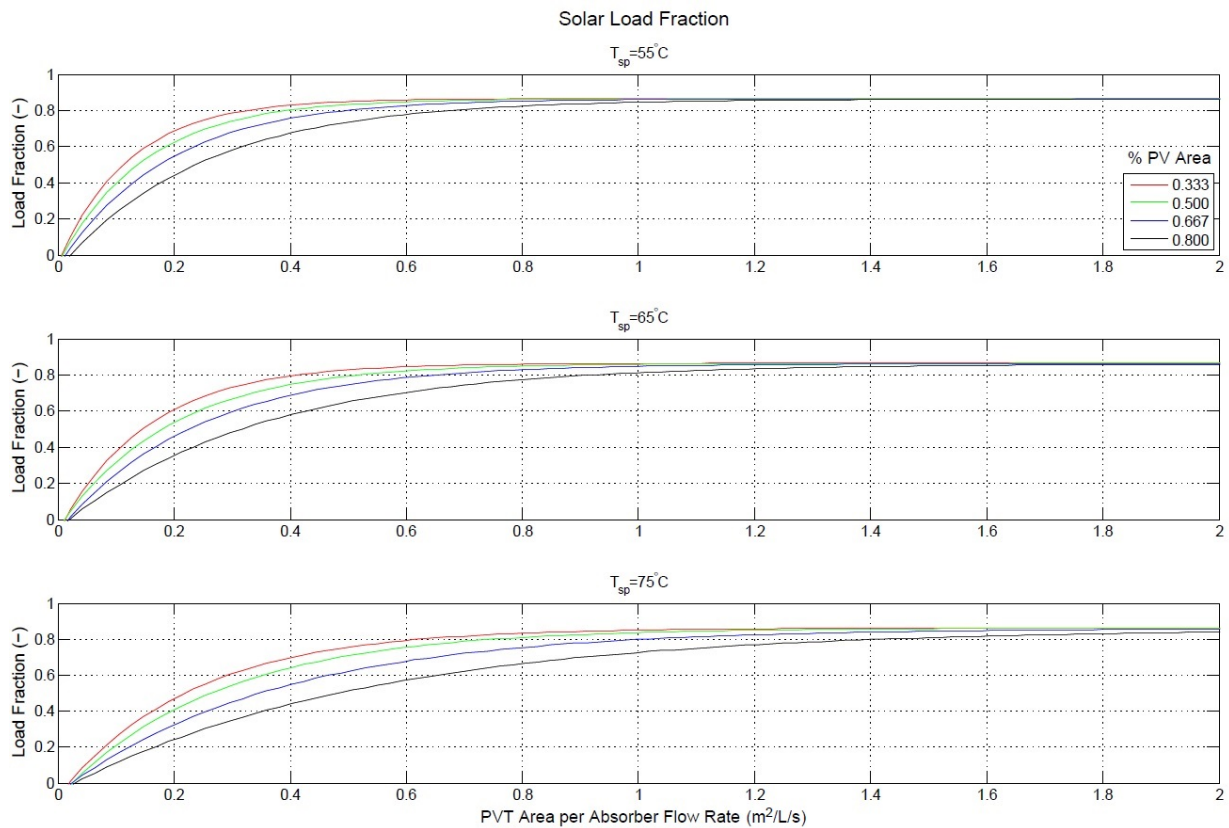


Figure 7.17: Solar Load Fraction - Atlanta

Figure 7.17 shows the solar load fractions for Atlanta at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Atlanta climate. Figure B.4 displays the curves fitted to the simulation data.

For the climate of Atlanta, the smallest array sizes for any absorber rate can be used with a PV/T array set-point of 55°C, which is the same as in Orlando and Houston. At the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately 0.574 - 1.17 $m^2/(L/s)$. For a typical 80% PV area system, a 105.3 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 351 m^2 PV/T array would be needed. These sizes are much smaller than in Orlando, and are close to fitting on multi-story buildings. For a 33% PV area system, a 51.66 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 172.2 m^2 PV/T array would be needed. These array sizes are much smaller and of adequate size for the example multi-story buildings.

Table 7.3: Absorbed Water Vapor - Atlanta

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	3580	0.41	1.21
117.99	8943	1.02	3.02
235.98	17847	2.04	6.03
471.95	35568	4.06	12.01

Table 7.3 contains the water vapor absorption rates for systems in Atlanta. All values refer to PV/T array sizes where the maximum load fraction is met. Atlanta requires significantly less dehumidification than Orlando and Houston, and this is reflected in absorption totals that are nearly half of that of Orlando.

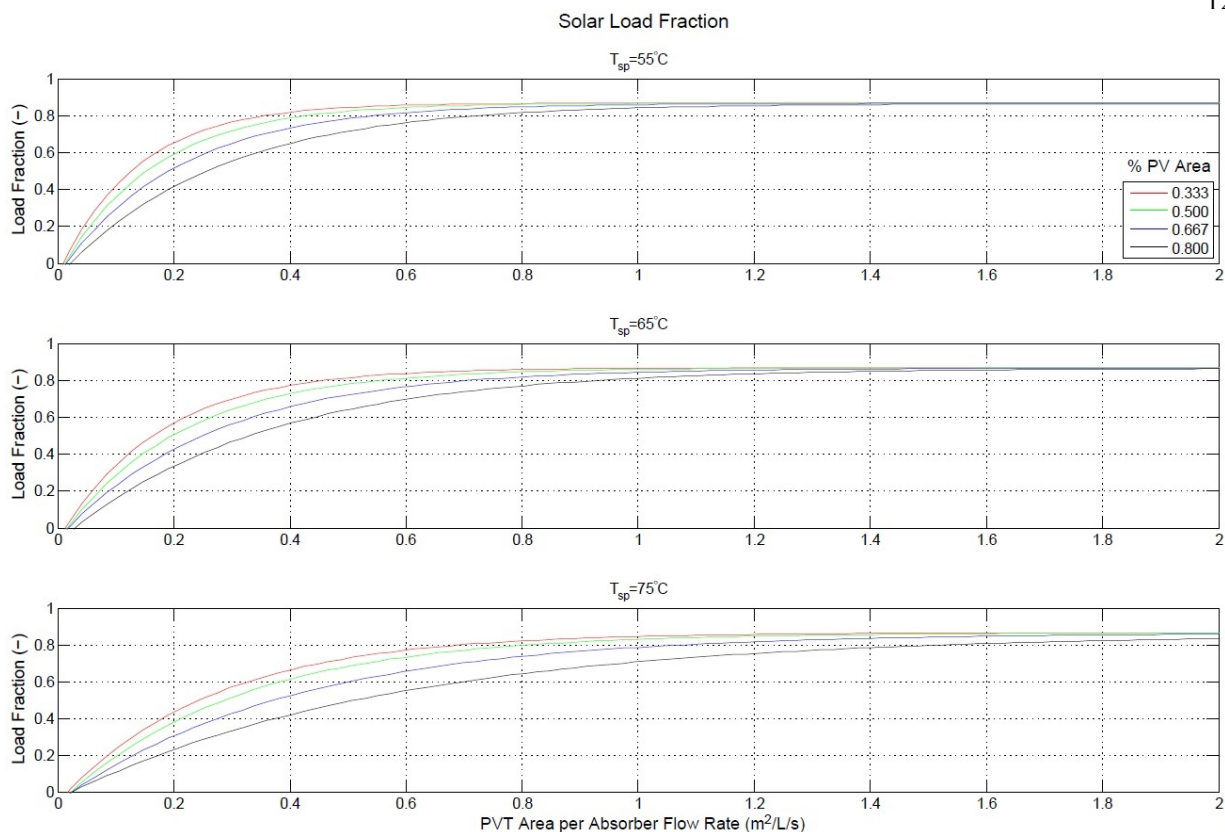


Figure 7.18: Solar Load Fraction - Washington DC

Figure 7.18 shows the solar load fractions for Washington DC at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Washington DC climate. Figure B.6 displays the curves fitted to the simulation data.

For Washington DC climate, the smallest array sizes for any absorber rate can be used with a PV/T array set-point of 55°C , which is the same as the other climates. At the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately $0.64 - 1.28 \text{ m}^2/(\text{L/s})$. For an 80% PV area system, a 115.2 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 384 m^2 PV/T array would be needed. Again, these sizes are much smaller than in Orlando, but still too large for the given building sizes. For a 33% PV area system, a 57.6 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 192 m^2 PV/T array would be needed. These values and curves are very similar to Atlanta, and sizing is approximately the same. This means that Atlanta and DC can use smaller

PV/T array sizes than in Orlando or Houston, due to the lower humidity and duration of the year that absorption needs to occur.

Table 7.4: Absorbed Water Vapor - Washington DC

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	2746	0.31	1.33
117.99	6864	0.78	3.19
235.98	13692	1.56	6.26
471.95	27256	3.11	12.18

Table 7.4 contains the water vapor absorption rates for systems in Washington. All values refer to PV/T array sizes where the maximum load fraction is met. Almost no absorption is necessary in the colder months, so absorption rates are significantly lower than the other climates. In fact, a 500 *L/s* absorber in Washington DC provides as much water absorption as an absorber at half the flow rate in Orlando.

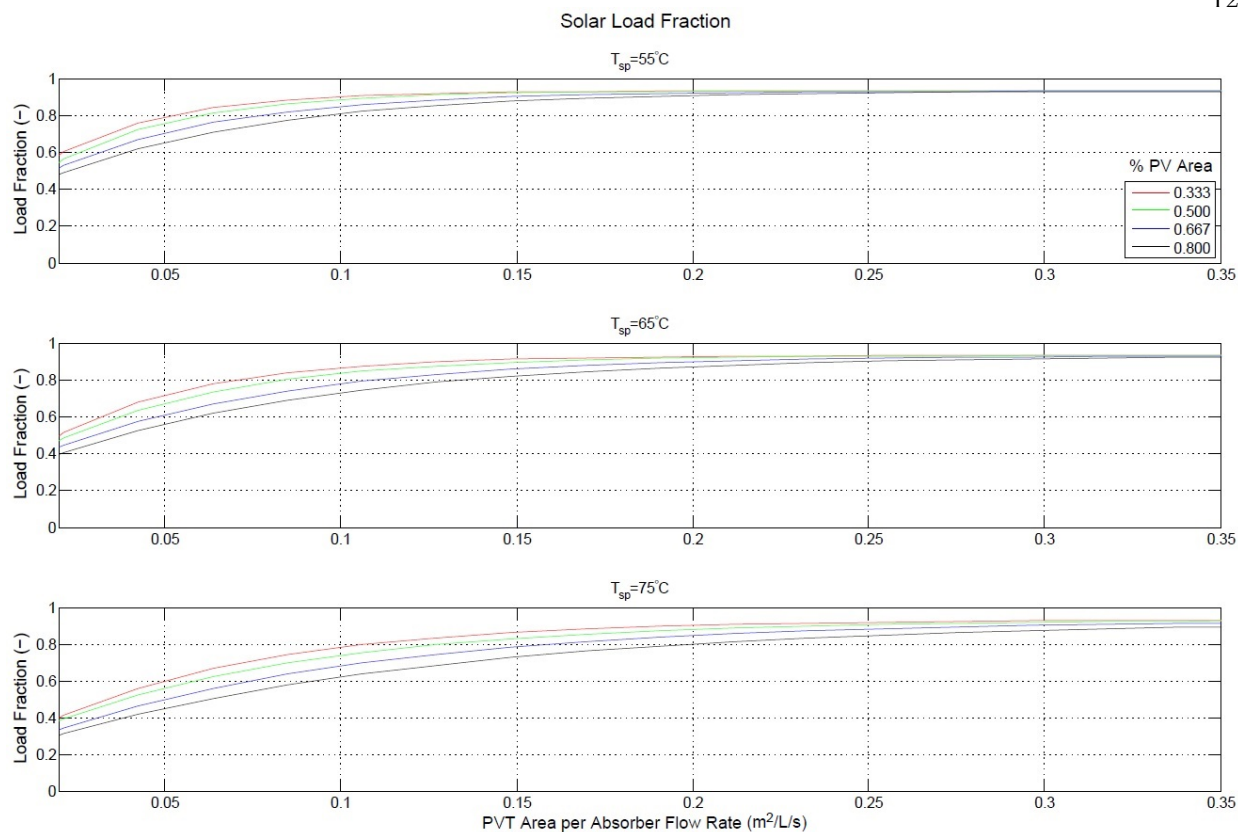


Figure 7.19: Solar Load Fraction - Phoenix

Figure 7.19 shows the solar load fractions for Phoenix at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Phoenix climate. Figure B.7 displays the curves fitted to the simulation data.

For the Phoenix climate, the smallest array sizes for any absorber rate can be used with a PV/T array set-point of 55°C , which is the same as all other climates. However, the regenerator flow rates are very high for the PV/T array sizes necessary in this climate. The high flow rates are due to the need to maintain the temperature set-point, even for small arrays. Much more heat is produced behind the PV/T array than in any other climate discussed in this thesis. A more appropriate set-point of 65°C could be used instead, but for comparison to other systems 55°C will still be used. The maximum load fraction is then reached between approximately $0.32 - 0.78 \text{ m}^2/(\text{L/s})$. For an 80% PV area system, a 70.2 m^2 PV/T array would be used for a 300 m^2 building.

For a 1000 m^2 building, an 234 m^2 PV/T array would be needed. For a 33% PV area system, a 27.9 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 96 m^2 PV/T array would be needed. These array sizes are very small compared to the others, so they can fit on any typical roof footprint. Unfortunately, Phoenix has such a short amount of the year that absorption can occur, so this type of system may be too costly and unnecessary.

Table 7.5: Absorbed Water Vapor - Phoenix

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	1102	0.13	1.19
117.99	2753	0.31	2.95
235.98	5506	0.63	5.90
471.95	11004	1.26	11.79

Table 7.5 contains the water vapor absorption rates for systems in Phoenix. All values refer to PV/T array sizes where the maximum load fraction is met. As in Washington DC, almost no absorption is necessary for most of the year. Due to the lower humidity and lack of a need to dehumidify for most of the year, a desiccant system does not seem necessary in Phoenix.

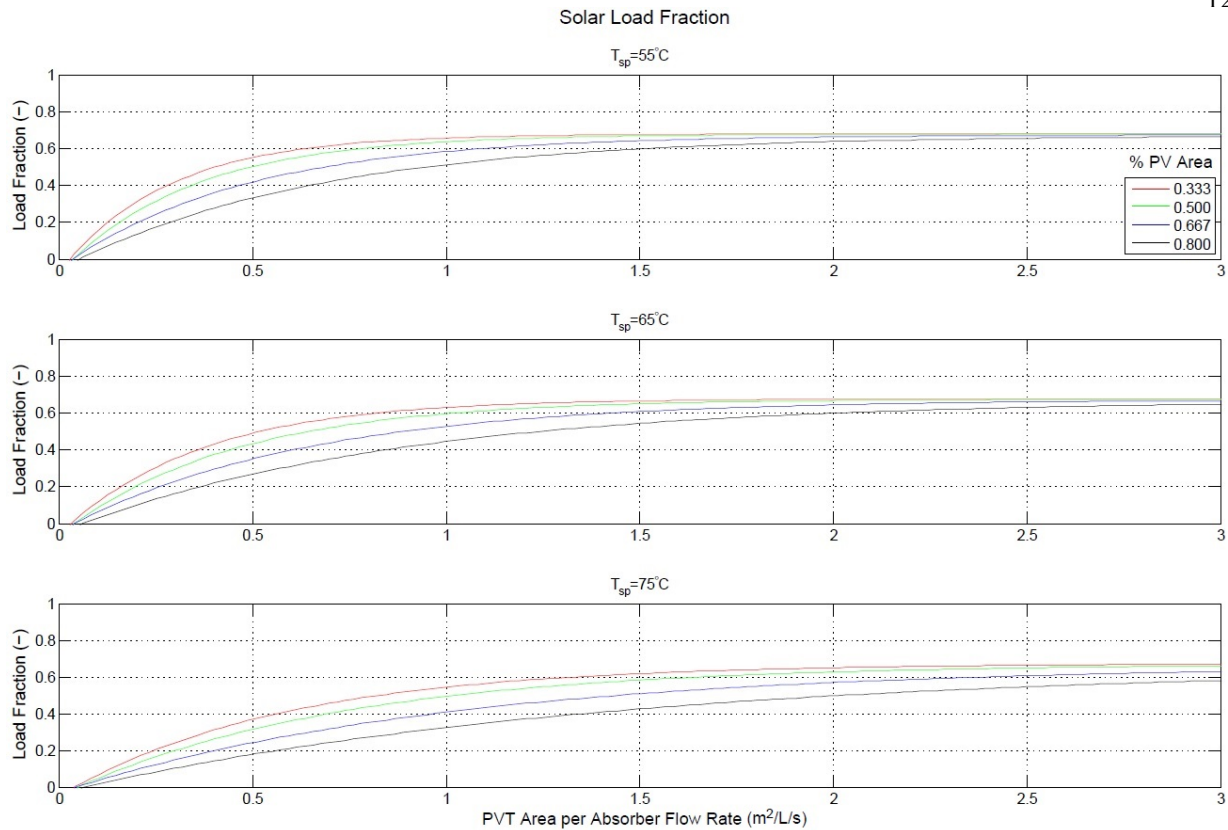


Figure 7.20: Solar Load Fraction - Guam

Figure 7.20 shows the solar load fractions for Guam at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Guam climate. The four different absorber CFM rates were simulated for a variety of PV/T array sizes and PV percentages. Figure B.2 displays the curves fitted to the simulation data.

In Guam, the smallest array sizes for any absorber rate can be found with a PV/T array set-point of 55°C , as in all of the other climates. At the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately $1.32 - 3.06 \text{ m}^2/(\text{L/s})$. For an 80% PV area system, a 275.4 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 915 m^2 PV/T array would be needed. For a 33% PV area system, a 118.8 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 396 m^2 PV/T array would be needed. Even at these absorber flow rates and PV/T array sizes, the arrays are too large for the example buildings and would only meet the maximum load fraction of 0.68. The humidity

set-point of $0.008 \text{ kg}_v/\text{kg}_a$ is much more difficult to reach for any sized system. However, using a smaller than ideal PV/T array size will still allow a partial load to be met, which would greatly reduce the load on the additional HVAC system.

Table 7.6: Absorbed Water Vapor - Guam

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	9830	1.12	1.48
117.99	24537	2.80	3.68
235.98	48890	5.58	7.34
471.95	93883	10.72	14.29

Table 7.6 contains the water vapor absorption rates for systems in Guam. All values refer to PV/T array sizes where the maximum load fraction is met. The absorption totals show a great deal of absorption occurring even though the maximum load fraction is not met. A PV/T-desiccant system would provide reasonable dehumidification that significantly reduces the latent load on an HVAC system.

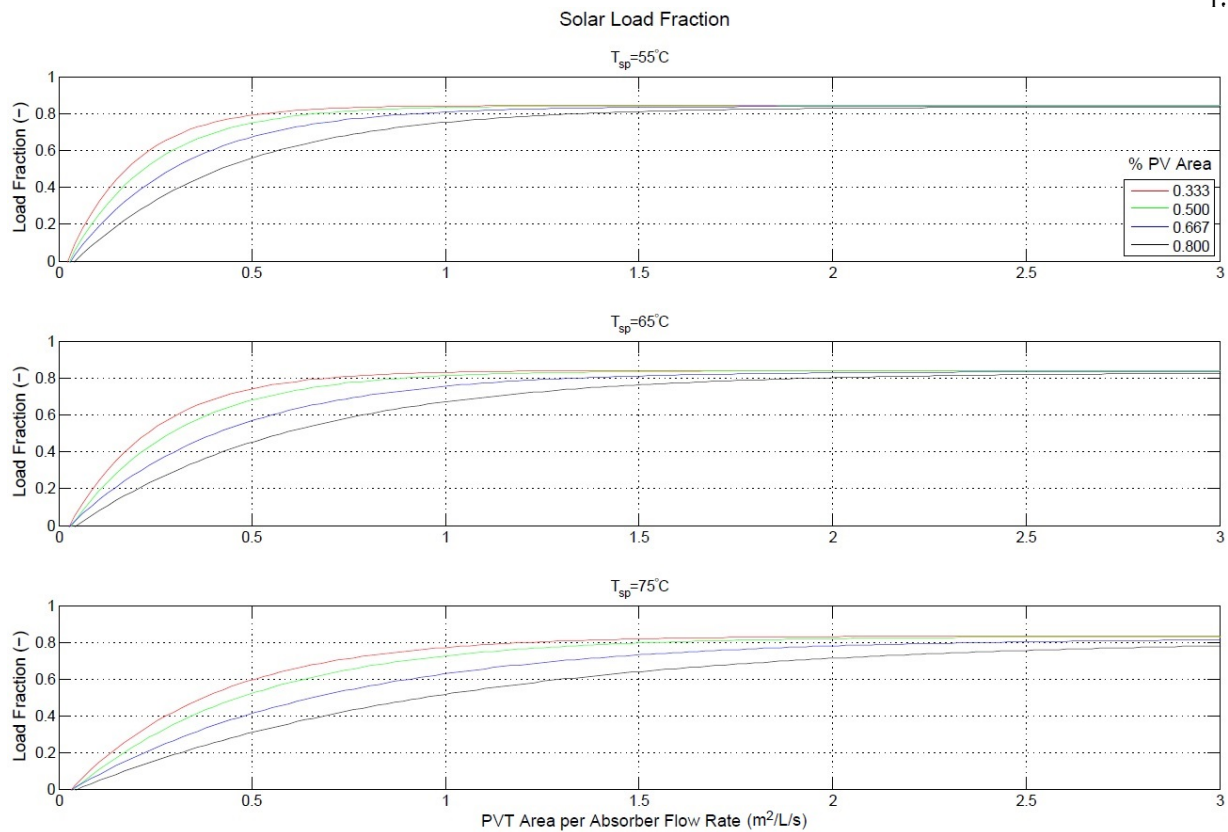


Figure 7.21: Solar Load Fraction - Honolulu

Figure 7.21 shows the solar load fractions for Honolulu at different PV/T array and absorber fan sizes. Each curve is fitted to data provided from the parametric simulations of the Honolulu climate. Figure B.3 displays the curves fitted to the simulation data.

As for all climates, the smallest array sizes for any absorber rate can be used with a PV/T array set-point of 55°C. As in Orlando, at the temperature set-point of 55°C and any PV area percentage, the maximum load fraction is reached between approximately 0.8 - 1.95 m^2/CFM . For an 80% PV area system, a 175.5 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 585 m^2 PV/T array would be needed. For a 33% PV area system, a 72 m^2 PV/T array would be used for a 300 m^2 building. For a 1000 m^2 building, an 240 m^2 PV/T array would be needed. Array sizing will be the similar for Honolulu as Orlando and Houston, but the overall maximum load fraction reached is larger than those other two climates. This is due to the Honolulu's overall lower humidity and the ability to reach the humidity ratio set-point for more of

the year.

Table 7.7: Absorbed Water Vapor - Honolulu

Absorber Flow Rate (<i>L/s</i>)	Total Absorbed (<i>kg</i>)	Average Absorbed (<i>kg/hr</i>)	Peak Absorption Rate (<i>kg/hr</i>)
47.20	7012	0.80	1.25
117.99	17505	1.99	3.10
235.98	34951	3.97	6.15
471.95	69538	7.80	12.20

Table 7.7 contains the water vapor absorption rates for systems in Honolulu. All values refer to PV/T array sizes where the maximum load fraction is met. Honolulu, according to the absorption totals, is nearly identical to Orlando. System sizing would be the same, and both would benefit from the high amounts of absorption that occur with a desiccant system.

7.4 Tank Sizing

The PV/T array, regenerator fan, and absorber flow rates were previously discussed and sized to maintain the maximum possible load fraction. These sizes were determined when using a large tank of greater than 50L of desiccant per L/s of absorber flow rate. For the baseline system of 104 m^2 , a tank with a capacity of 10,000 Liters was simulated. However, the tank capacity can potentially limit the amount of water vapor that is absorbed or regenerated, which will reduce the load fraction for the system. An acceptable tank capacity does not significantly reduce the load fraction met by the system. Various tank sizes were simulated to analyze how tank size affects the load fraction.

This section discusses tank sizing for various PV/T areas in all climates. The capacities are displayed in $L/(L/s)$.

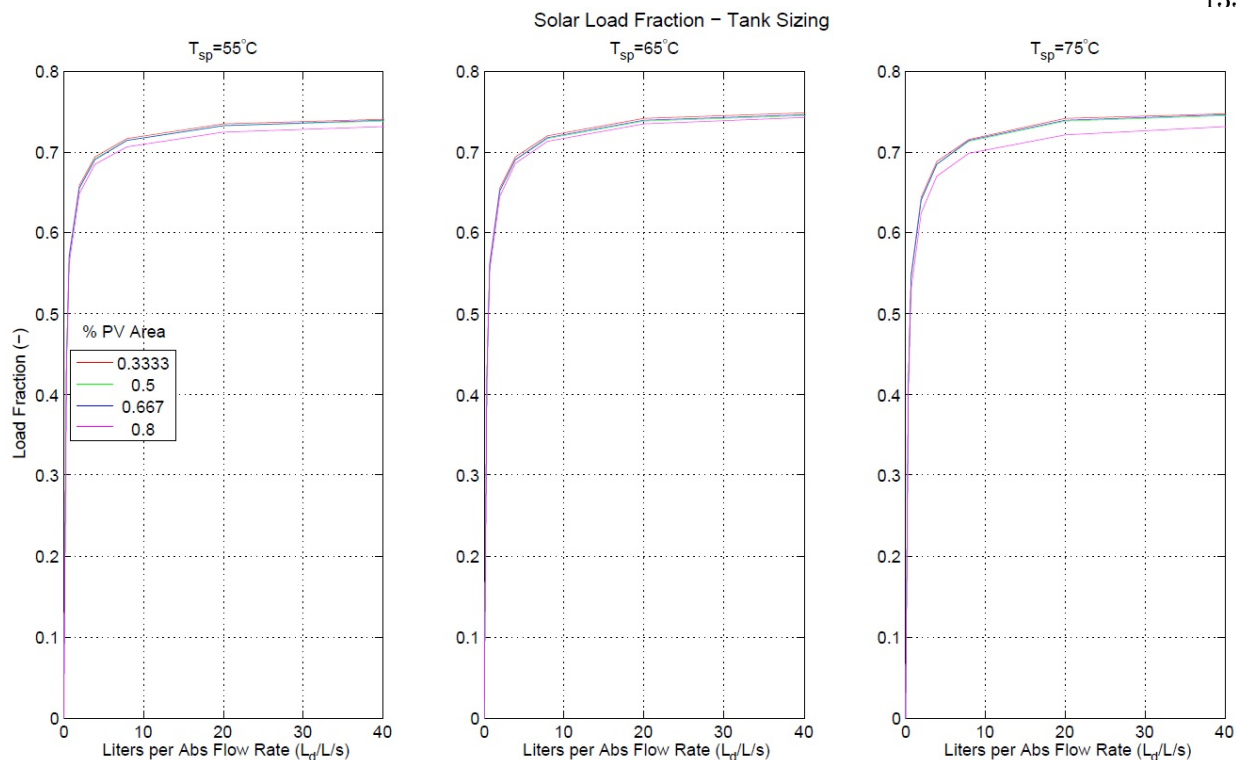


Figure 7.22: Solar Load Fraction for Tank Sizing - Orlando

The peak regeneration rate, how much water vapor is regenerated per hour, is approximately 26 kg/hr , for a 47.2 L/s absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 30 times the peak absorption rate. If using the regeneration rate to choose a tank size, a 2362L tank would be necessary for one day of desiccant storage. By adding more desiccant storage, the system can operate for longer periods of time when either the absorber or regenerator are not operating.

Simulations of various tank sizes are also used to determine more appropriate sizing. Figure 7.22 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section.

For all temperature set-points, a majority of the load fraction is reached when using a tank capacity of approximately $20\text{ L}/(\text{L}/\text{s})$, and no change in load fraction is seen at or above $40\text{ L}/(\text{L}/\text{s})$. For any 300m^2 building with a absorber flow rate of $90\text{ L}/\text{s}$, a tank size of 1800L (475 gallons) is sufficient. For 1000m^2 building with a absorber flow rate of $300\text{ L}/\text{s}$, a tank size of 6000L (1585

gallons) is needed. Larger tanks may allow additional regeneration/absorption during peak times, but 20 $L/(L/s)$ provides enough capacity to meet a majority of the load throughout the year.

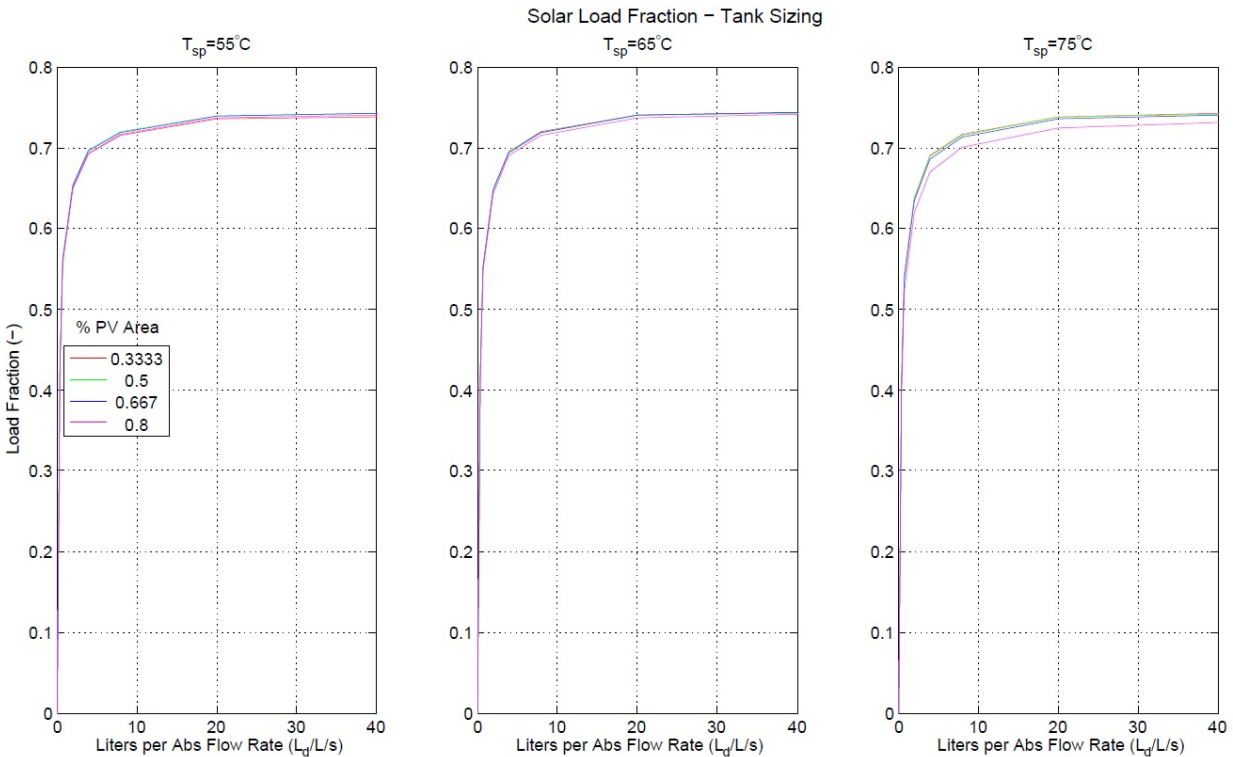


Figure 7.23: Solar Load Fraction for Tank Sizing - Houston

The peak regeneration rate in Houston is approximately 29 kg/hr , for a 47.2 L/s absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 40 times the peak absorption rate. The peak absorption and regeneration rates are similar to Orlando, requiring over 2650 L of capacity at worst-case conditions, for only one day of storage. However, the conditions are again worst-case, so simulations are used to determine appropriate sizing.

Figure 7.23 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section.

As in Orlando, for all temperature set-points, a majority of the load fraction is reached when using a tank capacity of approximately 20 $L/(L/s)$. No change in load fraction is seen at or above this capacity.

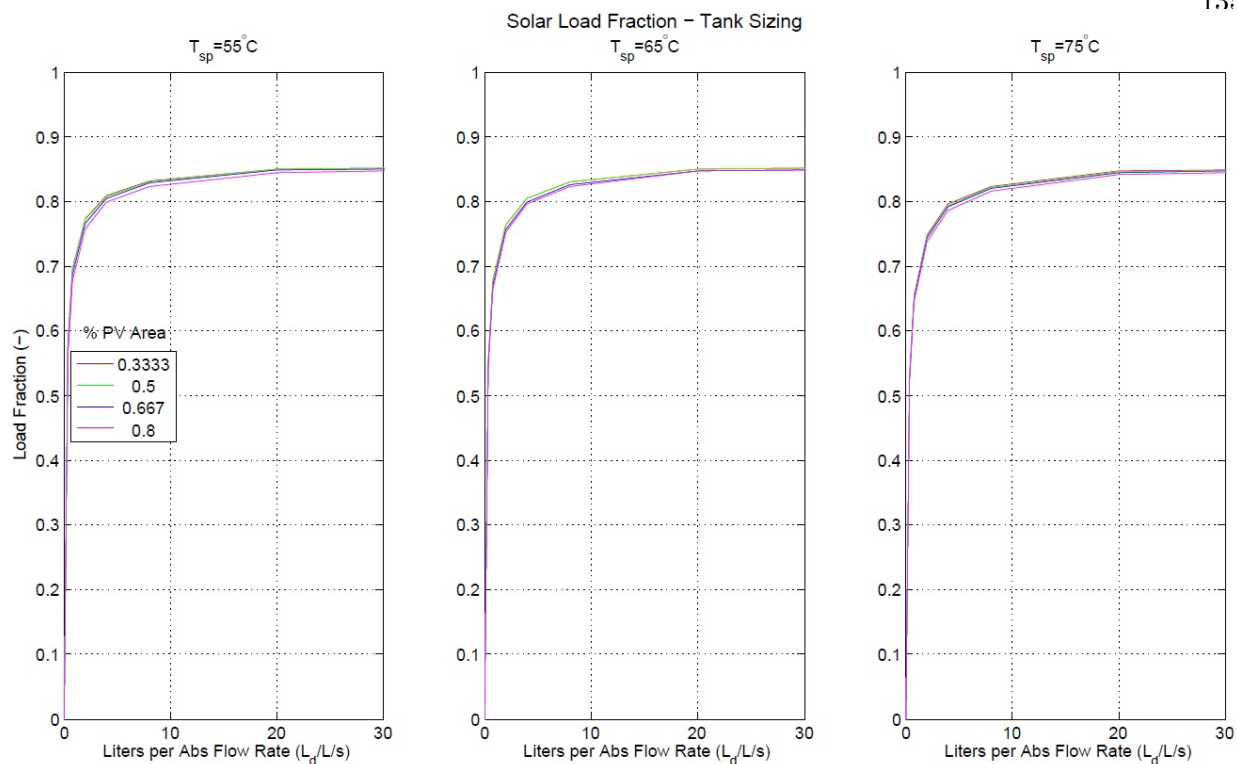


Figure 7.24: Solar Load Fraction for Tank Sizing - Atlanta

The peak regeneration rate in Atlanta is approximately 12 kg/hr , for a 47.2 L/s absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 30 times the peak absorption rate. For Atlanta, this only requires 1090 gallon tank at worst-case conditions, for one day of storage.

Other factors also affect tank sizing, and are seen in simulations. Figure 7.24 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section.

Maximum load fraction is reached when using a tank capacity of approximately $20 \text{ L}/(\text{L/s})$. Tank capacities for Atlanta are the same as in Orlando and Houston.

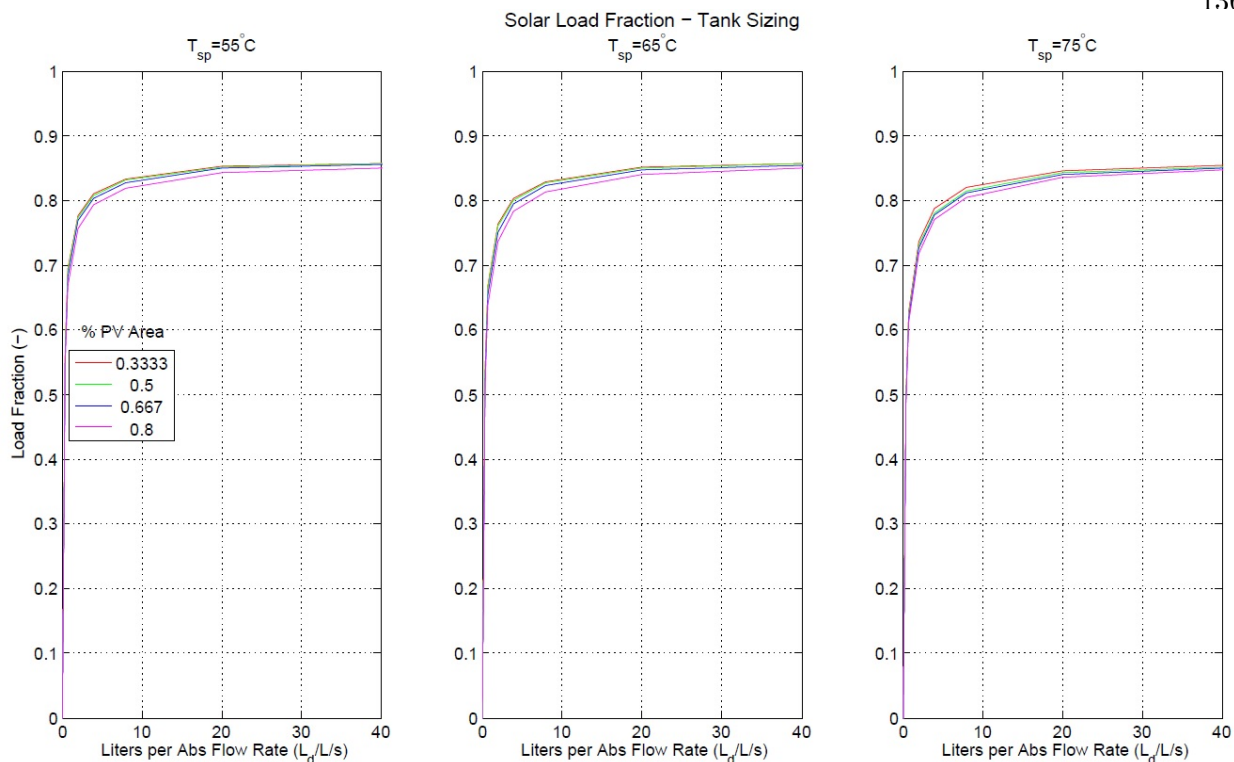


Figure 7.25: Solar Load Fraction for Tank Sizing - Washington DC

The peak regeneration rate in Atlanta is approximately 4.5 kg/hr , for a 47.2 CFM absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 14 times the peak absorption rate. For Washington DC, this only requires a 408L tank at worst-case conditions, for one day of storage.

Figure 7.25 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section. Maximum load fraction is again reached when using a tank capacity of approximately $20 \text{ L}/(L/s)$, which is the same for the other climates discussed so far.

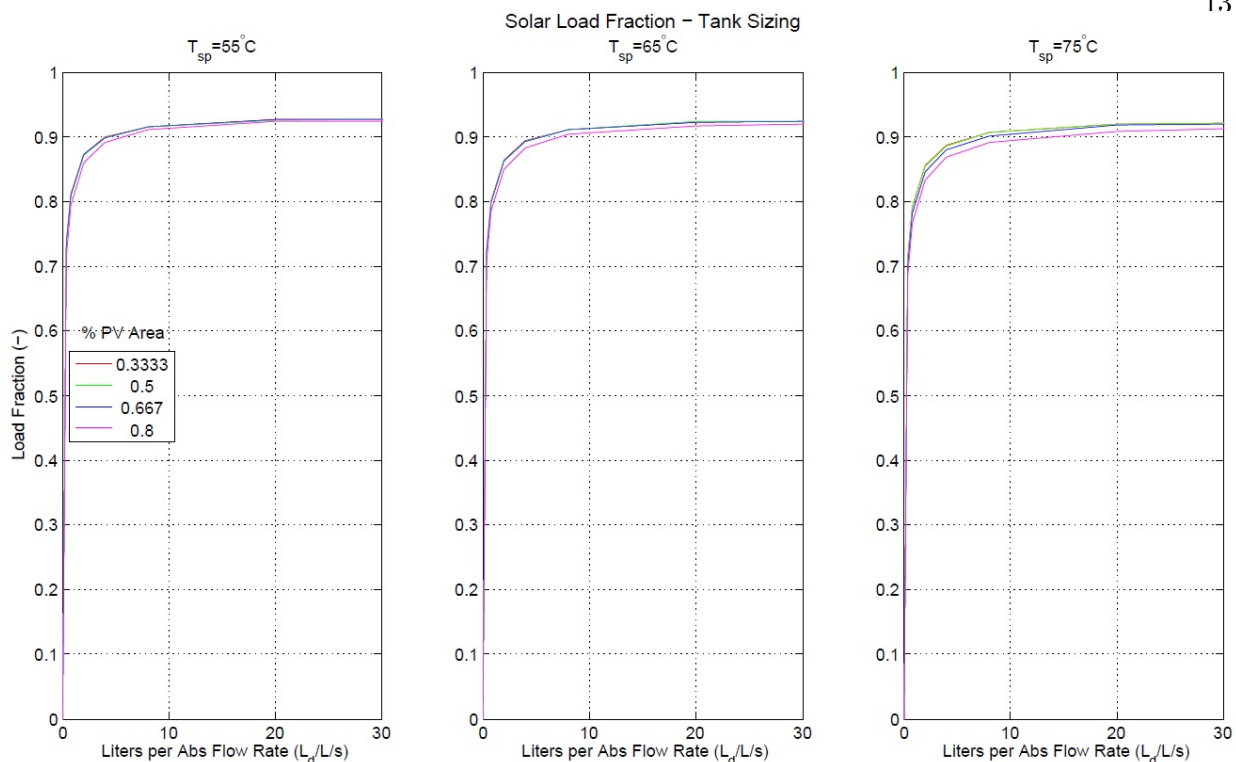
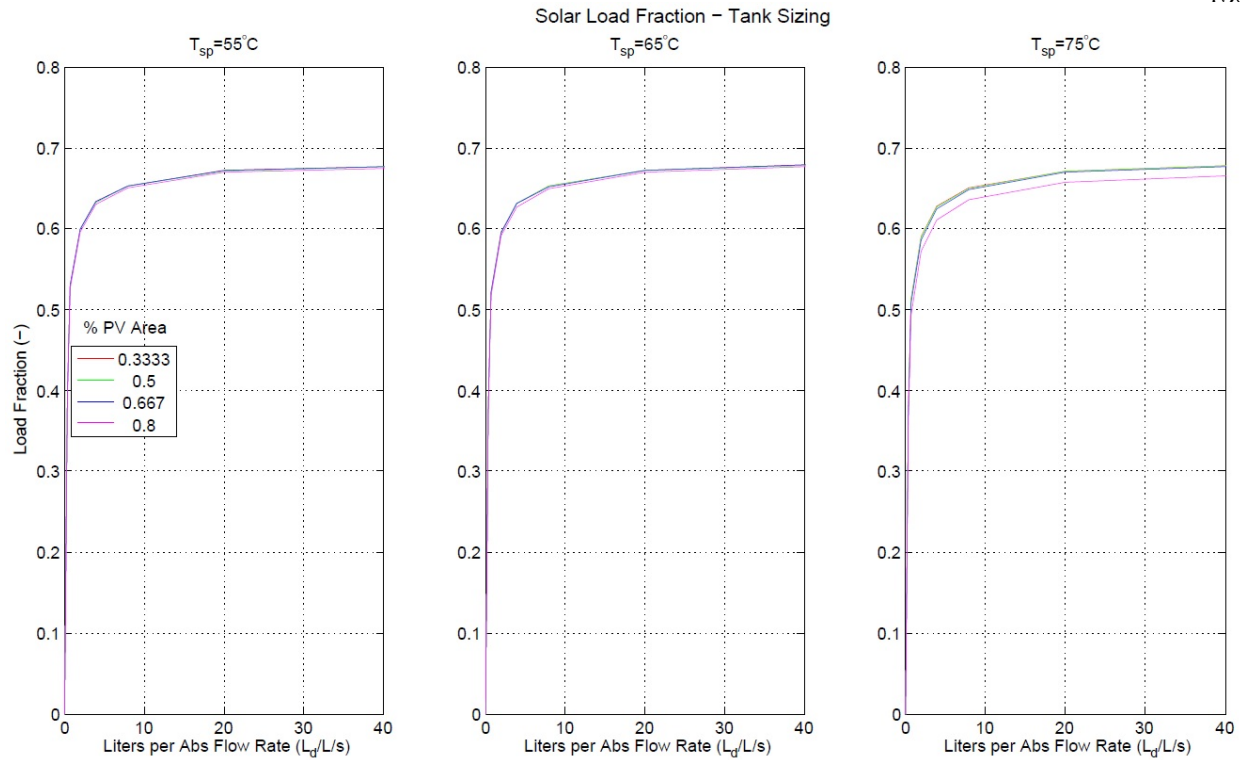


Figure 7.26: Solar Load Fraction for Tank Sizing - Phoenix

The peak regeneration rate in Phoenix is approximately 8 kg/hr , for a 47.2 CFM absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 61 times the peak absorption rate. In Phoenix, this only requires a 727 gallon tank at worst-case conditions, for one day of storage. The peak regeneration rate is much greater than that of the absorption rate, which may require additional storage to account for tank level variations. Simulations are used to analyze these conditions.

Figure 7.26 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section.

For all temperature set-points, the load fraction is reached when using a tank capacity of approximately $20 \text{ L}/(\text{L}/\text{s})$. As in the other climates, 1800L and 6000L are needed for 90 and 300 L/s flow rates, respectively.



The peak regeneration rate in Guam is approximately 24.5 kg/hr , for a 47.2 CFM absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 20 times the peak absorption rate. Guam would require a 2226L tank at worst-case conditions, for one day of storage. However, the peak regeneration rate and absorption rate are much closer to running at a similar rate, so extra storage is less likely to be required. Additionally, absorption is consistently required and occurs for the entire year. Regeneration is also able to occur consistently. Therefore, a large tank size is likely not necessary, except to account for the large amounts of absorbed vapor.

Figure 7.27 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section. As in all other climates, the load fraction is reached when using a tank capacity of approximately $20 \text{ L}/(\text{L}/\text{s})$.

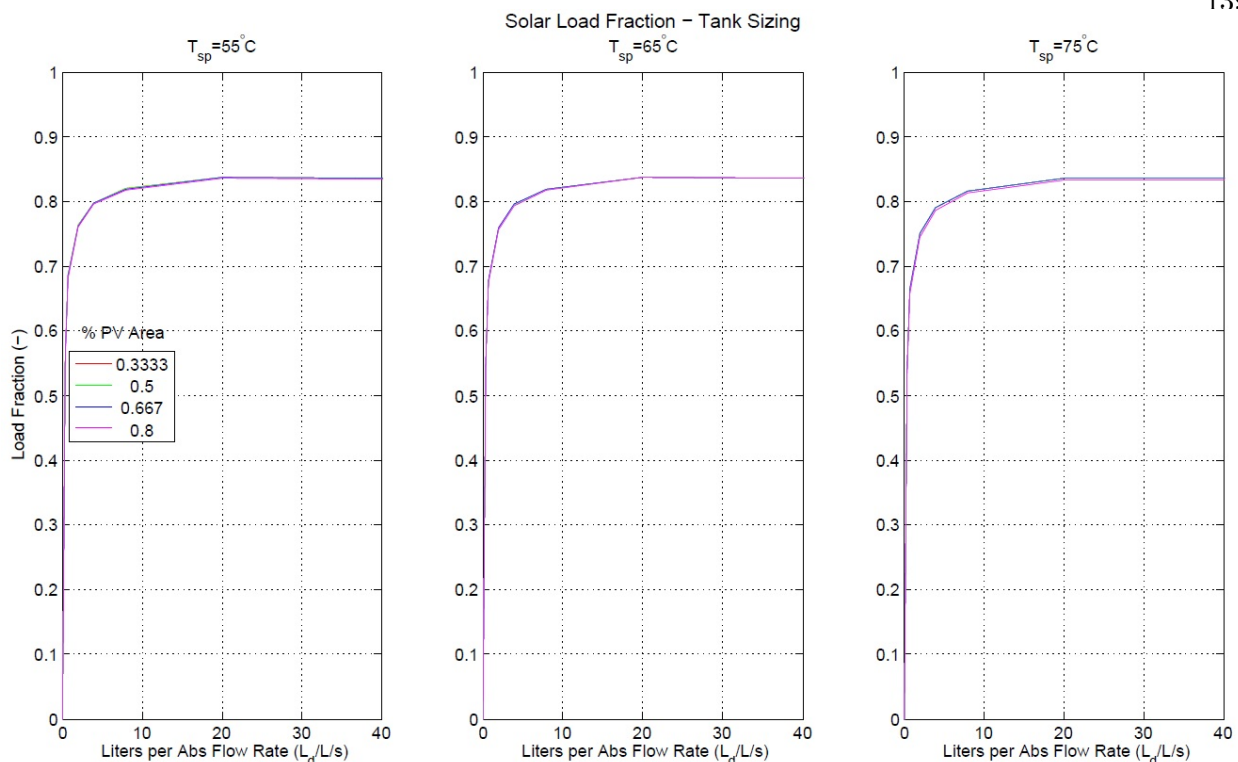


Figure 7.28: Solar Load Fraction for Tank Sizing - Honolulu

The peak regeneration rate in Honolulu is approximately 25 kg/hr , for a 47.2 CFM absorber and a PV/T array sized to meet the maximum load fraction. The rate is approximately 31 times the peak absorption rate. Honolulu would require a 2271L tank at worst-case conditions, for one day of storage. Regeneration and absorption are fairly consistent in Honolulu, a large tank size is likely not necessary.

Figure 7.28 displays how tank sizing affects load fraction for the PV/T array configuration and absorber flow rates discussed in the absorber sizing section. Again, the required tank size matches that of all the other climates.

7.5 System Energy Discussion

Energy used by all the pumps and fans for all system configurations analyzed was calculated and displayed in figures 7.29-7.35. Using the previously discussed absorber sizes for each climate, the total energy use for each climate can be found from the figures. The electrical energy use is

described in $kWh/(L/s)$, which is the total kWh of energy use over the course of a year for a given absorber flow rate. For example, a system with a 100 m^2 PV/T array with a $100L/s$ absorber flow rate may use approximately $14kWh/(L/s)$ of energy, which is $1400kWh$ of system energy used throughout the year. These scales are used so that the same values from the absorber sizing section can be used for comparison. Using the examples and values found in the absorber sizing section, PV/T-desiccant system energy use is compared to the energy use estimates for standard air-conditioning mentioned in the “Load Calculations” section, 7.1. For more accurate comparisons, both systems are assumed to be used for dehumidification only, with no additional cooling. Also, the original energy estimate for removing the latent load with standard air-conditioning will be reduced to match the same latent load met with the PV/T-desiccant system. A system meeting the full latent load should not be compared to a system meeting only $3/4$ of the load. As an additional assumption and to determine electrical energy use of the air-conditioning system, a COP of 3.0 is used. ASHRAE minimum COP requirements vary across system sizes and climate zones. Since a range of system sizes and climates are analyzed, a median COP value is chosen.

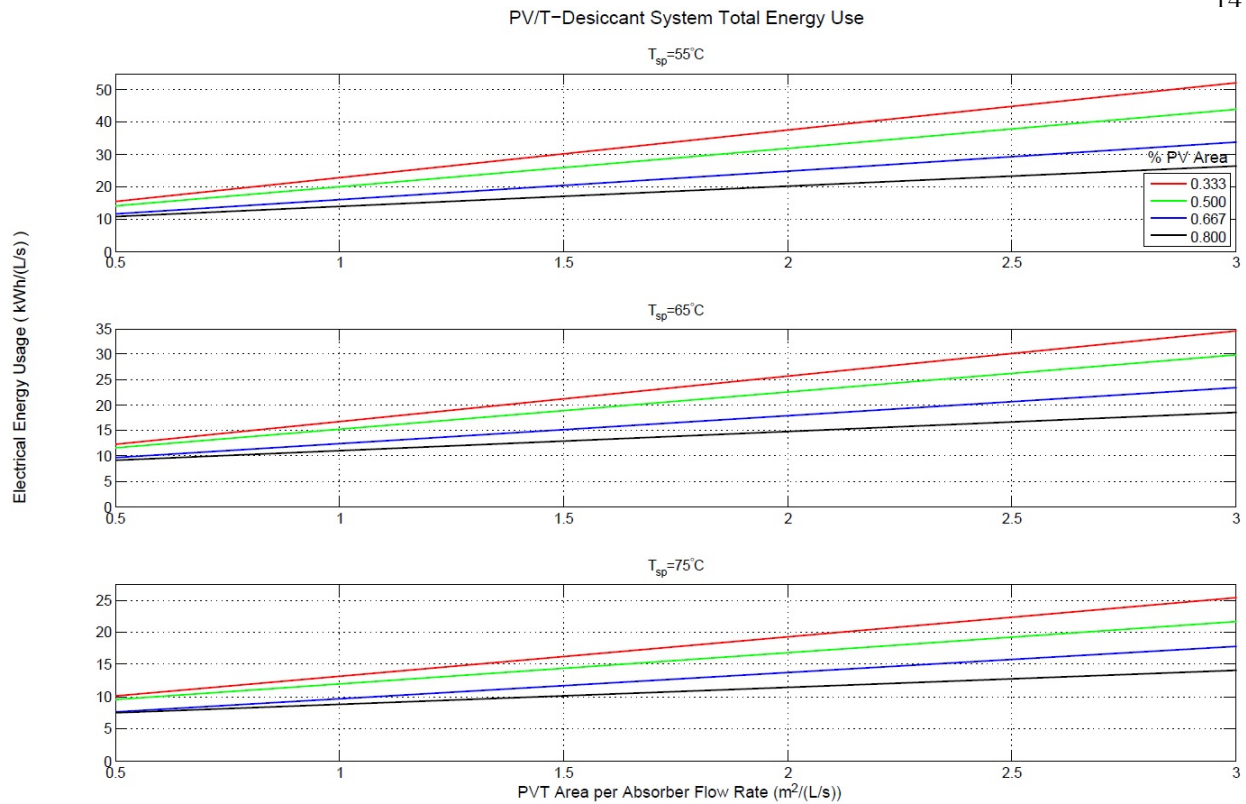


Figure 7.29: Electrical Energy Use - Orlando

Using the smallest system to meet the ventilation requirements, Orlando required a 33% PV system. The example building sizes were 300m^2 and 1000m^2 with 90 and 300L/s flow rates. In Orlando, the absorber was sized at $1.00\text{ m}^2/(\text{L/s})$. According to Figure 7.29, approximately $23\text{ kWh}/(\text{L/s})$ of energy is used, corresponding to 2070kWh for the 300m^2 building, and 6900kWh for a 1000m^2 building. System sizes ranged from $1.00 - 2.21\text{m}^2/(\text{L/s})$ at full load, and $0.25 - 0.6\text{m}^2/(\text{L/s})$ at 50% load.

The estimated energy required for the latent load of a standard air-conditioning was estimated at $145.25\text{ kWh}/(\text{L/s})$. Assuming a COP of 3.0, the same maximum load fraction reached with the absorber (0.77), and the same ventilation rates (90 and 300L/s), the estimated electrical energy required for removing the latent load using standard air-conditioning is 3355kWh for 90L/s and 11184kWh for 300L/s . According to these estimates, there is a 39.1% savings in electrical energy usage when using the 33% PV/T-desiccant system at full load instead of a standard air-conditioning

for dehumidification or removing the latent load. Savings ranged from 39.1% - 42.4%, with more savings and less electrical energy used as PV percentage increases from 33% to 80%.

By reducing the load on the PV/T-desiccant system, the sizes of the PV/T arrays are drastically decreased along with energy consumption. When reducing the load, it is assumed that more of the latent load will be handled by a standard air-conditioner instead of the PV/T-desiccant system. At reduced load, energy savings decrease and range from 33.6% - 34.8%. Again, higher PV percentage has more savings.

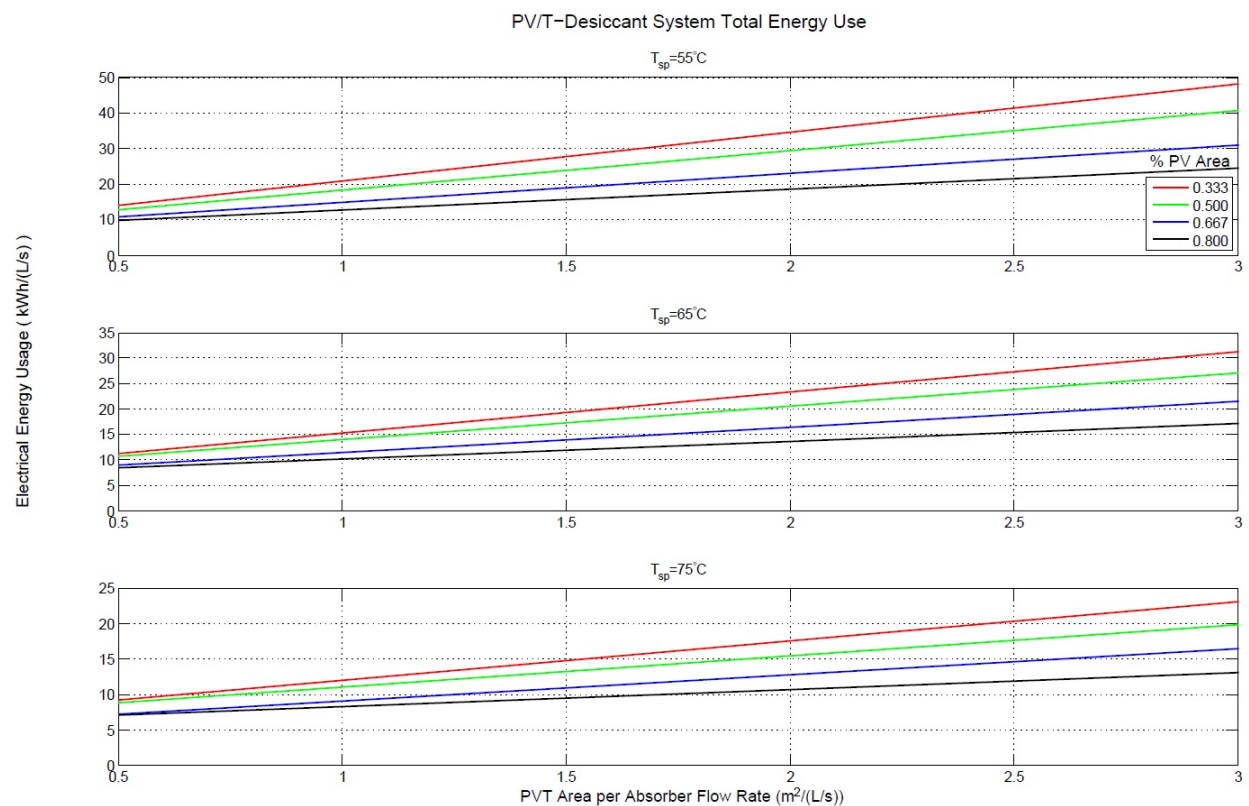


Figure 7.30: Electrical Energy Use - Houston

Houston requires similar sizing to Orlando. In Houston, the absorber sizes ranged from 0.86 - 2.23 $m^2/(L/s)$. Higher PV percentage arrays always require larger sizes. According to Figure 7.30, approximately 20 - 19.5kWh/(L/s) of energy is used, depending on PV percentage. As before, more PV decreases electrical usage and increases energy savings.

The estimated energy required for the latent load of a standard air-conditioning was estimated at $128.3 \text{ kWh}/(L/s)$, with a maximum load fraction of 0.78. The estimated electrical energy required for removing the latent load using standard air-conditioning is 3002 kWh for $90 L/s$ and 10007 kWh for $300 L/s$. These estimates show a 40 - 41.5% savings in electrical energy usage for full load systems. Reduced loading has savings ranging from 32.6 - 33.8%, for PV/T array sizing of $0.225 - 0.55 \text{ m}^2/(L/s)$. The values for array sizing and energy savings are almost identical to Orlando.

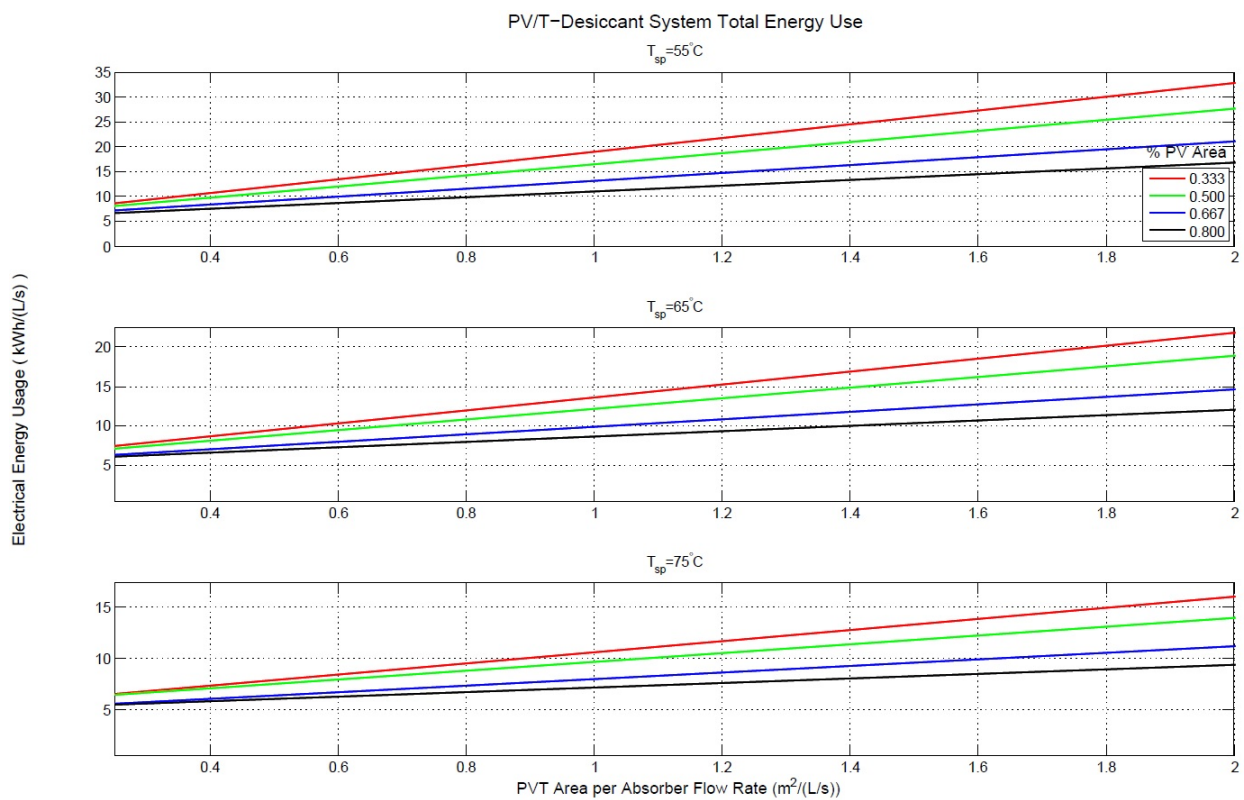


Figure 7.31: Electrical Energy Use - Atlanta

Atlanta requires much less dehumidification than Houston or Orlando. In Atlanta, the absorber sizes ranged from $0.57 - 1.17 \text{ m}^2/(L/s)$. According to Figure 7.31, approximately $13.15 - 11.97 \text{ kWh}/(L/s)$ of energy is used, depending on PV percentage.

The estimated energy required for the latent load of a standard air-conditioning is 70.23

$kWh/(L/s)$, with a maximum load fraction of 0.87. The estimated electrical energy required for removing the latent load using standard air-conditioning is $1833kWh$ for $90L/s$ and $6110kWh$ for $300L/s$. These estimates show a 35.4 - 41.2% savings in electrical energy usage for full load systems. Reduced loading has savings ranging from 23.1 - 25.1%, for PV/T array sizing of $0.12 - 0.25m^2/(L/s)$.

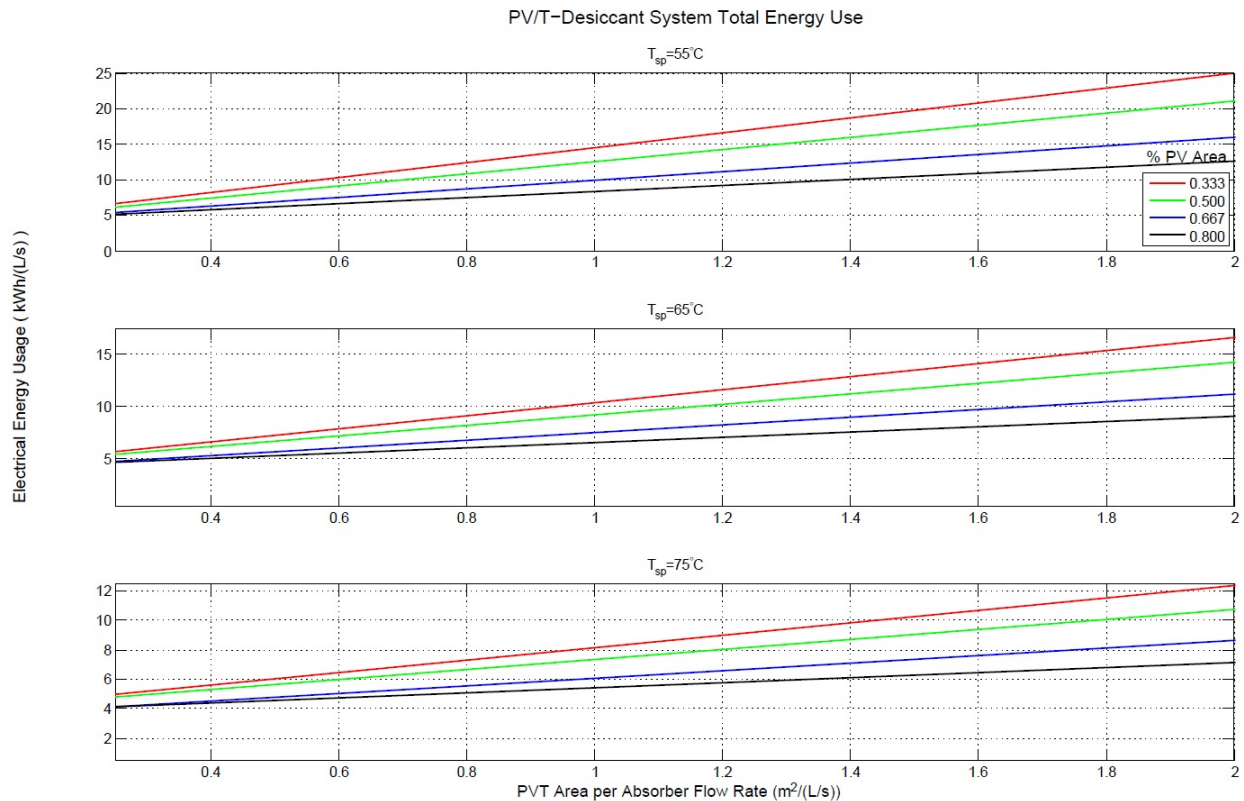


Figure 7.32: Electrical Energy Use - Washington DC

Washington DC requires similar systems to those in Atlanta. In DC, the absorber sizes ranged from $0.64 - 1.28 m^2/(L/s)$. According to Figure 7.32, approximately $10.75 - 9.5kWh/(L/s)$ of energy is used, depending on PV percentage.

The estimated energy required for the latent load of a standard air-conditioning is $49.58 kWh/(L/s)$, with a maximum load fraction of 0.89. The estimated electrical energy required for removing the latent load using standard air-conditioning is $1324kWh$ for $90L/s$ and $4413kWh$

for $300L/s$. These estimates show a 26.9 - 35.2% savings in electrical energy usage for full load systems. Reduced loading has savings ranging from 19.8 - 21.5%, for PV/T array sizing of $0.125 - 0.25m^2/(L/s)$.

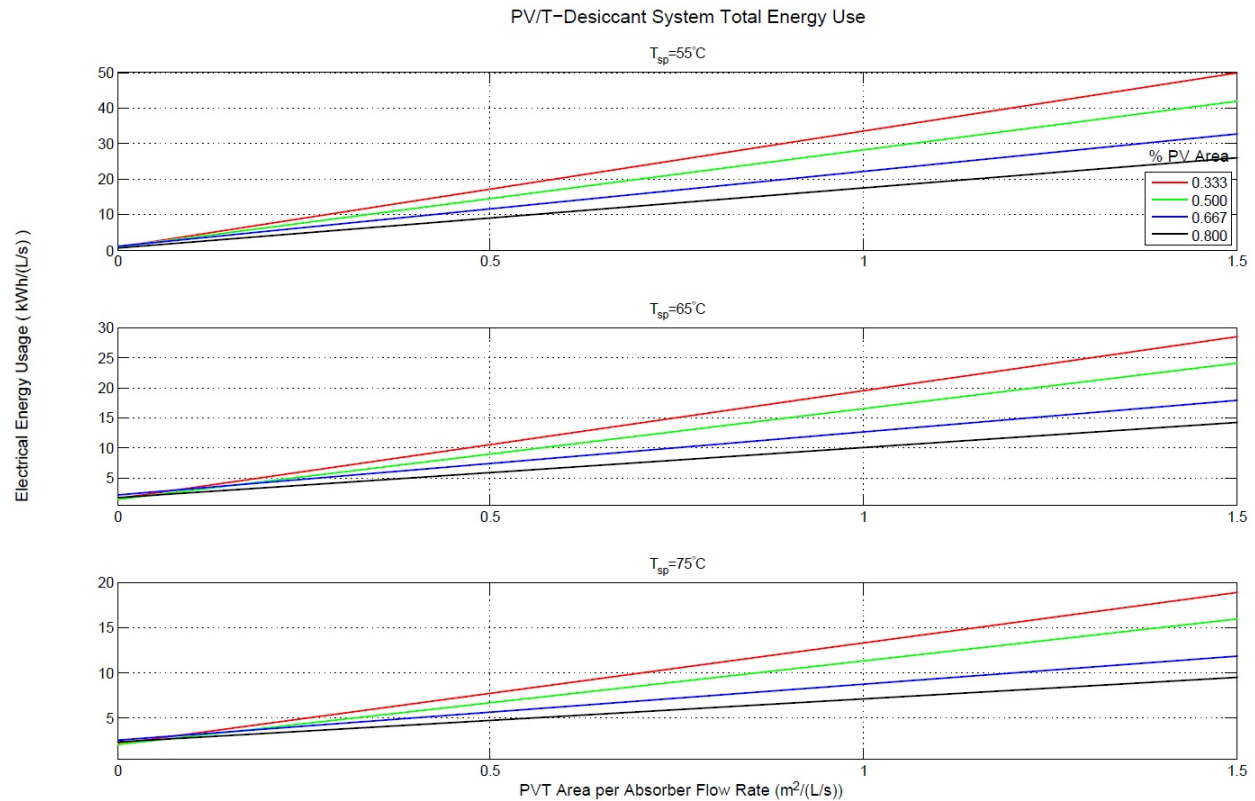


Figure 7.33: Electrical Energy Use - Phoenix

Phoenix requires almost no dehumidification and is the only dry climate analyzed. In Phoenix, the absorber sizes ranged from $0.32 - 0.78 m^2/(L/s)$. According to Figure 7.33, approximately $11.3 - 13.5 kWh/(L/s)$ of energy is used, depending on PV percentage.

The estimated energy required for the latent load of a standard air-conditioning is $18.2 kWh/(L/s)$, with a maximum load fraction of 0.94. The estimated electrical energy required for removing the latent load using standard air-conditioning is $513.24 kWh$ for $90L/s$ and $1710 kWh$ for $300L/s$. These estimates show savings when using standard air-conditioning and not using the desiccant dehumidification. The components of the PV/T-desiccant system using the most power

are the fans. In Phoenix, much larger fans need to be used for a large portion of the year to maintain particular set-point temperatures. This contributes to the high electrical energy use of the Phoenix system. There is a 49.4 - 57.8% savings in electrical usage for full load systems when not using the desiccant system. Reduced loading has savings ranging from 25 - 32.1%, for PV/T array sizing of $0.02 - 0.04m^2/(L/s)$. Phoenix is the only climate that saves more energy without using a desiccant system.

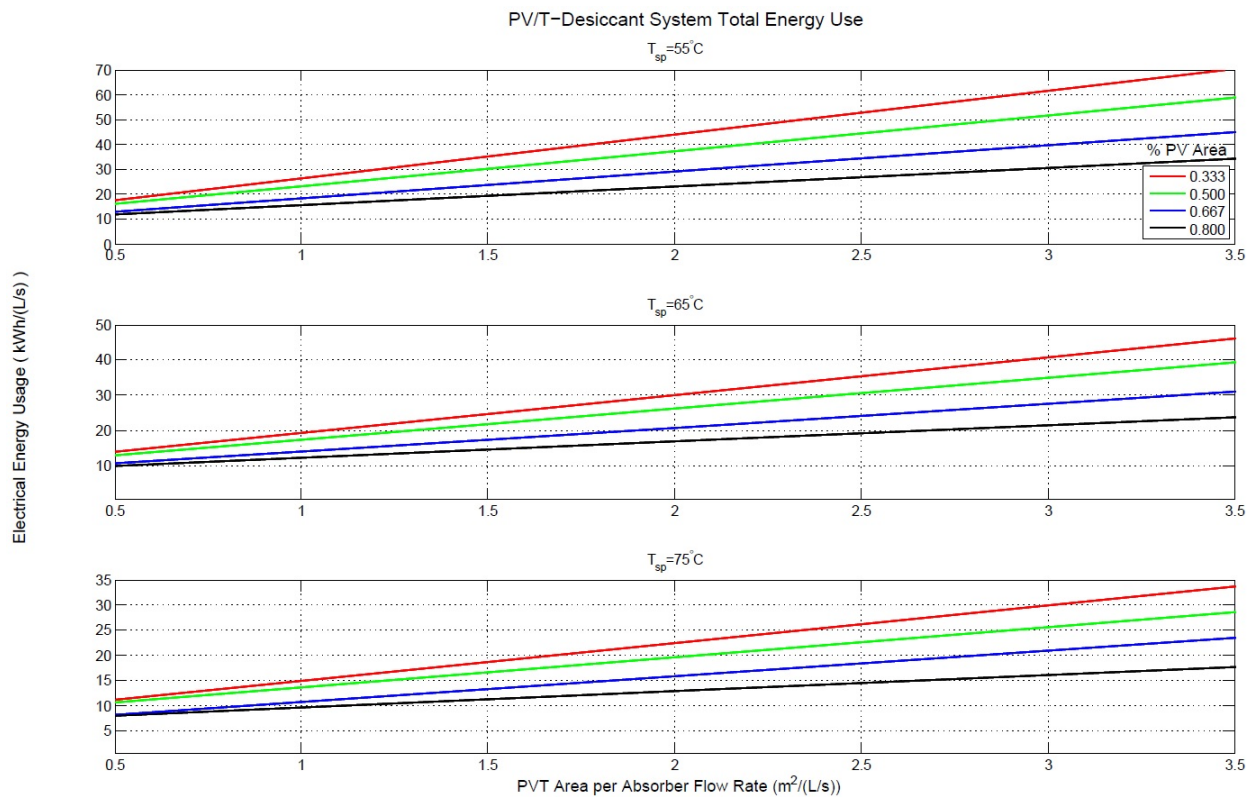


Figure 7.34: Electrical Energy Use - Guam

Guam requires the most dehumidification and shows the most electrical energy savings due to the high loads not dehumidified by standard air-conditioning. In Guam, the absorber sizes ranged from $1.33 - 3.06 m^2/(L/s)$. According to Figure 7.34, approximately $32.2 - 31kWh/(L/s)$ of energy is used, depending on PV percentage.

The estimated energy required for the latent load of a standard air-conditioning is 271.75

$kWh/(L/s)$, with a maximum load fraction of 0.68. The estimated electrical energy required for removing the latent load using standard air-conditioning is $5544kWh$ for $90L/s$ and $18479kWh$ for $300L/s$. These estimates show a 47.7 - 49.5% savings in electrical energy usage for full load systems. Reduced loading has savings ranging from 47.6 - 49.2%, for PV/T array sizing of $0.04 - 0.95m^2/(L/s)$.

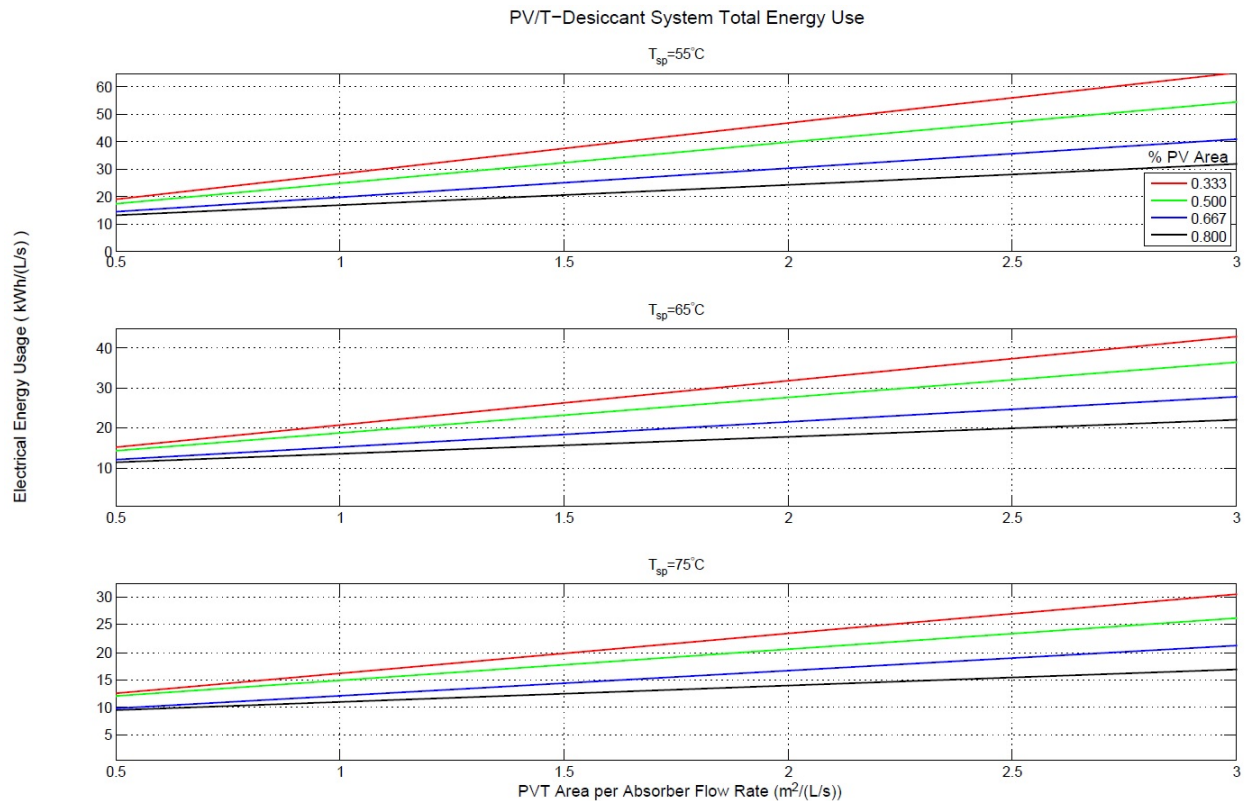


Figure 7.35: Electrical Energy Use - Honolulu

In Honolulu, absorber sizes ranged from $0.80 - 1.95 m^2/(L/s)$. According to Figure 7.35, approximately $24.6 - 23kWh/(L/s)$ of energy is used, depending on PV percentage.

The estimated energy required for the latent load of a standard air-conditioning is $140.04 kWh/(L/s)$, with a maximum load fraction of 0.84. The estimated electrical energy required for removing the latent load using standard air-conditioning is $3528kWh$ for $90L/s$ and $11760kWh$ for $300L/s$. These estimates show a 37.4 - 41.3% savings in electrical energy usage for full load

systems. Reduced loading has savings ranging from 27.4 - 28.2%, for PV/T array sizing of $0.15 - 0.4m^2/(L/s)$.

In all of the climates analyzed, except for Phoenix, energy savings are higher at higher loads and higher PV percentages. For any climate, lower loads are necessary to provide adequate roof space for the PV/T array. However, energy savings are reduced because standard air-conditioning must handle more of the latent load, which uses more electricity.

Fortunately, the PV panels produce enough electricity to power the pumps/fans plus more additional electricity. This additional electricity is shown in the figures below for each climate. Each figure displays the additional amount of electricity per month, to compare to general monthly electricity bills. The figures are an additional aid in sizing of the PV/T arrays for a given climate, since PV electricity production may be a factor in choosing a system size.

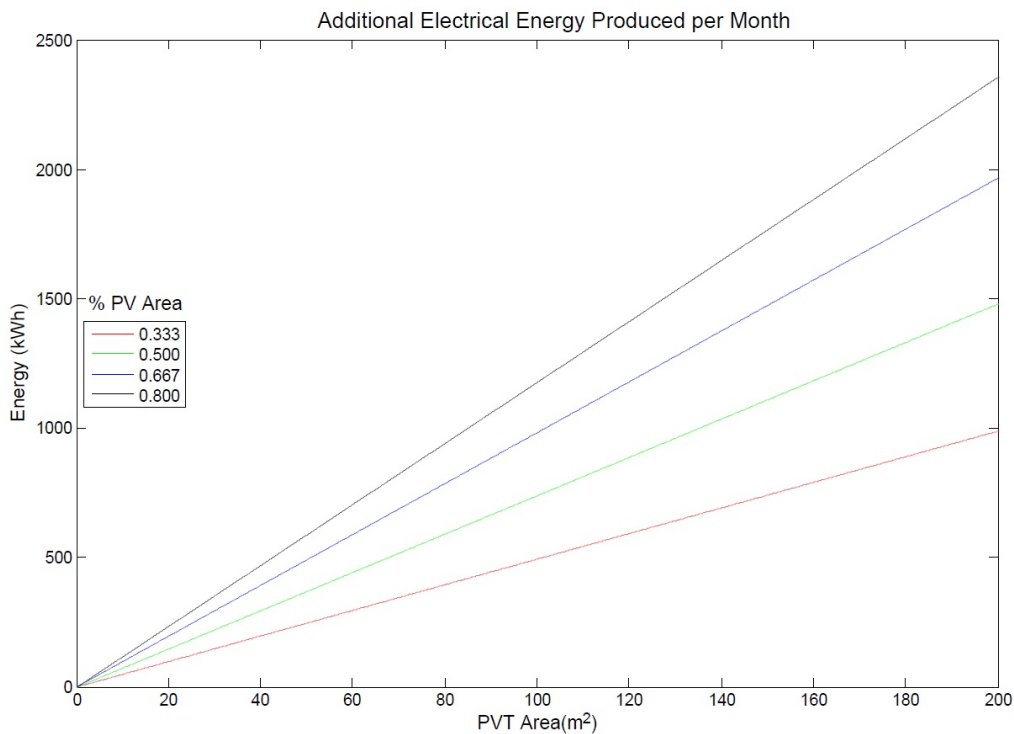


Figure 7.36: Electrical Energy Production - Orlando

Figure 7.36 shows the additional electrical energy produced by the PV panels for different

PV/T array sizes in Orlando. Energy values are additional energy produced per month (averaged), so pump/fan energy use is deducted from the total PV panel electricity production. Data shows that the 104 m^2 , 11.2 kW Orlando PV/T system produces approximately 15,000 kWh per year, an average of 1250 kWh per month. The figure shows nearly 1100 kWh hours of additional electricity can be produced for that system, which can obviously be used for providing electricity to the building it is installed on.

The pumps and fan use little electricity compared to how much PV electricity is produced, since this is a very large PV/T array. Also, the pumps are very small and use very little electricity. The regenerator fan uses a lot of power, but it only operates for part of the day and very rarely near the maximum speed. The absorber fan is much smaller than the regenerator fan, in all cases, and does not operate for approximately 20% of the year in Orlando. The same can be said for the cooling tower operation.

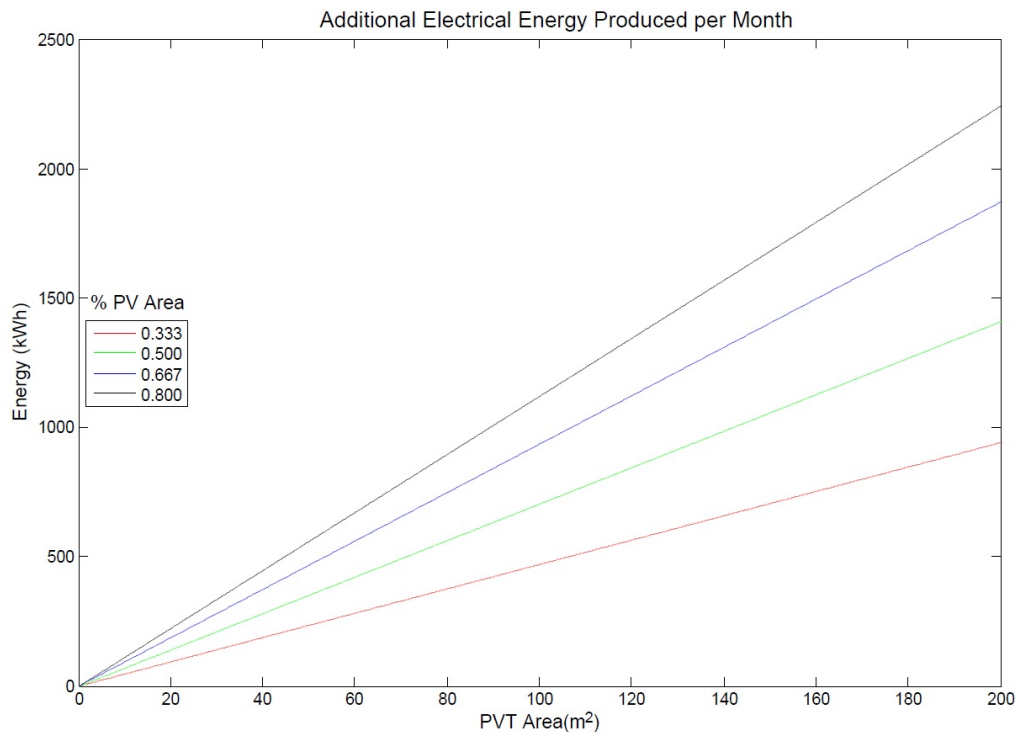


Figure 7.37: Electrical Energy Production - Houston

Figure 7.37 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Houston. As expected, less PV energy is produced, so additional energy is also less. An abundant amount of electricity is still produced for all array sizes.

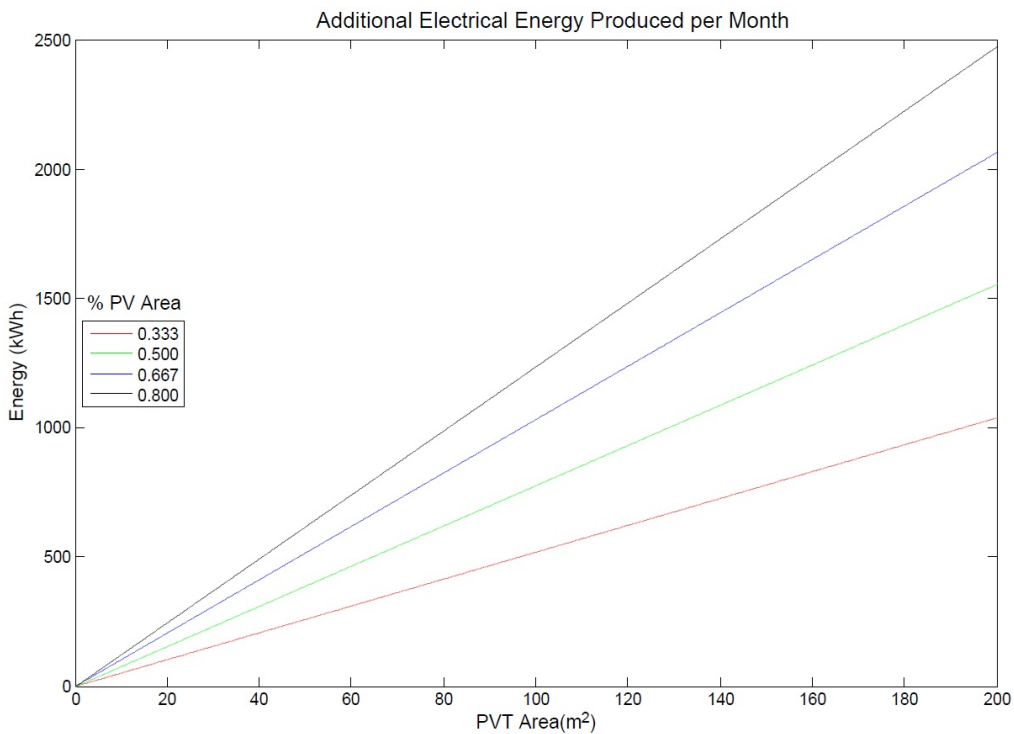


Figure 7.38: Electrical Energy Production - Atlanta

Figure 7.38 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Atlanta. More additional PV energy is produced in Atlanta than the other two climates, due to less power used by the absorber and plenty of sunlight produced throughout the year.

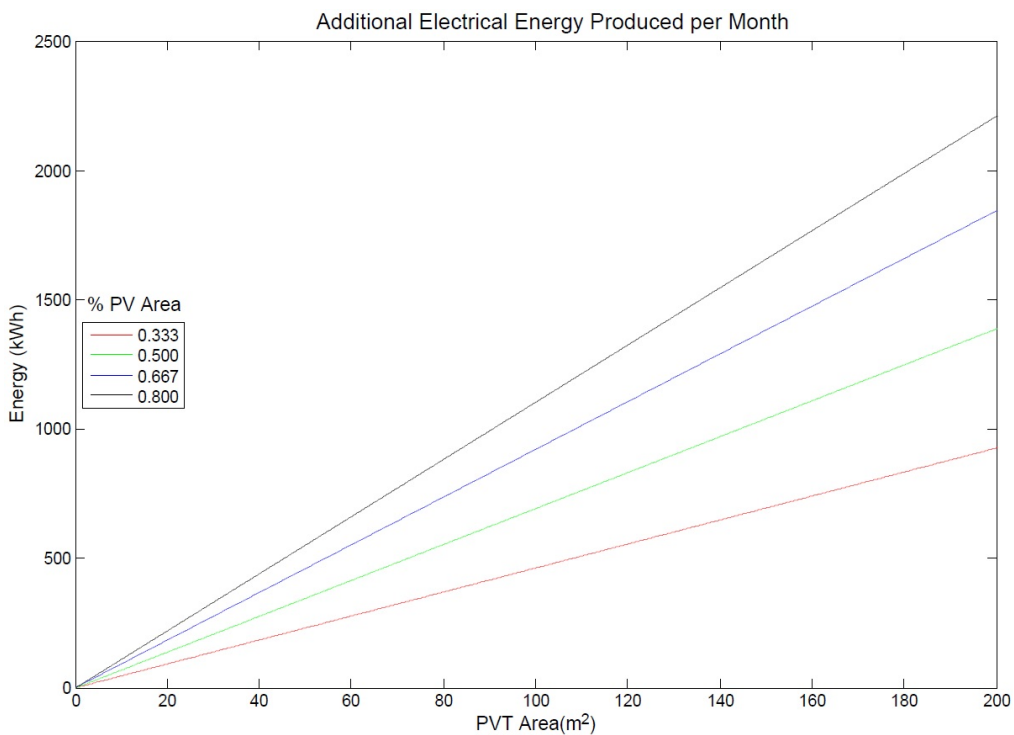


Figure 7.39: Electrical Energy Production - Washington DC

Figure 7.38 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Atlanta. More additional PV energy is produced in Atlanta than the other two climates, due to less power used by the absorber and plenty of sunlight produced throughout the year.

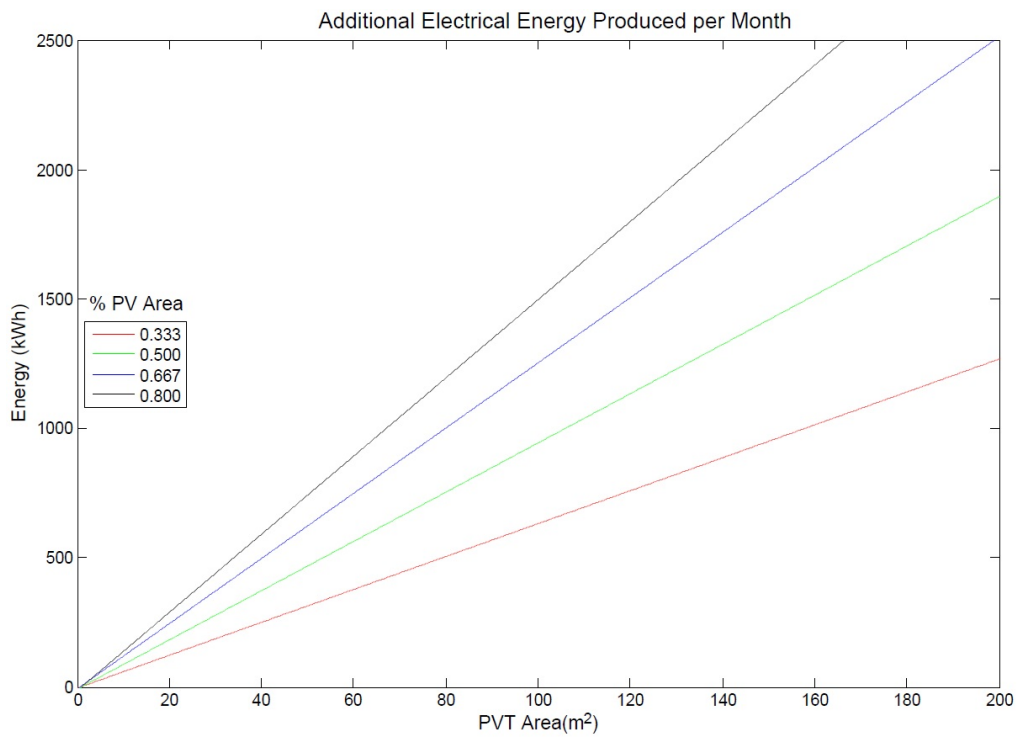


Figure 7.40: Electrical Energy Production - Phoenix

Figure 7.40 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Phoenix. Excellent solar insolation increases the total amount of PV electricity production for all sizes of arrays.

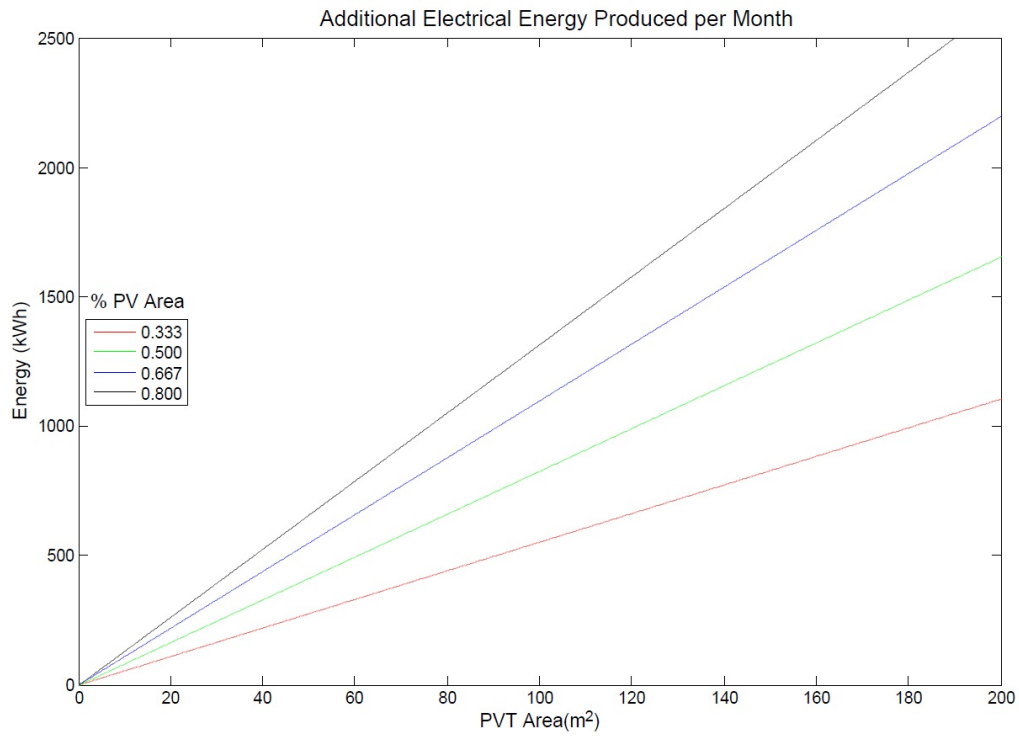


Figure 7.41: Electrical Energy Production - Guam

Figure 7.41 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Guam. PV production is much less than in Phoenix, but more absorption is required.

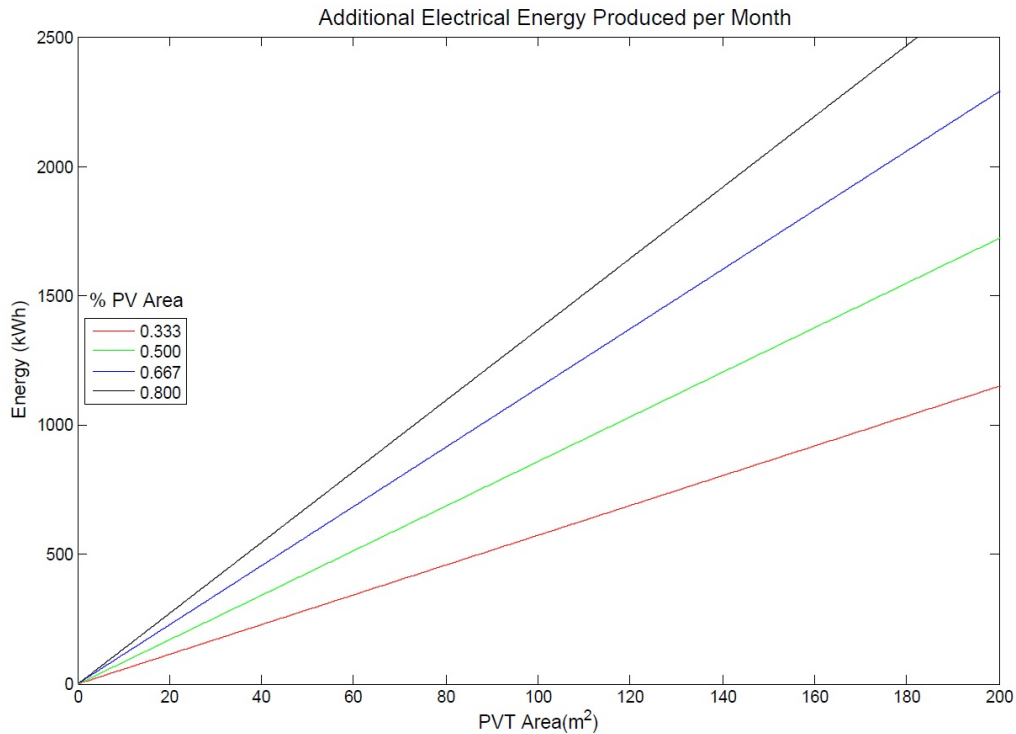


Figure 7.42: Electrical Energy Production - Honolulu

Figure 7.42 shows the additional electrical energy produced by the PV panels for different PV/T array sizes in Honolulu. Additional PV energy production is more than in Guam, since less absorption occurs and less absorber and cooling tower energy is required.

None of the figures in any of the climates vary significantly, because the regenerator-PV/T fans are scaled to the array size. However, climate differences cause the absorber and cooling tower equipment for different amounts of the year. The average values do not show PV production differences at peak times. The additional electricity production figures are simply another consideration in choosing PV/T array size and % PV area.

Chapter 8

Conclusions and Future Work

8.1 Conclusions

This thesis has examined the effects of component sizing on the performance of a liquid desiccant system regenerated using solar thermal energy. Performance is based on how much of the potential load can be met by a given system configuration. The total possible load is the amount of water vapor that the absorber needs to remove from the air to reach an outlet humidity ratio set-point, for a given absorber or ventilation flow rate. For all climates and systems, the absorber outlet humidity set-point is $0.008 \text{ kg}_v/\text{kg}_a$. The absorber does not operate when outdoor air humidity is below the set-point. For each climate tested, only a fraction of the calculated load can be met, regardless of system component sizes. The goal of choosing the sizes for the PV/T array, regenerator fan, absorber, and tank is to ensure that the maximum possible load fraction can be achieved. If any component is not appropriately sized, regeneration and/or absorption may stop under certain conditions, and less of the possible load will be reached.

While each component limits the capacity of each of the other components, the PV/T array size is a limiting factor in these systems. The PV/T array is, by far, the largest and most expensive component in the system, which must also be sized to fit on a building. The PV/T array sizes, with high PV percentage, needed to meet the minimum outdoor ventilation rate and maximum latent load may be too large for multi-story buildings. Single-story buildings with much larger roof areas are more appropriate for larger, higher PV percentage systems. Some lower PV percentage arrays, such as 33% PV, allow for sizes that fit on most roof sizes. As PV percentage increases, there is

less heat produced, lower airflow, and less regeneration that occurs. Higher PV percentage arrays simply cannot be sized small enough to fit on all roofs and meet the maximum load. Fortunately, array sizes can be greatly reduced when only meeting a partial load. At 50% load in any given climate, arrays with less than 80% PV are likely small enough for any roof. Near 80% PV or above, arrays are less likely to be acceptable. The maximum size solely depends on available roof space, and the array must be sized as large as possible for a given roof area.

Absorbers, regenerators, fans, and tanks can be sized to meet a load without the same physical size limitations. Since the PV/T array size is a major cost factor and size may be a concern in these systems, the other components are sized in relation to the PV/T array size. Tank size was found to only be affected by absorber/ventilation flow rate, and needs to be sized for 20 Liters for every L/s of ventilation required. Tank sizes are basically negligible compared to the PV/T array size.

The regenerator fan sizes must be chosen so that enough heat can be produced to regenerate a desiccant, and so an outlet set-point temperature can be maintained. A specific temperature must be maintained in order to not damage the regenerator, and also because the regenerator is more effective at certain temperatures. However, there are trade-offs in using different flow rates and temperatures. To maintain a lower set-point temperature, flow rate must increase to remove more heat from the back of the PV/T array. Higher set-point temperatures require less air flow. However, at increased air flow rates, a larger masses of air pass through the regenerator so larger amounts of water vapor are absorbed. Basically, lower temperature set-points create more airflow and more regeneration capacity. The data from the experimental Orlando system showed that, at a constant air flow rate, the effectiveness of the regenerator peaked at approximately 55 - 57°C. The optimal temperature set-point is 40°C, which is the lowest temperature that still allows for constant regeneration with any outdoor air conditions. A 40°C set-point, the regenerator is nearly as effective as temperatures up to 65°C. PV/T arrays with the lowest set-point and highest airflow will have the highest regeneration capacity. Unfortunately, there are limitations to fan size because of the maximum airflow through ducting associated with the PV/T array. Another down-side of using

higher air flows is that larger fans are required and more power is used. The lowest temperature set-point used in the simulations was 55°C so that the set-point could always be maintained without excessively large fan sizes. Also, the regenerator effectiveness peaks around 55°C.

Regenerator fans are sized to meet the specific set-point temperatures mentioned above at worst-case conditions. Since each climate receives varying degrees of solar insolation and has different outdoor air temperatures, the fans are sized separately for each climate. For larger PV/T arrays, such as at 100 m^2 , fan sizes range from 450-1500L/s, regardless of climate or percentage that is PV area, with the exception of Phoenix. Reducing PV area and increasing the amount of thermal module area, for the same total array size, also requires larger fan sizes. Specific fan sizing information is shown in Figures 7.8 - 7.14. Each figure shows the regenerator fan size as a function of total PV/T array area. Any PV/T array analyzed provides enough electrical energy to run all of the pumps and fans and will produce additional electricity. The amount of electricity produced does not vary significantly between location. However, lower percentage PV systems use much more fan power and electricity, which results in less energy savings compared to standard air-conditioning systems. Also, duct sizing and maximum duct flow rates were not analyzed, which would be factor to consider in regenerator fan sizing and meeting the temperature set-point.

The performance of different PV/T arrays and absorber flow rates are compared in Figures 7.15 - 7.21. Using a set-point of 55°C for each climate, the energy use and savings are shown for various PV/T array sizes and configurations in Table 8.1. The energy savings were calculated by comparing the electrical energy used in the simulated PV/T systems to the electrical energy estimate for a standard air-conditioning system latent load. At 50% load, less dehumidification is done by the PV/T system and more with the standard A/C.

Table 8.1: PV/T Array and Absorber Sizing

Location (-)	% PV Area (%)	Max Load			Half Load		
		Array/Absorber Size ($m^2/(L/s)$)	Energy Used (kWh/(L/s))	Energy Savings (%)	Array/Absorber Size ($m^2/(L/s)$)	Energy Used (kWh/(L/s))	Energy Savings (%)
Orlando	33.3	1.00	22.7	39.1	0.25	11.7	33.6
	50.0	1.21	22.55	39.5	0.3	11.65	33.7
	66.7	1.63	21.65	42.0	0.4	11.5	34.1
	80.0	2.21	21.45	42.4	0.6	11.25	34.8
Houston	33.3	0.86	20.0	40.0	0.225	10.5	32.6
	50.0	1.19	19.75	40.8	0.30	10.45	32.8
	66.7	1.59	19.65	41.1	0.40	10.2	33.5
	80.0	2.23	19.5	41.5	0.55	10.1	33.8
Atlanta	33.3	0.57	13.15	35.4	0.12	7	23.1
	50.0	0.70	13.05	35.9	0.15	6.95	23.3
	66.7	0.87	12.10	40.7	0.175	6.65	24.8
	80.0	1.17	12.00	41.2	0.25	6.6	25.1
Washington DC	33.3	0.64	10.75	26.9	0.125	5.35	19.8
	50.0	0.77	10.55	28.3	0.15	5.3	20.1
	66.7	0.97	9.75	33.6	0.195	5.15	21.2
	80.0	1.28	9.55	35.2	0.25	5.1	21.5
Phoenix	33.3	0.32	11.25	-49.4	0.02	1.60	-25.0
	50.0	0.42	12.5	-54.4	0.025	1.50	-27.1
	66.7	0.49	13	-56.1	0.03	1.45	-27.8
	80.0	0.78	13.5	-57.8	0.04	1.2	-32.1
Guam	33.3	1.33	32.2	47.7	0.4	16	47.6
	50.0	1.61	31.8	48.4	0.5	15.75	48.0
	66.7	2.25	31.6	48.7	0.95	15	49.2
	80.0	3.06	31.1	49.5	0.95	15	49.2
Honolulu	33.3	0.80	24.50	37.4	0.15	12.6	27.4
	50.0	1.01	23.95	38.9	0.2	12.5	27.6
	66.7	1.36	23.5	40.1	0.3	12.4	28.2
	80.0	1.95	23	41.3	0.4	12.3	28.2

From the table, it can be seen that Orlando, Houston, Guam, and Honolulu require very similar sized arrays and absorber rates to meet the maximum load fraction. These climates also have the most savings in electrical energy as compared to a standard system. Basically, in climates zones 1A and 2A, 30-50% energy savings can be achieved. Other humid climates, such as Washington DC and Atlanta, also show energy savings, but only between 20-25%. Phoenix is the only dry climate analyzed, and is also the only climate where standard A/C would use less energy in the

dehumidification process. Basically, there is very little latent load, but the regenerator fan would use much more electricity just to maintain the set-point temperature of the solar array. In all cases, increasing the PV percentage reduced overall energy use of the PV/T system and more energy was saved. However, the difference in percent savings of varying PV percentages was small, 1-10%. The high energy savings were found in the larger arrays, which are much too large to fit on building roofs. When reduced to 50% load, the array sizes were cut to 60% but savings also decreased. When meeting a 50% absorber load, most roofs would be able to accommodate the PV/T arrays while also meeting the minimum ventilation load and saving energy as compared to standard air-conditioning.

When sizing a PV/T-desiccant system, there are several known factors: building size, roof area, and minimum ventilation requirement. The main factor to determine is the percentage of the latent load that will be removed by the PV/T-desiccant system. After choosing a load percentage, the optimal system would be the highest percentage PV system and largest for the given roof area, operating at the lowest set-point above 40°C that can be maintained by the regenerator fan. While the higher PV percentage and lower set-points (larger regenerator fan) have opposite effects, the options are constrained by the roof area and limitations of the air ducts. The regenerator fan size will be limited by the maximum pressure and velocity of the air ducts. Fortunately, designing a system for a 50% load at the highest PV percentage and lowest set-point will give the greatest savings in energy compared to a standard COP 3.0 air-conditioning system. These savings will be greater than 25% in 1A and 2A climate zones. In other words, regeneration capacity will be maximized by using a lower set-point and higher air flow for a given PV/T area and percentage. The sizing charts are only necessary to help determine how much of the load will be reached.

8.2 Future Work

This thesis examined the appropriate sizing of components for a PV/T and liquid desiccant system, and will allow someone to determine the PV/T array size necessary for a specific building load. However, no specific buildings, building types, or space types were tested with this type of

system. Also, all of the climates studied also require a great deal of cooling throughout the year. Estimates and simulations only analyzed the latent load, with no additional cooling.

Simulation models should be created that analyze a combined PV/T-desiccant system with an HVAC cooling system. Additionally, another model should be created that includes an HVAC cooling system with a typical AIL liquid-heated regenerator setup. By creating both models, the energy use of a realistic combination dehumidification/cooling system could be examined and compared to understand which type of system provides the most energy savings. Further modeling could also include various building models, of different types, such as an office building, gymnasium, auditorium, etc.

In addition to simulating a more realistic, rather than experimental system, future work should involve testing of different control strategies. For this thesis, the absorber simply shuts off when the outdoor humidity is below a set-point. The simple controller does not adjust to changing outdoor air conditions. However, the flow rate could be adjusted to maintain an outlet humidity ratio, or the system could work in conjunction with the HVAC system to maintain comfort in a space.

Finally, future work should include system costs and energy savings to understand the pay-back of using such a system. Costs, especially of the PV/T array, should be determined, along with all component costs. By analyzing the system cost and the energy models of a combined desiccant-cooling system, the true cost benefits could be assessed.

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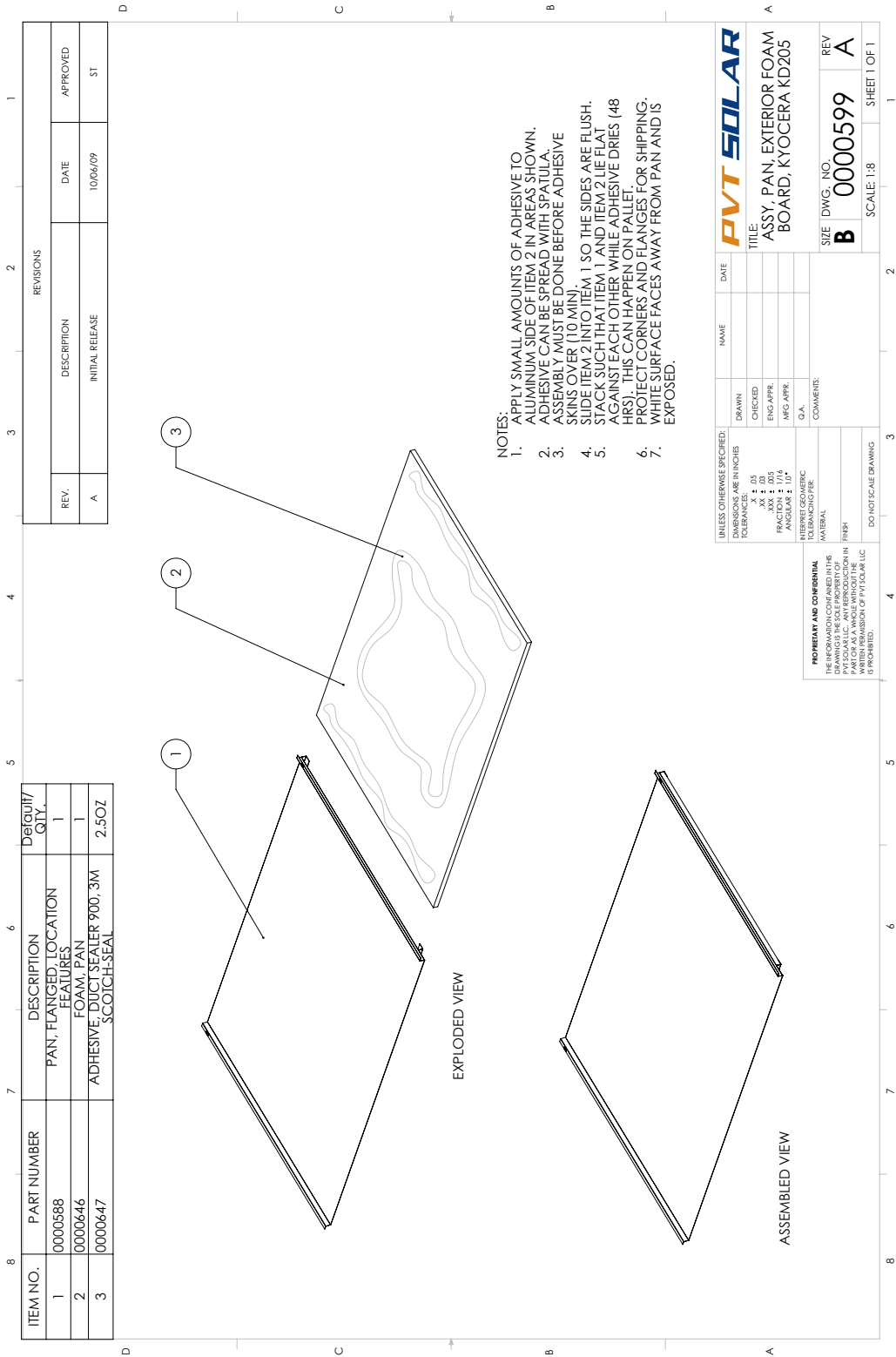
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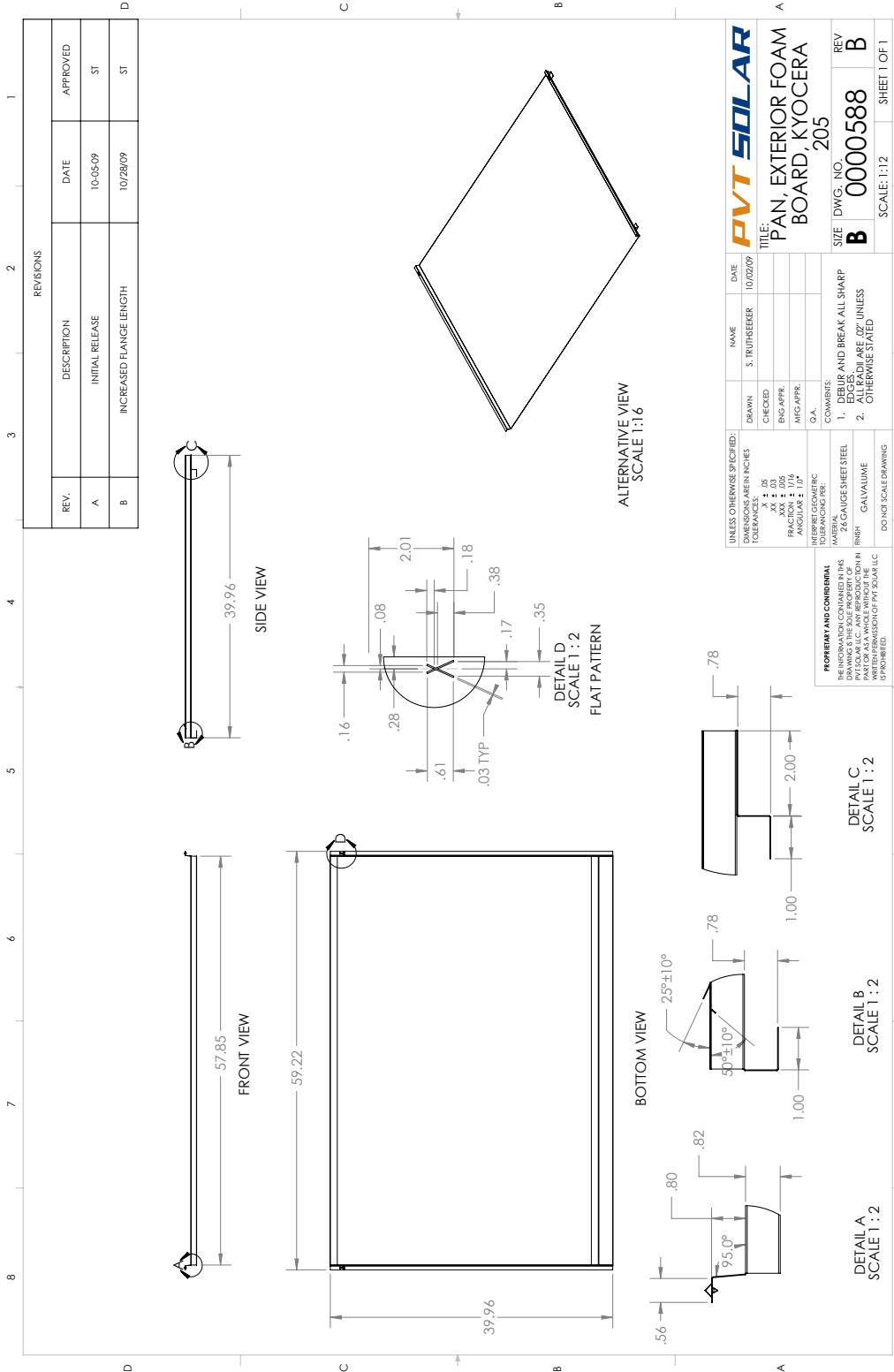
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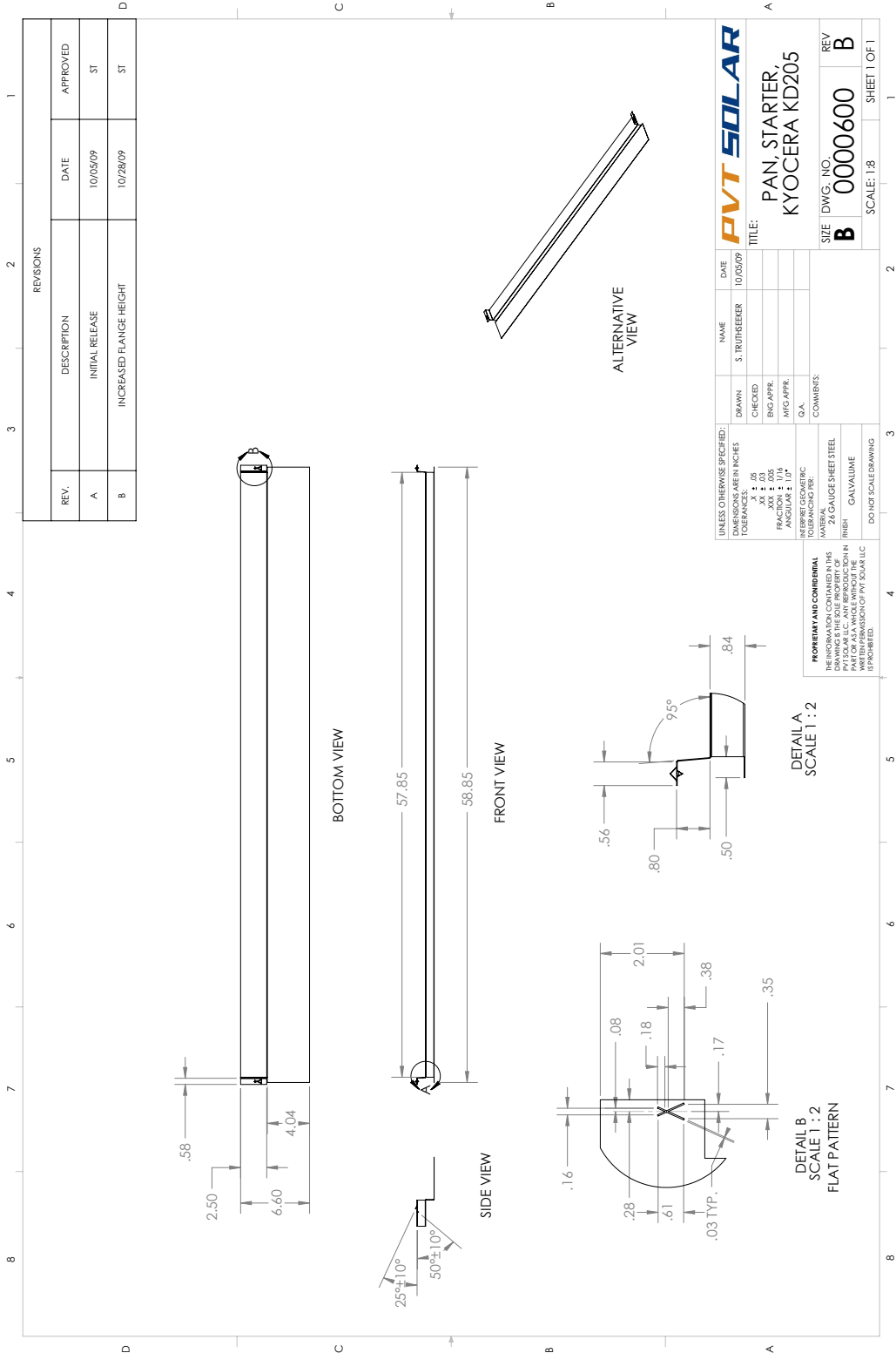
Appendix A

PVT System Documentation

A.1 Drawings







REVISIONS			
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B	INCREASED FLANGE HEIGHT	10/28/09	ST

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES:		DATE	10/05/09
XXX ± .05	XXX ± .005	NAME	S. TRUBREEKER
FRACTION ± 1/16	DECIMAL ± .001	DRAWN	
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:	CHECKED	
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FINISH: GALVALUME		MFG APPR.	
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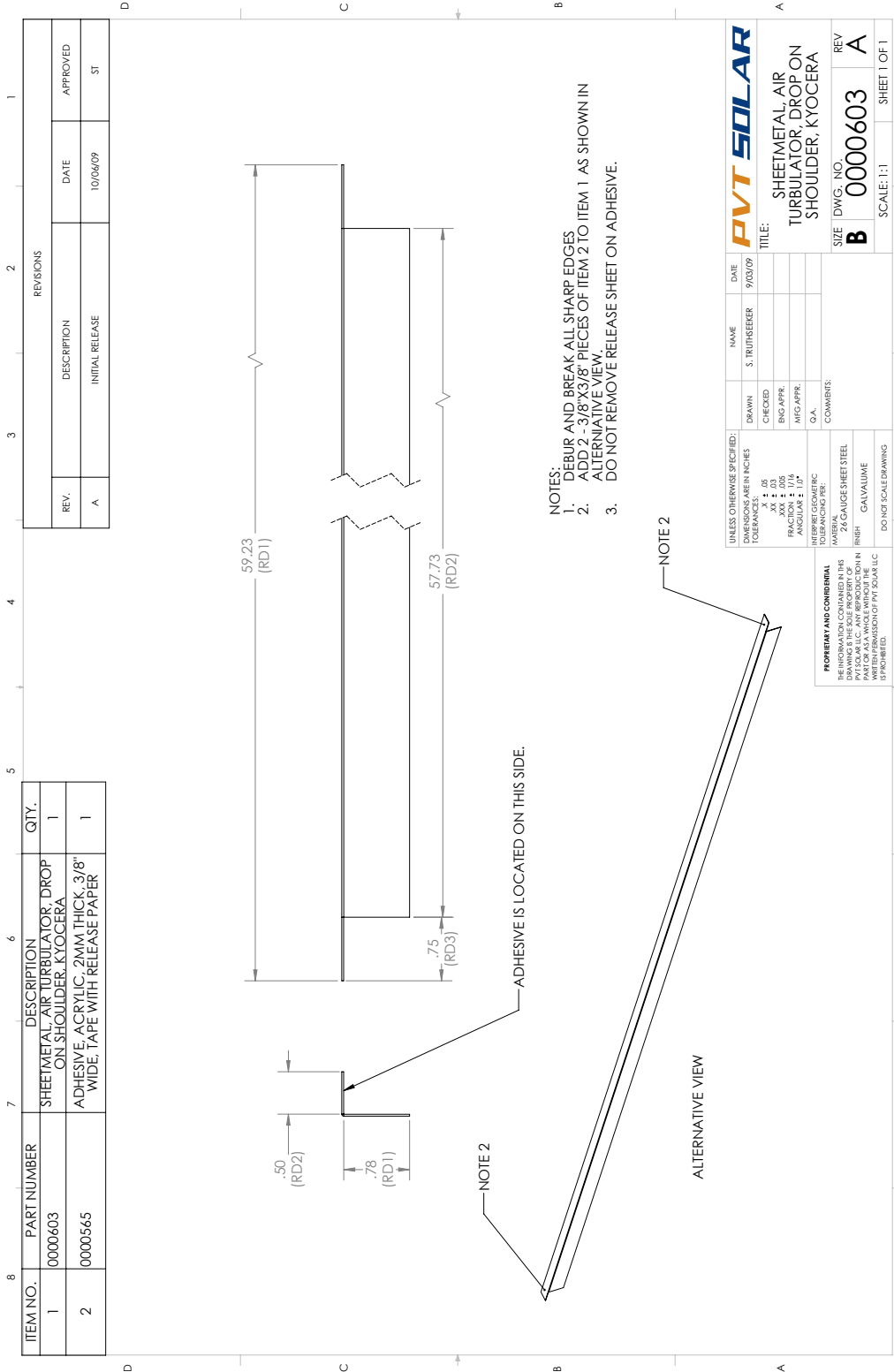
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DETAIL A
 SCALE 1 : 2

DETAIL B
 SCALE 1 : 2
 FLAT PATTERN

PVT SOLAR
 TITLE:
PAN STARTER, KYOCERA KD205

SIZE DWG. NO. **B 0000600** REV **B**
 SCALE: 1:8 SHEET 1 OF 1



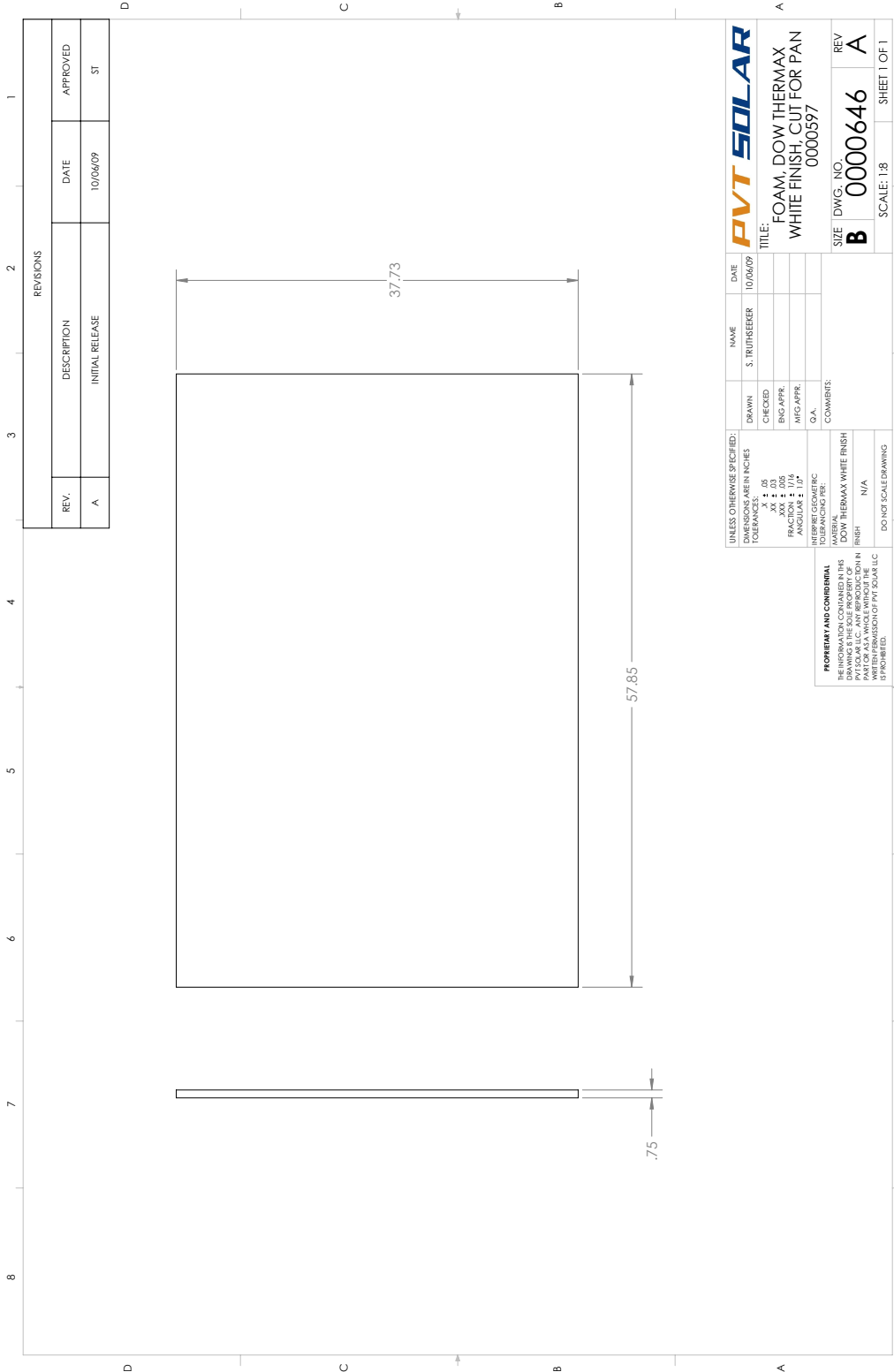
REVISIONS	
REV.	DESCRIPTION
A	INITIAL RELEASE
	DATE: 10/06/09
	APPROVED: ST

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
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2	0000565	ADHESIVE ACRYLIC .2MM THICK 3/8" WIDE TAPE WITH RELEASE PAPER	1

- NOTES:
- DEBUR AND BREAK ALL SHARP EDGES
 - ADD 2 - 3/8"x3/8" PIECES OF ITEM 2 TO ITEM 1 AS SHOWN IN ALTERNATIVE VIEW.
 - DO NOT REMOVE RELEASE SHEET ON ADHESIVE.

UNLESS OTHERWISE SPECIFIED: DIMENSIONS IN INCHES TOLERANCES: X .05 XX .005 XXX .005 FRACTIONS: 1/16 DECIMALS: 1/16 INTERPRET GEOMETRIC TOLERANCING PER: MATERIAL: 28 GAUGE SHEET STEEL FINISH: GALVALUME DO NOT SCALE DRAWING	DRAWN: S. TRUMBEEKER CHECKED: [] ENG APPR: [] MFG APPR: [] O.A.: [] COMMENTS: []	DATE: 9/03/09 NAME: S. TRUMBEEKER TITLE: PVT SOLAR SHEETMETAL AIR TURBULATOR DROP ON SHOULDER, KYOCERA SIZE: DWG. NO. B REV: 0000603 SCALE: 1:1 SHEET 1 OF 1
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REVISIONS		
REV.	DESCRIPTION	DATE
A	INITIAL RELEASE	10/06/09
ST		

PVT SOLAR

TITLE: **FOAM, DOW THERMAX WHITE FINISH, CUT FOR PAN**
0000597

DATE: 10/06/09

NAME: S. TRUBSEEKER

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COMMENTS:
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FINISH: N/A
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Appendix B

Additional Figures

B.1 Orlando

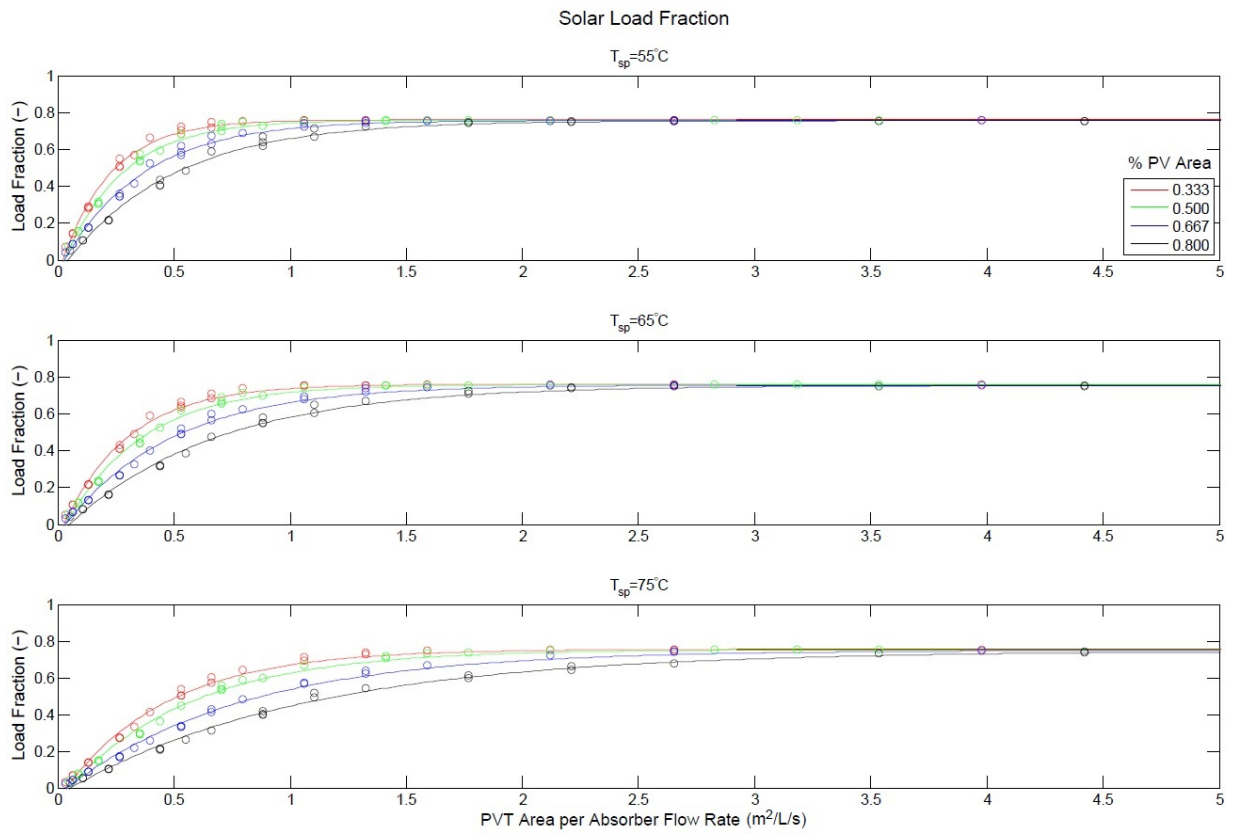


Figure B.1: Fitted Solar Load Fraction Curves by Area- Orlando

B.2 Guam

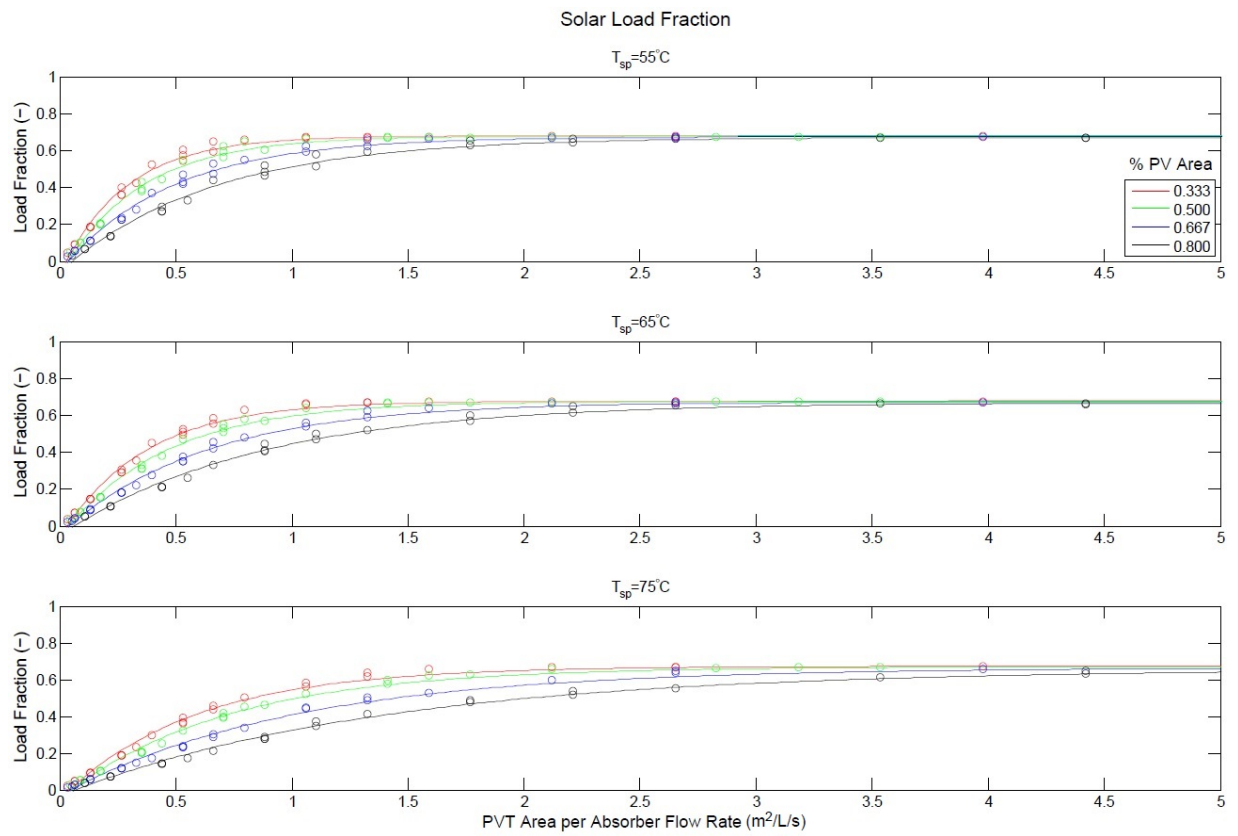


Figure B.2: Fitted Solar Load Fraction Curves by Area- Guam

B.3 Honolulu

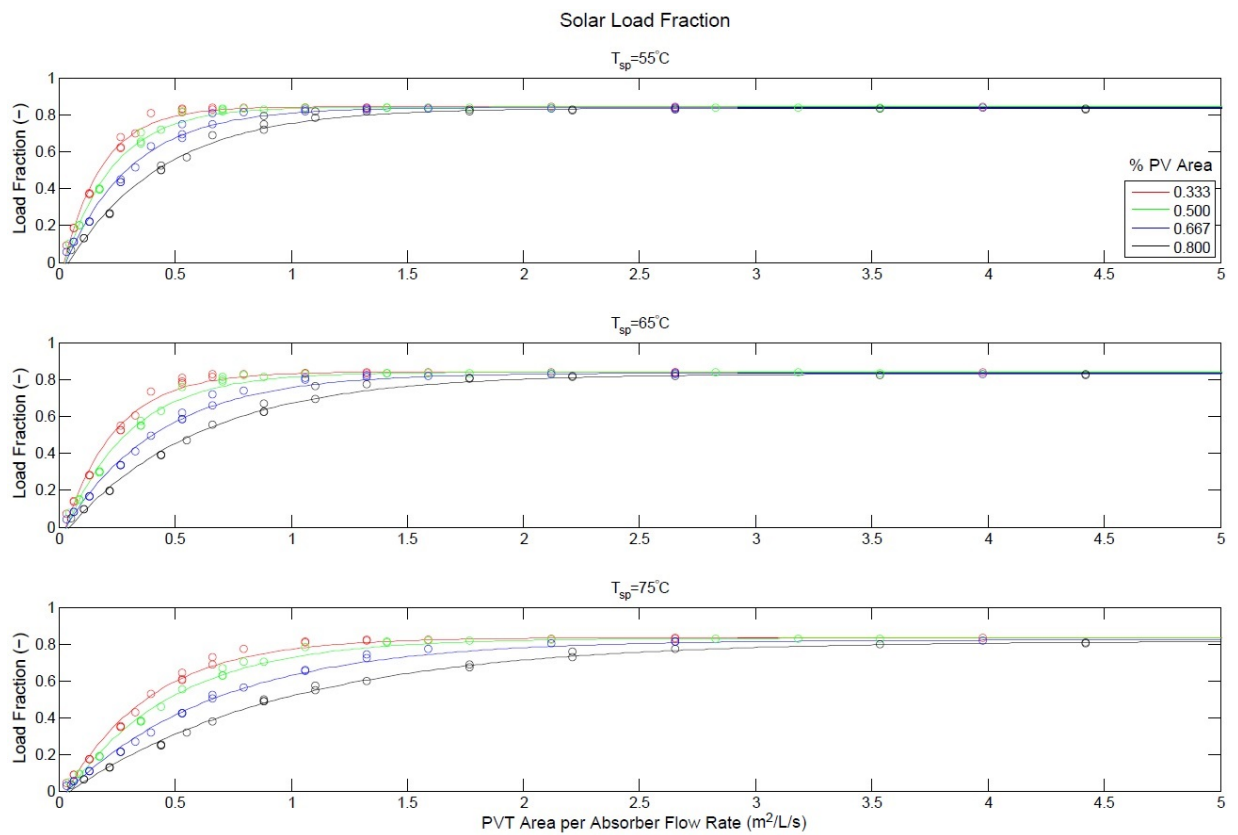


Figure B.3: Fitted Solar Load Fraction Curves by Area- Honolulu

B.4 Atlanta

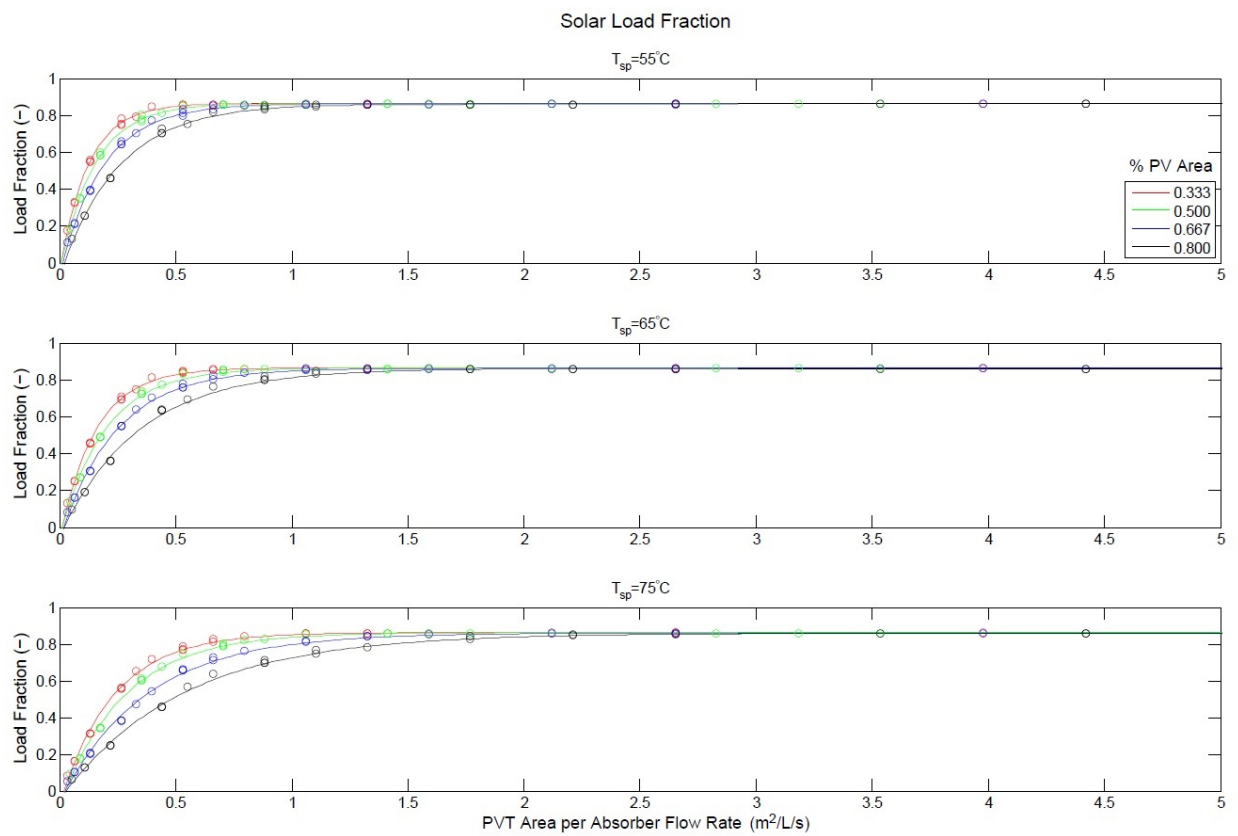


Figure B.4: Fitted Solar Load Fraction Curves by Area- Atlanta

B.5 Houston

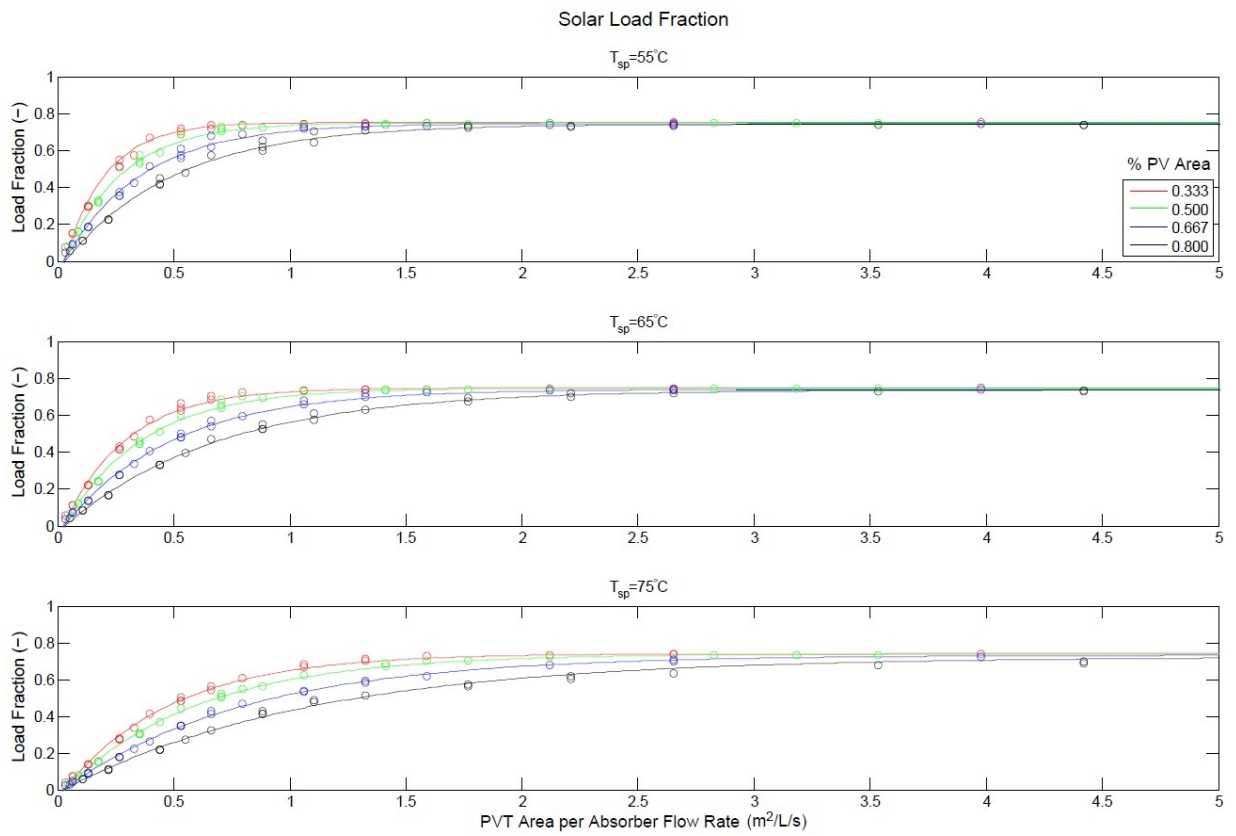


Figure B.5: Fitted Solar Load Fraction Curves by Area- Houston

B.6 Washington DC

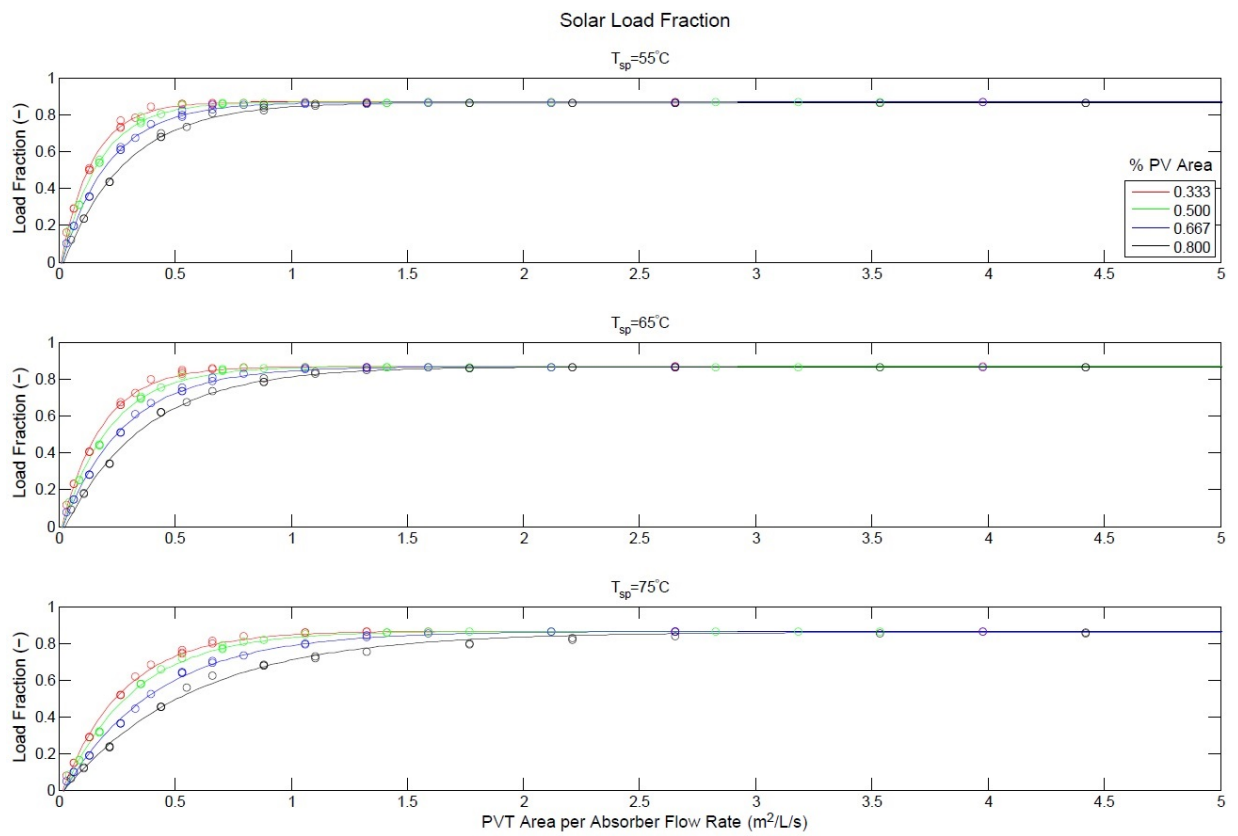


Figure B.6: Fitted Solar Load Fraction Curves by Area- Washington DC

B.7 Phoenix

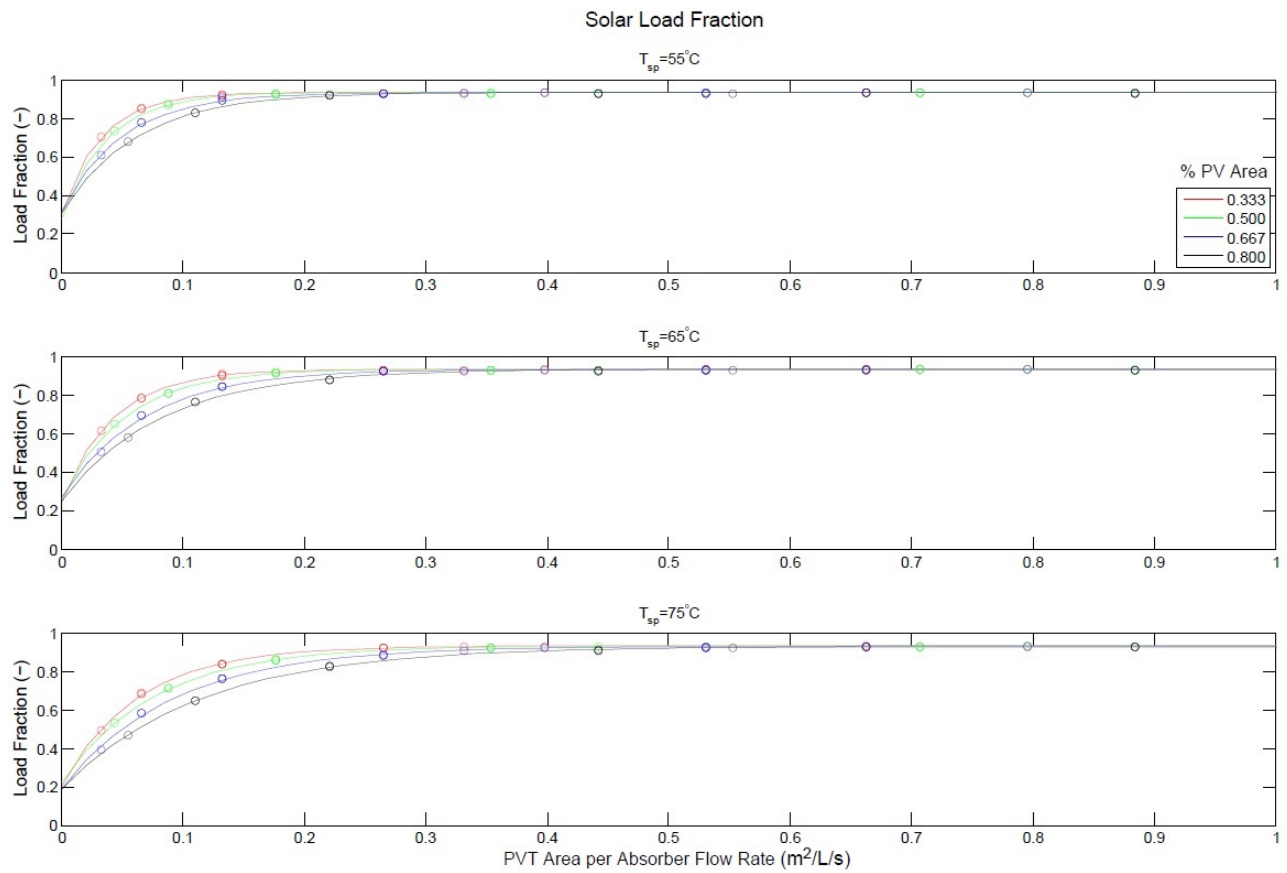


Figure B.7: Fitted Solar Load Fraction Curves by Area- Phoenix

Appendix C

TRNSYS Fortran Files

C.1 Type 937 - Warm Air Regenerator Model

```
SUBROUTINE TYPE937 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*****
C Object: Warm-Air Regenerator
C Simulation Studio Model: Type937
C Original:
C Author:
C
C Date: May 12, 2011 last modified: March 20, 2012
C
C
C ***
C *** Model Parameters
C ***
C C_p_d kJ/kg.K [-Inf;+Inf]
C
C ***
C *** Model Inputs
C ***
C P_atm_k kPa [0;200]
C T_in C [0;100]
C RH_in - [0;1]
C m_dot_a kg/hr [0;+Inf]
C xi_in - [0;1]
C T_d_in C [0;100]
C m_dot_LD_in kg/hr [0;+Inf]
C E_des - [0;1]
C E_h - [0;1]
C
C ***
C *** Model Outputs
C ***
C W_out - [0;+Inf]
C h_out - [0;+Inf]
C RH_out - [0;1]
C T_out C [0;100]
```

```

C m_dot_des kg/hr [0;+Inf]
C T_d_out C [0;100]
C m_dot_sd kg/hr [0;+Inf]
C xi_out - [0;1]

C ***
C *** Model Derivatives
C ***

C (Comments and routine interface generated by TRNSYS Studio)
C*****

C TRNSYS access functions (allow to access TIME etc.)
    USE TrnsysConstants
    USE TrnsysFunctions

C-----
C REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
    !DEC$ATTRIBUTES DLLEXPORT :: TYPE937 !SET THE CORRECT TYPE NUMBER HERE
C-----
C-----
C TRNSYS DECLARATIONS
    IMPLICIT NONE !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM
    DOUBLE PRECISION XIN !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
    DOUBLE PRECISION OUT !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
    DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET IT!
    DOUBLE PRECISION PAR !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
    DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP
    DOUBLE PRECISION T !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION SOLVER
    DOUBLE PRECISION DTDI !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
    INTEGER*4 INFO(15) !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND FROM THIS TYPE
    INTEGER*4 NP,NI,NOUT,ND !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
    INTEGER*4 NPAR,NIN,NDER !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
    INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
    INTEGER*4 ICNTRL !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW SOLVER
    INTEGER*4 NSTORED !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF STORAGE
    CHARACTER*3 OCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE OUTPUTS
    CHARACTER*3 YCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE INPUTS
C-----
C-----
C USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
    PARAMETER (NP=2,NI=8,NOUT=15,ND=0,NSTORED=0)
C-----
C-----
C REQUIRED TRNSYS DIMENSIONS
    DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
1 STORED(NSTORED),T(ND),DTDI(ND)
    INTEGER NITEMS

```

```
C-----  
C-----  
C   ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE  
C   PARAMETERS  
      DOUBLE PRECISION P_atm_k  
          DOUBLE PRECISION C_p_d  
  
C   INPUTS  
      DOUBLE PRECISION T_in  
          DOUBLE PRECISION RH_in  
DOUBLE PRECISION m_dot_a  
DOUBLE PRECISION xi_in  
          DOUBLE PRECISION T_d_in  
DOUBLE PRECISION m_dot_LD_in  
DOUBLE PRECISION E_w  
DOUBLE PRECISION E_h  
C   INTERNAL VARIABLES  
      DOUBLE PRECISION PSYDAT(9)  
          DOUBLE PRECISION STATUS  
  
          DOUBLE PRECISION P_atm  
          DOUBLE PRECISION P0  
DOUBLE PRECISION P1  
DOUBLE PRECISION P2  
DOUBLE PRECISION P3  
DOUBLE PRECISION P4  
DOUBLE PRECISION P5  
DOUBLE PRECISION P6  
DOUBLE PRECISION P7  
DOUBLE PRECISION P8  
DOUBLE PRECISION P9  
  
DOUBLE PRECISION T_w_crit  
DOUBLE PRECISION P_w_crit  
  
DOUBLE PRECISION phi  
DOUBLE PRECISION tau  
  
DOUBLE PRECISION A  
DOUBLE PRECISION B  
DOUBLE PRECISION F  
DOUBLE PRECISION F_2  
DOUBLE PRECISION P_25  
  
          DOUBLE PRECISION A0  
DOUBLE PRECISION A1  
DOUBLE PRECISION A2  
DOUBLE PRECISION A3  
DOUBLE PRECISION A4  
DOUBLE PRECISION A5
```



```

DOUBLE PRECISION P_v_rel
DOUBLE PRECISION P_v_w
DOUBLE PRECISION P_v_eq

      DOUBLE PRECISION W_eq
DOUBLE PRECISION h_eq
DOUBLE PRECISION RH_eq
DOUBLE PRECISION W_1
DOUBLE PRECISION W_2
DOUBLE PRECISION W_max

DOUBLE PRECISION W_in
DOUBLE PRECISION W_what
DOUBLE PRECISION h_in

DOUBLE PRECISION Q_a
DOUBLE PRECISION m_dot_LD_out
C   OUTPUT VARIABLES
      DOUBLE PRECISION W_out
DOUBLE PRECISION h_out
DOUBLE PRECISION RH_out
DOUBLE PRECISION T_out
DOUBLE PRECISION m_dot_des
DOUBLE PRECISION T_d_out
DOUBLE PRECISION m_dot_sd
DOUBLE PRECISION xi_out
DOUBLE PRECISION LD_ratio

C-----
C   READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
P_atm_k      = PAR(1)
C_p_d        = PAR(2)

C-----
C   RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER
T_in         = XIN(1)
      RH_in      = XIN(2)
m_dot_a      = XIN(3)
xi_in        = XIN(4)
      T_d_in     = XIN(5)
m_dot_LD_in  = XIN(6)
E_w          = XIN(7)
E_h          = XIN(8)

IUNIT        = INFO(1)
ITYPE        = INFO(2)

C-----
C   SET THE VERSION INFORMATION FOR TRNSYS
      IF(INFO(7).EQ.-2) THEN
          INFO(12)=16
      RETURN 1

```

```

ENDIF
C-----
C-----
C DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(8).EQ.-1) THEN
        RETURN 1
      ENDIF
C-----

C-----
C PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C e.g. save variables to storage array for the next timestep
      IF (INFO(13).GT.0) THEN
        NITEMS=0
C        STORED(1)=... (if NITEMS > 0)
C        CALL setStorageVars(STORED,NITEMS,INFO)
        RETURN 1
      ENDIF
C
C-----

C-----
C DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(7).EQ.-1) THEN
C        SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
        INFO(6)=NOUT
        INFO(9)=1
        INFO(10)=0 !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C        SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN THE INPUT FILE
C        IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS MODEL....

        NIN=NI
        NPAR=NP
        NDER=ND

C        CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C        THE TRNSYS INPUT FILE
        CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C        SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
        NITEMS=0
C        CALL setStorageSize(NITEMS,INFO)
C        RETURN TO THE CALLING PROGRAM
        RETURN 1
      ENDIF
C-----

C-----
C DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INTIAL TIME

```

```

        IF (TIME .LT. (getSimulationStartTime() +
        .   getSimulationTimeStep()/2.D0)) THEN

C       SET THE UNIT NUMBER FOR FUTURE CALLS
        IUNIT=INFO(1)
        ITYPE=INFO(2)

C       CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE SUBROUTINE IF AN ERROR IS FOUND
C       IF(...) CALL TYPECK(-4,INFO,0,"BAD PARAMETER #",0)

C       PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE
C       Find max humidity ratio from Conde (2003)
C       Fitted constants for empirical relation
        P0 = 0.31
        P1 = 3.698
        P2 = 0.60
        P3 = 0.231
        P4 = 4.584
        P5 = 0.49
        P6 = 0.478
        P7 = -5.20
        P8 = -0.40
        P9 = 0.018

C       Critical Temp (K) and Pressure (kPa) of water
        T_w_crit = 647.1
        P_w_crit = 22064.

C       Equations from Conde
        phi = (T_d_in + 273.15)/T_w_crit
        tau = 1. - phi
        A = 2. - (1. + (xi_in/P0)**P1)**P2
        B = (1. + (xi_in/P3)**P4)**P5 - 1.
        F = A + B*phi
        P_25 = 1. - (1. + (xi_in/P6)**P7)**P8 -
        1      P9*exp(-((xi_in - 0.1)**2.)/0.005)

C       Relative Vapour Pressure of LiCl (Relative to water at same temp)
        P_v_rel = P_25*F

C       Find vapour Pressure of water over liquid phase (Conde - Appendix A)
C       Fitted constants from Conde - Appendix A
        A0 = -7.858230
        A1 = 1.839910
        A2 = -11.781100
        A3 = 22.670500
        A4 = -15.939300
        A5 = 1.775160

C       Equation from Conde
        F_2 = (A0*tau + A1*tau**1.5 + A2*tau**3. +

```

```

1      A3*tau**3.5 + A4*tau**4. + A5*tau**7.5)/(1.-tau)

C      Water vapor pressure over liquid phase
P_v_w = exp(F_2)*P_w_crit

C      Desiccant vapor pressure
P_v_eq = P_v_w*P_v_rel

C      Calc air properties of air at same vapor pressure as Ca-Cl solution
W_eq = (0.6219*P_v_eq)/(P_atm_k - P_v_eq)

P_atm = P_atm_k/101.325

PSYDAT(1) = P_atm
PSYDAT(2) = T_d_in
PSYDAT(6) = W_eq
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,4,0,PSYDAT,2,STATUS,*101)
CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
101    CONTINUE

IF(STATUS.GT.0) THEN
    CALL MESSAGES(-1,'psych101','WARNING',IUNIT,ITYPE)
ENDIF

C      Max enthalpy
h_eq = PSYDAT(7)
RH_eq = PSYDAT(4)

C      Calculate inlet air properties
PSYDAT(1) = P_atm
PSYDAT(2) = T_in
PSYDAT(4) = RH_in

C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,2,0,PSYDAT,2,STATUS,*201)
CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
201    CONTINUE

IF(STATUS.GT.0) THEN
    CALL MESSAGES(-1,'psych201','WARNING',IUNIT,ITYPE)
ENDIF

C      Entering Air Enthalpy
W_in = PSYDAT(6)
h_in = PSYDAT(7)

PSYDAT(1) = P_atm
PSYDAT(4) = RH_eq
PSYDAT(7) = h_in

C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,8,0,PSYDAT,2,STATUS,*301)
CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
301    CONTINUE

```

```

IF (STATUS.GT.0) THEN
  CALL MESSAGES(-1,'psych301','WARNING',IUNIT,ITYPE)
ENDIF
W_max = PSYDAT(6)

IF((m_dot_a.EQ.0).OR.(m_dot_LD_in.EQ.0)) THEN
  W_out = W_in
h_out = h_in
  T_out = T_in
  RH_out = RH_in

  m_dot_des = 0.0
  Q_a = 0.0
  m_dot_LD_out = m_dot_LD_in
  T_d_out = T_d_in
  m_dot_sd = xi_in*m_dot_LD_in
  xi_out = xi_in
  LD_ratio = 0
ELSE
C    Calculate outlet air properties
  W_out = W_in + E_w*(W_max - W_in)
h_out = h_in - E_h*(h_in - h_eq)

PSYDAT(1) = P_atm
  PSYDAT(6) = W_out
  PSYDAT(7) = h_out
C    DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
  CALL PSYCHROMETRICS(TIME,INFO,1,6,0,PSYDAT,2,STATUS,*401)
  CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
401  CONTINUE

IF (STATUS.GT.0) THEN
  CALL MESSAGES(-1,'psych401','WARNING',IUNIT,ITYPE)
ENDIF
C    Leaving Air Temperature
  T_out = PSYDAT(2)
  RH_out = PSYDAT(4)

C    Rate of desorbed moisture
  m_dot_des = m_dot_a*(W_out - W_in)

C    Heat transfer between desiccant and air
  Q_a = m_dot_a*(h_out - h_in)

C    Outlet desiccant flow rate
  m_dot_LD_out = m_dot_LD_in - m_dot_des

C    Desiccant outlet temp
  T_d_out = (m_dot_LD_in*C_p_d*T_d_in - Q_a)/(m_dot_LD_out*C_p_d)

```

```

C      Flow rate of the solution
      m_dot_sd = xi_in*m_dot_LD_in

      xi_out = (m_dot_sd/(m_dot_LD_out))
      LD_ratio = xi_in/xi_out

      ENDIF

C      PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE
      NITEMS=0
C      STORED(1)=...

C      PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY
      CALL setStorageVars(STORED,NITEMS,INFO)

C      RETURN TO THE CALLING PROGRAM
      RETURN 1

      ENDIF
C-----
C-----
C      *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C-----
C-----
C      RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION
C      NITEMS=
C      CALL getStorageVars(STORED,NITEMS,INFO)
C      STORED(1)=
C-----
C-----
C      CHECK THE INPUTS FOR PROBLEMS
C      IF(...) CALL TYPECK(-3,INFO,'BAD INPUT #',0,0)
C      IF(IERROR.GT.0) RETURN 1
C-----
C-----
C      *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
C-----
C      ADD YOUR COMPONENT EQUATIONS HERE; BASICALLY THE EQUATIONS THAT WILL
C      CALCULATE THE OUTPUTS BASED ON THE PARAMETERS AND THE INPUTS. REFER TO
C      CHAPTER 3 OF THE TRNSYS VOLUME 1 MANUAL FOR DETAILED INFORMATION ON
C      WRITING TRNSYS COMPONENTS.
C
C      Find max humidity ratio from Conde (2003)
C      Fitted constants for empirical relation
      P0 = 0.31
      P1 = 3.698
      P2 = 0.60
      P3 = 0.231
      P4 = 4.584
      P5 = 0.49

```

```

P6 = 0.478
P7 = -5.20
P8 = -0.40
P9 = 0.018

C      Critital Temp (K) and Pressure (kPa) of water
T_w_crit = 647.1
P_w_crit = 22064.

C      Equations from Conde
phi = (T_d_in + 273.15)/T_w_crit
tau = 1. - phi
A = 2. - (1. + (xi_in/P0)**P1)**P2
B = (1. + (xi_in/P3)**P4)**P5 - 1.
F = A + B*phi
P_25 = 1. - (1. + (xi_in/P6)**P7)**P8 -
1      P9*exp(-(xi_in - 0.1)**2.)/0.005

C      Relative Vapour Pressure of LiCl (Relative to water at same temp)
P_v_rel = P_25*F

C      Find vapour Pressure of water over liquid phase (Conde - Appendix A)
C      Fitted constants from Conde - Appendix A
A0 = -7.858230
A1 = 1.839910
A2 = -11.781100
A3 = 22.670500
A4 = -15.939300
A5 = 1.775160

C      Equation from Conde
F_2 = (A0*tau + A1*tau**1.5 + A2*tau**3. +
1      A3*tau**3.5 + A4*tau**4. + A5*tau**7.5)/(1.-tau)

C      Water vapor pressure over liquid phase
P_v_w = exp(F_2)*P_w_crit

C      Desiccant vapor pressure
P_v_eq = P_v_w*P_v_rel

C      Calc air properties of air at same vapor pressure as Ca-Cl solution
W_eq = (0.6219*P_v_eq)/(P_atm_k - P_v_eq)

P_atm = P_atm_k/101.325

PSYDAT(1) = P_atm
PSYDAT(2) = T_d_in
PSYDAT(6) = W_eq
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,4,0,PSYDAT,2,STATUS,*151)
CALL LINKKCK('TYPE 937','PSYCHROMETRICS',1,99)

```

```

151      CONTINUE
        IF (STATUS.GT.0) THEN
            CALL MESSAGES(-1,'psych151','WARNING',IUNIT,ITYPE)
        ENDIF
C      Max enthalpy
        h_eq = PSYDAT(7)
        RH_eq = PSYDAT(4)

C      Calculate inlet air properties
        PSYDAT(1) = P_atm
        PSYDAT(2) = T_in
        PSYDAT(4) = RH_in

C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
        CALL PSYCHROMETRICS(TIME,INFO,1,2,0,PSYDAT,2,STATUS,*251)
        CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
251      CONTINUE

        IF (STATUS.GT.0) THEN
            CALL MESSAGES(-1,'psych251','WARNING',IUNIT,ITYPE)
        ENDIF
C      Entering Air Enthalpy
        W_in = PSYDAT(6)
        h_in = PSYDAT(7)

        PSYDAT(1) = P_atm
        PSYDAT(4) = RH_eq
        PSYDAT(7) = h_in

C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
        CALL PSYCHROMETRICS(TIME,INFO,1,8,0,PSYDAT,2,STATUS,*351)
        CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
351      CONTINUE

        IF (STATUS.GT.0) THEN
            CALL MESSAGES(-1,'psych351','WARNING',IUNIT,ITYPE)
        ENDIF
        W_max = PSYDAT(6)

        IF ((m_dot_a.EQ.0).OR.(m_dot_LD_in.EQ.0)) THEN
            W_out = W_in
            W_what = W_out
h_out = h_in
            T_out = T_in
            RH_out = RH_in

        m_dot_des = 0.0
        Q_a = 0.0
        m_dot_LD_out = m_dot_LD_in
        T_d_out = T_d_in
            m_dot_sd = xi_in*m_dot_LD_in
            xi_out = xi_in
            LD_ratio = 0

```



```

ELSE
C      Calculate outlet air properties
      W_out = W_in + E_w*(W_max - W_in)
      h_out = h_in - E_h*(h_in - h_eq)

      W_what = W_out

PSYDAT(1) = P_atm
      PSYDAT(6) = W_out
      PSYDAT(7) = h_out
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
      CALL PSYCHROMETRICS(TIME,INFO,1,6,0,PSYDAT,2,STATUS,*451)
      CALL LINKCK('TYPE 937','PSYCHROMETRICS',1,99)
451    CONTINUE

      IF(STATUS.GT.0) THEN
          CALL MESSAGES(-1,'psych451','WARNING',IUNIT,ITYPE)
      ENDIF
C      Leaving Air Temperature
      T_out = PSYDAT(2)
      RH_out = PSYDAT(4)

C      Rate of desorbed moisture
      m_dot_des = m_dot_a*(W_out - W_in)

C      Heat transfer between desiccant and air
      Q_a = m_dot_a*(h_out - h_in)

C      Outlet desiccant flow rate
      m_dot_LD_out = m_dot_LD_in - m_dot_des

C      Desiccant outlet temp
      T_d_out = (m_dot_LD_in*C_p_d*T_d_in - Q_a)/(m_dot_LD_out*C_p_d)

C      Flow rate of the solution
      m_dot_sd = xi_in*m_dot_LD_in

      xi_out = (m_dot_sd/(m_dot_LD_out))
      LD_ratio = xi_in/xi_out
      ENDIF
C-----
C-----
C-----
C      SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
C      NITEMS=
C      STORED(1)=
C      CALL setStorageVars(STORED,NITEMS,INFO)
C-----
C-----
C      REPORT ANY PROBLEMS THAT HAVE BEEN FOUND USING CALLS LIKE THIS:

```

```

C      CALL MESSAGES(-1,'put your message here','MESSAGE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','WARNING',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','SEVERE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','FATAL',IUNIT,ITYPE)
C-----
C-----
C      SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT
C      Set output variables

OUT(1)=W_out
OUT(2) = h_out
OUT(3)= RH_out
OUT(4)=T_out
OUT(5)=m_dot_des
OUT(6)=T_d_out
OUT(7)=E_w
OUT(8)=xi_out

OUT(9) = W_max
OUT(10)= h_eq

OUT(11)= W_in
OUT(12)= m_dot_LD_out

OUT(13)= m_dot_a
OUT(14)= m_dot_LD_in

OUT(15)= LD_ratio

C-----
C      EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
C      RETURN 1
C      END
C-----

```

C.2 Type 974 - Cross Flow Absorber Model

```

SUBROUTINE TYPE974 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*****
C Object: Cross-Flow Absorber
C Simulation Studio Model: Type974
C Original:
C Author:
C Editor:
C Date: May 13, 2011 last modified: 1/20/2011
C
C
C ***
C *** Model Parameters
C ***

```

```

C C_p_d kJ/kg.K [-Inf;+Inf]
C
C
C ***
C *** Model Inputs
C ***
C P_atm kPa [0;200]
C T_in C [0;100]
C RH_in - [0;1]
C m_dot_a kg/hr [0;+Inf]
C xi_in - [0;1]
C T_d_in C [0;100]
C m_dot_LD_in kg/hr [0;+Inf]
C E_abs - [0;1]
C E_h - [0;1]

C ***
C *** Model Outputs
C ***
C W_out - [0;+Inf]
C h_out - [0;+Inf]
C RH_out - [0;1]
C T_out C [0;100]
C m_dot_abs kg/hr [0;+Inf]
C T_d_out C [0;100]
C m_dot_sd kg/hr [0;+Inf]
C xi_out - [0;1]

C ***
C *** Model Derivatives
C ***

C (Comments and routine interface generated by TRNSYS Studio)
C*****

C TRNSYS access functions (allow to access TIME etc.)
    USE TrnsysConstants
    USE TrnsysFunctions

C-----
C REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
    !DEC$ATTRIBUTES DLLEXPORT :: TYPE974 !SET THE CORRECT TYPE NUMBER HERE
C-----
C-----
C TRNSYS DECLARATIONS
    IMPLICIT NONE !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM
    DOUBLE PRECISION XIN !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
    DOUBLE PRECISION OUT !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
    DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET IT!
    DOUBLE PRECISION PAR !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
    DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP

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DOUBLE PRECISION T !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION SOLVER
DOUBLE PRECISION DTDI !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
INTEGER*4 INFO(15) !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND FROM THIS TYPE
INTEGER*4 NP,NI,NOUT,ND !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
INTEGER*4 NPAR,NIN,NDER !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
INTEGER*4 ICNTRL !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW SOLVER
INTEGER*4 NSTORED !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF STORAGE
CHARACTER*3 OCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE OUTPUTS
CHARACTER*3 YCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE INPUTS
C-----
C-----
C USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
      PARAMETER (NP=3,NI=11,NOUT=18,ND=0,NSTORED=0)
C-----
C-----
C REQUIRED TRNSYS DIMENSIONS
      DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
1 STORED(NSTORED),T(ND),DTD(ND)
      INTEGER NITEMS
C-----
C-----
C ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE
C PARAMETERS
      DOUBLE PRECISION P_atm_k
      DOUBLE PRECISION C_p_d
      DOUBLE PRECISION C_p_cw
C INPUTS

      DOUBLE PRECISION T_in
      DOUBLE PRECISION RH_in
DOUBLE PRECISION m_dot_a
DOUBLE PRECISION xi_in
      DOUBLE PRECISION T_d_in
DOUBLE PRECISION m_dot_LD_in
DOUBLE PRECISION T_cw_in
DOUBLE PRECISION m_dot_cw
DOUBLE PRECISION E_abs
DOUBLE PRECISION E_h
DOUBLE PRECISION E_cw
C INTERNAL VARIABLES
      DOUBLE PRECISION PSYDAT(9)
      DOUBLE PRECISION STATUS

      DOUBLE PRECISION P_atm
      DOUBLE PRECISION P0
DOUBLE PRECISION P1
DOUBLE PRECISION P2

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```
DOUBLE PRECISION P3
DOUBLE PRECISION P4
DOUBLE PRECISION P5
DOUBLE PRECISION P6
DOUBLE PRECISION P7
DOUBLE PRECISION P8
DOUBLE PRECISION P9

DOUBLE PRECISION T_w_crit
DOUBLE PRECISION P_w_crit

DOUBLE PRECISION phi
DOUBLE PRECISION tau

DOUBLE PRECISION A
DOUBLE PRECISION B
DOUBLE PRECISION F
DOUBLE PRECISION F_2
DOUBLE PRECISION P_25

DOUBLE PRECISION A0
DOUBLE PRECISION A1
DOUBLE PRECISION A2
DOUBLE PRECISION A3
DOUBLE PRECISION A4
DOUBLE PRECISION A5

DOUBLE PRECISION P_v_rel
DOUBLE PRECISION P_v_w
DOUBLE PRECISION P_v_min

DOUBLE PRECISION W_min
DOUBLE PRECISION h_min

DOUBLE PRECISION W_in
DOUBLE PRECISION h_in

DOUBLE PRECISION Q_a
DOUBLE PRECISION Q_cw
DOUBLE PRECISION T_cw_out
DOUBLE PRECISION LD_ratio
DOUBLE PRECISION m_dot_LD_out
C  OUTPUT VARIABLES
DOUBLE PRECISION W_out
DOUBLE PRECISION h_out
DOUBLE PRECISION RH_out
DOUBLE PRECISION T_out
DOUBLE PRECISION m_dot_abs
DOUBLE PRECISION T_d_out
DOUBLE PRECISION m_dot_sd
DOUBLE PRECISION xi_out
```

```

C-----
C  READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
  P_atm_k    = PAR(1)
  C_p_d      = PAR(2)
  C_p_cw     = PAR(3)
C-----
C  RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER
  T_in       = XIN(1)
  RH_in      = XIN(2)
m_dot_a     = XIN(3)
xi_in       = XIN(4)
  T_d_in     = XIN(5)
m_dot_LD_in = XIN(6)
T_cw_in     = XIN(7)
m_dot_cw    = XIN(8)
E_abs       = XIN(9)
E_h         = XIN(10)
E_cw        = XIN(11)

  IUNIT      = INFO(1)
  ITYPE      = INFO(2)
C-----
C  SET THE VERSION INFORMATION FOR TRNSYS
  IF(INFO(7).EQ.-2) THEN
    INFO(12)=16
  RETURN 1
ENDIF
C-----

C-----
C  DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
  IF (INFO(8).EQ.-1) THEN
    RETURN 1
  ENDIF
C-----

C-----
C  PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C  e.g. save variables to storage array for the next timestep
  IF (INFO(13).GT.0) THEN
    NITEMS=0
C    STORED(1)=... (if NITEMS > 0)
C    CALL setStorageVars(STORED,NITEMS,INFO)
  RETURN 1
  ENDIF
C
C-----
C-----
C  DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE

```

```

IF (INFO(7).EQ.-1) THEN
C     SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
      INFO(6)=NOUT
      INFO(9)=1
      INFO(10)=0 !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C     SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN THE INPUT FILE
C     IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS MODEL....

      NIN=NI
      NPAR=NP
      NDER=ND

C     CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C     THE TRNSYS INPUT FILE
      CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C     SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
      NITEMS=0
C     CALL setStorageSize(NITEMS,INFO)
C     RETURN TO THE CALLING PROGRAM
      RETURN 1
ENDIF

C-----
C-----
C DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INITIAL TIME
      IF (TIME .LT. (getSimulationStartTime() +
.      getSimulationTimeStep()/2.DO)) THEN

C     SET THE UNIT NUMBER FOR FUTURE CALLS
      IUNIT=INFO(1)
      ITYPE=INFO(2)

C     CHECK THE PARAMETERS FOR PROBLEMS AND RETURN FROM THE SUBROUTINE IF AN ERROR IS FOUND
C     IF(...) CALL TYPECK(-4,INFO,0,"BAD PARAMETER #",0)

C     PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE

C Find max humidity ratio from Conde (2003)
C Fitted constants for empirical relation
P0 = 0.31
P1 = 3.698
P2 = 0.60
P3 = 0.231
P4 = 4.584
P5 = 0.49
P6 = 0.478

```

```

P7 = -5.20
P8 = -0.40
P9 = 0.018

C      Critital Temp (K) and Pressure (kPa) of water
T_w_crit = 647.1
P_w_crit = 22064.

C      Equations from Conde
phi = (T_cw_in + 273.15)/T_w_crit
tau = 1. - phi
A = 2. - (1. + (xi_in/P0)**P1)**P2
B = (1. + (xi_in/P3)**P4)**P5 - 1.
F = A + B*phi
P_25 = 1. - (1. + (xi_in/P6)**P7)**P8 -
1      P9*exp(-(xi_in - 0.1)**2.)/0.005

C      Relative Vapour Pressure of CaCl (Relative to water at same temp)
P_v_rel = P_25*F

C      Find vapour Pressure of water over liquid phase (Conde - Appendix A)
C      Fitted constants from Conde - Appendix A
A0 = -7.858230
A1 = 1.839910
A2 = -11.781100
A3 = 22.670500
A4 = -15.939300
A5 = 1.775160

C      Equation from Conde
F_2 = (A0*tau + A1*tau**1.5 + A2*tau**3. +
1      A3*tau**3.5 + A4*tau**4. + A5*tau**7.5)/(1.-tau)

C      Water vapor pressure over liquid phase
P_v_w = exp(F_2)*P_w_crit

C      Desiccant vapor pressure
P_v_min = P_v_w*P_v_rel

C      Calc air properties of air at same vapor pressure as Ca-Cl solution
W_min = (0.6219*P_v_min)/(P_atm_k - P_v_min)

P_atm = P_atm_k/101.325

PSYDAT(1) = P_atm
PSYDAT(2) = T_in
PSYDAT(6) = W_min

C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,4,0,PSYDAT,2,STATUS,*101)
CALL LINKCK('TYPE 974','PSYCHROMETRICS',1,99)
101      CONTINUE

```



```

C      Min enthalpy
      h_min = PSYDAT(7)

C      Calculate inlet air properties
      PSYDAT(1) = P_atm
      PSYDAT(2) = T_in
      PSYDAT(4) = RH_in
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
      CALL PSYCHROMETRICS(TIME,INFO,1,2,0,PSYDAT,2,STATUS,*201)
      CALL LINKCK('TYPE 974','PSYCHROMETRICS',1,99)
201    CONTINUE
C      Entering Air Enthalpy
      W_in = PSYDAT(6)
      h_in = PSYDAT(7)

      IF((m_dot_a.EQ.0).OR.(m_dot_LD_in.EQ.0)) THEN
          T_out = T_in
          W_out = W_in
      h_out = h_in
          RH_out = RH_in
          T_d_out = T_d_in
          m_dot_abs = 0.0
          m_dot_sd = xi_in*m_dot_LD_in
          m_dot_LD_out = m_dot_LD_in
          xi_out = xi_in
          LD_ratio = 0.0
          T_cw_out=T_cw_in
          Q_a = 0.0
          Q_cw = 0.0
          ELSE
C      Calculate outlet air properties
      W_out = W_in - E_abs*(W_in - W_min)
      h_out = h_in - E_h*(h_in - h_min)

      PSYDAT(1) = P_atm
      PSYDAT(6) = W_out
      PSYDAT(7) = h_out
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
      CALL PSYCHROMETRICS(TIME,INFO,1,6,0,PSYDAT,2,STATUS,*301)
      CALL LINKCK('TYPE 974','PSYCHROMETRICS',1,99)
301    CONTINUE
C      Leaving Air Temperature
      T_out = PSYDAT(2)
      RH_out = PSYDAT(4)

C      Desiccant outlet temp
      T_d_out = T_d_in - E_cw*(T_d_in - T_cw_in)

C      Rate of desorbed moisture
      m_dot_abs = m_dot_a*(W_in - W_out)

```

```

C      Heat transfer between desiccant and air
Q_a = m_dot_a*(h_in - h_out)

C      Outlet desiccant flow rate
m_dot_LD_out = m_dot_LD_in + m_dot_abs

C      Heat transfer into cooling water
Q_cw = (m_dot_LD_in * C_p_d * T_d_in - m_dot_LD_out *
1      C_p_d * T_d_out) + Q_a

C      Cooling water outlet temp
T_cw_out = T_cw_in + (Q_cw / (m_dot_cw * C_p_cw))

C      Flow rate of the solution
m_dot_sd = xi_in*m_dot_LD_in

C      xi_out = (m_dot_sd/(m_dot_LD_out))

LD_ratio = m_dot_LD_out/(2*m_dot_LD_in)

ENDIF

C      PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE
NITEMS=0
C      STORED(1)=...

C      PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY
C      CALL setStorageVars(STORED,NITEMS,INFO)

C      RETURN TO THE CALLING PROGRAM
RETURN 1

ENDIF

C-----
C-----
C      *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C-----
C-----
C      RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION
C      NITEMS=
C      CALL getStorageVars(STORED,NITEMS,INFO)
C      STORED(1)=
C-----
C-----
C      CHECK THE INPUTS FOR PROBLEMS
C      IF(...) CALL TYPECK(-3,INFO,'BAD INPUT #',0,0)
C      IF(IERROR.GT.0) RETURN 1
C-----
C-----

```

```

C      *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
C-----
C ADD YOUR COMPONENT EQUATIONS HERE; BASICALLY THE EQUATIONS THAT WILL
C CALCULATE THE OUTPUTS BASED ON THE PARAMETERS AND THE INPUTS. REFER TO
C CHAPTER 3 OF THE TRNSYS VOLUME 1 MANUAL FOR DETAILED INFORMATION ON
C WRITING TRNSYS COMPONENTS.
C
C      Find max humidity ratio from Conde (2003)
C      Fitted constants for empirical relation
P0 = 0.31
P1 = 3.698
P2 = 0.60
P3 = 0.231
P4 = 4.584
P5 = 0.49
P6 = 0.478
P7 = -5.20
P8 = -0.40
P9 = 0.018
C
C      Critical Temp (K) and Pressure (kPa) of water
T_w_crit = 647.1
P_w_crit = 22064.
C
C      Equations from Conde
phi = (T_cw_in + 273.15)/T_w_crit
tau = 1. - phi
A = 2. - (1. + (xi_in/P0)**P1)**P2
B = (1. + (xi_in/P3)**P4)**P5 - 1.
F = A + B*phi
P_25 = 1. - (1. + (xi_in/P6)**P7)**P8 -
1      P9*exp(-(xi_in - 0.1)**2.)/0.005
C      Relative Vapour Pressure of CaCl (Relative to water at same temp)
P_v_rel = P_25*F
C      Find vapour Pressure of water over liquid phase (Conde - Appendix A)
C      Fitted constants from Conde - Appendix A
A0 = -7.858230
A1 = 1.839910
A2 = -11.781100
A3 = 22.670500
A4 = -15.939300
A5 = 1.775160
C
C      Equation from Conde
F_2 = (A0*tau + A1*tau**1.5 + A2*tau**3. +
1      A3*tau**3.5 + A4*tau**4. + A5*tau**7.5)/(1.-tau)

```

```

C      Water vapor pressure over liquid phase
P_v_w = exp(F_2)*P_w_crit

C      Desiccant vapor pressure
P_v_min = P_v_w*P_v_rel

C      Calc air properties of air at same vapor pressure as Ca-Cl solution
W_min = (0.6219*P_v_min)/(P_atm_k - P_v_min)

P_atm = P_atm_k/101.325

PSYDAT(1) = P_atm
PSYDAT(2) = T_in
PSYDAT(6) = W_min
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,4,0,PSYDAT,2,STATUS,*151)
CALL LINKCK('TYPE 974', 'PSYCHROMETRICS', 1,99)
151    CONTINUE
C      Min enthalpy
h_min = PSYDAT(7)

C      Calculate inlet air properties
PSYDAT(1) = P_atm
PSYDAT(2) = T_in
PSYDAT(4) = RH_in
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
CALL PSYCHROMETRICS(TIME,INFO,1,2,0,PSYDAT,2,STATUS,*251)
CALL LINKCK('TYPE 974', 'PSYCHROMETRICS', 1,99)
251    CONTINUE
C      Entering Air Enthalpy
W_in = PSYDAT(6)
h_in = PSYDAT(7)

IF((m_dot_a.EQ.0).OR.(m_dot_LD_in.EQ.0)) THEN
    T_out = T_in
    W_out = W_in
h_out = h_in
    RH_out = RH_in
    T_d_out = T_d_in
    m_dot_abs = 0.0
    m_dot_sd = xi_in*m_dot_LD_in
    m_dot_LD_out = m_dot_LD_in
    xi_out = xi_in
    LD_ratio = 0.0
    T_cw_out=T_cw_in
    Q_a = 0.0
    Q_cw = 0.0
    ELSE

C      Calculate outlet air properties
W_out = W_in - E_abs*(W_in - W_min)

```

```

h_out = h_in - E_h*(h_in - h_min)

PSYDAT(1) = P_atm
    PSYDAT(6) = W_out
    PSYDAT(7) = h_out
C      DO NOT PRINT ERROR WARNINGS ON THE FIRST CALL
      CALL PSYCHROMETRICS(TIME,INFO,1,6,0,PSYDAT,2,STATUS,*351)
      CALL LINKCK('TYPE 974','PSYCHROMETRICS',1,99)
351   CONTINUE
C      Leaving Air Temperature
      T_out = PSYDAT(2)
      RH_out = PSYDAT(4)

C      Desiccant outlet temp
      T_d_out = T_d_in - E_cw*(T_d_in - T_cw_in)

C      Rate of desorbed moisture
      m_dot_abs = m_dot_a*(W_in - W_out)

C      Heat transfer between desiccant and air
      Q_a = m_dot_a*(h_in - h_out)

C      Outlet desiccant flow rate
      m_dot_LD_out = m_dot_LD_in + m_dot_abs

C      Heat transfer into cooling water
      Q_cw = (m_dot_LD_in * C_p_d * T_d_in - m_dot_LD_out *
1       C_p_d * T_d_out) + Q_a

C      Cooling water outlet temp
      T_cw_out = T_cw_in + (Q_cw / (m_dot_cw * C_p_cw))

C      Flow rate of the solution
      m_dot_sd = xi_in*m_dot_LD_in

C      xi_out = (m_dot_sd/(m_dot_LD_out))

      LD_ratio = m_dot_LD_out/(2*m_dot_LD_in)

      ENDIF
C-----
C-----
C-----
C      SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
C      NITEMS=
C      STORED(1)=
C      CALL setStorageVars(STORED,NITEMS,INFO)
C-----
C-----

```

```

C      REPORT ANY PROBLEMS THAT HAVE BEEN FOUND USING CALLS LIKE THIS:
C      CALL MESSAGES(-1,'put your message here','MESSAGE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','WARNING',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','SEVERE',IUNIT,ITYPE)
C      CALL MESSAGES(-1,'put your message here','FATAL',IUNIT,ITYPE)
C-----
C-----
C      SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT
C      Set output variables

OUT(1)=T_out
OUT(2)=W_out
OUT(3) = h_out
OUT(4)= RH_out
OUT(5)= m_dot_a

OUT(6)=T_d_out
OUT(7)=m_dot_abs
OUT(8)=m_dot_sd
OUT(9)= m_dot_LD_out
OUT(10)=xi_out
OUT(11)=LD_ratio

OUT(12) = T_cw_out
OUT(13) = m_dot_cw

OUT(14)= P_v_min

OUT(15) = W_min
OUT(16)= h_min
OUT(17)= Q_a
OUT(18)= Q_cw

C-----
C      EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
C      RETURN 1
C      END
C-----

```

C.3 Type 991 - Regenerator Controller

```

SUBROUTINE TYPE991 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICtrl,*)
C*****
C Object: PVT-LD System Controller
C Simulation Studio Model: Type991
C
C Description
C The operational logic of this model was to drive the array fan

```

```

C at variable speed such that the array outlet temperature was maintained at
C   a desired setpoint for liquid desiccant regeneration. The pumps and dampers for the
C   liquid desiccant system are also controlled based on the actual outlet temperature achieved.
C
C LAST MODIFIED:
C   AUGUST 2009 - ROSS CASEY -ORIGINAL CODING
C   NOVEMBER 2011 - MICHAEL HATT - Using same modulation control, but modified for specific system
C*****

C TRNSYS access functions (allow to access TIME etc.)
C   USE TrnsysConstants
C   USE TrnsysFunctions

C-----
C   REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
C   !DEC$ATTRIBUTES DLLEXPORT :: TYPE991 !SET THE CORRECT TYPE NUMBER HERE
C-----
C-----
C TRNSYS DECLARATIONS
C   IMPLICIT NONE !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM

DOUBLE PRECISION XIN !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION OUT !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET IT!
DOUBLE PRECISION PAR !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP
DOUBLE PRECISION T !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION SOLVER
DOUBLE PRECISION DTDI !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
INTEGER*4 INFO(15) !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND FROM THIS TYPE
INTEGER*4 NP,NI,NOUT,ND !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
INTEGER*4 NPAR,NIN,NDER !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
INTEGER*4 ICtrl !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW SOLVER
INTEGER*4 NSTORED !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF STORAGE
CHARACTER*3 OCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE OUTPUTS
CHARACTER*3 YCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE INPUTS
C-----
C-----
C USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
C   PARAMETER (NP=14,NI=6,NOUT=11,ND=0,NSTORED=0)
C-----
C-----
C REQUIRED TRNSYS DIMENSIONS
C   DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
1 STORED(NSTORED),T(ND),DTDI(ND)
C   INTEGER NITEMS
C-----
C-----

```

```

C   ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE

C   PARAMETERS
      DOUBLE PRECISION
      1 Mode,FanSizeCFM,RegMinCFM,RegMaxCFM,TDiff,
      1 X,Band,SampleTime,TankHLimit,TankLLimit,PumpTime,Tminr,Tmaxr,
      1 Tmaxabs,Tset

C   INPUTS
      DOUBLE PRECISION
      1 Tplate,Tamb,Tout,PVOn,TankC

C   LOCAL AND OUTPUT VARIABLES
      DOUBLE PRECISION
      1 CtrlMax,CtrlMin,TimerPrev,TimerNew,Ctrl,CtrlPrev,JustOn,
      1 RegCtrlPrev,RegCtrlState,CtrlOther,
      1 Storage(7)
      INTEGER
      1 PumpsPrev,DamperFreshPrev,DamperBypprev,DamperBypprev,DamperFresh,
      1 RegPumpsOn

C-----
C   ERROR MESSAGES
CHARACTER(LEN=MAXMESSAGELENGTH)::MESSAGE1
      MESSAGE1='The relative humidity of the array outlet air is greater
      1 than 100 percent (during night time TimerPrev). Therefore Type 982 and 961
      1 are overpredicting the TimerPrev performance due to their lack of
      1 moist air calculations.'
C-----

C-----
C   READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER
      Mode=PAR(1)
      FanSizeCFM=PAR(2)
RegMinCFM=PAR(3)
      RegMaxCFM=PAR(4)
      Tminr=PAR(5)
      Tmaxr=PAR(6)
      Tmaxabs=PAR(7)
      TDiff=PAR(8)
      X=PAR(9)
      Band=PAR(10)
      SampleTime=PAR(11)
      PumpTime=PAR(12)
      TankHLimit=PAR(13)
      TankLLimit=PAR(14)

C-----
C   RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER

      Tplate=XIN(1)

```



```

    Tamb=XIN(2)
    Tout=XIN(3)
    Tset=XIN(4)
    PVOn=XIN(5)
    TankC=XIN(6)
IUNIT=INFO(1)
ITYPE=INFO(2)

C-----
C  SET THE VERSION INFORMATION FOR TRNSYS
    IF (INFO(7).EQ.-2) THEN
        INFO(12)=16
        RETURN 1
    ENDIF
C-----

C-----
C  DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
    IF (INFO(8).EQ.-1) THEN
        RETURN 1
    ENDIF
C-----

C-----
C  PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C  e.g. save variables to storage array for the next timestep
    IF (INFO(13).GT.0) THEN
        NITEMS=7
        C IF (Cool.EQ.1.0) THEN
            C Cool=0.0
        C IF (RHarray.GE.100.0) THEN
            C CALL MESSAGES(-1,MESSAGE1,'WARNING',IUNIT,ITYPE)
            C Storage(1)=Storage(1)+1
        C ENDIF
        C ENDIF

        Storage(1)=JustOn
        Storage(2)=RegCtrlState
        Storage(3)=TimerNew
        Storage(4)=Ctrl
        Storage(5)=RegPumpsOn
        Storage(6)=DamperFresh
        Storage(7)=DamperByp
        CALL setStorageVars(Storage,NITEMS,INFO)
        RETURN 1
    ENDIF
C
C-----

C-----
C  DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
    IF (INFO(7).EQ.-1) THEN

```

```

C      SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
      INFO(6)=NOUT
      INFO(9)=1
      INFO(10)=0 !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C      SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN THE INPUT FILE
C      IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS MODEL...
      NIN=NI
      NPAR=NP
      NDER=ND

C      CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C      THE TRNSYS INPUT FILE
      CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C      SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
      NITEMS=7
      CALL setStorageSize(NITEMS,INFO)

C      RETURN TO THE CALLING PROGRAM
      RETURN 1

      ENDIF
C-----
C-----
C      DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INTIAL TIME
      IF (TIME .LT. (getSimulationStartTime() +
. getSimulationTimeStep()/2.D0)) THEN

C      SET THE UNIT NUMBER FOR FUTURE CALLS
      IUNIT=INFO(1)
      ITYPE=INFO(2)

C      PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE
      OUT(1)=0.
      OUT(2)=0.
      OUT(3)=TIME
      OUT(4)=0.
      OUT(5)=0.

C      PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE
      NITEMS=7
      Storage(1)=0.
      Storage(2)=0.
      Storage(3)=TIME
      Storage(4)=0.
      Storage(5)=0.
      Storage(6)=0.

```

```

Storage(7)=0.
JustOn=Storage(1)
RegCtrlPrev=Storage(2)
TimerPrev=Storage(3)
CtrlPrev=Storage(4)
PumpsPrev=Storage(5)
DamperFreshPrev=Storage(6)
DamperBybPrev=Storage(7)

C      PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY
      CALL setStorageVars(Storage,NITEMS,INFO)

C      RETURN TO THE CALLING PROGRAM
      RETURN 1

ENDIF

C-----

C-----

C      *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***
C-----

C-----

C      RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION
      NITEMS=7
      CALL getStorageVars(Storage,NITEMS,INFO)
      JustOn=Storage(1)
      RegCtrlPrev=Storage(2)
      TimerPrev=Storage(3)
      CtrlPrev=Storage(4)
      PumpsPrev=Storage(5)
      DamperFreshPrev=Storage(6)
      DamperBybPrev=Storage(7)

C-----

C-----

C      *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***
C-----

      IF (Mode.EQ.0) THEN
          CtrlMax=MIN(1.0,MAX(0.0,RegMaxCFM/FanSizeCFM))
          CtrlMin=MAX(0.0,MIN(1.0,RegMinCFM/FanSizeCFM))
C      CONTROLLER PREVIOUSLY OFF
      IF (PVOn.GT.0) THEN
          IF (RegCtrlPrev.EQ.0.) THEN
              IF (TIME.GE.(TimerPrev+SampleTime)) THEN ! TURN ON
                  RegCtrlState=1.
                  TimerNew=TIME
              ELSE
                  ! KEEP OFF
                  RegCtrlState=0.
          ENDIF
      ENDIF

```

```

TimerNew=TimerPrev
ENDIF
C CONTROLLER PREVIOUSLY ON
ELSEIF (RegCtrlPrev.EQ.1) THEN
    RegCtrlState=2.
    TimerNew=TimerPrev
ELSEIF (RegCtrlPrev.EQ.2) THEN
IF (Tplate.GE.Tminr) THEN ! KEEP ON
    RegCtrlState=3.
TimerNew=TIME
ELSE ! TURN OFF
RegCtrlState=0.
TimerNew=TIME
ENDIF
ELSEIF (RegCtrlPrev.EQ.3.) THEN
    IF (TIME.GE.(TimerPrev+PumpTime)) THEN ! PUMP TIMER EXPIRED
        RegCtrlState=4.
        TimerNew=TIME
    ELSE
        ! CHECK PUMP TIMER
        RegCtrlState=3.
        TimerNew=TimerPrev
    ENDIF
ELSEIF (RegCtrlPrev.EQ.4.) THEN
    IF ((Tplate.GE.Tminr).AND.(Tplate.LT.Tmaxabs)) THEN ! KEEP ON
        RegCtrlState=4.
        TimerNew=TIME
        !ELSEIF (Tplate.GE.Tmaxabs) THEN ! TURN OFF-TOO HIGH
!   RegCtrlState=0.
!   TimerNew=TIME
    ELSE ! TURN OFF
        RegCtrlState=0.
        TimerNew=TIME
    ENDIF
ENDIF
ELSE
    RegCtrlState=0.
    TimerNew=TIME
ENDIF

C ITERATIVE CONTROL
IF(RegCtrlState.EQ.1.) THEN
    Ctrl=CtrlMin
ELSEIF(RegCtrlState.EQ.2.)THEN
    Ctrl=CtrlMin
ELSEIF(RegCtrlState.EQ.3.)THEN
    Ctrl=CtrlMin
ELSEIF(RegCtrlState.EQ.4.) THEN
    IF(Tout.GT.Tset) THEN
    IF (DABS(Tout-Tset) .LT. Band) THEN
        Ctrl=Ctrl
    ELSE

```

```

    Ctrl=MIN(CtrlMax,Ctrl+X)
ENDIF
ELSEIF(Tout.LT.Tset) THEN
IF (DABS(Tout-Tset) .LT. Band) THEN
    Ctrl=Ctrl
ELSE
    Ctrl=MAX(CtrlMin,Ctrl-X)
ENDIF
ELSE
    Ctrl=Ctrl
ENDIF
ELSE
    Ctrl=0.
ENDIF
C Pump Control
    IF(PumpsPrev.EQ.0.)THEN
        IF((RegCtrlState.GE.3.) .AND. (TankC.LE. (TankLLimit-0.005))) THEN
            RegPumpsOn=1.
        ELSE
            RegPumpsOn=PumpsPrev
        ENDIF
ELSEIF(PumpsPrev.EQ.1.) THEN
    IF((RegCtrlState.NE.3) .AND. (RegCtrlState.NE.4))THEN
        RegPumpsOn=0.
    ELSEIF(TankC.GT. TankLLimit)THEN
        RegPumpsOn=0.
    ELSE
        RegPumpsOn=PumpsPrev
    ENDIF
ELSE
    RegPumpsOn=PumpsPrev
ENDIF

C Fresh Air Damper Control
    IF(DamperFreshPrev.EQ.0)THEN
        IF(RegCtrlState.EQ.4.)THEN
            IF((Ctrl.GE.CtrlMax) .AND. (Tout.GT. Tmaxr))THEN
                DamperFresh = 1.
            ELSE
                DamperFresh=DamperFreshPrev
            ENDIF
        ELSE
            DamperFresh=DamperFreshPrev
        ENDIF
    ELSEIF(DamperFreshPrev.EQ.1)THEN
        IF(Tout.LE.(Tmaxr-10.0))THEN
            DamperFresh=0.
        ELSE
            DamperFresh=DamperFreshPrev
        ENDIF
    ELSE

```

```

        DamperFresh=DamperFreshPrev
    ENDIF
C Bypass Damper Control
    IF(DamperBypPrev.EQ.0)THEN
        IF(RegCtrlState.EQ.4.)THEN
            IF((Ctrl.GE.CtrlMax).AND.(Tout.GT.Tmaxabs))THEN
                DamperByp = 1.
            ELSE
                DamperByp=DamperBypPrev
            ENDIF
        ELSEIF(RegCtrlState.EQ.3.)THEN
            DamperByp=1.
        ELSE
            DamperByp=DamperBypPrev
        ENDIF
    ELSEIF(DamperBypPrev.EQ.1)THEN
        IF(RegCtrlState.EQ.4.)THEN
            IF((Ctrl.GE.CtrlMax).AND.(Tout.GT.Tmaxabs))THEN
                DamperByp = 1.
            ELSE
                DamperByp=DamperBypPrev
            ENDIF
        ELSEIF(RegCtrlState.EQ.3.)THEN
            DamperByp = 1.
        ELSE
            DamperByp = 0.
        ENDIF
    ELSE
        DamperByp=DamperBypPrev
    ENDIF

    ELSEIF(Mode.EQ.1)THEN

        CtrlMax=MIN(1.0,MAX(0.0,RegMaxCFM/FanSizeCFM))
        CtrlMin=MAX(0.0,MIN(1.0,RegMinCFM/FanSizeCFM))
C CONTROLLER PREVIOUSLY OFF
    IF (PVOn.GT.0) THEN
        IF (RegCtrlPrev.EQ.0.) THEN
            RegCtrlState=1.
            TimerNew=TIME                ! KEEP OFF
C CONTROLLER PREVIOUSLY ON
        ELSEIF (RegCtrlPrev.EQ.1) THEN
            IF(TankC.LE.(TankLLimit-0.005)) THEN
                IF ((Tplate.GE.(Tset+TDiff)).AND.(Tplate.LT.Tmaxabs)) THEN ! KEEP ON
                    RegCtrlState=2.
                    TimerNew=TIME
                ELSE ! TURN OFF
                    RegCtrlState=1.
                    TimerNew=TIME
                ENDIF
            ELSE

```

```

    RegCtrlState=1.
    TimerNew=TIME
ENDIF
ELSEIF (RegCtrlPrev.EQ.2) THEN
    IF(TankC.LE.TankLLimit) THEN
    IF ((Tplate.GE.Tminr).AND.(Tplate.LT.Tmaxabs)) THEN
        RegCtrlState=3.
        TimerNew=TIME
    ELSE ! TURN OFF
        RegCtrlState=1.
        TimerNew=TIME
    ENDIF
ELSE
    RegCtrlState=1.
    TimerNew=TIME
ENDIF
ELSEIF (RegCtrlPrev.EQ.3.) THEN
    IF(TankC.LE.TankLLimit) THEN
    IF ((Tplate.GE.Tminr).AND.(Tplate.LT.Tmaxabs)) THEN
        RegCtrlState=3.
        TimerNew=TIME
    ELSE ! TURN OFF
        RegCtrlState=1.
        TimerNew=TIME
    ENDIF
ELSE
    RegCtrlState=1.
    TimerNew=TIME
ENDIF
ENDIF
ELSE
    RegCtrlState=0.
    TimerNew=TIME
ENDIF
ENDIF

C ITERATIVE CONTROL
    IF(RegCtrlState.EQ.2.) THEN
        Ctrl=CtrlMin
    ELSEIF(RegCtrlState.EQ.3.) THEN
        IF(Tout.GT.Tset) THEN
        IF (DABS(Tout-Tset) .LT. Band) THEN
            Ctrl=Ctrl
        ELSE
            Ctrl=MIN(CtrlMax,Ctrl+X)
        ENDIF
    ELSEIF(Tout.LT.Tset) THEN
        IF (DABS(Tout-Tset) .LT. Band) THEN
            Ctrl=Ctrl
        ELSE
            Ctrl=MAX(CtrlMin,Ctrl-X)
        ENDIF
    ENDIF

```

```

ELSE
Ctrl=Ctrl
ENDIF
ELSE
Ctrl=0.
ENDIF

C Pump Control
IF (PumpsPrev.EQ.0.) THEN
IF (RegCtrlState.GE.2.) THEN
RegPumpsOn=1.
ELSE
RegPumpsOn=0.
ENDIF
ELSEIF (PumpsPrev.EQ.1.) THEN
IF (RegCtrlState.GE.2.) THEN
RegPumpsOn=1.
ELSE
RegPumpsOn=0.
ENDIF
ELSE
RegPumpsOn=PumpsPrev
ENDIF

C Fresh Air Damper Control

DamperFresh=0.

C Bypass Damper Control

DamperByb = 1.

C -----Simplified Controls-----

ELSE
CtrlMax=MIN(1.0,MAX(0.0,RegMaxCFM/FanSizeCFM))
CtrlMin=MAX(0.0,MIN(1.0,RegMinCFM/FanSizeCFM))
CtrlOther=300/FanSizeCFM
C CONTROLLER PREVIOUSLY OFF
IF (PVOn.GT.0) THEN
IF (RegCtrlPrev.EQ.0.) THEN
RegCtrlState=1.
TimerNew=TIME ! KEEP OFF
C CONTROLLER PREVIOUSLY ON
ELSEIF (RegCtrlPrev.EQ.1) THEN
C IF (TankC.LE.(TankLLimit-0.005)) THEN
IF ((Tplate.GE.(Tset+TDiff)).AND.(Tplate.LT.Tmaxabs)) THEN ! KEEP ON
RegCtrlState=2.
TimerNew=TIME
ELSE ! TURN OFF
RegCtrlState=1.

```



```

    TimerNew=TIME
ENDIF
C ELSE
C   RegCtrlState=1.
C   TimerNew=TIME
C ENDIF
ELSEIF (RegCtrlPrev.EQ.2) THEN
C   IF(TankC.LE.TankLLimit) THEN
   IF ((Tplate.GE.Tminr).AND.(Tplate.LT.Tmaxabs)) THEN
       RegCtrlState=3.
       TimerNew=TIME
   ELSE ! TURN OFF
       RegCtrlState=1.
       TimerNew=TIME
   ENDIF
C ELSE
C   RegCtrlState=1.
C   TimerNew=TIME
C ENDIF
ELSEIF (RegCtrlPrev.EQ.3.) THEN
C   IF(TankC.LE.TankLLimit) THEN
   IF ((Tplate.GE.Tminr).AND.(Tplate.LT.Tmaxabs)) THEN
       RegCtrlState=3.
       TimerNew=TIME
   ELSE ! TURN OFF
       RegCtrlState=1.
       TimerNew=TIME
   ENDIF
C ELSE
C   RegCtrlState=1.
C   TimerNew=TIME
C ENDIF
ENDIF
ELSE
   RegCtrlState=0.
   TimerNew=TIME
ENDIF

C ITERATIVE CONTROL
   IF(RegCtrlState.EQ.2.) THEN
       Ctrl=CtrlMin
   ELSEIF(RegCtrlState.EQ.3.) THEN
       IF(Tout.GT.Tset) THEN
           IF (DABS(Tout-Tset) .LT. Band) THEN
               Ctrl=Ctrl
           ELSE
               Ctrl=MIN(CtrlMax,Ctrl+X)
           ENDIF
       ELSEIF(Tout.LT.Tset) THEN
           IF (DABS(Tout-Tset) .LT. Band) THEN
               Ctrl=Ctrl
           
```

```

ELSE
    Ctrl=MAX(CtrlMin,Ctrl-X)
ENDIF
ELSE
    Ctrl=Ctrl
ENDIF
ELSEIF(RegCtrlState.EQ.1.) THEN
    Ctrl=CtrlMin
ELSE
    Ctrl=0.
ENDIF

C Pump Control
IF(PumpsPrev.EQ.0.)THEN
    IF(RegCtrlState.GE.2.) THEN
        IF(TankC.LE.(TankLLimit-0.005)) THEN
            RegPumpsOn=1.
        ELSE
            RegPumpsOn=0.
        ENDIF
    ELSE
        RegPumpsOn=0.
    ENDIF
ENDIF
ELSEIF(PumpsPrev.EQ.1.) THEN
    IF(RegCtrlState.GE.2.) THEN
        IF(TankC.LE.TankLLimit) THEN
            RegPumpsOn=1.
        ELSE
            RegPumpsOn=0.
        ENDIF
    ELSE
        RegPumpsOn=0.
    ENDIF
ENDIF
ENDIF

C Fresh Air Damper Control
IF(DamperFreshPrev.EQ.0) THEN
    IF(RegCtrlState.EQ.3.)THEN
        IF((Ctrl.GE.CtrlMax).AND.(Tout.GT.Tmaxabs))THEN
            DamperFresh = 1.
        ELSE
            DamperFresh=DamperFreshPrev
        ENDIF
    ELSE
        DamperFresh=DamperFreshPrev
    ENDIF
ENDIF
ELSEIF(DamperFreshPrev.EQ.1)THEN
    IF(Tout.LE.(Tmaxabs-10.0))THEN
        DamperFresh=0.
    ELSE

```

```

        DamperFresh=DamperFreshPrev
    ENDF
ELSE
    DamperFresh=DamperFreshPrev
ENDF

C Bypass Damper Control
C   IF(RegCtrlState.GE.2.)THEN
C   DamperByp=1.
C ELSE
C   DamperByp=0.
C ENDF
    IF(DamperBypPrev.EQ.0)THEN
        IF(TankC.LE.(TankLLimit-0.005)) THEN
            DamperByp = 1.
        ELSE
            DamperByp=0.
        ENDF
    ELSEIF(DamperBypPrev.EQ.1)THEN
        IF(TankC.LE.TankLLimit) THEN
            DamperByp = 1.
        ELSE
            DamperByp=0.
        ENDF
    ELSE
        DamperFresh=DamperFreshPrev
    ENDF

    ENDF

C-----
C SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
Storage(1)=JustOn
Storage(2)=RegCtrlState
Storage(3)=TimerNew
Storage(4)=Ctrl
Storage(5)=RegPumpsOn
    Storage(6)=DamperFresh
Storage(7)=DamperByp
CALL SetStorageVars(Storage,NITEMS,INFO)

C-----
C   SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT

C Control Signal
OUT(1)=Ctrl
OUT(2)=TimerNew

```

```

OUT(3)=TimerPrev
OUT(4)=RegCtrlState
OUT(5)=RegCtrlPrev
OUT(6)=RegPumpsOn
OUT(7)=PumpsPrev
OUT(8)=DamperFresh
OUT(9)=DamperFreshPrev
OUT(10)=DamperByp
OUT(11)=DamperBypPrev

```

```

C-----
C  EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON
      RETURN 1
      END SUBROUTINE Type991

```

C.4 Type 992 - Absorber Controller

```

SUBROUTINE TYPE992 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICtrl,*)
C*****
C Object: Absorber System Controller
C Simulation Studio Model: Type992
C
C The operational logic of this model was to drive the absorber fan, cooling tower fan,
C and associated pumps at constant speed, and only operate under specific conditions.
C
C LAST MODIFIED:
C FEBRUARY 2012 - MICHAEL HATT - Using same modulation control as TYPE991, but modified for specific system
C*****

C TRNSYS access functions (allow to access TIME etc.)
      USE TrnsysConstants
      USE TrnsysFunctions

C-----
C  REQUIRED BY THE MULTI-DLL VERSION OF TRNSYS
      !DEC$ATTRIBUTES DLLEXPORT :: TYPE992 !SET THE CORRECT TYPE NUMBER HERE
C-----
C-----
C TRNSYS DECLARATIONS
      IMPLICIT NONE !REQUIRES THE USER TO DEFINE ALL VARIABLES BEFORE USING THEM

DOUBLE PRECISION XIN !THE ARRAY FROM WHICH THE INPUTS TO THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION OUT !THE ARRAY WHICH WILL BE USED TO STORE THE OUTPUTS FROM THIS TYPE
DOUBLE PRECISION TIME !THE CURRENT SIMULATION TIME - YOU MAY USE THIS VARIABLE BUT DO NOT SET IT!
DOUBLE PRECISION PAR !THE ARRAY FROM WHICH THE PARAMETERS FOR THIS TYPE WILL BE RETRIEVED
DOUBLE PRECISION STORED !THE STORAGE ARRAY FOR HOLDING VARIABLES FROM TIMESTEP TO TIMESTEP
DOUBLE PRECISION T !AN ARRAY CONTAINING THE RESULTS FROM THE DIFFERENTIAL EQUATION SOLVER
DOUBLE PRECISION DTDT !AN ARRAY CONTAINING THE DERIVATIVES TO BE PASSED TO THE DIFF.EQ. SOLVER
INTEGER*4 INFO(15) !THE INFO ARRAY STORES AND PASSES VALUABLE INFORMATION TO AND FROM THIS TYPE
INTEGER*4 NP,NI,NOUT,ND !VARIABLES FOR THE MAXIMUM NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES
INTEGER*4 NPAR,NIN,NDER !VARIABLES FOR THE CORRECT NUMBER OF PARAMETERS,INPUTS,OUTPUTS AND DERIVATIVES

```

```

INTEGER*4 IUNIT,ITYPE !THE UNIT NUMBER AND TYPE NUMBER FOR THIS COMPONENT
INTEGER*4 ICtrl !AN ARRAY FOR HOLDING VALUES OF CONTROL FUNCTIONS WITH THE NEW SOLVER
INTEGER*4 NSTORED !THE NUMBER OF VARIABLES THAT WILL BE PASSED INTO AND OUT OF STORAGE
CHARACTER*3 OCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE OUTPUTS
CHARACTER*3 YCHECK !AN ARRAY TO BE FILLED WITH THE CORRECT VARIABLE TYPES FOR THE INPUTS

```

C-----

C-----

```

C USER DECLARATIONS - SET THE MAXIMUM NUMBER OF PARAMETERS (NP), INPUTS (NI),
C OUTPUTS (NOUT), AND DERIVATIVES (ND) THAT MAY BE SUPPLIED FOR THIS TYPE
  PARAMETER (NP=10,NI=5,NOUT=11,ND=0,NSTORED=0)

```

C-----

C-----

```

C REQUIRED TRNSYS DIMENSIONS
  DIMENSION XIN(NI),OUT(NOUT),PAR(NP),YCHECK(NI),OCHECK(NOUT),
1 STORED(NSTORED),T(ND),DTDT(ND)
  INTEGER NITEMS

```

C-----

C-----

```

C ADD DECLARATIONS AND DEFINITIONS FOR THE USER-VARIABLES HERE

```

```

C PARAMETERS

```

```

  DOUBLE PRECISION

```

```

1 FanSizeCFM,AbsMinCFM,AbsMaxCFM,

```

```

1 SampleTime,TankHLimit,TankLLimit, HystEn, TankHyst, X

```

```

C INPUTS

```

```

  DOUBLE PRECISION

```

```

1 AbsOn,AbsCFM,TankC, RHset, RHout, Band

```

```

C LOCAL AND OUTPUT VARIABLES

```

```

  DOUBLE PRECISION

```

```

1 CtrlMax,CtrlMin,TimerPrev,TimerNew,Ctrl,CtrlPrev,JustOn,

```

```

1 AbsCtrlPrev,AbsCtrlState, AtLimit,

```

```

  1 Storage(7)

```

```

  INTEGER

```

```

1 PumpsPrev,DamperFreshPrev,DamperBypPrev,DamperByp,DamperFresh,

```

```

1 AbsPumpsOn

```

C-----

```

C ERROR MESSAGES

```

```

CHARACTER(LEN=MAXMESSAGELENGTH)::MESSAGE1

```

```

  MESSAGE1='The relative humidity of the array outlet air is greater

```

```

1 than 100% (during night time TimerPrev). Therefore Type 982 and 961

```

```

1 are overpredicting the TimerPrev performance due to their lack of

```

```

1 moist air calculations.'

```

C-----

C-----

```

C READ IN THE VALUES OF THE PARAMETERS IN SEQUENTIAL ORDER

```

```

      FanSizeCFM=PAR(1)
AbsMinCFM=PAR(2)
      AbsMaxCFM=PAR(3)
      SampleTime=PAR(4)
      TankHLimit=PAR(5)
      TankLLimit=PAR(6)
      HystEn=PAR(7)
      TankHyst=PAR(8)
      X=PAR(9)
      Band=PAR(10)

```

```

C-----
C  RETRIEVE THE CURRENT VALUES OF THE INPUTS TO THIS MODEL FROM THE XIN ARRAY IN SEQUENTIAL ORDER
      AbsOn=XIN(1)
      AbsCFM=XIN(2)
      TankC=XIN(3)
      RHset=XIN(4)
      RHout=XIN(5)
      IUNIT=INFO(1)
      ITYPE=INFO(2)

```

```

C-----
C  SET THE VERSION INFORMATION FOR TRNSYS
      IF (INFO(7).EQ.-2) THEN
        INFO(12)=16
        RETURN 1
      ENDIF

```

```

C-----
C  DO ALL THE VERY LAST CALL OF THE SIMULATION MANIPULATIONS HERE
      IF (INFO(8).EQ.-1) THEN
        RETURN 1
      ENDIF

```

```

C-----
C  PERFORM ANY 'AFTER-ITERATION' MANIPULATIONS THAT ARE REQUIRED HERE
C  e.g. save variables to storage array for the next timestep
      IF (INFO(13).GT.0) THEN
        NITEMS=7
        C IF(Cool.EQ.1.0) THEN
        C Cool=0.0
        C IF(RHarray.GE.100.0) THEN
        C CALL MESSAGES(-1,MESSAGE1,'WARNING',IUNIT,ITYPE)
        C Storage(1)=Storage(1)+1
        C ENDIF
        C ENDIF
          Storage(1)=JustOn
        Storage(2)=AbsCtrlState
        Storage(3)=TimerNew

```

```

Storage(4)=Ctrl
Storage(5)=AbsPumpsOn
Storage(6)=DamperFresh
Storage(7)=DamperByp
CALL setStorageVars(Storage,NITEMS,INFO)
    RETURN 1
ENDIF
C
C-----
C-----
C DO ALL THE VERY FIRST CALL OF THE SIMULATION MANIPULATIONS HERE
    IF (INFO(7).EQ.-1) THEN

C     SET SOME INFO ARRAY VARIABLES TO TELL THE TRNSYS ENGINE HOW THIS TYPE IS TO WORK
        INFO(6)=NOUT
        INFO(9)=1
        INFO(10)=0 !STORAGE FOR VERSION 16 HAS BEEN CHANGED

C     SET THE REQUIRED NUMBER OF INPUTS, PARAMETERS AND DERIVATIVES THAT THE USER SHOULD SUPPLY IN THE INPUT FILE
C     IN SOME CASES, THE NUMBER OF VARIABLES MAY DEPEND ON THE VALUE OF PARAMETERS TO THIS MODEL...
        NIN=NI
        NPAR=NP
        NDER=ND

C     CALL THE TYPE CHECK SUBROUTINE TO COMPARE WHAT THIS COMPONENT REQUIRES TO WHAT IS SUPPLIED IN
C     THE TRNSYS INPUT FILE
        CALL TYPECK(1,INFO,NIN,NPAR,NDER)

C     SET THE NUMBER OF STORAGE SPOTS NEEDED FOR THIS COMPONENT
        NITEMS=7
        CALL setStorageSize(NITEMS,INFO)

C     RETURN TO THE CALLING PROGRAM
        RETURN 1

    ENDIF
C-----
C-----
C DO ALL OF THE INITIAL TIMESTEP MANIPULATIONS HERE - THERE ARE NO ITERATIONS AT THE INITIAL TIME
    IF (TIME .LT. (getSimulationStartTime() +
        . getSimulationTimeStep()/2.DO)) THEN

C     SET THE UNIT NUMBER FOR FUTURE CALLS
        IUNIT=INFO(1)
        ITYPE=INFO(2)

C     PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL VALUES OF THE OUTPUTS HERE
        OUT(1)=0.

```

OUT(2)=0.

OUT(3)=TIME

OUT(4)=0.

OUT(5)=0.

C PERFORM ANY REQUIRED CALCULATIONS TO SET THE INITIAL STORAGE VARIABLES HERE

 NITEMS=7

 Storage(1)=0.

 Storage(2)=0.

 Storage(3)=TIME

 Storage(4)=0.

 Storage(5)=0.

 Storage(6)=0.

 Storage(7)=0.

 JustOn=Storage(1)

 AbsCtrlPrev=Storage(2)

 TimerPrev=Storage(3)

 CtrlPrev=Storage(4)

 PumpsPrev=Storage(5)

 DamperFreshPrev=Storage(6)

 DamperByPrev=Storage(7)

C PUT THE STORED ARRAY IN THE GLOBAL STORED ARRAY

 CALL setStorageVars(Storage,NITEMS,INFO)

C RETURN TO THE CALLING PROGRAM

 RETURN 1

 ENDIF

C-----

C-----

C *** ITS AN ITERATIVE CALL TO THIS COMPONENT ***

C-----

C-----

C RETRIEVE THE VALUES IN THE STORAGE ARRAY FOR THIS ITERATION

 NITEMS=7

 CALL getStorageVars(Storage,NITEMS,INFO)

 JustOn=Storage(1)

 AbsCtrlPrev=Storage(2)

 TimerPrev=Storage(3)

 CtrlPrev=Storage(4)

 PumpsPrev=Storage(5)

 DamperFreshPrev=Storage(6)

 DamperByPrev=Storage(7)

C-----

C-----

C *** PERFORM ALL THE CALCULATION HERE FOR THIS MODEL. ***

C-----

```

CtrlMax=MIN(1.0,MAX(0.0,AbsMaxCFM/FanSizeCFM))
CtrlMin=MAX(0.0,MIN(1.0,AbsMinCFM/FanSizeCFM))

```

C CONTROLLER PREVIOUSLY OFF

IF (AbsOn.GT.0) THEN

IF (AbsCtrlPrev.EQ.0.) THEN

IF (RHset.GE.4.44) THEN

AbsCtrlState=2.

TimerNew=TimerPrev

AtLimit = 0.

ELSE

AbsCtrlState=1.

TimerNew=TIME

AtLimit = 0.

ENDIF

C CONTROLLER PREVIOUSLY ON

ELSEIF (AbsCtrlPrev.EQ.1) THEN

IF (RHset.GE.5.44) THEN

IF (HystEn.EQ.1) THEN

IF (TankC.LE.(TankHLimit+TankHyst)) THEN

AbsCtrlState=1.

AtLimit = 1.

ELSEIF (TankC.GT.(TankHLimit+TankHyst)) THEN

AbsCtrlState=2.

AtLimit = 0.

ELSE

AbsCtrlState=1.

AtLimit = 0.

ENDIF

ELSE

IF (TIME.GE.(TimerPrev+SampleTime)) THEN ! TURN ON

AbsCtrlState=2.

TimerNew=TIME

AtLimit = 0.

ELSE ! KEEP OFF

AbsCtrlState=1.

TimerNew=TimerPrev

AtLimit=1.

ENDIF

ENDIF

ELSE

AbsCtrlState=1.

AtLimit=0.

ENDIF

ELSEIF (AbsCtrlPrev.EQ.2) THEN

```

    IF (RHset.GE.4.44) THEN
IF (TankC.LE.TankHLimit) THEN !
    AbsCtrlState=1.
TimerNew=TIME
AtLimit = 1.
ELSE ! TURN OFF
AbsCtrlState=3.
TimerNew=TIME
AtLimit = 0.
ENDIF
ELSE
    AbsCtrlState=1.
    TimerNew=TIME
    AtLimit=0.
ENDIF

ELSEIF (AbsCtrlPrev.EQ.3.) THEN
    IF (RHset.GE.4.44) THEN
IF (TankC.LE.TankHLimit) THEN !
    AbsCtrlState=1.
TimerNew=TIME
AtLimit = 1.
ELSE ! TURN OFF
AbsCtrlState=2.
TimerNew=TIME
AtLimit = 0.
ENDIF
ELSE
    AbsCtrlState=1.
    TimerNew=TIME
    AtLimit=0.
ENDIF
ELSE
    AbsCtrlState=1.
    TimerNew=TIME
    AtLimit=0.
ENDIF

ELSE
    AbsCtrlState=1.
    TimerNew=TIME
    AtLimit=0.
ENDIF
ELSE
    AbsCtrlState=1.
    TimerNew=TIME
    AtLimit=0.
ENDIF

C FAN SPEED CONTROL
    IF (AbsCtrlState.EQ.0.) THEN
Ctrl=0
    ELSEIF (AbsCtrlState.EQ.1.) THEN
Ctrl=CtrlMin
    ELSEIF (AbsCtrlState.GE.2.) THEN
Ctrl=AbsCFM/FanSizeCFM
    ELSE

```

```

Ctrl=Ctrl
ENDIF

C Pump Control
  IF(PumpsPrev.EQ.0.)THEN
    IF(AbsCtrlState.GE.2.) THEN
      AbsPumpsOn=1.
    ELSE
      AbsPumpsOn=PumpsPrev
    ENDIF
  ELSEIF(PumpsPrev.EQ.1.) THEN
    IF(AbsCtrlState.LE.1.)THEN
      AbsPumpsOn=0.
    ELSE
      AbsPumpsOn=PumpsPrev
    ENDIF
  ELSE
    AbsPumpsOn=PumpsPrev
  ENDIF

C Fresh Air Damper Control

C Bypass Damper Control

C-----

C SET THE STORAGE ARRAY AT THE END OF THIS ITERATION IF NECESSARY
Storage(1)=JustOn
Storage(2)=AbsCtrlState
Storage(3)=TimerNew
Storage(4)=Ctrl
Storage(5)=AbsPumpsOn
Storage(6)=DamperFresh
Storage(7)=DamperByp
CALL SetStorageVars(Storage,NITEMS,INFO)

C-----

C SET THE OUTPUTS FROM THIS MODEL IN SEQUENTIAL ORDER AND GET OUT

C Control Signal
OUT(1)=Ctrl
OUT(2)=TimerNew
OUT(3)=TimerPrev
OUT(4)=AbsCtrlState

```

```
OUT(5)=AbsCtrlPrev  
OUT(6)=AbsPumpsOn  
OUT(7)=PumpsPrev  
OUT(8)=AtLimit  
OUT(9)=AtLimit  
OUT(10)=AtLimit  
OUT(11)=AtLimit
```

```
C-----  
C  EVERYTHING IS DONE - RETURN FROM THIS SUBROUTINE AND MOVE ON  
C  RETURN 1  
C  END SUBROUTINE Type992  
C-----
```

Appendix D

TRNSYS Simulation File

D.1 Full System DCK file

```
VERSION 16.1
*****
*** TRNSYS input file (deck) generated by TrnsysStudio
*** on Monday, December 03, 2012 at 11:30
*** from TrnsysStudio project: D:\TRNSYS\FullSys\FullSys_Para\Flows\RegFlow\Orlando\Test2\FullSystem_Para.tpf
***
*** If you edit this file, use the File/Import TRNSYS Input File function in
*** TrnsysStudio to update the project.
***
*** If you have problems, questions or suggestions please contact your local
*** TRNSYS distributor or mailto:software@cstb.fr
***
*****

*****
*** Units
*****

*****
*** Control cards
*****
* START, STOP and STEP
CONSTANTS 12
START=0
STOP=8760
STEP=TIME_STEP
* User defined CONSTANTS
M_REG_PARA=500
M_ABS_PARA=500
TSP_PARA=65
PV_L_PARA=3.24
T_L_PARA=0.81
TANK_DIA_PARA=81
TANK_H_PARA=142
```

```

TANK_CAP_PARA=1000
TANK_B_PARA=5

SIMULATION  START  STOP  STEP ! Start time End time Time step
TOLERANCES 0.0001 0.0001 ! Integration  Convergence
LIMITS 30 100000 30 ! Max iterations Max warnings Trace limit
DFQ 1 ! TRNSYS numerical integration solver method
WIDTH 80 ! TRNSYS output file width, number of characters
LIST ! NOLIST statement
! MAP statement
SOLVER 0 1 1 ! Solver statement Minimum relaxation factor Maximum relaxation factor
NAN_CHECK 0 ! Nan DEBUG statement
OVERWRITE_CHECK 0 ! Overwrite DEBUG statement
TIME_REPORT 0 ! disable time report
EQSOLVER 0 ! EQUATION SOLVER statement

* EQUATIONS "PVParams"
*
EQUATIONS 28
PV_Length = PV_L_PARA ! m
PV_Width = 25.76 ! m
PV_CoverEmissivity = 0.84
PV_CoverConductance = 3.6 ! kJ/h.m.K
PV_CoverThickness = 0.003 ! m
PV_CoverDensity = 2470 ! kg/m3
PV_CoverSpecHeat = 0.75 ! kJ/kg.K
PV_CoverAbsorptance = 0.0402
PV_SubstResistance = 0.001762641 ! h.m2.K/kJ
PV_SubstThickness = 0.0008 ! m
PV_SubstDensity = 1623.75 ! kg/m3
PV_SubstSpecHeat = 2.359125 ! kJ/kg.K
PV_UChanEmiss = 0.87
PV_LChanEmiss = 0.25
PV_BackResistance = 0.81185 !h.m2.K/kJ
PV_BackThickness = 0.0704 ! m
PV_BackDensity = 97.12295 ! kg/m3
PV_BackSpecHeat = 1.46252 ! kJ/kg.K
PV_BackEmissivity = 0.90
PV_ChannelThickness = 0.04826 ! m
PV_NormalTauAlpha = 0.85115
PV_IncAngleModType = 0
PV_RefEfficiency = 0.13424
PV_RefTemp = 25 ! C
PV_RefRad = 3600 ! kJ/h.m2
PV_EffTempMod = -0.00370 ! 1/C
PV_EffRadMod = 0.00002431 ! h.m2/kJ
PV_TurbFactor = 1
*$UNIT_NAME PVParams
*$LAYER Main
*$POSITION 325 58

```

```

*-----

* EQUATIONS "TParams"
*
EQUATIONS 24
T_Length = T_L_PARA ! m
T_Width = 25.76 ! m
T_CoverEmissivity = 0.84
T_CoverThickness = 0.003 ! m
T_CoverDensity = 2470 ! kg/m3
T_CoverSpecHeat = 0.75 ! kJ/kg.K
T_CoverAbsorptance = 0.0402
T_CoverPlateSpacing = 0.02 !m
T_PlateEmissivity = 0.345
T_SubstResistance = 0.0000001414 ! h.m2.K/kJ
T_PlateThickness = 0.0002 ! m
T_PlateDensity = 8910 ! kg/m3
T_PlateSpecHeat = 1.42327 ! kJ/kg.K
T_UChanEmiss = 0.345
T_LChanEmiss = 0.25
T_BackResistance = 0.81185 !h.m2.K/kJ
T_BackThickness = 0.0704 ! m
T_BackDensity = 97.12295 ! kg/m3
T_BackSpecHeat = 1.46252 ! kJ/kg.K
T_BackEmissivity = 0.90
T_ChannelThickness = 0.04826 ! m
T_NormalTauAlpha = 0.82802
T_IncAngleModMult = 0.136
T_TurbFactor = 2.2
*$UNIT_NAME TParams
*$LAYER Main
*$POSITION 407 58

*-----

* EQUATIONS "RegenParams"
*
EQUATIONS 29
m_dot_a_design_CFM = M_REG_PARA
Reg_Scale = m_dot_a_design_CFM/1775
Reg_mrEffectiveness = max((p1*[13,1]^2 + p2*[13,1] + p3)/([13,1]^4 + q1*[13,1]^3 + q2*[13,1]^2 + q3*[13,1] + q4),0)
Reg_hEffectiveness = 0.0
Reg_EwEff = (1-exp(-m_dot_a_design/m_dot_a_in_check))*(Reg_mrEffectiveness/norm_const)
Reg_EhEff = 0.0
m_dot_a_design_lps = m_dot_a_design_CFM*1000./35.31467/60.
m_dot_a_design_kghr = m_dot_a_design_lps*(3600./1000.*rhoAir)
m_dot_a_design = m_dot_a_design_kghr
m_dot_d_design = Pump_GPM*0.0037854*60*Des_Density

```

```

C_p_a_design = 1.0035
C_p_d_design = 2.5833
C_a_design = m_dot_a_design*C_p_a_design
C_d_design = m_dot_d_design*C_p_d_design
UA = 1
C_a_in = [26,3]*C_p_a_design
C_d_in = [61,4]*C_p_d_design
p1 = 393.8
p2 = 621
p3 = 23.35
q1 = -109.7
q2 = 3592
q3 = 75.86
q4 = -0.3841
A_regen_orig = 0.3123 !m2
U_regen = UA/A_regen_orig
A_regen_new = 0.3123
norm_const = 1-exp(-1)
m_dot_a_in_check = max([26,3],0.0000001)
*$UNIT_NAME RegenParams
*$LAYER Main
*$POSITION 494 58

*-----

* EQUATIONS "AbsParams"
*
EQUATIONS 23
m_dot_a_abs_design = M_ABS_PARA
Abs_Scale = m_dot_a_abs_design/2550
Abs_absEffectiveness = 0.63
Abs_hEffectiveness = 0.45
Abs_cwEffectiveness = -0.0125*[24,6]*1000+0.9009
C_p_w_design = 4.183
C_p_a_abs_design = 1.0035
m_dot_cw_design = CTPump_GPM*0.0037854*60*CW_Density
C_a_abs_design = m_dot_a_abs_design*C_p_a_abs_design
C_cw_design = m_dot_cw_design*C_p_w_design
UA_abs_Ew = -min(C_a_abs_design,C_cw_design)*ln(1-Abs_absEffectiveness)
UA_abs_Eh = -min(C_a_abs_design,C_cw_design)*ln(1-Abs_hEffectiveness)
UA_abs_Ecw = -min(C_a_abs_design,C_cw_design)*ln(1-Abs_cwEffectiveness)
C_a_abs_in = [52,4]*C_p_a_abs_design
C_cw_abs_in = [66,2]*C_p_w_design
A_abs_orig = 0.3123
U_abs_Ew = UA_abs_Ew
U_abs_Eh = UA_abs_Eh
U_abs_Ecw = UA_abs_Ecw
A_abs_new = 0.3123
Abs_EwEff = 1-exp(-(U_abs_Ew*A_abs_new)/max(min(C_a_abs_in,C_cw_abs_in),0.00001))
Abs_EhEff = 1-exp(-(U_abs_Eh*A_abs_new)/max(min(C_a_abs_in,C_cw_abs_in),0.00001))

```



```
Abs_EcwEff = 1-exp(-(U_abs_Ecw*A_abs_new)/max(min(C_a_abs_in,C_cw_abs_in),0.00001))
```

```
*$UNIT_NAME AbsParams
```

```
*$LAYER Main
```

```
*$POSITION 585 61
```

```
*-----
```

```
* EQUATIONS "ArrayParams"
```

```
*
```

```
EQUATIONS 2
```

```
ArraySlope = 10
```

```
ArrayAzimuth = 0
```

```
*$UNIT_NAME ArrayParams
```

```
*$LAYER Main
```

```
*$POSITION 237 58
```

```
*-----
```

```
* EQUATIONS "LocParams"
```

```
*
```

```
EQUATIONS 6
```

```
Latitude = [24,32]
```

```
Longitude = [24,33]
```

```
P_atm = 1
```

```
LongStd = 75
```

```
MeridianShift = [24,34]
```

```
P_atm_kPa = P_atm*101.3
```

```
*$UNIT_NAME LocParams
```

```
*$LAYER Main
```

```
*$POSITION 152 58
```

```
*-----
```

```
* EQUATIONS "TimeParams"
```

```
*
```

```
EQUATIONS 2
```

```
StartingDay = 1
```

```
TIME_STEP = 15/60
```

```
*$UNIT_NAME TimeParams
```

```
*$LAYER Main
```

```
*$POSITION 661 61
```

```
*-----
```

```
* EQUATIONS "GndParams"
```

```
*
```

```
EQUATIONS 1
```

```

GRef1 = 0.2
*$UNIT_NAME GndParams
*$LAYER Main
*$POSITION 733 61

*-----

* EQUATIONS "TankParams"
*
EQUATIONS 27
H_buffer_in = TANK_B_PARA
TankCap = TANK_CAP_PARA !gallons
TankDia_in = TANK_DIA_PARA
TankHeight_in = TANK_H_PARA !inches
TankCArea = TankDia*TankHeight
TankCirc = 2*3.14159*(TankDia/2)
TankCapacity = TankCap/264.172
TankMin = TankCapacity/100
TankMax = TankCapacity
TankWetLoss = 55.354
TankDryLoss = 55.354
TankDia = TankDia_in/39.3701 !inches to m
TankHeight = TankHeight_in/39.3701 !inches to m
TankArea = 3.14159*((TankDia/2)^2)
H_max = TankCapacity/TankArea
H_max_in = H_max/0.0254
H_low = H_buffer_in*0.0254
V_low = H_low*TankArea
V_low_gal = V_low*264.172
m_low = V_low*Des_Density
xi_low = Tank_LLimit
m_salt_low = xi_low*m_low
m_des_init = m_salt_low/Cnc_Start
V_init = m_des_init/Des_Density
V_init_gal = V_init*264.172
H_init = V_init/TankArea
H_init_in = H_init/0.0254
*$UNIT_NAME TankParams
*$LAYER Main
*$POSITION 800 61

```

```

*-----

* EQUATIONS "DesParams"
*
EQUATIONS 8
DesVolume = V_init
DesTempInit = 25
SaltMass = m_salt_low

```

```

DesMass = m_des_init
Des_Density = 1375.6
Des_SpecHeat = 2.5833
DesVolume_gal = V_init_gal
Cnc_Start = (Tank_LLimit + Tank_HLimit)/2
*$UNIT_NAME DesParams
*$LAYER Main
*$POSITION 872 61

*-----

* Model "PV" (Type 982)
*

UNIT 12 TYPE 982 PV
*$UNIT_NAME PV
*$MODEL .\Custom\Type982.tmf
*$POSITION 585 181
*$LAYER Main #
PARAMETERS 28
PV_Length ! 1 Collector Length
PV_Width ! 2 Collector Width
PV_CoverEmissivity ! 3 Cover Emissivity
PV_CoverConductance ! 4 Cover Conductivity
PV_CoverThickness ! 5 Cover Thickness
PV_CoverDensity ! 6 Cover Density
PV_CoverSpecHeat ! 7 Cover Specific Heat
PV_CoverAbsorptance ! 8 Cover Absorptance
PV_SubstResistance ! 9 Substrate Resistance
PV_SubstThickness ! 10 Substrate Thickness
PV_SubstDensity ! 11 Substrate Density
PV_SubstSpecHeat ! 12 Substrate Specific Heat
PV_UChanEmiss ! 13 Channel emissivity - top
PV_LChanEmiss ! 14 Channel emissivity - bottom
PV_BackResistance ! 15 Back (Roof) Resistance
PV_BackThickness ! 16 Back (Roof) Thickness
PV_BackDensity ! 17 Back (Roof) Density
PV_BackSpecHeat ! 18 Back (Roof) Specific Heat
PV_BackEmissivity ! 19 Back emissivity
PV_ChannelThickness ! 20 Channel height
PV_NormalTauAlpha ! 21 Normal transmittance-absorptance
PV_IncAngleModType ! 22 IAM Coefficient Cell Type
PV_RefEfficiency ! 23 Reference PV efficiency
PV_RefTemp ! 24 Reference temperature
PV_RefRad ! 25 Reference radiation
PV_EffTempMod ! 26 Efficiency modifier - temperature
PV_EffRadMod ! 27 Efficiency modifier - radiation
PV_TurbFactor ! 28 Air Channel Turbulence Factor
INPUTS 15
22,1 ! RegArrFan:Outlet air temperature ->Inlet temperature

```

```

22,4 ! RegArrFan:Outlet flow rate ->Inlet flowrate
24,1 ! Type15-2:Dry bulb temperature ->Ambient temperature
SkyT ! Convert:SkyT ->Sky temperature
24,1 ! Type15-2:Dry bulb temperature ->Zone temperature
24,1 ! Type15-2:Dry bulb temperature ->Back radiant temperature
24,24 ! Type15-2:Total tilted surface radiation for surface ->Incident solar radiation
24,18 ! Type15-2:Total horizontal radiation ->Total horizontal radiation
24,22 ! Type15-2:Total diffuse radiation on the horizontal ->Horizontal diffuse radiation
0,0 ! [unconnected] Ground reflectance
24,29 ! Type15-2:Angle of incidence for surface ->Incidence angle
24,30 ! Type15-2:Slope of surface ->Collector slope
Top_kJ ! Convert:Top_kJ ->Top heat loss coefficient
Top_kJ ! Convert:Top_kJ ->Bottom heat loss coefficient
24,10 ! Type15-2:Atmospheric pressure ->Atmospheric pressure
*** INITIAL INPUT VALUES
20.0 100.0 10.0 10.0 20.0 20.0 0. 0.0 0.0 GRefl 45.0 10 25.0 11.0 1

```

```

*-----

```

```

* Model "T" (Type 961)

```

```

*

```

```

UNIT 13 TYPE 961 T
*$UNIT_NAME T
*$MODEL .\Custom\Type961.tmf
*$POSITION 734 181
*$LAYER Main #
PARAMETERS 24
T_Length ! 1 Collector Length
T_Width ! 2 Collector Width
T_CoverEmissivity ! 3 Cover Emissivity
T_CoverThickness ! 4 Cover Thickness
T_CoverDensity ! 5 Cover Density
T_CoverSpecHeat ! 6 Cover Specific Heat
T_CoverAbsorptance ! 7 Cover Absorptance
T_CoverPlateSpacing ! 8 Cover Plate Spacing
T_PlateEmissivity ! 9 Plate Emissivity
T_SubstResistance ! 10 Substrate Resistance
T_PlateThickness ! 11 Plate Thickness
T_PlateDensity ! 12 Plate Density
T_PlateSpecHeat ! 13 Plate Specific Heat
T_UChanEmiss ! 14 Channel Emissivity - Top
T_LChanEmiss ! 15 Channel Emissivity - Bottom
T_BackResistance ! 16 Back Resistance
T_BackThickness ! 17 Back (Roof) Thickness
T_BackDensity ! 18 Back (Roof) Density
T_BackSpecHeat ! 19 Back (Roof) Specific Heat
T_BackEmissivity ! 20 Back Emissivity
T_ChannelThickness ! 21 Channel Thickness
T_NormalTauAlpha ! 22 Normal Transmittance Absorptance
T_IncAngleModMult ! 23 Incidence Angle Modifier Multiplier
T_TurbFactor ! 24 Turbulence Factor

```

```

INPUTS 15
12,1 ! PV:Outlet fluid temperature ->Inlet Fluid Temperature
12,2 ! PV:Outlet fluid flow rate ->Inlet Flow Rate
24,1 ! Type15-2:Dry bulb temperature ->Ambient Temperature
SkyT ! Convert:SkyT ->Sky Temperature
24,1 ! Type15-2:Dry bulb temperature ->Zone Temperature
24,1 ! Type15-2:Dry bulb temperature ->Zone Radiative Temperature
24,24 ! Type15-2:Total tilted surface radiation for surface ->Incident Solar Radiation
24,18 ! Type15-2:Total horizontal radiation ->Total Horizontal Radiation
24,22 ! Type15-2:Total diffuse radiation on the horizontal ->Horizontal Diffuse Radiation
0,0 ! [unconnected] Ground Reflectance
24,29 ! Type15-2:Angle of incidence for surface ->Incidence Angle
24,30 ! Type15-2:Slope of surface ->Slope
Top_kJ ! Convert:Top_kJ ->Top Convective Heat Transfer Coefficient
Top_kB ! Convert:Top_kJ ->Back Convective Heat Transfer Coefficient
24,10 ! Type15-2:Atmospheric pressure ->Atmospheric Pressure
*** INITIAL INPUT VALUES
20.0 100.0 10.0 10.0 20.0 20.0 0.0 0.0 0.0 GRef1 45.0 10 25.0 11.0 1
*-----

* EQUATIONS "Convert"
*
EQUATIONS 2
Top_kJ = Hw*3.6
SkyT = [24,1]-10
*$UNIT_NAME Convert
*$LAYER Main
*$POSITION 329 149
*-----

* Model "Type15-2" (Type 15)
*
UNIT 24 TYPE 15 Type15-2
*$UNIT_NAME Type15-2
*$MODEL .\Weather Data Reading and Processing\Standard Format\TMY2\Type15-2.tmf
*$POSITION 55 245
*$LAYER Main #
PARAMETERS 9
2 ! 1 File Type
42 ! 2 Logical unit
3 ! 3 Tilted Surface Radiation Mode
0.2 ! 4 Ground reflectance - no snow
0.7 ! 5 Ground reflectance - snow cover
1 ! 6 Number of surfaces
1 ! 7 Tracking mode
ArraySlope ! 8 Slope of surface
ArrayAzimuth ! 9 Azimuth of surface
*** External files

```

ASSIGN "C:\Trnsys16\Weather\WashingtonDC.TMY2" 42

*|? Which file contains the TMY-2 weather data? |1000

*-----

* EQUATIONS "PumpFanParams"

*

EQUATIONS 31

Arr_FanMax_CFM = m_dot_a_design_CFM

Arr_FanMax_W = Arr_FanMax_CFM/3

Arr_FanMax_Lps = Arr_FanMax_CFM*1000./35.31467/60.

Arr_FanMax_kghr = Arr_FanMax_Lps*(3600./1000.*rhoAir)

Arr_FanMax_kJhr = Arr_FanMax_W*3.6

Arr_FanMax_WCFM = Arr_FanMax_W/Arr_FanMax_CFM

rhoAir = 1.0035

Pump_GPM = 1*Reg_Scale

Pump_kghr = Pump_GPM*0.0037854*60*Des_Density

Pump_kJhr = Pump_W*3.6

Pump_WpGPM = 5

Pump_W = Pump_GPM*Pump_WpGPM

AbsPump_GPM = 1.4*Abs_Scale

AbsPump_kghr = AbsPump_GPM*0.0037854*60*Des_Density

AbsPump_kJhr = AbsPump_W*3.6

AbsPump_WpGPM = 5

AbsPump_W = AbsPump_GPM*AbsPump_WpGPM

CTPump_GPM = 34*Abs_Scale

CTPump_kghr = CTPump_GPM*0.0037854*60*CW_Density

CTPump_kJhr = CTPump_W*3.6

CTPump_WpGPM = 10

CTPump_W = CTPump_WpGPM*CTPump_GPM

Abs_DrainPump_kghr = 2*AbsPump_kghr

Abs_DrainPump_kJhr = 2*AbsPump_kJhr

Abs_FanMax_CFM = m_dot_a_abs_design

Abs_FanMax_W = Abs_FanMax_CFM/3

Abs_FanMax_Lps = Abs_FanMax_CFM*1000./35.31467/60.

Abs_FanMax_kghr = Abs_FanMax_Lps*(3600./1000.*rhoAir)

Abs_FanMax_kJhr = Abs_FanMax_W*3.6

CTFan_m3hr = 8326*Abs_Scale

CTFan_kW = 0.8*Abs_Scale

*\$UNIT_NAME PumpFanParams

*\$LAYER Main

*\$POSITION 950 61

*-----

* Model "Type991" (Type 991)

*

UNIT 21 TYPE 991 Type991

*\$UNIT_NAME Type991

*\$MODEL .\Custom\Type991.tmf

```

*$POSITION 86 349
*$LAYER Main #
PARAMETERS 14
RegMode ! 1 Mode
RegFanSize ! 2 Rated Max Fan Size
RegMinFlow ! 3 Min Flow Rate
RegMaxFlow ! 4 Max Flow Rate
RegMinTemp ! 5 Min Regen Temp
RegMaxTemp ! 6 Max Regen Temp
RegMaxShutoff ! 7 Regen Shutoff Temp
RegTempDOn ! 8 Temp Diff Turn On
RegCtrlStep ! 9 Control Step Size
RegHyst ! 10 Hysteresis Band Size
RegSampleTime ! 11 Array Sample Increment
RegFanStartTime ! 12 Fan Startup Time
Tank_HLimit ! 13 Tank High Limit Conc
Tank_LLimit ! 14 Tank Low Limit Conc
INPUTS 6
13,1 ! T:Outlet Temperature ->Plate Temperature
24,1 ! Type15-2:Dry bulb temperature ->Ambient Temperature
13,1 ! T:Outlet Temperature ->Array Outlet Temperature
0,0 ! [unconnected] Array Outlet Setpoint
12,5 ! PV:Power production ->PV On
CncTank ! CncCalc:CncTank ->Tank Concentration
*** INITIAL INPUT VALUES
30 30 30 RegTsetpoint 0 0.38
*-----

* Model "RegArrFan" (Type 662)
*

UNIT 22 TYPE 662 RegArrFan
*$UNIT_NAME RegArrFan
*$MODEL .\Hydronics Library (TESS)\Fans\Sets the Flowrate\Variable-Speed\RH inputs\Type662-2.tmf
*$POSITION 443 207
*$LAYER Main #
*$$ VARIABLE-SPEED FAN
PARAMETERS 8
2 ! 1 Humidity mode
Arr_FanMax_kghr ! 2 Rated flow rate
Arr_FanMax_kJhr ! 3 Rated power
0.90 ! 4 Motor efficiency
0.0 ! 5 Motor heat loss fraction
2 ! 6 Number of power coefficients
0 ! 7 Power coefficient-1
1.0 ! 8 Power coefficient-2
INPUTS 7
24,1 ! Type15-2:Dry bulb temperature ->Inlet air temperature
24,6 ! Type15-2:Humidity ratio ->Not used (w)
24,7 ! Type15-2:Percent relative humidity ->Inlet air %RH
22,4 ! RegArrFan:Outlet flow rate ->Air flow rate

```

```

24,10 ! Type15-2:Atmospheric pressure ->Inlet air pressure
21,1 ! Type991:Control Signal ->Control signal
0,0 ! [unconnected] Air-side pressure increase
*** INITIAL INPUT VALUES
20.0 0.008 50. 2000.0 1. 1.0 0.0
*-----

* EQUATIONS "RegCtrlParams"
*
EQUATIONS 15
RegMode = 2
RegFanSize = Arr_FanMax_CFM*1.0
RegMinFlow = 0.05*Arr_FanMax_CFM
RegMaxFlow = 1.0*Arr_FanMax_CFM
RegMinTemp = 45
RegMaxTemp = 120
RegMaxShutoff = 120
RegTempDDn = 3
RegCtrlStep = 30/RegFanSize
RegHyst = 1
RegSampleTime = 20/60
RegFanStartTime = 3/60
Tank_HLimit = 0.30
Tank_LLimit = 0.42
RegTsetpoint = TSP_PARA
*$UNIT_NAME RegCtrlParams
*$LAYER Main
*$POSITION 1041 61
*-----

* Model "FreshAirDamper" (Type 11)
*
UNIT 25 TYPE 11 FreshAirDamper
*$UNIT_NAME FreshAirDamper
*$MODEL .\Hydronics\Flow Mixer\Moist Air\Type11c.tmf
*$POSITION 929 261
*$LAYER Main #
PARAMETERS 1
8 ! 1 Controlled flow mixer mode
INPUTS 7
13,1 ! T:Outlet Temperature ->Temperature at inlet 1
22,2 ! RegArrFan:Outlet humidity ratio ->Humidity ratio at inlet 1
FreshDamperPVTin ! FreshDamperCalc:FreshDamperPVTin ->Flow rate at inlet 1
24,1 ! Type15-2:Dry bulb temperature ->Temperature at inlet 2
24,6 ! Type15-2:Humidity ratio ->Humidity ratio at inlet 2
FreshDamperOAIN ! FreshDamperCalc:FreshDamperOAIN ->Flow rate at inlet 2
21,8 ! Type991:Fresh Air Damper Control Signal ->Control signal
*** INITIAL INPUT VALUES

```


20.0 0.006 100.0 20.0 0.006 100.0 0

*-----

* Model "BypDamper" (Type 11)

*

UNIT 26 TYPE 11 BypDamper

*\$UNIT_NAME BypDamper

*\$MODEL .\Hydronics\Flow Diverter\Moist Air\Type11e.tmf

*\$POSITION 1050 261

*\$LAYER Main #

PARAMETERS 1

7 ! 1 Controlled flow diverter mode

INPUTS 4

25,1 ! FreshAirDamper:Outlet temperature ->Inlet temperature

25,2 ! FreshAirDamper:Outlet humidity ratio ->Inlet humidity ratio

25,3 ! FreshAirDamper:Outlet flow rate ->Inlet flow rate

BypDampSig ! CtrlSigInv:BypDampSig ->Control signal

*** INITIAL INPUT VALUES

20.0 0.006 100.0 0

*-----

* EQUATIONS "FreshDamperCalc"

*

EQUATIONS 2

FreshDamperOAin = ([21,8])*([13,2])

FreshDamperPVTin = (1-[21,8])*([13,2])

*\$UNIT_NAME FreshDamperCalc

*\$LAYER Main

*\$POSITION 804 240

*-----

* Model "Type33c" (Type 33)

*

UNIT 29 TYPE 33 Type33c

*\$UNIT_NAME Type33c

*\$MODEL .\Physical Phenomena\Thermodynamic Properties\Psihrometrics\Dry Bulb and Humidity Ratio Known\Type33c.tmf

*\$POSITION 1046 421

*\$LAYER Main #

PARAMETERS 3

4 ! 1 Psychrometrics mode

1 ! 2 Wet bulb mode

2 ! 3 Error mode

INPUTS 3

26,1 ! BypDamper:Temperature at outlet 1 ->Dry bulb temp.

26,2 ! BypDamper:Humidity ratio at outlet 1 ->Absolute humidity ratio

22,5 ! RegArrFan:Outlet air pressure ->Pressure

*** INITIAL INPUT VALUES

22.0 0.006 1

*-----

* Model "Regenerator" (Type 937)

*

UNIT 32 TYPE 937 Regenerator

*\$UNIT_NAME Regenerator

*\$MODEL .\Custom\Type937.tmf

*\$POSITION 928 517

*\$LAYER Main #

PARAMETERS 2

P_atm_kPa ! 1 Atmospheric Pressure

Des_SpecHeat ! 2 Specific Heat Capacity

INPUTS 8

26,1 ! BypDamper:Temperature at outlet 1 ->Inlet Air Temperature

RH_frac_out ! RHconv:RH_frac_out ->Inlet Air RH

26,3 ! BypDamper:Flow rate at outlet 1 ->Air Mass Flow Rate

CncTank ! CncCalc:CncTank ->Entering Desiccant Concentration

61,3 ! Type5b:Cold-side outlet temperature ->Inlet Desiccant Temperature

61,4 ! Type5b:Cold-side flow rate ->Desiccant Mass Flow Rate

Reg_EwEff ! RegenParams:Reg_EwEff ->Humidity Ratio Effectiveness

Reg_hEffectiveness ! RegenParams:Reg_hEffectiveness ->Enthalpy Effectiveness

*** INITIAL INPUT VALUES

25 0.5 887 0.38 25 312.42 0.65 0

*-----

* Model "BypDamp" (Type 11)

*

UNIT 33 TYPE 11 BypDamp

*\$UNIT_NAME BypDamp

*\$MODEL .\Hydronics\Flow Mixer\Moist Air\Type11c.tmf

*\$POSITION 1050 752

*\$LAYER Main #

PARAMETERS 1

8 ! 1 Controlled flow mixer mode

INPUTS 7

32,4 ! Regenerator:Leaving Air Temperature ->Temperature at inlet 1

34,1 ! Type33e:Humidity ratio ->Humidity ratio at inlet 1

32,13 ! Regenerator:Air Mass Flow Rate ->Flow rate at inlet 1

26,4 ! BypDamper:Temperature at outlet 2 ->Temperature at inlet 2

26,5 ! BypDamper:Humidity ratio at outlet 2 ->Humidity ratio at inlet 2

26,6 ! BypDamper:Flow rate at outlet 2 ->Flow rate at inlet 2

0,0 ! [unconnected] Control signal

*** INITIAL INPUT VALUES

20.0 0.006 100.0 20.0 0.006 100.0 0.5

*-----

* Model "Type33e" (Type 33)

*

```

UNIT 34 TYPE 33 Type33e
*$UNIT_NAME Type33e
*$MODEL .\Physical Phenomena\Thermodynamic Properties\Psihrometrics\Dry Bulb and Relative Humidity Known\Type33e.tmf
*$POSITION 1025 621
*$LAYER Main #
PARAMETERS 3
2 ! 1 Psychrometrics mode
1 ! 2 Wet bulb mode
2 ! 3 Error mode
INPUTS 3
32,4 ! Regenerator:Leaving Air Temperature ->Dry bulb temp.
RH_perc_out ! RHconv:RH_perc_out ->Percent relative humidity
0,0 ! [unconnected] Pressure
*** INITIAL INPUT VALUES
22.0 60.0 1
*-----

* Model "RegenPump" (Type 3)
*

UNIT 41 TYPE 3 RegenPump
*$UNIT_NAME RegenPump
*$MODEL .\Hydronics\Pumps\Single Speed\Type3b.tmf
*$POSITION 686 592
*$LAYER Main #
PARAMETERS 5
Pump_kghr ! 1 Maximum flow rate
Des_SpecHeat ! 2 Fluid specific heat
Pump_KJhr ! 3 Maximum power
0.05 ! 4 Conversion coefficient
0.5 ! 5 Power coefficient
INPUTS 3
46,3 ! Type647:Outlet temperature-2 ->Inlet fluid temperature
46,4 ! Type647:Outlet flow rate-2 ->Inlet mass flow rate
21,6 ! Type991:Pump Control Signal ->Control signal
*** INITIAL INPUT VALUES
37 100.0 0
*-----

* EQUATIONS "CtrlSigInv"
*
EQUATIONS 1
BypDampSig = not([21,10])
*$UNIT_NAME CtrlSigInv
*$LAYER Main
*$POSITION 453 413
*-----

```

```
* EQUATIONS "RHconv"
```

```
*
```

```
EQUATIONS 2
```

```
RH_frac_out = [29,6]/100.0
```

```
RH_perc_out = [32,3]*100.0
```

```
*$UNIT_NAME RHconv
```

```
*$LAYER Main
```

```
*$POSITION 1067 520
```

```
*-----
```

```
* Model "RegDrainPump" (Type 110)
```

```
*
```

```
UNIT 43 TYPE 110 RegDrainPump
```

```
*$UNIT_NAME RegDrainPump
```

```
*$MODEL .\Hydraulics\Pumps\Variable Speed\Type110.tmf
```

```
*$POSITION 701 794
```

```
*$LAYER Main #
```

```
*$# VARIABLE-SPEED PUMP
```

```
PARAMETERS 6
```

```
Pump_kghr ! 1 Rated flow rate
```

```
Des_SpecHeat ! 2 Fluid specific heat
```

```
0 ! 3 Rated power
```

```
0.0 ! 4 Motor heat loss fraction
```

```
1 ! 5 Number of power coefficients
```

```
1.0 ! 6 Power coefficient
```

```
INPUTS 5
```

```
61,1 ! Type5b:Hot-side outlet temperature ->Inlet fluid temperature
```

```
61,2 ! Type5b:Hot-side flow rate ->Inlet fluid flow rate
```

```
RegDrainPump_Ctrl ! TankCtrl:RegDrainPump_Ctrl ->Control signal
```

```
0,0 ! [unconnected] Total pump efficiency
```

```
0,0 ! [unconnected] Motor efficiency
```

```
*** INITIAL INPUT VALUES
```

```
20.0 0.0 1.0 1 1
```

```
*-----
```

```
* Model "IntAbsReg" (Type 24)
```

```
*
```

```
UNIT 49 TYPE 24 IntAbsReg
```

```
*$UNIT_NAME IntAbsReg
```

```
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
```

```
*$POSITION 452 603
```

```
*$LAYER Main #
```

```
PARAMETERS 2
```

```
STOP ! 1 Integration period
```

```
0 ! 2 Relative or absolute start time
```

```
INPUTS 2
```

```
m_dot_abs ! AbsRegRates:m_dot_abs ->Input to be integrated-1
```

```
m_dot_reg ! AbsRegRates:m_dot_reg ->Input to be integrated-2
```

```
*** INITIAL INPUT VALUES
```

```
0.0 0.0
```

```
*-----
```

```
* EQUATIONS "AbsRegRates"
```

```
*
```

```
EQUATIONS 2
```

```
m_dot_abs = [62,9]-[54,2]
```

```
m_dot_reg = [32,14]-[32,12]
```

```
*$UNIT_NAME AbsRegRates
```

```
*$LAYER Main
```

```
*$POSITION 452 517
```

```
*-----
```

```
* EQUATIONS "CncCalc"
```

```
*
```

```
EQUATIONS 4
```

```
CncTank = SaltMass/m_tank
```

```
m_tank = DesMass - [49,2] + [49,1]
```

```
At_HighLimit = le(CncTank,Tank_HLimit)
```

```
At_LowLimit = ge(CncTank,Tank_LLimit)
```

```
*$UNIT_NAME CncCalc
```

```
*$LAYER Main
```

```
*$POSITION 546 603
```

```
*-----
```

```
* Model "ReCircPump" (Type 3)
```

```
*
```

```
UNIT 45 TYPE 3 ReCircPump
```

```
*$UNIT_NAME ReCircPump
```

```
*$MODEL .\Hydronics\Pumps\Single Speed\Type3b.tmf
```

```
*$POSITION 498 795
```

```
*$LAYER Main #
```

```
PARAMETERS 5
```

```
Pump_kghr ! 1 Maximum flow rate
```

```
Des_SpecHeat ! 2 Fluid specific heat
```

```
Pump_KJhr ! 3 Maximum power
```

```
0.05 ! 4 Conversion coefficient
```

```
0.5 ! 5 Power coefficient
```

```
INPUTS 3
```

```
46,5 ! Type647:Outlet temperature-3 ->Inlet fluid temperature
```

```
46,6 ! Type647:Outlet flow rate-3 ->Inlet mass flow rate
```

```
ReCirc_Pump_Ctrl ! TankCtrl:ReCirc_Pump_Ctrl ->Control signal
```

```
*** INITIAL INPUT VALUES
```

```
20.0 Pump_kghr 1.0
```

```

*-----
* Model "Type647" (Type 647)
*

UNIT 46 TYPE 647 Type647
*$UNIT_NAME Type647
*$MODEL .\Hydraulics Library (TESS)\Valves\Diverting Valve (100 Ports)\Other Fluids\Type647.tmf
*$POSITION 585 719
*$LAYER Main #
PARAMETERS 1
3 ! 1 Number of outlet ports
INPUTS 5
47,1 ! LDTank:Fluid temperature ->Inlet Temperature
47,2 ! LDTank:Load flow rate ->Inlet flow rate
Tank_Frac_Abs ! TankCtrl:Tank_Frac_Abs ->Fraction of flow to outlet -1
Tank_Frac_Reg ! TankCtrl:Tank_Frac_Reg ->Fraction of flow to outlet -2
Tank_Frac_ReCirc ! TankCtrl:Tank_Frac_ReCirc ->Fraction of flow to outlet -3
*** INITIAL INPUT VALUES
20.0 500.0 0.5 0.5 0.5
*-----

* Model "LDTank" (Type 39)
*

UNIT 47 TYPE 39 LDTank
*$UNIT_NAME LDTank
*$MODEL .\Thermal Storage\Variable Volume Tank\Type39.tmf
*$POSITION 499 720
*$LAYER Main #
PARAMETERS 12
2 ! 1 Tank operation mode
TankCapacity ! 2 Overall tank volume
TankMin ! 3 Minimum fluid volume
TankMax ! 4 Maximum fluid volume
TankCirc ! 5 Tank circumference
TankCArea ! 6 Cross-sectional area
TankWetLoss ! 7 Wetted loss coefficient
TankDryLoss ! 8 Dry loss coefficient
Des_SpecHeat ! 9 Fluid specific heat
Des_Density ! 10 Fluid density
DesTempInit ! 11 Initial fluid temperature
DesVolume ! 12 Initial fluid volume
INPUTS 4
48,1 ! Type649:Outlet temperature ->Inlet temperature
48,2 ! Type649:Outlet flow rate ->Inlet flow rate
Tank_Load_Flow_Rate ! TankCtrl:Tank_Load_Flow_Rate ->Flow rate to load
24,1 ! Type15-2:Dry bulb temperature ->Environment temperature
*** INITIAL INPUT VALUES
25.0 100.0 75.0 15.0
*-----

```

```

* Model "Type649" (Type 649)
*

UNIT 48 TYPE 649 Type649
*$UNIT_NAME Type649
*$MODEL .\Hydronics Library (TESS)\Valves\Mixing Valve (100 Ports)\Other Fluids\Type649.tmf
*$POSITION 415 719
*$LAYER Main #
PARAMETERS 1
3 ! 1 Number of inlets
INPUTS 6
57,1 ! AbsDrainPump:Outlet fluid temperature ->Temperature at inlet-1
57,2 ! AbsDrainPump:Outlet flow rate ->Flow rate at inlet-1
43,1 ! RegDrainPump:Outlet fluid temperature ->Temperature at inlet-2
43,2 ! RegDrainPump:Outlet flow rate ->Flow rate at inlet-2
45,1 ! ReCircPump:Outlet fluid temperature ->Temperature at inlet-3
45,2 ! ReCircPump:Outlet flow rate ->Flow rate at inlet-3
*** INITIAL INPUT VALUES
20 200.0 20 200.0 20 200.0
*-----

* EQUATIONS "TankCtrl"
*
EQUATIONS 7
ReCirc_Pump_Ctrl = 1
Tank_Load_Flow_Rate = AbsPump_kghr*[58,6] + Pump_kghr*[21,6] + Pump_kghr*ReCirc_Pump_Ctrl
Tank_Frac_Abs = (AbsPump_kghr*[58,6])/Tank_Load_Flow_Rate
Tank_Frac_Reg = (Pump_kghr*[21,6])/Tank_Load_Flow_Rate
Tank_Frac_ReCirc = (Pump_kghr*ReCirc_Pump_Ctrl)/Tank_Load_Flow_Rate
AbsDrainPump_Ctrl = [62,9]/Abs_DrainPump_kghr
RegDrainPump_Ctrl = [32,12]/Pump_kghr
*$UNIT_NAME TankCtrl
*$LAYER Main
*$POSITION 502 872
*-----

* Model "AbsPump" (Type 3)
*

UNIT 54 TYPE 3 AbsPump
*$UNIT_NAME AbsPump
*$MODEL .\Hydronics\Pumps\Single Speed\Type3b.tmf
*$POSITION 709 688
*$LAYER Main #
PARAMETERS 5
AbsPump_kghr ! 1 Maximum flow rate
Des_SpecHeat ! 2 Fluid specific heat
AbsPump_KJhr ! 3 Maximum power

```

```

0.05 ! 4 Conversion coefficient
0.5 ! 5 Power coefficient
INPUTS 3
46,1 ! Type647:Outlet temperature-1 ->Inlet fluid temperature
46,2 ! Type647:Outlet flow rate-1 ->Inlet mass flow rate
58,6 ! Type992:Pump Control Signal ->Control signal
*** INITIAL INPUT VALUES
37 AbsPump_kghr 0
*-----

* Model "AbsDrainPump" (Type 110)
*

UNIT 57 TYPE 110 AbsDrainPump
*$UNIT_NAME AbsDrainPump
*$MODEL .\Hydronics\Pumps\Variable Speed\Type110.tmf
*$POSITION 371 850
*$LAYER Main #
*$# VARIABLE-SPEED PUMP
PARAMETERS 6
Abs_DrainPump_kghr ! 1 Rated flow rate
Des_SpecHeat ! 2 Fluid specific heat
0 ! 3 Rated power
0.0 ! 4 Motor heat loss fraction
1 ! 5 Number of power coefficients
1.0 ! 6 Power coefficient
INPUTS 5
62,6 ! Absorber:Outlet Desiccant Temperature ->Inlet fluid temperature
62,9 ! Absorber:Outlet Desiccant Flow Rate ->Inlet fluid flow rate
AbsDrainPump_Ctrl ! TankCtrl:AbsDrainPump_Ctrl ->Control signal
0,0 ! [unconnected] Total pump efficiency
0,0 ! [unconnected] Motor efficiency
*** INITIAL INPUT VALUES
20.0 0.0 1.0 1 1
*-----

* EQUATIONS "CWParams"
*
EQUATIONS 2
CW_SpecHeat = 4.183
CW_Density = 998.3
*$UNIT_NAME CWParams
*$LAYER Main
*$POSITION 1027 133
*-----

* Model "Type992" (Type 992)
*

```



```

UNIT 58 TYPE 992  Type992
*$UNIT_NAME Type992
*$MODEL .\Custom\Type992.tmf
*$POSITION 86 456
*$LAYER Main #
PARAMETERS 10
Abs_FanMax_CFM ! 1 Rated Fan Size
Abs_MinCFM ! 2 Min Fan Flow Rate
Abs_MaxCFM ! 3 Max Fan Flow Rate
AbsWaitTime ! 4 Absorption Wait Time
Tank_HLimit ! 5 Tank Concentration - High Limit
Tank_LLimit ! 6 Tank Concentration - Low Limit
Abs_Hyst_Enable ! 7 Enable Hysteresis
Abs_Hyst_Inc ! 8 Hysteresis Size
Abs_Ctrl_Inc ! 9 Control Increment
Abs_Ctrl_Band ! 10 Control Band Size
INPUTS 5
HumRat_Test ! Equa-3:HumRat_Test ->Absorber Enable
Abs_Flow_Rate ! AbsCtrlParams:Abs_Flow_Rate ->Load Flow Rate
CncTank ! CncCalc:CncTank ->Tank Concentration
24,1 ! Type15-2:Dry bulb temperature ->RH Set Point
62,7 ! Absorber:Absorption Rate ->Current RH Input
*** INITIAL INPUT VALUES
1 200 0.38 15 0.5
*-----

* Model "AbsFan" (Type 662)
*

UNIT 52 TYPE 662  AbsFan
*$UNIT_NAME AbsFan
*$MODEL .\Hydronics Library (TESS)\Fans\Sets the Flowrate\Variable-Speed\RH inputs\Type662-2.tmf
*$POSITION 128 551
*$LAYER Main #
*$$ VARIABLE-SPEED FAN
PARAMETERS 8
2 ! 1 Humidity mode
Abs_FanMax_kghr ! 2 Rated flow rate
Abs_FanMax_kJhr ! 3 Rated power
0.90 ! 4 Motor efficiency
0.0 ! 5 Motor heat loss fraction
2 ! 6 Number of power coefficients
0 ! 7 Power coefficient-1
1.0 ! 8 Power coefficient-2
INPUTS 7
24,1 ! Type15-2:Dry bulb temperature ->Inlet air temperature
24,6 ! Type15-2:Humidity ratio ->Not used (w)
24,7 ! Type15-2:Percent relative humidity ->Inlet air %RH
52,4 ! AbsFan:Outlet flow rate ->Air flow rate
24,10 ! Type15-2:Atmospheric pressure ->Inlet air pressure
58,1 ! Type992:Fan Control Signal ->Control signal

```

```

0,0 ! [unconnected] Air-side pressure increase
*** INITIAL INPUT VALUES
20.0 0.008 50. 2000.0 1. 1.0 0.0
*-----

* EQUATIONS "AbsCtrlParams"
*
EQUATIONS 11
AbsWaitTime = 0
Abs_MaxCFM = Arr_FanMax_CFM
Abs_MinCFM = 0
Abs_Hyst_Enable = 1
Abs_Hyst_Inc = 0.005
Abs_Enable = 1
Abs_Ctrl_Inc = -0.01
Abs_Ctrl_Band = 0.5
Abs_RHSet = 2
Abs_Flow_Rate = m_dot_a_abs_design
Abs_HumRat_SP = 0.008
*$UNIT_NAME AbsCtrlParams
*$LAYER Main
*$POSITION 1127 61

*-----

* Model "Type5b" (Type 5)
*
UNIT 61 TYPE 5 Type5b
*$UNIT_NAME Type5b
*$MODEL .\Heat Exchangers\Counter Flow\Type5b.tmf
*$POSITION 820 712
*$LAYER Main #
PARAMETERS 4
2 ! 1 Counter flow mode
Des_SpecHeat ! 2 Specific heat of hot side fluid
Des_SpecHeat ! 3 Specific heat of cold side fluid
0 ! 4 Not used
INPUTS 5
32,6 ! Regenerator:Outlet Desiccant Temperature ->Hot side inlet temperature
32,12 ! Regenerator:Outlet Desiccant Flow Rate ->Hot side flow rate
41,1 ! RegenPump:Outlet fluid temperature ->Cold side inlet temperature
41,2 ! RegenPump:Outlet flow rate ->Cold side flow rate
0,0 ! [unconnected] Overall heat transfer coefficient of exchanger
*** INITIAL INPUT VALUES
20.0 100.0 20.0 100.0 10.0
*-----

* Model "Absorber" (Type 974)
*

```

```

UNIT 62 TYPE 974 Absorber
*$UNIT_NAME Absorber
*$MODEL .\Custom\Type974.tmf
*$POSITION 335 520
*$LAYER Main #
PARAMETERS 3
P_atm_kPa ! 1 Atmospheric Pressure
Des_SpecHeat ! 2 Desiccant Specific Heat Capacity
CW_SpecHeat ! 3 Cooling Water Specific Heat Capacity
INPUTS 11
52,1 ! AbsFan:Outlet air temperature ->Entering Air Temperature
Abs_RH_frac_out ! RH_Conv2:Abs_RH_frac_out ->Entering Air Relative Humidity
52,4 ! AbsFan:Outlet flow rate ->Air Flow Rate
CncTank ! CncCalc:CncTank ->Entering Desiccant Concentration
54,1 ! AbsPump:Outlet fluid temperature ->Entering Desiccant Temperature
54,2 ! AbsPump:Outlet flow rate ->Entering Desiccant Flow Rate
66,1 ! CTPump:Outlet fluid temperature ->Entering Cooling Water Temperature
66,2 ! CTPump:Outlet flow rate ->Entering Cooling Water Flow Rate
Abs_absEffectiveness ! AbsParams:Abs_absEffectiveness ->Absortion Effectiveness
Abs_hEffectiveness ! AbsParams:Abs_hEffectiveness ->Enthalpy Effectiveness
Abs_cwEffectiveness ! AbsParams:Abs_cwEffectiveness ->Cooling Water Effectiveness
*** INITIAL INPUT VALUES
20 0.5 500 0.38 20 5 20 1000 0.65 0.48 0.6
*-----

* EQUATIONS "RH_Conv2"
*
EQUATIONS 1
Abs_RH_frac_out = [52,3]/100.0
*$UNIT_NAME RH_Conv2
*$LAYER Main
*$POSITION 232 498
*-----

* Model "Type51a" (Type 51)
*

UNIT 65 TYPE 51 Type51a
*$UNIT_NAME Type51a
*$MODEL .\HVAC\Cooling Towers\External Performance File\Type51a.tmf
*$POSITION 225 626
*$LAYER Main #
PARAMETERS 11
2 ! 1 Calculation mode
1 ! 2 Flow geometry
1 ! 3 Number of tower cells
CTFan_m3hr ! 4 Maximum cell flow rate
CTFan_kW ! 5 Fan power at maximum flow

```

```

10 ! 6 Minimum cell flow rate
0.1 ! 7 Sump volume
20 ! 8 Initial sump temperature
54 ! 9 Logical unit
9 ! 10 Number of data points
1 ! 11 Print performance results?
INPUTS 6
62,12 ! Absorber:Leaving Cooling Water Temperature ->Water inlet temperature
62,13 ! Absorber:Cooling Water Flow Rate ->Inlet water flow rate
24,1 ! Type15-2:Dry bulb temperature ->Dry bulb temperature
24,3 ! Type15-2:Wet bulb temperature ->Wet bulb temperature
0,0 ! [unconnected] Sump make-up temperature
58,6 ! Type992:Pump Control Signal ->Relative fan speed for cell
*** INITIAL INPUT VALUES
20 7936 25 12 10 1
*** External files
ASSIGN "CoolTwrData2.csv" 54
!/? Which file contains the cooling tower performance data? |1000
*-----

* Model "CTPump" (Type 3)
*

UNIT 66 TYPE 3 CTPump
*$UNIT_NAME CTPump
*$MODEL .\Hydronics\Pumps\Single Speed - no Powercoefficients\Type3d.tmf
*$POSITION 225 744
*$LAYER Main #
PARAMETERS 4
CTPump_kg/hr ! 1 Maximum flow rate
CW_SpecHeat ! 2 Fluid specific heat
CTPump_W ! 3 Maximum power
0.05 ! 4 Conversion coefficient
INPUTS 3
65,1 ! Type51a:Sump temperature ->Inlet fluid temperature
65,2 ! Type51a:Sump flow rate ->Inlet mass flow rate
58,6 ! Type992:Pump Control Signal ->Control signal
*** INITIAL INPUT VALUES
20 CTPump_kg/hr 1
*-----

* Model "Type24" (Type 24)
*

UNIT 71 TYPE 24 Type24
*$UNIT_NAME Type24
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 585 335
*$LAYER Main #
PARAMETERS 2
PrintInterval ! 1 Integration period

```

```

0 ! 2 Relative or absolute start time
INPUTS 14
12,5 ! PV:Power production ->Input to be integrated-1
22,6 ! RegArrFan:Power consumption ->Input to be integrated-2
41,3 ! RegenPump:Power consumption ->Input to be integrated-3
43,3 ! RegDrainPump:Power consumption ->Input to be integrated-4
52,6 ! AbsFan:Power consumption ->Input to be integrated-5
54,3 ! AbsPump:Power consumption ->Input to be integrated-6
57,3 ! AbsDrainPump:Power consumption ->Input to be integrated-7
66,3 ! CTPump:Power consumption ->Input to be integrated-8
45,3 ! ReCircPump:Power consumption ->Input to be integrated-9
32,5 ! Regenerator:Desiccant Desorption Rate ->Input to be integrated-10
62,7 ! Absorber:Absorption Rate ->Input to be integrated-11
CT_kJhr ! CoolPower:CT_kJhr ->Input to be integrated-12
48,2 ! Type649:Outlet flow rate ->Input to be integrated-13
47,2 ! LDTank:Load flow rate ->Input to be integrated-14
*** INITIAL INPUT VALUES
0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0
*-----

* Model "Summary" (Type 25)
*

UNIT 72 TYPE 25 Summary
*$UNIT_NAME Summary
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 737 338
*$LAYER Outputs #
PARAMETERS 10
PrintInterval ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
59 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 16
PV_M_Out ! IntPrint:PV_M_Out ->Input to be printed-1
RegFan_M_Out ! IntPrint:RegFan_M_Out ->Input to be printed-2
RegPump_M_Out ! IntPrint:RegPump_M_Out ->Input to be printed-3
RegDrainPump_M_Out ! IntPrint:RegDrainPump_M_Out ->Input to be printed-4
AbsFan_M_Out ! IntPrint:AbsFan_M_Out ->Input to be printed-5
AbsPump_M_Out ! IntPrint:AbsPump_M_Out ->Input to be printed-6
AbsDrainPump_M_Out ! IntPrint:AbsDrainPump_M_Out ->Input to be printed-7
AbsCTPump_M_Out ! IntPrint:AbsCTPump_M_Out ->Input to be printed-8
ReCircPump_M_Out ! IntPrint:ReCircPump_M_Out ->Input to be printed-9
Reg_M_Out ! IntPrint:Reg_M_Out ->Input to be printed-10
Abs_M_Out ! IntPrint:Abs_M_Out ->Input to be printed-11

```

```

FanPumpE_Monthly ! IntPrint:FanPumpE_Monthly ->Input to be printed-12
WaterBal_Monthly ! IntPrint:WaterBal_Monthly ->Input to be printed-13
EDiff ! IntPrint:EDiff ->Input to be printed-14
MassDiff ! IntPrint:MassDiff ->Input to be printed-15
CT_Fan_M_Out ! IntPrint:CT_Fan_M_Out ->Input to be printed-16

*** INITIAL INPUT VALUES
PV RegFan RegPump RegDrainPump AbsFan AbsPump AbsDrainPump AbsCTPump
ReCircPump Reg Abs FanPumpT WaterBalance EDiff MDiff CTFan

*** External files
ASSIGN "***.out" 59

*!? Output file for printed results |1000
*-----

* EQUATIONS "IntPrint"
*
EQUATIONS 19
FanPumpE_Monthly = ([71,2] + [71,3] + [71,4] + [71,5] + [71,6] + [71,7] + [71,8] + [71,9] + [71,12])/3600
WaterBal_Monthly = [71,11] - [71,10]
PV_M_Out = [71,1]/3600
RegFan_M_Out = [71,2]/3600
RegPump_M_Out = [71,3]/3600
RegDrainPump_M_Out = [71,4]/3600
AbsFan_M_Out = [71,5]/3600
AbsPump_M_Out = [71,6]/3600
AbsDrainPump_M_Out = [71,7]/3600
AbsCTPump_M_Out = [71,8]/3600
ReCircPump_M_Out = [71,9]/3600
CT_Fan_M_Out = [71,12]/3600
Reg_M_Out = [71,10]
Abs_M_Out = [71,11]
EDiff = ([71,1] - FanPumpE_Monthly)/3600
MassFluidNow = [71,13] - [71,14]
MassDiff = MassFluidNow + Reg_M_Out - Abs_M_Out
PrintInterval = -1
DWorA = -1
*$UNIT_NAME IntPrint
*$LAYER Main
*$POSITION 664 338

*-----

* Model "PrintPumpFan" (Type 25)
*
UNIT 74 TYPE 25 PrintPumpFan
*$UNIT_NAME PrintPumpFan
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 822 157
*$LAYER Outputs #
PARAMETERS 10

```

```

STOP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
60 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels

INPUTS 26
Arr_FanMax_CFM ! PumpFanParams:Arr_FanMax_CFM ->Input to be printed-1
Arr_FanMax_W ! PumpFanParams:Arr_FanMax_W ->Input to be printed-2
Arr_FanMax_Lps ! PumpFanParams:Arr_FanMax_Lps ->Input to be printed-3
Arr_FanMax_kghr ! PumpFanParams:Arr_FanMax_kghr ->Input to be printed-4
Arr_FanMax_kJhr ! PumpFanParams:Arr_FanMax_kJhr ->Input to be printed-5
Arr_FanMax_WCFM ! PumpFanParams:Arr_FanMax_WCFM ->Input to be printed-6
rhoAir ! PumpFanParams:rhoAir ->Input to be printed-7
Pump_GPM ! PumpFanParams:Pump_GPM ->Input to be printed-8
Pump_kghr ! PumpFanParams:Pump_kghr ->Input to be printed-9
Pump_kJhr ! PumpFanParams:Pump_kJhr ->Input to be printed-10
Pump_W ! PumpFanParams:Pump_W ->Input to be printed-11
AbsPump_GPM ! PumpFanParams:AbsPump_GPM ->Input to be printed-12
AbsPump_kghr ! PumpFanParams:AbsPump_kghr ->Input to be printed-13
AbsPump_kJhr ! PumpFanParams:AbsPump_kJhr ->Input to be printed-14
AbsPump_W ! PumpFanParams:AbsPump_W ->Input to be printed-15
CTPump_GPM ! PumpFanParams:CTPump_GPM ->Input to be printed-16
CTPump_kghr ! PumpFanParams:CTPump_kghr ->Input to be printed-17
CTPump_kJhr ! PumpFanParams:CTPump_kJhr ->Input to be printed-18
CTPump_W ! PumpFanParams:CTPump_W ->Input to be printed-19
Abs_DrainPump_kghr ! PumpFanParams:Abs_DrainPump_kghr ->Input to be printed-20
Abs_DrainPump_kJhr ! PumpFanParams:Abs_DrainPump_kJhr ->Input to be printed-21
Abs_FanMax_CFM ! PumpFanParams:Abs_FanMax_CFM ->Input to be printed-22
Abs_FanMax_W ! PumpFanParams:Abs_FanMax_W ->Input to be printed-23
Abs_FanMax_Lps ! PumpFanParams:Abs_FanMax_Lps ->Input to be printed-24
Abs_FanMax_kghr ! PumpFanParams:Abs_FanMax_kghr ->Input to be printed-25
Abs_FanMax_kJhr ! PumpFanParams:Abs_FanMax_kJhr ->Input to be printed-26

*** INITIAL INPUT VALUES
Arr_FanMax_CFM Arr_FanMax_W Arr_FanMax_Lps Arr_FanMax_kghr Arr_FanMax_kJhr
Arr_FanMax_WCFM rhoAir Pump_GPM Pump_kghr Pump_kJhr Pump_W AbsPump_GPM
AbsPump_kghr AbsPump_kJhr AbsPump_W CTPump_GPM CTPump_kghr CTPump_kJhr
CTPump_W Abs_DrainPump_kghr Abs_DrainPump_kJhr Abs_FanMax_CFM Abs_FanMax_W
Abs_FanMax_Lps Abs_FanMax_kghr Abs_FanMax_kJhr

*** External files
ASSIGN "***.ou9" 60

*!? Output file for printed results |1000
*-----

* EQUATIONS "CoolPower"
*
EQUATIONS 1

```

```

CT_kJhr = 3600*[65,3]
*$UNIT_NAME CoolPower
*$LAYER Main
*$POSITION 259 381

```

```

*-----

```

```

* Model "PrintRegAbs" (Type 25)

```

```

*

```

```

UNIT 77 TYPE 25 PrintRegAbs

```

```

*$UNIT_NAME PrintRegAbs
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 1154 701
*$LAYER Outputs #
PARAMETERS 10
PrintInterval ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
63 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 6
78,1 ! Type24-2:Result of integration-1 ->Input to be printed-1
78,2 ! Type24-2:Result of integration-2 ->Input to be printed-2
78,3 ! Type24-2:Result of integration-3 ->Input to be printed-3
78,4 ! Type24-2:Result of integration-4 ->Input to be printed-4
78,5 ! Type24-2:Result of integration-5 ->Input to be printed-5
78,6 ! Type24-2:Result of integration-6 ->Input to be printed-6
*** INITIAL INPUT VALUES
RegIn RegOut Reg AbsIn AbsOut Abs
*** External files
ASSIGN "***.ou5" 63
*!? Output file for printed results |1000

```

```

*-----

```

```

* Model "Type24-2" (Type 24)

```

```

*

```

```

UNIT 78 TYPE 24 Type24-2

```

```

*$UNIT_NAME Type24-2
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 1150 602
*$LAYER Main #
PARAMETERS 2
PrintInterval ! 1 Integration period

```



```

0 ! 2 Relative or absolute start time
INPUTS 6
32,14 ! Regenerator:Inlet Desiccant Mass Flow Rate ->Input to be integrated-1
32,12 ! Regenerator:Outlet Desiccant Flow Rate ->Input to be integrated-2
32,5 ! Regenerator:Desiccant Desorption Rate ->Input to be integrated-3
54,2 ! AbsPump:Outlet flow rate ->Input to be integrated-4
62,9 ! Absorber:Outlet Desiccant Flow Rate ->Input to be integrated-5
62,7 ! Absorber:Absorption Rate ->Input to be integrated-6
*** INITIAL INPUT VALUES
0.0 0.0 0.0 0.0 0.0 0.0

```

```

*-----

```

```

* Model "Type24-3" (Type 24)

```

```

*

```

```

UNIT 79 TYPE 24 Type24-3
*$UNIT_NAME Type24-3
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 691 933
*$LAYER Main #
PARAMETERS 2
PrintInterval ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 12
48,2 ! Type649:Outlet flow rate ->Input to be integrated-1
47,2 ! LDTank:Load flow rate ->Input to be integrated-2
46,2 ! Type647:Outlet flow rate-1 ->Input to be integrated-3
46,4 ! Type647:Outlet flow rate-2 ->Input to be integrated-4
46,6 ! Type647:Outlet flow rate-3 ->Input to be integrated-5
57,2 ! AbsDrainPump:Outlet flow rate ->Input to be integrated-6
43,2 ! RegDrainPump:Outlet flow rate ->Input to be integrated-7
45,2 ! ReCircPump:Outlet flow rate ->Input to be integrated-8
62,7 ! Absorber:Absorption Rate ->Input to be integrated-9
32,5 ! Regenerator:Desiccant Desorption Rate ->Input to be integrated-10
62,9 ! Absorber:Outlet Desiccant Flow Rate ->Input to be integrated-11
32,12 ! Regenerator:Outlet Desiccant Flow Rate ->Input to be integrated-12
*** INITIAL INPUT VALUES
0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0

```

```

*-----

```

```

* Model "ValvTankMass" (Type 25)

```

```

*

```

```

UNIT 80 TYPE 25 ValvTankMass
*$UNIT_NAME ValvTankMass
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 801 936
*$LAYER Outputs #
PARAMETERS 10
PrintInterval ! 1 Printing interval
START ! 2 Start time

```

```

STOP ! 3 Stop time
64 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels

INPUTS 14
79,1 ! Type24-3:Result of integration-1 ->Input to be printed-1
79,2 ! Type24-3:Result of integration-2 ->Input to be printed-2
79,3 ! Type24-3:Result of integration-3 ->Input to be printed-3
79,4 ! Type24-3:Result of integration-4 ->Input to be printed-4
79,5 ! Type24-3:Result of integration-5 ->Input to be printed-5
79,6 ! Type24-3:Result of integration-6 ->Input to be printed-6
79,7 ! Type24-3:Result of integration-7 ->Input to be printed-7
79,8 ! Type24-3:Result of integration-8 ->Input to be printed-8
79,9 ! Type24-3:Result of integration-9 ->Input to be printed-9
79,10 ! Type24-3:Result of integration-10 ->Input to be printed-10
79,11 ! Type24-3:Result of integration-11 ->Input to be printed-11
79,12 ! Type24-3:Result of integration-12 ->Input to be printed-12
AbsDrainPump_Ctrl ! TankCtrl:AbsDrainPump_Ctrl ->Input to be printed-13
RegDrainPump_Ctrl ! TankCtrl:RegDrainPump_Ctrl ->Input to be printed-14

*** INITIAL INPUT VALUES
TankIn TankOut ValveOut1 ValveOut2 ValveOut3 ValveIn1 ValveIn2 ValveIn3
Abs Reg AbsOut RegOut AbsSig RegSig

*** External files
ASSIGN "***.ou3" 64
*! ? Output file for printed results |1000
*-----

* Model "AbsRegCncPrint" (Type 25)
*

UNIT 81 TYPE 25 AbsRegCncPrint
*$UNIT_NAME AbsRegCncPrint
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 204 936
*$LAYER Outputs #
PARAMETERS 10
STEP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
65 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels

INPUTS 4

```

```

62,7 ! Absorber:Absorption Rate ->Input to be printed-1
32,5 ! Regenerator:Desiccant Desorption Rate ->Input to be printed-2
CncTank ! CncCalc:CncTank ->Input to be printed-3
47,9 ! LDTank:Level indicator ->Input to be printed-4

```

```

*** INITIAL INPUT VALUES

```

```

AbsRate DesRate CncTank LevelInd

```

```

*** External files

```

```

ASSIGN "***.ou1" 65

```

```

*!? Output file for printed results |1000

```

```

*-----

```

```

* Model "PVTTemps" (Type 25)

```

```

*

```

```

UNIT 67 TYPE 25 PVTTemps

```

```

*$UNIT_NAME PVTTemps

```

```

*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf

```

```

*$POSITION 1132 136

```

```

*$LAYER Outputs #

```

```

PARAMETERS 10

```

```

-1 ! 1 Printing interval

```

```

START ! 2 Start time

```

```

STOP ! 3 Stop time

```

```

67 ! 4 Logical unit

```

```

0 ! 5 Units printing mode

```

```

0 ! 6 Relative or absolute start time

```

```

OWorA ! 7 Overwrite or Append

```

```

-1 ! 8 Print header

```

```

0 ! 9 Delimiter

```

```

1 ! 10 Print labels

```

```

INPUTS 6

```

```

13,1 ! T:Outlet Temperature ->Input to be printed-1

```

```

RegMinTemp ! RegCtrlParams:RegMinTemp ->Input to be printed-2

```

```

RegMaxTemp ! RegCtrlParams:RegMaxTemp ->Input to be printed-3

```

```

RegTsetpoint ! RegCtrlParams:RegTsetpoint ->Input to be printed-4

```

```

RegMaxShutoff ! RegCtrlParams:RegMaxShutoff ->Input to be printed-5

```

```

21,1 ! Type991:Control Signal ->Input to be printed-6

```

```

*** INITIAL INPUT VALUES

```

```

Toutlet Trmin Tmax Tsetp Tshutoff Control

```

```

*** External files

```

```

ASSIGN "***.ou4" 67

```

```

*!? Output file for printed results |1000

```

```

*-----

```

```

* Model "ParamsCheck" (Type 25)

```

```

*

```

```

UNIT 69 TYPE 25 ParamsCheck

```

```

*$UNIT_NAME ParamsCheck

```

```

*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf

```

```

*$POSITION 940 157

```

```

*$LAYER Outputs #
PARAMETERS 10
STOP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
68 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
OWorA ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels

INPUTS 24
A_regen_new ! RegenParams:A_regen_new ->Input to be printed-1
PV_Width ! PVParams:PV_Width ->Input to be printed-2
PV_Length ! PVParams:PV_Length ->Input to be printed-3
T_Width ! TParams:T_Width ->Input to be printed-4
T_Length ! TParams:T_Length ->Input to be printed-5
RegTsetpoint ! RegCtrlParams:RegTsetpoint ->Input to be printed-6
TankDia_in ! TankParams:TankDia_in ->Input to be printed-7
TankHeight_in ! TankParams:TankHeight_in ->Input to be printed-8
TankCap ! TankParams:TankCap ->Input to be printed-9
DesVolume_gal ! DesParams:DesVolume_gal ->Input to be printed-10
Tank_HLimit ! RegCtrlParams:Tank_HLimit ->Input to be printed-11
Tank_LLimit ! RegCtrlParams:Tank_LLimit ->Input to be printed-12
m_des_init ! TankParams:m_des_init ->Input to be printed-13
m_salt_low ! TankParams:m_salt_low ->Input to be printed-14
V_init_gal ! TankParams:V_init_gal ->Input to be printed-15
Cnc_Start ! DesParams:Cnc_Start ->Input to be printed-16
H_buffer_in ! TankParams:H_buffer_in ->Input to be printed-17
H_low ! TankParams:H_low ->Input to be printed-18
V_low_gal ! TankParams:V_low_gal ->Input to be printed-19
H_init_in ! TankParams:H_init_in ->Input to be printed-20
H_max_in ! TankParams:H_max_in ->Input to be printed-21
RMSE_Tot ! RMSE:RMSE_Tot ->Input to be printed-22
m_dot_a_design_CFM ! RegenParams:m_dot_a_design_CFM ->Input to be printed-23
m_dot_a_abs_design ! AbsParams:m_dot_a_abs_design ->Input to be printed-24

*** INITIAL INPUT VALUES
A_regen_new PV_W PV_L T_W T_L Tsp Tank_Dia Tank_H Tank_Cap Tank_VolInit
C_High C_Low m_des_Init m_salt_init Tank_V_init_gal Cnc_Start H_buffer
H_low V_low_gal H_init H_max RMSE mdot_reg mdot_abs

*** External files
ASSIGN "***.txt" 68
*! ? Output file for printed results |1000
*-----

* Model "Type24-4" (Type 24)
*

UNIT 70 TYPE 24 Type24-4
*$UNIT_NAME Type24-4

```

```

*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 307 943
*$LAYER Main #
PARAMETERS 2
STOP ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 2
At_HighLimit ! CncCalc:At_HighLimit ->Input to be integrated-1
At_LowLimit ! CncCalc:At_LowLimit ->Input to be integrated-2
*** INITIAL INPUT VALUES
0.0 0.0
*-----

* Model "AtCncLimits" (Type 25)
*

UNIT 68 TYPE 25 AtCncLimits
*$UNIT_NAME AtCncLimits
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 396 946
*$LAYER Outputs #
PARAMETERS 10
STOP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
69 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 2
70,1 ! Type24-4:Result of integration-1 ->Input to be printed-1
70,2 ! Type24-4:Result of integration-2 ->Input to be printed-2
*** INITIAL INPUT VALUES
High Low
*** External files
ASSIGN "***.o10" 69
*!? Output file for printed results |1000
*-----

* EQUATIONS "SPDiff"
*
EQUATIONS 2
RMSE_Diff = RegTsetpoint-[13,1]
RMSE_Sq = RMSE_Diff*RMSE_Diff
*$UNIT_NAME SPDiff
*$LAYER Main
*$POSITION 1007 338

```

```

*-----

* Model "Type24-5" (Type 24)
*

UNIT 84 TYPE 24 Type24-5
*$UNIT_NAME Type24-5
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 1065 346
*$LAYER Main #
PARAMETERS 2
STOP ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 1
RMSE_Sq ! SPDdiff:RMSE_Sq ->Input to be integrated
*** INITIAL INPUT VALUES
0.0
*-----

* EQUATIONS "RMSE"
*
EQUATIONS 1
RMSE_Tot = ([84,1]/max(TIME,0.001))^(1/2)
*$UNIT_NAME RMSE
*$LAYER Main
*$POSITION 1157 381
*-----

* EQUATIONS "WindConv"
*
EQUATIONS 1
Hw = (5.7 + 3.8*[24,8])
*$UNIT_NAME WindConv
*$LAYER Main
*$POSITION 227 146
*-----

* EQUATIONS "Equa"
*
EQUATIONS 4
kg_w_now_abs = [52,2]*[52,4]
kg_w_now_reg = [26,2]*[26,3]
PVT_area_tot = PV_Length*PV_Width+T_Length*T_Width
Rad_Rate = PVT_area_tot*[24,24]
*$UNIT_NAME Equa
*$LAYER Main

```

*\$POSITION 806 424

*-----

* Model "Type24-6" (Type 24)

*

UNIT 86 TYPE 24 Type24-6

*\$UNIT_NAME Type24-6

*\$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf

*\$POSITION 894 421

*\$LAYER Main #

PARAMETERS 2

PrintInterval ! 1 Integration period

0 ! 2 Relative or absolute start time

INPUTS 3

kg_w_now_abs ! Equa:kg_w_now_abs ->Input to be integrated-1

kg_w_now_reg ! Equa:kg_w_now_reg ->Input to be integrated-2

Rad_Rate ! Equa:Rad_Rate ->Input to be integrated-3

*** INITIAL INPUT VALUES

0.0 0.0 0.0

*-----

* Model "Type25c" (Type 25)

*

UNIT 87 TYPE 25 Type25c

*\$UNIT_NAME Type25c

*\$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf

*\$POSITION 972 424

*\$LAYER Outputs #

PARAMETERS 10

PrintInterval ! 1 Printing interval

START ! 2 Start time

STOP ! 3 Stop time

70 ! 4 Logical unit

0 ! 5 Units printing mode

0 ! 6 Relative or absolute start time

OWorA ! 7 Overwrite or Append

-1 ! 8 Print header

0 ! 9 Delimiter

1 ! 10 Print labels

INPUTS 3

86,1 ! Type24-6:Result of integration-1 ->Input to be printed-1

86,2 ! Type24-6:Result of integration-2 ->Input to be printed-2

86,3 ! Type24-6:Result of integration-3 ->Input to be printed-3

*** INITIAL INPUT VALUES

AbsWater_In RegWater_In Rad_In

*** External files

ASSIGN "****.arr" 70

```

*|? Output file for printed results |1000
*-----

* Model "Type25c-3" (Type 25)
*

UNIT 89 TYPE 25 Type25c-3
*$UNIT_NAME Type25c-3
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 108 754
*$LAYER Outputs #
PARAMETERS 10
-1 ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
72 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 5
62,2 ! Absorber:Leaving Air Humidity Ratio ->Input to be printed-1
HumRatAbove ! Equa-2:HumRatAbove ->Input to be printed-2
HumRatBelow ! Equa-2:HumRatBelow ->Input to be printed-3
91,1 ! Type24-7:Result of integration-1 ->Input to be printed-4
91,2 ! Type24-7:Result of integration-2 ->Input to be printed-5
*** INITIAL INPUT VALUES
Leaving Above Below Above_hr Below_hr
*** External files
ASSIGN "***.hmr" 72
*|? Output file for printed results |1000
*-----

* EQUATIONS "Equa-2"
*
EQUATIONS 2
HumRatAbove = ge([62,2],0.008)
HumRatBelow = lt([62,2],0.008)
*$UNIT_NAME Equa-2
*$LAYER Main
*$POSITION 102 829
*-----

* Model "Type24-7" (Type 24)
*

UNIT 91 TYPE 24 Type24-7

```



```

*$UNIT_NAME Type24-7
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 41 751
*$LAYER Main #
PARAMETERS 2
-1 ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 2
HumRatAbove ! Equa-2:HumRatAbove ->Input to be integrated-1
HumRatBelow ! Equa-2:HumRatBelow ->Input to be integrated-2
*** INITIAL INPUT VALUES
0.0 0.0
*-----

* Model "Type25c-4" (Type 25)
*

UNIT 92 TYPE 25 Type25c-4
*$UNIT_NAME Type25c-4
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 225 274
*$LAYER Outputs #
PARAMETERS 10
-1 ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
73 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 4
24,1 ! Type15-2:Dry bulb temperature ->Input to be printed-1
24,24 ! Type15-2:Total tilted surface radiation for surface ->Input to be printed-2
12,4 ! PV:Thermal efficiency ->Input to be printed-3
13,4 ! T:Thermal Efficiency ->Input to be printed-4
*** INITIAL INPUT VALUES
Dry Rad PVeff Teff
*** External files
ASSIGN "***.max" 73
*! ? Output file for printed results |1000
*-----

* EQUATIONS "Equa-3"
*
EQUATIONS 6
HumRat_Test = ge([24,6],Abs_HumRat_SP)
HumRat_Diff_Max = ([24,6] - Abs_HumRat_SP)*HumRat_Test*Abs_FanMax_kghr
MassWater = [62,5]*[24,6]

```

```

HumRat_Diff_L = ([24,6]*[62,5] - [62,2]*[62,5])
HumRatOut_Test = lt([62,2],Abs_HumRat_SP)
HumRat_Diff_Extra = (Abs_HumRat_SP - [62,2])*HumRatOut_Test*HumRat_Test*Abs_FanMax_kghr
*$UNIT_NAME Equa-3
*$LAYER Main
*$POSITION 91 904

```

```

*-----

```

```

* Model "Type24-8" (Type 24)

```

```

*
```

```

UNIT 99 TYPE 24 Type24-8
*$UNIT_NAME Type24-8
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 51 954
*$LAYER Main #
PARAMETERS 2
STOP ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 4
HumRat_Diff_Max ! Equa-3:HumRat_Diff_Max ->Input to be integrated-1
62,7 ! Absorber:Absorption Rate ->Input to be integrated-2
62,5 ! Absorber:Air Flow Rate ->Input to be integrated-3
HumRat_Diff_L ! Equa-3:HumRat_Diff_L ->Input to be integrated-4
*** INITIAL INPUT VALUES
0.0 0.0 0.0 0.0

```

```

*-----

```

```

* Model "Type25c-2" (Type 25)

```

```

*
```

```

UNIT 96 TYPE 25 Type25c-2
*$UNIT_NAME Type25c-2
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 998 957
*$LAYER Outputs #
PARAMETERS 10
-1 ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
76 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 6
62,7 ! Absorber:Absorption Rate ->Input to be printed-1

```

```

97,1 ! Type24-9:Result of integration-1 ->Input to be printed-2
97,2 ! Type24-9:Result of integration-2 ->Input to be printed-3
HumRat_Diff_L ! Equa-3:HumRat_Diff_L ->Input to be printed-4
97,3 ! Type24-9:Result of integration-3 ->Input to be printed-5
97,4 ! Type24-9:Result of integration-4 ->Input to be printed-6
*** INITIAL INPUT VALUES
AbsRate Absorbed1 Absorbed2 AbsRate2 Load ExtraLoad
*** External files
ASSIGN "***.abs" 76
*!? Output file for printed results |1000
*-----

* Model "Type24-9" (Type 24)
*

UNIT 97 TYPE 24 Type24-9
*$UNIT_NAME Type24-9
*$MODEL .\Utility\Integrators\Quantity Integrator\Type24.tmf
*$POSITION 894 943
*$LAYER Main #
PARAMETERS 2
STOP ! 1 Integration period
0 ! 2 Relative or absolute start time
INPUTS 4
62,7 ! Absorber:Absorption Rate ->Input to be integrated-1
HumRat_Diff_L ! Equa-3:HumRat_Diff_L ->Input to be integrated-2
HumRat_Diff_Max ! Equa-3:HumRat_Diff_Max ->Input to be integrated-3
HumRat_Diff_Extra ! Equa-3:HumRat_Diff_Extra ->Input to be integrated-4
*** INITIAL INPUT VALUES
0.0 0.0 0.0 0.0
*-----

* Model "Type25c-5" (Type 25)
*

UNIT 88 TYPE 25 Type25c-5
*$UNIT_NAME Type25c-5
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 1078 882
*$LAYER Outputs #
PARAMETERS 10
STEP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
77 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels

```

```

INPUTS 2
62,2 ! Absorber:Leaving Air Humidity Ratio ->Input to be printed-1
24,6 ! Type15-2:Humidity ratio ->Input to be printed-2
*** INITIAL INPUT VALUES
WLeaving WEntering
*** External files
ASSIGN "***.wte" 77
*! ? Output file for printed results |1000
*-----

* Model "Type25c-6" (Type 25)
*

UNIT 94 TYPE 25 Type25c-6
*$UNIT_NAME Type25c-6
*$MODEL .\Output\Printer\Unformatted\No Units\Type25c.tmf
*$POSITION 1153 829
*$LAYER Outputs #
PARAMETERS 10
STEP ! 1 Printing interval
START ! 2 Start time
STOP ! 3 Stop time
78 ! 4 Logical unit
0 ! 5 Units printing mode
0 ! 6 Relative or absolute start time
-1 ! 7 Overwrite or Append
-1 ! 8 Print header
0 ! 9 Delimiter
1 ! 10 Print labels
INPUTS 3
Abs_FanMax_CFM ! PumpFanParams:Abs_FanMax_CFM ->Input to be printed-1
Abs_FanMax_Lps ! PumpFanParams:Abs_FanMax_Lps ->Input to be printed-2
Abs_FanMax_kghr ! PumpFanParams:Abs_FanMax_kghr ->Input to be printed-3
*** INITIAL INPUT VALUES
CFM LPS KGHR
*** External files
ASSIGN "***.flw" 78
*! ? Output file for printed results |1000
*-----

END

```